STEAM TURBINES

THEIR DEVELOPMENT

STYLES OF BUILD, CONSTRUCTION

AND USES

W. GENTSCH

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BY

WILHELM GENTSCH

KAISERL. REGIERUNGSRAT UND MITGLIED DES PATENTAMTS

TRANSLATED FROM THE GERMAN BY

ARTHUR R. LIDDELL

WITH NUMEROUS ILLUSTRATIONS IN THE TEXT AND 19 PLATES

LONGMANS, GREEN, AND CO.
39 PATERNOSTER ROW, LONDON
NEW YORK AND BOMBAY
1906

PREFACE

The rapid progress made in the introduction of the Steam Turbine and its increasing commercial importance have created an interest in this invention far beyond the specialistic circles immediately affected by it, which aim at the perfection of its construction and the extension of its use. On the other hand, the knowledge existing as to the new domain created by it, which, until recently so scantily appreciated, has now grown to such dimensions, is comparatively slender. The author has accordingly approached the present task with the determination to treat it in a popular manner, and, as far as possible, to cover all the ground. The discussion of the various methods which in practice have been adopted in the construction of the Steam Turbine proved to be insufficient, and it became necessary to refer, as far as possible, to all the varieties of the different systems.

The development of the Steam Turbine may be divided into two periods of time, the first of which, extending over two thousand years, shows only a desultory searching for successes, while the second, covering the last decade, begins with results of practical value, which again have given the spur to systematic scientific research. By virtue of the latter, as also of the principles of thermodynamics and of the construction of water turbines, it has been possible to evolve theories for the steam turbine systems in use, thus sparing designers the trouble of experimenting, and to these theories we shall, on occasion, refer. As to the manner in which the scientific treatment of the subject-matter provides a basis for the calculations, a knowledge of the earlier attempts—however awkward these may sometimes appear—must exert an inspiring influence on the searcher; for even in the earliest times attempts were made to solve definite problems, to remove defects, and to achieve successes.

The author has further endeavoured to divide the systems of construction hitherto known into groups, for which two new designations have been made use of—those of the "Velocity Turbine" and the "Pressure Turbine." The grouping has indeed been the more difficult, that the experience of to-day has in many cases provided, for the modes of working of the motors, explanations differing from those which their designers had in view. It may here be observed that the researches on the domain of steam turbine construction are by no means to be looked upon as completed, and that it is not impossible that opinions now prevailing, as to the mode of working of the steam in this or that turbine, may so change that removals from the one group to the other become necessary.

In connection with the various systems, the peculiarities of construction, working, and application to purpose in view, have in each case been noticed. Attention continues to be paid more and more to the careful and suitable working out of all the parts of the different turbines in order to increase their efficiency and reliability, and at the same time reduce their cost of production. In the following work special attention is paid to the regulating, which in the working of dynamos is of special interest, and to the reversing, which plays an important part in the propulsion of vehicles and vessels on land and sea.

In view of the great commercial importance of the Steam Turbine, the marked tendency, which is showing itself on this particular domain, to protect rights of manufacture by letters patent is sufficiently accountable. It has not indeed been the author's object to go into the question of the patent rights of the different designs. Where, however, the patent specifications have been able to afford material, these have been duly referred to. The engineer will thus, without difficulty, be able to inform himself as to the legal aspects of the designs patented.

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INTRODUCTION

HISTORICAL NOTES

On the domain of steam-engine construction a profound revolution is just now in progress, the object of which is the turning to account, for the performance of useful work, of the energy possessed by currents of steam. Considering that the steam-engine constitutes our most important motor, and as such has found the most extended use, the change must be looked on as one of very considerable importance.

The far-reaching nature of the metamorphosis may be better appreciated when it is considered that the properties of the steam-engine have exerted a commanding influence on the development of other important branches, such as, in particular, those of dynamo construction and naval architecture. It thus becomes accountable that the commercial interest of large circles is excited by the whole transaction. A short span of time—that of the last five years—has sufficed to bring about a crisis in steam-engine construction; but the kind of engine which is now on the point of achieving such rare triumphs—the turbine—is the oldest steam-engine of all.

Indeed, Hero the Elder, who lived in Alexandria in the time of Ptolemy Philadelphus, in about the year 200 B.C., described in his "Spiritalia" an apparatus (Fig. 1) which was capable of giving life to doll-like figures and making them dance. These were placed upon a disc, the hollow vertical axis of which communicated at its open upper end with a furnace chamber, while its lower end was worked into two arms, which were bent in a direction tangential to the disc. When the upper end of the tube became heated by the fire in the furnace chamber, a circulation of the air was produced, and a turning motion was thereby imparted to the disc.

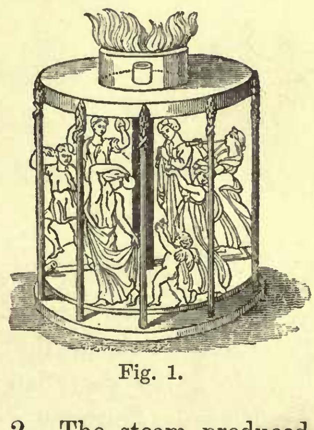

Fig. 1.

Hero further mentions a turbine of the reaction-wheel pattern, the appearance of which must, according to the description given, have been as shown in Fig. 2. The steam produced in the water boiler appears to have ascended through the hollow bearer into the hollow globe, which rested on the opposite side on a pointed spindle-bearing.

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In issuing from the arms the steam produced a back-pressure, which imparted a turning motion to the globe. More than this the Hero apparatus does not seem to have effected. Almost two thousand years later, in the year 1597, another invention was made public in Leipsic. A hollow globe (Fig. 3), with spindle-points borne by a pair of standards, revolved when water contained in it was made to produce steam, the latter then escaping through tubes projecting at the sides. For this apparatus it was claimed that it could be made to perform extraneous work, such as the turning of a spit attached to its spindle, in the absence of the housewife. Passing reference may here be made to the Äolipyl Cardans, pub-

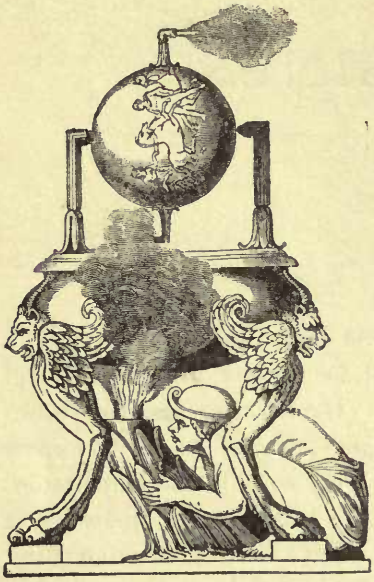


Fig. 2.

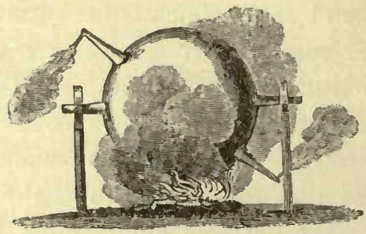


Fig. 3.

lished in 1550. Also, in the "Recréations mathématiques," Rouen (1627), reference is made to the self-turning globes. Of greater interest, however, is the report of the Italian mathematician Giovanni Branca, submitted to the Governor of Loretto, Canci (1628), which bore reference to a free-current turbine[1] (Fig. 4). In this case motion was to be transmitted by means of toothed wheels from a vertical shaft to a horizontal one. An alteration of the Branca design was made by Kircher, who directed two currents of steam on the turbine.[2] Grimaldi and Pereira are said to have made similar experiments in 1694, in presence of the Emperor Cang Hi. A light wooden car carried a receptacle filled with coals. Above this was placed a boiler, the exhaust from which was directed against the wheel of a windmill. The latter was set in motion by the steam issuing from the pipe, and in turn imparted its motion to the driving axle of the car, which could thus, it is said, be kept running for hours.[3] A few years later, in 1699, Amontons presented his steam wheel to the Académie des Sciences.

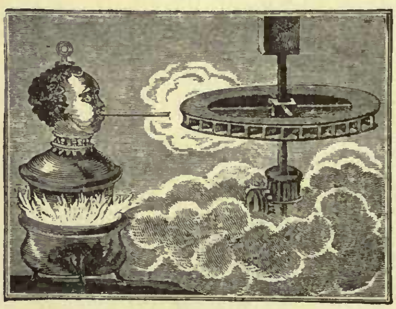


Fig. 4.

[1] Le Machine diverse del Signor Giovanni Branca, Rom, 1829.
[2] De Arte Magnetica, Colon, 1643, p. 413.
[3] Biographie Universelle, art. "Grimaldi."—Du Halde, Déscription de la Chine, 1736, vol. 111, p. 76.

An imitation of the undershot water-wheel was produced by Cook, who, in 1787 explained the engine before the Irish Royal Academy. The wheel, which, with its spindle, was free to revolve, was provided at its periphery with vanes, which were arranged to fold down as soon as they entered the casing. The steam, entering through a nozzle, acted against the vanes and escaped through a channel to the condenser. A wedge was so arranged as to oblige the vanes at the entrance to the case to lie close against the periphery of the wheel. The engine also worked the condenser. Condensation was also contemplated by Sadler in 1791, whose reaction wheel is illustrated in Figs. 5 and 6. The steam issued from the pipe *c* through the hollow spindle into the revolving arms *a* and *b*. On the other hand, cold water was injected through the end of the spindle *w* and the small pipe *s* into the nozzles *x*, and was intended to provide for the condensation of the steam here exhausting. The condensed product was then made to fall through the channel *ofh* into the reaction wheel *i*. The casing of the working wheel *n* was in turn enclosed in a protecting cover, *y*. A reversing motion was to be effected by the admittance of the steam into the casing. In such case—so thought Sadler—a reaction would take place, due to the pressure exerted by the steam when condensed at its entrance into the arms. Apparently the various steps were not very clear to the inventor himself. Kempel adopted a safer plan in 1794 in going back to the Hero engine, which, however, he altered somewhat (Fig. 7). In this case the boiler *a* ended above in a bottle-neck-like tube, which could be shut off by a cock, *d*, and on which the two-armed reaction wheel *b c* was free to turn. The arms were provided at their ends with disc-formed weights, and at a certain distance from the turning axis with a bolt, *x*, from which motion could, by means of chains or other such appliance, be transmitted to other machinery.

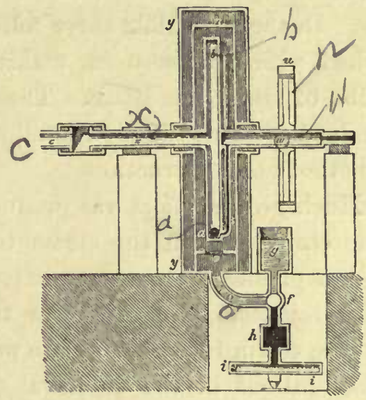


Fig. 5.

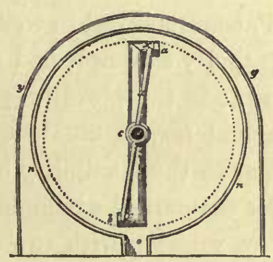


Fig. 6.

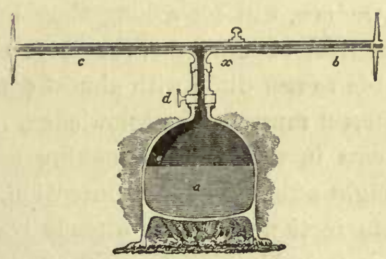


Fig. 7.

When Watt's inventions came out, the reciprocating steam-engine became the centre of interest, first with a revolving motion, and afterwards with a piston moving backwards and forwards in a straight path. The latter kind of engine then formed the central point round which the practical improvements and theoretical discussions grouped themselves, and the numerous proposals meanwhile made for steam turbines, which it was assumed could be lightly treated as unpractical constructions, were neither seriously taken up in practice nor thought worthy of attention in theory. It was not until quite recently that the Englishman, Parsons, obtained experience with an axial turbine of several stages (of pressure) of 10 H.P. and 18,000 revolutions per minute, built in 1884, such as in 1888 encouraged him to proceed to the construction of a larger one. The latter, which was several times repeated, was of 120 H.P., and was used to drive a continuous-current dynamo without condensation. In the year 1892 there followed a 200 H.P. turbine of the radial type, which worked with condensation, making 4800 revolutions per minute, and using only 16 lbs. of steam per I.H.P. These witnesses to the Parsons achievement, however, remained too much hidden to allow them to give the impulse to a general revolution in methods of construction.

Much greater effect was produced by the invention of the Swede, de Laval, who made clever use, in the steam turbine, of the already known laws relating to the passage of steam through tubes varying in diameter, and came before the public with a complete proposal. For the latter de Laval had furnished abundant proof that the steam turbine could be worked on a practical basis. The very large number of revolutions of the de Laval turbine, however, restricted it to a small field of application, in which it was not necessary to go beyond 350 H.P., so that it seemed as if the steam turbine would not attain to any far-reaching importance. At this point Parsons stepped in to dispel the doubts. To his endeavours and successes must the so rapidly accomplished revolution of the existing conditions be attributed, and the change in the prevailing views brought about by the practical results achieved. Only actual results obtained with finished engines had been able to produce conviction as to the possibility of their practical application. There existed no reliable bases for theoretical examinations of steam turbines, such as have brought the engine of to-day, working with the expansive energy of the steam, to its present scarcely betterable stage of perfection. Further, the complicated conditions which the application of the current-energy of the expansive and condensible motive-medium introduce, will for a long time to come oblige turbine constructors to depend upon practical experiments, the more so that the bases on which calculations might be made to rest differ with almost every one of the very numerous systems. Of further interest must be the knowledge, in which directions and with what means improvements in the engines making use of the current-energy of the steam have been sought after. It is the intention, then, in the following, along with the systems of construction that have already been adopted in practice, to discuss others which, as impracticable, or apparently impracticable, solutions, have been plunged into the sea of oblivion, though many of these, in perhaps somewhat altered form, have fair prospect of yet finding practical application.

CLASSIFICATION OF STEAM TURBINES

It is the custom at present, in accordance with the conditions of water turbine construction, to distinguish between action and reaction turbines. In the case of the first-named the back-pressure of the expanding steam which streams through the channel of the working wheel, is intended to supply the motive power; in that of the second, the mass-action of the steam is reckoned upon. What Zeuner says in regard to the water turbine may apply here also: "Reaction is the motive power in the case of the action turbines no less than in that of the others. If the usual distinguishing characteristics be observed, that in the reaction turbine the pressure and velocity of the motive medium during the passage of the latter through the working wheels undergo simultaneous alteration, while in the action turbine only the velocity changes, two kinds of steam turbine are obtained, the one in which the steam passes from the leading apparatus into the working wheel, the vanes of which must take the form peculiar to the reaction turbine, and the other in which the pressure of the steam is, by the help of the Laval nozzles, transformed entirely into velocity before it streams into the channels of the working wheels. Now, it is a well-known fact that, particularly in the case of the most promising full-pressure turbines in which the steam enters the wheel with undiminished pressure, the mass-action is preceded by reaction, and that it is in consequence customary to try to arrange the vanes for both kinds of action. In regard to the Laval nozzles, again, their mode of action, which will be described later, for the most part has the effect that the steam streams into the wheel with a certain pressure which is not applied to the performance of extraneous work. Finally, in the turbines arranged for action, and not in possession of Laval nozzles, there remains over a certain pressure, which must be used up by the wheel for extraneous work. Under these circumstances the over-pressure at the clearance space, as special characteristic of the reaction turbines, falls away.

The working of the steam turbine is in principle to be distinguished from that of the water turbine, in that it depends upon the help of an agent that can expand. This peculiarity can also be made the basis of classification of the steam turbines. The current-energy of the steam is intended, indeed, to be made use of in all turbines, but it is possible to pass the medium into the wheel under over-pressure so that this latter also must be used up in it. In such case it is imaginable, either that the steam passes at full pressure into the wheel, or that, even in the absence of Laval nozzles, the pressure is to a greater or less extent translated into velocity in the leading apparatus. In all cases there will be complete impingement upon the wheel channels. The steam may also, before doing its work, be allowed to expand to such an extent that its velocity alone is, as a form of energy, to be reckoned with, such over-pressure as still remains being left to the overcoming of inner resistances. The impingement upon the wheel channels may then be a partial one. Turbines of the first-named description might then be termed *pressure turbines*, and those of the last-named *velocity turbines.* The attempt may here be allowed to get this distinction introduced into practice. It may further be added that combinations also of the two kinds thus distinguished are imaginable, and, indeed, occur.

I

PRESSURE TURBINES

(*a*) **Pressure Turbines without Leading Apparatus.**—The simplest form of the steam turbine is probably that in which engine and boiler are in one: no connection between these two parts is then necessary. Its application, however, is not to be thought of for extensive or continuous work. Continuous working can be attained when the working-wheel is separated from the steam generator. This, indeed, entails the making tight of the moving part where it adjoins the fixed part that supplies the steam, and frictional resistance is thus set up. The frictional losses thus produced will, under conditions that are in other respects similar, evidently be smallest in amount when the path of the resistance is the shortest possible—in the present case when the steam supply is in the direction of the axis. This is the case in a considerable number of known systems of construction, which make use of cross-like arms like those of the Segner water wheel (Baker's mill).

Amongst the large number of these original *back-pressure* (*reaction*) *wheels*, *Lyman's* arrangement[1] (1876) (Fig. 8) may here be referred to, which consists in a straight tube working on a spindle at the middle. The ends of the latter are covered in and fitted only with ejection nozzles, *c*, for the steam, which is introduced at the middle.

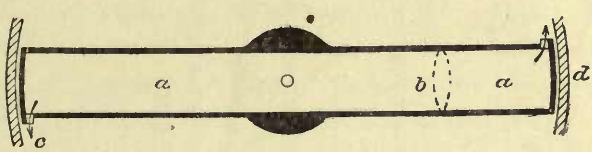


Fig. 8.

In order to reduce the outer air resistance to this rapidly revolving tube, Lyman provides it with a pointed oval section, as shown by dotted lines at *b*. The steam issuing from the nozzles *c* sweeps along the cold sides of the casing *d*, and is intended to be condensed thereby.

Parsons also (1884) made experiments with a reaction wheel, the arms of which were formed by steel tubes of pointed oval section. The wheel was enclosed in a cast-iron case which was in connection with the condenser. Parsons made use of steam of 100 lbs. per sq. in. pressure and a vacuum of 27 ins. He obtained an efficiency of 20 E.H.P. at a speed of 5000 revolutions per minute, and a steam consumption of 40 lbs. per E.H.P.

Emerson[2] (1886) reduced the air resistance to a frictional resistance by making use of a smooth revolving disc (Fig. 9). The channel *b*, which is connected with the hollow spindle is worked into the solid disc *a*. The latter is closed by a lid,

[1] A. P. 187,154. [2] A. P. 334,854.

which is worked tight on the disc at the side. The leverage with which the useful back pressure of the outflowing steam works, here forms only a fraction of the diameter of the disc.

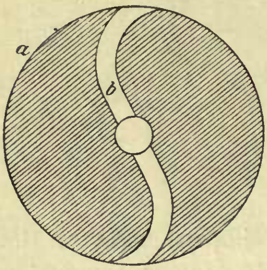


Fig. 9.

In all these arrangements the total quantity of steam which is contained in the hollow arms or channels, takes part in the spinning motion. Also the channels must have a distinctly defined direction depending upon the velocity of the steam and the angular velocity of the wheel, if the regular course of the current is not to be disturbed by portions of the steam in the channel becoming accelerated or retarded. Experiments have, however, been made with arrangements by which the steam is led almost to the back-pressure surfaces without its being, up to this point, accelerated by the action of the moving parts further than is due to friction. These alterations are in general accompanied by attempts to equalize the back-pressure forces acting on the periphery of the wheel.

To this category belongs *Ericsson's* back-pressure wheel[1] (Fig. 10).

In the case *a* is seated the hollow cone-shaped body *bc*, which by means of the nave *d* has solid connection with the wheel. The diaphragm *b* is provided with channels *f*, which afford passage from the case *a* to the inside of the body *bc*. The point of the latter is in connection with the outlet *g*, while the steam enters the case *a* at *h*. The steam contained in the latter makes its way through the channels *f* into the hollow cone *bc*, communicates a turning motion to the latter by means of back pressure on the channel walls, and then escapes through the outlet *g*. Within the hollow cone *bc*, meanwhile, fixed vanes, *k*, are arranged, which are worked upon the bearing tube *l*, and the purpose of which is to prevent the steam, as soon as it has left the channels *f*, from sweeping along the diaphragm *b* in a direction opposite to that of its motion and diverting it from the outlet *g*.

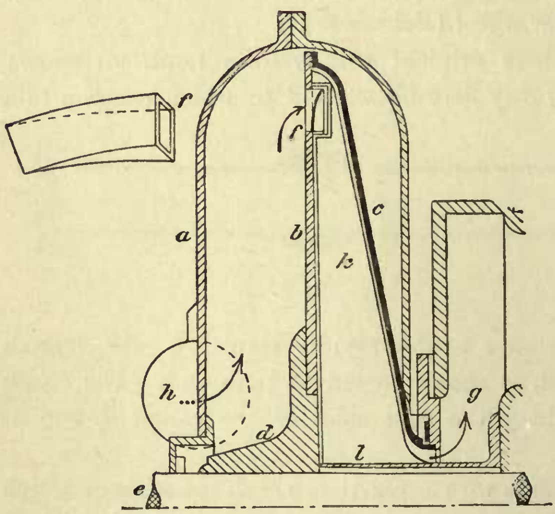


Fig. 10.

Here may also be mentioned *Gravier's* wheel[2] (1900) (Figs. 11 and 12).

Out of the hollow shaft *j*, which is packed into the nave *h* of the turbine, the steam makes its way through the slot-like openings *k* in the nave, into the free

[1] Engl. Pat., 5961 of the year 1830.
[2] D. R. P. 133,738.

inner space of the turbine, which serves as collecting chamber, and is contained by the plates *d* and *c*, which are provided with stiffening ribs, and are held together by the bolts *g*. Between these plates, at the outer periphery of the turbine, are inserted the grids *a*, which serve for the distribution and return of the steam, and a section of which is shown by Fig. 12. These grids, *a*, which are provided with rounds and hollows that fit loosely into one another and the interstices between them, form

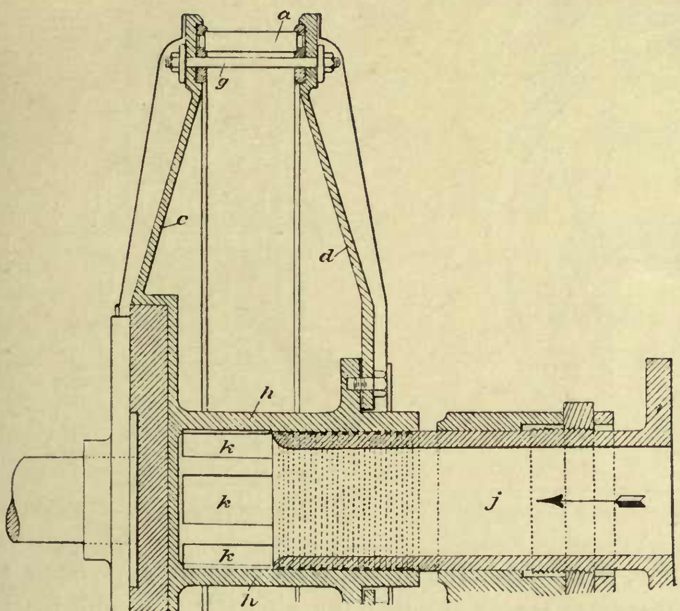


Fig. 11.

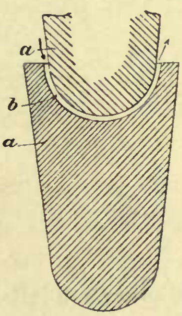


Fig. 12.

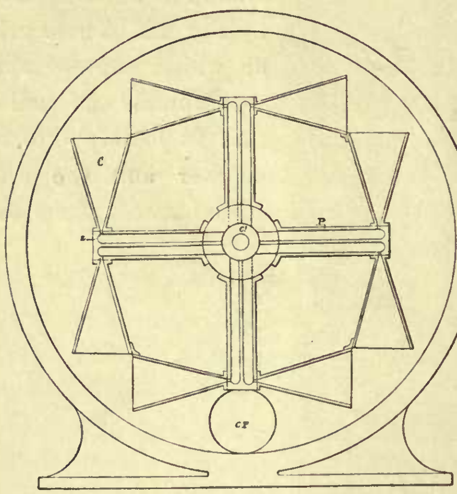
a row of narrow channels, the sections of which are exactly regulated by means of fillings, *b*, and they serve the purpose of compelling the return of the steam that flows through them, almost to 180°. The large number of these narrow channels brings about an almost molecular distribution of the steam. The U-like shape of the discharge channels is intended to bring about a considerably greater reduction in the absolute outflow speed of the steam and a speed of turning that is admissible in practice.

If the steam reach the outlet holes at full pressure and be allowed without further ado to exhaust, the magnitude of the motive back pressure will be in proportion to the velocity of the exhaust. After being exhausted from the engine, the steam expands, and as regards the power of the engine this expansion clearly represents a loss. In order to make the expansion available as power it must be made to take place within the motive parts of the engine. To effect this all sorts of means have been tried.

Reference may here be made to the proposals of *von Rathen*,[1] who caused the steam to expand in funnels, the sides of which were to take up the back pressure of the expanding motive medium. In Figs. 13 and 14 a turbine of this kind with a reversing arrangement is illustrated. According to the direction of turning motion, the steam is led, either through the pipe A^1, the chamber B^1, and the channel C^1,

[1] E. P. 11,800 of the year 1847.

or through A^2, B^2, and C^2, to the arms P, which are divided radially, so that the motive medium can stream either towards the funnel arranged for motion to the right or towards that for motion to the left. With the steam outlets *a* of these arms the expansion funnels *c* are here connected—the one group for right-hand,



Figs. 13.

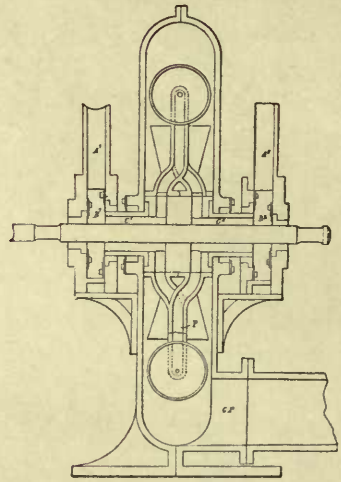


Fig. 14.

the other for left-hand motion, from which the steam, after expanding, proceeds to exhaust into the case, and thence to escape through the pipe CP. In addition to that of steam, the designer had in view the use of such media as compressed and atmospheric air for the production of the motive power. For the application of these the inside of the case would have to be in a constant state of vacuum. The form of the funnel must, of course, be made to suit the process of expansion. It may, for instance, be as given in Figs. 15 and 16.

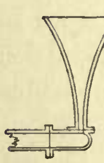


Fig. 15.

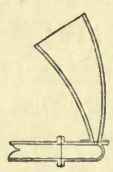


Fig. 16.

Again, the inlet opening to each funnel may be divided into several smaller openings. To promote a better distribution and utilization of the warmth, and especially in connection with the use of compressed air, in order to prevent a too rapid cooling, the funnels are provided with insertion pieces, which reach from the inlet to the outlet in each case. It is noteworthy that von Rathen further had in view the driving of locomotives by compressed and atmospheric air with the help of his turbines.

A further development of the *von Rathen* style of construction is shown by Fig. 17. The expansion chambers are bounded in this case by the inner ring C, the outer ring A, the steam chambers *a* with their slit-like openings, and the walls B, which are arranged spirally round the ring C. Finally, reference may be made to the reversible turbine (Fig. 18), in which the funnels traversed by open tubes A, P are arranged with their outlets together, and the steam has to escape through the openings O. The arms R, which again are each provided with two channels, receive the steam from the shaft. Only one of the steam pipes A[1] is shown. The steam passes from the arms R into the funnel through holes which are spaced at intervals round the tubes A, P.

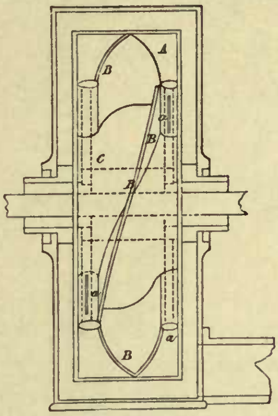


Fig. 17.

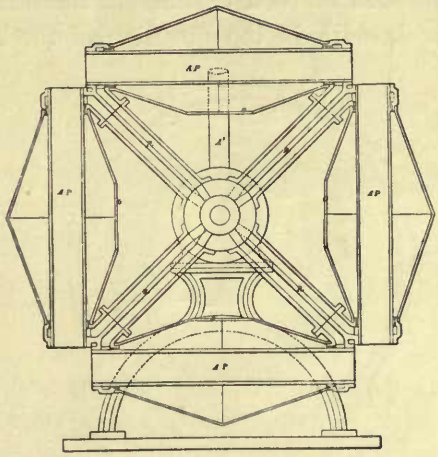


Fig. 18.

These expansion funnels, it may be observed, are not confined to a circular form of base, which may, for instance, be rectangular. An arrangement of this kind admits of a style of construction, useful in particular for ship purposes, in which the diameter is kept relatively small, and the engine is extended out in the direction of its axis.

Another arrangement, which is illustrated in Fig. 19, is that of *Cahen*[1] (1884). In the fixed case *a*, which is connected by means of a projecting tube *b* with a condenser, is set the revolving drum *c*, the walls of which are of non-conducting material. Against the inner wall of the drum is arranged the pipe *d*. The steam introduced through the hollow shaft *e* into the drum *c* enters the tube *d* at the point *f*, and leaves it again at *g*, to pass to the condenser. The space round the drum, then, is subject to about the same pressure as the condenser. The tube *d*, which is at first led cylindrically, becomes wider towards the outlet in the direction of the axis of rotation, and, as Cahen expresses it, in the sense of the theoretic curve of expansion. It may here be remarked in passing that the insertion of porous substances in the mouth of the pipe *d* is recommended, which, however, can scarcely assist the reactional effect of the outflowing steam. At the best it may be assumed that the influence of the widened outlet upon the velocity of outflow of the steam is thereby counterbalanced.

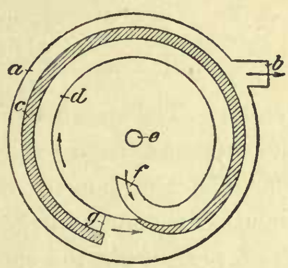


Fig. 19.

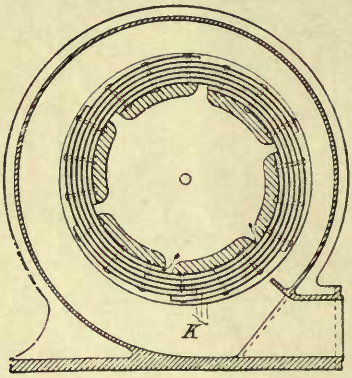
The design of *Husberg*[2] (1893), (Figs. 20, 21, and 22), is on much the same lines. The wheel is so formed that the circumferential containing-wall of the fresh-

[1] D. R. P. 32,560.

[2] A. P. 525,105.

steam chamber contains radially arranged openings, each connected with an expansion channel K, led spirally round the wheel in several turns, and discharging in a direction tangential to it.

The section of each channel K increases from the inlet to the outlet of the steam, its breadth in the direction of the axis of the wheel becoming gradually



Figs. 20.

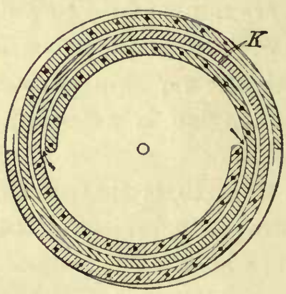


Fig. 21.

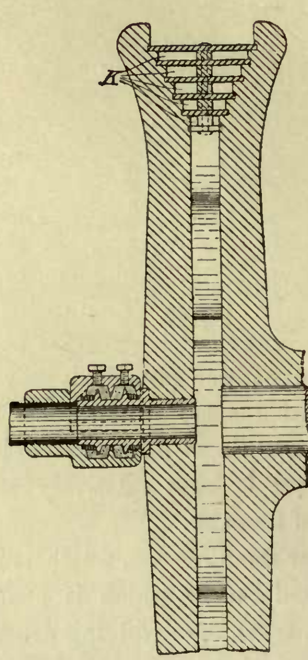


Fig. 22.

greater. The channels thus form chambers, during the passage through which the steam must expand before it escapes or passes into the condenser. The expansion channels, the length of which is, in proportion, considerable, are in the radial direction narrow, and they increase in the axial direction in order that the effective steam pressure in the channels may exert its force at as great a distance as possible from the shaft, and that the velocity of the steam may remain approximately the same in all parts of the channels. This last condition is fulfilled as soon as, for a given speed of the wheel, the increase of section of the channels keeps pace with the increase of volume of the steam streaming through.

Vissel[1] (1901) (Figs. 23 and 24) has adopted a reaction steam turbine, which consists in a spiral channel wound round a shaft in a plane, or in a truncated cone. In contradistinction to other arrangements, the steam outlet *f* of the spiral channel *c* is arranged as near to the shaft *a* as possible, *i.e.* at the inner end of the spiral, while the outer end of the latter serves as steam inlet *e*. Meanwhile the variations

[1] D. R. P. 131,660.

of sectional area of the spiral channel are so chosen that the velocity of the steam, during the passage of the latter through the channel, suffers considerable diminution. The object of this arrangement is the adjustment of the circumferential

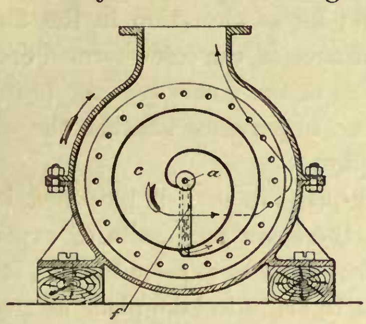


Fig. 23.

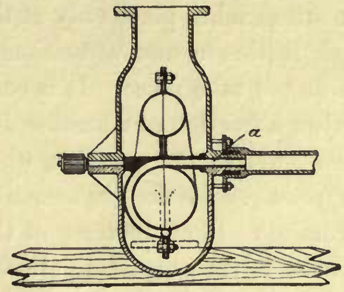


Fig. 24.

velocities at the various sections of the spiral channel in such a manner that at any point at which the steam has a high pressure and a correspondingly high velocity the spiral also rapidly revolves, while, on the other hand, at points at which, by reason of large increase of section, the steam has low velocities, the rate of rotation of the sections in question is slow. The great reduction of velocity of the steam within the spiral tube is intended to have the effect, that the work due and equivalent to it is made to help to turn the shaft of the engine, thus producing a higher efficiency than is possessed by those engines in which the steam does not alter its velocity, nor its corresponding kinetic energy, until it has emerged into the air or passed into the condenser.

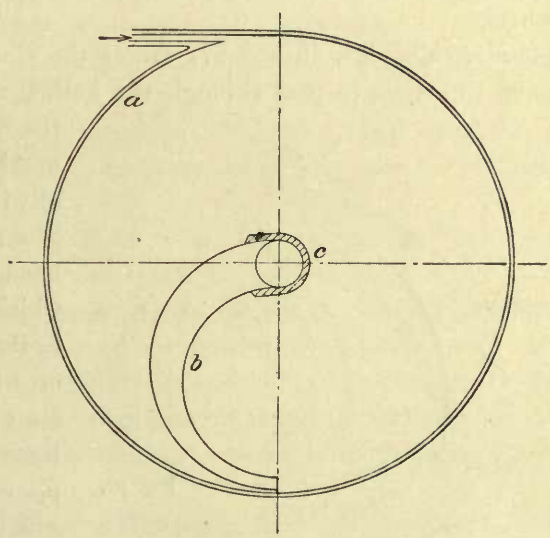


Fig. 25.

Another inversion of ordinary method is given by *Felderhoff*[1] (1877) (Fig. 25). The steam introduced tangentially into the case *a* at its periphery assumes a rotary motion. It streams against the mouth of the tube *b*, which leads into the shaft *c*. The latter is hollow at the one end, and is provided with a belt-wheel or other similar appliance. The tube *b* is set in motion by the steam; its section, however, increases as it nears the shaft. The steam which streams towards the exhaust loses in velocity in proportion as the section of the tube increases. This loss of velocity is intended to answer to the kinetic energy imparted to the tube.

[1] D. R. P. 249.

Mention may, in this connection, be made of the screw turbines in which the steam has to stream through screw-like passages, led either round cylinders (*Hewitt*,[1] *Canning*[2]), or round cones (*Wahle*,[3] *Astor*[4]). For steam turbines, however, these systems of construction are completely valueless; for an alteration in the direction of the steam takes place only at the point of entrance of the screw-formed channels. In their further course alterations in the direction of the steam tending to turn the wheel do not take place. It is consequently of no importance whether the channel comprises a fraction of a screw-spire or several spires.

In answer to the question, whether the back-pressure work is increased in such spiral passages by the expansion of the steam, there are no results of experiment forthcoming. If an increase of the working effect is to be expected at all, it would seem more to the purpose to allow the expansion to begin to take place at the point at which the back pressure itself comes into action, this being effected by the addition, at the outflow openings, of chambers increasing correspondingly in sectional area. To this end also many proposals have been made, which in part have attained the value of practical experiments.

Thus *Meade*[5] bends the outer ends of the back-pressure arms round to an angle of 90°, and attaches nozzles bent in a direction concentric with the axis of rotation, which also expand in the direction parallel with the axis in such a manner that the expansion of the steam can proceed with due regard to the speed of rotation of the wheel.

Morton[6] (Figs. 26 and 27) causes the steam, which streams continuously in an axial direction, to pass through the hollow narrowing arms *a* to the periphery of the wheel. On the periphery are arranged nozzle-shaped hoods *b*, which widen gradually in the direction of the current of steam. The steam passes from the arms *a* into the hoods *b*, in which it can expand, and from which it can issue in a direction tangential to the wheel. In order to reduce the loss of warmth imparted to the case by the arms *a*, which act as leads for the steam, these are covered in by disc-shaped side-plates *c*. Fig. 28 shows an arrangement, which is in so far an inversion of the foregoing, as the steam admitted into the casing enters, through the nozzles *d* on the periphery of the wheel, into the hollow expansion chamber arms *e*, and is led off in an axial direction.

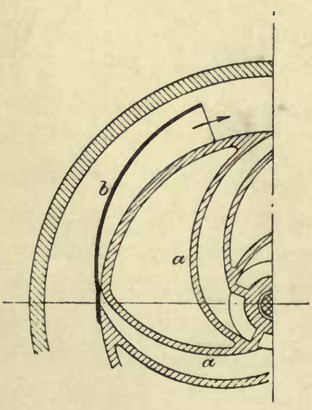


Fig. 26.

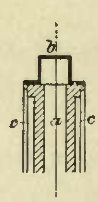


Fig. 27.

Weichelt's wheel[7] (1897) is shown in Fig. 29. At the outlet opening proper *a*,

[1] E. P. 11,709 of the year 1895. [2] E. P. 11,159 of the year 1898. [3] A. P. 634,969.
[4] E. P. 19,568 of the year 1901. [5] E. P. 10,765 of the year 1845.
[6] E. P. 17,299 of the year 1888. [7] E. P. 2817 of the year 1897.

a channel is attached, which grows gradually larger, and in which the steam can expand so much as to give forth all its energy before it leaves the wheel. The steam, which may be replaced by compressed air, gas, or other substitute, flows towards the opening *a* through the channel *c*, which is relatively wide at the inlet end, and narrows towards the opening *a*. The intention here is, that the expansion of the steam be completely effected in the channel *b*. The widths of the channel *b* (which is of circular section) at its commencement *a*, and at its end *d*, are so chosen as to correspond with the varying pressure of the steam.

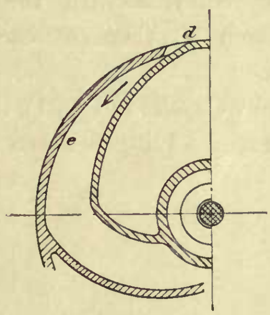


Fig. 28.

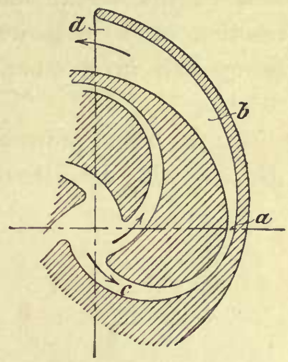


Fig. 29.

Whatever form the above-described endeavours may take, it will be feasible to carry the expansion of the steam only so far that it leaves the revolving parts at a pressure which is at least equal to that of the outer air or of the condenser. Should it sink below this, the result would be an opposing action of the pressure without, and a consequent reduction of the motive power, or even the stoppage of the engine. The degree up to which the expansion of the steam in the reaction wheel may be carried depends to a considerable extent upon the speed of the wheel. The finding of the right proportions in this respect, and of the means of regulating the amount of work performed by the engine without its efficiency being impaired, may be mentioned as further problems which will have to be solved in practice.

Meanwhile, there is more to be said for these attempts to utilize the expansion of the steam than for the ones to be mentioned below, the aim of which is to cause the steam issuing at full pressure from the back-pressure nozzles to strike against resisting fixtures. The expectation, which would appear to be an unreasonable one, seems to be that points of support will thus be obtained, which will compel the steam expanding, after its outflow, in some degree to expand backwards in the direction of the rotating nozzle, and thus to offer a resistance to the following steam as it streams out.

An arrangement of this kind (*Perkins's*)[1] is illustrated by Fig. 30. The reaction nozzle *a* of the working wheel *b*, which is of the same breadth as the latter, glides with its outer wall over the teeth *c* of the large case *d*. The steam issuing from the nozzle strikes against the teeth *c*.

In an arrangement made by *Ivory*,[2] only a fixed toothless ring is made use of, against which the steam is apparently expected to meet with frictional resistance. *Thévenet*,[3] again (1886), immerses the working wheel in a liquid, which on its part keeps the outflowing steam under a corresponding back pressure.

[1] E. P. 7242 of the year 1836. [2] E. P. 2076 of the year 1857. [3] D. R. P. 37,428.

In the examples hitherto adduced, a continuous flow of the steam through the channels of the wheel has been assumed. If the steam be allowed to enter these latter only at certain intervals, the expansion of this motive medium while the further supply of it is shut off will naturally follow.

This manner of working is, for instance, to be seen in the wheel proposed by *Slate.*[1] The fixed horizontal hollow shaft, which is arranged to serve as the supply lead of the steam, has radially laid openings, which during the rotation of the reaction arms periodically coincide with the steam-inlet openings. The arms being two in number, each of them at each revolution receives fresh steam twice.

In the wheel proposed by *Capell*[2] (Fig. 31), the central steam inlet a rotates with the outer ring c, while the ring b lying between these remains fixed, or turns in the opposite direction. The ring b can also turn alone, in which case the parts a and c remain fixed. It will be seen that the communication channels laid through the ring b widen out from the point of inlet a to the points where they meet with the channels of the ring c.

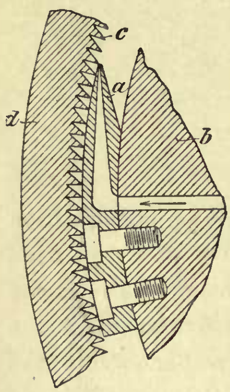


Fig. 30.

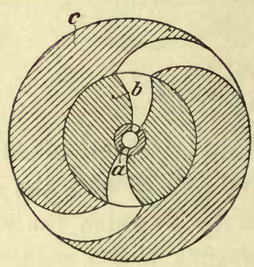


Fig. 31.

Since, under conditions in other respects the same, the speed of rotation of a turbine is the greatest when the utilization of the energy of the steam is most complete, it follows that every small abstraction of energy brings with it a corresponding small reduction of the velocity. Further, such part of its energy as a working wheel allows to pass on unused may be made use of by another wheel succeeding the first one. The coupling of several working wheels one behind the other, each of which takes up a fraction of the energy of the steam streaming through the engine, produces the multi-stage back-pressure wheels or pressure turbines without leading appliances. The introduction of multi-stage turbines, in which the energy of the steam is distributed over several wheels, accompanies the most recent practical successes of the steam turbine. The increase or reduction of the number of the stages affords the means of diminishing or increasing the speed of rotation of each stage, and consequently of the whole engine, without destroying the possibility of using the steam to the full in all the stages collectively.

Such multi-stage turbines are to be found amongst the earliest reaction turbines. That of *Gilman,*[3] for instance, may be pointed to, in which several wheels were set side by side on a shaft, and allowed to run in separate chambers of a common case, The supply of steam was given axially by means of tubes fitted concentric with the

[1] E. P. 1083 of the year 1852. [2] E. P. 911 of the year 1883.
[3] E. P. 7417 of the year 1837.

shaft with a certain amount of play, and each of which connected the wheel belonging to it with the chamber of the wheel previously acted upon by the steam.

The steam turbine of *Scherrer*[1] (Figs. 32–34) also has several reaction wheels set one behind another, and separated from each other by partition walls. The connection between the partitioned-off chambers *h*, *i* is effected by means of the channels *l*, which are so arranged that the motive medium passes from one steam chamber to another in a direction tangential to the casing wall. Within the turbine chamber the steam is intended always to preserve the same direction of motion, so that losses due to change in the direction of the current are not to be reckoned with.

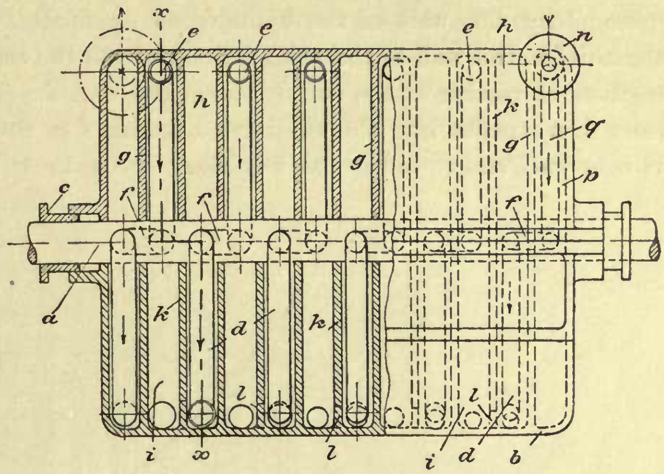


Fig. 32.

The motor shaft *a* rests tightly packed in stuffing boxes or otherwise in the cylindrical case *b*, which is constructed in two parts. On this shaft *a* are arranged the separate wheels composing the engine. Each wheel is made up of two semicircular bent hollow arms *d* and *e*, as shown in Fig. 33, lying in different planes side by side, and

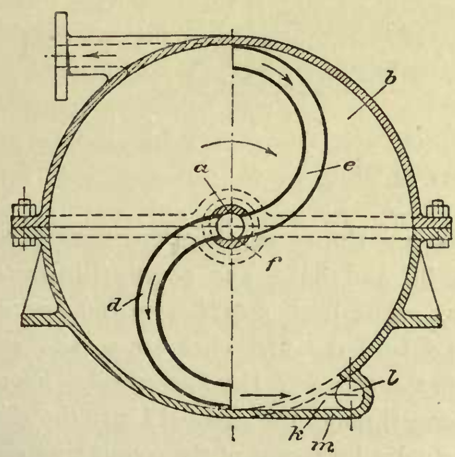


Fig. 33.

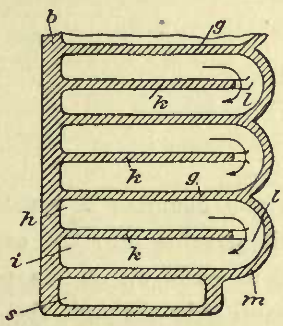


Fig. 34.

together forming an S. Between them a partition-wall *k* is provided in the cylindrical case *b*. The connection between the two tubes of each wheel, or between the steam chambers, is formed by the channels *l*. The tube *d* moves in the chamber *h*,

[1] D. P. R. 142,662.

the tube *e* in the chamber *i*. The connection between the chamber *i* and the adjoining chamber *h* of the next following section is effected by means of a short channel *f* arranged within the shaft *a*. The channels *l* are in a chamber *m* cast on the cylindrical case *b*, so that the tubes *d* and *e* can move past it with clearance, and so that the steam, streaming from the arm *d* into the chamber *h*, within which it moves along the wall of the cylinder so as finally to flow out in a direction tangential to this wall at the point *l*, shall, within the case, always preserve the same direction of motion with which it leaves the arm *d*. Through the channel *l* the steam then streams into the adjoining chamber *i* in a direction tangential to the cylinder wall, and continues its way along the same till it enters the arm *e*, thus,

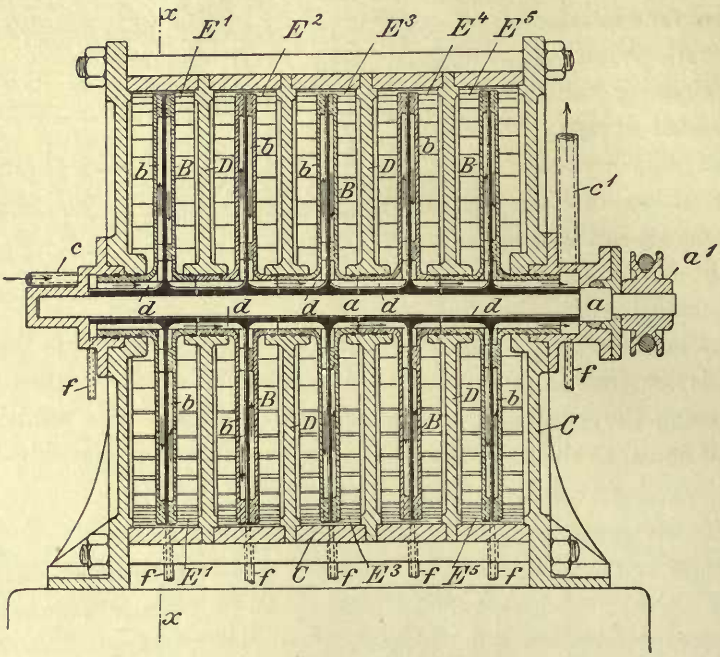


Fig. 35.

also in the chamber *i*, constantly preserving the same direction of motion with which it streams into the arm *e* (Figs. 33 and 34). The engine illustrated by Fig. 32 is made up of five pairs of arms. The fresh steam is introduced at the right-hand side through the short tube *n* into the first chamber *p*, and streams through the tube *q* and the channel *f* into the arm *d* of the first wheel. From this the steam streams into the chamber *h*, passes through the channel *l* into the chamber *i*; then proceeds through the arm *e*, and finally by means of the connecting channel *f*, arranged within the shaft *a*, reaches the arm *d* of the next following wheel. After making its way through the arms *d* and *e*, which together form an S-shaped tube, the steam performs the work of reaction as it passes out of it. Since the velocity of the steam becomes gradually less and its volume greater, the sectional areas of the tubes *d* and *e* must proportionately increase.

A design which, as far as the action of the steam is concerned, remains unclear, is that of *Benze* and *Bachmayr*[1] (1896), in which action and reaction of the steam both take place, but in which probably only an imperfect back-pressure remains available as motive power. Fig. 35 is a vertical section, and Fig. 36 a cross section at *xx* (Fig. 35). On a shaft *a* common to both are fixed the turbine discs B, on which at each side of the plane through their middle are rows of working cells *b* arranged in rings and bounded by curved partition walls. The cells are connected with each other by rows of channels b^1 (Fig. 36). The steam from the boiler passes through a pipe *c* into the first disc B, and streams from the last one into the exhaust c^1. The turbine is enclosed in a case *c*, which, by means of a partition-wall, D, fitting close against the nave portions of the turbine disc B, is divided into a number of chambers, E^1, E^2, E^3, E^4, etc., each having a disc B, which, when the engine is at work, is filled with steam. Each two successive discs B have communication with each other by means of the steam channels *d* arranged in the nave portions. Fig. 36 shows the section of the working cells at the left-hand side of a turbine disc of this kind. The fresh steam proceeds through the pipe *c* into the central channel *d*, streams in the direction indicated by the full-lined arrows in Fig. 36, into the cells *b* at the left-hand side of the disc B, and passes through the channels *e* into the chamber E^1. From here the current streams through the channels e^1 in the direction of the dotted arrows through the cells *b* at the right-hand side of the same turbine disc, B, and passes into the hollow of the nave portion of the latter, whence it is passed on to the second disc B. After the steam has thus passed through all the discs B, it finally reaches the pipe c^1, which leads either into the outer air or into the condenser. In the proportion in which the expansion of the steam progresses, the cross sections of the current channels also increase in area, so that the velocity of the current due to the number of the cells remains the same in all the parts of the turbine. At the same time it is intended that the kinetic energy of the steam, at its outflow into the chambers and at its entrance into the discs, be converted by the latter into motion, and thus made available for work.

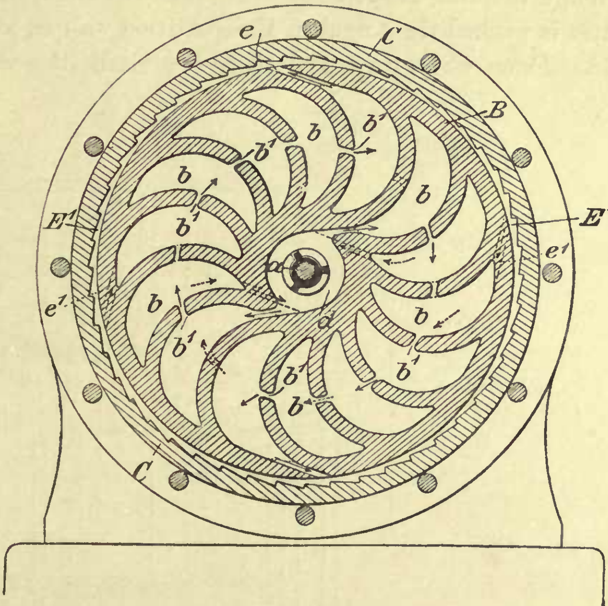


Fig. 36.

In the *Parsons* method of construction[2] (Fig. 37), also, several two-armed

[1] D. R. P. 90,777.

[2] E. P. 8854 of the year 1893.

reaction wheels *d*, *e*, *f*, arranged in separate chambers *a*, *b*, *c*, work in stages. The steam outlets *g* of the several arms are, in consequence, increased in size from wheel to wheel. The steam proceeds from the chamber *h*, which, by means of labyrinth stuffing boxes at the shaft bearings, is well packed off from the first chamber, through passages bored radially and axially in the shaft *k*, into the arms of the first wheel *d*. From this it passes into the first chamber, and thence into the wheel *e* through the ring-shaped space between the shaft and the concentric tube *l*, which latter is worked tight against the partition wall *m*, and turns with the wheel *e* aforesaid. From wheel to wheel there is a diminution in the steam pressure, which is

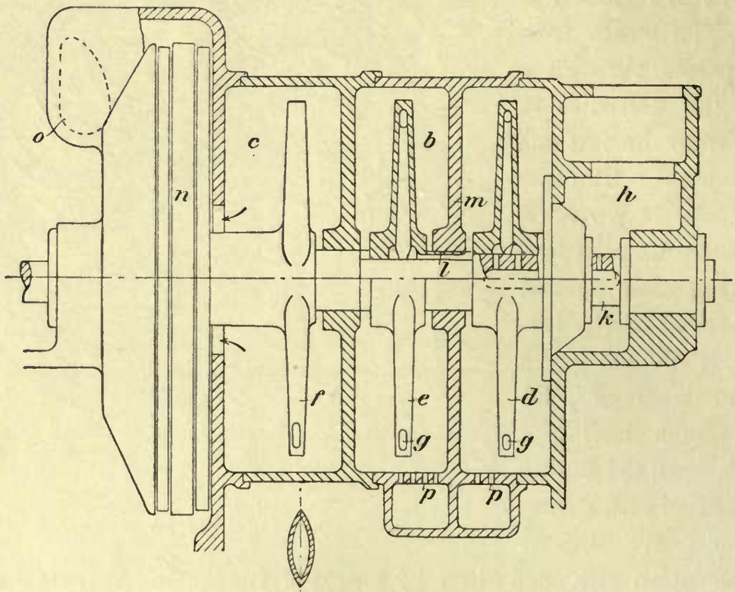


Fig. 37.

utilized only by means of the reaction wheels. Meanwhile Parsons has also thought of combining a set of reaction wheels with a radial turbine *n*. In this case the steam was to expand, through the wheels *d*, *e*, *f*, up to a certain point, and then to impinge on the turbine *n* from the inside. It was then to escape through the outlet *o*, either into an exhaust tube or into the condenser. The condensed water was to run off from the chambers through the holes *p*. This multi-stage arrangement of the Parsons reaction wheel, however, gave a less favourable result than that with the simple wheel, since the resistance offered to the revolution of the arms in the more compressed motive medium clearly more than counterbalanced the gain attained by the staging.

Morton's arrangement [1] differs in so far from the above that he arranges the stages conaxial with the shaft (Figs. 38–41). Between two discs B he inserts two rings A, which are conaxial with one another, and the spaces between which increase from the inside radially outwards. The one disc B is connected by means of narrow arms

[1] E. P. 9158 of the year 1888.

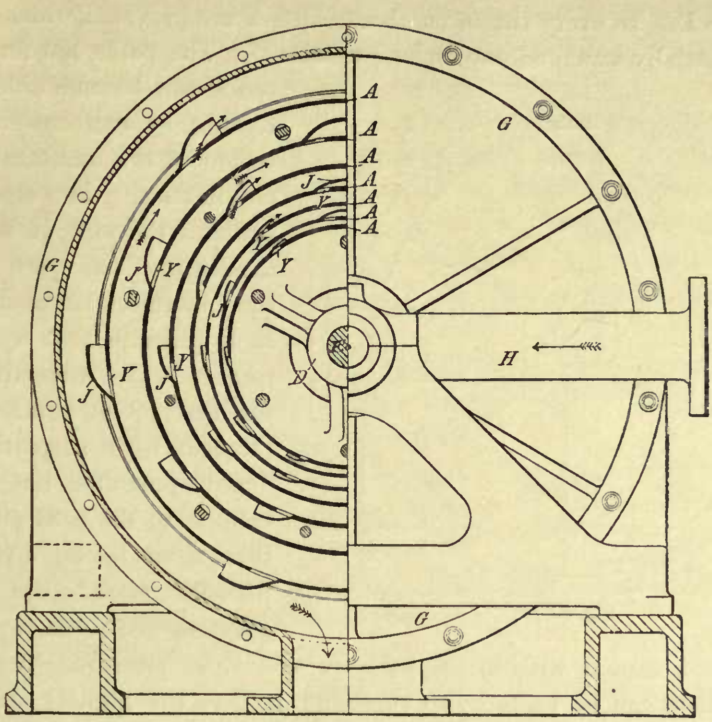


Fig. 38.

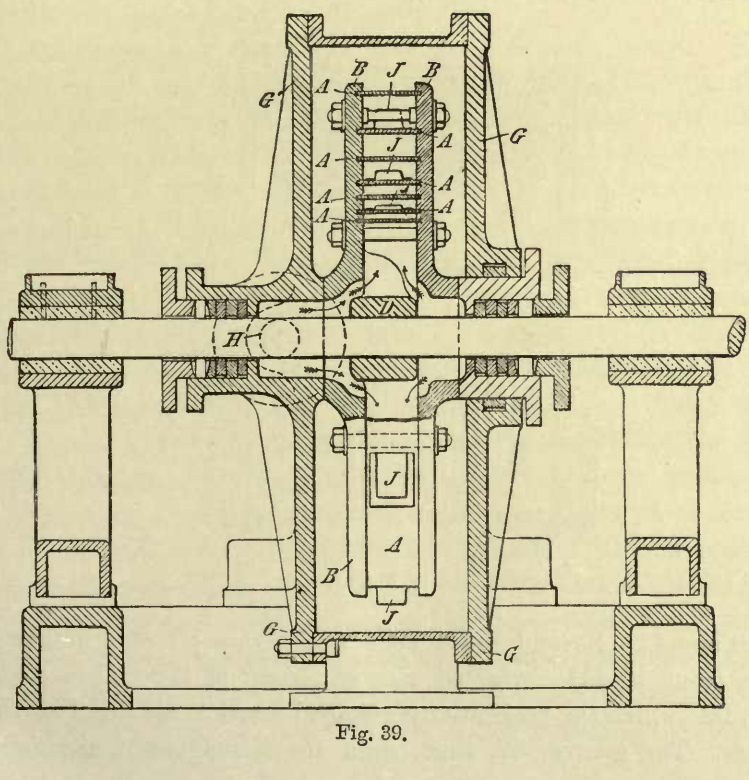


Fig. 39.

with the nave D. In every ring a number of slits Y are provided, which are covered by the tangentially arranged cap-formed nozzles J. The paths left for the flow of the steam become wider from ring to ring by increase either of the size or of the numbers of the slits and nozzles. The steam entering through the pipe H is introduced axially into the space enclosed in the innermost ring. It exerts its back pressure first upon the first ring A while streaming through the first set of nozzles, and then proceeds with proportionately reduced pressure, but in greater volume, to the next set of nozzles, there again having a portion of its pressure transformed into back pressure, and so on, until it passes into the closed case G with atmospheric or condenser pressure. A variation of this arrangement can be made by so reversing it that the highest stage is made the outer one, and the lowest the inner one.

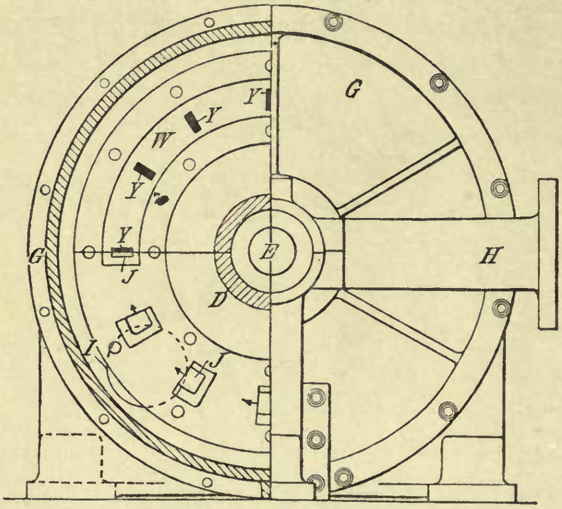


Fig. 40.

A further embodiment of the Morton idea in the form of axial turbines is shown

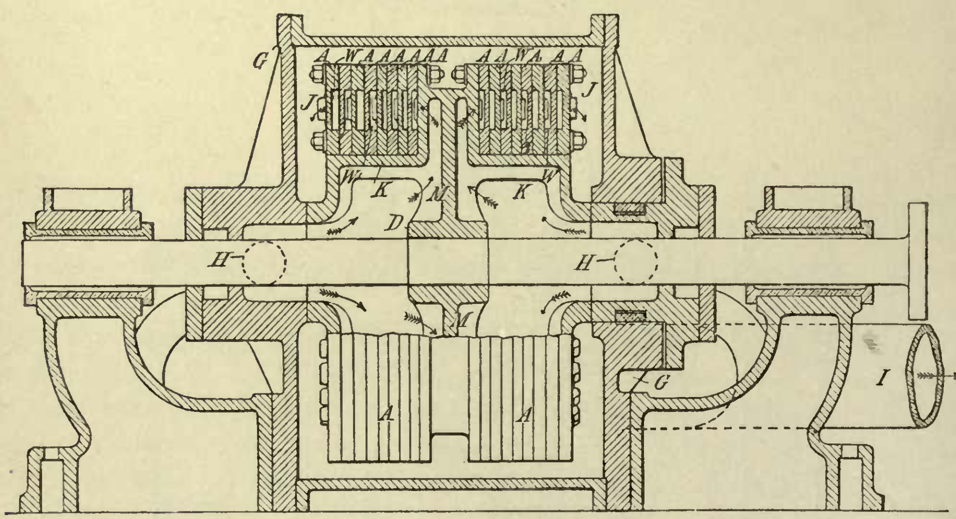


Fig. 41.

in Figs. 40 and 41. Several ring-shaped discs A are placed together in a row on the hollow trunk K, and connected with it. Each of these discs contains a ring-groove W, into which the tangentially arranged reaction nozzles J of the adjoining disc project. The grooves W, then, form the impingement chambers for those groups of nozzles which are opposite the (partially) grooved discs, and at the same time serve as the outflow chambers for the preceding nozzles. The hollow trunk K is divided into two equal parts by the wall M, which forms the connection with the nave D. It also bears two sets of discs A, which contain nozzles pointing to the right in the case of the one set, and to the left in the case of the other. In other words, the one side is arranged for right-hand and the other for left-hand motion. In accordance with this, two steam inlets H are provided. The steam sweeps through the nozzles from disc to disc, and loses in pressure from step to step. The case G, into which both the right-hand and the left-hand turbines deliver their outflowing steam, has only one outlet J.

In the engine designed by *Robinson*[1] (Fig. 42), also, the axial arrangement of the stages is met with, ring-like chambers *a*, *b*, *c* being arranged conaxially round the shaft. The steam introduced through the hollow shaft into the space *a* reaches a small chamber *d*, and proceeds from this through the nozzles *e* into the next chamber *b*, at the same time exerting against the wall *f* a motive back pressure. The force of this is proportionate to the loss of pressure experienced by the steam in passing into the chamber *b*. From the chamber *b* the steam streams through the nozzle *g* into the chamber *c*, and from this through the nozzle *h* into the ring-like chamber *k*, the outward boundary of which is the ring wall *l*, that turns with the rest. The concentric rings *l*, *m*, *n*, *o* are placed between radially arranged discs *p*, which at the same time form the side boundaries of the ring chambers. From the chamber *k* the steam is led radially back to the close vicinity of the shaft, whence it streams through the next set of wheels, which are similar to the ones first traversed. These groups of wheels may, however, be so put together that the steam first proceeds through all the chambers of like diameter lying side by side in an axial direction, before it passes into the outer chambers. The condensed water is led off through the outlets *q*, which are provided in all the ring chambers.

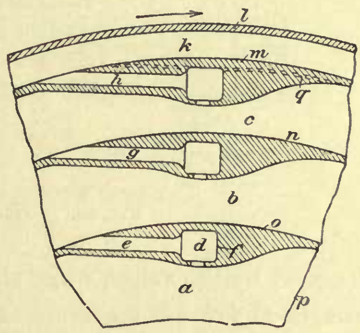


Fig. 42.

Of interest is the method of construction adopted by *Girard*.[2] In this, vaneless ring chambers, in which the steam is accelerated, alternate, within the working wheel, with wreaths of vanes in which this velocity is destroyed. The radial wheel, the shaft of which is vertical (Figs. 43 and 44), and the inside of which is everywhere impinged upon, consists of ring chambers *a*, *b*, *c*, *d*, *e*, *f*, which are fitted with vanes, and of the vaneless ring chambers *g*, *h*, *i*, *k*, *l*, which lie between these. The vanes are curved in the manner shown in Fig. 44. The channels belonging to each ring preserve the same section throughout, and in consequence of this they are made to increase in width from the point of inlet of the steam to that of the outlet, in accordance with the narrowing of the steam passages which is produced by the radial

[1] E. P. 15703 of the year 1893.

[2] E. P. 30 of the year 1855.

curving of the vanes. The steam streaming from the fixed steam chamber *m* to the working wheel is intended to receive a certain acceleration in the narrow ring channel *n*, and to lose velocity in passing the first group of vanes *a*, and thereupon, at a pressure somewhat less than that in the chamber *m*, to enter the ring chamber *g*, where it takes no part in the turning of the wheel. This chamber *g*, however, narrows somewhat towards the next following group of vanes, and the narrowing has the effect of again giving the steam a certain acceleration before it passes with reduced pressure among the vanes *b*, to suffer a further loss of velocity in streaming through these. The game thus continues to repeat itself with alternating acceleration of the steam in the vaneless ring chambers and loss of velocity in the channels

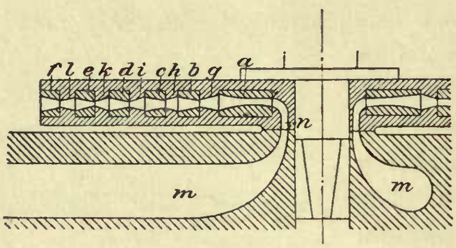


Fig. 43.

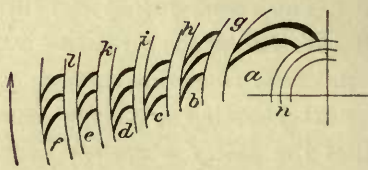


Fig. 44.

bounded by the vanes, while the difference in pressure between each two successive wreaths of vanes is made use of in the ring-chamber between them for the acceleration. From the last group of vanes the steam flows, deprived of its energy, either into the outer air or into the condenser.

(*b*) **Pressure Turbines with Leading Apparatus.**—Since every concussion is accompanied by loss of energy, the endeavour must be made to bring the steam into the channels of the working wheel with as few concussions as possible. Means of effecting this are at hand in the leading apparatus well known in the construction of water turbines. With the introduction of these the turbine loses the character of a simple back-pressure wheel, in so far that the steam in impinging upon the vanes of the working wheel already performs work by virtue of its volume, and reaction does not take place until it leaves the channel. The place of the reaction arms is assumed by the vanes, which are distributed over the periphery of the wheel, and, for the production of back pressure, are provided with curvatures, essentially as required by the theory of the water turbine. Here also the assumption is made, that the steam from the leading apparatus passes under pressure into the working-wheel channels formed by the vanes, and loses an amount of pressure proportionate to the energy given off by it in flowing through these channels. The full temperature of admission takes into account the temperature of the fresh steam and of the subsequent clearance space losses—losses undergone by the steam at the points where it passes from the leading apparatus into the wheel—which are sought to be minimised by the use of special contrivances.

In the *Heilmann* turbine[1] (1898) (Figs. 45 and 46) the middle fixed cylindrical part F, which supplies the steam, is completely enclosed in the revolving part M, which forms the turbine proper. This latter is in its turn enclosed in a steam-tight covering, which at the same time forms the outlet for the steam, and through which one of the shafts of the turbine projects. In the turbine illustrated in Fig. 45 the steam passes through the central inlet passage T into the fixed cylindrical steam chamber or guide-vane apparatus denoted by F, out of which it streams through the channels D. In this the steam impinges against the vanes M of a working wheel R, that is free to revolve round the chambers F. According to the drawing, the

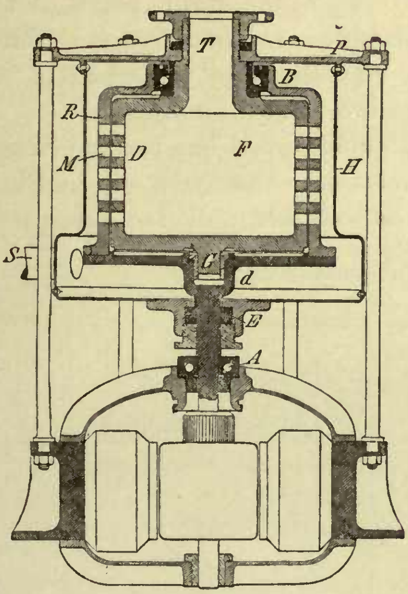


Fig. 45.

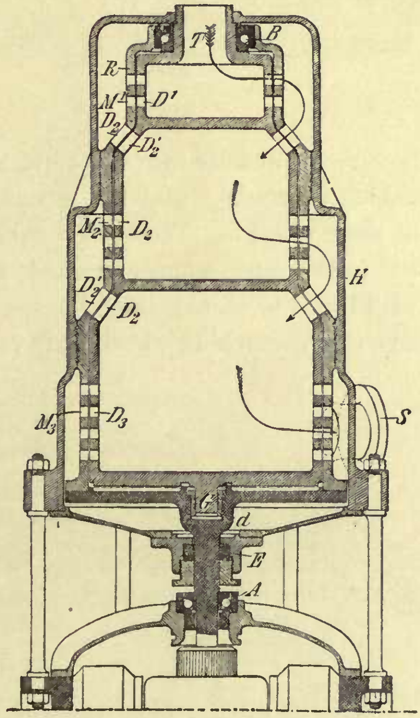


Fig. 46.

working wheel M runs in ball bearings at A and B. After the steam has parted with its energy it streams into the covering H, and from there exhausts through the pipe S into the air. A stuffing box E prevents its finding another way of escape. In Fig. 46 the same idea is applied to a triple-expansion steam turbine, which may here be mentioned. The steam passes at T into the turbine D^1M^1, in which the first expansion takes place. It then streams through the openings D_2, D'_2 to the turbine D_2M_2, expands for the second time, and passes on to the low-pressure turbine D_3M_3, in which the third expansion takes place. From here the steam is led into the air or into the condenser. The three guide wheels of the turbine are combined in a single casting, as are also the three working wheels.

[1] D. R. P. 102,255.

So far as an opinion can at this stage be formed, the idea of allowing the steam to work by stages is attaining practical importance, not in connection with the reaction wheels first treated of, but only in connection with those pressure turbines in which leading apparatus is fitted.

Hoehl, Brakell, and *Günther*[1] have worked out a turbine in the manner shown in Fig. 47. In this design a central ring chamber *b* is provided, in which the steam finds entrance through the openings *a*, and is then led through the guide vanes C to the vanes *d* of the working wheel belonging to the first stage. After having accomplished its work in this step the steam passes at about half its original pressure into the outer ring chamber *e*, and from this goes through the working channels *f* into the second stage represented by the wreath of vanes *g* of the same wheel *h*. The outflow takes place through the openings *k*.

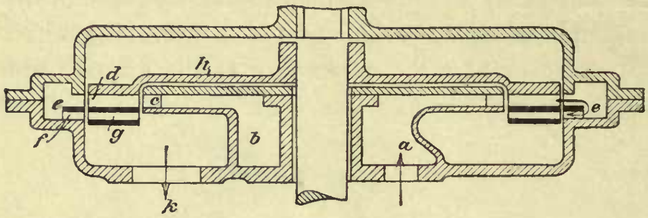


Fig. 47.

Last[2] in the same manner arranges several radial wheels in a row upon a single shaft with the view of reducing the speed of rotation. As may be seen in Fig. 49, the disc *e* of the working wheel bears vanes *v* on each side, while the corresponding

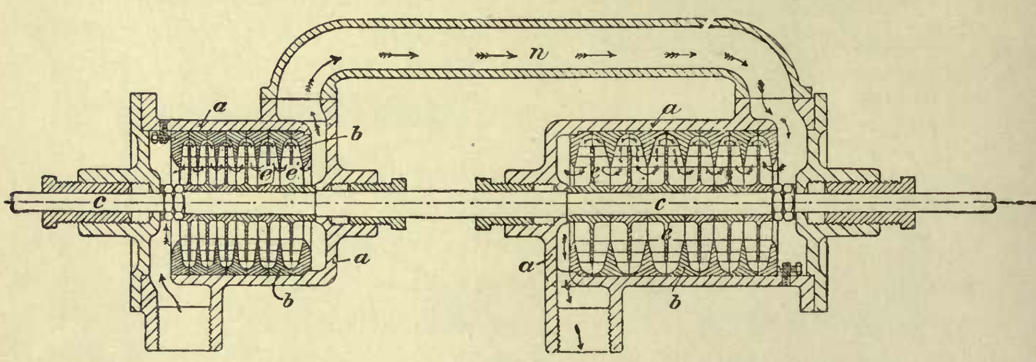


Fig. 48.

guide vanes *g* are arranged on the case *b*. Since the steam working by reaction streams in a radial direction alternately inwards and outwards, a radial turbine impinged upon from within is, in the case of each working wheel, combined with another radial turbine impinged upon from without. Fig. 48 shows several working wheels of this kind placed side by side in a single case. The steam pressure decreases from stage to stage, its volume meanwhile increasing, of which circumstance due account is taken in the widening of the channels of the working wheels and leading apparatus. Moreover, the pressure stages are arranged in two cases, partitioned off from one another, which are connected together in such manner by the pipe *n* that the axial direction of the steam in the second set of machinery is diametrically

[1] E. P. 2429 of the year 1863.

[2] E. P. 3885 of the year 1885.

opposed to that in the first set. By this arrangement the axial effort of the steam is intended to be neutralized.

In the radial turbine designed by *Mackintosh*[1] several working wheels are arranged side by side on a single shaft. In front of each working wheel is arranged a leading apparatus, which consists in a ring provided with steam-passage holes. The steam is, for instance, first led to the first stage in which impingement takes place in a radial direction from within. On leaving the first stage it collects in a ring chamber the outward boundary of which is the turbine case. From this chamber the steam passages of the next-following guide ring admit the steam, now reduced in energy, to the second working wheel, on which it impinges in a radial direction

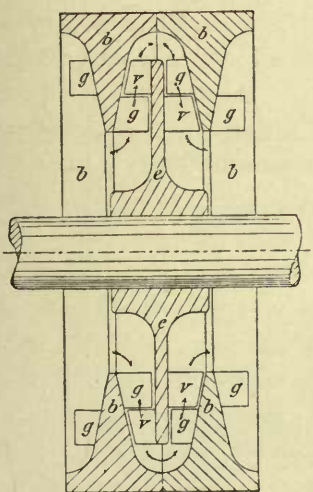


Fig. 49.

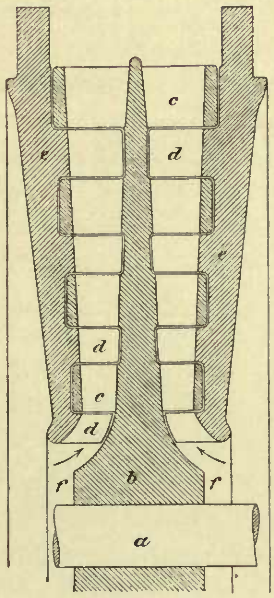


Fig. 50.

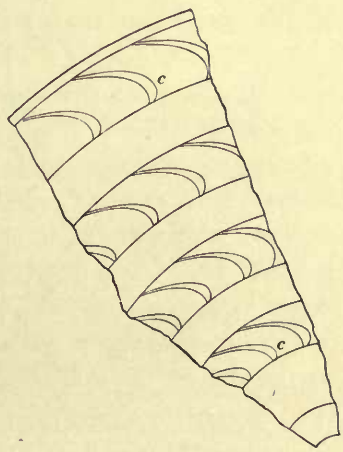


Fig. 51.

from without. In this way the wheels are impinged upon from without and from within alternately. The arrangement may also be such that the wheels are impinged upon in succession from within only or from without only. In all cases, however, the diameters of the wheels decrease in the direction of flow of the steam, while on the other hand their breadths increase in due relation to its increase in volume.

An arrangement in which the radial direction of flow of the steam is not reversed may be made in the manner chosen by *Cutler*,[2] amongst others (Figs. 50 and 51). The shaft *a* bears a disc *b*, on which concentric wreaths of vanes *c* are arranged. Between these pass the guide vanes *d*, which are attached to the side discs *e* fixed in the case. Similar guide vanes are also fixed in front of the first group of guide vanes *c*. The steam enters at *f*, sweeps through the working wheel in a radial direction, and escapes at the periphery into the air, or into the condenser. The vanes and the chambers bounded by them widen out in the direction of flow of the steam,

[1] E. P. 22,369 of the year 1896. [2] E. P. 5022 of the year 1879.

in order to give the latter the chance of expanding, and in order at the same time to increase the area of the steam-pressure surfaces. In case of a reversal of the direction of flow of the steam, *i.e.* when a radial turbine with impingement from without is to be constructed, the vanes will have to be widened in an increased degree in the axial direction from the periphery towards the shaft. A further retardation of the speed of rotation (and at the same time a compounding effect) is thereby attained, that the discs bearing the guide vanes are in a suitable manner arranged to revolve in a direction opposite to that taken by the working wheel. It is claimed that axial turbines may be designed in a similar manner by the fitting

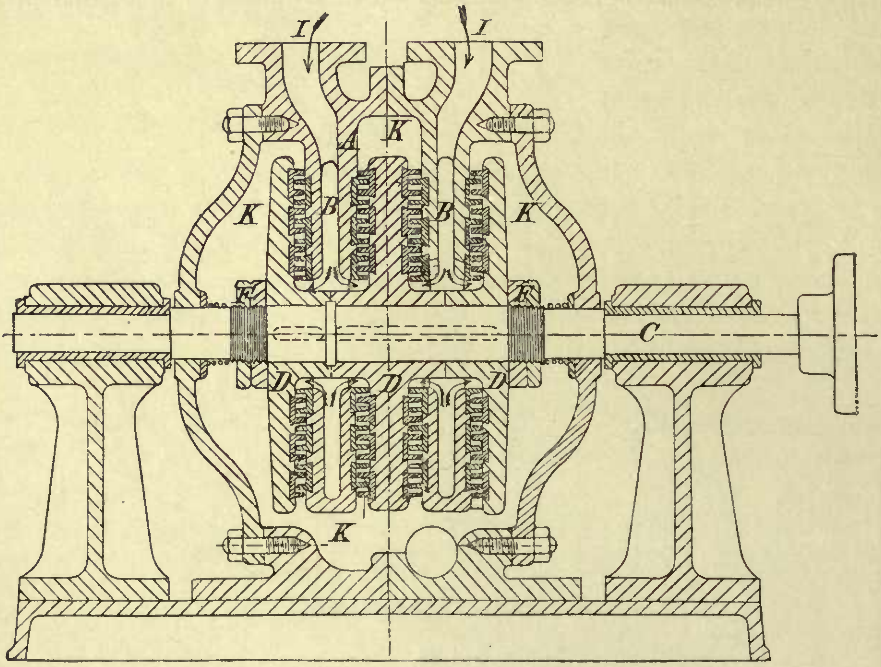


Fig. 52.

of cylinders or cones with vanes. Cutler also thought of the employment of steam jackets.

Clarke[1] avoids the use of the hollow shaft for the supply of the steam for radial wheels that are impinged upon only from within, by the arrangement of a steam pipe in the manner shown in Fig. 52. To the inner ends of the steam inlets I (within the turbine chamber) are connected the flat ring-shaped steam chambers B, the walls A of which reach so far towards the grooves of the working wheels D that sufficiently large ring-like slits remain for the passage of the steam. The walls A bear the axially arranged guide vanes, between which the vanes of the working wheels D then pass. The vanes are mounted, a pair on each ring, which latter is let into a working wheel or into a guide wheel A. The steam streaming through the wheels from within and from without is able, on its way, to expand before it

[1] E. P. 10,630 of the year 1889.

arrives in the turbine chamber K. The working wheels being impinged upon in pairs, no one-sided axial pressure is set up. The guide and working wheels can be shipped, one after another, on the shaft, and be fastened together by means of the nuts F.

Lount[1] also makes use of a disc as bearer of the working-wheel wreaths. He causes the steam to enter in an axial direction at both sides, and leads it off at the periphery of the case. The clear section of the latter, however, is curved out in parabolic form. The object of this measure is not quite clear. The obvious effect of it is only a narrowing of the free passage for the working steam, and in conjunction with this, apparently, an increase of the resistance only within the sets of working wheels. Possibly, however, effects are here produced resembling those in the Laval nozzles. The steam can in the first place experience an acceleration resulting from the narrowing of the clear section of the wheel, and then a second one due to its expansion in the widening portion of the turbine. This process would indeed stamp the Lount turbine as a combination one, since it begins with the pure pressure stages, but in its later phase shows the velocity stages, which will be considered further on.

Here, also, the earlier *Parsons*[2] method of construction may appropriately take its place. In this a number (5) of multi-wreath radial turbines of like diameter (for high pressure) are traversed, one after another, by the steam, which impinges upon the various sets of wheels from within. Following the latter are two larger wheels for low pressure. In 1892 the first engine of this kind with condensation was constructed, for the purpose of driving an alternating-current dynamo of 150 Kw. It was of 200 H.P., and ran at 4800 revolutions per minute. When moderately superheated steam of 100 lbs. per sq. in. pressure and 28 ins. vacuum was made use of, the turbine consumed 27 lbs. per Kw. per hour, which is equivalent to 16 lbs. per H.P. per hour.

Dow and *Chisholm*[3] (1890 and 1893) find room also for the pressure stages in the radial wheels (Fig. 53). These wheels E are placed in a row on the shaft D. The guide-vane bearers are in each case in one with the sections A of the case. In the middle only, at the place where the steam inlet is arranged, two ring-shaped guide-vane bearers *c* are specially screwed into the case section containing the steam inlet N and the outlet M. The leading channels *c* and the working-wheel channels *e* take the forms shown in Fig. 54. The channels *e* are made V-shaped, in order to give them greater strength. They overlap one another in so far that the knee projects more or less between the flanges of the next-lying vanes, but lies somewhat nearer to the inner flange of the next vane, so that the inner part of the passage is somewhat narrower than the outer part. Moreover, the clear widths of the channels gradually increase in due relation to the expansion of the steam. On the shaft D a ring disc F is then screwed in such a manner that between it and two other wheel discs G (Figs. 53 and 55) slight amounts of play are left for the passage of the steam from

[1] E. P. 10,296 of the year 1899. [2] E. P. 10,940 of the year 1891. See also Chap. XIII.
[3] D. R. P. 53,711 and 75,389. E. P. 16,072 of the year 1888.

the ring channel L into the chamber O, from which the latter proceeds into the stages of the turbine. The rings G are perforated by axial borings and radial

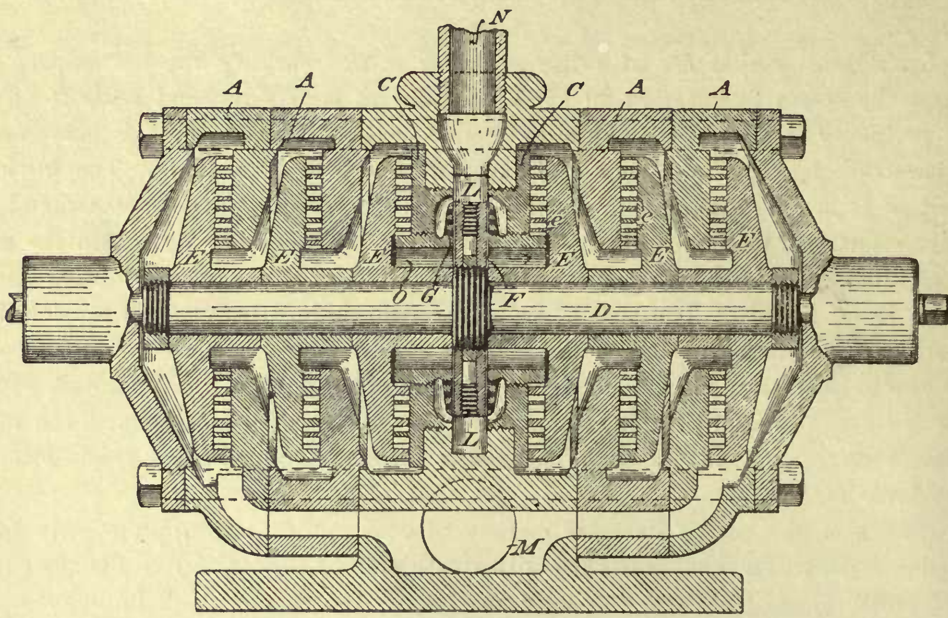


Fig. 53.

channels, and have, at the sides presented towards the ring disc F, ring grooves, which connect the radial channels with one another. Arrangement is thus made for

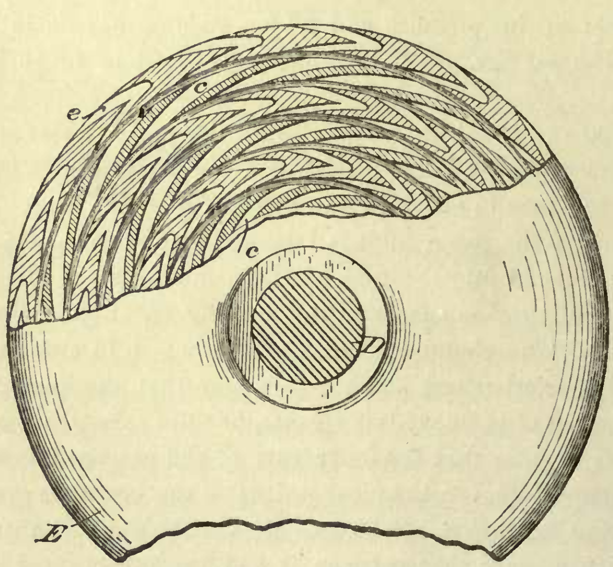


Fig. 54.

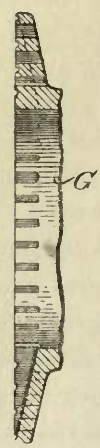


Fig. 55.

the free through-passage of the steam. The shaft bearings are arranged to give a little, so that an axial movement can on occasion take place; but this is, on account of the small amount of play allowed between the guide vanes and the working-wheel vanes, to be avoided. This purpose is served by the disc F. So long as everything is in order, the disc F revolves at equal distances from the rings G. Should, however, the shaft D be subjected to a movement to one of the sides, the passage of the steam on that side is throttled, and the over-pressure on the opposite side causes the wheels to return to the middle position. This regulating appliance can be divided and removed to the ends in cases in which the steam is introduced at the ends of the turbine and flows out at the middle.

If, as in the arrangement made by *Pyle*[1] (1894), the steam is allowed to enter the wheel in a radial direction from without, a turbine is obtained which is, in a

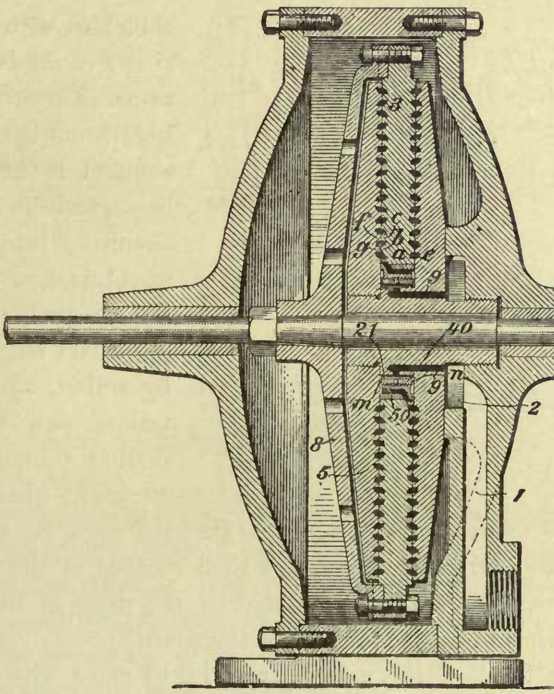


Fig. 56.

manner, self-regulating, as is the case also in the well-known water-power engines. The breadth of the revolving guide-wheel wreaths of the Pyle turbine increases gradually in the direction leading radially inwards.

In the wheel designed by *Edwards*[2] (1892) (Figs. 56 and 57), which is intended to allow action and reaction of the steam to take place in separate periods, the circumstances are different. The fixed guide discs 4 and 5 are attached to the case, and between them the working wheel 3 is free to move. Into the latter concentric ring grooves *a* are screwed, which cover the ring grooves of the guide discs. The grooves leave between them the rings *b* and *c*, of which *c* is divided into segments, which form channels leading in a direction at an angle with the radius in each case—

[1] A. P. 550,564. [2] A. P. 485,536.

viz. the steam passages—while the ring *c* is complete. A ring *c* is followed by a ring *b*, and so on. On the other hand, rings *f* and *g*, which are alternately complete and divided into segments, are in like manner formed by the grooves *e* in the guide discs 5; and the arrangement, indeed, is such that in all cases a complete ring *b* of the working wheel coincides with a divided ring *g* of the guide disc, and, conversely, that a divided ring *c* of the working wheel coincides with a complete ring *f* of the leading apparatus. The steam proceeding through the inlet, the channel 1, the ring chamber 2, and the longitudinal channel *g*, into the working wheel, is now to pass into the channels of the working-wheel rings *c* in order, by action, to produce rotation, and then, in its further passage from the working-wheel channels into the leading channels of the rings *g*, again to perform work by reaction. In this way the steam loses its energy by stages, and then escapes at the periphery of the working wheel. Here, also, is the attachment of the arms of the disc 8, by means of which the revolving wheel is attached to the shaft.

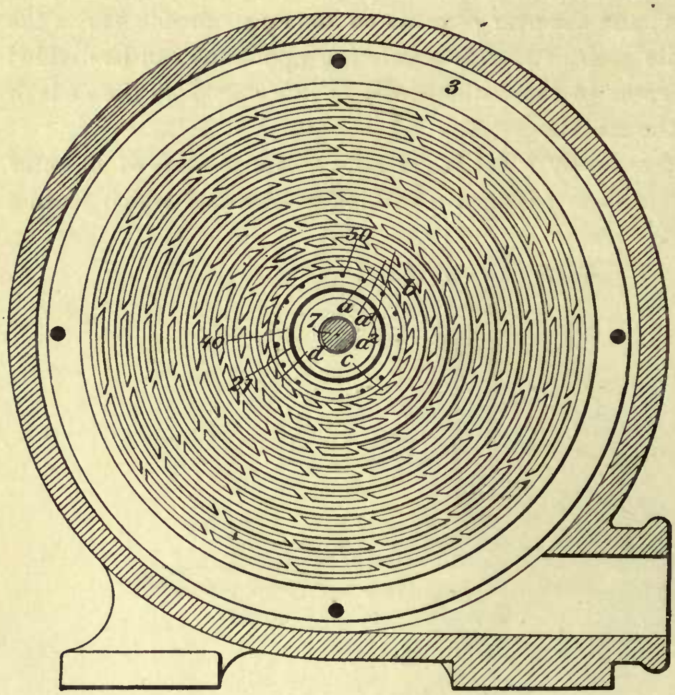


Fig. 57.

Edwards now proposes to allow the working wheel 3 to move with play, to the amount of from 0·001 to 0·0015 ins., between it and the guide disc, and to make the action of the working parts themselves preserve the play thus given. To this end the following arrangement[1] is made (1893): the steam, on issuing from the axial borings 9, first arrives in the ring-shaped collecting chamber 21, which is bounded by a ring 40 connected with the working wheel, but kept somewhat narrower than the latter. This ring allows the steam to pass in the direction of the one guide disc to the channel *m*, and in the direction of the other guide disc to the channel *n*. It further bears the channels 50, which lead the steam crosswise from the channel *m* against disc 4, and from the channel *n* against disc 5. When, therefore, for some reason, the working wheel approaches the disc 5, for instance, the passage *m* becomes narrowed, and the passage *n* widened. It follows, then, that more steam will be given to the side of disc 5, and the wheel

[1] A. P. 505,350.

3 thereby pressed off from this disc until the conditions have righted themselves again.

In order to attain a stage arrangement in axial turbines, *Wilson*[1] (Fig. 58) placed on a cylindrical roller *a* several rings of vanes *b*, and inserted between these the guide-vane rings *c*. The steam issuing from the chamber *d* must, in getting to the outlet *e*, impinge successively upon the various working-wheel vanes and guide vanes. In the course of this the roller *a* is caused to rotate. This is a case of a multi-stage axial pressure turbine with complete impingement. The radial vane breadths increase gradually in the direction of flow of the steam, so that the latter is able to expand. The current energy and the work of expansion are thus both made use of. Wilson, moreover, also provides for the alternative arrangement, in which

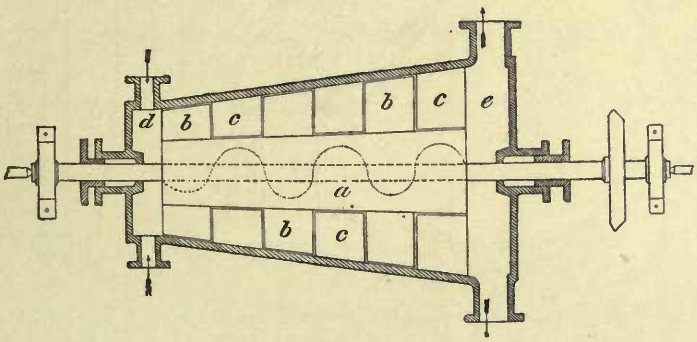


Fig. 58.

the roller is conical in shape, and in which the guide-vane bearer is fitted as a reverse-motion wheel.

Burdin and *Tournaire*[2] (1853) discuss in considerable detail the multi-stage axial pressure turbine, from which the steam is said to issue pressureless and with the least possible velocity. Fig. 60 is an elevation showing the foundation plate; Fig. 59 a second such, showing the impingement side uncovered; Fig. 61 a longitudinal section through a single turbine; and Fig. 62 shows the forms of the vanes. The turbine is divided into several sections, arranged parallel with one another, which have a case in common. This latter rests upon the foundation blocks L^1, while the turbine shafts are grouped around the shaft X that is to be driven. Each of the separate turbines consists of the stage-wheels B, C, E, F, which are held together by tension rods, T, and which are impinged upon from the leading apparatus K, G, H, I. The steam entering through the inlet A streams into the impingement chamber M of the first single turbine, does work in the stages of the latter, and is led back in the axial direction through a communication channel U into the impingement chamber M of the next single turbine, and so on till it arrives almost devoid of energy at the exhaust E. In due relation to the expansion of the steam, the clear widths of the working-wheel channels and leading channels increase from stage to stage; but this increase also takes place in the passing of the steam

[1] E. P. 12,026 of the year 1848. [2] F. P. 8081.

from one single turbine to the other. Further, as regards the vanes, due consideration is taken, in the choice of the forms and positions of these, of a concussionless working of the steam, and of the decrease of its velocity. The shaft O of each single turbine rests at one end in the solid disc L of the first leading apparatus K, and bears at the other end with a hardened point in a hardened metal plate. (The plate may also be replaced by hard stone.) At this latter end, *i.e.* without the steam chamber, sits the driving wheel R, which gears into the main wheel V. The shaft is

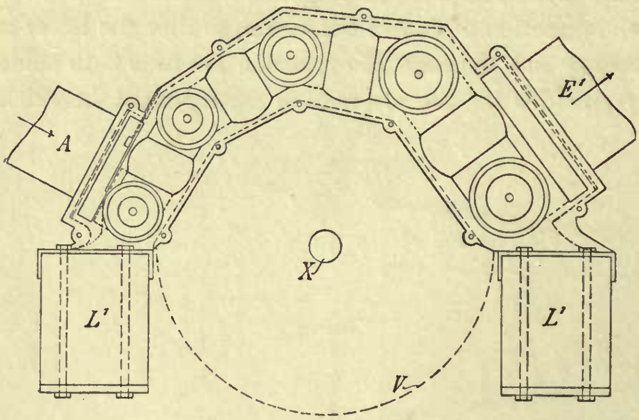


Fig. 59.

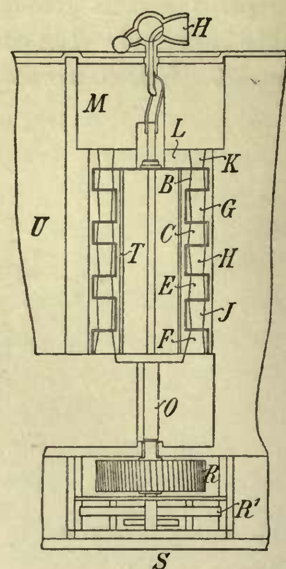


Fig. 61.

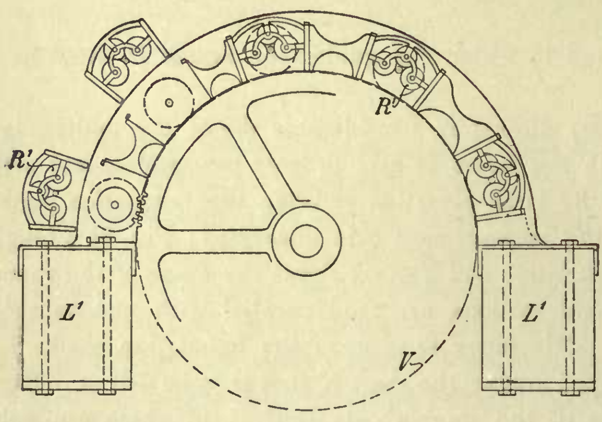


Fig. 60.

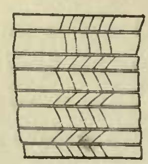


Fig. 62.

meanwhile secured in its place by discs R^1, which grip round it. At the point at which the shaft projects through the case in order to take the driving wheel R, a spring-metal packing is inserted. For the lubrication of the bearing the lubricant can be applied through a funnel H. By a turn of a cock introduced into the supply pipe for the lubricant, so that the latter is put in connection with the steam piping or other source of pressure, the oil may be applied to the bearings under pressure.

A. Müller[1] (1877) sought to accomplish the staging in the manner illustrated in

[1] D. R. P. 196.

Fig. 63. On the shaft *a* several working wheels *c*, of gradually increased diameters, are wedged. In front of each working wheel *c* is arranged a leading apparatus *b* of suitably proportioned diameter, which is in one with the respective section of the case. The steam passes from each working wheel with reduced pressure into the succeeding leading apparatus, which, on account of its larger dimensions (clear width), can give the steam a better chance of expanding. Proportionate to the reduction of pressure suffered by the steam in streaming through any one of the wheels, is the increase in the size of the next following wheel, in order that the smaller power now available may act with a greater leverage. It is the aim, in fact, to equalize the work done in the various turbines. The steam enters through the valve *g* and the inlet pipe *d*, and escapes by the exhaust *e*. The shaft is provided with a belt wheel *f*. Condensed water can be let off by the cock *h*.

A similar arrangement shows itself in the turbine designed by *Weichelt*.[1]

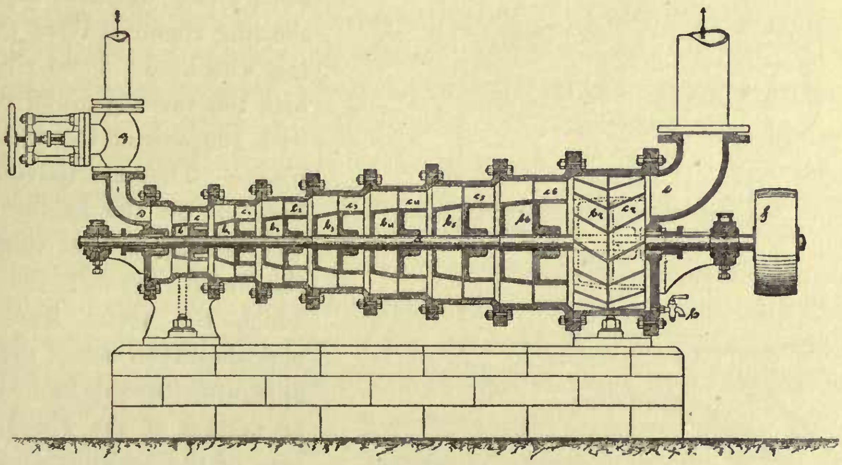


Fig. 63.

In the present *Parsons* turbine,[2] also, axial wheels arranged in stages of pressure are met with, having a case in common, and indeed Parsons, in allowing for the expansion and decreasing pressure of the steam, places together a series of stages of like diameter, followed by another series of larger diameter. Parsons employs a large number of stages, his medium-sized turbines having one hundred. In addition to a diminution of the number of revolutions, he is able to work with a relatively small reduction in pressure between the steps, and likewise with a relatively small excess of pressure at the clearance space. Any steam that may find its way through the clearance space without performing work meets, in the next stage, steam of only slightly less pressure, with which it can mingle without considerable loss of energy.

Huggins and *McCallan*[3] make the working-wheel and guide-wheel cells of the steam inlet and outflow increase gradually in width in the radial direction. The centre lines of the channels are, in each case, at an angle with the axis of the turbine, and the steam is so led as to describe a spiral round the shaft.

[1] E. P. 11,557 of the year 1899. [2] See Chap. XXI. [3] E. P. 23,832 of the year 1897.

Ashton[1] has aimed at the production of a turbine of small compass, in which it is possible to remove the working wheel from the case without it and the guide vanes interfering with one another (Figs. 64 to 67). The working wheel B consists of an inner conical portion and an outer cylindrical one. In the former the fixed cone F is attached to the lid of the case, and the latter is surrounded by the cape H. The cone F and the cape H at the same time bear the leading apparatus. They are provided with ring channels C, the centre lines of which have not the same positions along the axis as those of the ring channels B^1 of the working wheel. In these ring channels the vanes (parallel or askew with the axis of the turbine) are placed. They are halved by low bars, one of the halves of each vane having a level ridge, while the other, towards the end at which the steam leaves it, is curved—in the case of the leading apparatus, forwards in the direction of motion of the wheel; in the case of the working wheel, backwards. To each level vane of the working wheel a curved guide vane is thus opposed, and *vice versâ*, in such manner that level vanes receive the steam delivered to them by the curved vanes. The motive medium, entering through the inlet E, first fills the ring groove of the working wheel. It acts by means of back pressure as it passes into the first ring groove of the leading apparatus, is reversed in direction by the latter, and then acts by pressure on the level-ridged vanes of the next-following ring groove of the working wheel, and so on. Thus the reaction of the steam passing

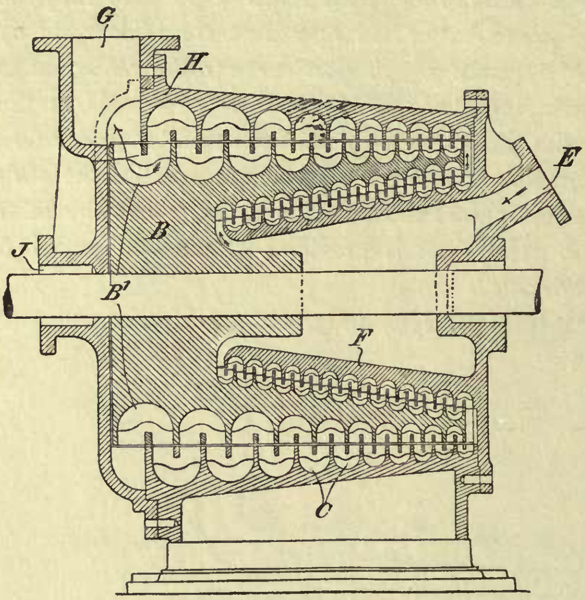


Fig. 64.

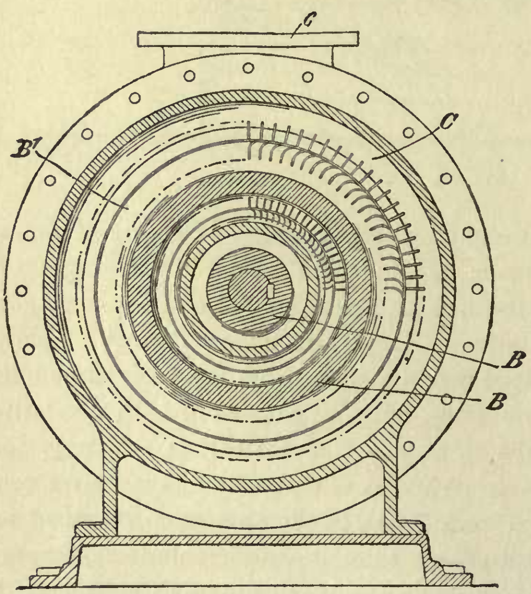


Fig. 65.

[1] E. P. 5198 of the year 1900.

from the working wheel is followed by its action in passing into the latter again, so that it influences first the inner, and then the outer wheel vanes, and finally escapes through the exhaust G. The expansion of the motive medium is proportionate to the widening of the working chambers. Here there are several radial

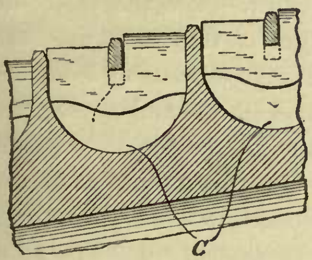


Fig. 66.

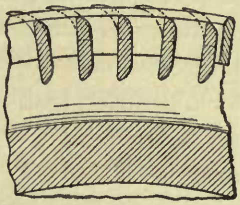


Fig. 67.

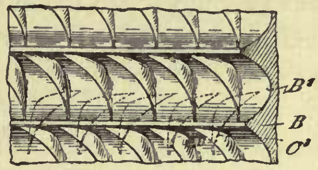


Fig. 68.

wheels set one after another in the axial direction. Since the vanes leave the hollows of the ring grooves free, an adjustment of pressure can take place in each of the latter. In the alteration illustrated in Fig. 68 the vanes are set in the ring grooves B^1 of the working wheel B without the dividing bars, but, to make up for these, are so curved that their level-ridged portions, which receive the steam, gradually merge into the curved portions which guide the motive medium onward. The corresponding guide vanes C^1 are shown dotted.

Finally, attention may be directed to the unimportant combination of *Linscott* and *Hunt*,[1] in which several sets of axial wheels are grouped concentrically round the shaft. In this case the steam, after it has streamed through the inner set in the direction of the axis of rotation, has to pass through an outer wreath in the opposite

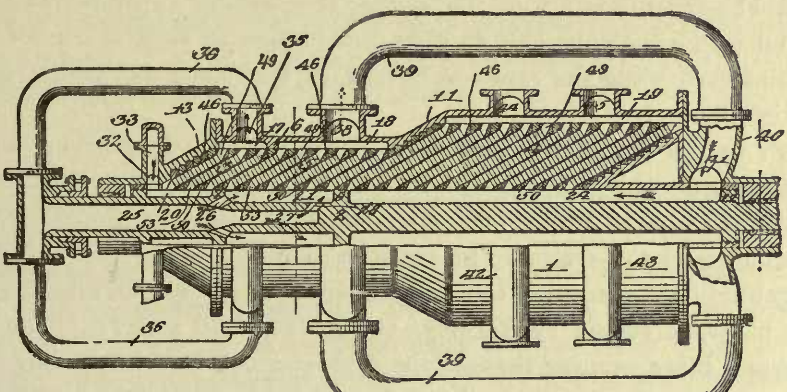


Fig. 69.

direction, but for the most part parallel with the axis, to be then made to impinge upon a further set with the original direction of flow, and so on.

Intermediate between a radial and an axial turbine is the arrangement made by *Asbury*[2] (1902), intended for marine work (Fig. 69). On the shaft 2 several working

[1] E. P. 6497 of the year 1902. [2] A. P. 719,295.

wheels 50 are placed in a row, each of which has a number of conaxial wreaths of vanes. The case 1 also bears a corresponding number of leading apparatus 46, with conaxially arranged wreaths of guide vanes. The working wheels and leading apparatus, however, take the forms of funnels, which are shipped into one another, and the points of which (in the drawing) look to the right, while the bases are towards the left. As may be seen in the drawing, the edges of the vanes are each set at an angle with the axis of rotation, and the steam, always streaming from the shaft towards the wall of the case, likewise does so in funnel-like chambers shipped into one another, the points of which, however, look to the left. The steam chambers are separated by bars, which are set between the channels of each working wheel and each leading apparatus. The turbine is divided into three chambers—a high-pressure chamber 3, an intermediate chamber 5, and a low-pressure chamber 4. The fresh steam, entering through the pipe 33 into the ring channel 32, fills the longitudinal grooves 20 of the shaft 2, which are closed towards the right by the band-projection 7. It proceeds through the openings 53 in the first place to three groups of guide and working wheel channels, out of which it flows through the openings 49. From the communicating ring chambers 17 and 35 the pipes 36 lead into the passage 25 in the shaft 2, which, by means of the holes 26, 27, 28 communicates with the longitudinal grooves 21 of the shaft bounded by the bands 7 and 9. The steam now performs work in six working wheels of the intermediate chamber 5, passes into the ring chamber 18, the end of which is closed by the ring 6, and is led by the pipes 38, 39, 40, 41 into the grooves 24 of the shaft, whence it proceeds to perform work in 12 wheels of the low-pressure chamber 4. Thence it passes to the chambers 19, and it finally escapes through the pipes 42, 43, 44, 45. Where the case widens out—at 13 and 11—the diameters of the wheels and leading apparatus also increase. Further, Asbury claims to be able to alter the turbine in such a manner that the shaft remains fixed while the case revolves, or so that shaft and casing revolve in opposite directions.

Asbury's design cannot be regarded as advantageous. It provides an engine, indeed, which, other things being equal, is smaller in diameter than a radial turbine, and shorter than an axial turbine, but the inlets and outlets for the steam, and the packings which have to be resorted to, are open to question.

A tangential impingement of the working wheel C, and an axial leading along of the steam has been chosen by *McElroy*[1] (1892) (Figs. 70 and 71). The working-wheel channels H are arranged between the guide channels G of the leading apparatus M, which is fixed to the case B. All the channels widen out in the direction of flow of the steam, in accordance with its increase in volume. The steam enters the ring channel F^1, and then divides its flow to pass through the turbine in both axial directions. By this means a one-sided axial effort is prevented. The turbine shaft A runs on ball bearings. In order to further relieve these from the weight of the working wheel, McElroy proposes to throttle the exhaust, so that a contrary pressure is produced, which is intended to slightly raise the wheel (?).

[1] A. P. 510,483.

Although *Lindmark*[1] (1902), in his method for the driving of multi-stage turbines, has the employment of back-pressure wheels immediately in view, yet his invention is applicable to the turbine proper, and can therefore appropriately be considered here. The gist of the method is that the motive medium, after it has streamed through a turbine section, and given over to the latter a certain amount of work, is introduced into a channel of special form, in which an absolute velocity of outflow is again transformed into pressure. On leaving the channel the motive medium thus has a reduced velocity and a greater pressure than when passing from the turbine section into the channel. From the latter the motive medium is led to a second turbine section, or to a second part of the same turbine section. After it has streamed through this second turbine section, it is led, in the same manner as before, into a channel of such form that its outflow velocity is again transformed into pressure. In this way the motive medium is passed through a number of turbine sections, which are thereby made to rotate, and while it is streaming through the specially formed channels leading from section to section of the turbine, its velocity is on each occasion converted into pressure, until, finally, it is in the condition of expansion of the outer air or of the condenser. Fig. 72 illustrates the application of the method to a radial

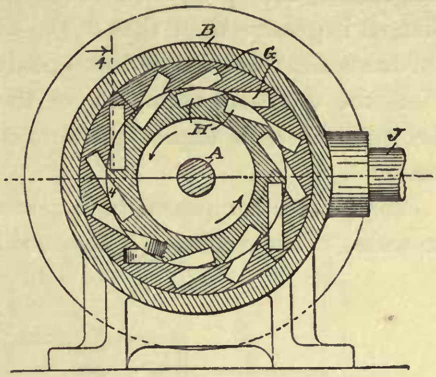


Fig. 70.

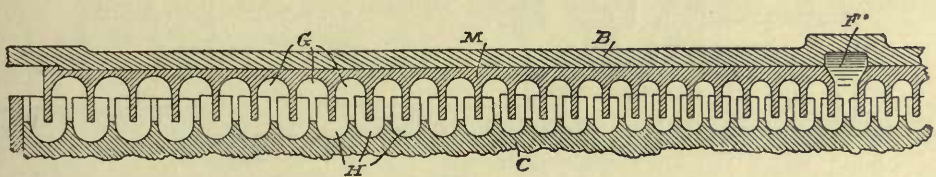


Fig. 71.

compound turbine with impingement from within, the course taken by the steam being clearly shown.

A number of turbine sections I, I^1, I^2, I^3, I^4 are fixed on a common shaft N, and arranged in the respective cases Q, Q^1, Q^2, Q^3, Q^4. Each section is provided at its periphery with a number of vanes, the entrance holes of which are larger than the outlet holes. The turbine-section cases Q, Q^1, Q^2, Q^3, Q^4 are provided at their peripheries with channels C, C^1, C^2, C^3, C^4 which widen in the outward direction. The channels C, C^1, C^2, C^3, C^4 are fitted in such a manner that their openings *o* lie opposite the openings *k* of the working-wheel channels, and as near the latter as possible, the breadth of each of the openings *k* being suited to that of the corresponding opening *o* aforesaid. In the outlet holes of the vanes of the first working wheel I the pressure of the steam is at least 58 per cent. of that in the

[1] D. R. P. 142,964.

entrance holes *o*. During the passage of the current through the channel C the absolute outflow velocity of the steam from the turbine section is wholly or partly transformed into pressure in such manner that the pressure in the outlet of the channel is greater than that in the outlet-passage openings, this being due to the circumstance that the outlet openings of the next-following turbine section I^1 effect the necessary throttling of the escaping steam. The steam enters through the inlet L^1 at a higher pressure than that with which it leaves the turbine section I.

The process is repeated in the same way in the other wheels until the steam has expanded to the condition it assumes in the outer air or in the condenser.

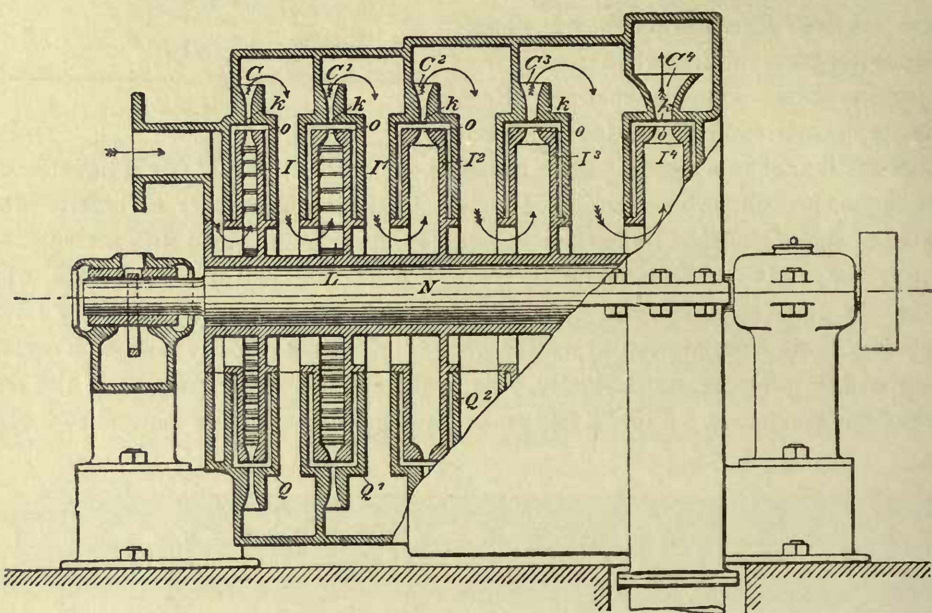


Fig. 72.

Should only one turbine section, or trunk, be employed, the result would be a style of construction as shown in Figs. 73 and 74, the principle of which is likewise that of the pressure turbine. The vanes 1 reach from the periphery of the wheel to its nave, and the parts of them denoted by 2 project into the central inlet L as far as flange 3, which is in one with the nave. A ring-formed channel C is fixed opposite to the outlet openings of the turbine. The case, or more precisely the chamber V, into which the above-named channel leads, is, however, divided by the partition-walls 4, 5, 6, 7, 8, which reach from the periphery wall of the chamber to the central inlet L, into four compartments, 9, 10, 11, and 12. By means of the walls 13, 14, 15, 16, 17, 18 the channel is divided into five compartments, the first four of which 19, 20, 21, 22 are connected, each with one of the compartments 9, 10, 11, 12 of chamber V, while the fifth compartment 23 is connected with the outlet 24. By means of a radial partition 25 and a partition 26 which is concentric with the central inlet, is formed compartment 27, into which the steam pipe A delivers. The course followed by the steam through the turbine is as follows: Through the steam pipe A it enters compartment 27 and proceeds to the central inlet, streaming thereby into as many of the vane-chambers as the distance between the partition walls 25 and 4 will allow. The steam, then, enters a segment of the

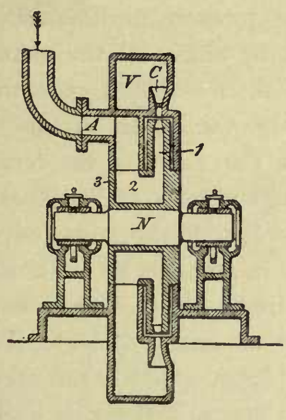


Fig. 73.

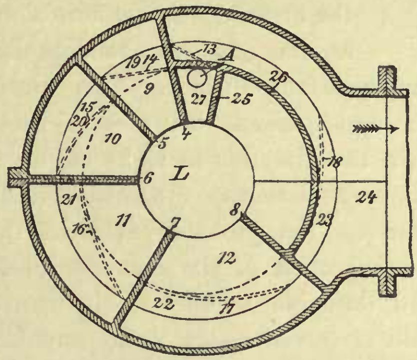


Fig. 74.

turbine-trunk, and passes from it through the compartment 19 of the channel C into the compartment 9 of the chamber V. From this compartment 9 it streams through the central inlet L to the turbine trunk, and there enters the next-following larger segment, which is, in size, proportioned to the distance between the partitions 4 and 5, measured at the central inlet L. From this segment the steam streams into compartment 20 of the channel C to compartment 10 of chamber V, and takes its way into the third section of the turbine trunk, passing in turn through compartment 21 of the channel, compartment 11 of chamber V, the next-following segment, compartment 22 of the channel, compartment 12 of chamber V, the last segment, and compartment 23 of channel C to the outlet 24.

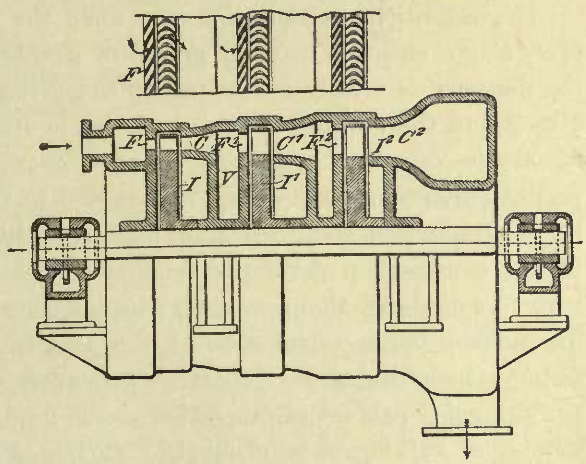


Fig. 75.

Fig. 75 shows an axial pressure turbine with leading apparatus, in which the *Lindmark* method of working has been applied. F, F^1, and F^2 are the leading channels, by which the throttling of the steam, and therewith the increase of its pressure in the channels C and C^1, is effected.

In order, be it remarked, that, in the pressure turbine, the steam issuing from the working wheel 6 (Fig. 76) may with certainty pass into the adjoining leading apparatus *a*, Lindmark (1903) provides the latter with a widened mouth *c*, which catches the steam, even though the wheel 6 should be somewhat diverted in the axial direction.[1]

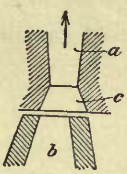


Fig. 76.

For a given performance of the turbine an exactly proportioned quantity of steam is required. In the pressure turbines, in which complete impingement takes place, and in which all the channels of the first stages, as shown above, receive their steam at once, the clear section of these channels determines also the width of the inlet of the steam into the turbine, which must surround the shaft in the form of an annular clearance space. The clear width of the clearance space is regulated by the breadth (in the direction at right angles to the direction of the steam current) of the working-wheel vanes. Since the breadth cannot, in practice, go below a certain limit—Parsons has probably attained this lower limit—there only remains an eventual diminution of the diameter of the annular clearance space, if it be not considered desirable to rely solely upon an extra large quantity of steam. A diminution of the diameter of the annular clearance space, however, is necessarily accompanied by a diminution of that of the working wheel. The result, then, is a reduction in the size of the working wheels, and to obviate an increase in the number of revolutions per minute of the turbine shaft it becomes necessary to increase the number of the stages to such a degree that the circumferential velocity of the individual stage-wheels is proportionately reduced. This method, then, leads to the use of a large number of stages.

The conditions become different when the fresh steam is allowed to impinge upon single channels or upon groups of channels of the first stage. In such case the designer is clearly not bound by the given quantity of steam to a particular diameter of the working wheel. In order to attain few revolutions of the shaft per minute, he can, on the contrary, adopt large wheels, and will thereby require proportionately fewer stage wheels. In regard to the action of the steam in working, there arise, indeed, from this alteration other disadvantages which, while not affecting the construction of the turbine, may form sources of constant loss. For it must here be considered that a working wheel, during the time in which it is entering or leaving the impingement zone of the leading apparatus, only receives a partial filling, which causes an ineffective premature expansion of the steam. The loss thus caused is reduced in proportion as the impingement is spread over an increased number of working-wheel channels. Further, a clearance-space loss will have to be reckoned upon in the direction of the channels that are not impinged upon. If the working-wheel channels are left open towards the chamber to such an extent that they remain constantly filled with steam, though the latter be of low pressure, the importance of the channels that are not completely or not at all impinged upon, as disadvantageous spaces, indeed sinks, but to make up for this the ventilation

[1] A. P. 735,889.

resistance comes more into prominence. There are thus a large number of opposing considerations to be taken into account.

The most important representative of this kind of pressure turbine that is introduced into practice is shown by Fig. 78 in longitudinal section, and by Fig. 77 in cross section, the latter partly following the line X^3X^3 and partly the line X^4X^4.

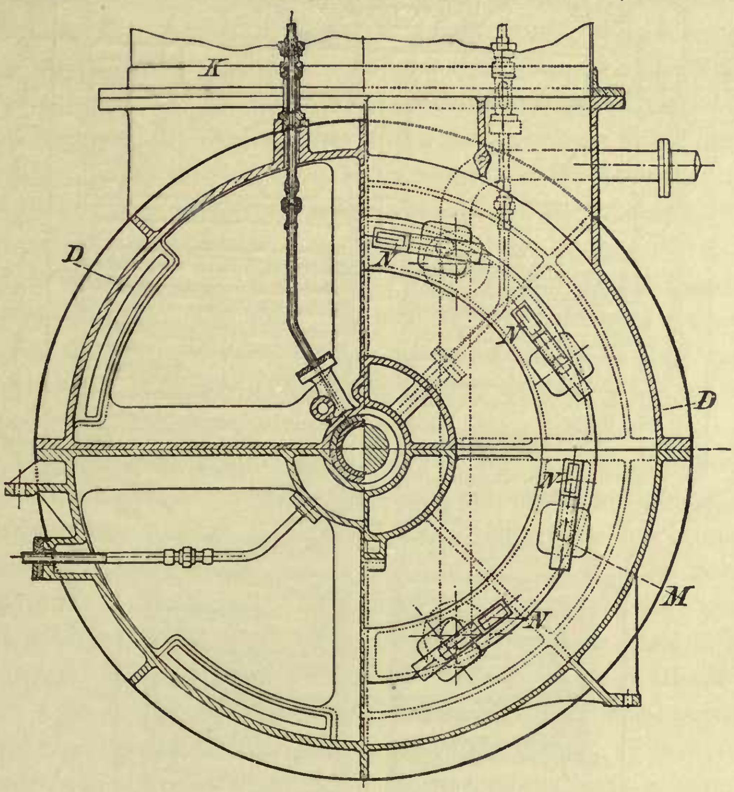
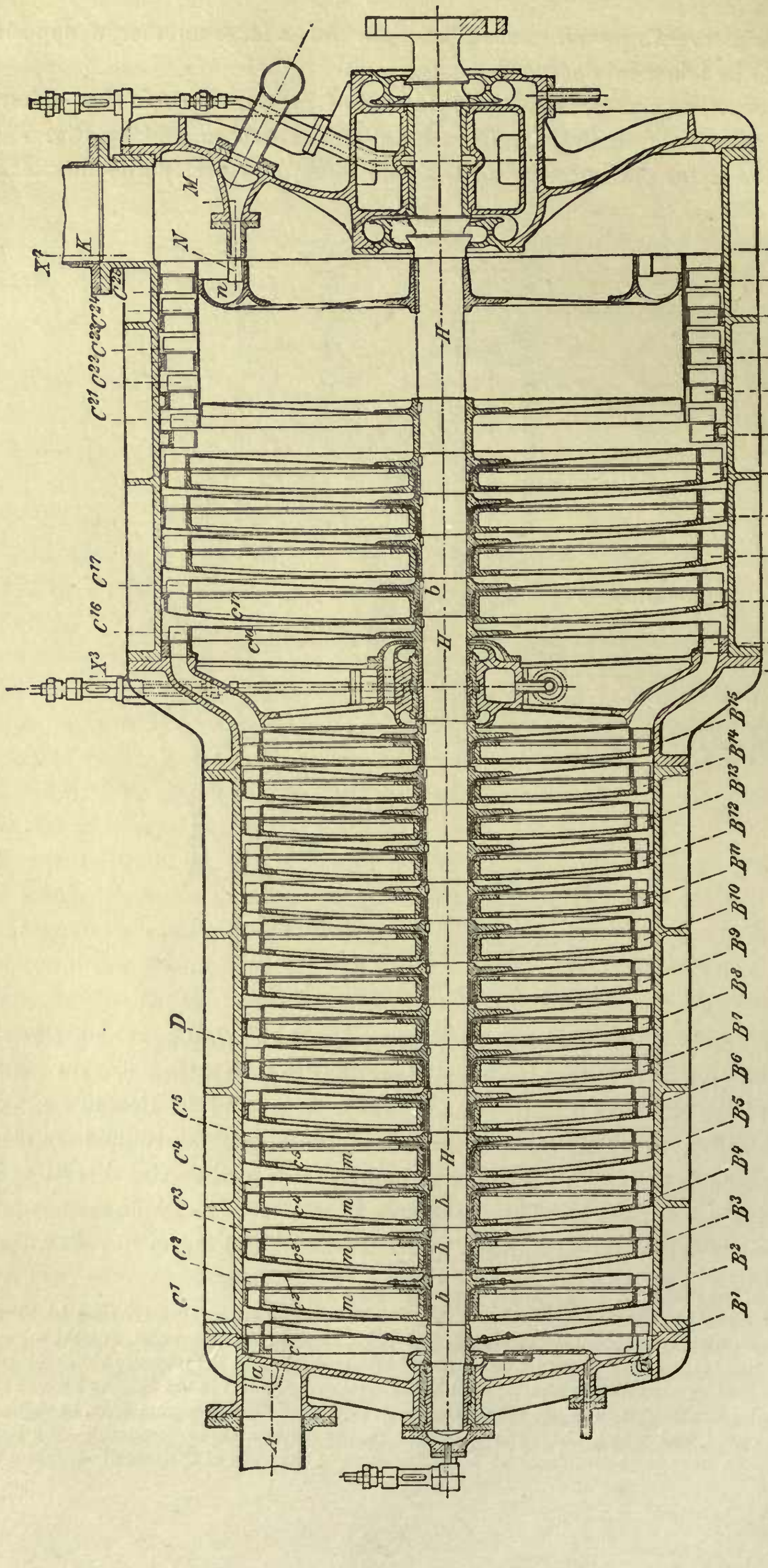


Fig. 77.

This is a turbine designed by *Rateau*, and *Sautter, Harlé and Co.*,[1] in which likewise, according to the view here entertained, the steam has to pass with a certain residue of pressure into the working wheel, which, however, is treated in Rateau's calculations as a pure action turbine. The utilization of the steam is attained by means of the working wheels C^1 to C^{25}, which are impinged upon in the axial direction, and opposed to which are set the fixed guide wheels B^1 to B^{25}. The impingement takes place in such manner that the first guide wheel B^1 allows the steam to pass through

[1] E. P. 24,204 of the year 1898. See also Chapters XIV and XXI.

NOTE.—In order to make the author's intention quite clear, it may here be observed that he assumes that in the first pressure-stages, if considerable ventilation-resistances are to be avoided, at least so large a proportion of the whole fall in pressure must be converted into velocity, that the pressure obtaining before the mouth of each leading apparatus is smaller than the critical pressure. If in the Rateau turbine these prove to be so inconsiderable that the stage subdivision may, even in the high-pressure zone, be reduced—which Rateau intended, and which according to recent results appears to be assumable—the Rateau turbine will, of course, have to be considered as belonging more to the class of the velocity-turbines with pressure-stages.



A
D
H
K
M
N
X³
X⁴
C¹ C² C³ C⁴ C⁵
C¹⁶ C¹⁷
C²¹ C²² C²³ C²⁴
B¹ B² B³ B⁴ B⁵ B⁶ B⁷ B⁸ B⁹ B¹⁰ B¹¹ B¹² B¹³ B¹⁴ B¹⁵
B¹⁶ B¹⁷ B¹⁸ B¹⁹ B²⁰ B²¹ B²² B²³ B²⁴ B²⁵

Fig. 78.

channels which are put together in sectors and occupy a portion only of the periphery. The sectors increase in proportion to the gradual expansion of the steam in its passage through the turbines until in the end they extend over the whole working-wheel wreath. The widening of the steam-passage ways, however, is in the lower stages also effected by an increase in the diameter of the wheel, and, indeed, also by a lengthening of the vanes in the radial direction. From the inlet A the steam reaches the ring channel *a*, and then streams through the various guide and working wheels, gradually expanding and undergoing losses of pressure by stages till it escapes through the outlet K. This mode of working of the motive medium assumes the existence, for the working wheels, of chambers which are formed by the bodies of the guide wheels. With the exception of the first leading apparatus, indeed, the guide rings fixed in the cylinder wall D are connected with vaulted plates M, to which pieces of pipe *b* are riveted. These fit round the naves of the working wheels with only a slight amount of play, forming clearance-openings which alone can give the steam the chance of making an unintended passage through. The vaulting of the plates *m* is in every case in the direction of the chamber having the higher pressure. Plates similarly vaulted and riveted to the naves form the working-wheel bodies, which can thus be made light and stiff, the balancing of the parts being at the same time facilitated. Only in the last stages are several working-wheel vane wreaths attached to a single drum, the partition walls of the leading apparatus being in consequence there dispensed with. At the exhaust side the shaft H bears a single working wheel, which is fed through the nozzles M, and the vanes of which are so formed that it brings about the backward rotation of the shaft. Through the surfaces *n* the outflow from this working wheel is led into the exhaust K direct. From this backward-motion turbine only a small amount of work is expected. According to the drawing, the shaft H, it may be observed, has three bearings, and thus requires lubrication[1] at as many places. The bearing at the condenser side is so arranged that the oil flowing through the lubrication pipe at the same time forms a close packing, since it passes through the ring spaces at both ends of the bearing in each case.

It is worthy of remark that *Parsons*[2] also has thought of having the wreath of vanes impinged on in the first stage by segments only. The numbers of the leading channels were to increase from stage to stage in relation to the expansion of the steam. For this very system of construction, however, Parsons has taken into consideration the application of special clearance-space packing.[3]

This partial utilization of the working wheel for direct impingement in the first stage has suggested the idea of not leading the steam onward to other wheels, but of reversing its direction instead, and leading it back to renewed work in the channels of the same wheel. In this way the pressure turbines with several stages in a single working wheel have been produced.

Among these may be classed the turbine designed by *Wilson*[4] (Figs. 79 and

[1] See Chap. XVIII.
[2] E. P. 7066 of the year 1901.
[3] See Chap. XVI.
[4] E. P. 12,026 of the year 1848.

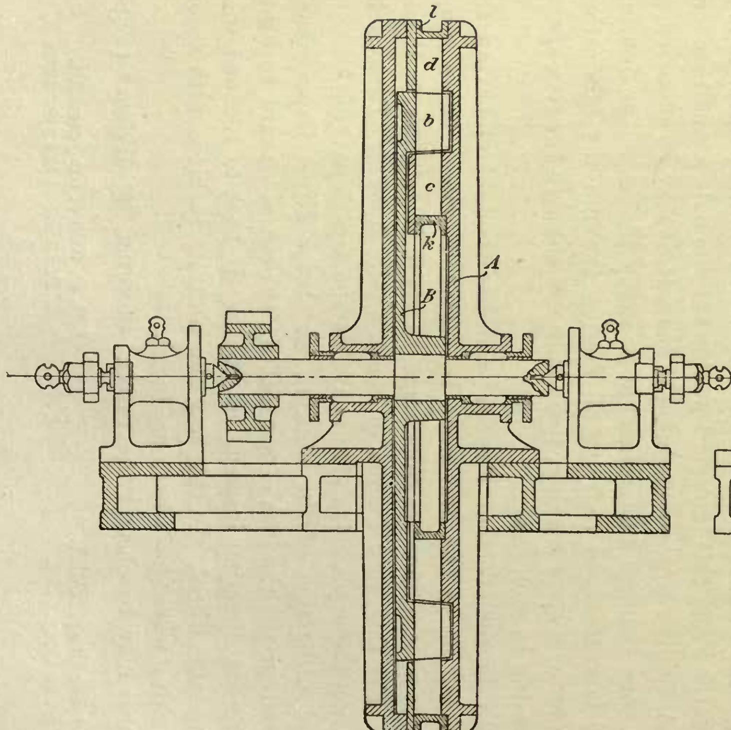


Fig. 79.

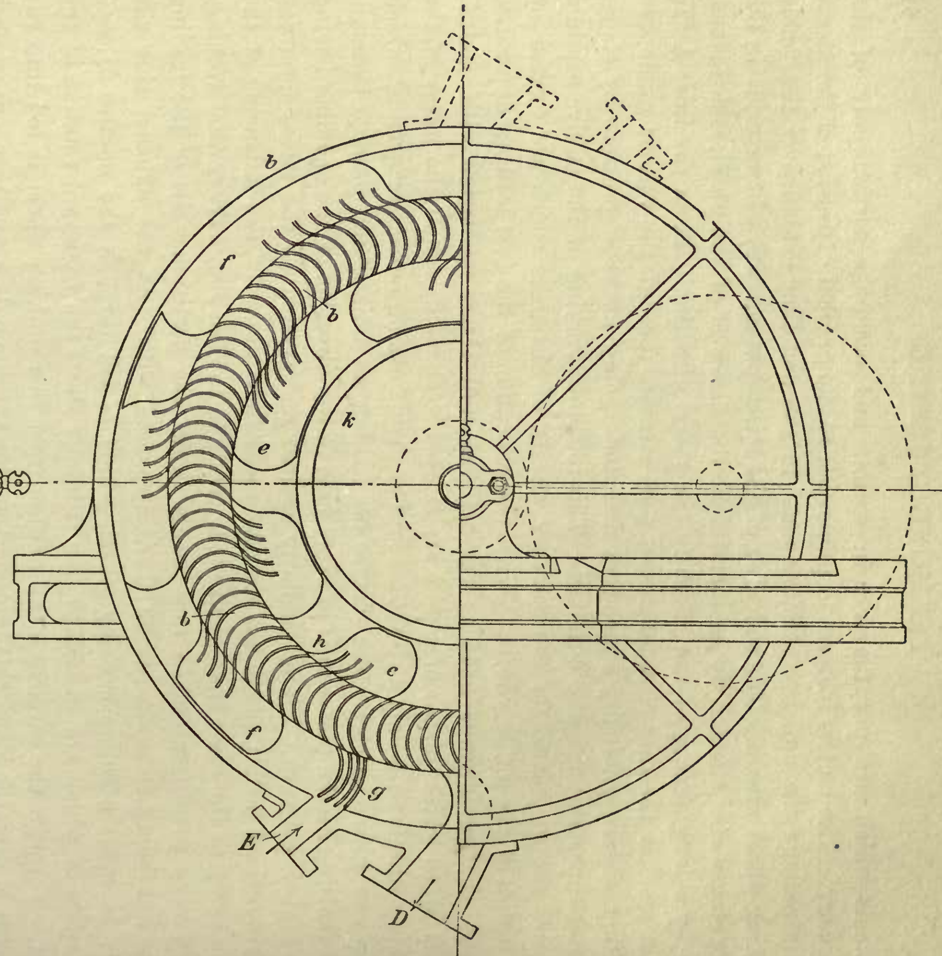


Fig. 80.

80). In the fixed case A the working wheel B is free to turn, its wreath of vanes *b* pursuing a circular path between the concentrically arranged guide wreaths *c* and *d*. The working-wheel vanes are curved in approximately semicircular form. The guide wreaths *c* are provided with the return chambers *e*, and the guide wreath *d* with the exactly similar chambers *f*, while the portions of the guide wreaths between the *e* and *f* chambers respectively are solid. The intended mode of working is, that the steam entering by the inlet E is introduced without concussion through the guide vanes *g* into the working-wheel channels, and after streaming through these arrives in the first return chamber *e*, in which it to a certain extent expands. At the same time, however, the direction of flow of the steam is reversed, and after passing through the fixed vanes *h* the current proceeds in increased volume to impinge upon a proportionately increased number of working-wheel channels. In the next return chamber *f* a further expansion of the steam takes place, after which it again passes through an increased number of working-wheel channels into the second chamber *e*. The motive medium thus takes its way, expanding in each return chamber, in snake-like fashion through the working wheel, till it escapes through the discharge D. Instead of the steam being allowed to enter by one inlet only, and having to make its way right round the axis, an arrangement with two inlets and two discharges (as shown by dotted lines in Fig. 80), or a still further subdivision, could be made. Should it be proposed, as also contemplated by Wilson, to substitute an axial turbine for the radial one, the return and expansion chambers will lie in an axial direction at both sides of the working-wheel vane wreath. Wilson speaks of an action effect of the steam entering the wheel channels, and of the reaction of the outflowing steam. Further, according to the description, the return-chambers of the inner wreath are closed at the inner side of the ring *k*, those of the outer wreath at the outer side by the ring *l*. The covering ring of the working wheel turns in a corresponding groove of the case. The turbine shaft is borne between pointed pintles.

The reduction of the steam pressure by stages in a single-wheel construction is to be met with also in the turbine of *Dumoulin*[1] (1884), (Figs. 81 to 83). The engine is said to bear a general similarity to the *Fourneyron* water turbine. The fixed leading apparatus AA′A″ is made of bronze. The steam passes through the pipe T, which is provided with a regulating cock, through the supply chamber MM and the channels *m*, *m*, into the ring chamber DD. Within the bronze trunk, to right and left of the disc-shaped parts A, four (eventually more or less) groups of eight channels each, exactly alike, are provided, the supply channels *a*, *b*, *c*, *d*, *e*, *f*, *g*, *h*, on the right, and the discharge channels *a′*, *b′*, *c′*, *d′*, *e′*, *f′*, *g′*, *h′*, on the left side. Each of the supply channels communicates at *a′/b*, *b′/c*, *c′/d*, *d′/e*, *e′/f*, *f′/g* or *g′/h* respectively with the base of the preceding discharge channel, with the exception of the first supply channel *aa″* and of the last discharge channel *h′h″* of each group. The first of these channels *aa″* communicate with the ring-shaped chambers DD′ lying to the right of the disc A (Fig. 81), the last *h′h″* with the

[1] D. R. P. 31,095.

ring-shaped chambers EE′ lying to the left. This chamber, which is continued through the channels *nn* to the discharge chamber M′M′ below, leads either into the atmosphere or into a condenser.

The parts MM and M′M′ belong to one and the same long chamber within the central hollow trunk borne by the standards CC, and are separated from one another only by the partition walls P, P¹. The supply channels *a, b, c . . . h,* which from the base to their outflow ends become gradually narrower, as also the similar discharge channels *a′, b′, c′ . . . h′,* increase gradually in section in the order given. The end of each of the supply channels of one and the same group meets

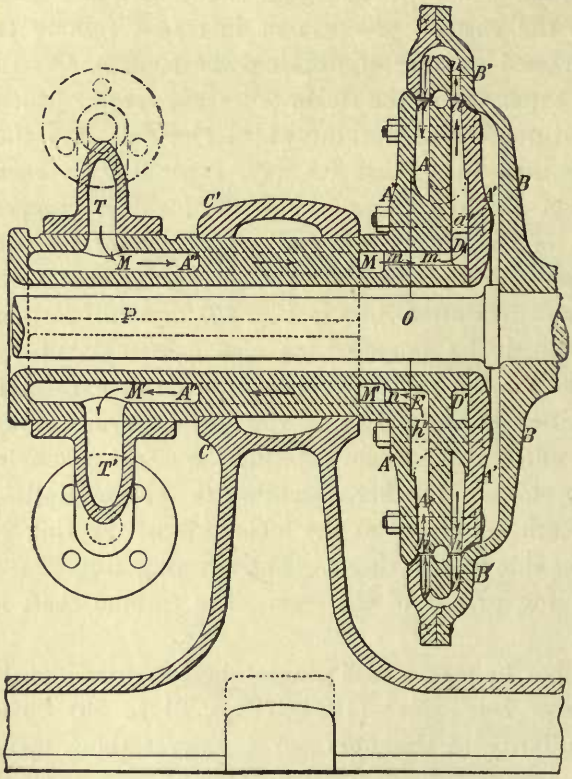


Fig. 81.

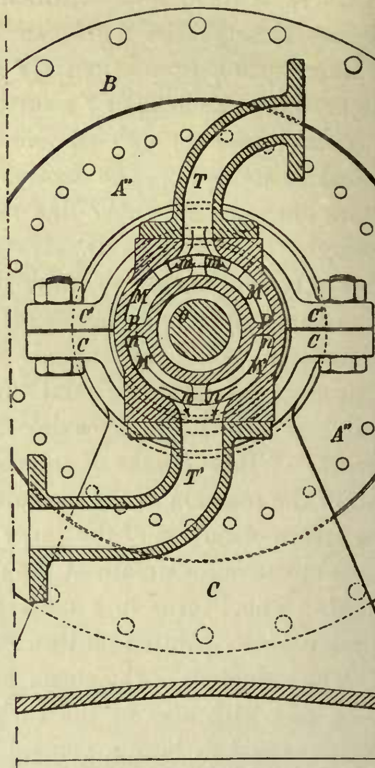


Fig. 82.

at the periphery of the distributing trunk, with the end of the corresponding discharge channel, which, on its part, is in connection with the next-following supply channel at the base, and so on from the first channel *a* to the last channel *h* of each group. All the channels, the outflow ends of which are at the periphery of the distribution trunk, are curved at this point in such a manner that the mean current of steam in each case leaves the supply channels in a direction which forms an angle of from 14° to 16° with a tangent to the periphery, while the direction of the mean current of the steam led off by the discharge channel forms an angle of from 16° to 18° with the same tangent. The cast-iron working wheel BB is wedged on to the engine shaft O, and provided at its periphery with the bronze wreath B'B', containing the receiving parts proper. Only a small amount of play ($\frac{1}{2}$ to 1 mm. at most, according to the diameter) is left at the point of transmission of the current. The wreath BB contains channels u, v of like size and form, running from the one side to the other in curves, which in each case are three times deflected, and which in Fig. 81 show a U-shaped section. The right-hand branches u, the entrance to which is somewhat widened, are so placed as exactly to meet the supply channels a, b, c . . . h, while the left-hand branches v, which are not widened at the points of transmission, meet the discharge channels a', b', c' . . . h', which on their part are so widened at their points of entrance. The curvatures of the branches u at their entrance, and of the branches v at their points of discharge, are so dimensioned that here the mean direction of the motive-medium streaming through them in each case forms the same angle with the tangent to the periphery as do the respective mean directions of the currents issuing from the discharge channels of the distributing trunk and of those entering the discharge channels of the same. The respective tangents to the mean-current directions always coincide with one another, *i.e.* they lie in one and the same direction. The steam, then, imparts a first direct motive impulse at its entrance into the right-hand branch u (Fig. 81), and a second such by the equal and opposite back pressure acting in the same direction at its point of exit through the branches v on the left-hand side. The steam which issues from the ring chamber DD¹, and which is supplied through the first leading channels a'' to the four groups of channels, streams through the latter in the following manner:—

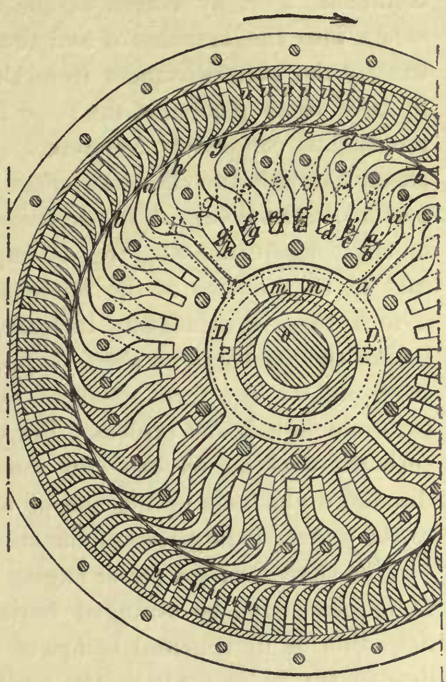


Fig. 83.

$$a''auva'/b,\ buvb'/c,\ cuvc'/d,\ duvd'/f,\ fuvf'/g,\ guvg'/h,\ huvh'h''$$

—and finally passes into the free atmosphere, or into a condenser. The steam thus imparts to the working wheel B'B, in the direction of its rotation, eight direct impulses and eight back pressures in each group, in the course of which it is intended to expand step by step to the pressure of the atmosphere. While streaming through, the motive medium suffers a gradual reduction of its velocity relative to that of the channels. The supply and discharge channels are arranged in such a manner that the coursing of the motive medium from one to another of a group of channels takes place by means of what, in regard to the direction of rotation of the working wheel BB', are backward movements. The arrangement can be such, however, that the steam moves with a forward motion. The detrimental action of the centrifugal force on the steam during its passage through the channels *u*, *v* is intended to be neutralized by the U-form and by the threefold curvatures of the channels. For, by reason of the circumstance that the point of entrance of the steam into the branches *u* and the point of discharge from the branches *v* are at exactly the same distances from the axis of rotation, the centrifugal forces acting in both these places are the same in amount, but have opposite directions. These forces thus completely counteract each other, and their action vanishes. A loss of steam through the clearance space is not anticipated, because the free flow of the current at the points of transmission is so ensured that the action of a side pressure of any consequence whatever is impossible. A means to this end, for instance, is said to be a choice of the sections of the channels of each group, such that from *a* to *h'* these continuously increase, according to the laws of the flow of currents and of the discharge of elastic liquids in relation to their expansion, as also to those in regard to the form assumed by currents of gas when being discharged from certain channels and orifices. For the attainment of a reversal of the motion, two turbines working on one and the same shaft are contemplated, the supply and discharge channels of the one of which are arranged to lead in directions opposed to the corresponding ones of the other.

The shaft O of the power engine is not necessarily horizontal, but may, according to requirements, be arranged horizontally or vertically. In order to avoid condensation or detrimental escape of steam, where such is to be feared, or, again, to deaden the noise made by the motion of the apparatus, the latter may eventually be provided with a mantle-shaped or other suitable cover.

Kemble[1] arranges at each side of the vane wreath of an axial turbine leading channels, which admit of an expansion of the steam in the direction of rotation of the shaft and of its snake-like passage through the cells of the working wheel.

Hewson, Whyte, and *de Rome*[2] (1896) (Figs. 84 to 86) arrange the stages in radial proximity to one another. On the shaft A is fitted the circular disc B, which is thickest at its nave, and becomes gradually thinner towards the periphery. This disc B is provided with rows of indentations, c, c^1, c^2, c^3, c^4, etc. The indentations or pockets c are of U-shape, so that the motive medium streaming through them reverses its direction of motion. The horns of the U lie in the direction of rotation of the wheel, and the deepest point in each case is at the turning-point of the arch of the same letter. The pockets are smallest at the periphery of the disc B where the steam enters, and increase in the direction of the disc B, *i.e.* towards the steam discharge, in suitable relation to the gradual expansion of the motive medium. They are symmetrically arranged on the two sides of the disc, so that the lateral pressure in the direction of the axis is neutralized. Into the case D, at both sides of it, are worked the channels E, E^1, E^2, E^3, E^4, etc., the outermost of which E receives the steam from the boiler, while the others guide it in each case from one of the rows of U-shaped pockets to the next, until the last of

[1] E. P. 18,420 of the year 1900.

[2] D. R. P. 91,619.

the channels delivers it into the open air. Ring-shaped packings between the case D and the disc B prevent the steam from passing directly through between them. From the channel E passages H lead the steam into the outer horns of the U-shaped pockets *c* in such a manner that, striking directly on the arch of the U, it is reversed in direction. It is then made to stream through the inner horns of the U-shaped pockets *c* into the passages I, and through these into the channels E^1. The steam is intended to act, not only by direct pressure, but also by the centrifugal force due to its arc-like course (?). From the channel E the steam, by reason of its expansion, proceeds with reduced pressure and increased volume through the passages H^1 into the U-shaped pockets c^1, and from these through the passages I^1 into the channel E^2. In due relation to the increase of volume of the steam, the volumes of the channels E, E^1, E^2, etc., of the pockets *c*, c^1, c^2, etc., and of the pressure surfaces increase in the direction of the centre. By this arrangement it

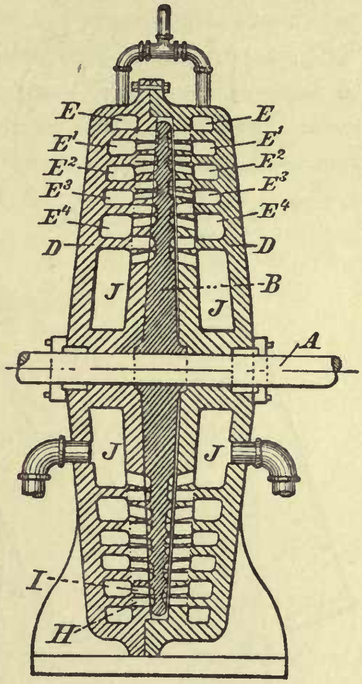


Fig. 84.

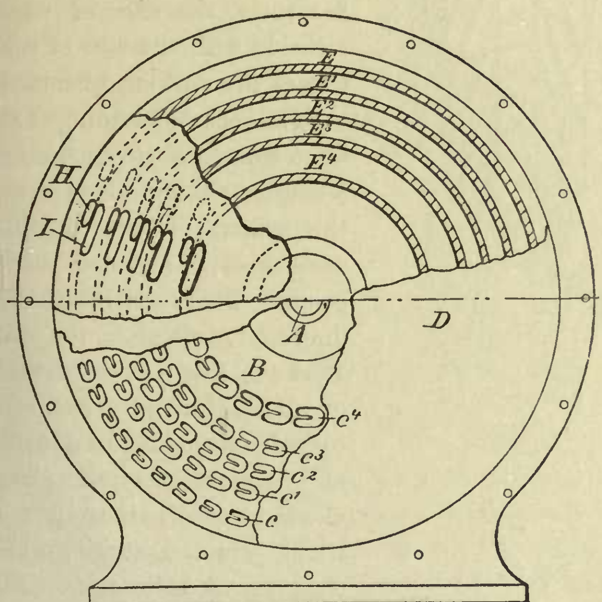


Fig. 85.

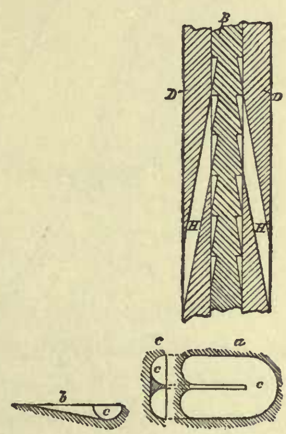


Fig. 86.

comes about that the steam, in proportion to its own decrease of velocity from the periphery towards the centre, streams through pockets of decreasing peripheral velocity, till it passes with its smallest velocity into the discharge channel J, which lies round about the axis A. This style of construction can, if only in view of the unavoidable packing between the case and the working wheel, be credited with no practical importance.

In like manner *Hewson*,[1] in a new design (1902), can hardly be said to have produced anything of practical value, when he arranges the working-wheel channels on the periphery of a roller, increasing in diameter in the direction of flow of the current, and, in due relation to this, also the leading channels, in such a manner that the steam practically follows a screw-like course round the shaft of the turbine.

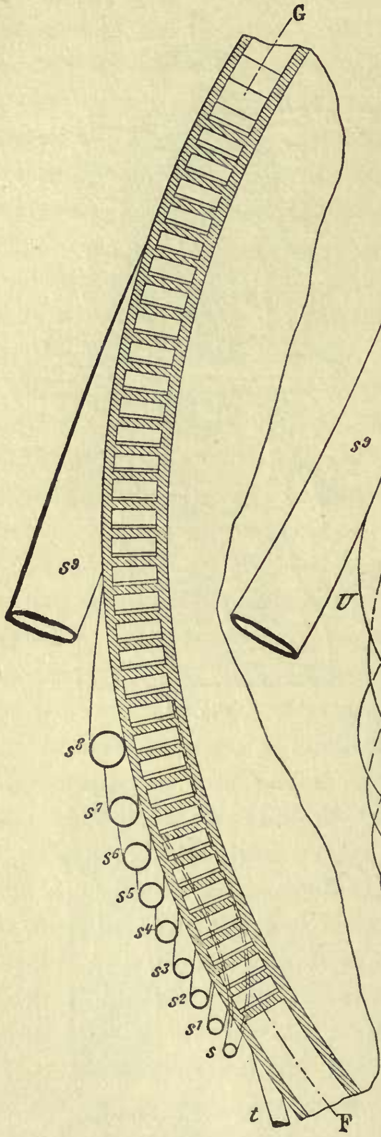


Fig. 87.

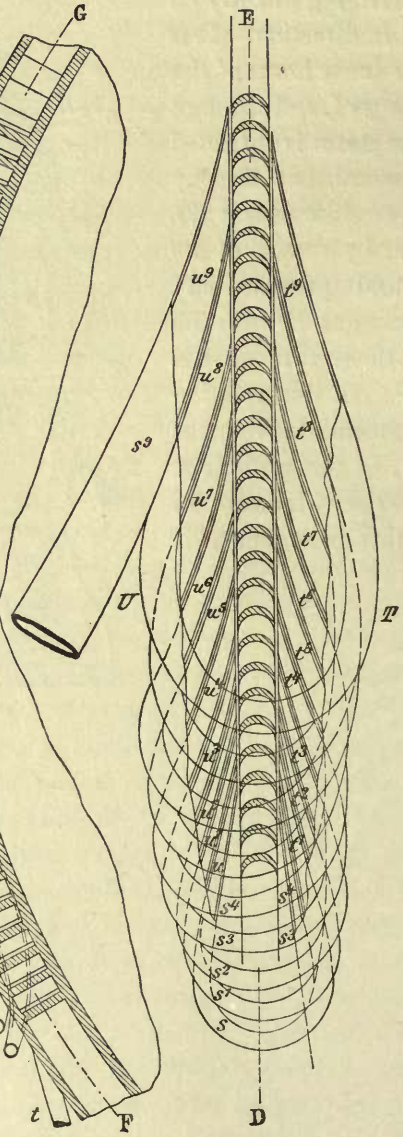


Fig. 88.

Daw[2] has arranged the channel mouths so as to cross each other, and placed them at both sides of a complete disc-formed revolving trunk. The channels run out within two flat rings, the axes of which coincide with the axis of rotation of the working wheel.

The repeated leading of the same volume of steam through a single working wheel is also the salient feature of older designs of *Farcot* and *Perrigault*.[3] Thus, according to a drawing[4] made about the years 1864–65, the wreath of vanes of a wheel axially impinged upon is surrounded by a bundle of circulating pipes placed close together, the ends of which join the working-wheel channels at both sides. Fig. 87 shows a section through DE in Fig. 88, which, on the other hand, shows a section through FG in Fig. 87. First, the steam is led by a pipe t towards

[1] A. P. 708,227.

[2] E. P. 20,907 of the year 1902.

[3] The author assumes that here also, for want of Laval nozzles, and because the working-wheel channels are constantly fully impinged upon, a certain amount of pressure will be used up by the working wheel at every stage.

[4] F. P. 65,640.

one of the working-wheel channels, and in its further course each circulating pipe s receives it through the arm u on the discharge side U of the channels, and leads it back again to the side T (of the wheel), at which the impingement takes place. The working-wheel channels and the pipes s thus form round the working wheel a closed spiral passage, which, with due regard to the expansion of the steam, and, of course, also to its retardation, gradually widens until the current passes off by the pipe s^9. Since the wheel channels retain a constant width, the increase of section is confined to the transmitting pipes, the mouths of which accordingly cover a varying number of working-wheel channels. The pipes lie close together, and their mouths are separated only by the thin walls of the pipes. The drawback is thus to be looked for, that the steam passes from one channel into two neighbouring pipes at once. It is worthy of note that the supply pipe t, as also the circulating pipes s and the discharge pipe s^9, are widened out on their course towards the channels in a direction radial to the working wheel, so that they always cover the whole breadth of the wheel channels. A corresponding narrowing in the directions at right angles to those above mentioned enables the sectional area of each pipe to be kept constant. Arrangements similar to the above, it may be observed, can also be applied to radial turbines. In Figs. 89–91 a working wheel is illustrated which has U-shaped steam channels arranged tangentially to the periphery of the wheel. Fig. 89 is a section at HI in Fig. 91, Fig. 91 a section at JK in Fig. 89, and Fig. 90 a section at $\omega\omega'$ in Fig. 89. In front of the horns of these channels are placed the circulating pipes t, u, so that here also the spiral passages round the working wheel are formed. The horns of the pipes are, however, transposed one with another, so that in each case the mouth of a channel comes just opposite to the entrance to the next tube. These U-channels may also be arranged to open towards the axis, in which case the bundle of pipes will have to be placed between the wreath of vanes and the axis.

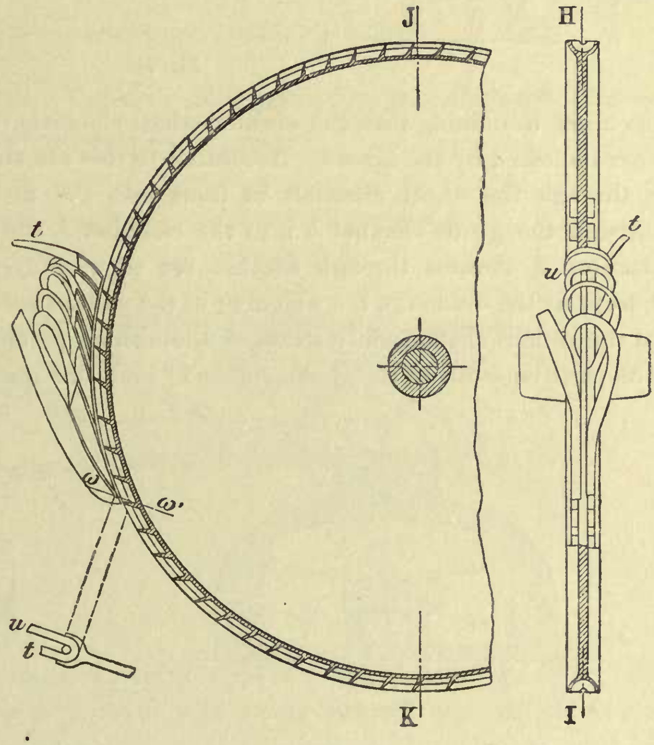


Fig. 90. Fig. 89. Fig. 91.

At a later date *Farcot* and *Perrigault*[1] attempted to improve these arrangements (Figs. 92 to 100). As shown in Figs. 92 to 94 the working wheel *b*, which is provided with obliquely arranged vanes *a*, is enclosed in a case, which, between the steam inlet *c* and the discharge *i* at the sides of the working wheel, bears the chambers *e*, *d*,

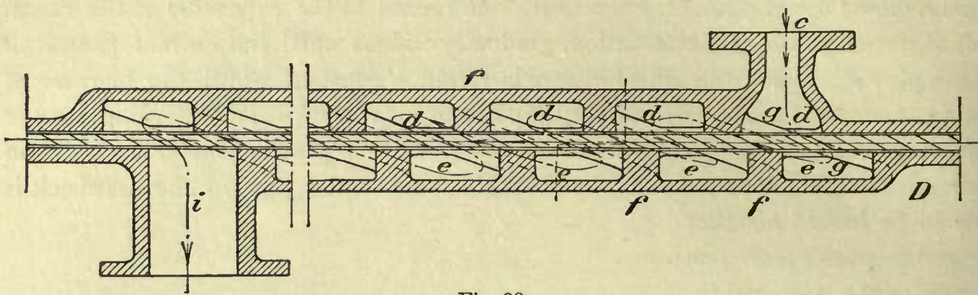


Fig. 92.

the latter combining with the working-wheel channels to produce the steam-passage ways indicated by the arrows. According to this the steam proceeds from the inlet *c*, through the wheel channels in front into the first chamber *e*; then streams through the girdle channel *h* into the chamber *d*, and passing over to the second chamber *e*, streams through another set of working-wheel channels, and so on. Except at the discharge *i*, a widening of the steam-passage way is not contemplated. In the various changes only steam of diminishing pressure is intended to be present. The partition-walls of the chambers are of such size that an exchange of contents of

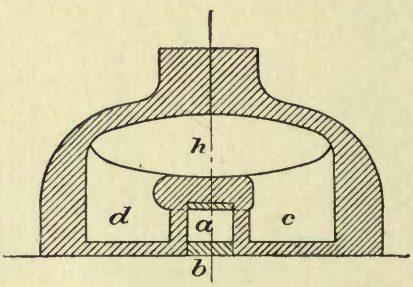


Fig. 93.

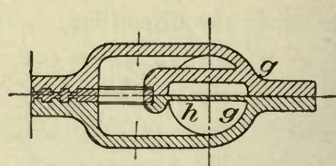


Fig. 94.

two adjacent chambers of one side of the case through the working-wheel channels cannot take place. The working-wheel disc is, moreover, made tight over against the case by means of labyrinth packing.

According to the alteration illustrated in Figs. 95 to 97, two conaxial working-wheel wreaths *a a'* are provided, which are fitted on one and the same disc, but the vanes of which are transposed one with another. The steam here, starting from the first chamber *d*, may, for instance, proceed through the outer wreath of vanes *a* into the chamber *e*, which lies opposite, and is then led through the girdle channel *h* to the lower chamber *e* for the vane wreath *a'*, through which it streams. It then arrives at the other side, and is led through a connecting girdle channel *h* of the second

[1] E. P. 1206 of the year 1866.

chamber d of the outer vane wreath a, whence it begins its second round, and so on. The designers, it may be observed, have pointed to the eventual use of more than

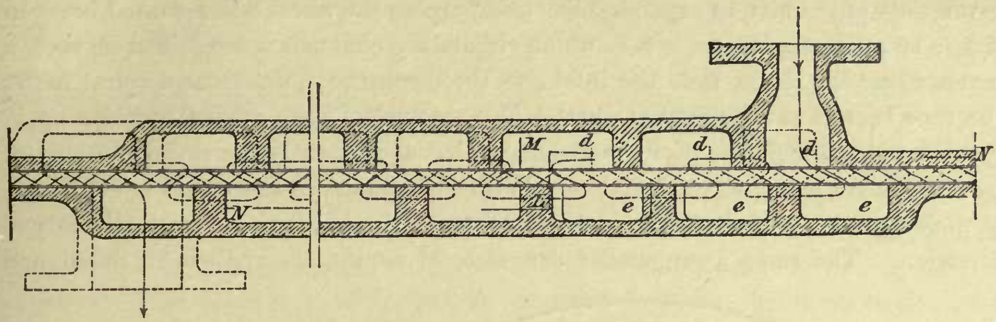


Fig. 95. (Section through GHIJ in Fig. 96.)

two conaxial wreaths of vanes. The style of construction just discussed likewise shows no widening of the steam-passage ways.

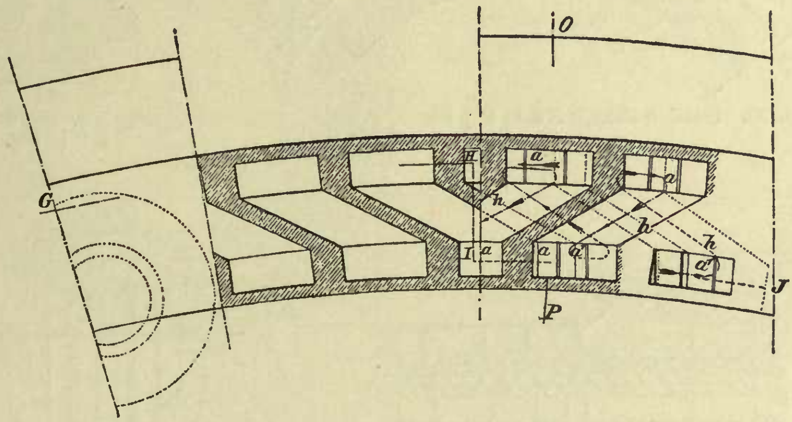


Fig. 96. (Section through NLMN in Fig. 95.)

Finally, mention may be made of a third style of construction (Figs. 98 to 100), in which the working wheel p is provided with steam channels that are oblique in

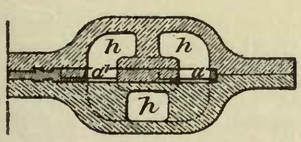


Fig. 97. (Section through OP in Fig. 96.)

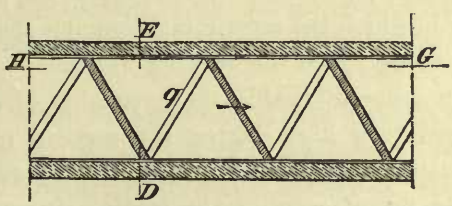


Fig. 98. (Section through ABC in Fig. 99.)

one direction, while the channels of the leading apparatus are oblique in the other direction. The leading apparatus here surrounds the vane wreath of the working wheel. A guidance of the steam worthy of the name is, in this case, not to be thought of.

Scheuber[1] also (Figs. 101 to 102), in order to reduce the velocity of the working wheel *a*, leads the steam repeatedly through the cells of the wheel, whereby it is given the opportunity to expand. For this purpose the wheel *a* is mounted between the fixed guide appliances *b*, *c*, to which circulating channels *d* are joined in such a manner that the steam, from the inlet *e* to the discharge *f*, describes a spiral round the vane bearers of the working wheel. The channels *d* keep gradually widening in the direction of the current of the steam, so that a constantly increasing number of guide cells are left open to the latter. With such guidance the steam is not forced to undergo a reversal, but takes its way through the wheel uninterruptedly in one direction. The same arrangement can also, of course, be applied to radial and

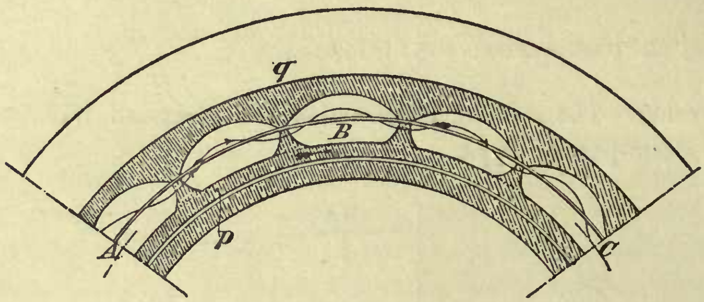


Fig. 99. (Section though HG in Fig. 98.

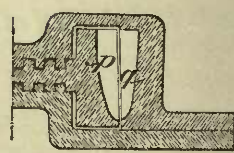


Fig. 100. (Section through ED in Fig. 98.)

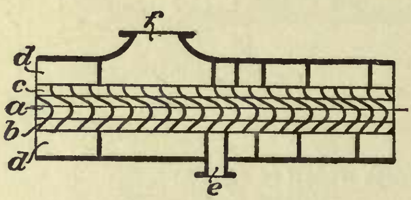


Fig. 101.

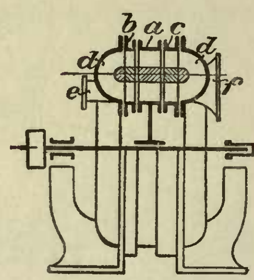


Fig. 102.

compound turbines. Further, it can be applied to wheels having single vane wreaths only, in which the circulating channels then lie close together, so as in each case to form a ring round the working wheel.

In order to enable the expansion of the steam in the close-coil system to be utilized to a high degree, the sections of the leading channels must, of course, gradually widen, and this widening may easily be the cause of difficulties, especially in the manufacture of the leading apparatus. Here the invention of *Zahikjanz*[2] (1902) comes into play, in accordance with which the increase of section of the guide channels is applied to groups of the latter. Fig. 103 shows a steam turbine with U-shaped guide and working-wheel channels, in which the coil ring is made up of four coil systems, *a*, *b*, *c*, and *d*. Each coil system is divided into three groups of

[1] E. P. 20,310 of the year 1902.

[2] D. R. P. 148,704.

coils, the sections of which bear to each other the proportion 1 : $1\frac{1}{2}$: 2. The current of steam enters the first coil system *a* at the point *e*, and in its course through it works successively with single, $1\frac{1}{2}$-fold, and 2-fold expansion. It then leaves the coil system *a* at *f*, and separates into three branches, which stream through the other coil systems running parallel with the first one. In this the whole current of steam works first with (3 × 1)-fold = 3-fold, then with (3 × $1\frac{1}{2}$)-fold = $4\frac{1}{2}$-fold, and finally with (3 × 2)-fold = 6-fold increase of section and of expansion. Fig. 104 is an outline illustration of a turbine with three coil rings, each having four coil systems. The coil systems of the first ring are arranged one above another in the manner described above, and illustrated in Fig. 103. On the other hand, the coil systems of the other two rings may, as desired, be arranged as if they belonged to the same ring, or as if they were separate turbines. Fig. 105 is an outline illustration of the

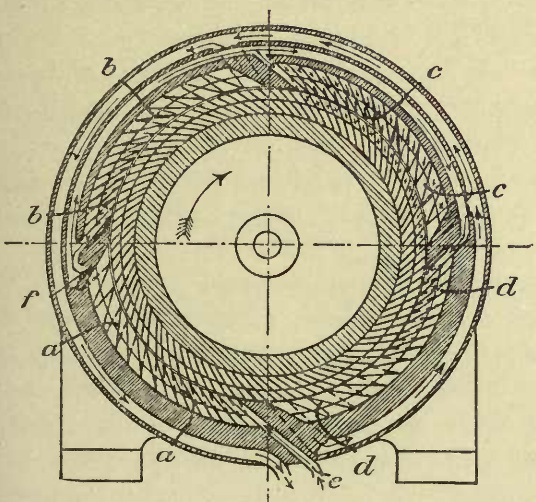


Fig. 103.

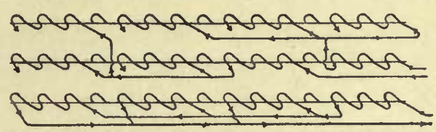


Fig. 104.

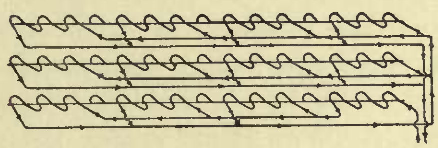


Fig. 105.

arrangement of a steam turbine with three coil rings, each having four coil systems for the attainment of a very considerable increase of section. For example, under the assumption that each coil system consists of three groups of coils, the sections of which bear the proportion 1 : $1\frac{1}{2}$: 2 to one another, a 16-fold increase of section (or expansion) is obtained, which, in the case of the use of very high pressure or of condensation, is found to be necessary.

The repeated coursing of the steam through the channels of one and the same wheel has, indeed, the advantage that for the attainment of the number of stages required for a given speed of rotation a comparatively small number of working wheels only are required. Also it enables the space occupied by the wheel to be very fully utilized. Further, if the steam pressure be reduced by stages to a small degree only, the loss at the clearance space between two neighbouring wheels becomes of relatively small importance. In addition, all the channels remain filled with steam, so that the injurious space, which is represented by an empty channel as it winds into the impingement zone, almost vanishes. There remain, however, the clearance-space overpressure between the highest and the lowest stages, which limits the total gradation attainable in a wheel, and the annular clearance spaces between the working wheels and the leading apparatus, so that, where several wheels are to be made use of, these must work in special chambers made thoroughly tight in relation to each other. Finally, the turns—generally sharp ones—to which the steam is subjected in the leading apparatus, and which form a source of loss of energy, should not be underrated.

II

VELOCITY TURBINES

When, in a water turbine, the clear width of the working wheel is greater than the section of the water jet which impinges on it, the arrangement is known as that of the partial or free-jet turbine. The partial impingement of the working-wheel channel, however, when applied to the steam turbine, brings a new factor into the question, namely, the expansion of the steam, which takes place without a simultaneous imparting of work to the wheel, and only results in the conversion of the pressure of the steam into velocity. To completely utilize this velocity and to take up an equivalent quantity of work in order to transfer it to the shaft, is the object of the velocity turbine.

(*a*) **Velocity Turbines with Acceleration of the Steam within the Working Wheel.**—Following the pressure turbines, the turbines with partial impingement of the working-wheel channels may here first be discussed, and it may be pointed out that the steam entering a channel under pressure undergoes a more or less complete expansion, depending on the width of the channel. If the clear width of the working-wheel channel be sufficiently great to allow the steam issuing from the leading channel to expand to the pressure of the medium which surrounds the working wheel, the latter will not have a remainder of pressure to give off, but only velocity.

Among the velocity turbines with partial impingement of older construction, that of *Schiele*[1] (Figs. 106 and 107) is of interest. In the case *a* (Fig. 106), the two working wheels *c* on the shaft *b* are arranged, the one for right-hand, the other for left-hand, rotation of the shaft. These wheels are so closely shut in by the steam-ring chambers *d* that they can only just run free. They bear double vanes, *e*, which branch from the middle of the wheels on each side in the form of curved channels. The curvatures of the vanes of the one wheel trend, in accordance with the different directions of motion of the wheels, in a direction opposite to that of the vanes of the other wheel. In accordance with this, also, the tangentially arranged steam-egress nozzles *f* of the two ring chambers *d* trend in different directions. The vanes are impinged upon in the middle. The jet of steam is parted, and issues at both sides of each wheel into the case *a*, thence to stream to the jet condenser *g*. By means of the cock *h* the steam can, as required, be led to the one or the other of the ring

[1] E. P. 1693 of the year 1855.

chambers *d*, so that either right-hand or left-hand motion of the shaft *b* is produced. The best utilization of the energy of the steam is now supposed to take place when, in leaving the cells of the wheel, it streams out in an axial direction. In order to

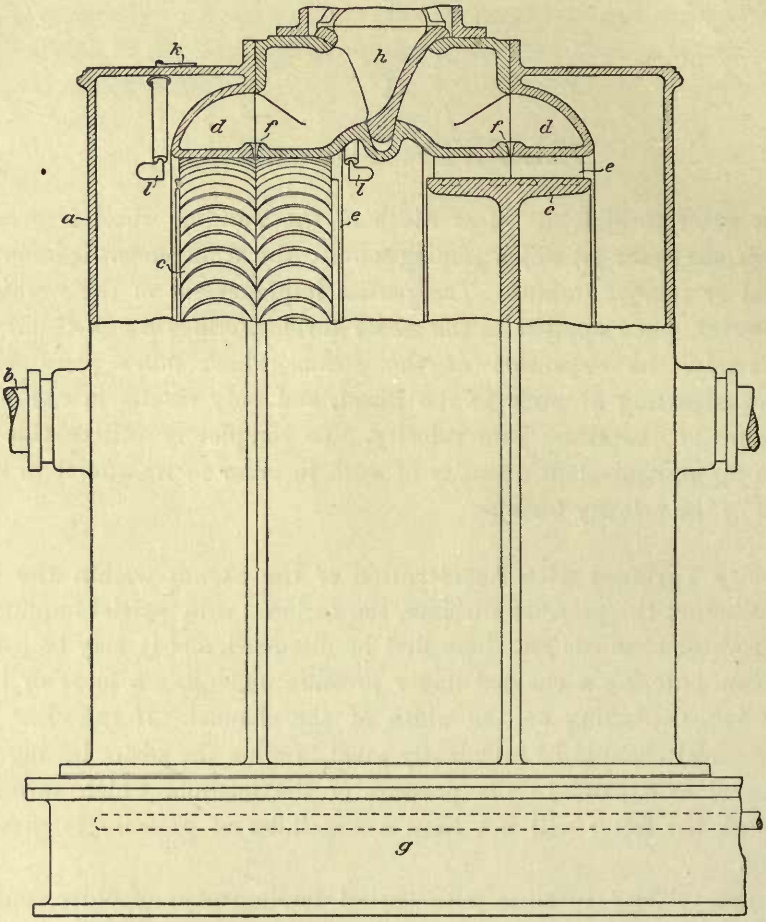


Fig. 106.

effect this, either the quantity of steam to be introduced or the power-development of the turbine is suitably adapted or altered. When the engine is at work, the directions of the steam jets as they leave the vanes are shown by the pointers *k* (one only shown in the figure), the shafts of which are passed through the walls of the case and end in the vanes *l*, which are adjusted by the jets of steam. It may further be pointed out that the steam nozzles *f* (Fig. 107) are of rectangular section and that their partition walls run to a point at the outer side in such a manner that the separate jets join together at the points of discharge into a complete ring. It may likewise be observed that the metal-plate working-wheel vanes *e* are, according to the designer, to be cast in one with the working-wheel wreath.

Fig. 107.

They may either keep their full breadth from the middle to the sides of the wheel, or their outer corners may be rounded off.

Another arrangement has been made by *Harthan*[1] (Fig. 108). In a manner similar to that in a centre-shot water wheel, the working wheel *a* is provided at its periphery with vane cells, *b*, which are open to the outside only, and the bottoms of which are curved in semicircular form in the direction of the axis of the wheel. The steam issuing from the nozzle *c*, which is narrowed and arranged as close as possible to the wheel, impinges upon one after another of the vanes on the one side. It is caused by the semi-cylindrical bottom to take a new direction, and passes off into the discharge channel *d*, which, like the nozzle *c*, is also arranged as close as possible to the periphery of the wheel (but not so as to rub against it), and which is wide enough to receive the steam from several of the cells *b*. It is here the intention that a direct pressure action of the fresh steam shall be combined with a reaction of the jets of steam issuing from the vanes. In the case of axial impingement the cells *c* are opened at the sides, but they are then closed at the periphery of the wheel. Harthan recommends that the velocity of the vanes be not allowed to exceed half that of the steam. He assumes the former at from $\frac{1}{3}$ to $\frac{4}{10}$ of the latter.

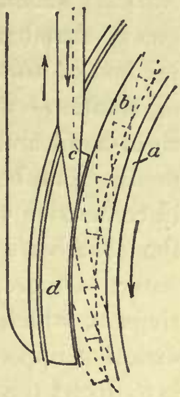


Fig. 108.

An intermittent steam supply allows of the expansion of the steam within the working-wheel cells after the closing of the steam inlet. Thus *Wrench*[2] (Figs. 109 and 110), by means of the leading apparatus *a*, divides the steam into separate jets. These issue from the nozzles *b*, which converge towards the working wheel. The

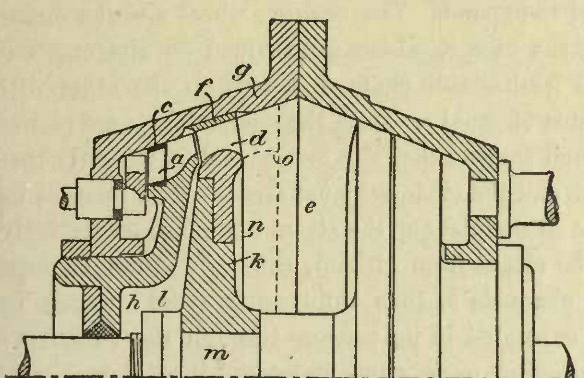


Fig. 109.

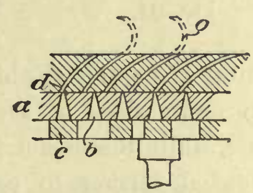


Fig. 110.

nozzles are covered by the slide *c*, which is adjusted by hand or by a regulator. The working-wheel channels *d* are likewise on the steam-inlet side at a considerable distance from it. They widen out towards the chamber *e*, which is in connection either with the exhaust into the air, or with the condenser. At the periphery the working-wheel channels are covered by a ring *f*. Further, the leading apparatus and the working wheel are, in accordance with the shape of the case *g*, widened out in conical form in the absolute direction of the current of steam. Between the fixed nozzle-bearer *h* and the working wheel *k*, a neutral chamber *l* is

[1] E. P. 144 of the year 1858. [2] E. P. 6248 of the year 1894.

formed, and it is asserted that the jets of steam passing from the nozzles *b* into the working wheel exercise a suction on the chamber *l*. The intention is that the exhaust steam in the chamber *e* be led through the chamber *l* (when *e* and *l* are in connection with one another), in order to join again in performing work in the working wheel. There are two turbines with separate steam inlets and an exhaust-steam chamber in common arranged on the shaft *m*. The assumption as to the method of working is that either both turbines influence the shaft in one direction of rotation—there is then no possibility of reversing—or that one wheel is for right-hand and the other for left-hand motion. In the latter case the part that is for the time being disconnected runs free as a flywheel, its chambers being then in connection with the exhaust or with the condenser. Wrench thought of arranging the chamber *e* at the same time as a jet condenser, the shaft then throwing off water. Of greater value is the proposal to insert behind the working wheels the rings *n*, which are provided with vanes *o*, the curvature of which trends in a direction opposed to that of the working-wheel vanes. The object of these vanes is to divert the steam issuing from the working wheel into the direction of rotation of the wheel.

In spite of the fact that *Henderson*[1] (1897) (Fig. 111) credits his turbine with a combined manner of working, his design may suitably take its place here. Over the periphery of the working wheel *a* are distributed the U-shaped channels *b*, which widen out in the direction of the current of the steam, so that opportunity is given to the latter to expand. The working wheel *a* works steam-tight in the case *c*, which is provided on the one side with the fresh-steam channel *d*, and on the other with the exhaust channel *e*. Both the channels *d*, *e* are carried right round the wheel. The first-named is narrower than the other, and the clear sectional areas of both increase towards the steam inlet and the steam discharge respectively. A number of nozzles *f*, which can be closed from without, allow the steam to enter the narrower sides of the wheel channels *b*, thus influencing these directly by pressure, while the steam that has expanded in its passage through the channels *b* is intended further to perform reaction work when it passes into the exhaust chamber *e*. In this latter, moreover, fixed guide vanes are introduced, which evidently prevent the steam issuing from the working wheel from rotating further in the exhaust chamber. In this case, then, the steam is introduced and also discharged at the periphery. Since the working wheel and the leading ring are made tight, one against the other, frictional work is performed over the longest possible surface.

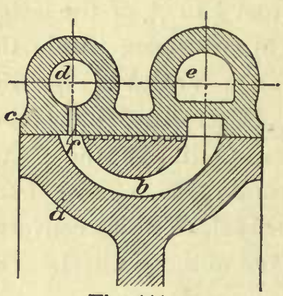


Fig. 111.

Lennox[2] also has aimed at passing the steam through the turbine by stages (Fig. 112). The working wheel consists of two discs *a*, between which the tubes *b* are arranged in a circle round the axis of the wheel, at an acute angle with it, the

[1] A. P. 634,170.

[2] E. P. 8850 of the year 1901.

discs lying close against the fixed walls *c*, *d*. These latter have oblong openings *e*, *f*, the holes *e* not being opposite to the holes *f*, but so removed from them in the direction of motion of the wheel that the tubes *b* pass quite clear of the steam-inlet holes *e* before they begin to communicate with the steam-discharge holes *f*, while, on the other hand, they continue in communication with these when they have moved clear of the holes *e*. For each tube there are thus following in succession a pressure-action of the inflowing steam while the tube is being filled, a free flow of current through the latter, and finally an expansion, towards the discharge side, of the steam remaining in the tube, *i.e.* a reaction. By the exchange of the functions of the steam inlet and of the steam outlet the reversal of the direction of rotation of the working wheel is intended to be effected. Meanwhile the circumstances are here such that it must be doubted whether it will be possible to make the wheel rotate at all.

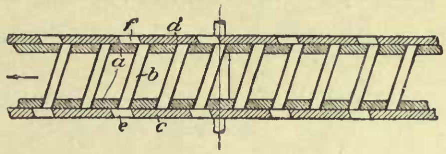


Fig. 112.

The multi-stage turbines with partial impingement offer greater opportunity than the single-stage ones for variations, but they also produce methods of construction, the modes of working of which are, in some cases, not at all clear.

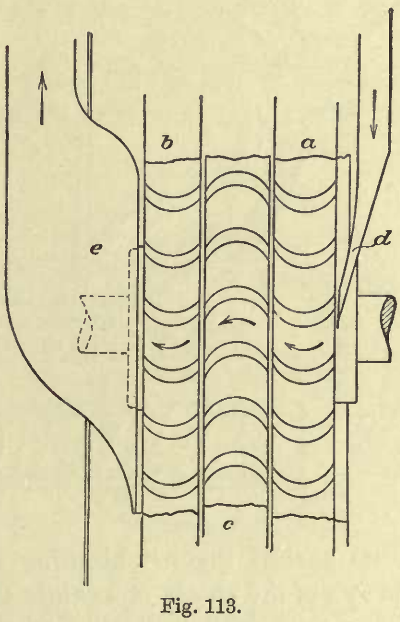


Fig. 113.

Harthan[1] (Fig. 113) places on one shaft, without, indeed, giving any special reason for so doing, two working wheels *a* and *b* with their vanes curving in the same direction and arranged for axial impingement. Between these wheels, however, there is arranged a group *e* of fixed guide vanes, which reverse the steam as it leaves the wheel *a*, and lead it to the wheel *b*. The fresh steam streams through the nozzle *d* to the wheel, while the exhaust steam passes into the pipe *e*.

Imray[2] (Figs. 114 to 116) brings the velocity grades together into a single wheel, and leads the steam again and again into the cells of the same working wheel with the object of reducing the latter's speed. Into the working wheel A is worked a ring groove, which corresponds with the ring groove in the cylinder wall D. The first-named ring groove is divided into cells by the plates *a*, the second by the plates *d*. The plates *a* and *d* are in each case placed at an inclination to the corresponding radii, and complement each other in pairs, so as to form circular surfaces, while the grooves in wheel A and in the cylinder wall D are of elliptical section. From the nozzle *e*, which is arranged at the side of and tangential to the

[1] E. P. 144 of the year 1858. [2] E. P. 177 of the year 1881.

above-mentioned ellipse, the steam streams out under pressure, and, after expanding, is reversed by the cell of the working wheel. It then enters the next-following cell, and is led by the latter into the working wheel again, and so on, until it is exhausted through the channel G. In order that a one-sided axial pressure may be avoided, two steam inlets and two outlets are arranged symmetrically to the axis of the wheel. Further, the working wheel A is provided with an odd number of vanes *a*, so that when the one nozzle *e* is just beginning with the impingement of a working-wheel cell, the other nozzle finds itself in the middle of a cell. Every nozzle mouth-piece L closes the working-wheel cell that is being played upon by the fresh

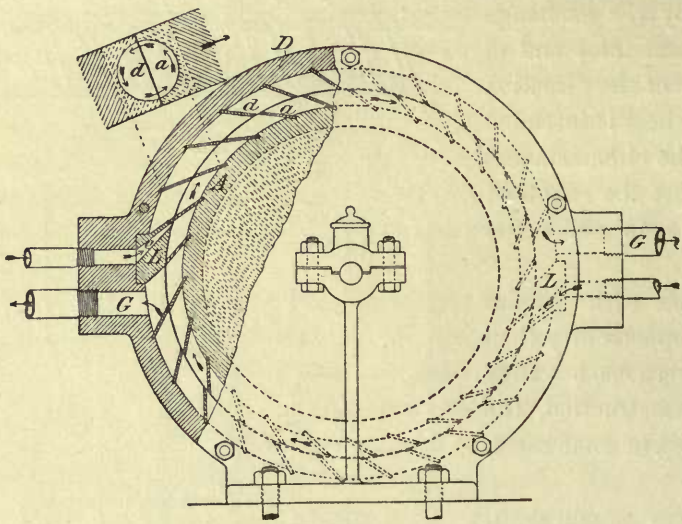


Fig. 114.

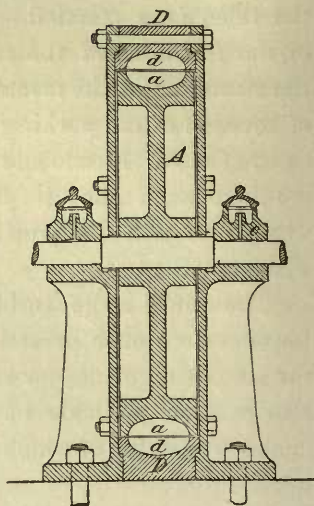


Fig. 115.

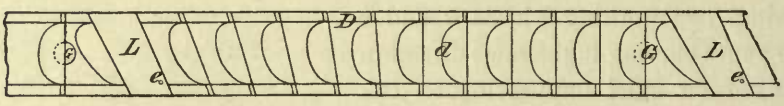


Fig. 116.

steam against the neighbouring exhaust G. The vanes *a* and *d* lie parallel with the axis of the wheel. Meanwhile, the guide vanes *d*, that lie nearest to the nozzles *e*, are set at a small inclination (Fig. 116), so that the steam issuing from one of the latter, after playing upon the cell of the working wheel, is guided by the next leading cell against a more advanced part of the working wheel. The sharpened edges of the vanes *a* and *d* are intended to assist in the reduction of the resistances.

Imray, be it observed, makes allusion to several alternative arrangements, in one of which the course of the steam lies, not at the periphery of the working wheel, but in a ring at the side of a disc.

Dow[1] (1886) (Fig. 117) has a varying arrangement, in which action is followed by reaction, and this by compression in the leading apparatus. The fixed leading apparatus *a* are provided with channels *b*, which become narrowed in the direction of the flow of the steam through them; the cells *c* of the working-wheel wreath *d*

[1] A. P. 360,766.

become wider. Further, the guide vanes and working-wheel vanes are so placed and formed that the latter can be impinged upon through the former at short regular intervals, and that the steam that has found its way into the working-wheel cells can expand in these, perform work by reaction, and pass free into the next-following leading channel. The first work done consists, then, in pressure exercised by the jet of steam from the first leading cell *b* streaming under pressure against the working-wheel vanes of the first wreath *d*, which lie directly opposite this cell. There then follows an expansion of the steam in the working-wheel cells *c*, at first accompanied by a further supply of fresh steam from the still open leading channel *b*, and afterwards, with this supply shut off, into the leading channels b_1 of the outer leading apparatus a_1. At the discharge side of the latter the steam again undergoes a certain compression, and so on.

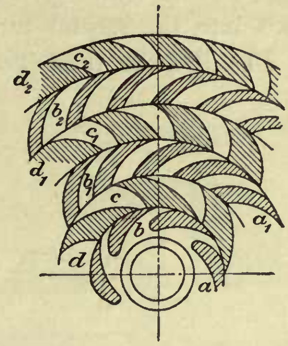


Fig. 117.

Of the mode of construction adopted by *Heinrichs*[1] (1897), Fig. 118 shows at its upper half a section at I, I, and at its lower half a section at II, II, in Fig. 119, in each case. On a hollow shaft A, which is immovably fixed in its bearings, there are placed, in the first design, three turbines differing from one another only in size, and arranged one behind another in such a manner that the leading wheel C of each of them is fixed to the shaft, while the working wheel D, which takes the form of a case enclosing the guide wheel, and is provided with a belt rim, E, rotates upon the shaft A. The shaft A, which for a single turbine serves the purpose at once of steam-supply and steam-discharge pipe, is made up of several parts, each of which is closed in the middle of a guide wheel against the next one, and each part of which is larger in diameter than the preceding one in due relation to the varying pressure of the steam. At the edge of the guide wheel C may be observed two sets of steam-passage channels. The one set of channels G pass the steam into the working wheel, and are formed like the leading channels of a Girard turbine, and the other set H are for the discharge. In conformity with these purposes the former are connected by pipes with the steam-supply portions of the shaft A, and the latter with the steam-discharge portions. The working wheel D, which, to the avoidance of detrimental spaces, lies as closely as possible against the sides of the guide wheel, has, running right round the periphery of the latter, a channel J in which the working-wheel vanes K are inserted. The steam proceeds at high

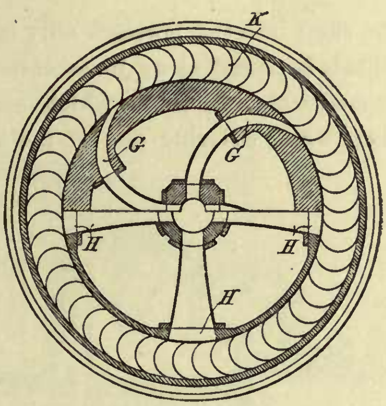


Fig. 118.

[1] D. R. P. 100,939.

pressure from the shaft A, through the nozzles G, into the cells of the first working wheel, which, while passing the discharge channel H, deliver their steam into the same, so that steam of reduced pressure is now led through the next portion of the shaft into the second turbine, and so on. According to the arrangement shown in Fig. 120, the three turbines are enclosed in a casing, D, so that a division of

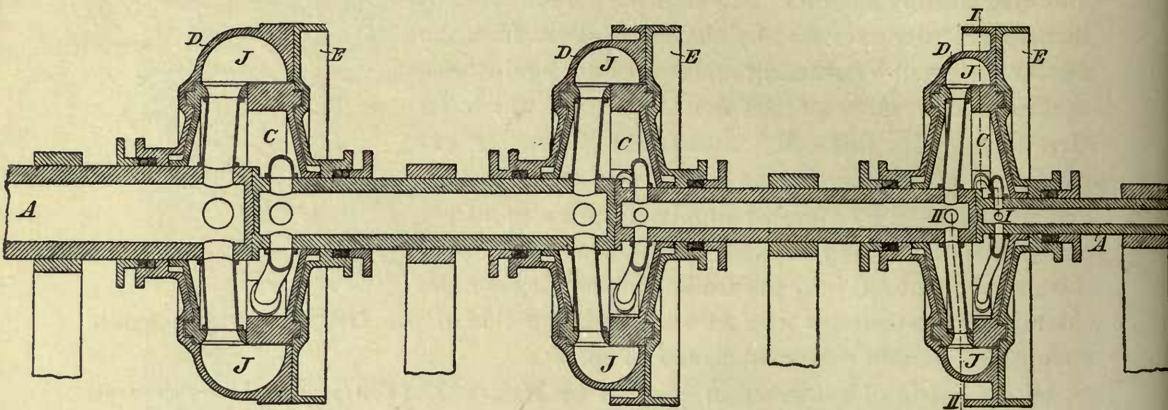


Fig. 119.

the shaft into two portions only is necessary. The steam escaping from the working wheels is passed along by sector-shaped leading apparatus into the ring channels H^1, immediately adjoining which are the impingement nozzles G^1. The steam will then, when entering one of the working-wheel cells, first perform work by action, after which it will expand to the volume of the cell, and finally, while streaming into the discharge channel, will again perform work by reaction. Meanwhile its pressure decreases from stage to stage.

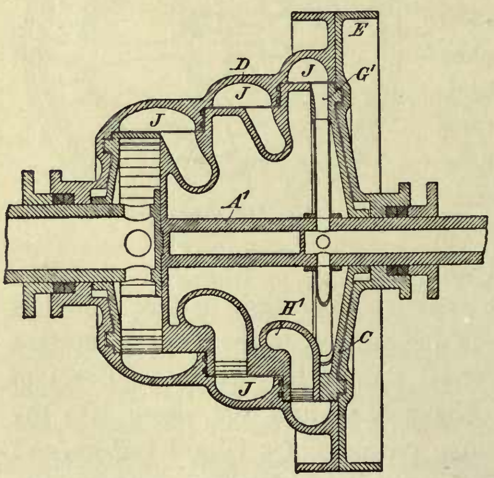


Fig. 120.

Pyle[1] (1894) arranges a wheel, composed of several wreaths of vanes concentric with one another, forming a multi-stage axial turbine. The side walls of the case, which closely adjoin the working wheel, are provided with channels, which lead by turns from one stage to the next. The steam, which streams out of several nozzles arranged in a circle, proceeds in the first place into the outermost vane wreath, and is then led by the channel before mentioned to successive performances of work in the stages trending radially inwards, till it escapes at a point in close proximity to the shaft.

The turbine designed by *Ferretti*[2] (Figs. 121 and 122) works in a similar manner.

[1] A. P. 549,277. [2] E. P. 19,839 of the year 1899.

The steam introduced into it streams successively through the working wheels *c, d, e, f*, etc., between which the fixed leading apparatus *g, h, k*, etc., are inserted. Each of the latter is of such capacity that the steam entering it expands to about half the pressure with which it entered the preceding working wheel. The guide wheels are of about twice the breadth of the working wheels, which are kept as narrow as possible. They have wide steam inlet openings (Fig. 122), but narrow outlets to the next-following wheel, which arrangement causes an acceleration of the steam in the direction of the current. As the diameters of the working wheels and leading apparatus increase, the vanes decrease in number.

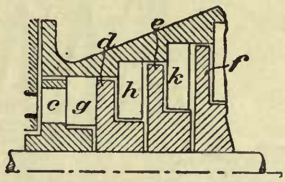


Fig. 121.

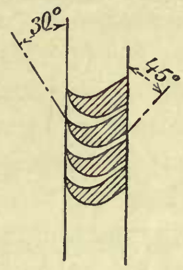


Fig. 122.

According to *Hoehl, Brakell*, and *Günther*,[1] the speed of rotation of radial turbines is to be reduced by an arrangement by which the steam is forced to impinge upon two sets of wheels one after another. According to the design illustrated in Fig. 123, the steam proceeds through the branches *a* into the annular chamber *b*, from

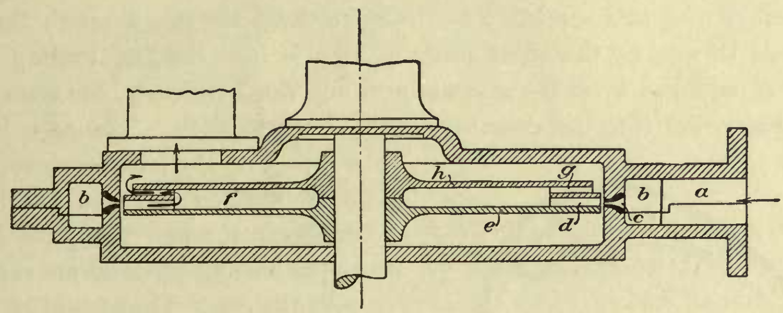


Fig. 123.

which tangentially arranged guide vanes lead it into the suitably curved vanes *d* of the lower working wheel *e*. Behind the vanes *d* the steam collects in the chamber *f*, and it streams thence through the vanes *g* of the second wheel *h*, outwards into the air, or into the condenser. It would, however, be feasible to arrange a larger number of wheels one behind another. In the chamber there is a pressure, which is intermediate between those of the annular channel *b* and the exhaust chamber. This mean pressure imparts a proportionate velocity to the steam streaming onward into the vanes *g*, but at the same time retards the flow of the current through the first working wheel *e*, and a lower speed of the turbine is attained than would be possible with only one working wheel.

Averseng[2] (1878) places on the shaft J several radial wheels A^1, B^1 . . . G^1 (Figs. 124 and 125) arranged for impingement from without. Fig. 124 shows a

[1] E. P. 2429 of the year 1863. [2] D. R. P. 2607.

section through *ab* of Fig. 125, while the latter shows a section through V*e* of Fig. 124. Each of the wheels is enclosed in one of the separate chambers A, B, C, D, E, F, and G. The chambers are enclosed in a steam-tight mantle RS. The working-wheel vanes are so curved as to stand radially at the outer ends, while their inner

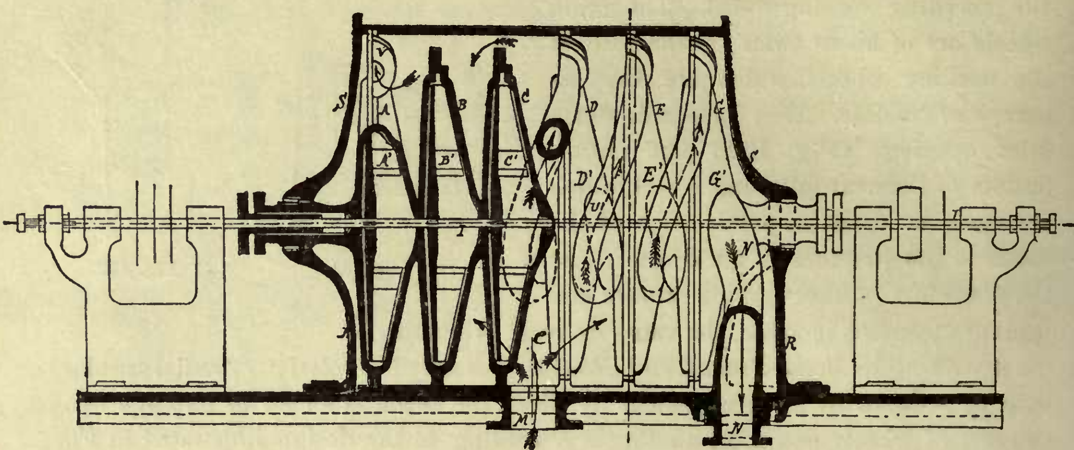


Fig. 124.

ends are tangential to a circle. The steam, entering the case through the branch M, first finds its way by the short piece of pipe V into the first leading channel. From this it impinges upon the separate working-wheel channels, one after another, and then passes out into the chamber surrounding the shaft. Thence it is led by

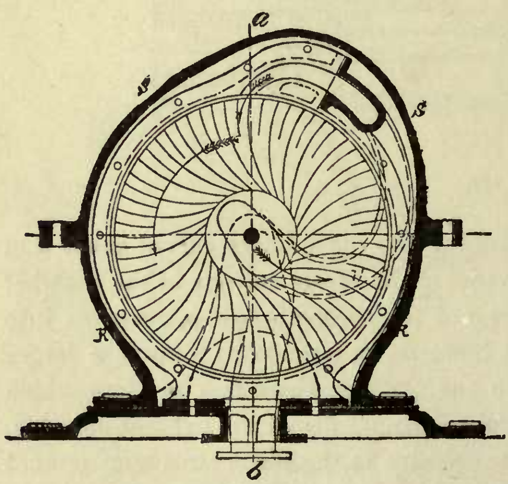


Fig. 125.

a pipe U to the leading channel of the next wheel, and so on till it finally exhausts through the pipe N. The upper leading pipes U are cast in one with the case. They could be replaced by copper pipes. The first leading channel is narrower than a working-wheel channel, so that an expansion of the inflowing steam must take place in the working-wheel channel. For the succeeding stages the conditions are similar. This is plainly a case of pressure stages combined with a conversion of a part of the steam pressure into velocity within each of the working wheels. The direction of the steam is opposed to that of the centrifugal forces exerted by means of the wheels. The wheel channels widen out in an axial direction, so that their sectional areas decrease from the inlet towards the discharge end. The power of the engine can be regulated by a widening or narrowing of the leading channels. In order that a constant pressure in all the wheels may be attained, the diameters of these are increased from stage to stage, by which arrangement also the increase of volume of the steam receives due consideration.

The mode of construction adopted by *Masters*[1] does not seem very advantageous (Figs. 126 and 127). His axial turbine has several working wheels *a* placed one behind another, between which lie the leading apparatus *b*. From the side of the lid *c* the steam is introduced through the nozzles *d*. The intervening leading apparatus *b* are likewise provided with nozzles *d*, the forms and positions of which may be seen in Fig. 127. Each nozzle converges in the direction of flow of the steam. At the inlet and outlet sides pocket-like holes are bored into the casting in such a manner that the steam issuing from the wheel channels lying opposite the nozzle in each case is made to strike perpendicularly against the surface *e*. The surface *f* is then intended to guide the steam issuing from the nozzle in the direc-

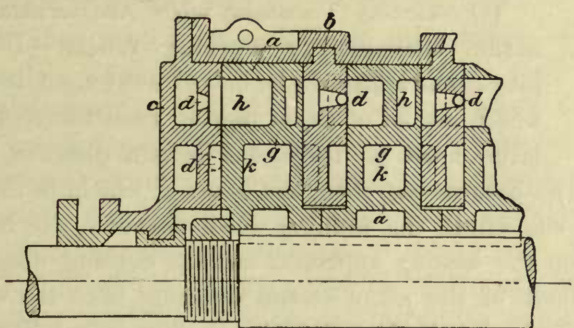


Fig. 126.

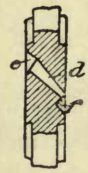


Fig. 127.

tion of the vanes of the next-following wheel. This arrangement is intended to increase the motive power of the wheel by means of reaction. Its chances of success, however, appear somewhat questionable. Further, the working wheels *a* are each provided with two concentric chambers *h* and *k* closed by ring walls *g*, and the vanes of the one annular chamber are in every case arranged in a direction opposed to that of the vanes of the other chamber. Similarly, the nozzles of each leading apparatus, which are divided into two annular groups, are arranged to point in opposite directions, so that the outer wreath of the turbine can be used for the one direction of rotation, and the inner wreath for the other. The possibility, however, of arranging both turbines for rotation in the same direction has also received attention.

In a still greater degree than the foregoing has *The Butler Turbine Engine Company*[2] (1901) failed to grasp the fundamental principles of a good engine (Fig. 128). Round the sleeve *b* fixed on the shaft *a* are laid the screw-like surfaces *c*, and these again are surrounded by the cylinder *d*. The sleeve *b* and the cylinder rotate together within the case. The screw surfaces *c* are crossed by the screw surfaces *e*, and these latter are provided with passage holes *f* for the steam, which has to wriggle through the surfaces *c*. At both ends of the cylinder *d* are arranged the

[1] E. P. 24,201 of the year 1901. [2] A. P. 712,119. D. R. P. 153,372.

steam chambers *g* (only one is shown in the illustration) into which the inlet tubes *h* deliver, while the steam discharges *k* lead direct from the working chamber. The inlets and outlets work together crosswise in pairs, in such a manner that the steam is obliged to stream axially through the whole cylinder *d*. The openings *m* arranged on the lids *l* are set obliquely, so that the jets of steam are directed against the surfaces *e*. The steam passages *f* of the latter are also set obliquely, so that the steam, through them, receives its direction against the surfaces *c*. The cylinder *d*, which takes part in the rotation, closes the working chambers against the case.

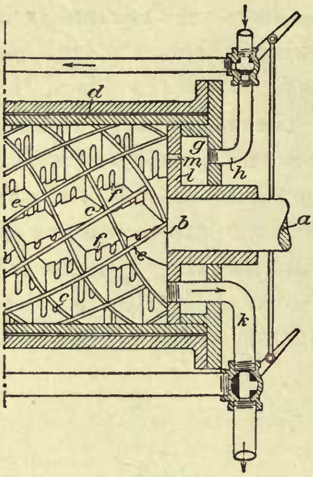


Fig. 128.

(*b*) **Velocity Turbines with Acceleration of the Steam without the Working Wheel.**—If the impingement apparatus be moved away from the working wheel, the jet of steam passing from the former to the latter must go through a certain distance, and is in so doing free to expand. The occurrences, then, the scene of which, in the velocity turbines first referred to, was within the working-wheel channels, are here transferred to the place between the leading apparatus and the working wheel. Since the whole increase of volume of the jet of steam, resulting from the expansion, takes place in front of the wheel, it is clear that the working-wheel channels will have to be just large enough to accommodate the thus expanded volume of steam, without the turbine being made to lose its character as a velocity turbine. The working steam jet may fill the channel without producing over-pressure; it can, however, impinge upon the channel only partially.

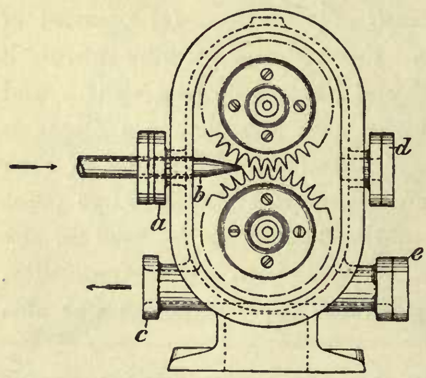


Fig. 129.

From the long series of impulse wheels, that of *Cordes* and *Locke*[1] may here be taken as an example of an engine formed in the manner of a centre-shot water turbine.

In Fig. 129 a design by *Schmid*[2] is illustrated. The steam acts on the point of contact of the arcs of two turbine wheels, the vanes of which work into one another, and their parallel shafts are coupled with each other. The steam nozzle *b*, fixed on the branch pipe *a*, corresponds with the steam discharge *c*. The branches *d* and *e* effect the reversal of the direction of rotation of the wheels.

Amongst other impulse wheels, that of *Hopkins*[3] (Figs. 130 and 131) may here

[1] E. P. 8572 of the year 1840. [2] D. R. P. 123,049.
[3] E. P. 11,880 of the year 1894.

be mentioned. The turbine, the outside of which is made very smooth, so as to offer as little resistance as possible, consists of the two parts *a* and *b*, in which radially arranged vanes *c* and *d* are fixed. Between these vanes a space is left free, so that there is the slightest possible play between them and the fixed steam nozzle *e*, which is to a certain extent enclosed in the turbine wheel. The parts *a* and *b* leave, at the periphery, only the slot *f* free for the insertion of the steam pipe *g*. A ring piece *h*, connected with the latter, closes the clearance space to the outside over about one-quarter of the circumference. The nozzle *e*, when arranged for right and left motion, has the arrangement shown in Fig. 131. The steam streams through one of the annular clearance spaces *k* which happens to be presented for its reception, and at the same time draws air through the inner chamber of the nozzle between the cone *l* and the wall *m*, whereby first the pressure on the back of the nozzle is reduced, and then a diverging current of steam is produced.

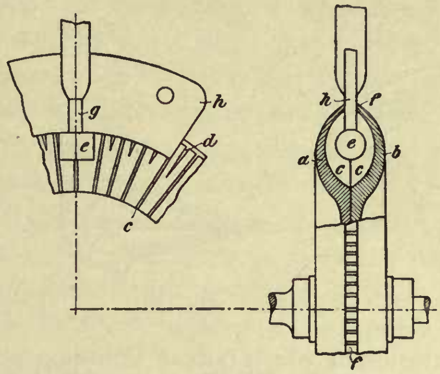


Fig. 130.

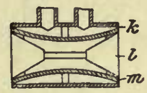


Fig. 131.

Prall's turbine[1] (1898) (Figs. 132 to 134) works in a similar manner. It is provided with two discs *a*, *b* having tooth-like, radially arranged projections *c*, which turn in a similarly shaped case. In the annular channel *e*, between the wall of the case and the revolving discs, is a chamber *f* serving for the leading of the steam in the one or the other direction, and so arranged that, while filling the channel, it does not hinder the motion of the rotating disc. The teeth glide past the chamber *f*. The steam, in the first place, acts directly against the side of a tooth, which may be straight or curved. As soon as this side has passed the chamber, the steam passes freely into the space between the discs. It moves forward then, expanding the while, in a zigzag course as far as the discharge pipe *g*, and takes the discs *a* and *b* along with it. For the opposite direction of rotation, the steam-injection pipe *h* and the exhaust pipe *k* come into action.

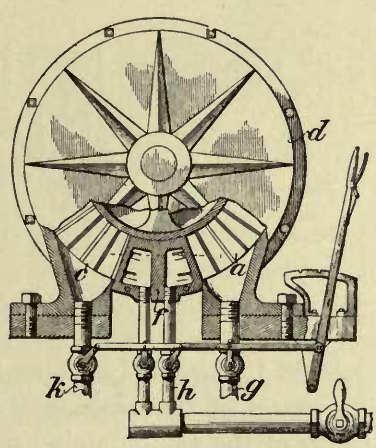


Fig. 132.

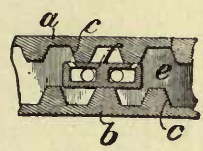


Fig. 133.

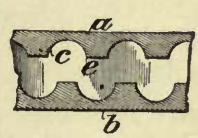


Fig. 134.

[1] D. R. P. 115,941.

Of this, the wheel designed by *Krissmanek*[1] (Vienna, 1900) (Figs. 135 and 136) forms, in a sense, the inversion. In this case, namely, the working wheel is a thin disc, *a*, which, on the side impinged upon by the steam jets, is roughened like a file, or checkered, or provided with holes *b*; arrow-like vanes may also be fitted. In the axis of symmetry of this disc *a* the jets directed against the wheel from both sides meet at the point *x*, and then join together to form a single jet, the direction of pressure of which falls in the same plane with the direction of rotation of the wheel. The double-impingement nozzle *l* is provided with intermediate webs *q* with holes *r* bored conically outwards, the sharp edges of which are intended to produce close jets.

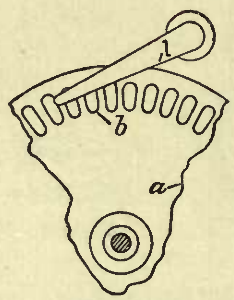


Fig. 135.

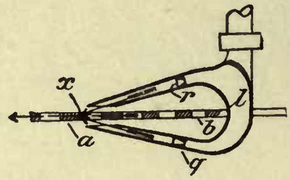


Fig. 136.

Fedeler also allows the motive medium to enter the working-wheel channels from both ends, so that the steam has to crowd together in these,[2] and then make an effort to stream forth in a tangential direction.

An action wheel, in which the steam is discharged from the side, has been designed by *Wise*[3] (1884), who has the following ideas (Figs. 137 and 138). The working wheel *a* is provided with cells *b* from the floors of which pipe branches *c* fixed at the sides of the wheel-wreath deliver in an outward direction. These are bent backwards at about 30° with the direction of rotation of the wheel. Over a small portion of the periphery of the wheel the cells *b* are covered on their outer sides. The nozzles of the steam pipes *d* are so placed that the jets of steam only strike the back surfaces of the vanes, but are not able to exert a braking pressure on the front sides of the next-following vanes. It is, indeed, assumed that the steam jets create a vacuum in front of the next-following vanes. The steam issuing from the pipe branches *c* is intended to strike against the plates *e* fixed on the wall of the case, and from this Wise expects to obtain an access of motive power.

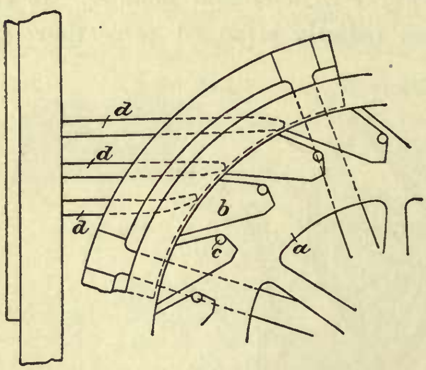


Fig. 137.

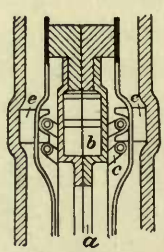


Fig. 138.

Hutchinson[4] makes use of an action wheel impinged upon radially from within (Fig. 139). The wheel *a*, fixed upon the axis, is provided with hollow vanes *b*, for

[1] Ö. P. 8500.
[2] See Chap. VII.
[3] A. P. 303,781.
[4] E. P. 7807 of the year 1893.

which space is left between the channels *c*. These channels also discharge, as far as possible, in a tangential direction. The wheel *d*, on the other hand, is free to turn on the axis, which is at rest. It has surfaces *e*, against which the steam streaming out of the channels *c* is made to strike, so that it is caused to rotate. The outer toothed periphery *f* is acted upon by the pinion wheel *g*, which turns the shaft of the engine. A further pinion is provided in case a reversal of the motion be required. The steam is led into the inside of the wheel *a*, and it is discharged in an axial direction, between the wheels *a* and *d*, into the case *k*.

It is of no advantage in the foregoing case to close the working-wheel cells at their outer sides, so that the steam is carried along, over as long a path as possible, to the exhaust, as is contemplated, for instance, by *Fidler*,[1] who for this purpose employs an annular spring pressed by screws against the periphery of the wheel. Arrangements of this kind are more likely to prevent the motion of the wheel altogether.

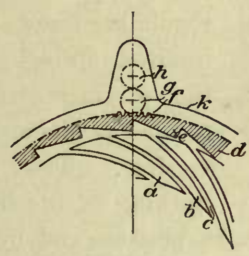


Fig. 139.

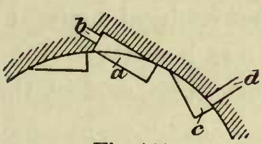


Fig. 141.

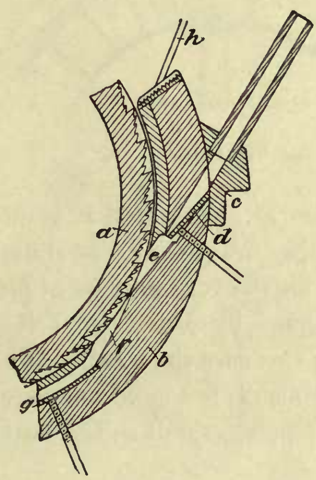


Fig. 140.

Not much more advantageous seems the arrangement made by *Boyd*[2] (Fig. 140). The working wheel *a* is provided with vanes, the front surfaces of which lean forward like the teeth of a saw, while the back surfaces are stepped so that they may offer resisting surfaces to the steam. The cells which are bounded by the vanes are closed at the sides by discs provided at their peripheries with annular grooves. Into these grooves springs belonging to the fixed ring-piece *b* are inserted, so that the latter is pressed steam-tight against the wheel *a*. The square-sectioned steam nozzle *c* can be made narrower or wider by means of the tongue-like spring *d*. The steam is led, in the first place, from the channel *e* to the cells of the working wheel. It can then escape through the channel *f*, the width of the outlet of which can, as before, be varied by means of the tongue *g*. The lubricating medium is introduced through the pipe *h*.

The steam will not succeed in producing a turning motion if it be forced, as

[1] E. P. 121 of the year 1902. [2] E. P. 28,812 of the year 1897.

mentioned above, to exert pressure upon the cells of the working wheel. This is the case, for instance, in the wheel designed by *Addington*[1] (Fig. 141). The intention here is that the working-wheel cell *a* shall stand full in front of the steam inlet *b* when the foregoing cell *c* has just left free the discharge *d*.

In like manner, the style of construction adopted by *Walther*[2] (Figs. 142 and 143) cannot claim to represent an efficient wheel. The box-like cells *f* bear a wreath, made in one piece, against which presses a distributing sliding piece C introduced into the case, which is built round and free of it. In way of the cells, the sliding piece has an arch-like groove *i*, and is pressed by means of regulating screws R and thin inserted springs K against the gliding surfaces. The turbine is claimed to be reversible. The manner of working of the steam is here not quite clear. Apparently, it is intended to work by pressure and by expansion, while the cells *f* are passing the sliding piece C. In this, however, the distributing channel *i* is for the very purpose of preventing a one-sided action of the steam. The turbine designed by *McGregor*[3] (New York, 1901) works with a similar pressure piece, but in this case the rotation of the wheel is not rendered doubtful by it.

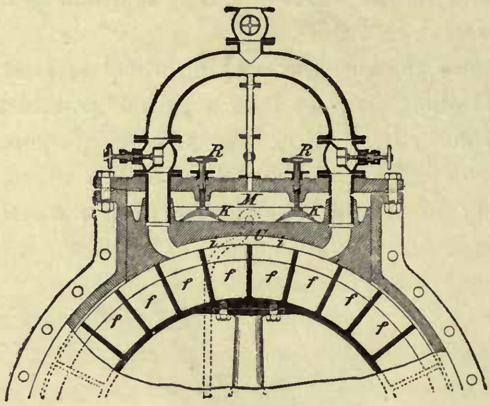


Fig. 142.

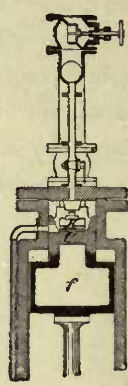


Fig. 143.

Ingham[4] also is not more successful (Fig. 144). It is claimed that the turbine utilizes the kinetic as well as the static energy of the steam, as does the backwards

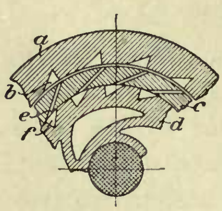


Fig. 144.

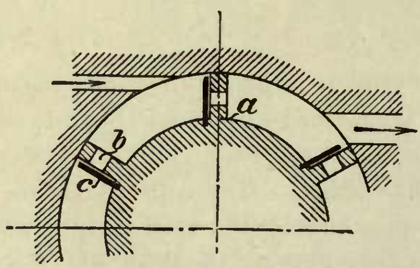


Fig. 145.

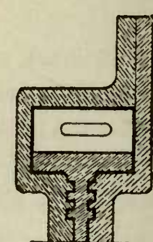


Fig. 146.

and forwards moving piston of the steam-engine. To effect this, chambers are constructed, which, on the one hand, offer a firm end wall to the expanding steam, and on the other are closed by a yielding wall. In Fig. 144 *a* is the fixed case, having indentations *b* with radially arranged sides, and *c* a ring closely encircling the working wheel *d*, and provided with tangentially arranged channels *e*. The

[1] E. P. 4714 of the year 1898.
[2] D. R. P. 100,336.
[3] A. P. 712,709.
[4] E. P. 27,733 of the year 1902.

cells of the wheel *d* likewise have radially arranged sides *f*. The steam streams in between the case *a* and the ring *c*. It is now assumed that each of the indentations *b*, together with one of the working-wheel cells and one of the channels *e*, forms an expansion chamber, in which the steam finds a firm surface of resistance against the radial sides of the indentations *b*, and that, pressing against the surface *f* of the working wheel, it causes the latter to rotate. The steam escapes from the working-wheel cells, through transverse holes bored through the ring *c*, either into the air or into the next stage.

A decidedly more correct appreciation of the problem underlies the ideas of *Willey*[1] (1893) (Fig 145) when he provides the vanes *a* with holes *b*, and closes these with spring plates *c*. These latter lie on the hinder sides of the vanes, so that the steam jet holds them closed. They open when the steam in the cell, which happens to be undergoing impingement, exercises a back pressure on the fore sides of the next-following vanes. The steam can then escape in the backward direction. In order to avoid a detrimental back pressure, *Farcot* and *Perigault*[2] have, indeed, proposed that holes be made in the vanes (Fig. 146).

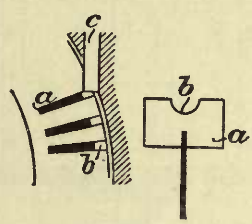


Fig. 147.

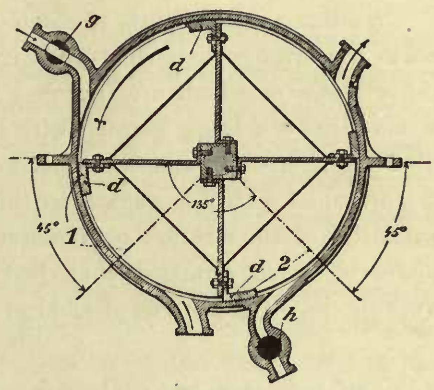


Fig. 148.

It is, further, a correct proceeding to allow the vane in the first place to go past the nozzle without being impinged upon, and not to let the jet of steam strike it until the distance between it and the nozzle is such that complete expansion may be counted upon as having taken place. Amongst others who have gone to work in the same way is *Seymour*[3] (1898) (Fig. 147). He provides the outer edges of the vanes *a* with segment-like indentations, which allow them to go freely past the nozzle *c*, but, on the other hand, also have the effect that the jet of steam does not strike the vanes until it has passed to a certain distance from the mouth of the nozzle.

Korn and *Reinhardt*[4] (1900) (Figs. 148 and 149) have similar aims in view. In their wheel the steam is led alternately through an inlet *g* and an inlet *h* set at an angle of 135° with each other, which are worked by a disc *l* provided with projections and fixed on the shaft of the engine against the working-wheel vanes *d*, which are set at right angles to one another. It then drives each of these 45° onward. The jet of steam is here intended to act on the working-wheel vanes at as advantageous an angle as possible. When the inlet *g* is opened, working steam streams

[1] A. P. 530,375.
[2] E. P. 1206 of the year 1866.
[3] A. P. 621,586.
[4] D. R. P. 125,166.

against the working-wheel vanes 1, and moves these 45° onward, on which the inlet *g* is closed and the inlet *h* opened. The working steam streaming through the last-mentioned inlet acts on the vanes 2, which have meanwhile also been moved 45° onward, and drives these with full power 45° further. After the inlet *h* is closed, the inlet *g* opens again, and so on.

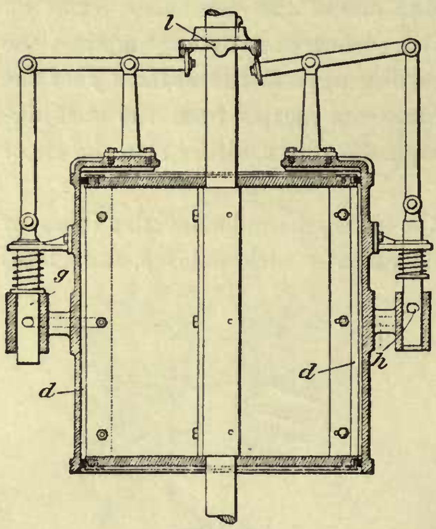


Fig. 149.

In connection with the foregoing, attention may here be directed to an old arrangement made by *Farcot* and *Perrigault* [1] (Figs. 150 and 151), in which a ring G, provided on both sides with vanes *u*, is caused to rotate by an impulse given by jets of steam issuing from the two nozzles *a* and *b*, each of which is distributed over a certain number of the vanes. To admit of this latter the vanes are made to take up the whole breadth of the ring-shaped steam chamber.

In order, as they think, to combine action with reaction, *House* and *Symon* [2] have made the arrangement shown in Fig. 152. The working wheel *a* bears the vanes *b*, the surfaces of which are arranged for right-hand and left-hand motion. Of the steam-pipes *c*, *d* lying parallel with the axis of the wheel, the one is intended to be used for right-hand and the other for left-hand motion. The steam discharges also are arranged with due regard to this. In order further to utilize to the full the expansion of the steam, fixed counter vanes *e* are fitted on the case. Each of these, moreover, is so arranged that, just as a working-wheel vane is receiving its full working jet of steam, its discharge end begins to pass the point of the counter

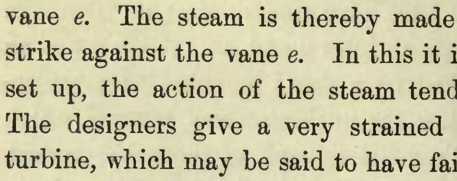


Fig. 150.

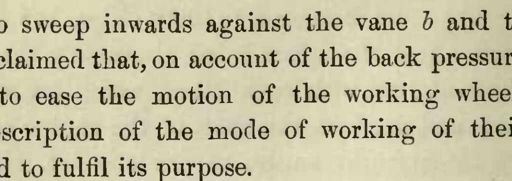


Fig. 151.

vane *e*. The steam is thereby made to sweep inwards against the vane *b* and to strike against the vane *e*. In this it is claimed that, on account of the back pressure set up, the action of the steam tends to ease the motion of the working wheel. The designers give a very strained description of the mode of working of their turbine, which may be said to have failed to fulfil its purpose.

Finally, mention may be made of the design sent in by *Scott*, [3] who proposes to save the vanes and to avoid jerks (Fig. 153). The working wheel *a* is provided at

[1] E. P. 1206 of the year 1866. [2] E. P. 8832 of the year 1896.
[3] A. P. 565,123.

its periphery with obliquely arranged openings *b*, which lie opposite counter-oblique openings *c* of the fixed annular chamber *d*. The fresh steam is led simultaneously through the pipe *e* to the chamber *d*, and through the partially hollowed shaft *f* to the working-wheel *a*. At first the steam issuing from the openings *b* works by reaction on the working-wheel *a*, propelling it forwards. Then, however, the jets of steam issuing from the wheel strike against the jets issuing from the openings *c* of the chamber *d*, so that the latter now exercise the motive pressure action on the wheel *a*.

In the case of impulsive wheels also, a graded transmission of the energy of the motive medium is imaginable. This could be effected in such manner that the jet of steam were directed again and again against the same wheel, or were made to strike several wheels one behind another, or a combination of these two methods might be made. Durability would, it is true, not be imparted to the impulse wheels by such arrangements. In deference, then, to ever-recurring attempts in this direction, a few typical cases only need here be picked out for illustration.

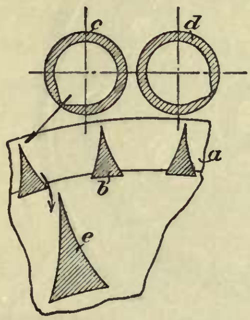


Fig. 152.

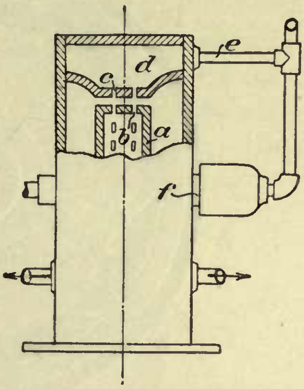


Fig. 153.

McAllister,[1] then, has placed several wheels upon the same shaft, but in separate chambers. The steam, after impinging upon the first wheel, passes through two or more channels arranged in the wall of the case to the nozzles of the second wheel, and so on.

In *White's* impulse wheel[2] (1893), on the other hand, the several-times-repeated action of the steam on one and the same wheel is met with. To this end a circulating channel is provided in the wall of the case, into which the steam, after it has streamed through the fresh-steam nozzle against one of the vanes, is made to pass, so that it may proceed to impinge upon another vane lying further forward in the direction of rotation.

It is of interest to note that quite recently the *British Thomson-Houston Company* has taken up the same idea.[3] Within the case they also lead a steam channel round the impulse wheel in such a way that the vanes of the latter are again and again struck by the steam. The channel widens from the entrance to the discharge end. The designers have also thought of allowing two or more wheels to work together on the compound principle, or of leading the steam to an axial or radial turbine after it has done its work in the impulse wheel.

In the arrangement made by *Graydon*[4] the steam, entering tangentially, meets with radially arranged vanes. After half a revolution has been accomplished, it

[1] E. P. 14,885 of the year 1897.
[2] A. P. 507,468.
[3] E. P. 26,646 of the year 1902.
[4] E. P. 17,098 of the year 1901.

leaves the first wheel to stream on through an overflow pipe into the casing of the neighbouring wheel, and there to complete the revolution, and so on. Meanwhile it is not intended that the whole of the steam which enters the first wheel shall pass on after a half rotation into the second wheel. A part of it is carried round by the cells of the first wheel, so that the fresh steam in entering finds the vane cells partly filled. Also the vanes are provided with perforations, through which the steam expands, so that its forward motion is quicker than that of the vanes, and does not tend to retard, but rather to assist the rotation.

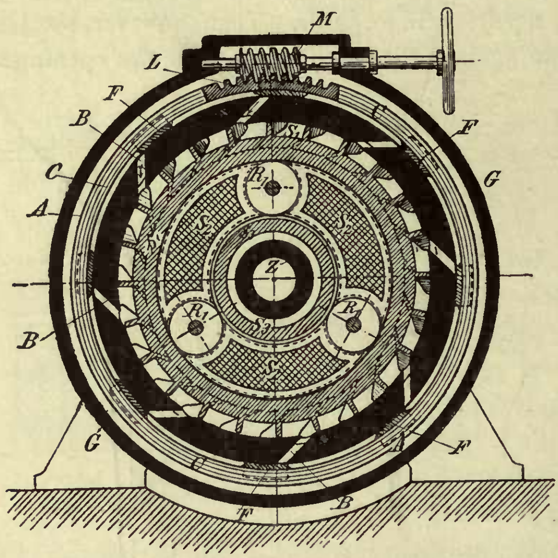


Fig. 154. (Section through *a*, *b* in Fig. 155.)

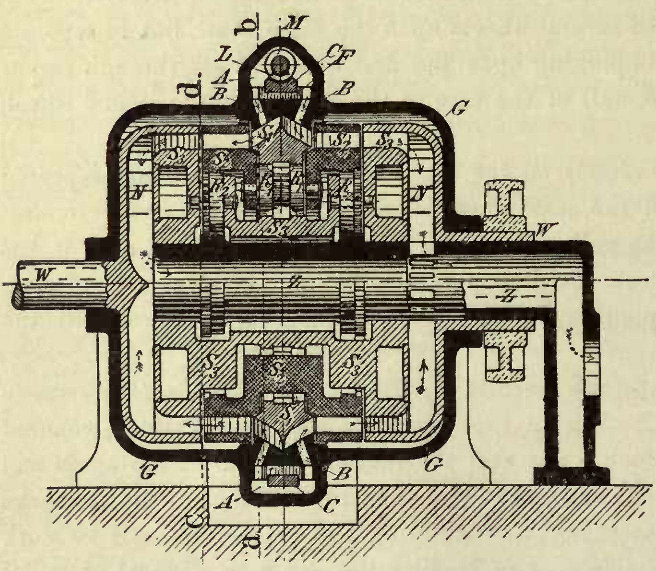


Fig. 155. (Section through *g*, *h* in Fig. 157.)

In the design of *Wirth and Co.*[1] (1898) the velocity of the steam, as well as that of the single vane wreaths, is said to gradually decrease. To avoid choking, the successive vane wreaths are provided with sectional areas at least equal to the foregoing ones, *i.e.* they are of the same diameter. On the other hand, however, a decrease of velocity is obtained by a forced reduction of the number of revolutions per minute. The arrangement is illustrated in Figs. 154 to 157. The motive jet proceeds from the distribution chamber A of the case G through the leading channels B, the openings of which can be regulated by a single valve F and a common ring C actuated by the tooth segments L and the worm M. It first takes

[1] D. R. P. 104,972.

a tangential direction towards the first vane wreath S_1, the vanes of which are arranged symmetrically (Fig. 155). After parting with equal portions of its kinetic energy towards the two sides, the jet is diverted from the tangential direction towards the second set of vane wreaths S_2, which are fixed together. Thence it proceeds to the third and last set, and it is finally made to take the axial direction. Adjoining each third (*i.e.* last) wreath of vanes is a neutral chamber N, which, in order to prevent parts of the jet from developing centrifugal force, is provided with smooth walls, and presents to the steam that has finished its work a free passage—now in the radial direction—to the fixed hollow shaft Z and its discharge opening. The fixed connection between the various vane wreaths can be made in the manner illustrated in Fig. 157. The first vane wreath S_1 is guided on rollers R_1, the inner

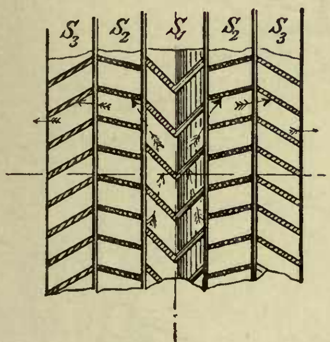


Fig. 156.

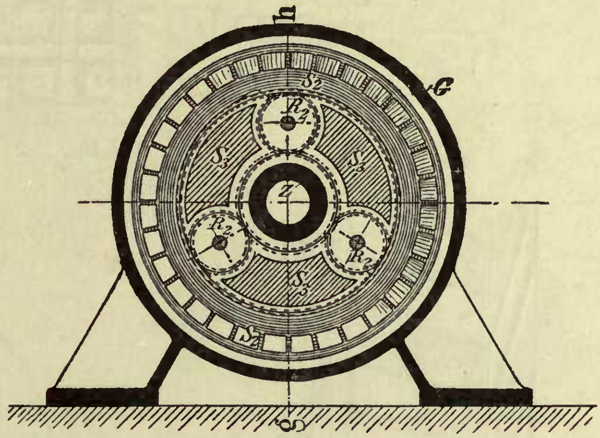


Fig. 157.

path of which lies on the rotating body of the second-following (*i.e.* of the third) vane wreath S_3, while the bearing ring for the rollers R_1 is formed by the rotating body of the second vane wreath S_2. This latter is further guided by the rollers R_2, the inner paths of which are on the fixed shaft Z, while the ground ring required for this second (last) system of rollers is formed by the third (last) vane-wreath system. Thus for each number of the vane-wreath systems the number of the corresponding system of rollers is smaller by one. The curved surfaces of all the rollers and those of their paths are provided, over a portion of their axial dimension, with teeth. This is intended to effect a combination of all the turning moments produced in the various vane wreaths by the impulse action of the working jet on the one of these latter which is for the time being the last, that is to say, on its concentric sets of shafts W. According to the choice of the diameters of the rollers of the various systems, the relative proportions borne by the numbers of revolutions per minute of the successive vane wreaths can be arranged as desired.

The practically unimportant method of construction proposed by *Müller*[1] (1898), which has for its object the retardation of the velocity of rotation, is given in Figs. 158 and 159. The case consists of several cylinders of different diameters,

[1] D. R. P. 105,688.

into which the fixed guide wheels *a*, *b*, *c*, *d*, *e* and the working wheels *h*, *i*, *k*, *l*, *m* are built. The sectional areas of the inlet-slots *p* are in the small guide wheels *a* quite narrow, and widen out in the direction of the largest guide wheels *e* in proportion to the increase in the diameters of these. The angle of the inlet slot *p* is as small as possible, and may be from 12° to 15°. The guide wheel *e* contains twelve such inlet slots. The working wheel *m* is divided into twenty-six cells, the radial walls *r* of which stand at angles of from 75° to 80°. The cells 1, 3, 5, 7, etc., are bounded at the side of the guide wheel *e* by the partition wall *s*, and the cells 2, 4, 6, etc., are bounded at the discharge side by the partition wall *t*. The end walls *r* are provided with numerous long, narrow slots, *n*, *n*. The partition walls *s*, together with the

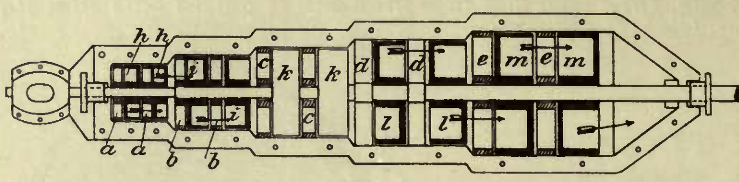


Fig. 158.

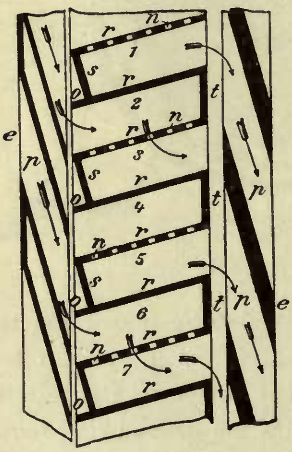


Fig. 159.

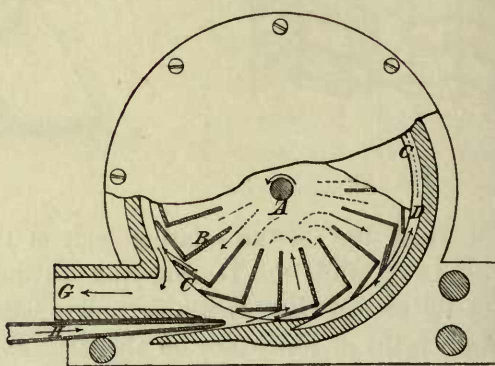


Fig. 160.

vane walls, form the cells *o*. The mode of working of the engine is as follows: The fresh steam streams through the guide wheel *a* against the cells and against twelve slotted end walls of the turbine working wheel *h*, shoots through the slots *n* against the next-following vanes, parting everywhere with some of its energy, and passes out of the first turbine at a reduced pressure, to repeat the same process afresh.

For practical purposes, only the jerkless action of the steam on the vanes comes into serious question. The impulse wheels become pressure wheels, the vanes of which are so formed that they begin with the direction of the jet of steam, and gradually divert the latter (tangential wheels).

A primitive embodiment of this idea is to be seen in the design of *Dufort*[1]

[1] E. P. 2368 of the year 1876.

(Fig. 160), in which, it is true, the alteration of the jet of steam from the tangential to the radial direction is not effected gradually. The working wheel C placed on the shaft A has vanes B, the inner-lying parts of which are arranged radially. Against these are placed fender plates at angles of nearly 90°, the extremities of which meet the periphery of the wheel at the points at which the prolongations of the radial parts would cut the same. The jet of steam issuing from nozzle H at first acts directly against the vanes that are, for the time being, just in front of it. The steam then proceeds through the wheel channels into the inside of the wheel, and expands through the channels in the annular chamber D, which for the moment are

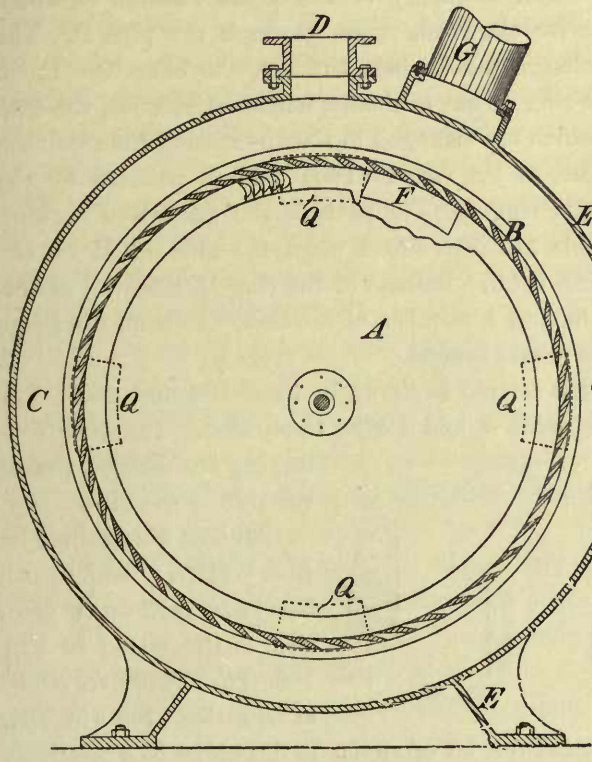


Fig. 161.

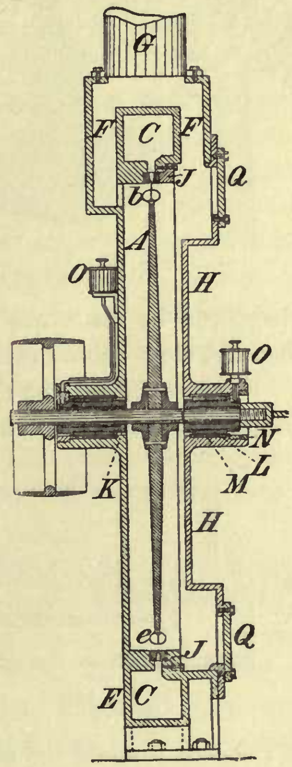


Fig. 162.

not being directly impinged upon, thus again performing work before it finally arrives at the exhaust G.

Maardt[1] (1895) aims at a reduction of the revolutions per minute of the working wheel, and seeks to prevent the steam from carrying too much air with it, and so losing kinetic energy. The solution of the first problem is sought in an increase of the working wheel disc, and that of the second by the use of specially formed vanes. The largely dimensioned steel working-wheel disc A (Figs. 161 and 162) is turned on both sides, and, in order to give strength, is made thicker at the nave than at the edge. In the latter is worked a groove, into which the feet *c* of the vanes are inserted (Fig. 163). The vane adopted by Maardt is punched out of the solid in the

[1] D. R. P. 87,519.

form of a double cup of steel or iron, and has a sharp edge along its circumference, as well as along the central rib near the point *a*. Each of the cups is of such form that the jet of steam which strikes at the point *a* is divided into two, each of which passes through its own cup in a direction of such obliquity that the used-up steam streaming out at the points *b*, *b* does not strike the edge of the turbine which bears the cups. At the same time the steam jet is intended to have room to spread out more and more during its passage through.

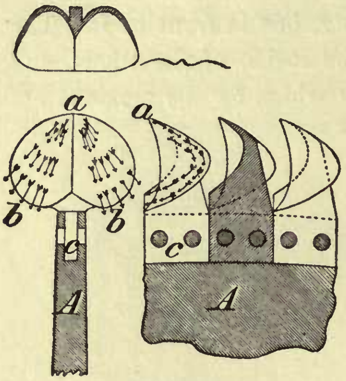


Fig. 163.

The ring B, lying close outside the vanes, forms the inner boundary of the steam channel C, which receives the fresh steam through the pipe D. The discharge takes place through the chambers F, G. The ring B has conically narrowed nozzles, the axes of which are arranged in the direction of the receiving surface of the vanes. It is held in position by the elastic ring J. The parts of the guide and working wheels are introduced when the side-lid H of the casing is off. Instead of the ring B, however, sector-pieces can be made use of, each having a number of nozzles. A steam discharge then lies close behind each of these nozzle bearers.

For an axial turbine the vanes are altered as shown in Figs. 164 and 165. The steam then enters the vanes at the point *a*, and leaves them at *b*. The ring containing the nozzles lies at the side of the disc.

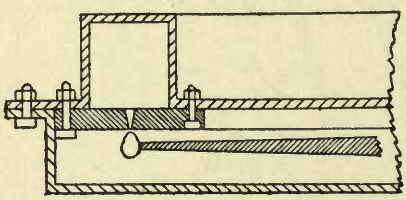


Fig. 164.

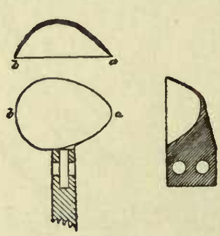


Fig. 165.

An idea entertained by *Kolb*[1] (1897), which can hardly be realized in practice, is illustrated in Fig. 166. The object here aimed at is, without the widening of the cells towards the sides, to cause the jet of steam to discharge in a relatively tangential direction, and thus, at the normal speed of the motor, to make the absolute discharge velocity = 0, thus giving the working wheel the benefit of the whole of the velocity of the jet. The working wheel consists of a circular channel *a* with a smooth inner surface, and with one or more vanes *b* provided at the steam-inlet side with knife edges. The jet of steam led from the nozzle *c* against the inner side of the wheel is divided, while the knife edge of one of the vanes is passing, into two parts. One of these strikes directly against the vane in question, while the other is led from the channel *a*, with an approximately constant relative velocity, to the next vane. In the illustration only one vane is shown, which, in addition to the direct-acting jet of steam, is struck by the part which is separated from it at its meeting with the knife edge of the vane, and carried round by the channel. It may well be

[1] D. R. P. 93,654.

imagined that a diverted jet of considerable thickness will be cut off by the knife-edges of several vanes in succession.

In view of the slender practical importance of turbines the motive media of which are allowed to expand in the atmosphere in front of the working wheel, it will here suffice to give a few examples only of multi-stage engines so designed. In this, however, it may be pointed out that the same mode of construction may be utilized for the pure velocity turbines to be discussed later.

As was done by Imray,[1] *Lilienthal*[2] (1890) also leads the steam in serpent-like coils round the working wheel, in order to produce velocity stages. For the vanes is chosen the open hollow form, which admits of the free circulation of the surrounding medium, and which, along the whole path through the cells of the wheel, does not hinder the action of the steam entering in the tangential direction. To enable the kinetic energy of the steam to be fully utilized, a turbine, having at its periphery the hollow vanes alluded to, would require to have a peripheral velocity equal to half the velocity of the jets. The attempt is here made to effect full utilization of the kinetic energy with a low velocity of the working wheel. The jet of steam issuing from a vane *a*, and still possessing energy, is, by means of an adjoining reversing vane, led back to be tangentially introduced into the next turbine vane, and so on, until the current energy of the steam is exhausted. As may be seen in Figs. 167 and 168, the steam proceeds from the side-nozzle first in a tangential direction into a pocket *s* of the wheel S, glides along the curved wall, and arrives in the fixed channel *g* of the leading apparatus G, which lies opposite, to be from here led back again into the next-following guide-wheel channel *s*. The steam thus streams through a cylindrical coil lying obliquely against the inner wall of a tube that is formed by the channels G and S.

Moorhouse[3] (Figs. 169 and 170) counts only upon pressure stages. He constructs the case in several pieces, inserted one after another between the end castings, which are held together by long bolts. The separate parts form chambers, the capacities of which increase from the steam-inlet side to the exhaust side. In each chamber is placed a working wheel *i*, the vanes of which are radially impinged upon from nozzles *n*. In the case of the higher stages the impingement takes place from within, while in the case of the two last stages it comes from without. The sectional areas of the nozzles *n* increase also from stage to stage. The process assumed by Moorhouse to be followed in working is, that the steam entering through the branch *h* reaches the first group of nozzles with undiminished pressure, and impinges upon the vanes of the first working wheel. At the same time, it parts with a certain amount of its pressure, and, thus reduced, it streams to the wider leading nozzles of the second group, and thence to the second working wheel. In this second stage also a diminution of pressure of fixed amount takes place, and so on. The naves of the working wheels run close against the fixed chamber walls. They are pressed against a spring resting against a ring-projection on the shaft.

[1] See p. 63. [2] D. R. P. 54,631.
[3] E. P. 2068 of the year 1876.

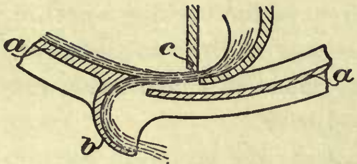


Fig. 166.

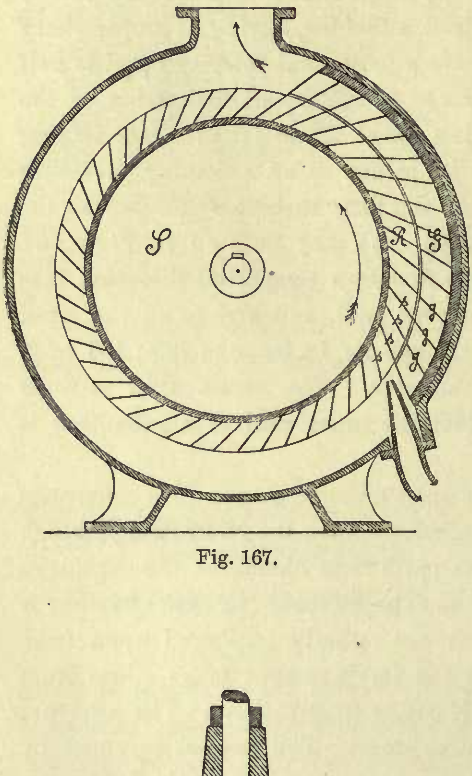


Fig. 167.

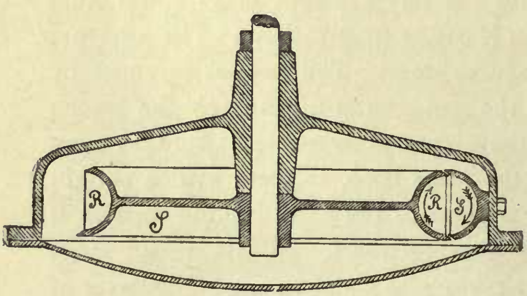


Fig. 168.

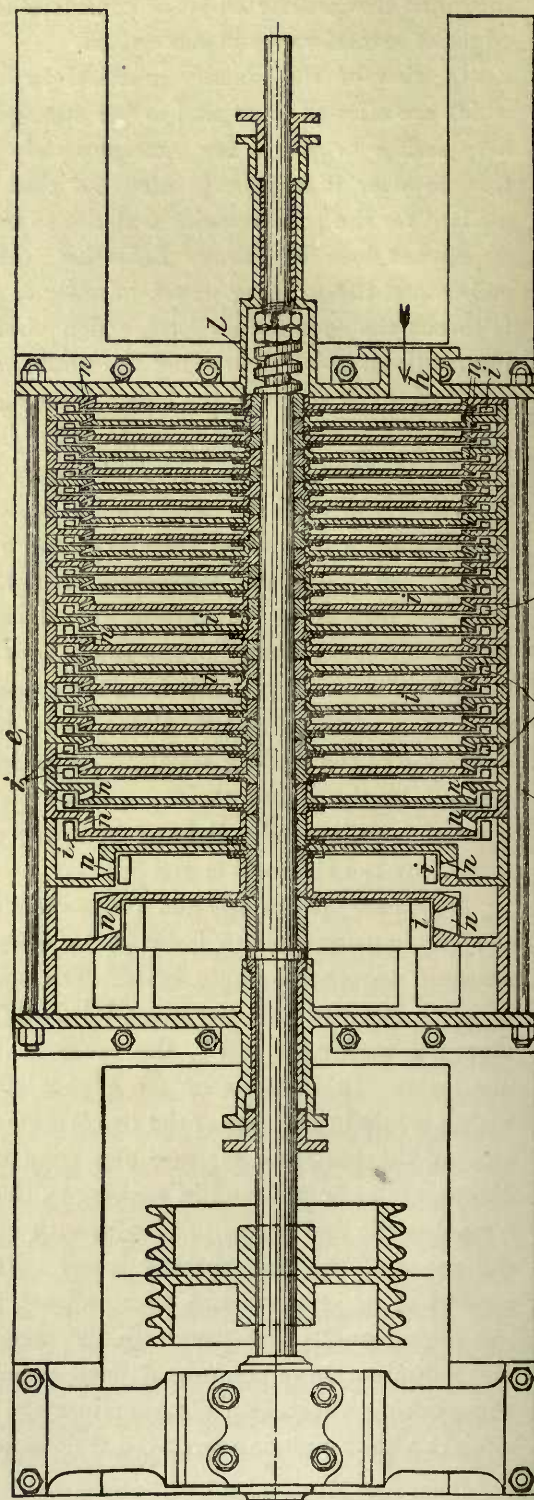


Fig. 169.

c. **Velocity Turbines with Acceleration of the Steam in the Leading Apparatus by means of Expansion.**—The steam which expands outside the nozzles, and which in the free-jet wheels is mostly made to perform work during the period of expansion, is able to convert only a small portion of its pressure energy into current energy, so that the working of the velocity turbines hitherto discussed has not given a satisfactory economical result. Experimental trials, indeed, have shown that cylindrical nozzles, and such as converge in form towards the mouth, admit of an expansion of the steam of at most 57·7 per cent. of the initial pressure. A better state of things was produced for the first time by the invention of *de Laval*,[1] which is based on the discovery that a complete expansion of the steam to the pressure of the atmosphere and below it may be obtained when it is caused to steam through a nozzle that is gradually increased in a suitable manner.[2] The nozzle *a* (Fig. 171), the mouth of which is brought close to the wreath of working-wheel

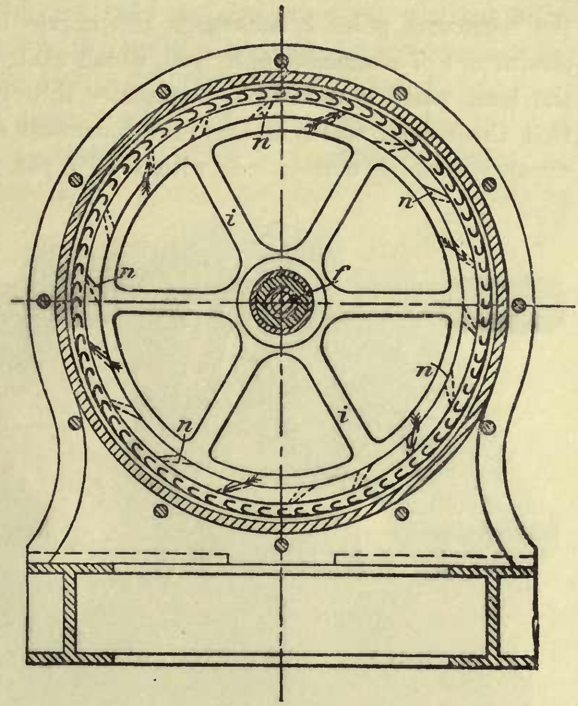


Fig. 170.

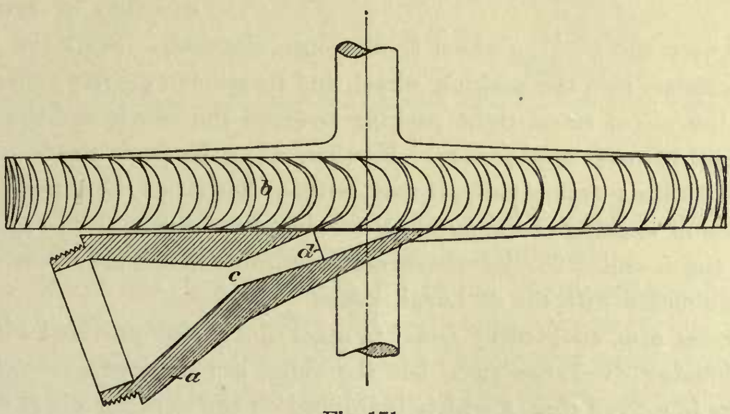


Fig. 171.

vanes *b*, has, in accordance with the theory, a narrowest point, *c*, which bears a certain relation to the quantity of steam that is to be dealt with. From the point *c* a conical increase of width takes place, which reaches right to the shaft *d*. The wheel is axially

[1] E. P. 7143 of the year 1889. A. P. 522,066. D. R. P. 84,153.

[2] Compare Chap. XIV.

impinged upon. The increase ceases on that side of it on which the vanes *b*, in their rotary course, arrive opposite the nozzles, while extensions in the lengths of the nozzles are carried to the points at which the vanes pass clear of them. From the narrowest point *c* backwards the nozzle is widened out with a view to the attainment of a steam-supply lead, which shall be as wide as possible, and occasion the least possible reduction of pressure through friction. De Laval then assumes that the steam, when leaving the narrowest section *c* of the nozzle, undergoes expansion which corresponds with a 57·7 per cent. fall in pressure in front of the point *c*, and that the remainder of the pressure is converted into velocity in the streaming of the current through the nozzle—*i.e.* up to the point *d*—so that the steam at the point of discharge is no longer under pressure (or is very slightly so). The nozzle is designed with a view to a possible increase of velocity of the steam. For this reason its sectional area decreases in a smaller ratio than that of the increase of volume of steam streaming through it at a constant velocity. The adoption of several nozzles distributed over the working wheel is, of course, desirable. Since the jet of steam passes pressureless into the working wheel, and its specific gravity suffers no further alteration, the use of steam-tight packing between the nozzle and the vanes of the working wheel becomes unnecessary. Further, since the fresh steam does not come in contact with the rotating parts, it may, without hesitation, be superheated.

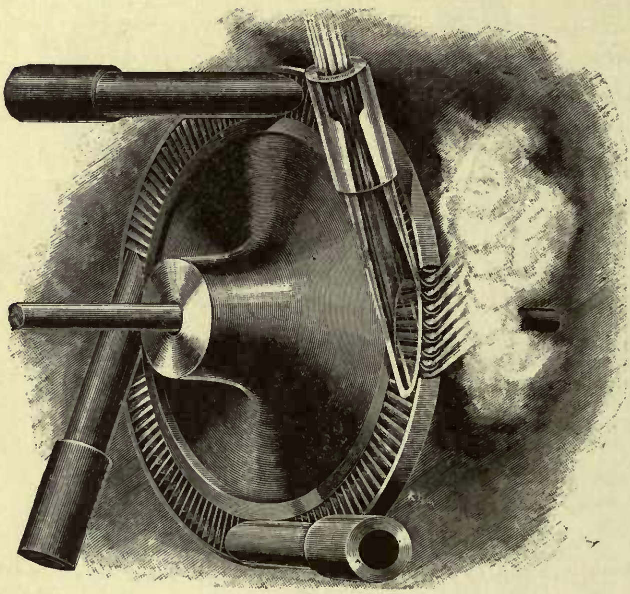


Fig. 172.

The mode of working of the de Laval turbine may be seen from Fig. 172.

Clearly the so-called free-jet wheels are practicable in steam-turbine construction only in combination with the de Laval nozzle.

The nozzles, also, adopted by *Lake*[1] (Figs. 173–175) are provided with shoulders at their widening discharge ends, but the vanes are impinged on radially. The nozzles *a* are in a fixed ring, *b*, which is shipped on the working wheel *c* with just so much play that the steam can escape from the pockets *d* in an axial direction. These pockets are of the form shown in Figs. 173 and 174. The jet is divided by the knife-edge *e*, so that the steam passes out on both sides of the wheel. This knife-edge is cut away at the point *f*, so that a still sharper separation between the

[1] E. P. 17,273 of the year 1894.

two pockets results. The annular chamber *g*, which surrounds the nozzles *a*, and is in connection with the steam inlet, is closed to the turbine case, into which the exhaust steam makes its escape.

Whitcher and *Roberts*[1] (Figs. 176 and 177) likewise cause the wheel *a* to be impinged upon from without, and this in the radial direction. The vanes *b* are set so as to form angles of about 60° with the radii. They form pockets, which are each divided into two equal parts by the annular knife-edge *e*, which is left standing between the annular grooves *c*, *d*. In the

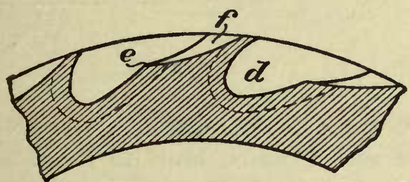


Fig. 173.

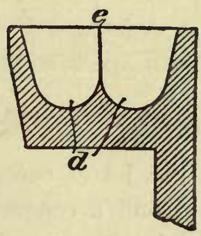


Fig. 174.

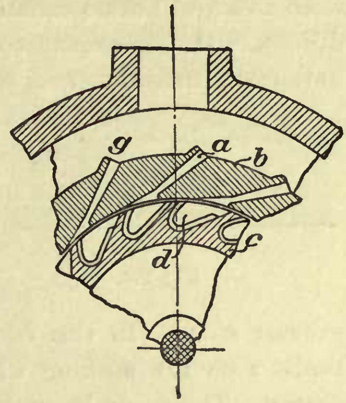


Fig. 175.

nozzle *f* the steam receives an acceleration caused by expansion. It strikes against the knife-edge *e*, and is guided by it into the two half-pockets, that is to say, it is divided by it in the direction of the axis of the wheel, in order to be subjected to equal alterations of direction by the two annular grooves *c*, *d*.

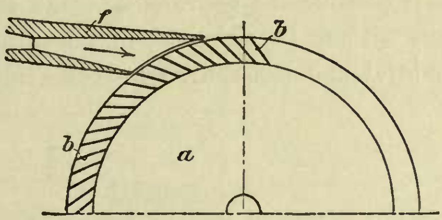


Fig. 176.

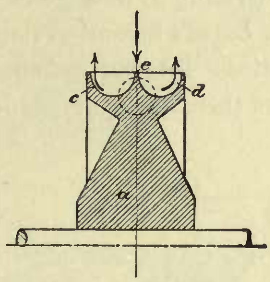


Fig. 177.

The *Riedler-Stumpf* turbine, which will be discussed in Section XXI., in connection with other turbine arrangements, may here be mentioned.

Attention should first, however, be drawn to the experiments of *Stumpf*,[2] who made use of expansion nozzles *a* (Fig. 178) of rectangular section, and had pockets *b* worked into the periphery of the working wheel to serve as steam channels.

It is here taken for granted that the steam in the nozzles expands to the atmosphere, or, if condensers be used, to the pressure obtaining in these, so that the pressure is thoroughly converted into velocity. If, then, the jet of steam, passing from the nozzle *a* into the vane *b*, has the same pressure as the space in which the working wheel runs, the jet will always have the same section as that of the nozzle *a*

[1] E. P. 2815 of the year 1900. [2] E. P. 17,951 of the year 1901.

(Fig. 179). If the pressure of the steam decreases below the pressure in the vicinity of the working wheel, compression will take place (Fig. 180); but should it exceed the pressure mentioned, a further expansion will take place (Fig. 181). Consequently, the pressure of the steam and the outer pressure will together affect the form of the passing jet. Slight, but continuous variations of the pressure of the fresh steam will cause variations in the jet. This is adapted to certain definite conditions, in the same manner as the variation of the counter-pressure, as, for instance, during the transition from work in the condenser to that done by the help of the

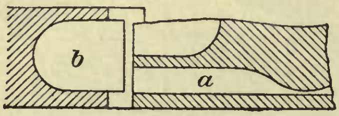


Fig. 178.

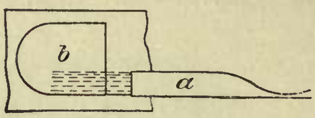


Fig. 179.

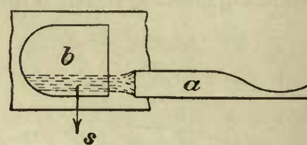


Fig. 180.

exhaust steam. In this case the jet of compressed steam will produce an axial thrust *s* on the working wheel and a concussive action, which leads to losses of energy. These may be avoided by the use of very broad vane cells *b*, arranged in the direction of the axis. Owing to the high velocity of the wheel and the great friction, this can, however, only be done within certain limits. On the other hand, if the jet of steam expand (Fig. 181), an axial strain *r* on the wheel will be produced in addition to a loss of pressure. All these drawbacks can be avoided if double **U**-shaped pockets *c* (Fig. 182) be made use of, and the jet of steam directed straight against the dividing knife edge of the two half-pockets.

While *de Laval's* invention renders possible the conversion of the energy of the compressed steam into current energy without the existence of the turbine as a profitable engine thereby becoming doubtful, and has achieved success in so far as the

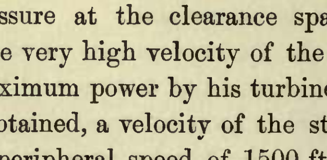


Fig. 181.

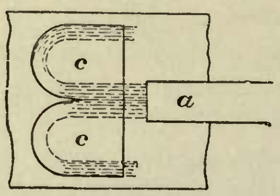


Fig. 182.

excess pressure at the clearance space—always present in pressure turbines—is avoided, the very high velocity of the steam prevents the attainment of more than a small maximum power by his turbine. If a theoretically perfect supply of energy is to be obtained, a velocity of the steam of, for instance, 3000 ft. per second, will require a peripheral speed of 1500 ft. in the working wheel, which explains the enormous number of revolutions per minute made by the de Laval turbine. By an increase in the diameter of the working wheel the number of revolutions can be reduced; but in practice this can only be done within certain limits. The attempt has been made to utilize the energy of a given quantity of steam by stages, and this step has brought both the pressure turbines and the velocity turbines—the latter in connection with the de Laval nozzle—to their present importance. The division of the energy into grades can be made in different ways, as will be shown.

It may first be assumed, that an impinging nozzle has to convert the total pressure of the current of steam into velocity. In this case there is no alternative but to gradually reduce the speed of the jet of steam, which is no longer under pressure, by causing it to act twice, three times, or several times in succession, on the vanes of the working wheel. By this means, *velocity turbines* with the *velocity grades* are obtained. According as two, three, or several grades are adopted, one half, one-third, or a still smaller fraction of the number of revolutions of the single-grade turbine may be obtained, the general conditions remaining the same.

The Paris firm of *Breguet*, which manufactured Laval turbines so long ago as the year 1894, proposed a development of these, which may in a certain sense be

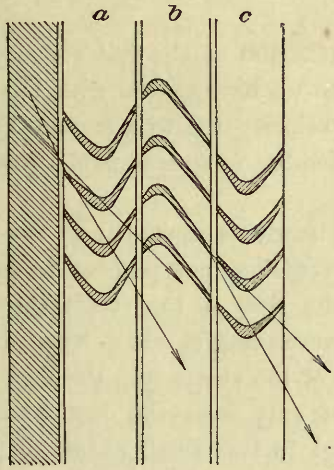


Fig. 183.

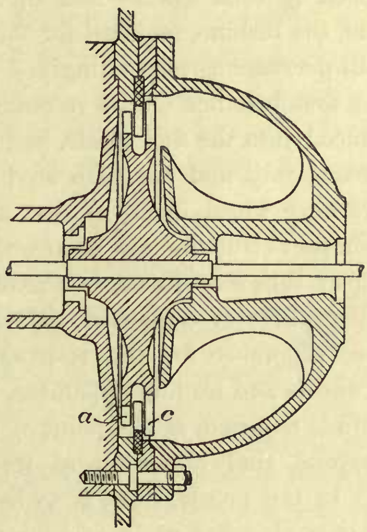


Fig. 184.

regarded as a forerunner of the present turbines, with several grades of velocity.[1] Even the Laval wheel with a circumferential velocity of 1400 ft. per second, leaves a considerable part of the steam energy unused, the steam leaving the channels of the wheels with considerable velocity. In order to utilize this remaining energy, Breguet has inserted a stationary leading apparatus *b* behind the wheel *a* (Fig. 183), which conveys the steam to the channels of a second working-wheel wreath *c*. As shown in Fig. 184, the wreaths *a*, *c* may be parts of the same working wheel. At any rate, they are intended to drive a single shaft only. If, therefore, the principle laid down here is to have practical importance, when the wheels *a*, *c* are of the same diameter, the rotations per minute of the two grades *a* and *c* must be the same. To allow the wheel *c* only a fraction of the steam velocity provided for the wheel *a* would only cause the wheel *c* to exercise a propulsive action on the steam which streams into it. This consideration led to the decision to give

[1] F. P. 236,883.

the two wheels a, c the same effective steam velocity. (Breguet contemplated the use of more than two wheels.)

The most important example of this type of turbine has been produced by *C. G. Curtis*[1] (1896), who conveys the steam through an expansion nozzle, so as to convert the pressure into current energy before the steam enters the first working wheel. This current energy, however, is not utilized in one wheel only with a single velocity grade.

Two or more working wheels are placed one behind another, each of which consumes part of the energy of the working steam, and correspondingly reduces its velocity. Between every two working wheels is inserted a leading apparatus, which alters the direction of the steam leaving the first wheel in suitable relation to the position of the vanes of the following one. As only a fraction of the steam velocity is consumed in each grade—half of it in the case of two grades—the speed of rotation of the turbine, required for the full utilization of the kinetic energy of the steam, will decrease correspondingly.

In the first instance Curtis recommends that steam of the full velocity-energy be introduced into the first grade, as in this case we have to deal with the velocity of the steam only, and not with an increased velocity due to a further expansion in the working wheel itself, so that a small number of grades, and consequently a more compact structure, can be arranged for.

To effect this, the stream in the expansion nozzle is expanded to the pressure of the atmosphere or condenser. Curtis, however, does not propose to go so far. He makes allowance for the resistances to the flow of the steam through the working wheels and leading apparatus, for the overcoming of which a certain excess of pressure is required, at the point of entrance of the steam into the first working wheel, beyond that obtaining at its point of exit from the last wheel. The expansion in the nozzle, then, is to be restricted to this small excess of pressure. As the latter is sufficient in amount to be further taken into consideration as an adjunct to the power required for the rotation of the wheels, the classification of the Curtis turbine amongst the velocity (pressure) turbines with pure velocity stages is a correct one. A gradual increase of section of the steam passages is required, not so much on account of the further expansion of the steam within the turbine due to the overpressure, as because of the retardation of the current velocity from working wheel to working wheel, occasioned by the conversion of the current energy, from grade to grade, into useful work. This has resulted in the arrangement, adopted by the designer and shown in Figs. 185 and 186, of a turbine with two grades. Fig. 186 shows a cross section through x, y in Fig. 185. According to this design the steam is first expanded in the nozzle G from d to e. Thence it passes at full velocity into the working wheel f, the cells of which expand radially. The steam then streams with only about half its velocity through the leading apparatus E into the working wheel G, on leaving which it arrives at the discharge pipe H with only the energy requisite to overcome the additional

[1] A. P. 566,968. D. R. P. 104,468. *Zeitschr. Ver. D. Ing.* 1903, Vol. 47, p. 1120.

resistances. The gradual increase of the volume has been effected by the enlargement of the working wheel and leading channels in the radial direction. This radial enlargement is also intended to render possible a reduction of the angle of discharge. In regard to the leading apparatus Curtis says: "If the stationary passage be curved around, so as to deliver the fluid to the second set of vanes at an angle equal to that of the nozzle, as is desirable, a considerable increase in the radial dimension is necessary to make up for the reduced width of the passage resulting from the reduced angle of its walls and to produce the additional enlargement necessary to permit the desired expansion." The channels are uncovered both inside and outside. The vanes of the working wheel are mounted on the common wheel C, whilst the leading apparatus is attached to the wall of the case B, which is made perfectly tight. When the outlet H is connected with a con-

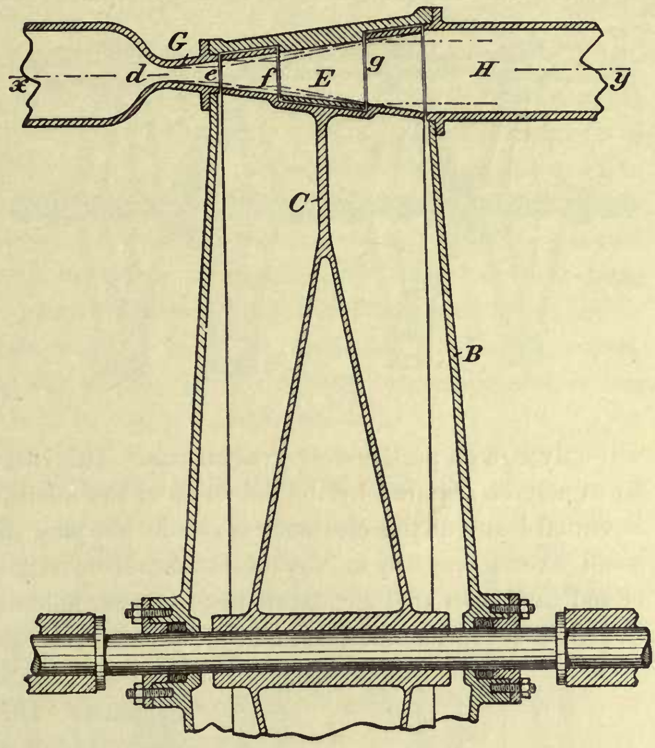


Fig. 185.

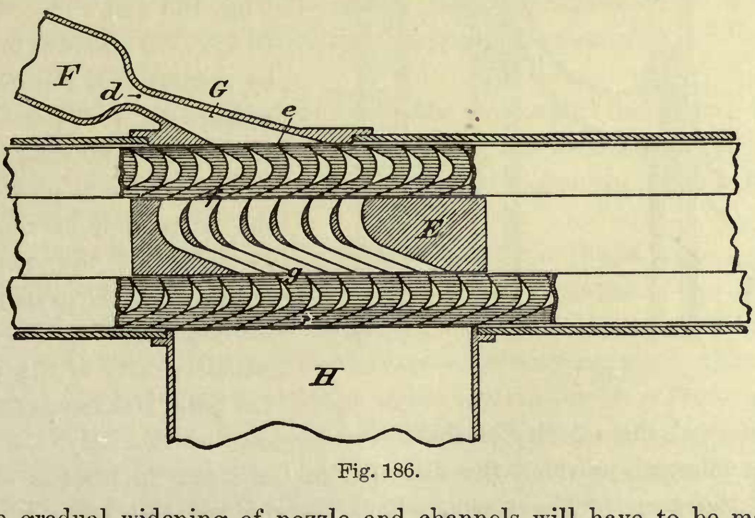


Fig. 186.

denser, the gradual widening of nozzle and channels will have to be made in the manner shown by the full lines. In case of discharge into the atmosphere the enlargement will have to be made as shown by the dotted line. The construction can, however, be made in such a manner that the channels of the working wheels are not widened. In this case the complete enlargement of the volume of steam must be effected by the widening of the leading channels in a still greater degree, as shown by the dot-and-stroke lines in Fig. 185.

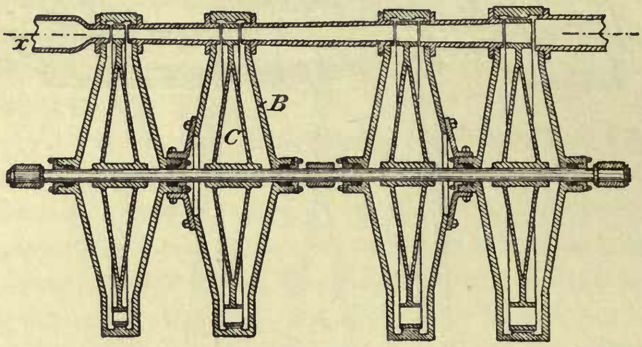


Fig. 187.

If more than two working wheels are chosen, so that the number of the leading apparatus have to be increased, the widening of the cross section of the flow will only be a proportionately gradual one. This can, however, be effected only in so far as may be required by the reduction of the number of revolutions of the turbine. Eventual losses at the clearance space do not play an important part, owing to the small excess pressure in way of these. Moreover, the latter themselves lie in a closed casing, so that a counter pressure soon holds the balance, no matter whether the complete turbine be placed in a case, nor whether each working wheel be built into a separate case. See Figs. 187 and 188, the latter of which is a section at x in Fig. 187.

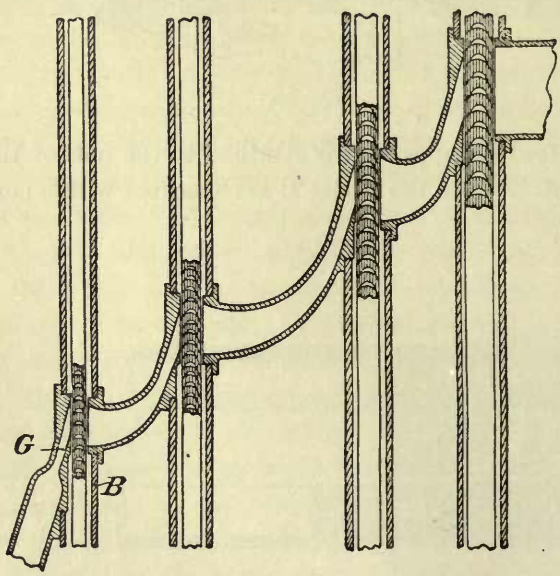


Fig. 188.

In these first designs, the cross section of the inlet nozzle G is not circular, but oblong. (See Figs. 185 and 186.) The surface friction was intended to be reduced by the shape of the nozzle. The passage of the steam into the vanes of the working wheel without concussion is provided for by the adaptation of the angle of incidence of the vanes of the working wheels to the velocity of the steam and to the circumferential velocity of the wheel. As the latter is the same for all wheels of a turbine, and the velocity of the steam decreases continuously, the angle will have to be enlarged, provided the direction of the steam in front of all working wheels be the same. The steam, which on striking the vanes is subject to compression, can then expand within the channels of the wheel to its normal volume.

Each of the stationary leading apparatus is sufficiently wide to receive the steam from all the cells impinged on, and is so arranged as to collect the single jets of steam and convey them to the next working wheel in the form of a single jet of normal volume.

In the case of the leading apparatus also the angle of entrance alters as the velocity of the steam becomes reduced. A suitable arrangement is that in which each leading projection is arranged to shift with the previous one in the direction of motion of the wheel, by which account is taken of the rotary motion imparted to the latter by the steam. For the avoidance of eddies, moreover, vanes are fitted in the leading channels, so that narrower channels are formed and the steam is separated into strips. Account must also be taken of the circumstance that the steam streaming against the concave vanes is, by reason of the centrifugal forces acting upon it, to a certain extent compressed. It is the intention, for the most part, to prevent this compression by the employment of a suitable arrangement of the guide vanes, which also enable it to be utilized in the acceleration of the steam on their outflow sides.

In the method of construction illustrated in Fig. 189 the Curtis idea again shows itself. In this case, however, fixed guide vanes are not fitted. Instead of these, two wheels, *a*, *b*,[1] running in opposite directions, are provided, the vanes of which are so curved that they lead the steam, the one to the other. Further, a radial turbine with impingement from without is adopted, the first clearance space of which, however, lies directly open to the case connected with the exhaust.

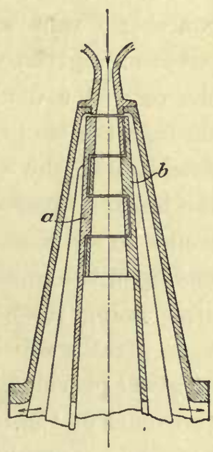


Fig. 189.

According to the older particulars taken from the source referred to, the steam-inlet nozzles cover about one-sixth of the periphery of the first working wheel. In engines up to 700 H.P. they are collected into a single group; in larger engines they are in a number of groups, which are distributed at regular intervals over the periphery.

In a subsequent alteration (1897) Curtis no longer makes use of single-expansion nozzles for the introduction of the steam, but adopts a complete annular opening, which widens out in the direction of flow of the steam in proportion to the expansion of the latter.[2]

Objections have been raised to the designation of the Curtis turbine as a pure velocity turbine, the pressure still existing after the discharge of the steam from the nozzle, small though it be, being considered, in view of the excess pressure at the clearance space, to mark the engine as an (excess-)pressure turbine. This cannot be admitted. The first essential to the idea of the (over-)pressure turbine is, that the steam in the working wheel shall have sufficient pressure to enable it to do propulsive work by means of expansion. The excess pressure at the clearance opening is, indeed, present as a subsidiary phenomenon; but, considering that the Curtis

[1] Compare Chap. IV. [2] A. P. 582,720.

system of construction does not fulfil the first essential conditions of a pressure turbine, it cannot brand it as belonging to this class. Curtis clearly recognized that a steam turbine with several velocity stages cannot, by reason of the centrifugal forces acting on the working jets of steam, in practice work satisfactorily with steam that is completely pressureless. The retention of a certain pressure by the steam will be the more necessary the greater the diverting influences to which the working medium is subject. The same observations have also been made by Stumpf[1] in connection with his velocity turbine. Of great practical utility is the adoption, in the Curtis turbine, of closed steam channels, which admit of a lead of the steam that is sure and free from obstructive eddies. On account of this very advantage, it may be expected that designers will be more and more impelled to arrange the sections of the steam channels so as to suit those of the jets of steam, or, in other words, to adopt the intermediate system of turbine construction.

Bök and *Robsahm*[2] (1898) also lead the expanded steam through several stages, but in the following manner: The axial jet wheel, consisting of two or more concentric vane wreaths, is so arranged that it can produce a maximum of horse power on a given diameter. The current of steam is, in consequence, divided. In the case of a double-wreath turbine, as shown in Fig. 190, part of the steam goes through the lead *m* to the nozzle *d*, and thence passes on to impinge upon the outer wreaths *g* of the working wheels *b*, fitted on the shaft *a* and the guide wheels *h* of the leading apparatus *c*. The other portion of the motive medium streams out of the lead *n* to the nozzle *f*, and impinges upon the inner wreaths *i* of the working wheels, and upon the guide wreaths *k*. The annular chambers *l*, in which the steam arrives after doing its work, are connected with a single exhaust. The counter-motion vane wreaths which widen out in a trapezium-like manner, are for a given diameter of wheel provided with a comparatively large section of passage. The positions of the vanes of the inner and outer wreaths will be such that they work in the same direction of motion. They can, however, be arranged in such a manner that one wreath can be used for the one direction and the other one for the other. In the altered arrangement last described the attainment of the greatest efficiency possible in a given size of wheel in one direction of motion is given up in exchange for the reversibility of the turbine.

Whitcher and *Roberts*[3] also divide the motive medium (Fig. 191). The steam issuing from the nozzles *a* is divided in such a manner that the one half passes through the wheel *b* in the one axial direction, the other half through the wheel *c* in the opposite one. Each wheel has rings of vanes *d* lying side by side, and corresponding rings of guide vanes *e*. The latter are shifted in relation to the former, so that the steam from the one working-wheel cell is led from the corresponding guide vane into the adjacent working-wheel cell of the axially further advanced ring *d*. The steam then expands in a spiral winding round each wheel *b* or *c*. The drawing shows conical working wheels. These and the leading apparatus may also be made cylindrical.

[1] Compare Chap. XIV. [2] D. R. P. 107,419. [3] E. P. 2815 of the year 1900.

It has hitherto been assumed that every velocity grade corresponds with a special working-wheel vane wreath. As in the case of the pressure turbines, however, the velocity grades of the velocity turbines may also be contained in a single working wheel, the steam which has gone through one set of channels being

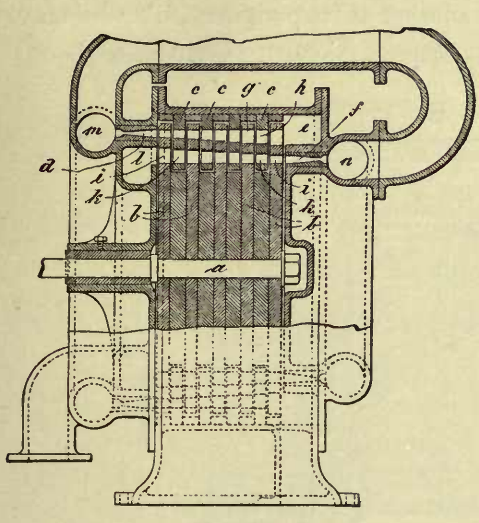


Fig. 190.

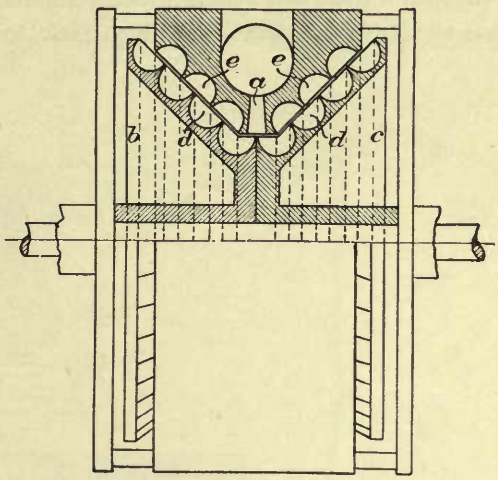


Fig. 191.

reversed in direction by means of a leading apparatus, and led into another set of channels of the same wheel wreath. The number of impingements of a wheel wreath corresponds, then, with the number of the stages, which, however, is only a small one, unless the proportion borne by the impinged part of the first stage to the whole wheel be very small. Account must further be taken of the circumstance that the number of working-wheel channels opened to the passage of the steam increases from stage to stage in proportion to the decrease of the velocity of the current. Further, it must be remembered that in the preceding case the jet of steam has to change its direction by 180° in the leading apparatus, so that the centrifugal forces exercise a detrimental influence of exceptionally great extent.

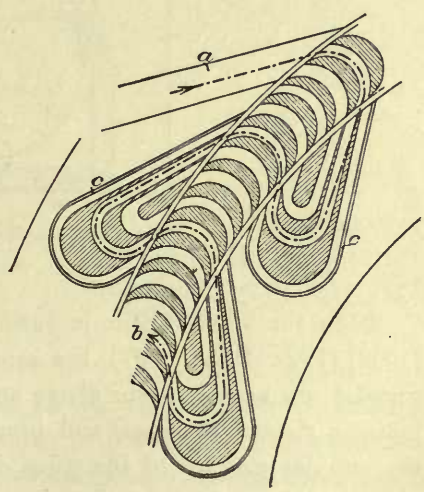


Fig. 192.

De Ferranti[1] (Fig. 192) first allows the steam to expand in a nozzle, or in a tube *a*, before an impingement of the working wheel *b* takes place. The latter is bounded by the leading apparatus *c*, which repeatedly guide the steam into the cells of the same wheel, and the stages, it may be observed, are stepped down in the backward direction of the motion of the wheel. The heights of the

[1] E. P. 2565 of the year 1895.

channels of the working wheel and of the leading apparatus meanwhile remain constant. Thus account is not taken of the increase of volume caused by the retardation of the flow of the motive medium.

Riedler and *Stumpf*, who slot out the working-wheel channels in the outer wheel of a disc, and arrange them one above another at its periphery, likewise make use of return vanes in arranging the velocity stages. (Compare Chap. XXI.)

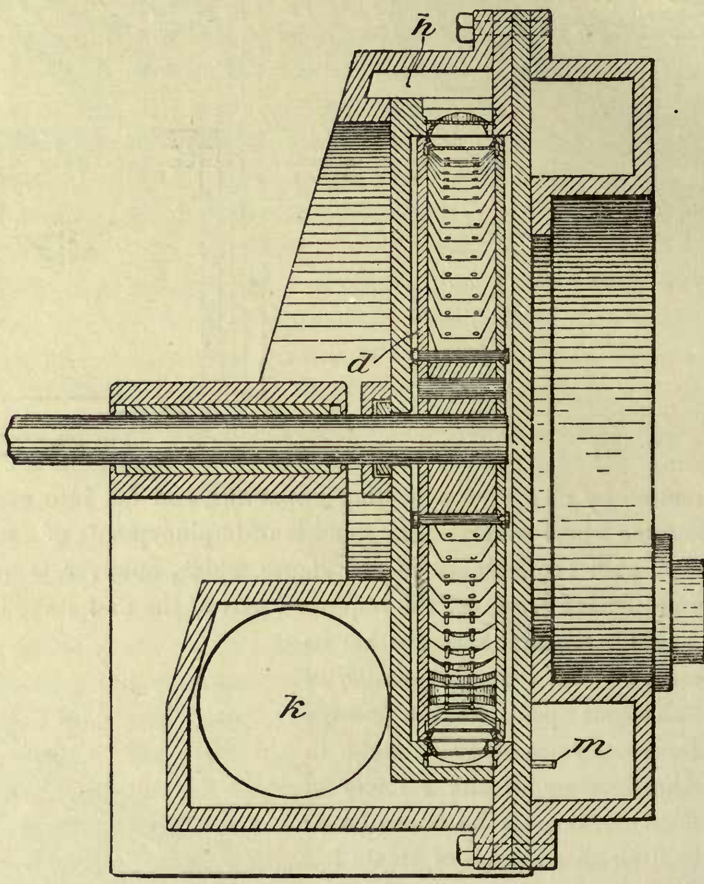


Fig. 193.

As in the Riedler-Stumpf turbine, so also in the turbine designed by *Terry*[1] (1902) (Figs. 193 to 197), the semi-cylindrical steam channels *a* of the working wheel *b* are arranged one above another. The channels, however, are formed by plates *c* riveted together, and provided with projections, which are inserted into the annular grooves of the discs *d* forming the body of the wheel. Between the plates *c* lie the leading plates *e*. Around the working wheel is arranged the ring of fixed leading channels *f*, which also are piled one above another, and so arranged that the steam-outlet sides are shifted with the inlet sides forwards in the direction of rotation of the wheel, a measure which requires a correct leading of the steam

[1] A. P. 741,385. F. P. 335,820.

from each working-wheel channel into the one next following. In Fig. 195, which represents a section through xx in Fig. 194, this displacement is illustrated. The side walls of the leading channels show sickle-shaped openings g, through which the steam, after doing its work in the annular collecting chamber h, is discharged on its way to the outlet k. At certain intervals, then, expansion nozzles l, which can be closed by the valves m, are inserted, the widening out of which takes place only

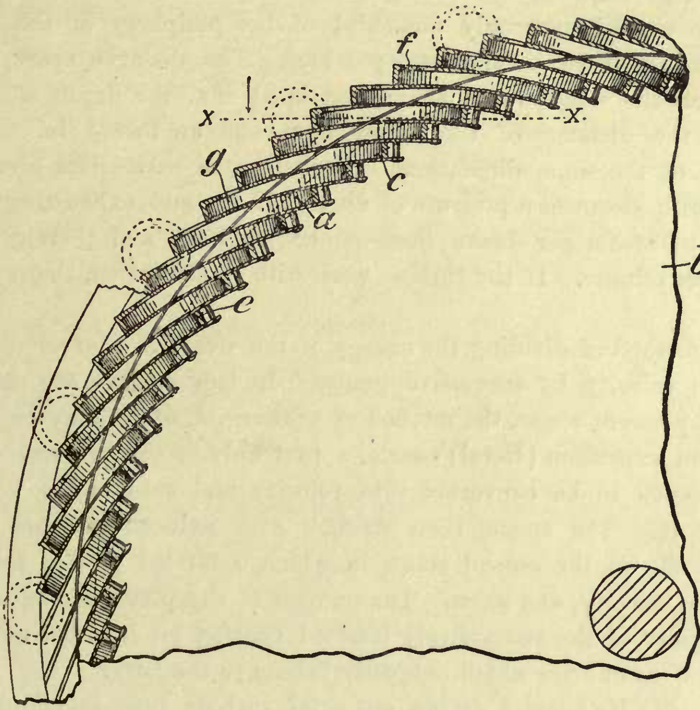


Fig. 194.

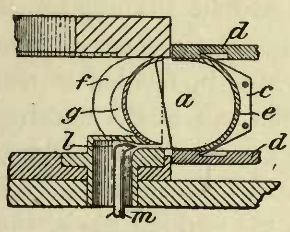


Fig. 195.

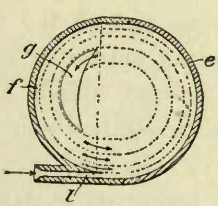


Fig. 196.

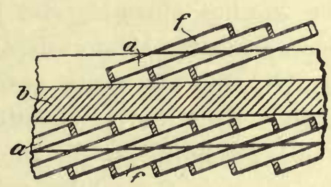


Fig. 197.

in the axial direction, so that the steam arrives at the working-wheel channel in the form of a riband corresponding with the width of the latter. The steam then passes several times through the leading channels in spiral windings, and is finally discharged through the openings g. Terry's idea is that the jet of steam arranges itself in spiral layers, as shown in Fig. 196, and thus proceeds gradually to the outlets g. This process is an unlikely one, and the method of leading the steam from the channels cannot be considered advantageous.

In Fig. 197 is shown a variation of the Terry method of construction. The working-wheel disc b is here provided with channels a, not at the periphery, but at both its sides. These channels correspond with the leading channels f, the intention being that one side of the turbine shall so work as to produce rotation to the right, and the other so as to produce rotation to the left.

A 30 H.P. experimental engine on the Terry system[1] has a working wheel 2 ft. in diameter, having seventy channels, only about one-third of which are, for the power given, at any one time traversed by the steam, so that the leading channels also extend over only one-third of the periphery of the wheel. For the impingement three nozzles are provided. The distance apart of the discs between which the wheel vanes are placed is $2\frac{1}{2}$ ins., the depth of the channels $1\frac{1}{4}$ in., and the distance of the vanes from one another 1 in. The leading channels are of the same dimensions. According to particulars given by Terry, the turbine with steam at a pressure of about 147 lbs. and exhausting into the air uses 32 lbs. of steam per brake horse-power per hour, and thereby makes 2600 revolutions per minute. If the turbine work with condensation, the revolutions rise to 3300.

Another method of dividing the energy of the steam is that of converting the pressure into velocity by successive grades. In this manner are created *velocity turbines with pressure stages*, the method of working of which may be somewhat as follows: In an expansion (Laval) nozzle, a part only of the pressure of the steam may be supposed to be converted into velocity and used in the working wheel provided for it. The steam then streams with reduced pressure towards the expansion nozzle of the second stage, in which a further part of the pressure is converted into velocity, and so on. The number of the pressure stages corresponds with the number of the successively inserted nozzles, or, in other words, with the number of sources of error which naturally belong to the latter.

Huggins and *McCallum*[2] divide an axial turbine into pressure stages. In accordance with the progressive expansion of the motive medium the number of the leading channels also increases, and at the same time the amount of the impingement. Meanwhile, the leading apparatus are so twisted in relation to one another that the steam passages describe spirals around the axis of the working wheel. Each leading channel forms a nozzle which widens out towards the working wheel belonging to it, so that in every pressure stage, in consequence of the expansion of the motive medium before its entrance into the working wheel, the latter undergoes an acceleration which destroys the increase of velocity. The effect of this is, that in every pressure stage a part of the pressure is converted into velocity. The radial heights of the leading and working-wheel channels increase continuously from the steam inlet to the steam outlet, but in such a manner that the centre lines drawn through the axis planes of the channels, which all lie one behind another, are inclined to the axis of the working wheel. The steam is thus pressed towards the shaft in a direction contrary to the centrifugal forces which are set up in the turbines.

[1] *The Iron Age*, Vol. 73, No. 11, p. 1.

[2] E. P. 23,832 of the year 1897.

A like division of the energy of the steam is also to be met with in the turbine of the *Aktiengesellschaft der Maschinenfabriken von Escher, Wyss and Co.*[1] (*Zoelly*) (1901) (Figs. 198 and 199). The outflow chamber of the one stage at the same time forms the nozzle-fed chamber or guide wreath of the next-following one. The turbine is characterized by the circumstance that the outflow chambers impart to

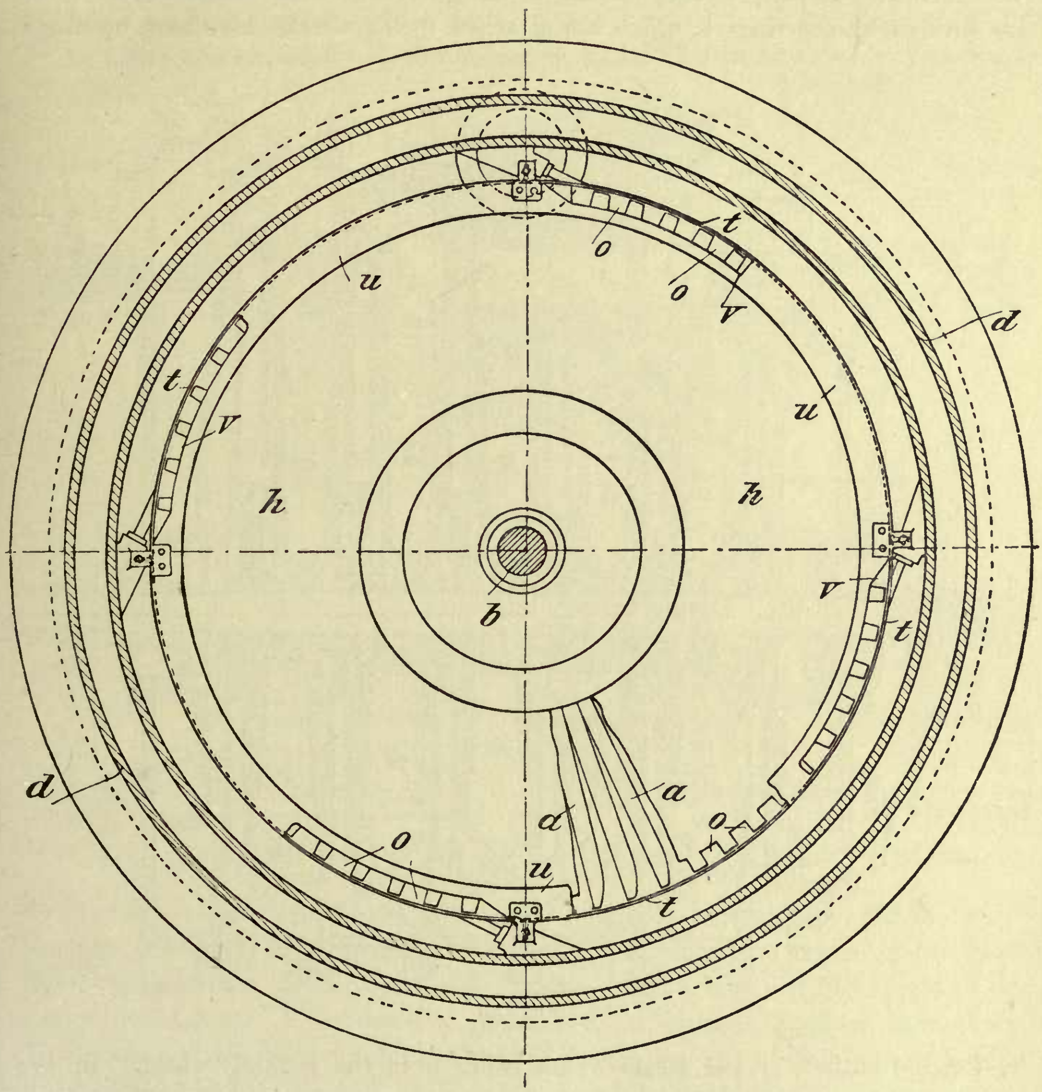


Fig. 198.

the jets of steam flowing away in different directions at both sides of the turbine wheel, the same direction of current, before they enter the next-following group of nozzles. The case of the turbine illustrated (Fig. 199 shows at its upper part a turbine with wheels that gradually increase in size, and at its lower part a turbine

[1] D. R. P. 141,492.

with wheels equal in diameter) is made up of sections in the forms of hollow castings notched into one another, which bear the nozzles for the turbines. Each two of these sections between them enclose a chamber for the accommodation of one of the turbine wheels. The radiating bars of these latter are in every case provided at each side with a disc *h*, which is of the same diameter as the turbine wheel, and has apertures *o* at its periphery for the discharge of the steam. These apertures *o* are covered by flat rings *u*, which are attached to the nozzles, and have openings

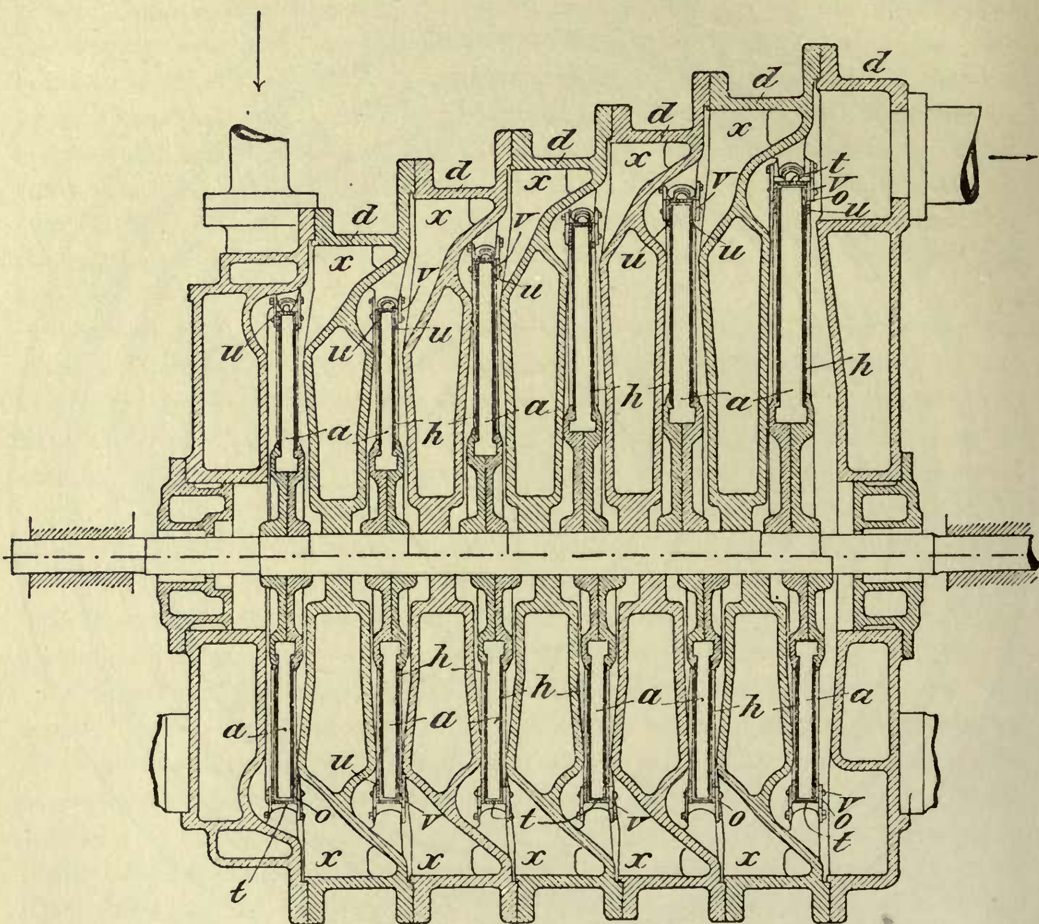


Fig. 199.

v, for the outflow of the steam which issues from the working wheel. In the arrangement under consideration, then, the jets of steam issuing from the working wheel leave the latter at both sides almost in axial, *i.e.* in opposite directions. At the periphery of the working wheel a ring *t* adjoins the latter with only a small amount of play, and covers the radiating bars over their whole breadths. Between each two nozzles a channel is thus formed, which is closed towards the working wheel, and which forms a connection between the outflow chambers on the opposite sides of the turbine wheel, so that the part of the steam issuing from the left-hand side is led through between the nozzles and joined with the other part of the steam which streams into the chamber x on the opposite side of the wheel. The two outflow chambers of a stage, which at the same time form the nozzle-supply chamber of the next-following stage, also serve to impart a common direction of current to the diametrically opposed jets of steam, before the entrance of these into the next-following group of nozzles.

In another arrangement *Zoelly*[1] combines radial with axial wheels. The steam

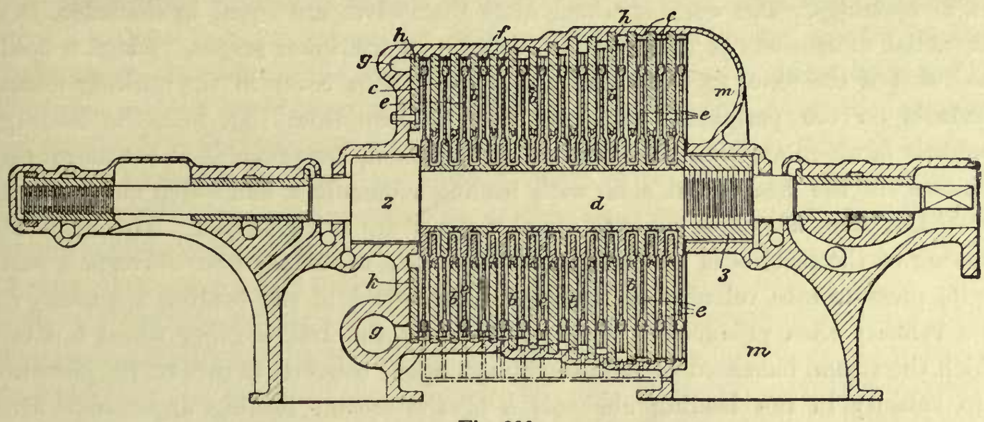


Fig. 200.

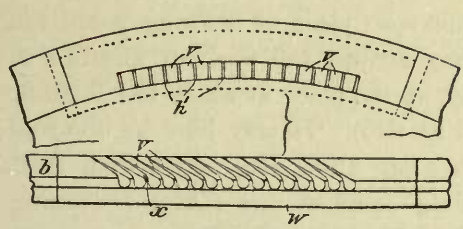


Fig. 201.

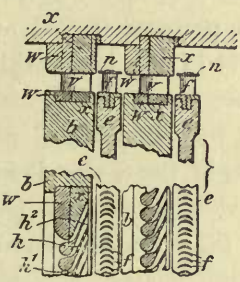


Fig. 202.

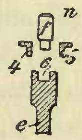


Fig. 203.

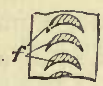


Fig. 204.

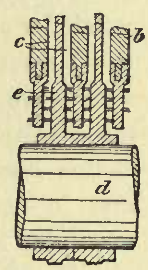


Fig. 205.

first passes through a number (3) of wheels of the first description, the discharge chamber of the one of which forms the supply channel for the nozzles of the next-following stage, and then passes on to impinge upon a series of (4) wheels of the second description. A division into pressure stages is made, however, throughout. The steam pressure is meanwhile submitted in the first set of wheels to expansion by stages in widening nozzles, while in the second set (according to the description) only a narrowing of the leading channels in the direction of the current of the steam is provided for. This arrangement may be of importance for driving the propellers of ships, in that it opposes an axial pressure to the thrust of these.

Fullagar[2] (1901) (Figs. 200 to 205) divides the case horizontally into two parts, the parting being at the height of the shaft. The case bears the annular steam-supply channel g and the outflow channel m, and is divided by fixed ring discs b

[1] E. P. 1062 of the year 1902. [2] S. P. 25,437.

into chambers *c*, into each of which a working wheel *e* is built. The working wheels forming the stages are placed one behind another between the collar 2 and the nut 3 on the shaft *d*. In the closing plate *k* on the inflow side and in the discs *b* steam-leading channels are provided, arranged in groups on separate segments. The numbers of the channels in the segment pieces decrease progressively from chamber to chamber, *i.e.* from stage to stage, until they form, in the segment pieces of the partition wall *b*, which also increase in length, a continuous, or almost continuous, row of openings. The working-wheel discs themselves are equal in diameter, but the radial dimensions of their floats *f* increase in the lower stages. Rings *n* hold the heads of the vanes together, and at the same time cover in the working-wheel channels on the peripheral side. As may be seen from Fig. 202, the leading channels *h*, which are arranged for axial impingement, are provided at the narrowest points h^1 of the steam-inlet sides with leading channels *h*, and widen out towards the outlet sides h^2. The working-wheel vanes *f* are sickle-shaped. The mode of working of the turbine is assumed to be such that the steam first converts a part of its pressure into velocity in the guide channels *h* of the leading apparatus *k*. The velocity thus produced is then taken up by the first working wheel *b*, after which the steam, increased as it is in volume, again converts a part of its pressure into velocity in the leading channels *h* of the second leading apparatus. The velocity thus won is absorbed again by the working wheel *b*, and so on, until the remainder of the steam pressure is used up in the last leading apparatus, and the remainder of the velocity in the last working wheel. In order that an overflow of steam from one chamber into another may not take place at a wrong point, the discs *b* are made tight by "labyrinth" packing against the working wheels *e* in the vicinity of the naves of these, that is to say, at the place at which the smallest motion towards either side takes place (see Fig. 205). It may here be observed that in regular work the separate chambers *c* are filled with steam from their respective stages of approximately corresponding pressure, and that the working wheels will accordingly have to contend with considerable ventilation resistances. The manner in which the guide vanes *v* are, with the help of the filling pieces *w*, *x*, collected together into the segments, and these again inserted in the discs *b*, is to be seen from Fig. 202. The working-wheel vanes *f* are let into the rings 4 and 5 (Fig. 203), and these again are pressed into the annular grooves 6 of the discs *e*. In order to lead off any condensed water which may collect, either the discs *b* are provided at their lowest points with horizontal boreholes, through which the water flowing from chamber to chamber can reach the channel *m*, or each chamber is connected by means of a special drain pipe with a channel outside the case.

A *combination* of the *pressure stages* with *velocity stages* is also clearly possible. The steam is then subject in the first leading apparatus to a certain drop in pressure, which is greater than in the case in which only one velocity stage is in question. It then impinges on the same working wheel, or on several working wheels, the number of such impingements corresponding with the number of the velocity stages, and streams on with reduced pressure to the next group of expansion nozzles, and so on. An arrangement of this kind offers advantages in certain cases, as for instance where turbines which alternately exhaust and condense their steam are in question. For if, in the latter case, two pressure stages be arranged for, the first with a drop to the pressure of the atmosphere, and the second with a further drop to that of the condenser, the second pressure stage can, in the transition from work with the condenser to that with the exhaust, be shut off without the conditions required for the velocity stages of the first pressure stage being altered thereby. The combination referred to appears, indeed, to be the most promising development of the velocity turbine, because in it the number of expansion nozzles required becomes less than that in the engines which work with pressure stages only, and the importance of the losses of energy which take place in velocity stages is diminished. In practice it is the custom to divide every pressure stage into two velocity stages.

A combination of pressure and velocity stages of this kind has been chosen by *Curtis*[1] (1897) also. As an example of this an arrangement with two pressure stages is given in Fig. 206. A part of the steam pressure is converted into kinetic energy in the nozzle *c*, and the remainder in the nozzle *d*. In front of the working wheel *a* with its two velocity stages is inserted the nozzle *c*, and in front of the similarly formed working wheel *b* is set the nozzle *d*. Both working wheels are placed on the same shaft; they might, however, be arranged to drive two shafts. In the assumed case the wheel *a* is intended to do twice as much work, and consequently to take up twice as much energy as the wheel *b*. In the nozzle *c*, therefore, two-thirds of the available drop in pressure is converted into kinetic energy, and in the nozzle *d* only one-third. For the wheel *a* steam and circumferential velocities twice as great as for the wheel *b* are obtained. If, then, the revolutions per minute are the same in both cases, wheel *a* must have twice the diameter of wheel *b*. The nozzles and the walls of the chambers, it may be observed, are here coated with insulating material.

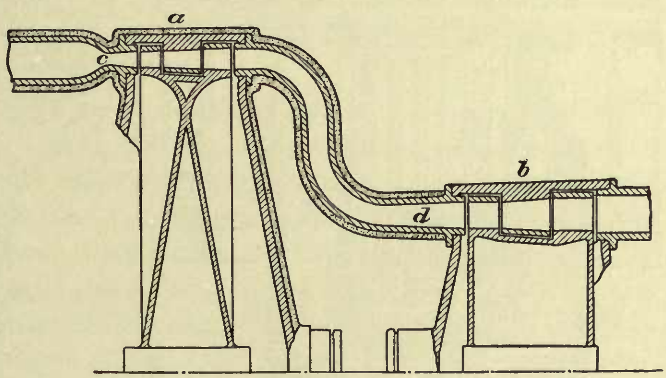


Fig. 206.

The axial turbine designed by *Shepard*[2] (1903) (Figs. 207 and 208), the two pressure stages of which are subdivided into equal numbers of velocity stages, has the peculiarity that the same set of working wheels with the leading apparatus between them are used for both pressure stages. The illustrations show two sections through the turbine at right angles to one another. The fresh steam makes its way through the branch 27 into an annular section 26, from which the 7 expansion

[1] A. P. 566,969. [2] A. P. 741,940.

nozzles 28 of the high-pressure stage are fed. The motive medium then streams through the working wheels 2, 3, 4, and 5, and the corresponding leading apparatus 23, 24, and 25 between them into the chamber 32. From the latter a pipe 34 leads to the chamber 35, from which the 14 expansion nozzles 36 of the second pressure stage take their steam. After another impingement of the working wheels the steam passes, shorn as far as possible of its energy, into the chamber 40, 41,

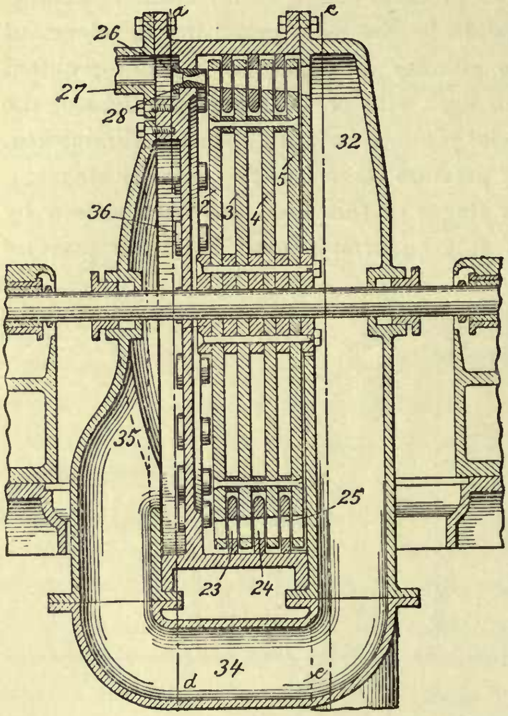


Fig. 207.

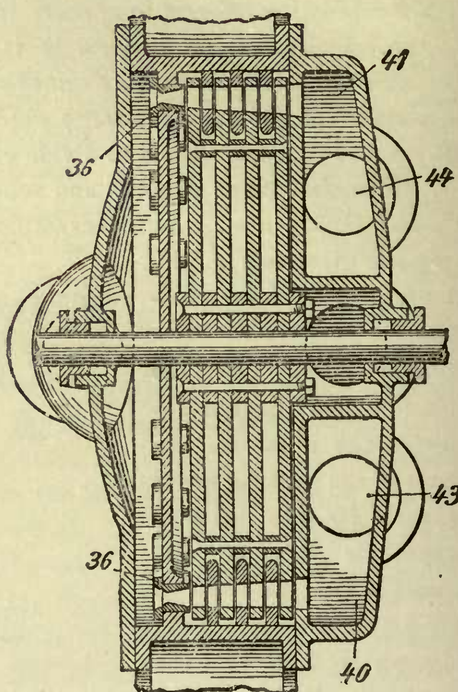


Fig. 208.

from which two outlets 43 and 44 at the sides of the overflow pipe 34 lead to the condenser or into the atmosphere. It will be seen that the radial breadths of the guide and working-wheel vanes increase in the direction of flow of the steam, the object of this being that in the passage of the steam through the working wheel and leading channels a small portion of its pressure may also be converted into velocity. Meanwhile the velocity produced by the expansion in the channels is intended to be used up by the working wheels at the end of every pressure stage, *i.e.* in the last velocity stage in each case. In order that the degree of increase of section of the channels may at once be suitable, the expansion of the steam in the nozzles 36 is to be carried further than in the nozzles 26.

III

COMBINATIONS OF PRESSURE AND VELOCITY TURBINES

PROPERLY speaking, every turbine combines the quantitative action of the steam with its back pressure. The attempt has, however, been made to construct turbines in such a manner that the two kinds of force are exerted in separate periods. These endeavours have been assisted by the faculty possessed by the motive medium of expanding and contracting. Combinations are thus obtained of the pressure turbine with the velocity turbine.[1]

When, after the ejection of the jet of steam, the cell of an *impulse wheel* is at once closed against the steam inlet on the side of the vane lying exactly opposite to the steam nozzle, before the inflowing steam has time to fill it, and a back pressure is thus set up against the walls of the remaining cells, a quantitative motive effect remains. The steam (now become neutral) remaining in the cell is carried along by the current until the opening in the cell comes opposite to the outlet. Then follows an expansion of the steam from the cell into the air, and thus a reaction. In most arrangements of the kind, indeed, it is very questionable whether a motive force is thus produced. No practical importance can be attached to them. Designers, however, whose work must be taken seriously, have down to the most recent times attempted to make the impulse wheel a successful construction.

Thus *Whele*[2] (Fig. 209) has covered the wheel *a* over only about one-third of its periphery with a ring *b*. The wheel cells *c* are so fashioned that, when they lie completely open to the steam inlet pipe *d*, the steam has the tendency to push the walls *e* forward, like pistons. The cells remain filled after the steam inlet *d* has passed by, until the lower ring *b* becomes free, after which the steam exhausts, and thus performs work by reaction.

The design of *Bergen*[3] (1877) (Fig. 210) is founded upon the same idea. The wheel *a* and the case *b* together form chambers. The steam entering through the pipe *c* first works by means of impulse on the vanes of the wheel. These then carry round with them fixed quantities of steam. When a chamber arrives at the outlet the steam escapes through the latter, in so doing still performing work by means of back pressure. Moreover, the case *b* contains two chambers, which are

[1] For the sake of simplicity pressure and back-pressure are placed in antithesis to one another.

[2] E. P. 149 of the year 1852.

[3] D. R. P. 2044.

separated from one another by a wall, the one containing a wheel for right-hand motion, and the other a wheel for left-hand motion. The reversing cock *e* then opens the steam inlet to the one or the other chamber, as the case may be. In order that the impulsive force of the steam may be efficiently utilized, the surfaces of the vanes are made of special form.[1]

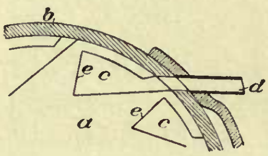


Fig. 209.

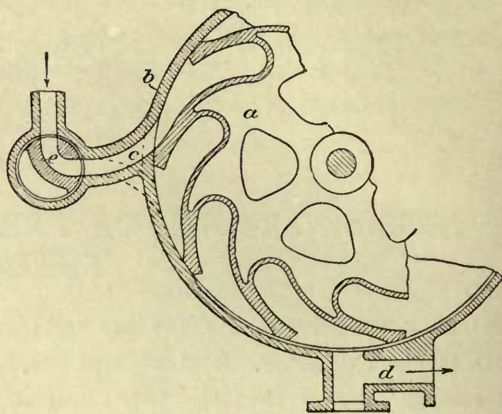


Fig. 210.

The wheel designed by *Yates* and *Bellis*[2] (Fig. 211) contains cells, the sides *a* of which are so set that, when passing the single steam-leading channels *b*, which can be shut off, they take the same direction as these. The sides *c* meanwhile run nearly in a radial direction. The cheek *d* lying close against the wheel is provided with perforations *e* opening into the chamber *f*, which in its turn is open to the exhaust chamber *g*. The steam proceeding through the channels *b* into the cells of the wheel performs work first by impulse, and then by back pressure, and escapes, partly through the perforations *e* into the chamber *f*, and partly into the chamber *g*.

Hoffbauer[3] (Fig. 212) constructs each working-wheel cell of two chambers, the

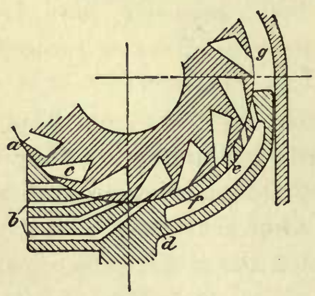


Fig. 211.

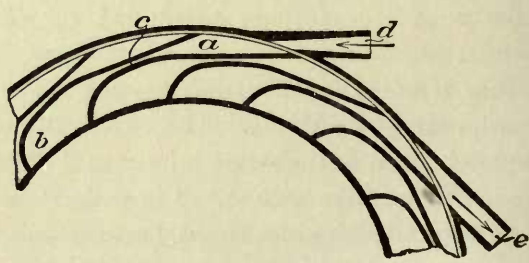


Fig. 212.

foremost of which *a* is connected with the after one *b* by means of a narrow channel *c*. As soon as a chamber *a* becomes open to the steam nozzle *d* steam streams in, which first performs work by direct pressure, and fills the chambers *a* and *b* while the former is passing the nozzle *d*. Then the steam collected in the two chambers *a* and *b* expands, and now works by reaction. The conditions

[1] D. R. P. 3127.
[2] E. P. 9220 of the year 1898.
[3] E. P. 25,144 of the year 1901.

are so adjusted that this reaction work lasts till the cell in question reaches the exhaust *e*.

Hakansson's wheel,[1] which will often be alluded to in the following pages, is arranged for axial impingement and for rotation in both directions, but is in other respects similar in its way of working to the ones above discussed.

As in the case of the impulse wheels, in which the inlet and the outlet for the steam are arranged at the same place, so amongst the wheels provided with grades, a considerable number of designs are to be found which aim at a combination of the manner of working of the steam in the piston engine with that in the turbine, but leave out of account the actions of pressure and back-pressure, which in such an arrangement counteract each other. To these belong the arrangements in which the utilization of the energy by successive grades is effected by an impulse wheel.

Thus, according to *Gray*,[2] the jet of steam in the first place acts on the first vane, which is situated just in front of the nozzle. The first cell, after being filled with steam, communicates with a channel in the wall of the case. Its steam is enabled by this channel to expand in the next cell in advance in the direction of rotation, and so on.

Durr[3] (1901) (Fig. 213) attempts the solution of the problem in a somewhat different, but not very successful, manner. On the inner peripheral surface of the case, and on the outer peripheral surface of a driving wheel contained by the latter, shell-formed working chambers are arranged, pointing almost in the tangential direction. The steam is led, at regular intervals only, to the shell chambers of the driving wheel, which are intermittently opened and closed to the shell chambers of the case, the object being to suffer only a few of the driving-wheel chambers to fill with fresh steam, and thus to make the most of the expansive force of the latter. It is said that this can be effected by the spacing of the shell chambers in the driving wheel more widely or more closely than those in the casing. If the shell chambers in the case and in the driving wheel are spaced at the same distances apart, the cover sections are made larger than the chambers. If the motive medium be admitted through the inlet channel *k* (which in the turbine illustrated takes place once during each rotation), it first passes into the driving-wheel shell chamber *g*, which is situated

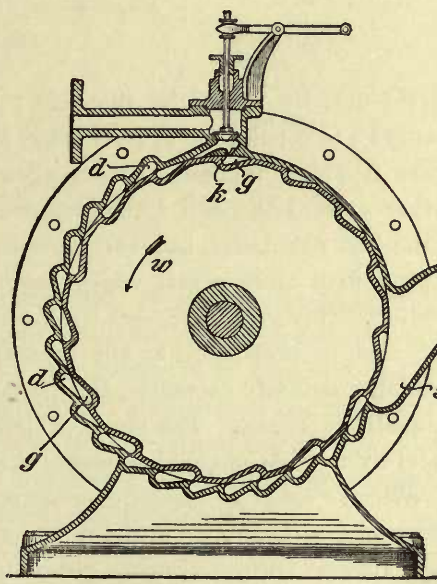


Fig. 213.

[1] E. P. 15,069 of the year 1897. [2] E. P. 15,600 of the year 1900.
[3] D. R. P. 137,128.

below the channel. When the driving wheel turns in the direction *w*, the chamber *g*, then filled with steam, obtains communication with the first chamber *d*, so that the steam enclosed in *g* passes over into the chamber *d*. Here it expands and, by reason of its expansion, exerts an almost tangential backward impulse on the driving wheel, by which the latter is impelled further onward in the direction of the arrow *w*. This is repeated in each shell chamber *d* till the motive medium at last reaches the limit of its expansibility and escapes to the atmosphere at *s*.

Tilp[1] (1899) leads the steam, after its first performance of work in the working wheel, to a point *in front of* the inlet. The cells of the working wheel, however, are closed, so that the steam on leaving these is able to perform work by reaction. In Fig. 214 a radial turbine is illustrated. The steam proceeding from the inlet *g* through the leading channel *h* into the working wheel *b* enters the cell *l* of the latter and performs work in it. As soon as the rotation of the working wheel moves the cell *l* onwards, the latter meets with the opening *n* of the first (inmost) arched return-lead channel *e*. Into this opening the expanding steam carried over by one of the channels *e* now enters, and is led back through the next channel *e* to a point in the working wheel situated *in front of* the inlet opening, where it again proceeds to act upon the working wheel. As the wheel moves farther onward the cell *l*, the pressure in which, by reason of the surrender of steam in the first channel *e*, had become reduced, arrives at the second channel *e*. Here it parts with more steam, which again is led back to a point in front of the inlet opening. This process repeats itself each time that the cell *l* reaches another channel *e*, so that in proportion as the pressure of the steam in the cells *behind* the inlet opening gradually decreases, the pressure in the cells *in front of* the inlet opening gradually increases. The steam which is not taken from the cells of the working wheel by the last channel *e* escapes through the discharge *m*.

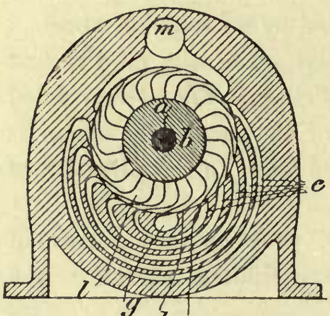


Fig. 214.

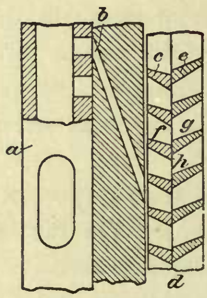


Fig. 215.

Nilsson[2] seeks to separate pressure from back pressure in an axial turbine by other means (Fig. 215). From the ring *a*, which for purposes of adjustment is made free to turn, the steam passes into the expansion channels *b*, and then acts directly on the vanes *c* of the working wheel *d*. These vanes *c*, which bound the supply channels of the wheel, and the vanes *e*, which are situated on the outlet side, are so placed and fashioned that the clear entrance into the wheel cells *f* is greater than the steam discharge from the wheel cells *g*, and they are also provided with the projections *h*. The effect of this arrangement is that the steam passing into the cells *g* immediately on entering these undergoes a sudden expansion, and then

[1] D. R. P. 111,278. [2] E. P. 23,759 of the year 1899.

continues to expand further as the cells g increase in section. This is intended to produce an intensified reaction, which, in conjunction with the pressure on the vanes c, furnishes the driving power of the wheel.

One of the most remarkable among these combination turbines is that of *Höltring*[1] (1899) (Figs. 216 to 219). Here also the intention is to combine action with reaction in a single wheel, but it is to be done in such a manner that the fall in pressure is regulated by the turbine itself, in order that, in comparison with wheels working by action or reaction, only the speed of rotation can be very considerably reduced. The motive medium effects its entrance from points at the outer periphery of the working wheel into the latter in a direction parallel to a plane

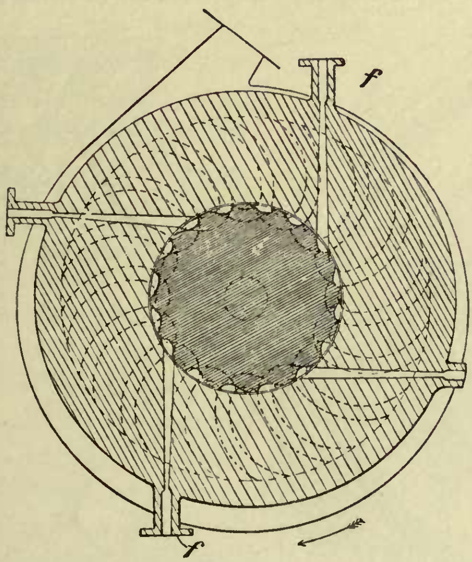


Fig. 216.

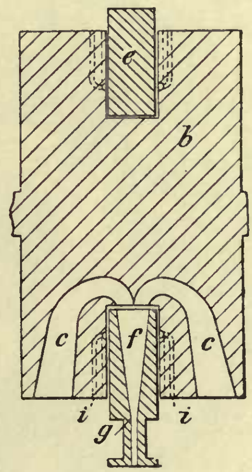


Fig. 217.

standing at right angles to the axis of the turbine. It streams through curved channels c and proceeds in a direction parallel with the plane through the inlet passage to the outside, where it discharges at a point situated at a certain distance from the inlet. The leading nozzles f are arranged in a ring-shaped casting e, which is placed in a correspondingly ring-shaped groove in the body of the turbine. The circular or cone-shaped surfaces of this groove are so near the corresponding surfaces of the nozzle ring that they serve as a sufficient packing for the clearance space at the point of inlet. The leading nozzles f are narrowest at the points g, and widen out in the inward direction in such a manner that their discharge ends are of the same section as the entrances of the channels c. An uncontracted entrance of the motive medium into the turbine channels is thus rendered possible, while the whole section of the leading nozzles f increases in such a manner that the desired degree of expansion is attained. In like manner the turbine channels c widen out towards their points of discharge to such a degree that they enable the

[1] D. R. P. 116,494.

complete expansion of the motive medium to take place in these channels, and the turbine to perform work by means of reaction. Meanwhile open groove-like channels *i*, curved like the turbine channels, can be provided in the circular or conical packing surfaces of the clearance opening. These enable the energy of the motive medium which escapes at the clearance opening to be utilized as far as possible in driving the turbine. Otherwise, in place of the groove-like channels, small closed turbine channels *o*, in connection with an annular groove-like collecting channel *h* (Fig. 218), are provided for the leading away of the steam that escapes at the clearance opening to a further utilization of its energy. This illustration also shows that the inner portion of the channel *c* may be made larger than the outer portion. Clearly, the two portions may also be made of equal length.

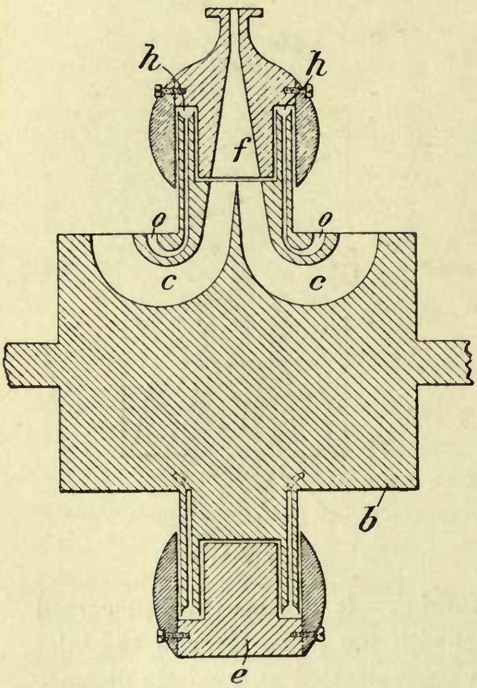


Fig. 218.

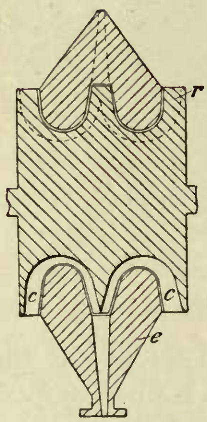


Fig. 219.

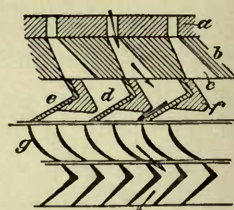


Fig. 220.

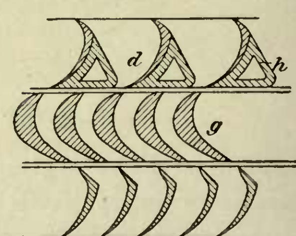


Fig. 221.

Finally, to enable the engine to be easily constructed, it may be so arranged [1] that the turbine channels *c* (Fig. 219) are worked into flutings in the periphery of the working wheel and take the form of open grooves separated by ribs *r*. They are then bounded at their outer sides by fixed nozzle-bearing castings *e* projecting into the flutings and serving to separate the inlets of the channels from the outlets of these.

Scott, Tyzack, and Summerfield,[2] on the other hand, go to work as follows (Fig. 220). The distributing disc *a* is adjoined by the fixed disc *b*, with its expansion nozzles *c*, from which the steam streams into the cells *d* of the working wheel *e*. These cells are contracted by means of inserted pieces *f*, so that the steam, first expanding in the cells *d*, suffers compression in front of the contractions, and

[1] D. R. P. 137,432. [2] E. P. 22,740 of the year 1902.

then enters the guide wheel *g* on its way to the next-following working wheel, and so on. In Fig. 221 the contractions are produced by the insertion of light, hollow, triangular-sectioned pieces placed close to the points of discharge. These inserted pieces have no sharp corners. They produce a choking and retarding effect.

Thormeyer[1] also (1902) seeks to produce action and reaction separately. His method of construction, shown in Fig. 222, is said to render possible the adoption of a reduced speed of rotation of the wheel. The steam introduced into the steam chamber E impinges, through the nozzles *e*, on the vanes of the vane wreath *a* of the working wheel, which turns with the shaft in the case A. The vanes of the wreath *a*, situated in front of the nozzle openings *e*, are, during the impingement, closed by the unperforated parts *f* of the leading apparatus *c*, so that the steam filling their hollow spaces cannot directly escape. After the nozzles have passed by, the vanes of the wreath *a*, which are filled with steam, are closed on the impingement side by the walls of the steam chambers E. In consequence, the compressed steam enclosed by the vanes is enabled to give forth its energy by reaction, without being subject to the influence of the steam nozzles. The outer vane wreath *b* is impinged on through the leading apparatus, and in it the last remainder of pressure is utilized by reaction. The resistance which the outflowing steam meets with in the channels when closed during the impingement is said to have an effect in reducing the speed of rotation of the wheel.

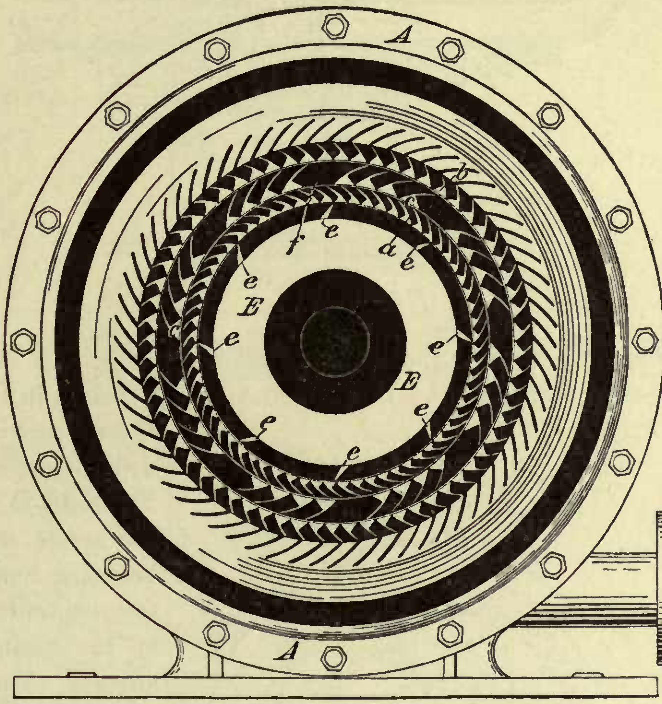


Fig. 222.

If it were proposed to perform work first by reaction and then by action on one and the same wheel, a method might be adopted like that of *Holland*[2] (1893) (Figs. 223 and 224), whose working wheel D, in a case C made in two parts, is made tight by the packing rings *h* and *n*. The wheel D has channels *c*. Each portion of the case contains pockets *d*, which open into the nearly semi-annular channel E. Adjoining this, and completing the ring, is the narrower channel *e*, which is open to the side vanes *f* of the wheel D, and delivers into the exhaust *g*. The steam proceeds

[1] D. R. P. 144,864. [2] A. P. 521,541.

through the slots a of the hollow shaft, or through the branch b, into the inner chamber of the working wheel D, expands first through the reaction channels c of the latter, and then from the pockets d into the wide channels E, whence it proceeds to stream through the channels e to the outlets g. In its course it takes the vanes f of the wheel D along with it.

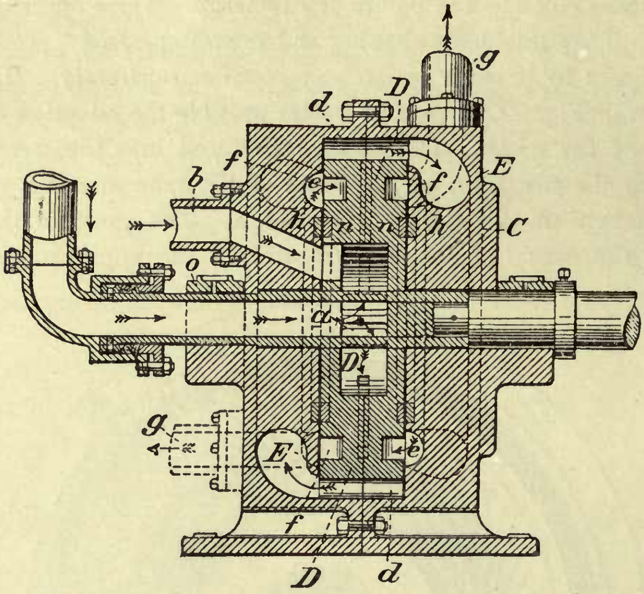


Fig. 223.

Morton[1] adopts an arrangement of pressure stages, but combines in each of the latter a pressure (reaction) wheel with velocity wheels. In the arrangement shown by Figs. 225 and 226 there are three working wheels in as many separate cases. Fig. 225 shows on the left a section through a, b of Fig. 226, and on the right a section through c, d. In the case A, which is made in two parts, the partition walls B form part of the castings, and together with the covers C form the three chambers. The shaft D runs in bearings E, held by the covers and walls of the case. The bearings, however, are so constructed as to leave central openings for the passage of the steam from one chamber into another. The three working wheels are fixed upon the shaft. A radial section through the first of them is shown in Fig. 226, in which a flat middle disc G and the conical side discs G^1 and G^2 are to be seen. At the periphery of the disc G there are at the one side the ringlike pieces H between which the channels I are left free, and at the other side the ringlike pieces H^1 between which lie the channels I^1. Further, there are the vanes M. Between these and the ring-pieces I the reversing and guide vanes L are inserted, which are borne by the further fixed disc K. The disc G thus forms what is really the

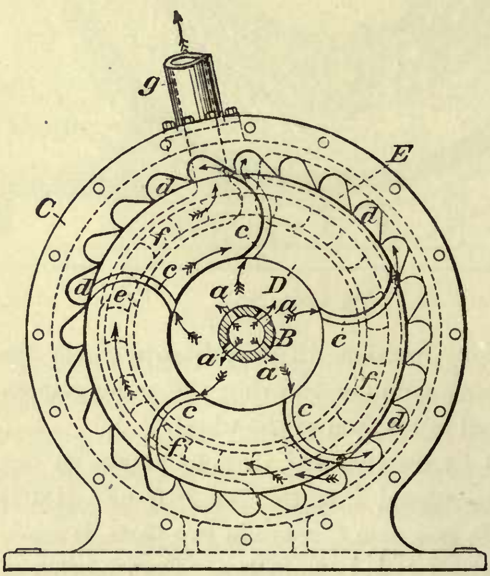


Fig. 224.

[1] E. P. 5820 of the year 1891.

working-wheel body, while the side discs G^1 and G^2, which rotate with it, serve to bound the passage ways of the steam. The steam, entering through the branch O, first proceeds in the axial direction into the space between the discs G and G^1, and then in the radial direction outwards. In passing through the channels I it performs work by reaction, losing a part of its pressure in so doing, and then repeats its reaction work in streaming through the channels I^1. Finally, it is led by the guide vanes L to the further performance of mass work on the vanes M. After this triple loss of energy the steam streams radially inwards, through between the discs G, G^2, on to the second and third working wheels. In these the process repeats itself until the steam escapes

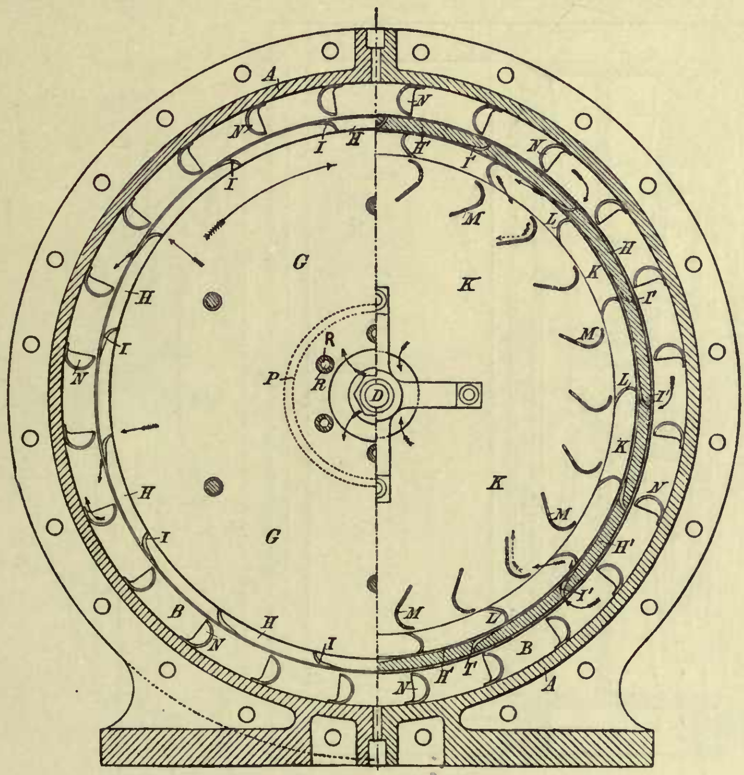


Fig. 225.

with exhaust or condenser pressure. The reaction channels I, I^1 gradually widen in the direction of flow of the steam, the result of which must be a corresponding acceleration of the current caused by expansion in the channels. Moreover, the sectional area of clear passage way decreases from stage to stage and from wheel to wheel. In order to promote the flow of the steam leaving the channels I into the channels I^1, and to prevent its rotation round the wheel, the case is provided with the fixed vanes N. Attention may here be directed to the relieving appliance, intended to neutralize the axial thrust of the steam, which is delivered in one direction only. To this end the ring P, which is pressed with a spring against the disc G^1, is made larger in diameter than the ring Q, which fits against the disc

G^2. The effect of this is that the steam pressure in the chamber bears against a larger part of the surface of the disc G^2 than of that of the disc G^1. The difference corresponds with the excess pressure of the steam on the working-wheel body G in the left-hand direction acting in opposition to the pressure in the right-hand direction, *i.e.* after the drop in pressure has taken place. Undoubtedly, any steam which presses in at the ring P will be carried away by the small pipe R into the chamber with the lower pressure. The cock S serves for the regulation of the supply of water for the stuffing-box packing.

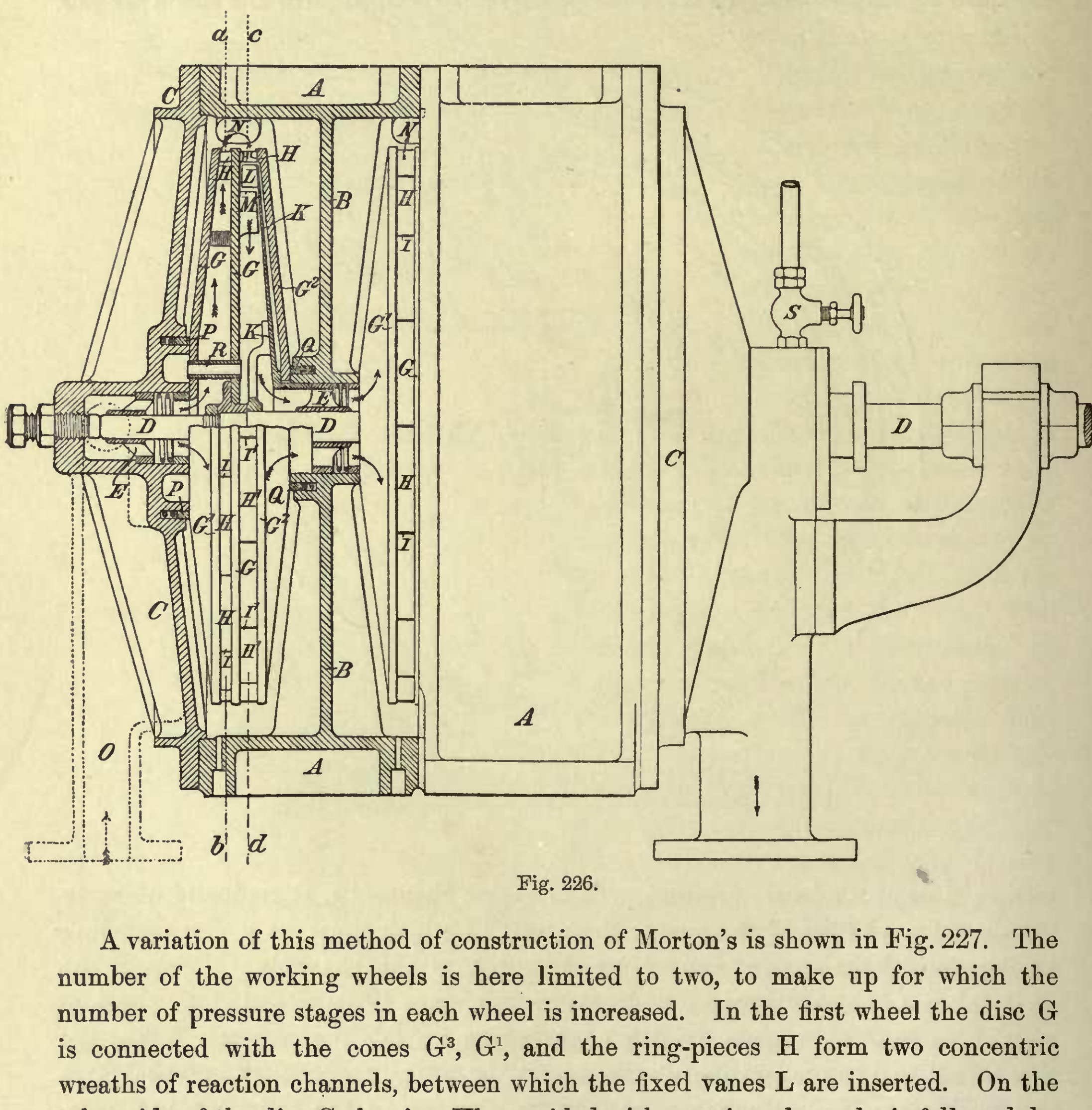


Fig. 226.

A variation of this method of construction of Morton's is shown in Fig. 227. The number of the working wheels is here limited to two, to make up for which the number of pressure stages in each wheel is increased. In the first wheel the disc G is connected with the cones G^3, G^1, and the ring-pieces H form two concentric wreaths of reaction channels, between which the fixed vanes L are inserted. On the other side of the disc G the ring H^1, provided with reaction channels, is followed by the vane wreath M for action work. The second wheel has only three conaxial rings for back pressure, which are built in between the rotating discs G^3, G^4, G^1, and G, and impinged on by the steam in the radial direction from within. The plates N fixed to the case A here only serve the purpose of preventing the exhaust steam from coursing round the wheel. The first wheel, again, requires the rings P in order to neutralize the excess pressure of the steam acting in the axial direction. Here, however, the chamber covered in by the ring is in connection with the atmosphere. On the second working wheel, on the other hand, no such unbalanced pressure in one direction is exerted by the steam.

A system of construction, the working of which is very obscure, is given by *G. Westinghouse*[1] (Fig. 228). The arrangement contemplated is that of a double turbine. The steam is introduced into the annular chambers 6, from which it passes into the portable expansion nozzles inserted in the annular discs 11. It then

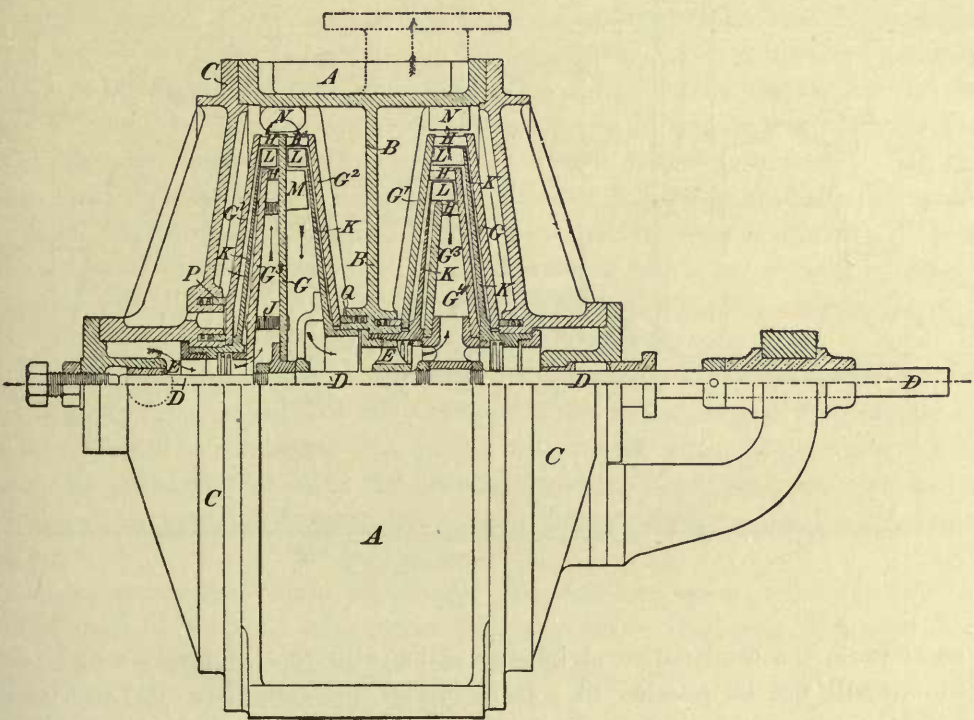


Fig. 227.

impinges on several velocity stages of the wheel wreaths, which, with their widening radial vanes 2, are worked on the drum 1. In this the steam brushes over the heating-coils 4, and takes up heat. Thereupon the steam has to proceed into the working-wheel wreaths 26, the channels of which present narrower passage ways than do those of the wreaths 2. To the wreaths 26 succeed the working-wheel wreaths 3, which are larger in diameter. The channels belonging to these increase in the radial direction from stage to stage, and this in a greater degree than do those of wreath 26. The exhaust steam flows through chamber 40 towards the discharge. The circulating channel 17 is intended to neutralize the pressure in the chambers 16. In consequence of the heat imparted by the

[1] E. P. 11,045 of the year 1902.

coils 4, and of the contracted entrance into the set of wheels 26, the steam that has just been robbed of its velocity evidently experiences an increase in pressure, which is made use of in the next-following pressure stages. From this combination no advantage is to be derived. Further, that a compact arrangement is thereby attained, as imagined by Westinghouse, does not appear.

If the principle be kept in view that concussions of the motive medium entail

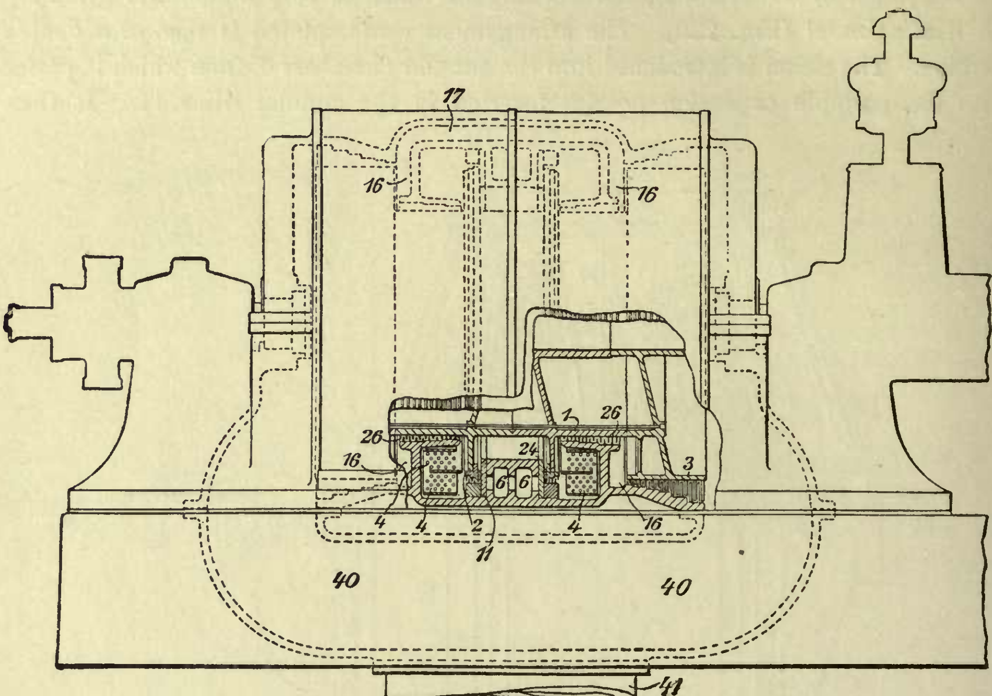


Fig. 228.

loss of work, the combination of impulse action with that of pressure and back-pressure will not be resorted to. It is further the case that every conversion of the one form of energy into the other is combined with a loss of useful work, which in some cases is of very considerable amount. The conversion of pressure into velocity, no less than that of velocity into pressure, is a source of unavoidable losses. So far as economy of working, however, is concerned, this amounts to the passing of sentence on all those methods of turbine construction in which suchlike conversions alternate with one another.

IV

COUNTER-DIRECTION WHEELS

If the guide wheels, instead of being fixed, be made free to turn, an arrangement of *counter-direction wheels* is obtained, in which one working wheel forms the impingement apparatus for the next-following one, which revolves in a direction contrary to that of the first. Amongst other advantages consequent on the abolition of the fixed leading apparatus is a diminution of resistances in the leading of the steam, which are then confined to the parts which have to take over its energy. On the other hand, for the greatest development of power, and under conditions otherwise the same, the speed of rotation required for each member of a pair of wheels running in contrary directions is only half that of a wheel with fixed leading apparatus. Practical difficulties, however, arise when the attempt is made to impart the work of a pair of wheels to one shaft, or to transmit it in one direction.

It is, of course, possible to couple together pressure and velocity wheels in pairs, and to apply these, or pairs of the former or latter, to the purpose just referred to. In this it has to be considered that, for the complete utilization of the energy of the steam, the speed of rotation of the pressure (reaction) wheel must be equal to the velocity of the steam, and that of the velocity (action) wheel to half the velocity of the steam.

In the arrangement made by *Gilman*[1] (Fig. 229) the steam is led through the hollow shaft to the arms a (one arm only is shown). The steam streaming from the nozzle c is not intended to give all its energy to reaction, *i.e.* to the driving of the wheel b; a part of it is to be applied to the efficient impingement of the vanes d of the wheel e. The vanes have square or triangular forms.

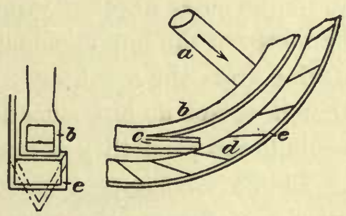


Fig. 229.

A similar arrangement, in which, however, the reaction nozzles are drawn out almost to the breadth of the turbine vanes, was adopted by *Messinger*[2] (1899).

On each of the arms a of the reaction (pressure) wheel, which are curved in the direction of the shaft, *Andrews*[3] (Figs. 230 and 231) has arranged a special outlet nozzle, pointing almost in a tangential direction, and having a cylindrical piece c following a slight contraction. The cylinder is followed by the piece d, which bounds

[1] E. P. 7417 of the year 1837. [2] A. P. 644,621

[3] E. P. 13,598 of the year 1851.

the issuing jet on one side only. Together with this reaction wheel works a second wheel bearing vanes. Fig. 230 shows one of these vanes in longitudinal section, while Fig. 231 shows it in elevation at right angles to the axis of rotation. The vanes are made fast between two rings, the motions of which are also imparted to

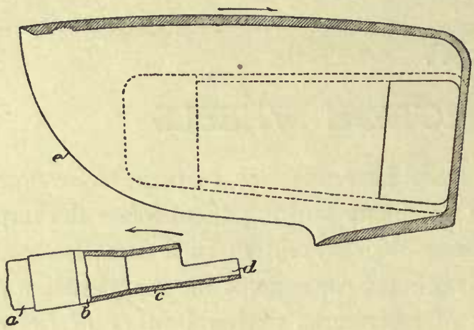


Fig. 230.

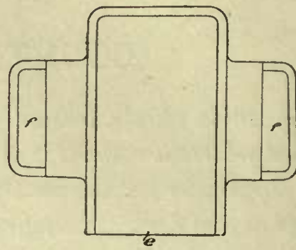


Fig. 231.

the shaft. The steam streaming from the reaction nozzle *b* passes into the vanes *e* just opposite, and presses against their radially arranged back wall, whereupon it parts asunder and escapes through the side channels *f*. The vane *e*, then, will be turned in the direction of the arrow, first by the pressure of the jet of steam issuing from the nozzle *b* against the back of the blade *e*, and then by the back pressure of the steam as it streams from the nozzle again. In a typical case, twenty-four vanes *e* were provided for four arms *a* of the reaction wheel.

The combination of *Jourdanet* and *Gauthier*[1] is illustrated in Fig. 232. The few outlet nozzles *a* of the inner wheel pass out in a direction as nearly as possible tangential, while the more numerous vanes *b* of the outer wheel are set tangential to the inner wheel, but curving outwards into the radial direction. The inner wheel is made to revolve in the one direction by reaction, while the outer wheel is turned in the other direction by means of the impingement of its vanes by the steam issuing from the nozzles *a*. The steam escapes from the vanes *b* in such a manner that it causes no hindrance to any of the revolving wheels. Incidental allusion is made to the possibility of throwing one or other of the wheels out of gear.

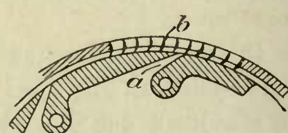


Fig. 232.

In the arrangement made by *Baker*[2] (Figs. 233 and 234), amongst others, the arms *a* of the reaction wheel are divided into pairs of chambers by the partition walls *c*. On the annular flange *b* they are fitted with the vanes *d* and *e*, which are curved in opposite directions. The chambers *f* lead into the reaction nozzles *g*, and the chambers *h* into the reaction nozzles *k*. The case *l*, which is likewise free to turn, contains, on its part, the groups of vanes *m* and *n*, which are curved in opposite directions. If the sleeve *o*, which can be moved in the axial direction, be so adjusted that the steam obtains access through the holes *p* to the arm chambers *f*, reaction

[1] E. P. 26,650 of the year 1897. [2] E. P. 6768 of the year 1899.

wheel and case will turn in the direction of the arrows. Meanwhile the vanes *m* guide the steam from the nozzles once more against the vanes *d*, to which the latter are fixed, thus producing a combination of the actions of pressure and velocity in one and the same wheel. If the sleeve *o* be so adjusted that the chambers *h* are

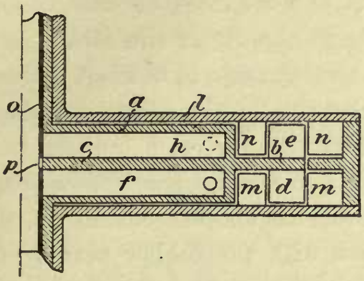


Fig. 233.

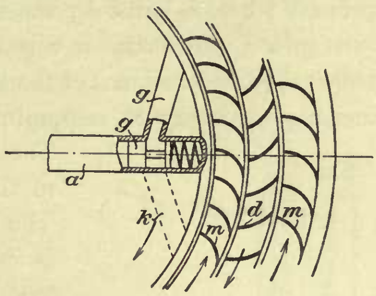


Fig. 234.

impinged upon, wheel and case will rotate in opposite directions. The reaction arms *a* are provided with spring slides *g*, which, after a certain circumferential velocity has been exceeded, contract the steam outlets more and more.

An essentially axial flow of the steam is to be seen in the design made by *Martindale*.[1] In this case also, in order further to utilize the energy of the outflowing steam, the reaction wheel running on the horizontal axis is combined with further axial turbine wheels, the first of which is impinged upon from the nozzles of the reaction wheel, and itself passes on the steam to the next-following wheel. The reaction wheel and the turbine wheel impinged upon from it turn in opposite directions, while the working wheel again moves in the same direction as the reaction wheel, and so on. The latter has the greatest diameter, and for the working wheels this dimension becomes gradually smaller in the direction of flow of the steam, so that the case takes the form of a cone, with its point towards the end at which the steam is discharged.

For axial turbines, the combination adopted by *Lohmann*[2] is also made use of (Fig. 235). The wheels *a* and *b* rotate on vertical axes. The upper wheel *a* has the vanes *c*, the lower one *b*, the vanes *d* turned in the opposite direction to the former. The size of the latter is so chosen that for each vane *d* there are two corresponding vanes *c*. Each of the vanes *d* has a cavity *e* with an opening pointing towards the upper wheel *a*. The steam is introduced into a central chamber below the wheel *b*, from which radial tubes lead it on into the cavities *e*. The steam issuing from these turns the wheel *a*, by means of pressure on the vanes *c*, in the one direction, and the wheel *b*, by means of reaction, in the other direction. The steam on leaving the wheel *a* is drawn off by a pipe fitted on the case of the turbine, and the effect is said to be thereby increased.

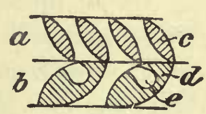


Fig. 235.

Parsons[3] (1902) also has recently proposed the adoption of counter-direction

[1] E. P. 2123 of the year 1897. [2] E. P. 16,635 of the year 1897.
[3] A. P. 729,215. E. P. 6142 of the year 1902.

wheels as a means of reducing the number of revolutions. In this, however, he assumes that the pressure of the steam is converted into velocity in expansion nozzles. These nozzles at the same time form the reaction nozzles of the first wheel. An arrangement embodying this idea is shown in Fig. 236. In the latter the primary wheel *b*, borne by the shaft *a*, consists of plate discs, between which, in the first place, the chamber *c* is formed. Through the latter the steam issuing from the tube-like portion *d* of the shaft passes to the nozzles *c*, in which it undergoes expansion preparatory to impinging upon the channels of the secondary wheel *f*.

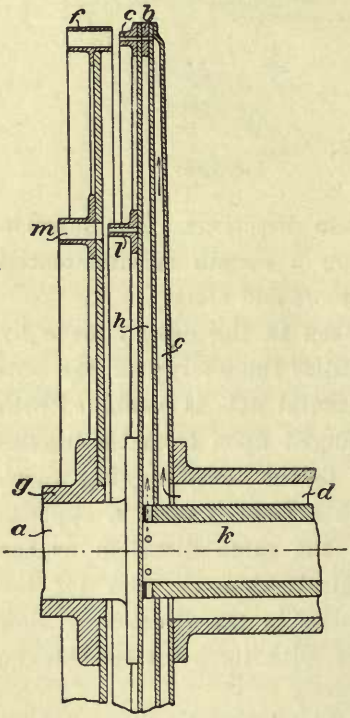


Fig. 236.

The wheel *f* rotates in the direction opposed to that of the wheel *b*. It is borne by the hollow shaft *g*. The wheel *b*, however, has another chamber *h*, which is in communication with the hollow part *k* of the shaft, and into which the nozzles *l* deliver. In order to reverse the direction of rotation of the working wheel *b*, these latter are made to point in the direction opposite to that of the nozzles *c*. The vanes *m* also, belonging to the wheel *f*, are arranged in a corresponding manner, so that when steam is introduced into the chamber *h*, and the chamber *c* is closed, the direction of rotation of both wheels, and consequently also of the shafts, is reversed. As an alternative arrangement to the one illustrated, the two wheels can both be mounted on the ends of the shaft without supports (flying), so that the one shaft is not obliged to pass through the other. Also, the nozzles *l* can be separately attached to the case, so that the steam impinges upon the backward-motion turbine vanes *m* from the left-hand side (of the drawing). In this case the wheel *b* will be put out of gear when the engine is reversed.

Radial pressure turbines with impingement from within are made by *Wilson*[1] with counter-direction wheels. In the case A (Figs. 237 and 238) the working-wheel discs B and C are mounted on the ends of the shafts *b* and *c*. The disc B bears the conaxial vane wreaths *d*, which alternate with the vane wreaths *e* of the disc C. The inner portions of the two discs are perforated, so that the steam admitted through the branches *f* can pass between the discs, and, streaming thence radially outwards, can impinge upon the vanes one after another. The curvature of the vanes *d* is here opposite in direction to that of the vanes *e*, and the wheels rotate in correspondingly different directions, as shown by the arrows in Fig. 238. The steam inlets *f* are widened out where they enter the case, in order that the rotating arms *g* of the wheel may not be able temporarily to stop the supply of the steam, but that the latter may proceed unhindered through the perforations *h*. The numbe

[1] E. P. 12,026 of the year 1848.

of the steam channels increases from vane wreath to vane wreath (from stage to stage), so that the expansion of the steam can be kept up as it proceeds along its path. The exhaust steam collects in the annular chamber *k*, from which the branches *l* lead it to the atmosphere or into the condenser. If need be, a further expansion of the motive medium could be attained by a broadening of the vanes in the axial direction from stage to stage. As already observed, the wheels are mounted loosely on the ends of the shafts. To enable the wheels to be centred one with the other, a bolt is inserted axially in one of the shaft ends, which fits into a correspond-

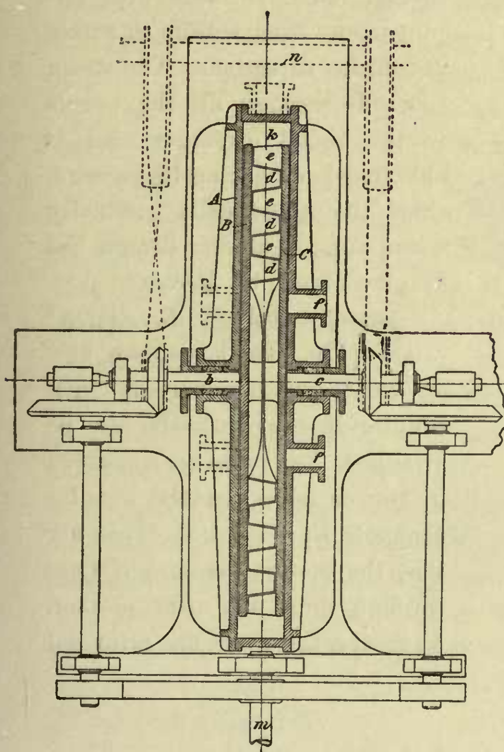


Fig. 237.

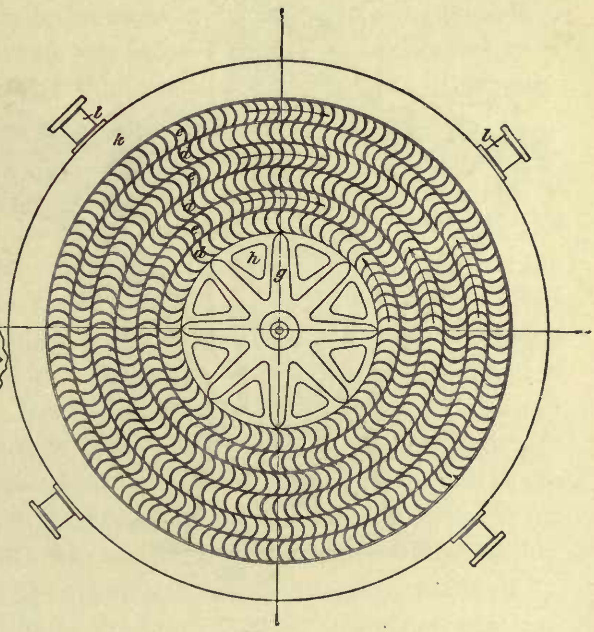


Fig. 238.

ing hole in the other shaft. The transmission of the work of the two wheels to a common shaft *m* is effected by means of toothed wheels; belting, however, with open and crossed belts might be applied, as shown for the shaft *n*.

Brady[1] (1899) (Fig. 239), amongst others, has produced a pressure turbine with counter-direction wheels. His steam turbine also possesses two concentrically arranged vane wreaths which rotate in opposite directions. For the attainment, however, of a symmetrical application of power the disc *a* turns between the two discs *b* and *c*. Disc *a* is borne on a hollow shaft which serves to lead the motive medium outward from the centre. Discs *b* and *c* together form a case, and are mounted on a hollow shaft, which ships over the one first mentioned. On the inner shaft, as also on the outer one, several discs can be arranged. This constitutes

[1] D. R. P. 122,103.

a special advantage possessed by the arrangement described, since the work which the engine can do may easily be increased by an addition to the number of the discs thus working in common. An increase of the diameter of the discs is not necessary; only the outer discs must be set further apart on the hollow bearing.

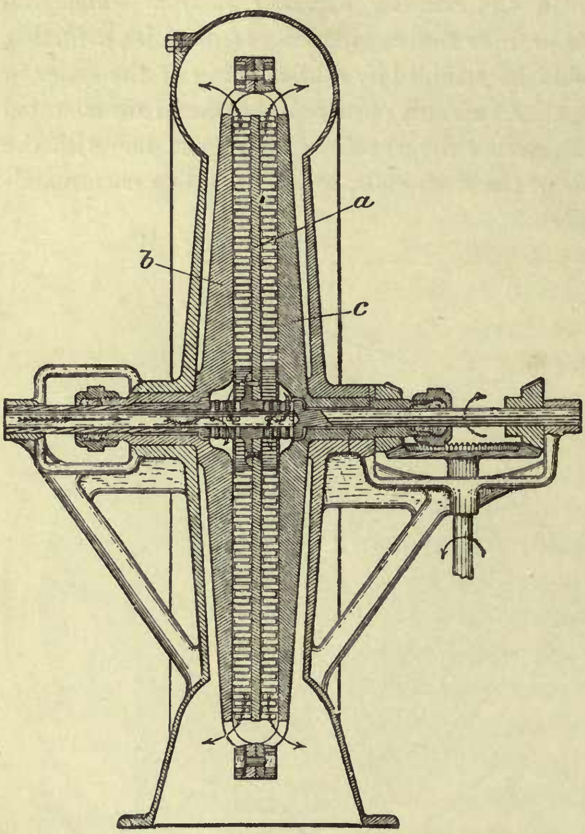


Fig. 239.

Terry[1] (1892) also causes several radial turbines with impingement from within to rotate in contrary directions. The steam proceeds through all the wheels in the radial direction, and is able to expand during its passage, since the wheel cells gradually widen out. A similar design has also been made by *Mercer*.[2]

Amongst others, *B. Ashton*[3] (1900) (Fig. 240) has shown how an axial pressure turbine with counter-direction wheels can be arranged. The vanes are contained in two or more conaxial annular channels F, G in such a manner that the motive medium courses through these one after another. Moreover, the vanes are arranged

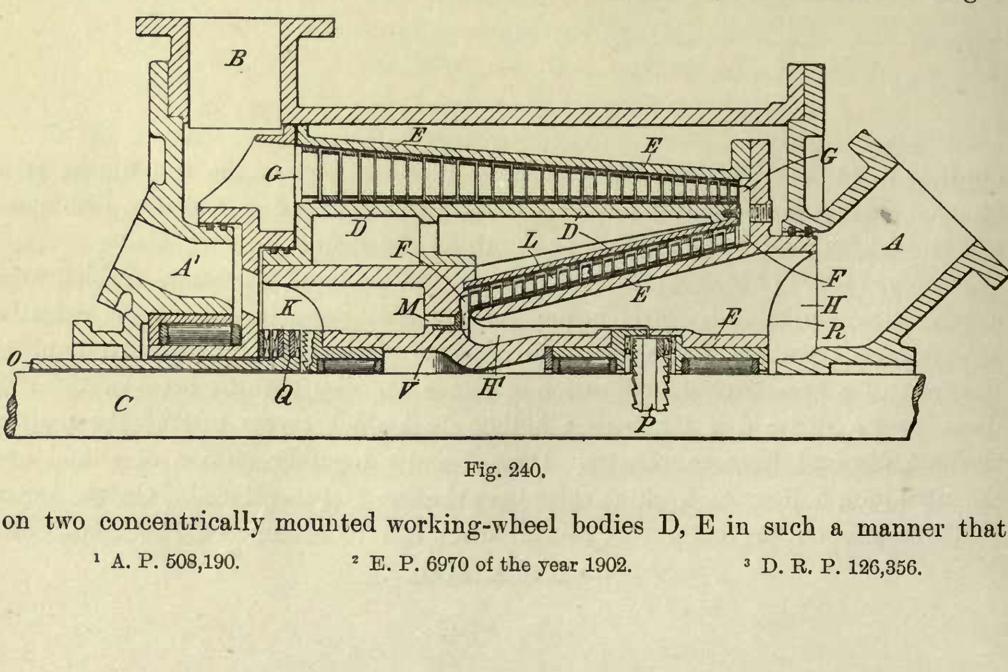


Fig. 240.

on two concentrically mounted working-wheel bodies D, E in such a manner that

[1] A. P. 508,190. [2] E. P. 6970 of the year 1902. [3] D. R. P. 126,356.

those of the one working wheel are attached to a removable support L for the purpose of exactly adjusting their positions with reference to the other wheel, and to enable the inner sets of wheels to be removed.

For motion in the forward direction the motive medium is led through the inlet A into the chamber H, whence it proceeds between the ribs R to the annular chamber H^1. At the openings M, which are arranged in a circle on the working-wheel body D, a valve V is provided, which opens to the chamber H^1. The motive medium thus has to stream through the annular chamber F, and then through the channel G in a direction practically parallel with the axis of the shafts C and O impinging on the vanes on its way, by which the working-wheel bodies D and E are caused to rotate in opposite directions, the vanes being so curved as to effect this. The motive medium then escapes through the exhaust B into the atmosphere, or is caught in a condenser. The wheels D and E transmit their rotary motion by means of the gearing coupling P on to the shafts C and O.

For reversing[1] the shafts C and O the motive medium is allowed to stream in at A^1, whence it passes into the annular chamber K, and there brings the valve V into the position shown by the broken lines, and closes the chamber H^1. The pressure in the chamber K pushes the working-wheel bodies D and E along in the longitudinal direction, by which the couplings of these, with the shafts O and C, are interchanged. From the chamber K the motive medium streams through the annular guides F, G to the exhaust B. Meanwhile the working wheels D and E rotate in the same direction as before, but the shafts C and O, as intended, turn in the opposite direction. If the steam is to be allowed to stream through both sets of wheels in an axial direction, the outer annular channel G must widen out in the same direction, and the steam must be led back between the channels F and G (in Fig. 240) to the left. In reversing, then, the use of a second inlet for the working steam becomes unnecessary as soon as the chamber K is closed against the inlet A^1, and the steam led to this place only effects the axial displacement of the working-wheel bodies D and E.

In the case of velocity turbines with counter-direction wheels the impinging apparatus of the first working wheel must be fixed.

So long ago as the year 1843, *Pilbrow*[2] makes use of counter-direction wheels in order to reduce the speed of rotation of the turbine. He uses axial wheels, the channels of which, as in the Seger turbine, which will be referred to farther on, overlap each other at one point, at which also the steam is delivered by means of a nozzle. With an increase in the number of the wheels on which the jet of steam has successively to impinge, the speed of the turbine is said to become reduced. For the prevention of a loss due to dispersal of the steam, the channels are to be made to run one in front of another as closely as possible. For the same purpose Pilbrow also recommends that the channels themselves be made narrow in order that they may be filled by the steam, thus recalling the transition turbine. It is

[1] See also D. R. P. 131,995 in the Chapter on Reversing Gear.

[2] E. P. 9658 of the year 1843.

even reckoned upon that the width of the channel diminishes from wheel to wheel in the direction of the flow of the steam, so that the latter also has to fill channels increasing in number from wheel to wheel. Pilbrow also pays heed to the influence of the centrifugal forces, which must not be allowed to throw the steam out of the channels. For this reason he sets the vanes, not in the radial direction, but at an

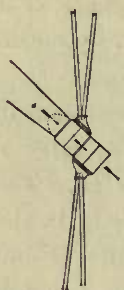


Fig. 241.

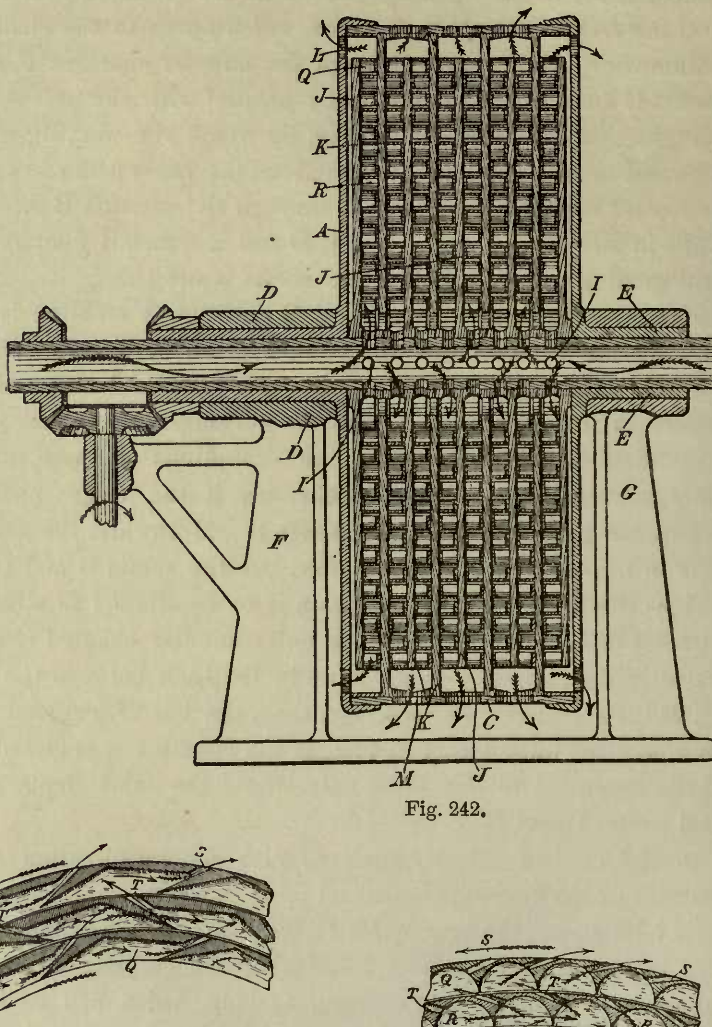


Fig. 242.

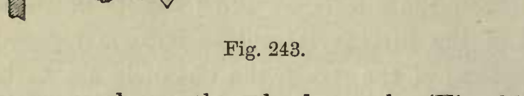


Fig. 243.

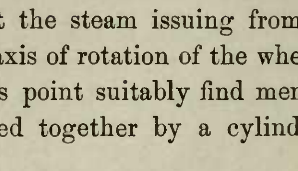


Fig. 244.

acute angle on the wheel wreaths (Fig. 241), so that the steam issuing from the nozzle p also assumes a direction at an angle with the axis of rotation of the wheels.

The turbine designed by *Brady*[1] (1900) may at this point suitably find mention (Figs. 242 to 244). The outer discs A are connected together by a cylindrical

[1] D. R. P. 123,933.

intermediate piece C, serving as bearer of a number of discs K, which have openings at their centre for the passage of the shaft. The discs A rest with their naves D and E in the frame FG. Passing through the naves is a hollow shaft, bearing a number of discs J, which run between the discs K, and with their concentrically arranged projections pass between the concentric projections of the discs K. The motive medium is introduced through the hollow shaft, and passes through holes I into the chamber between the discs, where it performs work in the chambers and then streams out through the openings L and M. The discs J and K are here driven in the contrary direction in the usual manner. The side projections of the discs J and K consist of concentric rings, which are provided with cavities Q and R (Figs. 243 and 244), and with channels S and T for the discharge of the motive medium. The chambers R and Q are at their sides completely closed, and are open only towards the inside (in the radial direction). The open side of the chamber is covered by the closed rear side of the foregoing ring of the neighbouring disc, so that the chambers R and Q may be looked upon as completely closed spaces. In order to enable the partition walls to be well shut off from the neighbouring disc, this latter is bored out somewhat more deeply at the places in question. The steam introduced at I passes from the chambers Q of the first ring through the channels S into the chambers R of the next-following ring, and drives the latter in the direction indicated by the arrows in Figs. 243 and 244. From the chambers R the steam escapes through the oblique channels T, and in so doing strikes the walls of the chambers Q of the next-following ring of a disc rotating in a direction contrary to its own, and so on until it finally passes through the radial openings of the closing ring and escapes through the holes L.

Attention may here also be called to the arrangement made by *Seger*[1] (1893) (Fig. 245), in which the steam acts on two axial wheels *a* and *b*, which are placed one behind another without the intervention of leading apparatus. In this case the diversion of the direction of the steam is not resorted to. The axes of the wheels *a*, *b* do not coincide, but lie parallel with one another, so that the wheels partly overlap. The steam streaming axially through, passes directly from the one wheel into the other, and accordingly turns these in opposite directions. The wheels, then, by means of gearing, act together in imparting their work to a separate shaft.

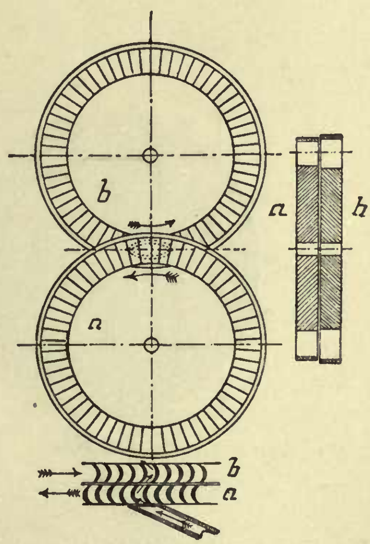


Fig. 245.

The conditions are similar in the arrangement made by *Ferranti*,[2] which, however, includes a radial widening out of the wheel cells both in the inward and in the outward directions.

[1] D. R. P. 70,551.

[2] E. P. 2565 of the year 1895.

The turbine adopted by *Altham*[1] (1892) is illustrated in Figs. 246 to 249 in which Fig. 249 represents a section through 4–4 in Fig. 248. The turbine consists of two wheels a, b, having the same axis and rotating in opposite directions, the wheel

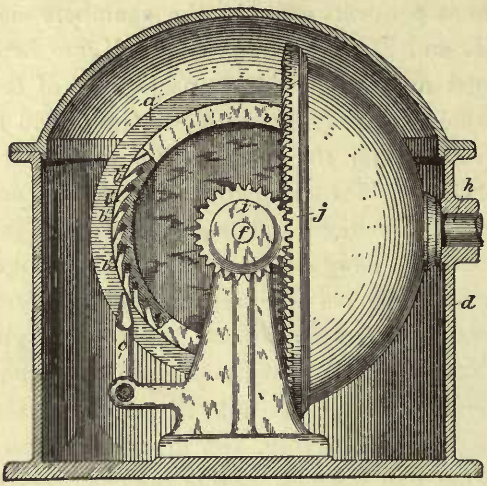


Fig. 246.

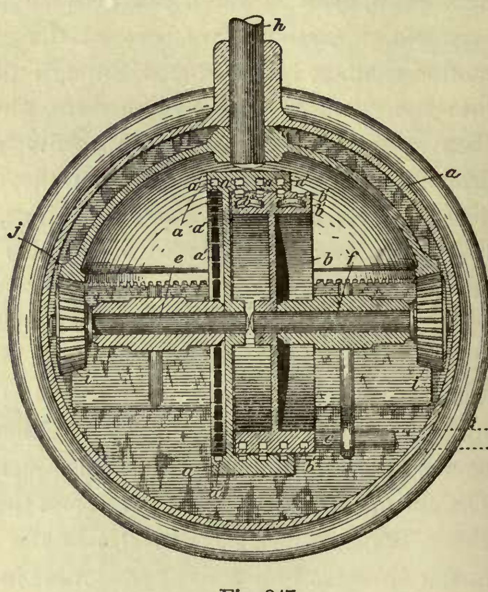


Fig. 247.

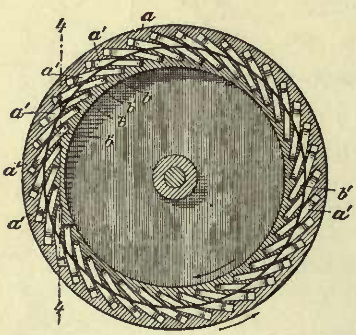


Fig. 248.

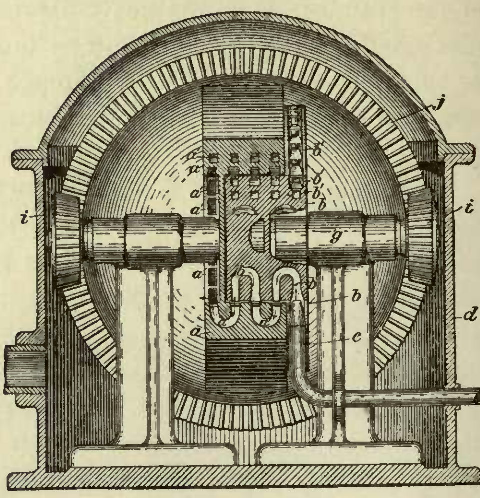


Fig. 249.

b lying partly within the wheel a. In both wheels channels (cavities) a^1 b^1 are distributed over the periphery. The channels a^1 are set approximately tangential to the periphery of the wheel b, and the channels b^1 to an imaginary circle within the wheel b. The steam nozzle c is so placed that it lies in a line with each successive cavity b^1, which comes opposite to it as the wheel b turns. The steam

[1] D. R. P. 68,787.

enters the widened mouth b^1 of the first channel, and passes from the latter into the channel a^1 of the wheel a, whence it again streams into the wheel b. The discharge into the case d takes place through the channel a. It will be seen that the cavities of both wheels are provided with inlet and outlet mouths, and the cavities of the outer wheel are arranged to shift with those of the inner one. The cavities are, also, in each case placed in rows that are parallel to the axis of the wheel in such a manner that the inlet mouths belonging to the outer wheel lie directly opposite the outlet mouths belonging to the inner wheel. In order that the entrance of the jet of steam which streams from the nozzle c may be facilitated, the inlets b^1 of the channels of the wheel b are widened out. The inserted pieces which produce the V-shaped channels might be dispensed with, though in this the leading of the steam would no doubt suffer. The jet of steam, then, works alternately on the wheels a and b, driving them in opposite directions. The shafts e and f transmit their motion by means of the gearing i, j to the shaft h.

V

FRICTION WHEELS

The high velocity of the jets and the capacity of the steam for expansion suggested the idea of allowing the working medium to act in a different manner. Instead of its being constrained by the organs which take over its energy to change its directions of current, it should then act on the body to be driven, solely by means of friction, so that concussions are avoided. For the purpose, however, of providing sufficient frictional resistance, appliances of various kinds have been proposed, which in part no doubt themselves occasion concussions and pressures. For reasons which lie on the surface, little practical value attaches to the friction wheels, and only a few examples of these will accordingly be here given.

According to the design of *Winkler*[1] (1885) ring-shaped discs are arranged parallel to one another, and provided with undulations of such a nature that the crests and hollows lie in the radial direction. To the steam entering in the tangential direction, *i.e.* parallel with the discs, the latter offer surfaces of ascending and descending slope, which entail concussive, pressure, and sucking actions of the steam, the sum of which constitutes the motive power. The impingement may be effected in the radial direction from within or from without.

Hammesfahr[2] (1887) makes use of circular, screw-shaped, or spirally bent pipes through which steam under pressure is caused to stream. By the friction of the steam on the smooth, rough, checkered, indented, or otherwise prepared pipe surfaces, or by its concussive action against impact surfaces, projections, contractions, valves, flaps, and suchlike appliances, or, finally, by combinations of the foregoing kinds of action, the pipes are caused to rotate in the direction of motion of the current.

An arrangement made by *Thrupp*[3] is shown in Fig. 250. By means of discs *a* placed side by side in the axial direction, and of the discs *b* placed between the former ones, narrow chambers are formed, through which the steam issuing from the nozzle *c* expands. This expansion, moreover, is intended to take place without the jet of steam from the nozzle *c* to the discharge being caused to change its direction. By means of the friction produced by its action on the discs *a* it carries these along with it. The arrangement illustrated is that of a compound turbine, since the jet of steam on leaving the discs *a* passes between the discs *d* of another wheel before it arrives at the discharge *e*. The compound turbine is, moreover,

[1] D. R. P. 35,783. [2] D. R. P. 43,726. [3] E. P. 6422 of the year 1901.

reversible, the steam for motion in the opposite direction being introduced through the nozzle *f*.

Roberts,[1] on the other hand, avails himself of the assistance of the centrifugal forces, which are assumed to press the jet of steam against a surface which, be it remarked, is smooth. By this the friction is said to be increased. The jet of steam is, accordingly, directed against the inner wall of a cylinder, so that it has to sweep along it.

In like manner *Nordenfelt* and *Christophe*[2] allow the steam to impart its kinetic energy to smooth, vaneless, working wheels. Several cup-shaped wheels, each able to rotate independently, are fitted concentrically, one within the other, and so arranged that the rim of each projects over that of the preceding one. The jet of steam directed against the rim of the innermost wheel sets the latter in motion and then, in issuing from it, acts against the rim of the second, and so on. The wheels

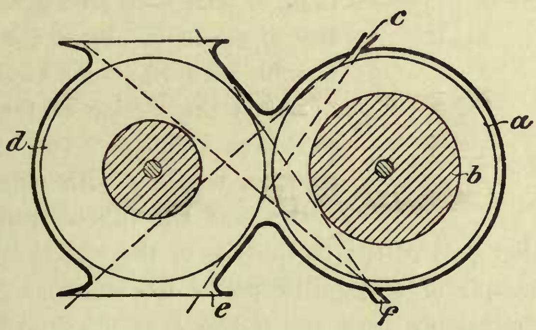


Fig. 250.

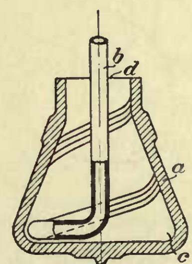


Fig. 251.

are made to rotate in one direction, but with different velocities. Their work is combined, by the help of suitable gearing, so as to act together on a shaft. In place of the cup-shaped working wheels smooth discs may be made use of, which overlap each other with their rims in step fashion, and against the surfaces of which the jet of steam, beginning with the centremost wheel, acts at a suitable angle. According to the embodiment of this idea given in Fig. 251, the various wheels, rotating at speeds diminishing in proportion to the decreasing force of the jet of steam, are combined in the form of a single conical wheel *a* without vanes. Into this the jet of steam is introduced through *a* pipe *b* at the point *c* of the greatest diameter. The smallest diameter of mouthpiece *d* is so dimensioned that the jet gliding upwards in a screw-like course along the inner wall of the wheel acts progressively against the surfaces of the wheel, as they gradually approach the axis of rotation. The circumferential velocities of these diminish in proportion to the gradual decrease of the force of the steam.

In the turbine of *Vojáček*[3] the pressure jet acts by means of friction, being admitted in a tangential or almost tangential direction through a nozzle *a* (Figs. 252 and 253) into the groove *b*, which runs round the periphery of the disc

[1] E. P. 1675 and 3134 of the year 1872. [2] D. R. P. 84,853.
[3] D. R. P. 92,372. E. P. 18,807 of the year 1894.

c, and is formed with an inward taper. By virtue of the pressure here exercised against the surfaces of the groove and the friction thus produced, in conjunction with the regular expansion or outward throw of the particles of the motive medium, the disc is made to rotate. The jet can be made to impinge at the same time upon two discs *d, e* (Fig. 254) which almost touch each other, and which rotate in opposite directions (Figs. 254 and 255). These may have various profiles. If the nozzle *a* (in the illustration, Fig. 254) be brought into the position a^1, a reversal of the direction of rotation of the wheel *d* is obtained, the wheel *e* then being carried round empty in the same direction. The aim will then, of course, be for a given quantity of steam to obtain the largest possible jet surfaces for friction, and to adjust the section of the nozzle to this requirement. In Fig. 256 is shown a nozzle *f* with ring-shaped outlet, and the tube-shaped jet of steam is pressed against a farther part of the peripheries of the wheels by means of an inverted cone *g*. An example of a step-like use of the steam is given in Fig. 257, in which the current streaming from the nozzle *h*, after parting with a portion of its energy to the wheels *k, l*, is taken up by the nozzle *m*, and led to a further pair of wheels *n, o*. In case weak pressures of steam are available, the

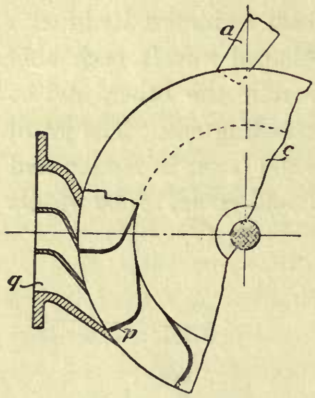


Fig. 252.

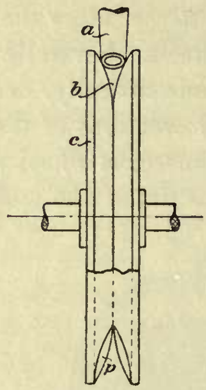


Fig. 253.

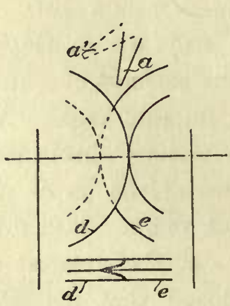


Fig. 254.

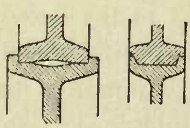


Fig. 255.

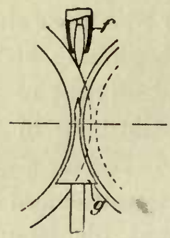


Fig. 256.

sides of the grooves are provided with vanes *p* (Fig. 252). The impingement can also be effected by the help of a leading apparatus *q*. A combination of the grooves with plough-like vanes (Fig. 258) enables the steam to act, first by means of friction in the groove *r*, and then by pressure. The sharp edge *s* of the vane *t* here divides the steam and throws it off to both sides of the wheel.

Winkler[1] (1885) (Fig. 259) seeks to reduce the high velocities of rotation of the

[1] D. R. P. 32,847 and 33,404.

steam turbine by causing the jets of steam issuing in front of the vanes to exercise a suction on these latter. The jets are made to create a vacuum, into which the blades are pressed by the air streaming in from without. In the case *a* a drum is fitted, which is free to rotate, and consists of two side discs *b*, with the cylinder *c* between them. Conaxial with the drum, but fixed against the case *a*, sits the cylinder *d*. The latter is supplied through the hollow shaft with steam, which streams out through the slot-like steam nozzles *e*. On the cylinder *d* parallel metal rings *f* are fixed side by side. Between these the steam must stream in order to be able to discharge through the openings *g*. In this it carries the rings *f* round with it by means of friction. The steam escaping radially from the cylinder *c* in front of the vanes *h* (in the direction of rotation) is now intended to produce under-pres-

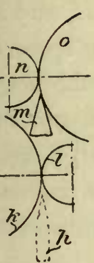


Fig. 257.

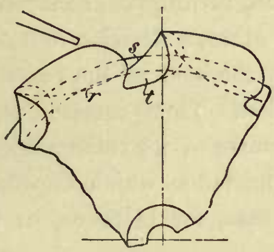


Fig. 258.

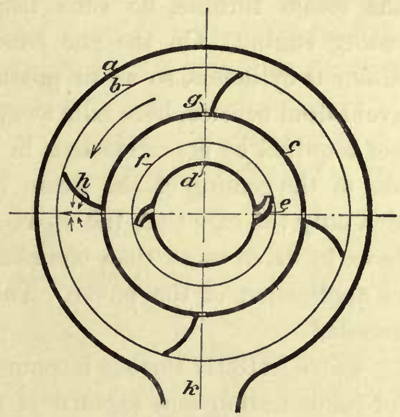


Fig. 259.

sure at these points, so that the air which streams in through a side opening of the case *a* is able to exert an excess pressure on the after-sides of the vanes *h*. The steam streams out through the branch *k*.

The sucking action of the jet of steam, in the absence of frictional work, is, moreover, to be found in the arrangement of *Fryklind* and *Laestadius*[1] (1900) (Fig. 260). On the hollow shaft *a*, which is free to rotate, at least one vane *b* is fixed. For the discharge of the motive medium from the shaft into the outer air an opening *c* is arranged beside the vane and near one end of it, the vane being set parallel with the geometrical axis of rotation. The idea involved is that by the outward pressure of the motive medium through the opening *a*, a rarefaction of the air on the one side of the vane is caused, so that the pressure of the air on the opposite side of the vane can cause the shaft to turn.

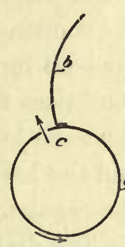


Fig. 260.

By means of steam streaming radially outwards, *Lohmann*[2] also proposes to create a vacuum, which occasions a pressure of the air tending to produce rotation of the working wheel.

[1] S. P. 21,145.

[2] E. P. 19,394 of the year 1898.

VI

COMPOUND ARRANGEMENTS

For compound arrangements, that is to say, for the taking of the work of a given quantity of steam from several cases at a distance from one another, there is, in the steam turbine, no such necessity as exists in that of the ordinary reciprocating engine. On the one hand, the necessity of allowing the steam to work in different cylinders, so as to produce an even turning moment on the shaft, and to avoid dead centres, here falls away; on the other, a division of the working space is not required by considerations in regard to the heat of the parts of the machinery, and of the cooling of the steam by expansion. There remains, then, for consideration only the case that the steam, in the course of its transmission of energy, may have to drive more than one shaft, a circumstance which depends on the manner of application of the power. This is the case, for instance, in the propulsion of vessels.[1]

Curtis[2] (1897) further recommends the compound arrangement, when the engine for condensation and exhaust is made use of. Two stage turbines are then fitted upon one shaft. The steam expands in the first turbine from the entrance pressure to about that of the atmosphere, and proceeds then to the second turbine, in which its pressure sinks to that of the condenser. In the passage from the first turbine to the second one an exhaust leading to the atmosphere is inserted. When opened, this allows the steam from the first turbine, reduced by its work to the pressure of the atmosphere, to escape into the latter direct. The first method of working—with the condenser—is, then, that of a compound turbine; the second, in which the low-pressure turbine is altogether shut off, is that of an exhaust engine. Meanwhile the conditions for the regulation and the performance of the first turbine remain exactly the same in both cases. The widths of the inlet nozzles of both the high-pressure and the low-pressure turbines can be varied by means of slides, which are moved by the regulator.

The division of the turbine case into two (or more) parts may also become necessary, when the shaft must, by reason of its length, be carried in more bearings than the two end ones. Bearings within the case are avoided as far as possible, partly on account of their inaccessibility, and partly to prevent the admixture of the lubricating material with the steam. In the case of three bearings, then, the only course open is to place the middle one between two sections of the case, *i.e.* to divide the turbine itself into two parts, set one behind the other.

[1] See Chap. XXIV.

[2] A. P. 590,211.

In regard to axial turbines, attention may also be called to the case in which, in order to prevent the axial thrust of the steam, half the engine rotates in the one direction, and the other half in the opposite one. Assuming that both sections of the turbine are mounted on the same shaft, they will be placed in separate cases and be made to communicate with one another by a connecting pipe. Finally, the adoption of the compound arrangement will be a matter for consideration, even if, as recently proposed by E. Lewicki, the steam of multi-stage turbines be considerably superheated when passing from one stage to another.[1]

[1] *Zeitschr. d. Ver. deutscher Ing.*, Vol. 47, p. 530.

VII

MEANS FOR REDUCING THE NUMBER OF REVOLUTIONS PER MINUTE OF TURBINES

THE large number of revolutions per minute which have to be reckoned with in turbine work was, at an early date, recognized as a drawback, and means were sought which would enable the speed to be reduced. The first idea hit upon was to utilize the energy of the steam in order to impart motion to bodies of high specific gravity, and then to let these bodies influence the turbines.

Thus the apparatus shown in Fig. 261 comes from *Watt*.[1] In the box *a*, which is for the most part filled with a liquid which can drop, such as water, quicksilver, or oil, the vessel *b* on the pedestal *c*, and with the throat-bearing *d*, is set free to rotate about a vertical axis. The vessel is divided by the partition *e* into two chambers, each of which has a bottom flap *f*, opening inwards. Each chamber is provided, at the height of the bottom, with a discharge pointing in a tangential direction, but closed at the top, and has only a side opening *g*, which, according to the position for the time being of the vessel, delivers either into the steam-supply pipe *h*, or above the liquid into the box *a*. The chambers become filled one after the other with the liquid from the box *a*, which drives the steam out again through the side opening arranged at the bottom, so that the vessel *b* is forced round by the pressure of the out-flowing jet against the fluid which surrounds it. If steam of atmospheric pressure be made use of, the box is closed air-tight and connected with the condenser by the pipe *k*. Otherwise the steam exhausts into the open air.

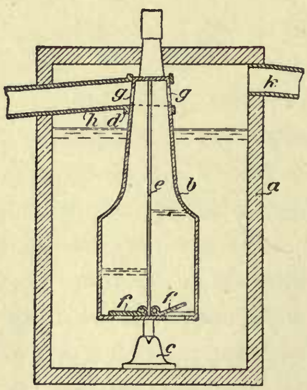


Fig. 261.

Experiments of this kind are to be met with down to the most recent times. The turbine designed by *Trossin*[2] (1896) (Fig. 262), also, is intended to be driven by molten metal. Meanwhile the steam is allowed to enter at intervals, and the vessel containing the metal is at the same time automatically closed. A is the leading apparatus, by means of which the fluid metal is introduced into the vaues F of the turbine. The leading apparatus is in connection with the steam chest G, which is provided with the valve *o*. The valve rod *h* is raised by means of the lever

[1] E. P. 1432 of the year 1784.

[2] D. R. P. 98,990.

p, the rod *q* and the roller *r* by the triangular projection *s* fixed on the fly-wheel B, and these are lowered again by means of weights or springs. The valve *o* is thereby opened or closed. The channels *f*, *u*, and *v* contain the molten metal. The steam-supply pipe D stands, by means of the passage E and of side channels, in connection with the steam-chest G. The engine works in the following manner: if the valve be opened for a short time by the motion of the engine, the steam streams into the channel *v* and separates the molten metal into two parts. One of these parts is thrown through the channel *u* against the vanes of the turbine, and imparts kinetic energy to these. The other part rises upwards in the channel *f*, but is prevented from escaping by the annular valve *y*, which swims on the metal. After the shutting-off and expansion of the steam, the metal sinks back, and with it the annular valve *y*. The metal can then assume its original level in the channels *f* and *u*.

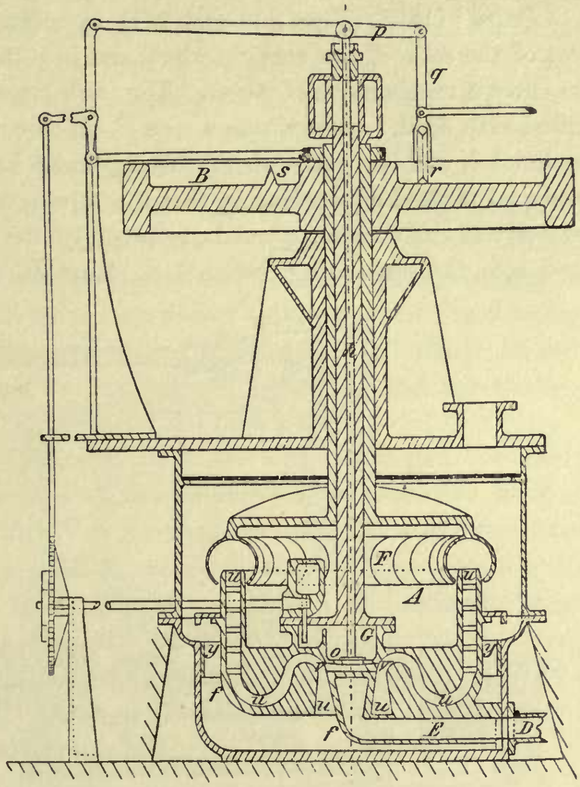


Fig. 262.

Fiedler[1] makes use of a Pelton wheel, against the vanes of which molten metal,

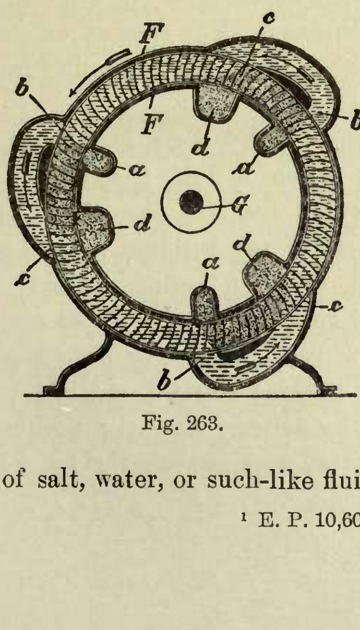


Fig. 263.

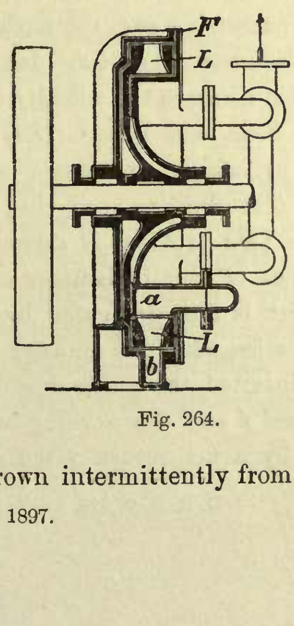


Fig. 264.

solutions of salt, water, or such-like fluids are thrown intermittently from a nozzle.

[1] E. P. 10,609 of the year 1897.

For this purpose the nozzle is fixed to a chamber, into which steam is then introduced through a geared valve, after a certain quantity of the liquid has passed through another valve.

Prásil[1] (1893) (Figs. 263 and 264), by means of steam pressure, forces a liquid out of the cells of the working wheel and into them again alternately, in order thus to effect a transmission of power. The working wheel L, the channels *c* of which are filled with fluid, rotates within a case F, which contains a steam inlet *a*, a circulating channel *b*, and an outlet *d*, the arrangement being several times repeated. The assumed method of working is, that the liquid, which arrives at the inlet *a* with the velocity of the working wheel L, is forced by the steam out of the cells of the wheel and into the circulating channel *b*. Here it experiences a certain acceleration,

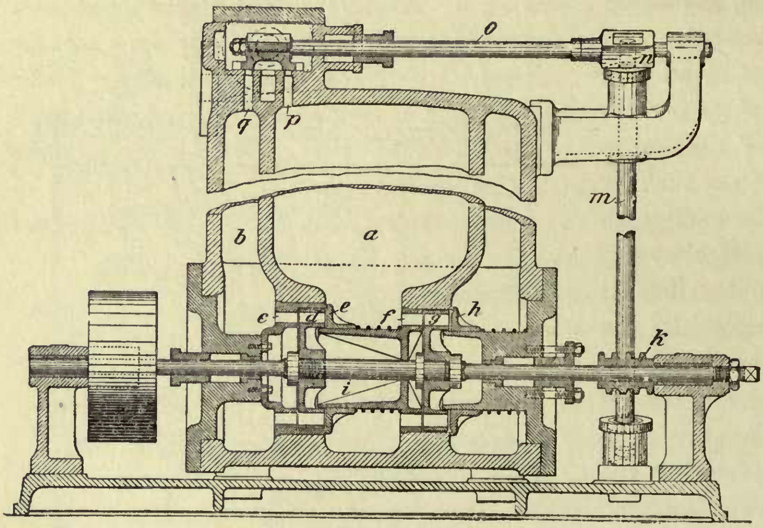


Fig. 265 a.

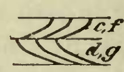


Fig. 265 b.

which finds expression in the form of a pressure on the vanes of the working wheel when the channels *c* pass the outlet of the channel *b*. The steam then escapes from the cells into the exhaust *d*. Its place is taken by the fluid pressing from behind, which again assumes the velocity of the wheel.

Fleck, Sons, and Voigt[2] (1897) (Figs. 265 *a* and *b*) have had an arrangement in view by which the steam alternately presses a fluid from one vessel into another. The fluid thus moving backwards and forwards then forms the motive medium. This problem is solved by the help of two turbines, in connection with which tubes of small diameter are avoided, in the following manner: The inner fluid holder is here indicated by *a* and the outer one by *b*. These are placed in communication with one another by means of two openings or channels, in one of which is inserted the guide wheel *c* and the working wheel *d*, and in the other the guide wheel *f* and the working wheel *g*. The delivery from the working wheel *d* is closed by a non-return valve *e*: that of the working wheel *g* by a non-return

[1] D. R. P. 76,177. [2] D. R. P. 99,515.

valve *h*. Both the wheels *d* and *g* are fixed on the same wheel *i*. On the latter is arranged a worm wheel *k*, which by means of a shaft *m* with its crank journal *n* imparts a back-and-forward motion to the connecting rod *o*. By this means the intermittent introduction of gas or steam under pressure through the channel *p* into the holder *a*, or through the channel *q* into the holder *b*, is forcibly effected. If, for instance, steam be introduced into the holder *b*, the non-return valve *h* will be closed, and the turbine *d* caused to turn. On the other hand, the steam pressure in the holder *a* shuts the valve *e* and drives the turbine *g*.

A favourite idea is that of keeping water, or some other fluid, in circulation by a device, by which the steam issuing from a nozzle sucks it up after the manner of an injector, and thus imparts a certain velocity to it. In the circuit formed by the fluid, the circulation of which takes place within a cylinder or annular chamber, the driving turbine is inserted. [*Bonnett*,[1] *Voigt*, and *Fleck Sons*[2] (1893).] In such cases a certain length of path must be left between the nozzle and the working wheel, along which the acceleration of the water can take place.

Ulffers[3] (1883) makes the matter more intricate. In his engine a working wheel (vertical or horizontal), provided at its periphery with teeth-like striking vanes, turns within a cistern filled with water. Upon the vanes of the latter fixed guide channels are directed, through which the water circulating in the cistern is led. To effect this, within the channels in the direction of motion of the flowing water steam is forced in through special nozzles, and imparts an acceleration to the water.

In the engines above discussed the object was to impart kinetic energy to a fluid which in its turn was to drive the turbine. Other designers have kept the steam as motive medium, but have mixed it with water in order to increase its specific gravity.

In this manner, for instance, *Hartung* and *Lenz*[4] (1877) have gone to work. Fig. 266 shows a horizontal section through one of the arms *a* and the plan of the nozzle *b*. The two-armed wheel is free to turn about a vertical axis. The steam is introduced through the hollow shaft *c* into the arms *a*. The injector nozzle *d* draws water through a pipe entering from below, and the mixture of steam and water flows out from the reaction nozzle *b*. The latter can, moreover, be turned round so that the direction of motion of the wheel can be reversed.

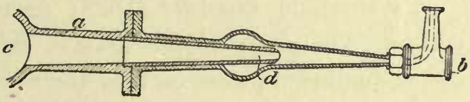


Fig. 266.

Bernstein and Wolfson[5] produce a homogeneous wet steam, which takes up hot water. The latter is injected into the jets of steam previous to their entrance into the nozzles. The mixture works upon an action wheel. For purposes of condensation a vane wheel, which is in the first place intended to eliminate the heavy particles of water, and to considerably condense the steam, is mounted on the shaft

[1] E. P. 1493 of the year 1873.
[2] D. R. P. 89,634.
[3] D. R. P. 25,383.
[4] D. R. P. 910.
[5] D. R. P. 56,023.

of the working wheel beside the turbine. It is said to be possible to work with a low pressure of steam in the boiler if fluids having high boiling points (strong brine) are made use of.

Scott[1] causes the steam to draw up hot water through an injector, and leads the mixture repeatedly to work against an impulse wheel.

The system adopted by *Grabe*[2] (1900) recalls the ordinary circulation in the steam engine. The turbines are driven by a mixture of fluid and steam in such a manner that the first of these, which is in constant circulation, is led by means of a pump to a pipe or system of pipes (Fig. 267), which are wholly or partly heated from without. After the mixture has experienced an acceleration in the pipe or pipe system by the gradual partial evaporation and the consequent expansion of the steam, it is thrown at a high velocity against the vanes of a turbine wheel.

Finally, we meet with the mingling of cold water with the steam in the turbine by *Lacavalerie*.[3]

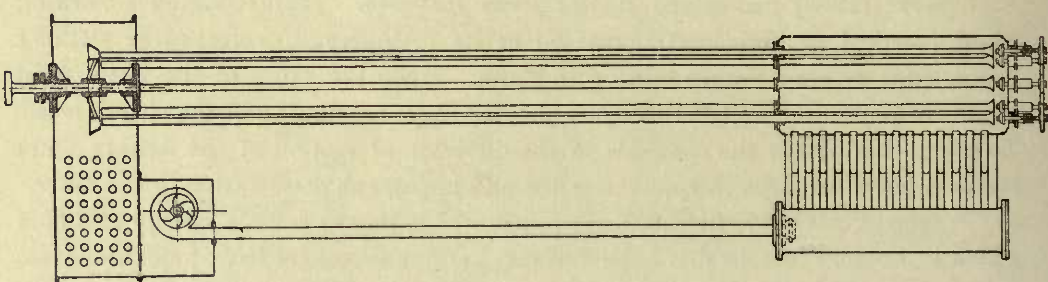


Fig. 267.

Attempts that must be taken more seriously are those in which steam of greater specific gravity, such as, for instance, exhaust steam, is mixed with the fresh working steam.

To this end *Lundell*[4] (1898) makes use of a turbine with impingement from within. The steam nozzles draw in the exhaust steam after the manner of injectors and the mixture passes on into the working wheel.

Stumpf[5] (1901) (Fig. 268) had the idea not to mix the fresh steam with heavier steam, which would have to be heated, and would often prevent the use of condensation, but to lead exhaust steam to join with the expanded fresh steam in a mixing nozzle. An increase of the volume of the steam thus takes place, accompanied by a corresponding reduction of the velocity of the current. The supply lead of the steam and the course of its current may be seen from Fig. 268, which shows the side view of the turbine. It will be seen that, in the first place, the steam passes from the narrowest point *a* of the nozzle *b* to the point *c* through the well-known closed cone-shaped nozzle-piece. In this section the steam expands adiabatically from its initial pressure to that of the condenser. From *c* to *d* the nozzle is then supplied

[1] E. P. 29,637 of the year 1897. [2] D. R. P. 134,617 [3] E. P. 12,060 of the year 1896.
[4] A. P. 613,694. [5] D. R. P. 141,784.

with perforations, which, again, may take nozzle form, and through which a part of the exhaust steam passes into the nozzle and mixes with the fresh steam. The supply-branch *f* for the nozzle *b*, as well as the nozzle, is fixed in the exhaust channel of the turbine. It may also be observed that the inlet-branch *f* for the fresh steam is arranged immediately below the outlet branch *h* for the exhaust steam. In the main leading channel *i* the various jets of steam are intended to assume equal velocities, with which they then pass through the branch *k* into the impingement channel, which is divided off by the spirally winding partition *m*. When the steam exhausts in the same direction as that followed by the jet of fresh steam, the exhaust energy of the exhaust steam could be utilized as well.

Bollmann[1] (Figs. 269 and 270) allows the steam streaming from the clearance space *a* to expand considerably and to draw air at both sides, with which the steam mixes on its way between the discs *b* and *c*, so that steam and air are forced at equal temperatures (together with a certain amount of condensed water) into the annular chamber *d*. From this the mixture proceeds through the channels *e* into the working wheel channels *f*, and expands then further through the fixed leading channels *g* and the working-wheel channels *h* till it reaches the exhaust *k*. The channels are of the forms given in Fig. 270, which shows a section parallel to the axis of the turbine. Meanwhile the working wheel and the leading apparatus may be put together to form a turbine with radial impingement from without. For this it is only necessary that a suitable return lead of the steam mixture radial to the axis be arranged for. This last system of construction gives courses for the steam that are more direct. Further, working wheel and guide wheel can be made to run in opposite directions. Also, it will be possible to make use of single stage turbines instead of multi-stage ones.

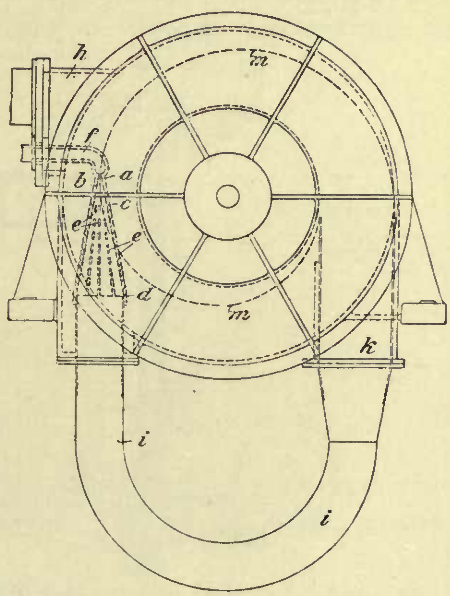


Fig. 268.

Bollmann and *Kohnberger*[2] afterwards (1896) sought to improve this turbine by means of arrangements which aimed at a better mixture of the steam with the air. The mixture of steam (or gas) and air is led through a long chamber R (Fig. 271) which is arranged in screw, spiral, zigzag, or other such form. This proportionately increases the length of the course to be pursued by the current before it enters the working wheel, and affords opportunity for the steam and the air thoroughly to mix together, so that the kinetic energy of the steam is wholly imparted to the mixture. According to the illustration, the fixed vanes L lead the mixture into the spiral

[1] E. P. 6822 of the year 1894. D. R. P. 84,908. [2] D. R. P. 93,462.

passages R, from which the guide vanes lead it against the vanes of the working wheel T. One of the plates K, between which the steam streams, is of annular form, so that the inner part falls away and the outer air is able to press against the

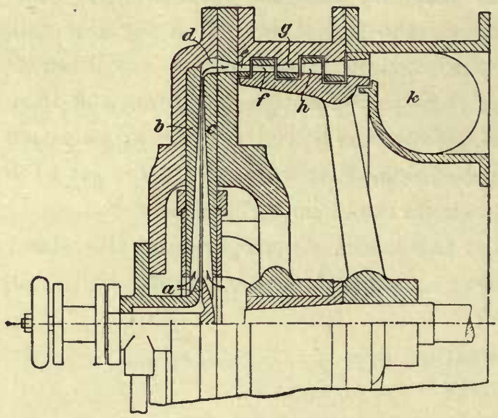


Fig. 269.

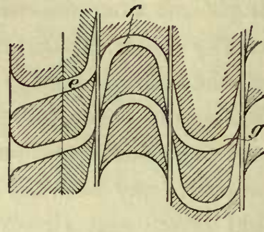


Fig. 270.

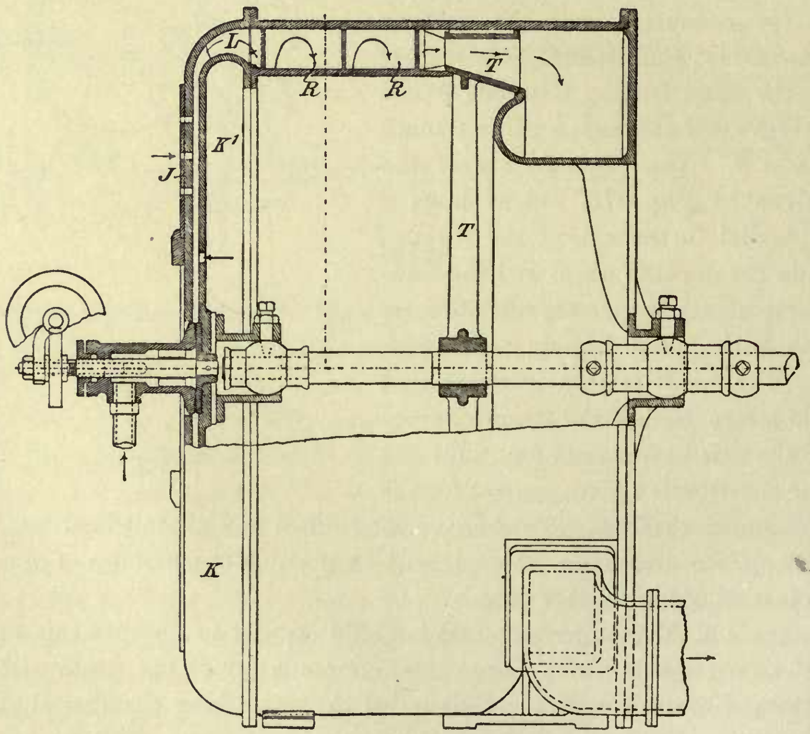


Fig. 271.

thus uncovered surfaces, or against conical, or steam (or gas) jets, by which the absorption of the air is assisted. Between the plates K and K^1 partitions can be inserted, forming spiral channels of gradually increasing width, that are provided with suction holes. In these the steam is mixed with the air, the noise here produced being sensibly diminished. If, then, the plates K, K^1 (or an alternate appliance) be arranged to revolve, they will act as reaction wheels, the wheel T meanwhile, with or without vanes, remains at rest and is firmly fixed to the case.

Mention may here suitably be made of those methods in which hot water is led under a high pressure into the turbine chambers, and there made to produce steam. Such are the conditions underlying the design of *James*[1] (Fig. 272). Out of a boiler hot water (or other fluid) mixed with steam is led into the hollow shaft *a*. The mixture flows through curved arms *b*, which are widened out at their discharge end. It works by back pressure, and passes into the case which encloses the wheel. Here the steam separates from the water, and escapes either to the condenser or into the atmosphere. The water meanwhile falls away, and is returned to the boiler. James does, indeed, make use of mixtures in the arrangement illustrated by Fig. 273. Through the hollow shaft only steam is introduced, which is led through the pipe *b* and blown into the ring *c*. Through the fixed pipe *d*

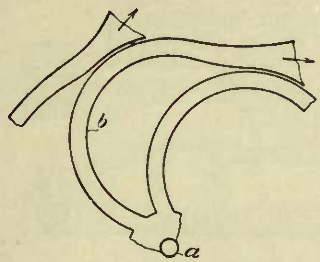


Fig. 272.

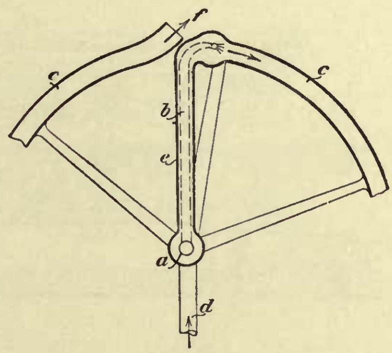


Fig. 273.

and the revolving pipe *e* the steam now continuously sucks up water from the chamber surrounding the wheel, so that a mixture of steam and water courses through the ring *c* and is discharged at *f*.

Mention must further be made of the method of *Hutin* and *Leblanc*[2] (1898), who likewise introduce into the turbine superheated water, which there gives off steam.

A variation from the foregoing is the method of *Hörenz*[3] (1899). Here the method of heating water in a boiler to a temperature sufficient to produce a high pressure of steam is not employed. A given quantity of water is pumped into a heater that stands close to the case of the turbine, and, after the openings are closed, is there brought to a very high temperature. Then the shut-off appliance fitted between the heater and the engine is opened, so that water of very high temperature and correspondingly high pressure is now, in entering the turbine case, changed into steam, and does work by expansion. Continuous working could, then, on this system, be maintained by the use of a number of charges, which would have to be fired off one after another alternately towards the turbine case in order to empty it, and towards the boiler to refill it.

[1] E. P. 7854 of the year 1838. [2] A. P. 639,394. [3] D. R. P. 121,722.

The method made use of by *Prall*[1] (1898) (Fig. 274) is characterized by the circumstance that the motive medium taken in liquid form from the boiler, and issuing from suitable guiding nozzles *l*, is gradually changed into steam through the reduction in pressure. Expanding gradually to the full, it is then thrown against the vanes *a* of several turbine wheels arranged one behind another. When, for instance, highly superheated water is used, it acts on the vanes of the first wheel, by reason of its liquid form, with a very considerable blow. It then acts on the succeeding wheels with blows which become gradually weaker in proportion as the water more and more assumes the form of steam, until it finally impinges on the vanes of the last wheel in the form of pure steam. In the case of the engine shown in Fig. 274 the superheated water entering at *r* fills the chamber *k*, and the steam is discharged from the branch *s*.

De Laval[2] also applied himself to similar problems, with the difference, however, that he provided for the development of the motive medium in his expansion

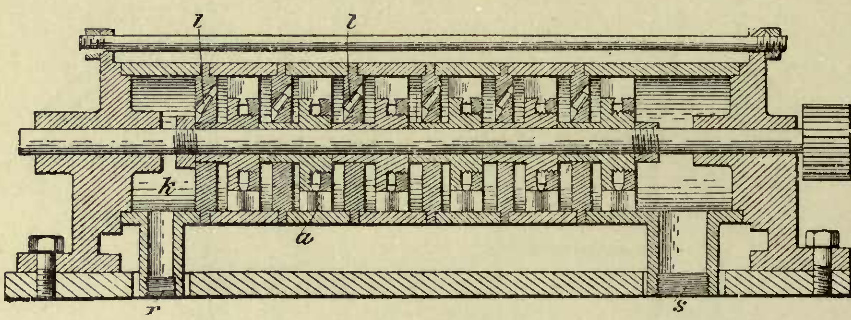


Fig. 274.

nozzle, in which a mixture of liquid and steam is intended to be produced. The highly superheated water, therefore, is led in the same physical condition up to the nozzle, which then first widens out in conical form in the direction of the mouth, and then merges into a (round, ring-shaped, square, etc.) section that remains constant. The passage through this nozzle is intended to produce steam, which mixes perfectly with the water that still remains. The pressure of the mixture decreases as the nozzle widens, but its velocity at the same time increases.

Finally, attention may be drawn to the method of *Beck*[3] (1900), in which the motive medium is intended to circulate within the turbine. A mixture of water, ammonia, and carbonic acid, which fills a portion of the turbine chamber that is shut off from the outer air, is evaporated by a direct heating of the chamber. The expanding vapours perform work on a set of vanes of the working wheel and pass into a second chamber, which is cooled from without. Here the expanding vapours experience condensation. The downward-flowing product of the latter drives the same working wheel by impingement upon a second set of vanes, and finally returns to the evaporation chamber.

In every case there will be a more or less complete mingling of the steam

[1] D. R. P. 125,114. [2] D. R. P. 84,153. [3] A. P. 666,637.

with a body of greater specific gravity. This takes place in accordance with the law of inelastic concussions, and must therefore be accompanied by losses of energy. On the other hand, the impossibility of separating again the added substances, such as quicksilver, etc., from the steam, does not give promise of economical success for this method. For practical purposes, when the diameter of the wheel becomes too great, the only economical means yet discovered for reducing the speed of rotation of steam turbines is, and remains, the utilization of the energy of the steam by stages already discussed.

VIII

CONDENSATION

In steam turbines special attention must be paid to efficient condensation. Every reduction of pressure in the condenser here produces a much greater increase in efficiency than in the case of the reciprocating engine. The reason for this is, that in the turbine large ranges of expansion have to be reckoned with. While in the reciprocating engine a given quantity of steam expands in working to fourteen times its volume, the steam, in its passage through the Parsons turbine for instance, has to expand 140 fold. The result is the long straggling diagram shown in Fig. 275.

For the condensation of the exhaust the appliances familiar in the case of the reciprocating engine can, of course, be adopted in the turbine as well. In such case, however, they will have to be adapted to the conditions of the current which streams towards them at high velocity. Rapidly running pumps and energetically working cooling surfaces will have to be reckoned with if the condenser is not to become larger than the motor itself. Under certain circumstances superheated steam will also have to be compressed.

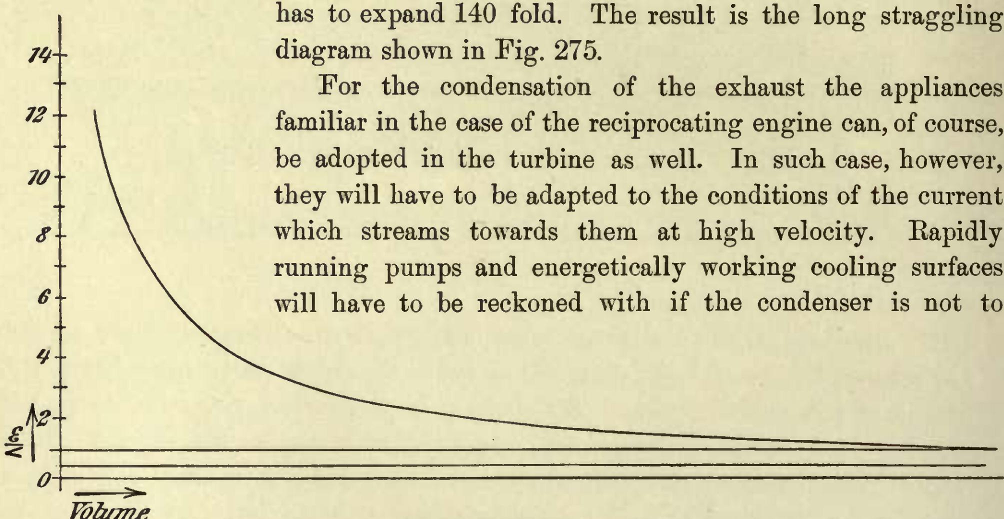


Fig. 275.

It may be observed, however, that in good systems of construction it will be impossible for dirt, in the form of cylinder oil, etc., to mingle with the steam. The condensed fluid can, then, without previous cleansing, and warm as it is, be led back into the boiler, unless, indeed, unwelcome additions be brought to it in the process of jet condensation. In this connection attention may be called to the advantageous peculiarity of the steam turbine, that in case breakdowns should occur in the condenser, the latter can be disconnected and the engine worked with the exhaust without a stoppage of work becoming necessary. For the time being, of course the advantageous utilization of the fall in pressure must be given up.

Amongst the early simple pressure wheels are to be found no less primitive jet condensers, in which the steam issuing from the steam channels is treated directly with a spray of cooling water.

For steam turbines in which the steam escapes at the periphery, *Vojáček*[1] (1896) proposes to arrange a condensing appliance in the form of a fixed pipe *d* (Fig. 276), which is provided with holes *w* opposite the points of outflow of the steam from the working wheel *c*. Water or cold air then comes in close contact with the steam. Otherwise, on the working wheel *ac* itself (Fig. 277) are fixed the mantles *g*, *h*, which are open towards the shaft *o*, and turn with it. In the direction looking towards the points of escape of the steam from the wheel *ac* holes *t* are arranged in the mantles *g*, *h*. The cooling medium ejected through the nozzles *s*

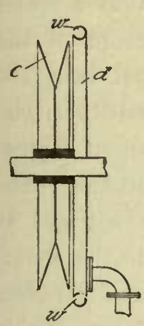


Fig. 276.

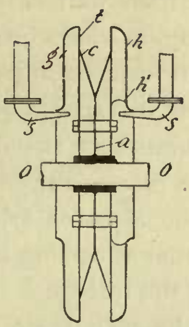


Fig. 277.

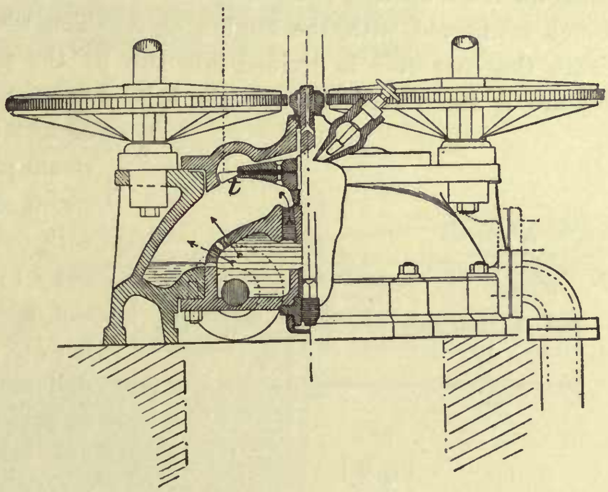


Fig. 278.

is driven outwards through the holes *t* into the air. A still more effectual spreading of the cooling medium is effected by means of the ribs or vanes h^1.

Buttenstedt and *Mewes*[2] (1900) (Fig. 278) form the lower space of the turbine chamber at once as a condenser, the double bottom being supplied with water under pressure. The steam issuing from the working wheel *t* is at once condensed by the cooling water injected in the direction of the arrows.

In the cases in which the Parsons turbine is used for driving electric current generators the pumps usually appertaining to reciprocating engines, and in this case driven by electro-motors, are used for the condensation. Fig. 279 (see Plate I.) shows a horizontally arranged condenser air-pump for a turbine of 5000 H.P., and Fig. 280 (see Plate I.) a vertical condenser air-pump for turbines of from 100 to 750 H.P., both of the Brown, Boveri-Parsons type.

The whole manner of working of turbines tends to bring about an alteration in the arrangements of the appliances for the condensation of the steam. In view of

[1] E. P. 18,807 of the year 1894. D. R. P. 92,373. [2] D. R. P. 135,333.

the necessarily rapid movements of the air and the water, the centrifugal wheels may with advantage be used as pumps.

In an earlier arrangement made by *Schiele*,[1] of which jet condensation forms a part, the centrifugal air-pumps required in connection with the latter, and the water-pump which also is provided with a centrifugal wheel, are driven by an extra turbine, which is mounted between the two working engines and on the same shaft with them.

From *Curtis*[2] (1886) comes a turbine which is driven by steam and water, and is said to be applicable as a condenser also (Fig. 281). It is then assumed that the exhaust steam, which is made use of to a certain degree in the main engine, is led away to be used directly in the driving of the condenser turbine. The steam, which for the reduction of its temperature is saturated with water, enters through a branch connected with the engine at *k*. The wheel *d* mounted on the shaft *c* within the case aa^1b is divided into two by the partition wall *e*. Water under pressure is introduced through the tube *k*; according to the adjustment of the cock *ml* it can be injected into the steam pipes in variable quantity. The steam saturated to the desired degree with water, proceeds through the channel *j* to the chamber *hf*, the water at the same time streaming into the chamber h^1f^1 of the turbine *d*. The impulses delivered by both these motive media at once against the vanes of the turbine are intended to make the wheel *d* rotate. In the further course of the steam and of the water the former becomes condensed in the annular chamber *i*; the condensed fluid then flows away to the outlet channel. As motive steam, the exhaust of a steam-engine may be used, so that the turbine thus represents a combination of a low-pressure engine with a condenser.

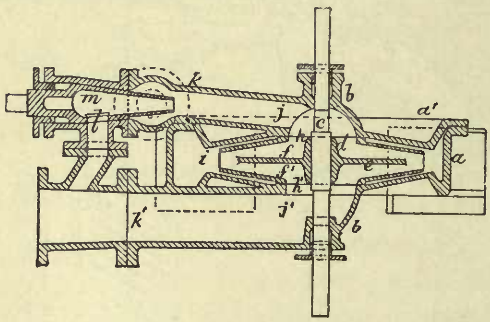


Fig. 281.

Among others, *De Ferranti*[3] also had a jet condenser in his mind. On the shaft of the turbine he places a centrifugal pump, which draws cold water from a tank and induces a small pressure jet. The latter now comes into action in an injector pointing vertically downwards, and meanwhile sucks up considerable quantities of cold water. The falling cooling water then takes up the exhaust, which is led into the injector through a pipe connected at the side of the latter. Into this exhaust pipe is introduced a non-return valve, which closes when the pressure in the injector exceeds that of the off-flowing steam.

Krank[4] (1902) places on the end of the turbine shaft a *vane pump*, which, in addition to its duty of forcing water through the bearings, has also to supply water to the condenser.

[1] E. P. 1693 of the year 1855.
[2] D. R. P. 38,266.
[3] E. P. 2565 of the year 1895.
[4] S. P. 25,914.

PLATE I.

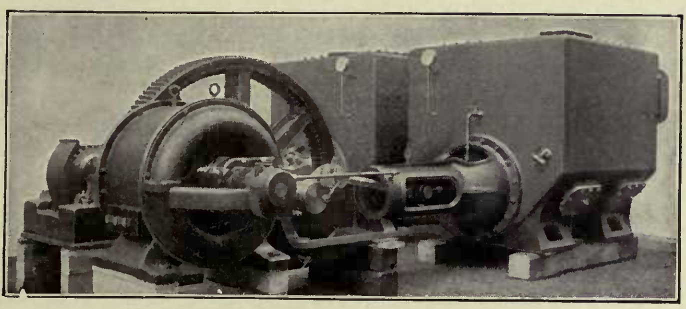


Fig. 279.
Horizontal Condenser Pump for a Turbine of 5000 H.P.

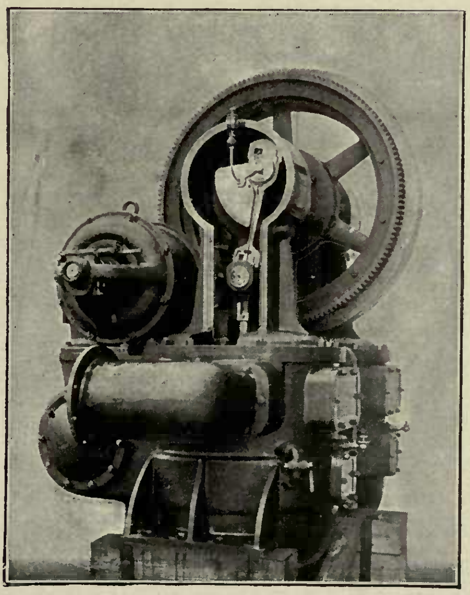


Fig. 280.
Vertical Condenser Air-Pump for Turbines of 100—750 H.P.

A style of construction which appears advantageous has been produced by *Stumpf*[1] (1901) (Fig. 282). One or more centrifugal wheels are arranged inside a closed turbine case in such a manner that the steam issuing from the vane system is admitted directly to the centrifugal wheels. Here it is condensed, and then carried off with the cooling water delivered by the last-mentioned wheels. The working wheel *a* may, for instance, be impinged upon from the nozzles *d* of an annular channel, so that the steam leaves the working-wheel vanes in a lateral direction.

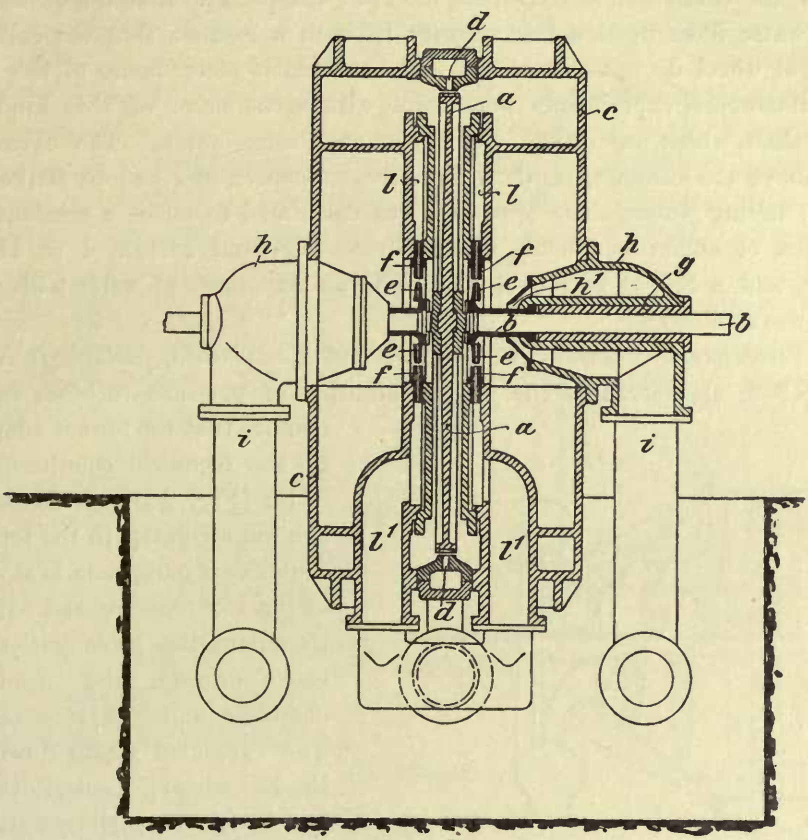


Fig. 282.

The steam has direct access, within the case *c*, to the centrifugal wheels *e*, or to the diffusers *f*. In the main the steam will come in contact with the cooling water at the point of its transition between the centrifugal wheels and the fixed vane wreaths. It will then condense between the water pistons in the channels of the diffuser. As may be seen from the figure, the bearings *g* of the shaft *b* are provided with hollow casings *h*. Through these hollow casings the water flows to the centrifugal wheels *e* of the condenser. The water is led to the latter through the two pipes *i*, and, cooling the bearings on its way, streams through the hollow bodies *h*. It then passes out through the annular openings h^1 surrounding the shaft *b*, and passing along the latter reaches the centrifugal wheels *e*. From the fixed guide

[1] D. R. P. 142,053. Compare also E. P. 14,679 of the year 1901.

vane wreaths f the cooling water proceeds, together with the water condensed from the steam, to the annular chamber l, from which it flows away through the branch l^1. The wreaths f may also, under certain circumstances, be dispensed with, and the centrifugal wheels e would then only have to effect the separation of the water, so that the resulting action would be similar to that of the jet condenser.

The centrifugal condenser is so arranged that the products of condensation flowing to it are thrown off to the diffuser, and the pressure of 1·47 lbs. per sq. in. in the condenser is converted into 14·7 lbs. per sq. in. at the outflow. The cooling water flows in at a low velocity in such a manner that the cells of the centrifugal wheel do not form vacua. In the Moabit power-house of the Berliner Elektricitätswerke experiments were made with a condenser of this kind with a vertical shaft, which was driven directly by an electro-motor. The exhaust here passed above the centrifugal wheel into the condenser, and had to strike against a veil of falling water. The apparatus was calculated to effect a condensation of 33,000 lbs. of steam per hour. It occupied a ground surface 4 ft. 11 ins. in diameter, and a height of 5 ft. 3 ins. With an abundance of water a 98 per cent. vacuum was attained.[1]

The *Vereinigte Maschinenfabrik Augsburg und Maschinenbaugesellschaft Nürnberg, A.-G.*[2] (1902) also arranges the jet condensation of pressure turbines in such a manner that the former adapts itself to the technical conditions of the latter (Fig. 283). With this view the wet air-pump, in the form of the centrifugal pump nmo, is so arranged at the low pressure end as directly to adjoin the prolongation of the case forming the condensation chamber and the chamber itself. The expanded steam flowing from the last wheel g meets, in the chamber f, with the cooling water, which is led by the pipe d into the receptacle c, and thence ejected through the holes e. A worm i mixes the water with the steam, and sends on the condensed water to the leading apparatus n of the centrifugal pump. The last guide wheel g of the turbine may be provided in the channels in its lower half with valve flaps (such for instance as indiarubber plates fitted at the back walls of the channels), their object being to prevent the entrance of the surplus condensed water into the working wheels, especially while the engine is being set in motion.

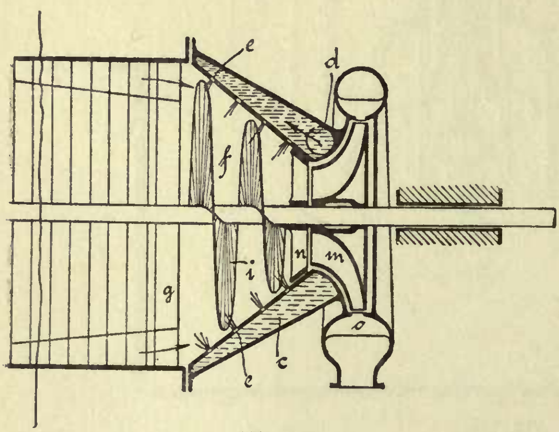
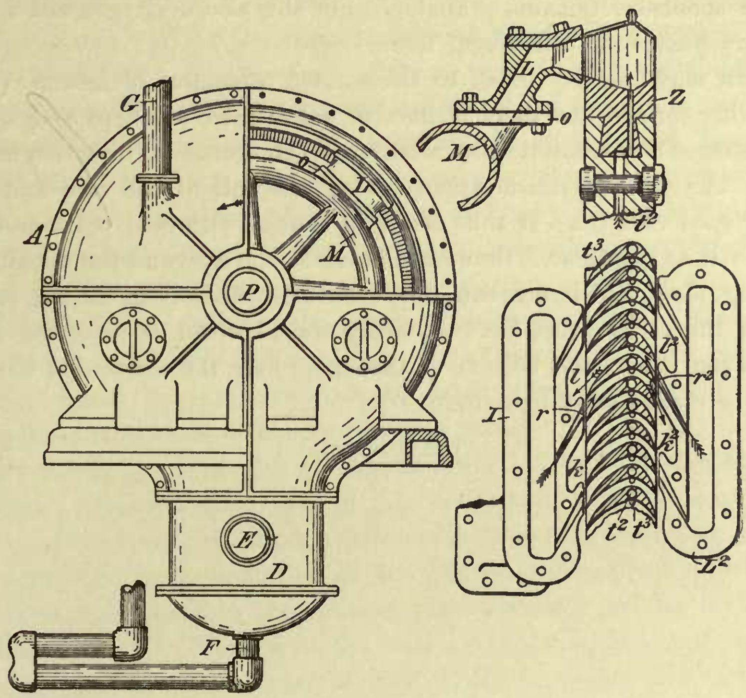


Fig. 283.

A peculiar design, in which the turbine working wheel itself takes the form of a centrifugal pump, comes from *Fedeler*[3] (1902) (Figs. 284–286). At both sides

[1] *Zeitschr. d. öst. Ing. u. Archit.-Vereins*, 1904, p. 215.
[2] D. R. P. 152,369.
[3] A. P. 700,314.

of the wheel Z are arranged the fixed leading apparatus LL^2, which are provided with expansion nozzles r, r^2. The appliances L^2, forming parts of the leading apparatus, are made to shift with the others, being set farther forward in the direction of rotation of the wheel. Also the centres of curvature of the vanes of the wheel lie nearer to these advanced leading apparatus L^2. Two hollow rings M—one for each side of the wheel—provide for the distribution of the steam to the leading apparatus. The motive medium is delivered to them through the pipes G. At the bottom of the case A a water receptacle D is provided, from which a pipe F leads to the turbine shaft P, half of which, *i.e.* the half extending to the middle of



Figs. 284–286.

the wheel, is hollow. The water can make its way through radial borings in the shaft P into the channels t^2, t^3 of the working-wheel wreath. If steam be introduced, the jets issuing from the nozzles rr^2, first meet with working-wheel vanes that are at rest. They stream through the channels and pass through the spaces k, l and k^2, l^2 lying between the nozzles into the case. As soon, however, as the working wheel rotates, the jets of steam passing from the nozzles r into the wheel channels are in part carried along by it, so that they can no longer reach the chambers l^2, k^2, but strike against the jets of steam issuing from the nozzles r^2. A damming back of the steam in the channels thus takes place. Meanwhile the water flowing in from the hollow shaft is driven outwards by the rotation of the wheel through the channels t^2, t^3, and it cools and condenses the steam that dams up the channels as above mentioned. The condensed fluid falls back into the chamber D, in which it is cooled, to be again led through the pipe F to the shaft. For the production of a vacuum within the case, an air-pump is fitted in front of the exhaust E. By means of the arrangement described, Fedeler seeks to enable the energy of the steam to be imparted in an advantageous manner to a slowly running wheel by arranging that the particles of steam shall be cooled while streaming over the vanes. This cooling process, however, is clearly tantamount to a destruction of energy, and economical working can hardly be assumed.

In cases in which economy in working requires the return of the pure product of condensation into the boiler, and where cooling water in sufficient quantity is not at command, as, for instance, under the conditions obtaining on shipboard, only the surface condenser remains available. For this also designers will seek to give the necessary pumps the centrifugal form.

Attention may here be drawn to the surface condenser of *Lyman* [1] (1876), who uses as cooling surfaces the turbine-chamber walls, which are kept cool by the outer air. He assumes a pressure turbine rotating in the horizontal direction and working in stages. The exhaust steam streams along the wall of the case and then rises to a cone-shaped case lid. It thus comes in contact with cold portions of the case, which cause it to condense. Between the lid of the case and the working wheel is a protecting plate, which prevents the condensed fluid from falling back on the wheel. On the other hand, however, every staged wheel is protected by a plate placed between it and the bottom of the case, where the condensed water collects in order to be carried away by a drain.

[1] A. P. 187,154.

IX

USE OF SUPERHEATED STEAM

THE prospect which the turbine presents of a much greater economy in working than the reciprocating steam engine at present in use shows itself, amongst other things, in the circumstance that no lubrication is required within the steam chamber, in consequence of which the use of very highly superheated steam becomes admissible. While in the reciprocating engine the temperature of the steam, in consideration of the behaviour of the materials necessary for the lubrication of the pistons, cylinders, etc., cannot at the present time much exceed 662° F., it can, in the case of the turbine, be very much increased. In the choice, then, of the materials of construction for the parts which come in contact with the superheated steam, the designer will have to be entirely guided by the influence exerted by the high degree of heat. For high temperatures bronze cannot be used for the nozzles and their valves; steel must be chosen for these parts.[1]

Amongst others, *De Ferranti*[2] intends to drive his turbine with highly superheated steam. Also *Parsons*[3] has, in his compound turbines, contemplated a superheating of the steam in the main steam pipe or in the connecting pipes.

The superheating may be carried so far that the more or less superheated outflowing exhaust has a higher temperature than the water in the boiler. In the Laval turbine *Lewicki*[4] observed, in the case of superheated steam that entered the engine at 828° F., temperatures of 280° F. in the boiler and 644° F. at the discharge respectively, so that the heat of the exhaust still represented a difference in temperature of 392° F. For the utilization of this heat in the process of working, provision must, of course, be made.

Lewicki, v. *Knorring*, *Nadrowski*, *and Imle*[5] (1900) accordingly go to work as follows (Figs. 287 and 288). The exhaust steam, which leaves the turbine while still in a superheated condition, is led through a system of heaters, which are streamed over by water or steam from the boiler. Provided the heating surface be sufficiently large, the exhaust here gives up a corresponding portion of its heat, and flows out with approximately the boiler temperature. It then makes its way

[1] Compare also J. Defays, "Actions de la vapeur sur chauffée sur les métaux industriels," *Revue indust.*, 1903, p. 478.

[2] E. P. 2565 of the year 1895.

[3] E. P. 11,086 of the year 1896.

[4] E. Lewicki, "Die Anwendung hoher Ueberhitzung beim Betrieb von Dampfturbinen," *Zeitschr. des Ver. deutscher Ing.*, 1903, Vol. 47, p. 441.

[5] D. R. P. 129,182.

through a feed-water warmer, before passing on to the condenser or escaping into the atmosphere. That is to say, the exhaust is used for generating or heating fresh steam. By means, then, of the combination of the system of heaters above mentioned with a turbine the end is attained, that the whole amount by which the heat of the exhaust is in excess of the boiler temperature is made directly useful

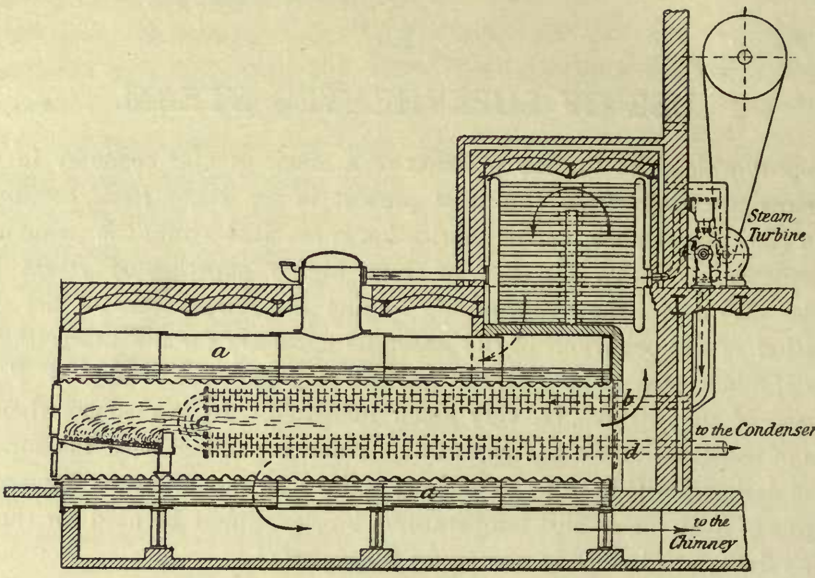


Fig. 287.

for the fresh steam, and so brought back to take its place in the working process, without being dependent upon the fall in temperature of the latter. This circumstance also makes it possible to set the upper limit of temperature very high indeed, which, as experiments are said to have shown, improves the efficiency of the engine. According to the arrangement shown in Fig. 287, the steam issuing from the exhaust pipe l is led into one of the heaters b, c, d fitted in the boiler a, and thence streams to the condenser. Otherwise it streams, as shown in Fig. 288, through the heater b lying in the water of the boiler, and then through the adjoining feed-water warmer c, after which it passes to the condenser. The heaters could also be placed in the steam space, and so on.

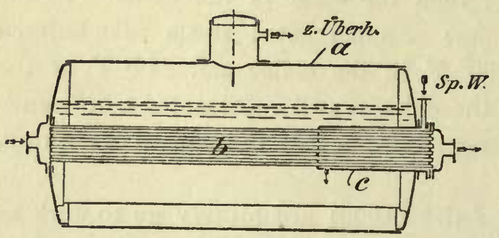


Fig. 288.

In connection with the above, *E. Lewicki* has made further proposals, one of which may, in the meantime, be picked out. The idea is to drive the first of two turbines, which work in combination with one another, with high pressure and a moderate degree of superheating (662° F.), and to superheat the steam before its entrance into the second turbine, to a high degree (930 to 1100° F.). Further, a combination with the reciprocating engine is to be driven in such a manner that the exhaust steam of the reciprocating engine is used for the performance of work in the turbine, and is superheated above the limit set for the first-mentioned engine.

In an early arrangement of his own, *Parsons* [1] proposes to prevent the steam within the turbine from being cooled below the saturation point. If carried to too great an extent, cooling would result in formation of particles of water, and therewith in a retardation of the current of the steam, and in the replacement of steam friction by fluid friction. From this arise losses in working, which can be avoided when the steam, before its entrance into the turbine, is just so much superheated that its temperature is not reduced by expansion in the turbine to the point at which particles of water are separated. The superheating may, for instance, be managed in such a manner that the steam is led through a heating coil, which lies in a

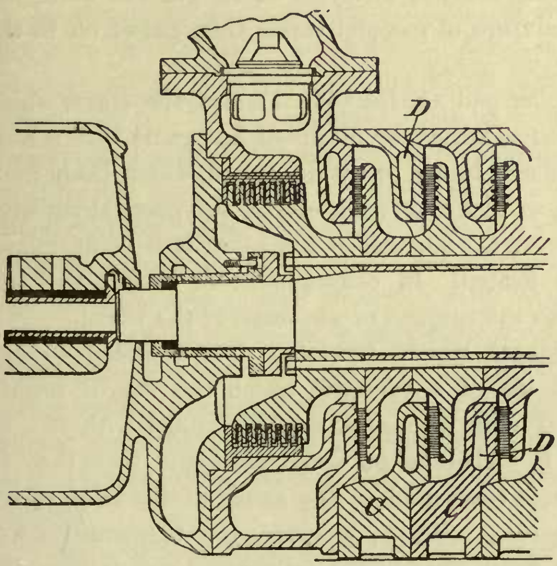


Fig. 289.

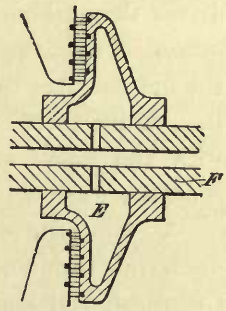


Fig. 290.

chamber placed in connection with the turbine and traversed by streams of hot gas. Without an increase of the inlet temperature the steam can also be kept so hot that fresh steam of higher pressure than the working medium may be allowed to circulate through the interconnected chambers D, arranged in the fixed leading apparatus C (Fig. 289). In place of the superheated steam hot gases or fluids could also be chosen.

An increase of the effect of the chambers D is attained by means of a special heating mantle laid round the case of the turbine. In Fig. 290 the design of a heating apparatus is illustrated, in which the working wheels themselves contain chambers E, which receive the heating medium through the hollow shaft F.

A similar arrangement is made by *Scott*, [2] whose turbine shows several working wheels placed in separate chambers. The exhaust space of the one chamber forms at once the impingement space for the leading channels of the next-following

[1] E. P. 5074 of the year 1891. [2] E. P. 22,428 of the year 1892.

working wheel, and in front of every leading apparatus is placed a heating coil, which is streamed through by hot steam, or by a hot fluid, and superheats the working steam as it flows towards the leading apparatus.

The special method of superheating, which consists in the mingling of hot gases with the fresh steam, is, by reason of the dirt thereby imparted to the latter, and in view of the loss of energy produced by the mixture of two gases of different densities, not to be recommended.

Amongst others, *Robertson* [1] contemplated a mixture of the kind just mentioned. From a nozzle under a high pressure he causes a fine thread of water to stream forth, and directs the latter towards the rather broad-lipped mouthpiece of an injector. The jet of water here streams through hot gases, which produce evaporation, and the steam thus generated, which preserves the direction of flow of the water jet, carries the gases along with it. A mixture of gas and steam then passes on to the performance of work in the turbine.

According to the plan of *Desmelles* and *Carlier* [2] (1881), also, the steam which drives the turbine wheel has to be superheated by means of hot gases. In consequence of this, the steam nozzle is placed beside or within a pipe, which leads from the fire chamber to the turbine case, so that the steam carries the gases along with it. While, then, the steam streams over the surfaces of the vanes it is being mixed with the gases, and thus becomes heated. In consequence of this it will not condense when coming in contact with the surfaces of the vanes of the wheel.

As the outcome of his experiments, which, however, were confined to one de Laval turbine, *Lewicki* [3] gives a list of advantages which the use of highly superheated steam in turbines fitted with expansion nozzles brings with it. In superheating he went up to 930° F. Thereby he established the fact that in general the gross consumption of heat (measured in the steam at the inlet valve) was reduced by increase of temperature. While, however, the temperature was increasing and the pressure of the steam surrounding the wheel was falling, the friction of the wheel also lessened. Also the circumstance that the wear of the vanes becomes reduced is to be attributed to the reduced friction of the dry, hot working steam. To this it may be added that superheating opens up the possibility of driving turbines by steam of atmospheric pressure with good steam and heat economies. Finally, the point established by *Lewicki* is of importance, that the outflow formulæ for the determination of the consumption of steam can be used as they are, if only nozzle area, pressure, and temperature be exactly measured.[4]

[1] E. P. 3146 of the year 1868. [2] D. R. P. 19,866. [3] *l. c.*

[4] Compare also Zeuner, "Technische Thermodynamik," Leipsic, 1901; Bach, "Zur Frage des Wärmewertes des überhitzten Wasserdampfes," *Zeitschr. des Ver. deutscher Ing.*, 1902, p. 729; Weyrauch, "Ueber die spezifischen Wärmen des überhitzten Wasserdampfes," *Z. d. V. d. I.*, 1904, p. 24.

X

UTILIZATION OF THE EXHAUST

THE use of the superheated steam, which, after performing its work in the turbine, is still hot, and consequently in possession of energy which may be made use of, makes it possible to keep the motive medium, free as it is from every admixture of oil, in complete circulation in the turbine. On the other hand, it greatly favours the employment of surface condensers, which may be used as evaporators for fluids of low boiling point. The attempts, then, in connection with turbine work to combine water steam-engines with cold steam-engines may be understood.

This was aimed at, for instance, in the arrangement made by *Lack*[1] (1900), in which a utilization of the available energy, to the greatest possible extent, is intended to be made possible. This power plant has a first steam turbine or group of turbines (hot steam turbines), and connected with this a second steam turbine or group of turbines, making use of the energy still retained by the exhausts of the first one. For this purpose the first-named turbine, which is fed with hot steam, is connected with a surface condenser, which is arranged for the evaporation of a fluid with low boiling point by means of the heat still retained by the exhausts from this hot steam turbine. The steam so won is used for the feed of the cold steam turbine or turbines.

This idea is further developed by *Steffen*,[2] in so far that several fluids with different boiling points are made use of, each of which can condense the exhaust steam of the turbine, before which it is inserted, and the exhausts from which, in their turn, can evaporate the motive medium of the turbine which follows it in the line of the current. This is a course of things which seems to point to perpetual motion.

Windhausen[3] (1900) (Fig. 291) sets forth the following train of reasoning. If a water-steam turbine and a cold steam turbine are brought into such dependence, the one of the other, that in ordinary work the exhaust of the former suffices to produce the cold steam for the latter, when the water-steam turbine has less work to do, and in consequence requires less steam, too little of the exhaust steam will pass off to produce heat, and a supply of fresh steam to the cold steam generator will become necessary. Again, when the work done by the water-steam turbine exceeds the ordinary amount, too much exhaust steam will be drawn off. On the other hand, with increasing or decreasing demands on the cold steam turbine in

[1] D. R. P. 127,257. [2] E. P. 5970 of the year 1900.
[3] D. R. P. 141,836.

regard to work, greater or less supplies of heat respectively will be required by it than the ordinary work of the water-steam turbine will produce.

In order to ensure uniformity, the water-steam turbine is coupled with the cold steam turbine. According to Fig. 291, compressed water steam enters the water-steam turbine A through the pipe 1, where it expands and performs work, and then makes its way through the pipe 2 into the evaporator, in which the exhaust steam transfers its heat to the cold steam fluid contained in the tubes (NH_3, SO_2, CO_2, or the like). The exhaust steam is condensed by contact with the pipes, and passes into the open air or to an air pump through the pipe 3. The cold steam, produced in the evaporator B, can either be conveyed directly to the cold-steam turbine D, or

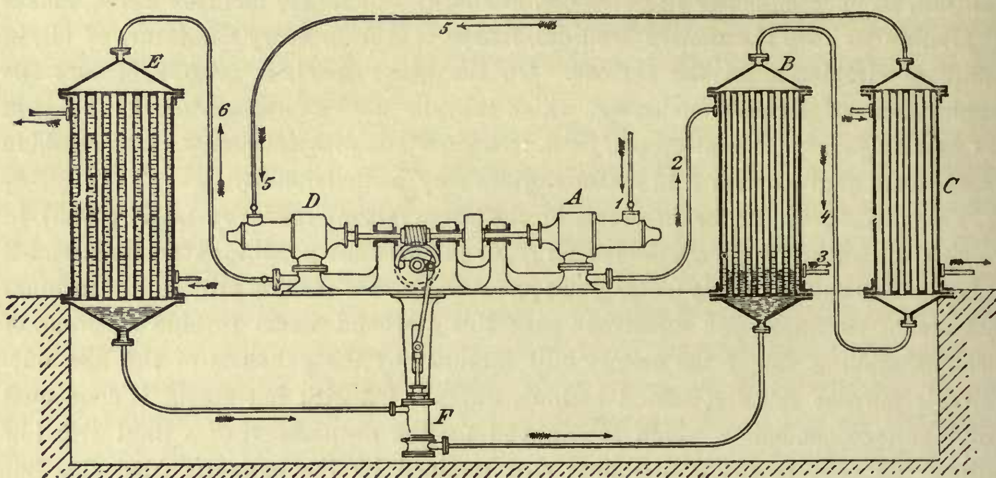


Fig. 291.

may first be conveyed to a superheater C through the tube 4, where it is superheated by means of water steam, combustion gases, or the like. The steam then makes its way through the pipe 5 into the cold-steam turbine D, which is coupled either directly or indirectly with the water-steam turbine A, that supplies the exhaust steam. The cold steam expands in the cold-steam turbine D until the condenser pressure is attained, when it passes through the pipe 6 into the condenser E, in which it is liquified by means of cooling water. The liquified cold steam is conveyed to a feed pump F, from which it is carried into the evaporator B to be again evaporized.

A peculiar domain of application has been opened up by *Rateau,* apparently with success in an economical point of view. Hauling machines in mines and the steam-reversing engines for the trains of rolls in rolling mills do not permit of a full utilization of the steam, by reason of their discontinuous style of working. The exhaust steam from these machines still contains large quantities of energy, which are not made use of, and the accumulated amounts thus wasted represent considerable losses in working. Rateau conveys this exhaust steam into low-pressure turbines —his own method of construction—in which a large amount of available work is utilized. The plan could only be carried out in connection with compensating devices, by means of which the energy which has been supplied in irregular manner by the primary engine is uniformly conveyed to the turbine. Among such devices are heat collectors, by means of which Rateau has probably overcome the difficulties in this direction. The principle on which these heat collectors are constructed is that the exhaust steam of the ironworks machines, which has about the pressure of the atmosphere, is conveyed through reservoirs filled with heat collectors consisting either of cast-iron and water or of water alone. If the steam engine at full load supplies more steam than is required by the turbine, the excess of steam is used for heating the collector masses. On the other hand, if the steam required by the turbine exceeds that supplied by the main engine, the pressure in the collector decreases, the result of which is a partial evaporation of its water.[1] Characteristic of the Rateau collector[2] is the arrangement of several reservoirs one above another in a vessel of any desired shape, so that the steam entering the vessel or condenser is caused to sweep over the surface of the fluid, or over the masses of metal contained in the reservoirs. In order to reduce the first cost the reservoirs may be dispensed with. In their place iron or cast metal parts, inserted between pairs of plates, are placed in the vessels. If t be the difference of temperature in the collector, corresponding with the variations of pressure (generally 37° to 41°) there obtaining, P, the weight of the heat collectors, having the mean specific heat C, the quantity of heat accumulated and delivered by the collector during each period will be $P \times C \times t$ calories. Consequently the condensed and re-evaporized weight of the steam is about $\frac{PCt}{L}$, L being the evaporating heat of the water.

Moreover, the low-pressure turbine is made to work independently of the primary engine by means of a valve attached to the collector, which maintains a certain degree of pressure, so that fresh steam can be conveyed to the conduit pipe leading to the turbine when the pressure decreases.

As we here have to do with the exhaust steam of piston engines, the accumulator has been connected with an oil separator. It has been proved that in working with low-pressure turbines great uniformity can be produced, provided sufficient collector masses are at disposal. The favourable views entertained in regard to the Rateau process are supported by the results obtained with the plant which has been in operation at the Bruay mines since August, 1902. Here a hoisting machine of 174 H.P. has been erected, the exhaust steam of which, at a pressure of about 12·8 lbs. per sq. in., is reduced to 2·13 lbs. per sq. in. by condensation. Commensurate with this drop in pressure, an additional 300 H.P. is made available for work in the turbine, and this efficiency can be increased to 450 H.P. when the hoisting machine is working at its full capacity. Collectors are used in the Bruay mines containing 40 tons of cast-iron, besides water. Rateau succeeded in varying the temperature in the collectors by from 37° to 43° only, which corresponds with a variation of the pressure of from 2·1 to 3·6 lbs. per sq. in. The turbine is directly

[1] *Le Génie civil*, Vol. XLIV., p. 293.

[2] D. R. P. 125,117 and 153,376.

connected with two direct-current dynamos, and there is also a connection with the main steam pipes, from which fresh steam with correspondingly reduced pressure is derived, providing the hoisting machine be not at work. According to Rateau, the consumption of steam, per electric H.P. per hour, in general varies between 26 and 37 lbs., provided exhaust steam at a pressure of 14·7 lbs. per sq. in. be at disposal.

At the Béthune mines (Pas-de-Calais) it is proposed to combine a turbine of 250 H.P. with a hoisting machine in order to drive a centrifugal air compressor which will supply air compressed to 13·2 lbs. At the Mines de la Réunion de la Société des chemins de fer de Madrid (at Saragossa, in Spain) a hoisting machine and a pumping engine deliver their exhaust steam into collectors for the supply of three turbines of 350 H.P. each, which are intended to drive 3-phase alternators of 220 kw. each. The steel works at Donetz are to be provided with three turbines, each of 750 electric H.P., which use the exhaust steam of two reversing machines. A plant of 400 H.P. is contemplated for the Mines de Fontaine l'Evêque (Belgium), etc.[1]

Attention may be drawn to the fact that the utilization of Rateau's invention in Germany has been transferred to the Kommanditgesellschaft Balcke and Co. of Bochum.

[1] *La Nature*, 32, p. 231.

XI

INTERNAL RESISTANCES

THE internal resistances of turbines can be divided into two groups. One of these, in addition to the resistances due as a matter of course to the imparting of energy, also comprises those which block the working steam. The other group comprises the resistances to which the revolving working wheels are subject.

As regards the steam, its friction on the surfaces over which it sweeps is to be reduced to a minimum by the careful smoothing of these. Special attention will have to be devoted to the inner walls of the expansion nozzles, since slight variations will result in very great losses, owing to the very rapid conversion of the pressure into velocity. Further, the surfaces of the vanes must not give rise to eddies, whether they be swept over by compressed steam of low velocity, or by pressureless steam of high velocity. For this purpose the surfaces of the vanes must be properly formed, so as to ensure that their points of entrance and exit, as well as their intermediate parts from the beginning to the end, are free from impact surfaces, besides being smooth in the direction of flow of the current. This condition must, of course, also be maintained in working. Consequently, if wet steam be used, a material of construction not subject to rust (preferably bronze) should be employed, whilst for considerably superheated steam, steel should be used. Additional improvements, such as those proposed by the *Société Maison Breguet* in 1894,[1] including the polishing of the outer sides of the vanes, and the nickel-plating or gilding them, entail expense that is out of all proportion to the advantages they bring.

The impact of the steam at the entering edges of the vanes forms a far greater source of loss than the friction. Each front edge of the vane affords a surface of impact, and the attempt has frequently been made to reduce the proportion borne by the entire surface of impact to the cross section of the current of steam by the adoption of a fewer number of vanes. This method does not, however, produce much result, since for the proper guidance of the steam the extensive sub-division of the current is a special requirement. In practice it does not seem advisable to make the edges very sharp when the velocity of the steam is to be very high, as in such case even steel blades will soon have their edges broken. On the other hand, it is of great importance that the edges be well rounded off, in order, especially in the case of pressure or reaction turbines, to prevent the contraction of the steam.

Owing to the gaseous nature of the motive medium, and the great speed of

[1] F. P. 237,267.

working, centrifugal forces, due to the possible changing of the direction of the steam, also play an important part. These forces tend to produce a compression of the steam, followed by a corresponding expansion after the jet has been reversed. This alternating play will result in losses of energy, which can be avoided or reduced if the change of direction be kept small, or if the pressure of the steam be not allowed to sink below a certain degree, which neutralizes the influence of the centrifugal forces.[1] This can especially be observed in the case of pure velocity turbines.

The steam leaving the working wheel must have sufficient energy left to carry it to the exhaust or to the condenser, *i.e.* to enable it to overcome the resistance in the exhaust pipe. This amount of energy means a loss of useful work, and should therefore be reduced to a minimum. For this reason, *Parsons* and *Wass*,[2] amongst others, have attached the (surface) condenser directly to the last stage. The steam can thus pass freely into the condenser by the shortest possible way.

Frequent attempts have been made to arrange a special condenser in the exhaust in such a manner that the vane wheel fixed on the turbine shaft conveys the escaping steam to the outlet (*Vojáček*[3]). Such means can, however, hardly be considered as improvements, since the ventilator must exercise a retarding force on the shaft of the machine at least proportionate to the increase of energy obtained from the steam in the turbine.

An arrangement which can be compared with the effuser or diffuser of the water turbine[4] is used by *Terry*[5] (1897) (Fig. 292). In order to utilize an additional part of the energy of the steam leaving the cells of the working wheel *b*, which is axially impinged upon by the nozzles *a*, Terry inserts behind the wheel, and in front of the exhaust chamber proper, in which atmospheric or condenser pressure obtains, an annular chamber *c*. This annular chamber is enlarged in the direction of flow of the steam, which, leaving the working wheel with a certain kinetic energy, passes onward to be consumed in the space *c*. Thus the steam does not enter the exhaust chamber proper directly from the working wheel, but must first expand in order to overcome the resistance of the external pressure of the atmosphere, or of the pressure in the condenser. It has thereby been assumed that the steam, after having performed its work in the wheel, still possesses sufficient energy to take it back to the condenser. Terry's construction apparently also gains a certain practical importance, when the annular chamber *c* is made to take up the exhaust steam immediately after it leaves the working wheel, and carry it away from the latter, so as to prevent a retarding counter-circulation around the wheel.

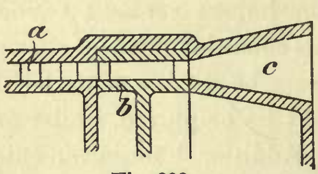


Fig. 292.

De Laval,[6] it may be observed, proceeds in a similar manner (Fig. 293) when he tries to dispense with the special condenser, and uses expansion nozzles for the

[1] See p. 97, and under "Leading Apparatus."
[2] E. P. 5881 of the year 1899.
[3] E. P. 18,807 of the year 1894.
[4] See Zeuner, Th. d. Turb., 1899, p. 176.
[5] A. P. 636,867.
[6] E.P. 20,603 of the year 1891.

exhaust steam. He uses the steam issuing from the working wheel for the production of an effective vacuum around the wheel, firstly in order to reduce the friction between the wheel and the medium surrounding it, and secondly to increase the effect of the motive medium as it sweeps over the working wheel by a further expansion. In Fig. 293, *a* represents the expansion nozzle, in which the fresh steam in its passage to the farthest point *b* expands isentropically to the pressure existing in the turbine chamber *c*. As the steam passes the vanes of the working wheel *d*, its full velocity is reduced, partly by friction, and partly by its conversion into work. On the steam outlet side an exhaust nozzle *e* is connected with the working wheel, the axis of which lies in the direction of the jets of steam issuing from the wheel, and the entrance *f* of which forms an annular section which is sufficiently large to receive all the steam, including that which has been dispersed by the wheel during its rotation.

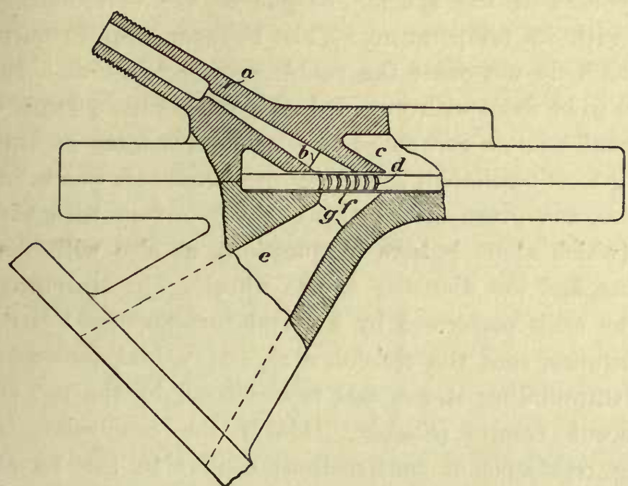


Fig. 293.

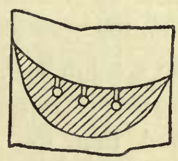


Fig. 294.

The narrowest passage *g*, however, is just wide enough to ensure the convenient passage of the steam admitted. Following the contraction *g* comes a uniform enlargement of the exhaust nozzle. From the steam entering this enlargement, De Laval expects an injector-like action on the turbine chamber *c*. According to the results of his experimental trials, this device, as compared with the production of a vacuum by condensation in the ordinary manner, is said to have proved itself an economical one.

A certain resistance is offered by the water separated by non-superheated steam while giving off its energy and carried away by the current, especially when the latter is led through expansion nozzles, in so far as such water is not again evaporated during the application of heat which accompanies the process of conversion. The re-evaporation, and also the movement of the particles of water, of course represent losses of energy. The separation of water can be prevented by the application of heat from the outside (see Section X). Where, however, waste water accumulates in the driving machinery, it must be removed as quickly as possible.

Dodge[1] (General Electric Company) (1903) even goes so far as to make the vanes, whether of the guide wheel or of both wheels, hollow (Fig. 294), and to

[1] A. P.,741,776.

provide the surfaces, over which the steam sweeps, with fine perforations, which run into the hollow spaces at the points at which the direction of the steam jets is changed. Thus, the particles of water are to be driven into the hollow spaces by centrifugal force. Suitably arranged collecting pipes lead off the water which has accumulated. This mechanical method of drying is effected by steps, and proceeds at the same rate as that in which water is separated.

As regards the resistance produced by the working wheel itself, the main point to be considered is the friction with the surrounding medium. In this case, attention must be paid to the well-known phenomenon, that the gas friction is independent of the pressure, and consequently also of the specific weight of the surrounding medium, but that it increases with its temperature. This, however, applies only when smooth bodies rotate, which do not cause the rubbing gases to move. In the case of turbines, wheels have to be dealt with provided with vanes which, according to their forms and arrangement, exert a greater or smaller motive force on the surrounding medium. By this means *ventilation resistances* are produced, which, in the case of wheels of similar form, and given arrangement of vanes, vary with the specific weight of the steam (which alone is here in question), as also with the number of revolutions per minute, and the diameter of the wheel. The searching tests made by *E. Lewicki,*[1] on the work performed by a Laval turbine of 30 H.P. when running free show, for instance, that the resistance at empty load increases with the specific weight of the surrounding steam, and is decreased by the use of superheated steam. From recent results (*Stodola,*[2] *Odell*[3]) the conclusion is warranted that the ventilating resistance of constructions similar to the Laval wheel increases with about the third power of the number of revolutions per minute, and with the 2·5th–3rd power of the diameter of the wheel.

The production of a vacuum in the turbine chamber is, of course, of great importance, as it also results in the reduction of the ventilation resistance. The attempt has been made to diminish the latter as far as possible by an arrangement of the vanes, such that their outside surfaces are smooth. As, however, working channels more or less open to the outside must be provided for the steam, the performance by them of injurious resistance work will be inevitable. Under equal conditions, the ventilation resistance of a wheel will be at a maximum when it is made to rotate at empty load in the direction opposed to its working load, as for instance in the case of reversing turbines with wheels running in both directions. In this case the wheel running at empty load must be in as perfect a vacuum as possible. For this reason the chambers of the wheels running at empty load are also placed under condenser pressure.

Vizet[4] (1899) (Fig. 295) likewise proposes to let the wheel, while at work, run with steam devoid, as far as possible, of pressure. The steam introduced into the turbine case by the branch *e* enters the annular nozzle in the case section *c* without reduction of pressure, to effect which the cross section of the nozzle chamber diminishes

[1] *Zeitschr. des Vereins deutscher Ingenieure*, 1901.
[2] Berlin, 1904.
[3] *Engineering*, 1904, Vol. LXXVII., p. 30.
[4] D. R. P. 116,512.

in the direction of the current. By means of the partition walls d arranged in the annular nozzle chamber with given curvature, spacing, and angles of inclination, the steam is divided into single jets, which leave the annular nozzle at an angle of only about 12° with the plane of rotation, and by virtue of the conservation of the initial or maximum pressure, and of the velocity at which the outflow takes place, act with full energy on the wheel a. As the said annular channel expands on the side directed towards the case section, and the cross section of each cell space between

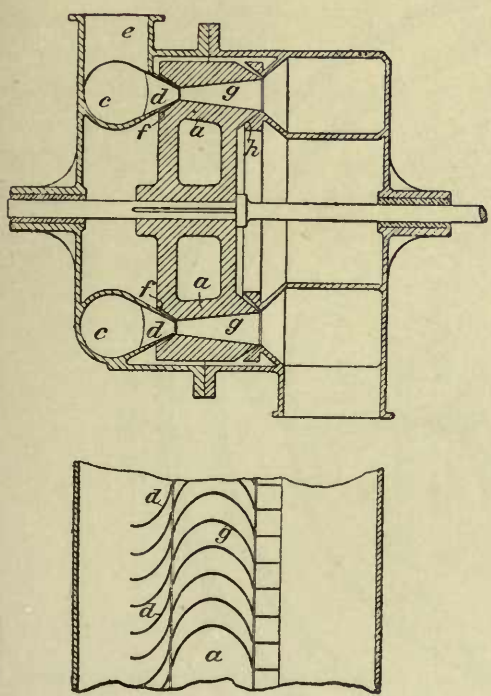


Fig. 295.

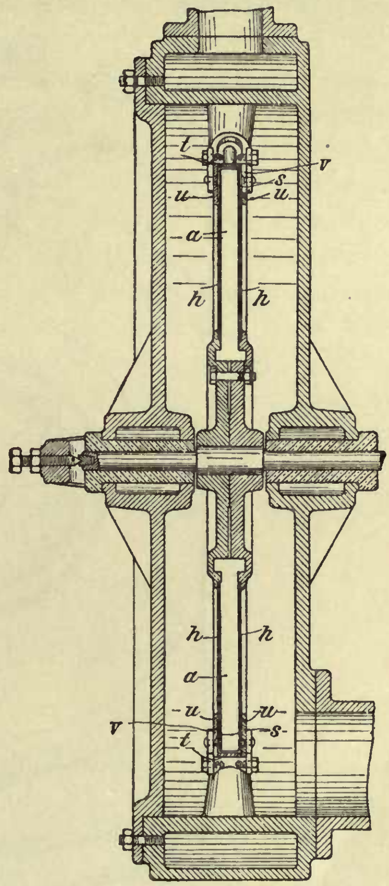


Fig. 296.

the vanes g increases from the entrance point onwards, an expansion of the steam passing through the wheel a will take place. The overlapping of the inlet nozzle by the wheel a (at f), and the engagement of the annular projection at the outlet side of the wheel a into the circular groove h in the case section, prevent losses of energy between the wheel a and the two case sections, and produce an injector-like action, by means of which the air, intermingling with the steam, is drawn from the turbine case, and a counter current round the turbine wheel is prevented.

To obviate suction between the radiating bars a *Veith*[1] (1899) (Figs. 296 and

[1] D. R. P. 112,724.

297) has closed the parts of these which do not act as vanes, by the attachment of thin discs h at the sides. This has been improved upon by the *Actiengesellschaft der Maschinenfabriken von Escher, Wyss and Co.*[1] (1901) by the employment of the same outside diameter for the lateral discs h as for the wheel, and by the arrangement, at the periphery, of openings for the outlet of the steam or gas. The same company has attached to the turbine case flat rings u (Figs. 296 and 297) connected by a ring t surrounding the working wheel at its circumference over its entire width,

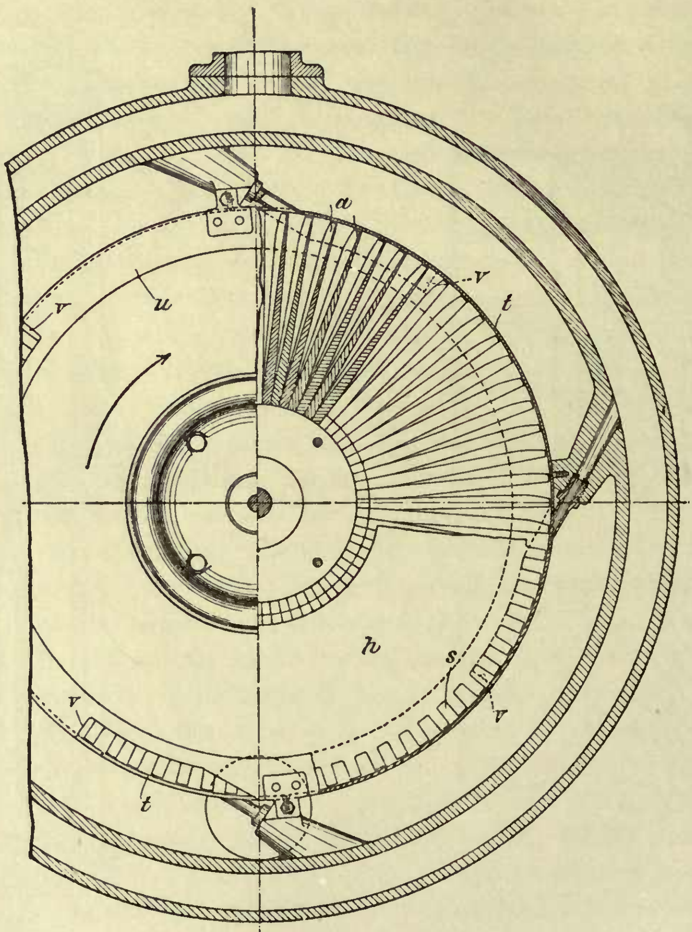


Fig. 297.

for the purpose of closing the clearance space s of the lateral discs h of the working wheel.

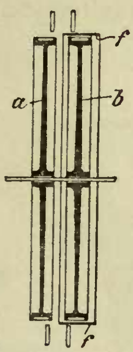
These flat rings u only contain openings v for the out-flowing motive medium, which, in case of partial impingement, will enable it to pass through at the impinged spots only.[2]

If only one disc, that is smooth on the outside, be allowed to rotate in the steam, the frictional resistance will be insignificant in amount, and a reduction of the

[1] D. R. P. 128,605. [2] D. R. P. 136,681.

steam pressure for the purpose of lessening it need not be resorted to. Amongst the different constructions of working wheels which have been made public, the smooth disc with tangentially arranged pockets at the circumference will, in the direction corresponding with the working one, no doubt have the lowest ventilation resistance. In case of empty load and motion in the opposite direction, when the reduction of the resistance in this condition is the object aimed at, the pockets which are open in the direction of rotation must necessarily produce an increased resistance. At any rate, the withdrawal of the pressure from the chamber will, when the wheel is running at empty load, become a necessity.

If the wheel running at empty load, or at least its channels, could be closed against the surrounding chamber, the ventilation resistance would clearly be reduced to a minimum. The method of steam supply, however, does not admit of such closing by means of the casing in an effectual manner. *Müller*[1] (1903) attempts to overcome this difficulty by uncovering the vane wreaths, which are in motion but not performing work, by means of movable ring slides, or other similar appliance. In Fig. 298 the ring slide *f* is shifted to a position exactly over the wheel *b*, which is running free. The assumption has here been made that the wheels *a* and *b*, which are intended to run in both directions, possess U-shaped pockets serving as steam channels. If an axial turbine of the de Laval type



Figs. 298.

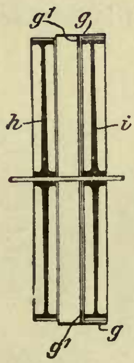


Fig. 299.

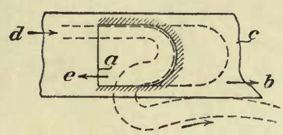


Fig. 300.

requiring lateral protection, be in question, the Müller device may then consist of a ring *g* (Fig. 299) of T-shaped section, with its flange g^1 lying between the wheels *h* and *i*. This flange g^1 will then also close the steam channels axially at one end, while at the other end they will have to be covered by a separate flat ring. This method of uncovering may be employed both for reversing turbines and for empty-running wreaths of stages that are to be disconnected.

Finally, the alternating action between the exhaust steam and the working wheel remains to be considered. In relation to this *Stumpf*[2] (Fig. 300) makes a statement somewhat as follows:—

"If a jet of steam be made to enter the cell *a* of the wheel *c* running in the direction *b*, its direction of flow will be reversed by the cell, so that the jet of steam *e* issuing from the latter has the tendency to stream in the direction opposed to that of the working wheel, and thereby of course to produce great resistance to its motion."

[1] D. R. P. 149,811.

[2] E. P. 14,155 of the year 1901.

Stumpf estimates the amount of the losses produced by the frictional resistances at twenty per cent. of the entire losses of energy occurring in the turbine. In order to avoid these sources of loss, he places opposite to the cell in the working wheel other cells in the concentrically arranged guide wheel for the reception of the jets of steam issuing from the cells *a* of the working wheel. These reverse the direction of the steam again, so that it now streams in the direction of the working wheel *c*, thereby helping the latter to rotate by means of the friction set up on one of its sides.

Amongst others, *Valley*[1] (1896) has attempted to avoid such counter currents around the working wheel by surrounding the internally supplied radial (reaction) wheel with a ring, which almost touches it, and which is provided with openings for the passage of the steam. Such a construction, however, is likely to offer resistance to the flow of the steam through the wheel.

The endeavour of *House*[2] (1897) to prevent the working wheel from producing a vacuum, by causing the steam jets of a reaction wheel, issuing in a direction tangential to the latter, to suck in air in regulated quantities, which, together with the exhaust steam was to be discharged at the periphery, seems to be a failure. The air is sucked up by the wheel, which it is intended at the same time to cool.

[1] A. P. 590,247. [2] A. P. 593,219.

XII

GOVERNING APPLIANCES

SINCE a distribution of the steam, such as is necessary in reciprocating engines, can be dispensed with in turbines, it can in connection with the governing appliances also be left out of account. The regulation of the supply of steam is effected by a direct influencing of the motive medium streaming into the working chamber.

Since the medium streams very rapidly through the turbine, any change made in the quantity of energy very quickly makes itself felt.

In regard to the governing of the *power* developed by the turbine, the simplest process is that of narrowing or widening the steam inlet. This *throttling* of the

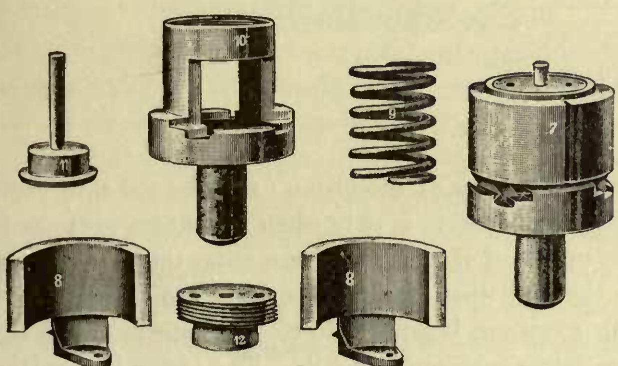


Fig. 301.

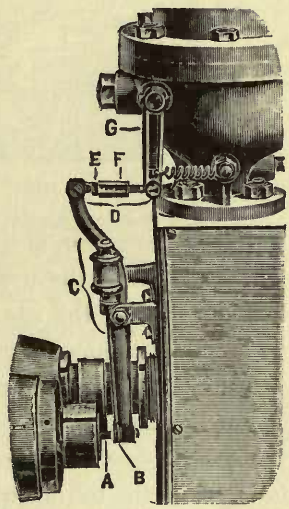


Fig. 302.

steam is, however, accompanied by an alteration of the admission pressure, and thus also of the velocity of the steam in the turbine itself, a circumstance which has an unfavourable influence on the proportion borne by the steam passage to the forms and positions of the vanes.

In the *de Laval* turbine which is being made in an up-to-date manner by the *Humboldt Engine Works,* the throttle system of regulation is applied, an axial governor being made to actuate a throttle valve. In Fig. 301 the different parts of the governor are illustrated, while Fig. 302 shows its arrangement with reference to the engine, and the actuating gear of the throttle-valve. In case of a too rapid rotation of the shaft which bears the governor, the pin A is pushed outwards, so as to act against a spring. The pin bears against the eye B of the double lever C, which, in turning, moves the crank G by means of the parts D, E, F, the lengths of which are adjustable. The crank G is held back by a tension spring. It actuates in its turn the inlet valve which is mounted on the case of the engine.

According to the arrangement made by the *Maschinenbau-Anstalt Humboldt*[1] (1899) (Fig. 303) the steam-pressure governor only allows steam of a certain fixed pressure to enter the turbine. It contracts or entirely closes the steam-discharge opening from the channel *a* to the steam nozzle *b* as soon as the pressure in the channel *a* has sunk to a certain minimum. When the pressure rises to, or exceeds, the minimum, the governor either widens the opening, or opens it to the full. The governor consists of the pointed cone-shaped end-piece *d* of a piston-rod *e*, which extends into the opening of the nozzle. Fixed at the other end of the rod *e* is a piston *i* guided in a cylinder which is in connection with one side of the steam channel. On the back of the piston presses an adjustable

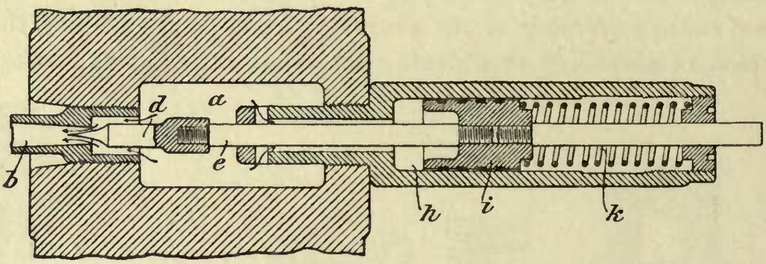


Fig. 303.

spring *k*, the action of which tends, by means of the piston *i* and the rod *e*, to press the cone *d* into the nozzle opening, so that it is only when the steam pressure is equal to that of the spring, or in excess of it, that steam can enter the nozzles. In case of fluctuations occurring in the load, when the latter is eased, the throttle valve will, under the influence of the governor, begin to close. Less steam enters the channel *a* and the pressure in the latter consequently falls. The piston is eased on the inflow side, and the spring *k* moves it along (in the illustration towards the left), while the entrance opening *c* of the steam nozzle becomes smaller and smaller. The turbine will then run more slowly, the governor in consequence fall away, and the throttle valve become opened. The pressure now rises again in the turbine; the piston *i* retreats; the nozzle passage widens and allows more steam to enter the engine. After this process has been repeated a few times, the nozzle-closing appliance gradually adjusts itself to the load, and its action tends to balance it. The process of working with throttled steam, *i.e.* with steam of reduced pressure, ceases, because considerable fluctuations in the number of revolutions of the engine and consequent falling back of the governor no longer occur. Only comparatively small differences of pressure now come in question in the channels *a*. In practice

[1] D. R. P. 111,493.

it proves desirable to fit several automatically working nozzle-closing appliances of this kind to a turbine case; for instance, about two with their spindles cut straight at the ends, which thus are completely open or closed by small movements of the piston, and one the spindle of which ends in a conical point (as in the illustration), which, with its variable width of opening of the steam entrance, provides for the intermediate stages of the two first ones. In this way only a quantity of steam, which is of approximately constant pressure, and which corresponds with the degree of load, is supplied to the engine.

The *Société Maison Breguet*[1] (1894) has preferred to regulate the steam of a de Laval turbine in such a manner that the impingement nozzles are successively closed and opened in accordance with the work required of the engine. The nozzles should therefore immediately adjoin a general annular steam chamber encompassing the (radially impinged) working wheel. The admission of the steam to this chamber would be influenced by a throttle valve actuated by the governor, so that the steam pressure in the annular chamber would fluctuate. The nozzle entrances were to be closed by valves, the spindles of which were loaded to different degrees by springs having the tendency to close the valves, while the steam tended to open these. In proportion as the steam pressure became weaker in the annular chamber, by reason of the throttling of the steam inlet by the governor, the nozzle valves had successively to be closed by their springs.

Olsson[2] (1901) completes the regulation of the de Laval turbine by means of a safety appliance, which comes into action in case a further increase in the revolutions of the working wheel should take place in spite of the governor having exercised its influence on the throttle valve. For if the engine has been completely relieved of its load by the action of the governor in closing the steam valve, and steam should for some reason or other continue to find its way through, it can bring about a dangerous speed of rotation of the wheel when the latter runs empty under condenser pressure, *i.e.* with small resistance. Olsson accordingly arranges on the turbine case an air-valve, the axis of which coincides with that of the governor. Should the increase in the revolutions take place after the load has been taken off, the pin of the governor goes beyond the position corresponding with the closure of the steam-valve, and opens the air-valve. Atmospheric air can then enter the turbine chamber, which, by reason of the ventilation resistances which now ensue, produce a retardation of the working wheel.

This appliance will indeed be able to exert an effective influence only in case the turbine fitted with it has its own condenser. As soon as central condensation is provided for several engines, the entrance of air into a turbine chamber will hardly produce an appreciable increase in the resistance of the wheel. For this case, then, *Olsson*[3] (1901) gives up the admission of air. In place of it he inserts in the transmission channel to the condenser a throttle flap, the axis of which can be turned by a piston. Under ordinary circumstances this latter is held, against the pressure of a spring, in such a position that the throttle flap holds the discharge

[1] F. P. 237,267. [2] A. P. 701,500. [3] A.P. 705,124.

pipe open. In order to effect this, the condenser is made to place the chamber shut off by the piston and its cylinder under a correspondingly low pressure. Should, however, as in the first instance, the governor open the air-valve when the number of revolutions has, after the closing of the steam-valve, been increased, the in-streaming air does away with the under-pressure in the chamber before mentioned below the piston of the throttle flap, and the latter is then closed. The unintentionally *inflowing steam* then fills the *turbine chamber*, so that the steam forms the resisting medium.

The closing of the lead to the condenser becomes a matter of importance when, for instance, the check apparatus, by reason of wear or of the presence of dirt in way of it, no longer closes quite tightly. In view of the latter eventuality, care

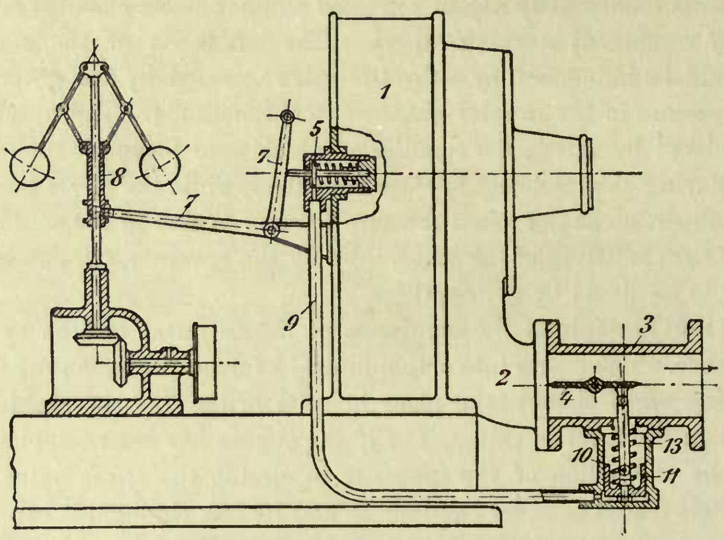


Fig. 304.

must be taken that this safety appliance is made very reliable in its action. The *Aktiebolaget de Lavals Angturbin* (1903), in view of this, calls in question the arrangements by which the chamber below the piston actuating the throttle valve is connected, on the one hand, with an air valve, and on the other with the condenser, and a spring is instrumental in the closing of the valve. Considering that at the starting of the engine there is no vacuum in the condenser, it is necessary that the throttle valve be kept open by an outward appliance, because it would, without this, be shut by the spring. If then, after a sufficient vacuum has been produced in the condenser, the throttle valve has, by mistake, not been set free, the whole air-valve arrangement fails to act as a safety appliance. Even in case the vacuum in the condenser should for a short time accidentally become bad, the throttle valve would be influenced by the spring, and the steam outlet be closed. The Company, indeed, also provides for a governing appliance (Fig. 304). In this there is fitted, in the condenser lead 3, a throttle valve 4, which, by the help of the air introduced into the cylinder, influences a piston 11 that is situated in the cylinder 10, and is loaded by means of the spring 13 or by means of a weight. Meanwhile the cylinder spaces on both sides of the piston 11 stand in connection with one another, and the spring or weight opens the throttle valve instead of closing it. By this means the throttle valve is made independent of the pressure in the condenser lead[1]; moreover, the air is introduced through the pipe 9 at the under side of the piston 11, when the regulator 8, by means of the lever 7, opens the air valve 5 arranged in the turbine case 1.

In connection with the foregoing, the method adopted by *Hellweg* (1902) may be referred to, the aim of which is to prevent the pressure and temperature in the turbine from falling when the governor is applied. The governor is here designed

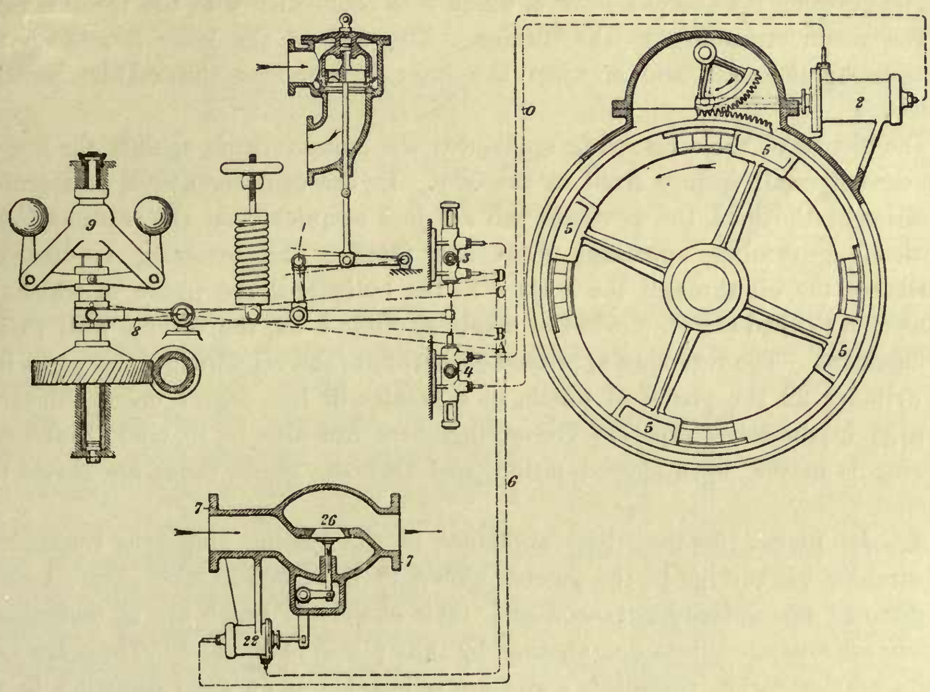


Fig. 305.

only to throttle the exhaust, so that the exertion of power by the engine is reduced only by the increase of the back pressure.

The automatic governing appliance adopted by *Rateau*[2] (1901) (Fig. 305) is characterized by the circumstance that, side by side with the familiar regulation by means of the constant pressure of the governor on a throttle valve, an additional regulation takes place in the end positions of the governor. This is of such a nature that, in case of a reduction of the load on the turbine, the governor, by means of special reversing appliances, closes covers fitted over the guiding vanes of the turbine, and, in case of an increase of the load, opens the check apparatus of a relief pipe. This admits the extra steam into the turbine. The new governor works in

[1] D. R. P. 152,476. [2] D. R. P. 143,618. Compare also S. P. 25,548.

the three stages AB, BC, and CD. In the middle stage BC, which can be altered at will, the governor works in the familiar manner only on the steam-supply valve, and regulates then only by throttling. In the stage AB, which corresponds with the greatest speed of the governor, the latter further closes the steam-supply valve, but at the same time acts on a reversing appliance, by means of which a motive fluid (oil, steam, or the like) can pass into an auxiliary machine, which opens and closes the covering (or several coverings) of the turbine vanes. In the stage CD, corresponding with the lowest speeds, the governor also acts on a reversing appliance, by which the motive fluid passes to an auxiliary machine, and opens or closes the check appliance of the relief pipe.

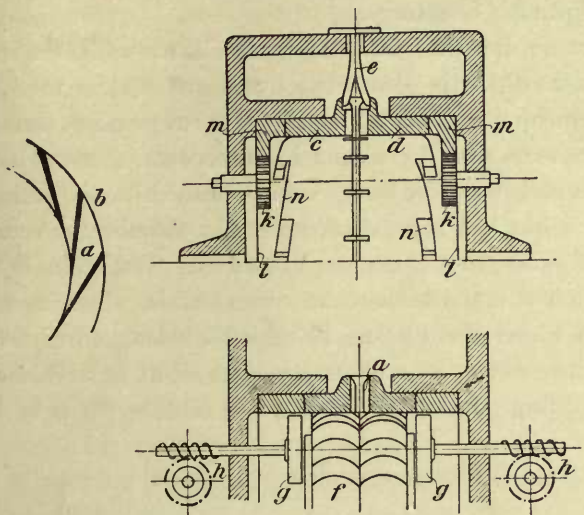
The governor 9 actuates a lever 8, which is in connection with the throttle valve for the steam streaming to the turbine. The end of the lever 8 acts on the reversing appliances 3 and 4, when the lever 8 oversteps the middle position C or B.

The pistons of these reversing appliances are brought back against the lever 8 by means of small springs fixed in the case. In the case, then, of a considerable alteration of the load, the governor can act just as quickly on the valve as if the additional governing apparatus were not fitted. The reversing appliance 4 distributes the oil through the leads *o* to the cylinder 2, the piston of which, by means of a toothed sector, moves the revolving slide 5 for the turbine with partial impingement. The reversing appliance 3 distributes the oil through the pipes 6 to the cylinder 22, the piston of which, to a greater or less degree, opens the check valve 26 in the relief pipe 7. The arrangement can also be so made that every covering is moved by a special piston, and that the guide vanes are closed one by one.

Schiele[1] places the throttling appliance in the leading apparatus themselves, and arranges his turbine in the manner shown in Figs. 306 to 308. The nozzles *a* are directed tangentially outwards, and their chambers, which are of rectangular section, are separated from one another by thin pieces of metal *b*. These are fixed in the nozzle-bearing ring-piece *c*, and are adjustable, in an axial direction, in the slits of the other nozzle ring *d*, so that the nozzle widths can thereby be altered, and that the nozzle rings *c*, *d* can be displaced in the axial direction, *i.e.* in the direction of the axis of the working wheel, towards or away from each other. A spring *e* (or several such) tends to keep the rings *c*, *d* apart. To bring the rings together, the following arrangement is made. A Schiele working wheel *f* (compare p. 59) is assumed, which is said to work best when the jets of steam leave the working-wheel vanes exactly in an axial direction. In this case no turning action is exercised on the small turbine wheels *g*, which, on the contrary, are held fast. The turbines *g* are first turned in the one or the other direction when the jets of steam assume a different direction. Their turning motion is communicated, by means of the worm gearing *h*, to the driving rods *k*, which engage into toothed segments worked on the rings *l*. The rings *l* are secured against axial displacement,

[1] E. P. 1693 of the year 1855.

but they bear with wedge-shaped surfaces n against corresponding surfaces of the nozzle rings c, d. From this it follows that the rotation of the rings l in the one direction causes a displacement of the nozzle rings c, d towards each other, *i.e.* a narrowing of the nozzle sections, while their rotation in the opposite direction allows the



Figs. 306–308.

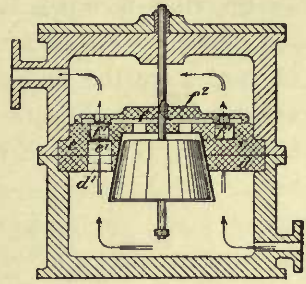


Fig. 309.

spring e to push the rings c, d apart, and thus to increase the widths of the nozzles. Both actions, then, are made to depend on the direction of the jets of steam as they leave the working wheel.

The steam turbine designed by *Koch*[1] (1897) (Fig. 309) also has a vertical shaft. The vanes f^1 fixed at the lower surface of the working wheel f, move in a slot in the upper of two supply-lead discs d, c, which are provided with channels d^1, c^1, pointing in like directions. The entrance of the steam can be regulated by the turning of the disc c.

In *Kummer's* arrangement[2] (1895) (Figs. 310 and 311) an impingement wheel c is assumed, to which the steam is led through the nozzle e, to be carried away again through the nozzle f. The working wheel c rotates between discs a, b which bear the nozzles, and are turnable—whether by hand or by the governor—in the slots g of the case. In the process of regulating, either the discs a with the supply-lead nozzles e are turned and the discs b held fast, or the latter are turned, and thereby the discharge-lead nozzles made to shift with the nozzles e in such a manner that communication with a greater or a smaller number of vane cells is

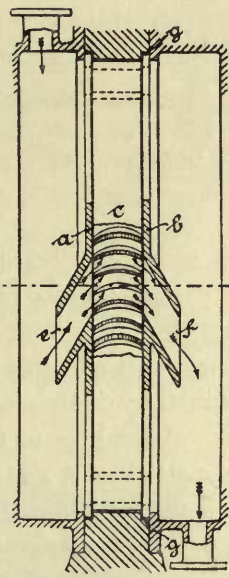


Fig. 310.

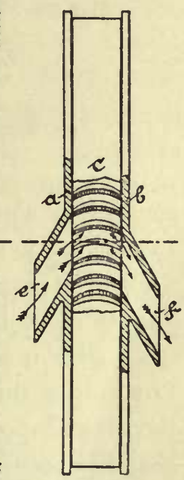


Fig. 311.

[1] D. R. P. 115,217.

[2] D. R. P. 83,412.

established or interrupted. Two wheels *a* may be assumed, which are impinged upon at both sides, and the guide rings of which must be relieved of all axial pressure of the steam by means of pairs of couplings.

In Fig. 312 is shown a governing appliance adopted by *McCollum and Forster*.[1] In the figure *a* is the turbine case, which is connected by means of the tube *d* with the slide-valve case *c*. The turbine shaft *b*, by means of worm gearing, causes the shaft *g* and the governor *z* to turn, and, by means of an eccentric, moves the slide pistons 17 and 18 backwards and forwards, the governor meanwhile displacing the tube slide 6. The valve case *c* has two separate fresh steam chambers covered by the slide 6, which is provided with the channels 7 and 8. The slide 6 is influenced by the governor in such a manner that the second slide, which is set in motion by the turbine and to which the pistons 17 and 18 belong, in case of an abnormal loading of the engine, covers the channels 7 or 8 at intervals or altogether. In case of normal loading, the slides 6 take the middle position, in

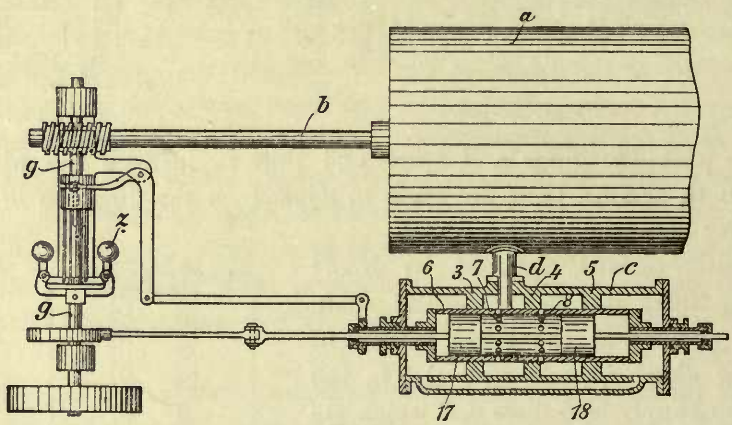


Fig. 312.

connection with which the channels 7 and 8 remain open. When the load on the engine is a normal one, its parts assume positions such that the channels 7 and 8 stand at equal distances from the middle annular partition 4. The pistons 17 and 18 move backwards and forwards, but will not pass over the channels 7 and 8 when slide 6 is in the middle position, because the stroke of the piston is too small to admit of this. According to this the steam passes into the chamber between the ring-shaped partitions 4 and 5, and through the channels 7 into the annular chamber between the walls 3 and 4, whence it proceeds through the pipe *d* into the engine. If the load on the engine be diminished, the governor *z* draws the slide 6 in the forward direction, and the row of channels 7 moves nearer to the wall 3, so that the piston 17 at intervals covers the channels 7, and will thus from time to time cut off the supply of the steam. The lengths of the periods of cut-off will naturally be determined by the greater or less reduction of the load. The supply of the steam through the channel 8 into the inside of the slide 6 will not be

[1] D. R. P. 146,999.

prevented; but, on the other hand, the flow of the steam to the cylinder through the pipe *d* in the manner required, and in exact proportion to the decrease of the load on the engine, will be interrupted. Meanwhile, should too great a load be put upon the engine, the slide 6 would be moved in the opposite direction by the governor, and the channels 8 would be brought close against the ring-shaped partition 5 and below the piston 18, so that this piston, by its movement, covers the channels, and thereby cuts off the steam.

The throttle regulation possessed by the turbine which was designed in 1884, and exhibited by Parsons before the Institution of Mechanical Engineers[1] in 1888 (with the current of steam divided), has rightly been given up. It showed, however, some constructive peculiarities. While subject to the alternating action, the valve stood between a spring tending to open and a leather diaphragm tending to close it. The influence of the latter began as soon as an exhauster mounted on the shaft rarefied the air in front of the diaphragm. The air had, however, to be drawn by the continuously rotating exhauster from a receptacle which by means of an opening stood in constant communication with the atmosphere. This opening was now closed by an electro-magnetically influenced revolving slide in proportion as the current passing through the electro-magnet became stronger, *i.e.* as the production of electric energy in the dynamo connected with the turbine varied from the normal amount. A weakening of the electric current caused the uncovering of the air opening of the receptacle above mentioned, and herewith the putting out of action of the diaphragm, *i.e.* the opening of the throttle valve. It may here be remarked that this regulation has been used in Parsons' turbines with from 6500 to 18,000 revolutions per minute.

The throttle regulation causes an alteration in the pressure of the steam. Since, now, the expansion nozzles with fixed walls and constant back pressure are only suitable for a certain fixed pressure of the inflowing steam, the attempt has been made to arrange the width of the nozzle in accordance with the alteration of the steam pressure, in order that the process of expansion for the conversion of the steam pressure into velocity may not thereby be intensified.

De Laval[2] (1894) (Fig. 313) has arranged his turbine with a reducing arrangement for running free. Into the nozzle A another nozzle B^1 can be shipped. Between the walls of these two a diverging channel B is formed, the flow of the steam towards which can be altered by an axial displacement of the nozzle B^1. When the nozzle B^1 has the position given in the illustration, the flow of the steam from the chamber A^1 to the channel B is interrupted. The steam

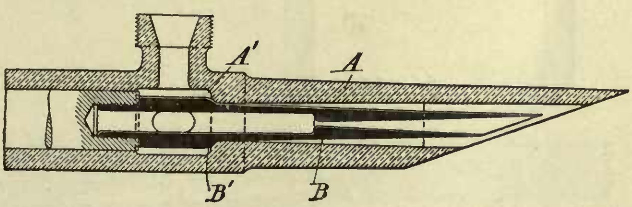


Fig. 313.

[1] E. P. 6734 of the year 1884. *Proceedings of the Inst. of Mech. Engrs.*, 1888.
[2] D. R. P. 81,783.

then streams out only through the nozzle B^1, which is adjusted to suit the power required for the empty load of the turbine.

Curtis[1] has, in the year 1896, given various methods for the construction of the nozzles. According to Fig. 314, the wall of the expansion nozzle G (of

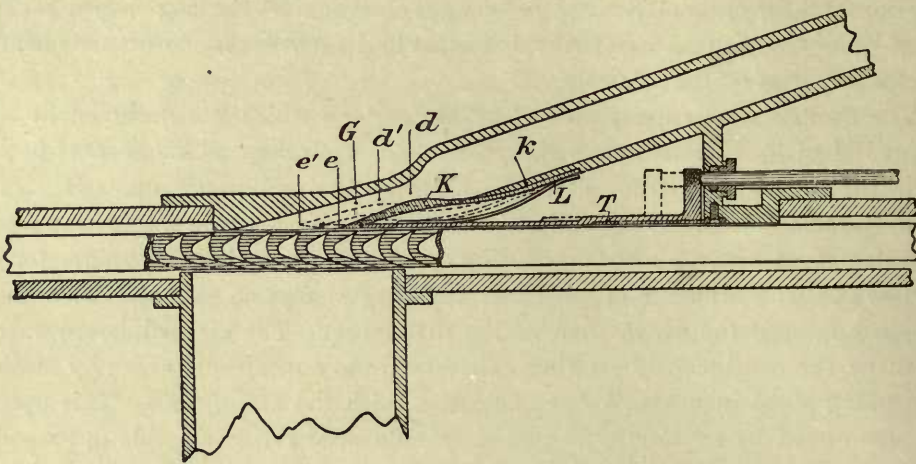


Fig. 314.

rectangular section) is formed by a tongue K, which can slide on the approximately circular surface *k*. A spring L presses the tongue K against the surface *k*, and by means of a slide T the tongue can be turned into the dotted position shown. If the motion of the tongue K is as it should be, the proportion between the steam inlet and the steam outlet openings of the nozzle remains the same, and $d : e :: d^1 : e^1$.

In order to relieve the tongue K of one-sided pressure of the steam, the side of the wall turned towards the latter is provided with an opening, which allows fresh steam to pass to the back of the tongue. In the arrangement shown in Fig. 315 the tongue K slides in a groove *k* in the wall of the nozzle. The displacement of the tongue is effected by the spur wheel N and the ratchet M. On the other hand, Fig. 316 shows a tongue *a* which is free to turn on the bolt *b*, and which also is provided with a relieving opening *c*. A number of partitions L (Fig. 317) may also, however, be arranged in the nozzle G, directed

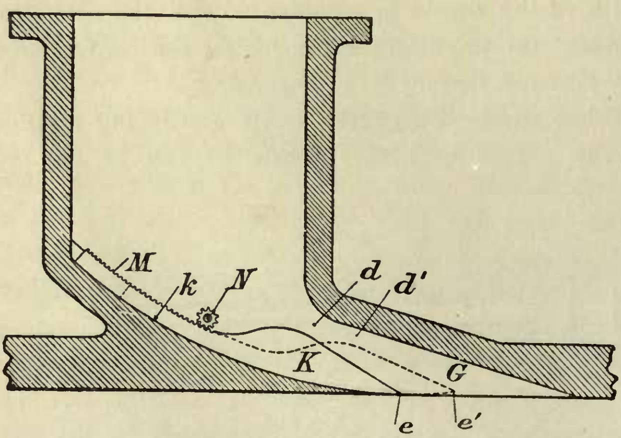


Fig. 315.

[1] D. R. P. 119,706. E. P. 19,284 of the year 1896.

towards a common trace point. A system of nozzle subdivisions are thus formed, which by means of a slide *s* can be successively shut off.

From the utterances made by Curtis[1] (1897) the following is worthy of remark. If the partitions of the steam-nozzle be allowed to run out, as shown by the full lines, up to the discharge *a* (Fig. 318), it is evident that the steam issuing at the short side of the nozzle has a greater pressure than that escaping at the long side, since an expansion of greater range has here taken place. A case can then be imagined in which the steam arriving at the short side of the nozzle in streaming past the long side first expands to the pressure of the steam here issuing before doing extraneous work. Further, the steam streams in at the short side at a greater angle, and therefore in greater quantity, than at the long side. These drawbacks Curtis proposes to meet by a special alteration of the nozzle. The latter is to be widened, as may be necessary for the required expansion to the dotted lines, as far as the position *b*. From here the nozzle walls are carried parallel as far as the point of discharge *c*. Otherwise (as shown by the stroke-and-dot line) the short side is, from the point *d* to the discharge *e*, made parallel with the long

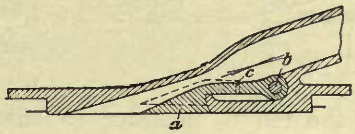


Fig. 316.

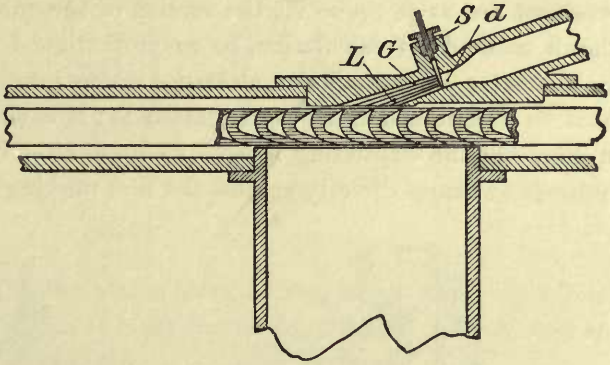


Fig. 317.

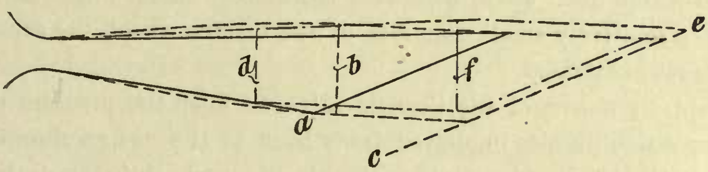


Fig. 318.

side beyond *f*. By the laying of the short side of the nozzle parallel to a part of the long side, the alteration of the section of the nozzle may be effected without modification of the conditions in regard to the expansion, as shown by Fig. 319, by means of the movable tongue *g*, the edge *h* of which moves along the line *l*.

In the regulating appliance designed by *Nadrowski* and *v. Knorring*[2] (1901) (Figs. 320 and 321) the principal feature is an annular adjustable leading apparatus

[1] A. P. 589,422. [2] D. R. P. 137,586.

with a through-passage chamber gradually increasing in size to a degree corresponding with the expansion of the motive medium. Forming part of the arrangement is a cone-shaped body K, the acute angle a of which is such that the proportion borne by the section of the outflow at the periphery of the guide wheel to the section of the clearance space at the narrowest point remains constant, when an axial displacement of the cone is effected by the rod S, in order to ensure, for all stages of the regulation, an equal fall of pressure of the motive medium in the leading apparatus. If in a turbine of this kind the pressure of the steam before entering the inlet apparatus be, for instance, 100 lbs. abs., and the condenser pressure be 1·43 lbs., so that the pressures are as $p_1 : p_2 = 70$, the section of the inlet apparatus must, by the well-known methods of calculation, be so proportioned that $F_2 : F_1 = 10{\cdot}4$; and taking the diameter of the annular clearance space into consideration, the acute angle a must be determined from the equation $R : H = \sin a$. In the arrangement shown in Fig. 320 the expanding motive medium, after its passage through the leading apparatus, streams directly against the first working wheel. Fig. 321 serves to show

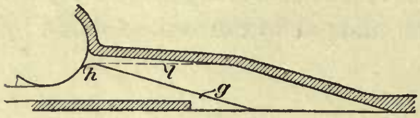


Fig. 319.

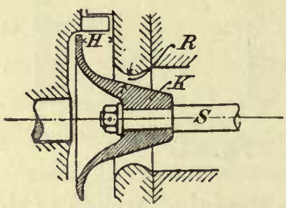


Fig. 320.

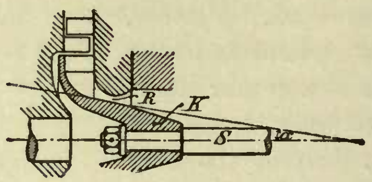


Fig. 321.

that a guide-vane wreath may, in addition, be set up before the first working wheel, its object being to divert the jet in a direction corresponding with the direction of rotation of the wheel. Fig. 322 shows the leading apparatus built into a turbine in which a multi-stage axial wheel adjoins a multi-stage radial wheel, so that, in conjunction with a relatively small diameter of the turbine wheel, the greatest possible number of stages is attained.

An attempt, by means of the throttle-valve, to keep the pressure of the steam before the expansion nozzles unaltered leads back to the design shown in Fig. 323, made by *Curtis*[1] (1902), who divides the steam nozzle into separate sections a, which are connected with the chamber k by suitable channels b. The partition walls c separating the sections are, at the points of outlet, kept as thin as possible, so that at all times a compact stream passes into the working wheel, and unconnected parallel threads of steam do not cause the working-wheel cells lying between the threads and the points not touched by the steam to be only partially filled. The governor is designed as an eased piston slide d, with a cushion e, which is adjusted by swinging-ball or other governing arrangement. In the case of compound

[1] A. P. 700,744.

turbines, governing appliances of this kind are placed before all the stages, and inter-connected so that they run together.[1] Here, indeed, it must not be lost sight

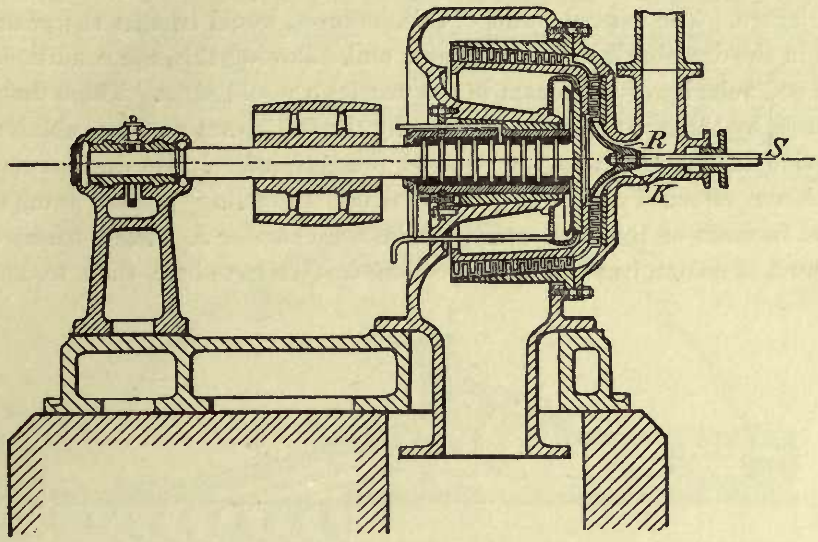


Fig. 322.

of that the dimensions and path of the piston for the low-pressure engine are larger than those of the high-pressure side. If a turbine be arranged for both exhaust and condensation, the groups of nozzles in question may be influenced by the same piston, in which case the one or the other group will be put out of action by the closing of the leading channel between the governor and the nozzles.

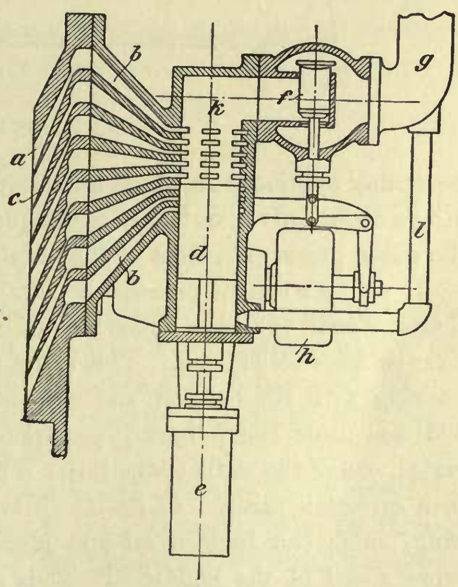


Fig. 323.

In Fig. 323 is shown an arrangement[2] made by *Curtis* in 1902, in which the piston slide *d* acts in conjunction with a valve *f*. It is here assumed that the steam lead *g* will show a constant pressure of steam, *i.e.* that it will, for instance, stand in connection with the boiler. The valve *f* is raised and lowered by the axial governor *h*. The upper side of the piston *d* is acted upon by the steam pressure obtaining in front of the channels *b* in the chamber *k*, while at the lower side steam of constant pressure from the lead *g* is admitted through the pipe *l*. When the turbine is set in motion, the piston *d* is at its lowest point. It leaves all the channels *b* open, so that

[1] Compare also E. P. 19,248 of the year 1896.

[2] A. P. 715,246.

the steam from the pipe *g* can stream unhindered into the nozzles *a* through the valve *f*, which is open to the full. As soon as the shaft of the turbine, and with it the axial governor *h*, attains a certain speed, the valve *f* will be closed to a corresponding degree. The natural result of this, however, would be that the pressure of the steam in the chamber *k* would then sink, and, following this, the conditions as to expansion and velocity of the steam in the nozzles *a* would alter. These drawbacks are prevented by the raising of the piston *d* by the full steam pressure, which acts at its lower side in opposition to the reduced pressure above. By degrees so many channels *b* are closed by it, that the resulting throttling process brings about the desired increase of the steam pressure in the chamber *k*. The surfaces of the piston *d* must, of course, be suitably proportioned. We have here, then, to do with a

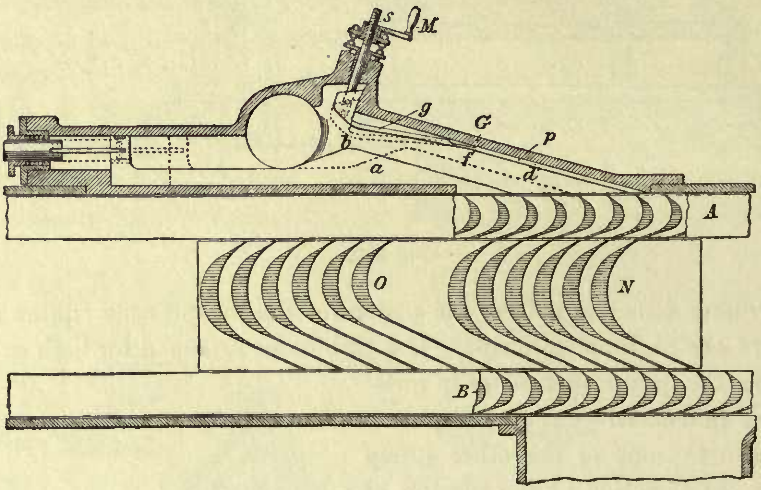


Fig. 324.

governing appliance in which an alteration of the load on the shaft of the turbine brings about only a variation in the quantity of the steam, and not a modification of the steam pressure in front of the nozzles.

If with a constant pressure of admission the power of the turbine is to remain constant when the back pressure varies, an alteration of the expansion nozzles will likewise have to be made. This refers to the disconnection of the condenser and the working with the exhaust, and *vice versâ*. For such cases *Curtis*[1] (Figs. 324 and 325) has made the following provision. To enable the width of the nozzle to be varied, one of the walls of the latter is formed by a tongue *a* capable of being moved in a direction parallel to itself. Into the opposite wall G, however, a spring *p* is sunk, which can be bent off and pressed inwards at the point *f* by the help of a screw *s* and of the handle M. This spring varies the conicity of the nozzle. It supplies a means by which the expansion of the steam in the nozzle, and with it the velocity and pressure of the same at the discharge, may be increased or diminished. If, for instance, the spring take the position *g* (the tongue *a* remaining unaltered), a

[1] E. P. 20,536 of the year 1897. D. R. P. 123,932.

larger quantity of steam will expand to a pressure in the nozzle less low than when the spring is pressed back to the position *b*. In the former case the form of nozzle is attained for a turbine working with the exhaust, and in the latter the form for a turbine working with condensation. If, for instance, the latter has two stages, A and B, the variation of the conicity of the nozzles will have to be accompanied by the adjustment of the leading apparatus, which is provided with two groups O, N of guide vanes, the one for the exhaust and the other for condensation. Further, it becomes possible to suit the widening of the nozzle to variations that may occur in

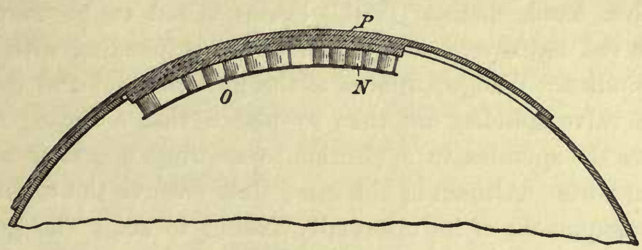


Fig. 325.

the boiler pressure when constant quantities of steam of constant velocity are required.

Another arrangement for driving multi-stage turbines, the first stage *a* of which, with or without condensation, is shown in Fig. 326, consists, according to *Curtis*[1] (1897), in the connection of two expansion nozzles *b*, *c*. The nozzle *b* is so formed as to admit just so much steam as work with the exhaust requires for a given power. The increase in width of the nozzle is so arranged that the expansion of the steam, after it passes through the last wheel, can only proceed as far as the pressure of the exhaust. The case is different with the nozzle *c* arranged for the same power, which has a narrower steam inlet than nozzle *b*, and admits of the steam expanding to a greater degree than in the first case, before it enters the wheel *a*. In view of the difference in volume of the steam in the two cases, the leading apparatus *d* and *e* are of correspondingly different sizes. The nozzles *b*, *c* have branch pipes *f*, *g*, which are capable of being shut off, and tongues *h*, *k* influenced by the governor, which throttle the steam to a degree corresponding with the varying load on the engine.

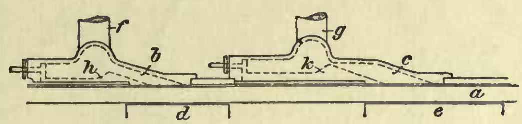


Fig. 326.

If, as is possible in the velocity turbine, the quantity of steam to be supplied to the engine be divided into separate parts, which are eventually to be led through nozzles, the *regulation* of the *quantity of steam* may be effected, without alteration of the initial pressure of the latter, by the successive opening or closing of the nozzles.

Raworth[2] covers the entrance to the nozzle with a ring slide having large slits of different sizes, the arrangement being such that when the slide is turned by the governor the nozzles can only be covered one after another.

[1] A. P. 590,210. [2] E. P. 25,090 of the year 1893.

Davidson's[1] aim also is to allow the steam from the nozzles just opened always to flow out with its full energy. With the adjusting-gear of the governor he accordingly also combines a ring slide, which opens or closes the nozzles one after another.

Similar in its method of working to the above is the arrangement made by *Griffin*[2] (1894), who, as check appliances for the valves, makes use of cocks, the cones of different groups of which are connected together by means of rings, so that only one group need be adjusted at a time.

On the other hand, *Altham*[3] (1893) causes a rod to be moved by the ball governor. The rod has several wedge surfaces corresponding with similar surfaces on the valve spindles. Wedge surfaces of the governor rod and the corresponding surfaces of the valve spindles are then so placed that a sliding movement made by the rod raises the spindles, in opposition to a spring, one after another, and thus opens the steam ports. Altham at the same time ensures the regulating appliance against the consequences of its eventually failing to act. The governor in such cases produces a change in the position of the rod by means of reversing gear. The latter is so disconnected by the failure, that the outer end of the rod, which influences the valve, moves the sleeve of the governor by means of gearing.

When, as in the case of the *Maardt*[4] turbine (1895), a ring of nozzles is put round the working wheel to act as impinging apparatus, the regulation could be effected by the arrangement shown in Fig. 327. The fresh-steam channel C is kept separate by means of a partition wall. On the other hand, two check-valves are inserted. By means of these a quadrant of the nozzle ring, or two such quadrants, may be disconnected. If, however, nozzles be provided only for two quadrants of the nozzle ring lying opposite to one another, it is possible by means of adjustable ring guards, suitably arranged round the periphery, to open or close successive pairs of nozzles.

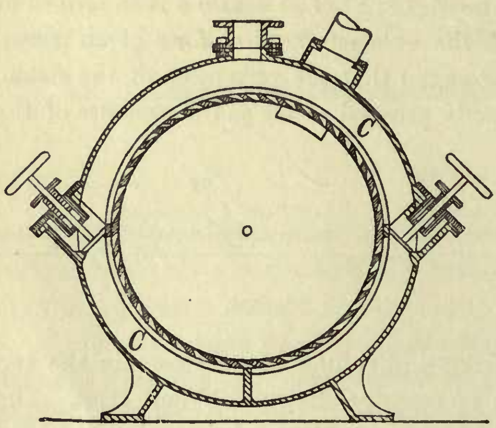


Fig. 327.

G. Westinghouse[5] (1897) adopts a radial turbine with impingement from within and with velocity stages (Figs. 328, 329, and 330). As working-wheel body he uses a disc 1, on which the vanes 4 are arranged axially. The corresponding guide vanes 10, again, are placed together on the ring 5. The expansion nozzles 15 are supplied with fresh steam from special chambers 14, each of which can be closed against the annular steam chamber 12 by means of a spring valve 16. The fresh steam in the annular chamber acts against the springs by pressure on the valve

[1] E. P. 15,501 of the year 1896. [2] A. P. 562,821. [3] A. P. 519,785; D. R. P. 82,215.
[4] D. R. P. 87,519. [5] A. P. 712,626.

piston 18 in trying to open all of the valves 16 to such an extent that it can, without loss of pressure, stream from chamber 12 into chamber 14. If, however, steam be introduced at the back of the valve piston 18, the latter pushes the valve against its seat. The chambers behind the valve piston 18 are covered by a ring 21, which is worked into the case cover 22 and contains grooves 24 running round it. It may now be assumed that each of the latter is placed in connection with the

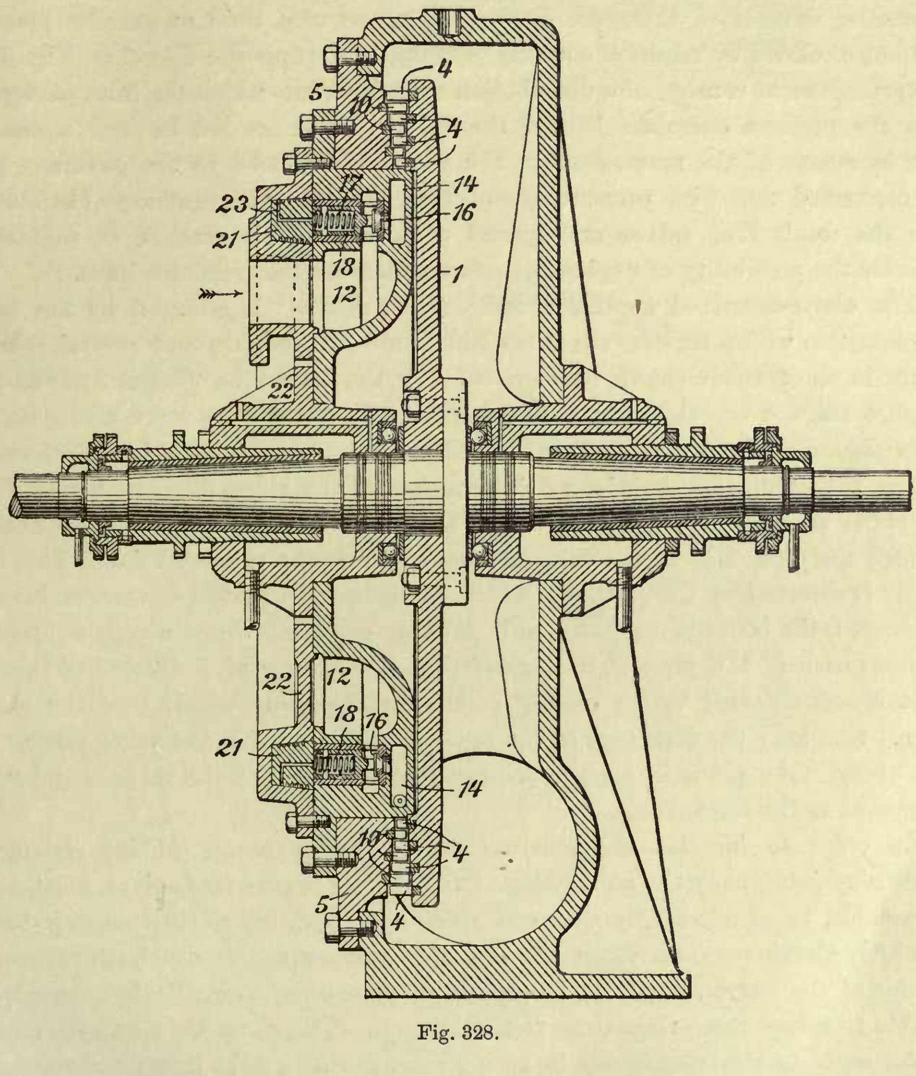


Fig. 328.

piston chambers of two valves 16 by means of cross-running holes 23, and with the channel 27 by means of a hole 25. The channel 27 opens on the one side into the steam chamber 12, and on the other by means of the passages 30 to the atmosphere. According to the adjustment of a slide 28, which is influenced by the governor, certain of the annular grooves 24 will now receive steam from the chamber 12, while others will be under the pressure of the atmosphere. In accordance with this the valves 16 belonging to the first-named groups of annular grooves will be closed and the others opened. The piston-slide rod 32, it may be observed, is hollow, so that the governor has to adjust a slide, the load on which is eased.

Another similar appliance which effects the introduction of auxiliary steam to the individual valve pistons is to be seen in the regulating arrangement of *Reuter*[1] (1901). It is applied to turbines with partial impingement and separate nozzles that can be disconnected by valves. According to the load on the engine, a larger or smaller number of these are opened by means of a fluid or gaseous pressure medium, or closed by means of a spring pressing in the opposite direction (Fig. 331). The arrangement consists of a distribution cylinder *i*, into which the inlet passages *h* from the pressure chambers behind the valve pistons are led in such a manner that, by means of the movements of the piston *l* influenced by the governor, they are connected now with pressure chamber *m*, now with the discharge chamber *n*, with the result that valves are opened or closed. This manner of regulating excludes the possibility of unpleasant reflex actions on the regulator itself.

The above-described appliance designed by *Reuter*[2] is adduced by the latter in connection with a turbine, which is divided into two pressure and several velocity stages in the manner shown by Fig. 332. In the latter the nozzles *b* of the first pressure stage *a* are closed by valves *d*, when only the springs *g* are acting on the valve pistons *e*. The nozzles b^1 of the second-pressure stage a^1 strive to close the springs f^1 pressing on the pistons e^1, by means of the slides d^1. The valves *d* and the slides d^1, however, are opened when a pressure fluid, such as water, is applied through the pipes h, h^1 to the valve pistons e, e^1 from the valve-rod side. This fluid supply is directed by the pistons *l*, which are influenced by the governors, into the cylinders *i*, the latter being provided in suitable number. Since a pipe h^1 branches off from each of the pipes *h*, it is clear that the closing of a nozzle *b* of the first stage is accompanied by the closing of the corresponding nozzle b^1 of the second stage. Similarly the discharge of the pressure medium below the valve pistons e, e^1 will always take place in such a manner that one each of the valves *e* and e^1 will be opened at the same time.

In order to increase the sensitiveness of the governor, rubbing resistances, which may accompany the adjustment of the closing organs themselves, must, as far as possible, be avoided. By means of electro-magnetically worked closing organs, in which electro-magnets effect the opening, and springs or other appliances the closing of the valves, singly or in groups, in succession, *Reuter*[3] (1902) expects to be able to reduce these resistances to a minimum. The governor closes the circuits of the separate electro-magnets in such a manner that a fluid interpolator attached to its lever is raised and lowered, a larger or smaller number of contacts, according to the load on the turbine, being dipped into the fluid.

As closing organs of the various nozzles *Wilkinson*[4] (1903) makes use of revolving slides, the axes of which lie at right angles to those of the nozzles. The slides are

[1] D. R. P. 144,102.
[2] E. P. 20,164 of the year 1903.
[3] D. R. P. 151,379.
[4] A. P. 741,426.

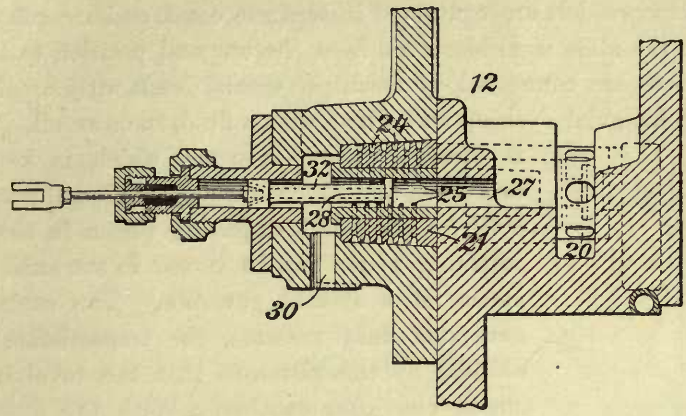


Fig. 329.

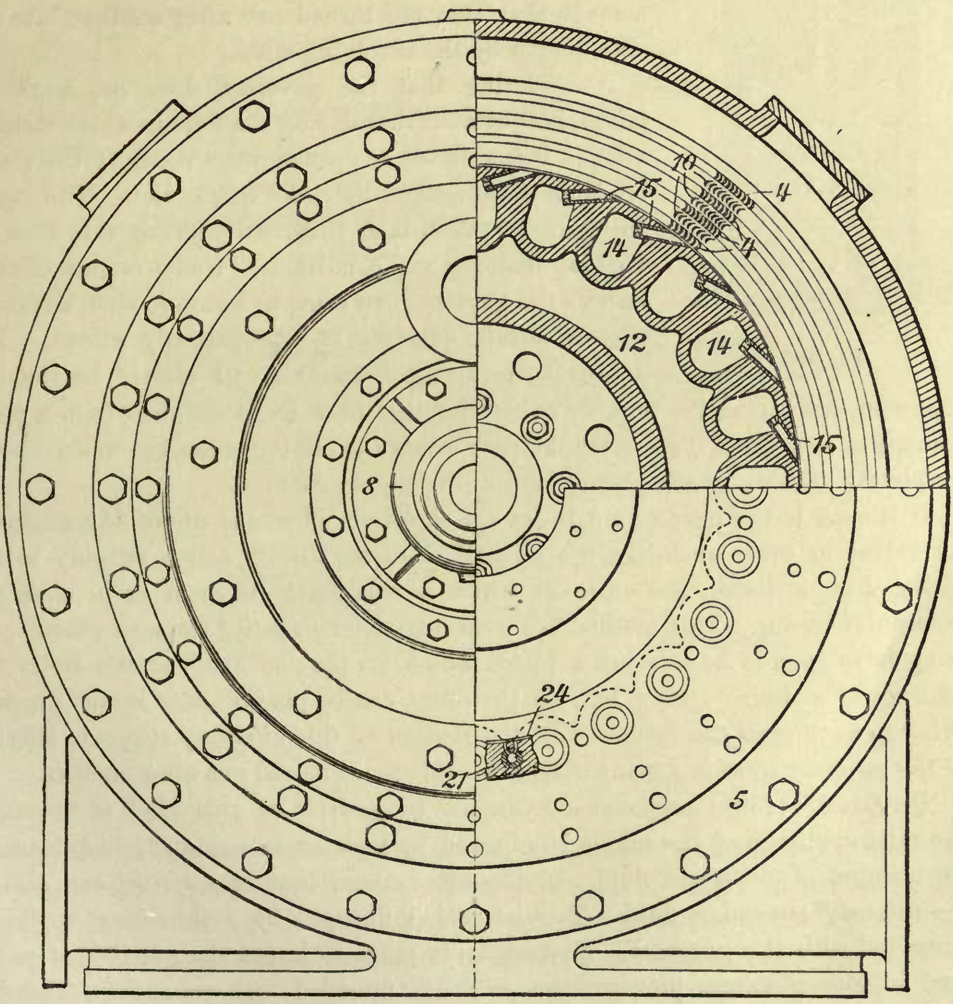


Fig. 330.

worked by pistons which are contained in separate cases, and are subjected to fluid pressure when the slide is to be moved from the one end position to the other one. The piston cases are connected by means of special leads with a cylinder that is provided with an axial division and filled with a fluid, such as oil. The cylinder is pierced by a piston rod which is kept constantly revolving, and which turns a centrifugal pump in the one cylinder and a slide-adjusting piston in the other. The adjusting piston can be moved in the axial direction by means of a special governor. The centrifugal pump generates fluid pressure, the transmission of which is effected by the piston so that the revolving slides are closed one after another. With the down stroke the adjusting piston establishes connection between the centrifugal pump and the cases of the pistons in such a manner that they are turned one after another into the open piston by the revolving slide.

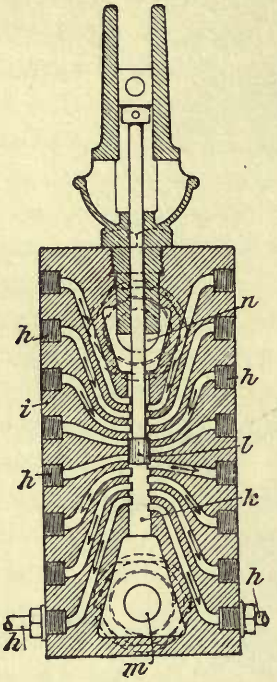


Fig. 331.

Considering that the governor does not work by jumps, it is evident that during the time in which a steam inlet is being closed or opened a throttling of the steam takes place in front of the inlet in question. The corresponding steam nozzle, then, will during this time be working under wrong conditions. Under certain circumstances the governor can come to a stop in such a manner that a nozzle entrance is kept partially closed. The drawbacks resulting from this will clearly be the less important, the larger the number of nozzles that have to be supplied from a given quantity of steam, and the shorter the path which the closing organ has to traverse in completely opening or completely closing the supply channel.

It is more to the purpose not to let the governor effect the direct adjustment of the valves or other such-like operation, but to confine its action entirely to the setting free of forces, the office of which is completely to open or to close the steam-inlet organs. This method has been adopted by *Zoelly*,[1] amongst others. To every valve spindle he attaches a piston which, on the one hand, stands under the influence of a closing spring, but, on the other, can be placed under steam pressure acting so as to open the valve. The distribution of this auxiliary steam is effected by the governor in such a manner that the valves are moved one after another.

De Laval[2] (1894) proposed a hydraulic transmitter of this kind as governor. The inlet appliance of the motor is adjusted by a piston, one side of which is under the pressure of the power fluid, and the other under that of a spring, and also of a constantly streaming fluid. The latter is influenced by a throttling appliance connected with the governor. Meanwhile, in order to lessen the overflow of power fluid in case of excess pressure, the piston is provided with an annular chamber

[1] E. P. 1250 of the year 1902. [2] D. R. P. 84,915.

extending along it. By means of a narrow channel, in which the amount of the flow can be regulated, the annular chamber stands in connection with the chamber for the streaming fluid, and is kept filled with a fluid that is under approximately the same pressure as that of the power fluid. In Fig. 333, A is the cylinder and B the piston. The steam streams into the motor through C, and out of it through D. The regulating valve E is fixed to the piston and has its seat in the cylinder. The

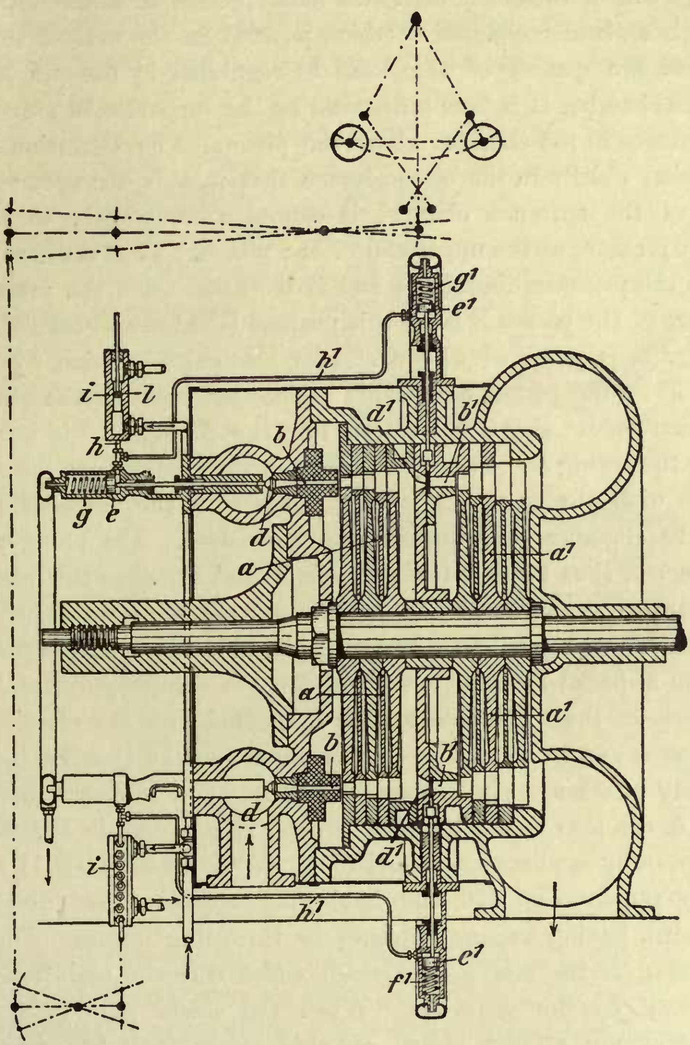


Fig. 332.

steam pressure at the one end of the piston has to preserve equilibrium between the pressure of the spring S and the pressure of the fluid, which streams through the upper part of the cylinder, and which is here introduced at F and led off at G. In the lead which is screwed on at G is inserted the brake organ, which is so influenced by the governor of the turbine as to vary the water pressure at the upper end of the piston, and therewith to regulate the supply of steam. To promote the mobility of the piston B the latter is hollowed out at H, and into the hollow a fluid is introduced. The latter may suitably consist of a part of the feed water which is forced into the boiler in connection with the engine in question. For the case in which the pressure of the feed water, as, for instance, in water-tube boilers with long coils, is considerably greater than the steam pressure, the water pressure is reduced by means of a special appliance so that it becomes approximately the same as that of the steam. Out of the chamber H this water presses on to the upper part of the piston through a slender channel L, which is fixed in the wall of the cylinder A, and the area of the opening of which can be regulated by the pin M. The water pressure in the chamber H is thus influenced by the variations in pressure which the regulator produces in the chamber above the piston. The alteration illustrated in Fig. 334 consists chiefly in the circumstance that in this arrangement the area of the opening of the entrance channel is caused automatically to vary with the fluctuations of pressure at the upper end of the piston. Thus the area of opening is reduced when the pressure diminishes, and is increased when the pressure increases. The upper part of the piston B is for this purpose bored out to take the plunger L^2. The latter, again, is provided with a slender channel L^1, which, by means of the side channel F^1 in the piston, establishes connection between the chamber H and the other chamber H^1 above the piston B. The plunger L^2 is influenced at its upper end by the spring S^1. It is here apparent that the plunger alters its position in accordance with the variation of the pressure in the chamber H^1, while the pressure in the chamber H is approximately constant. The plunger is caused to move upwards, and thus to close the channels F^1 and F^2, when this pressure becomes less, and to move downwards and open the channels referred to when the pressure increases. Instead of the plunger L^2 being placed in the hollowed piston B, it may be arranged in a special cylinder outside of the governing appliance A. The lower and upper parts of the cylinder are then connected with the chambers H and H^1 respectively, as shown in Fig. 335. In the same way the channel L, together with the pin M, may be arranged in a separate member apart from the cylinder A. The hollowing at H can also be made in the cylinder A, as shown in Fig. 335.

In the governing appliance designed by *A. C. Th. Müller*[1] (1901), an illustration of which is given in Fig. 336, the impinging apparatus are divided into two principal groups, having separate closing or throttling organs. The closing (or throttling) organ of the one group is influenced directly, and that of the other group indirectly, by the governor. When the steam pipe *g* is thrown open, all the valves *k* are at first closed, so that steam from the main steam lead *g* cannot at once pass to the turbine wheel. By means of the branch lead *h*, in which the throttling organ is brought into the closing position by the governor R, only a small quantity of steam finds its way into the annular chamber *e*, and thence by way of e^1 to the group of nozzles I. Since only considerably throttled steam enters the chamber *e*, while fresh steam at full pressure enters the chamber *d*, the valves *k* will, in consequence of the excess pressure in *d*, open

[1] D. R. P. 146,497.

quickly, and steam enters the various nozzles *a*, *b*, which are separated by the partitions *f* into groups II and III. The turbine will then start quickly, and in case of light load the number of its revolutions will increase so rapidly that the governor will come into action and the throttling organ *m* of the branch lead *h* will be gradually opened, by which the pressure in *e* will increase and the valves *k*, according to their dimensions and the strength of their springs, be closed all at once or one after another. For the real light load the nozzles of group I will generally be sufficient, which on the opening of the throttling organ *m* receive full steam through the branch lead *h*. According to the choice of the loading springs

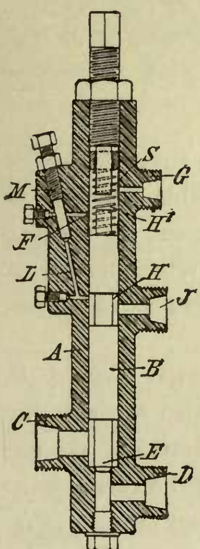


Fig. 333.

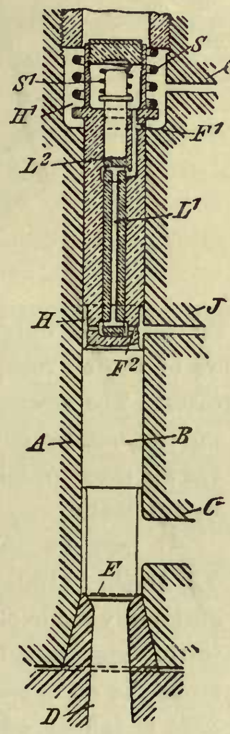


Fig. 334.

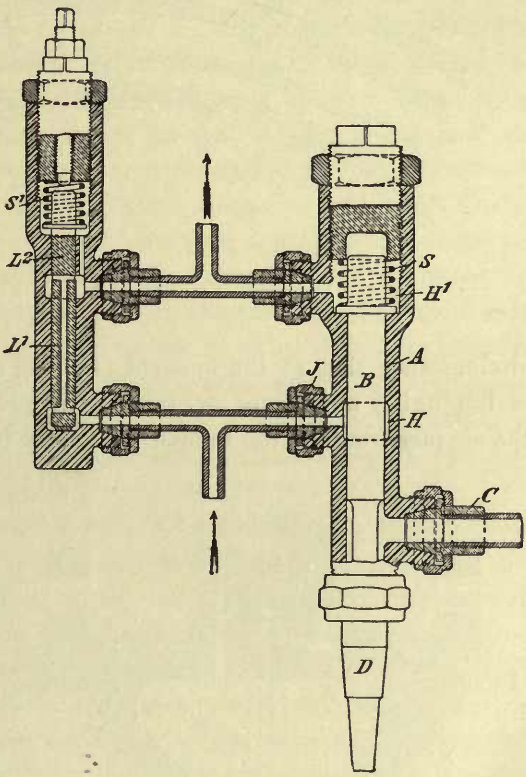


Fig. 335.

and of the proportion between the surfaces of the valves on which the various steam pressures are put, such arrangements must be made that the valves do not all act at once, but that single groups of the impinging appliance, according to the varying load on the turbine, receive steam one after another. In this manner an extremely exact regulation can be made. Those nozzles which, as in the case of Group I, receive steam from the outer annular channel *e*, *i.e.* independently of the valves *k*, can be used to overcome the light-load resistance, but in ordinary work they also take up the steam which, on account of leaks, as, for instance, past the moving piston of the closing organ *k*, passes from the chamber *d* to the chamber *e*.

To some extent a reversal of the foregoing, *i.e.* the preservation of the power of the turbine by means of an increase in the number of nozzles acting, is presented by the method invented by *Stumpf*[1] (1902) for improving the efficiency of fish torpedoes that are worked by means of compressed air turbines. It is characterized by the

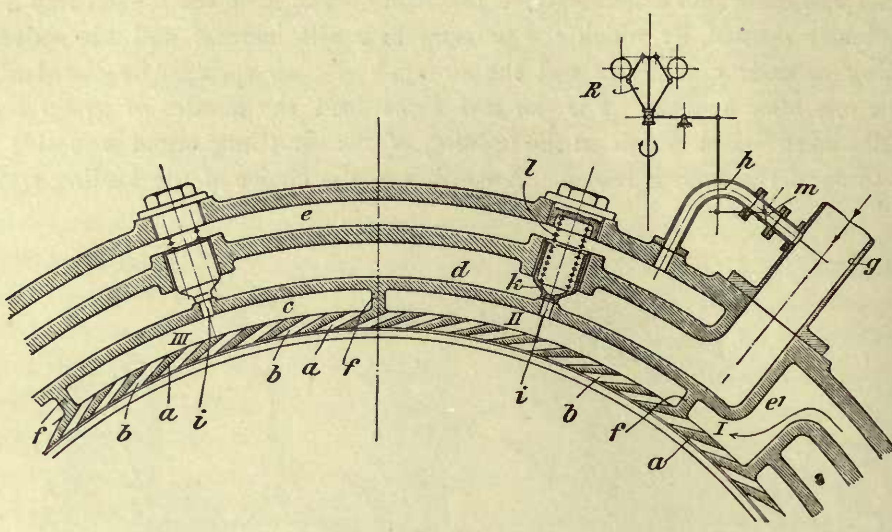


Fig. 336.

circumstance that at the moment of firing of the torpedoes only a limited number of impinging appliances receive the full pressure of the air receptable, and that as the air pressure falls the number gradually increases. The arrangement for turning on the nozzles one after another as the air pressure falls must, of course, be automatic.

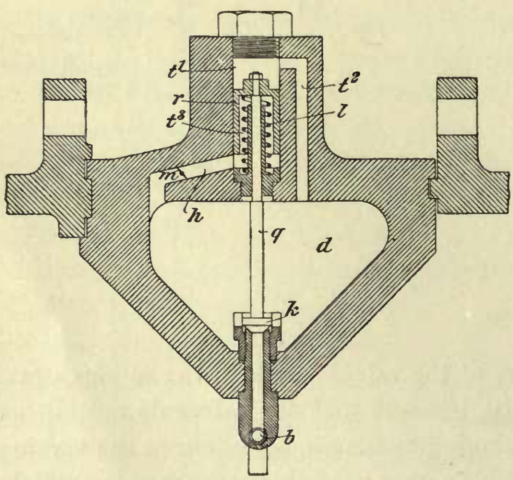


Fig. 337.

It may be observed that *A. C. Th. Müller*[2] (1901) also reverses the customary mode of action of the adjusting appliance for the valves in so far as the valve motion piston r (Fig. 337) is influenced at the side t^1 by fresh steam in the closing direction, and at the other side t^3 by the throttled steam and the spring l in the opening direction. Müller here comes to the following conclusion. If, in the case of turbines, a current of steam be made use of as varying pressure medium, its pressure must increase when the engine requires less steam, and decrease when it requires more steam. Since, now, the auxiliary steam which comes in question in the first line for the adjustment of the closing organ is for the most part also used

[1] D. R. P. 148,468. [2] D. R. P. 146,623.

in the turbine, where it is led to a number of nozzles, there arise, for these latter, fluctuations of pressure which stand in inverse proportion to the likewise varying power requirements. When the turbine has less work to do, these latter nozzles receive more steam, and when it has more to do they receive less. In his arrangement the circumstances are such that the nozzles fed from the regulating current also, in accordance with their increased and reduced work, receive more or less steam. The nozzles *b* stand in full communication with the fresh-steam supply channel *d* and are separated from it by separate valves *k*. Each valve is connected by a rod *q* with a piston *r*. All the pistons are placed in cylindrical borings in the case. The chamber t^1 outside of each piston *r* stands, by means of a transverse channel t^2, in direct communication with the fresh-steam channel *d*. The chamber t^3 within the piston *r* also stands, by means of a branch channel *h*, in communication with the fresh-steam chamber *d*. In this branch channel, however, is inserted a throttling organ *m*, which is actuated by the governor. In practice all the chambers t^3 are connected by a special steam channel, so that only one throttling organ is required. The piston *r* is further acted upon by a spring *l*, the tendency of which is to move the latter outwards. Since the full pressure of the fresh steam on the channel *d* constantly rests on the piston *r* from without at t^1, the latter can only be moved, *i.e.* the valve *k* can only be opened, when the pressure in the chamber t^3 with the help of the spring *l* overcomes the pressure in the chamber t^1. The throttling organ *m* must then be opened, and when, as before mentioned, a throttling organ serving for all the pistons is present, the latter must be opened more and more in proportion as the power requirement increases. Thus the more steam the turbine needs, the more the throttling appliance will have to be opened, and *vice versâ*.

Wilkinson[1] (1903) also tries to reduce the work to be performed by the adjusting gear of the governor as much as possible. He accordingly ships the spindle of the turning disc which acts as steam-checking organ with a squared end into one-half of a coupling, the other half of which is kept in constant rotation by the turbine. The two parts of the coupling are held apart by a spring. The half coupling, which by means of the squared end acts upon the slide rod, may now be provided with electromagnets, in which case the governor closes his circuit for the latter and the coupling is completed. Otherwise the half coupling is moved by hydraulic power, in which case the governor gives admission to the pressure medium. In each case the rotation of the coupling and thus also of the admitting organ is produced by the engine itself.

Now an unvarying maximum hydraulic effect in a turbine can only be attained when the velocity of the working steam, even in case of a fall in pressure, can be kept constant, whether such fall be produced by fluctuation of the initial pressure or whether by a change from work with the condenser to work with the exhaust or the reverse. To a certain degree this may also be effected by the admission, to the current of steam, of fresh steam. *Steinle*[2] (1902) applies this to the regulating of the current velocity of a compound turbine which works with expansion nozzles in

[1] A. P. 743,125.

[2] D. R. P. 151,678.

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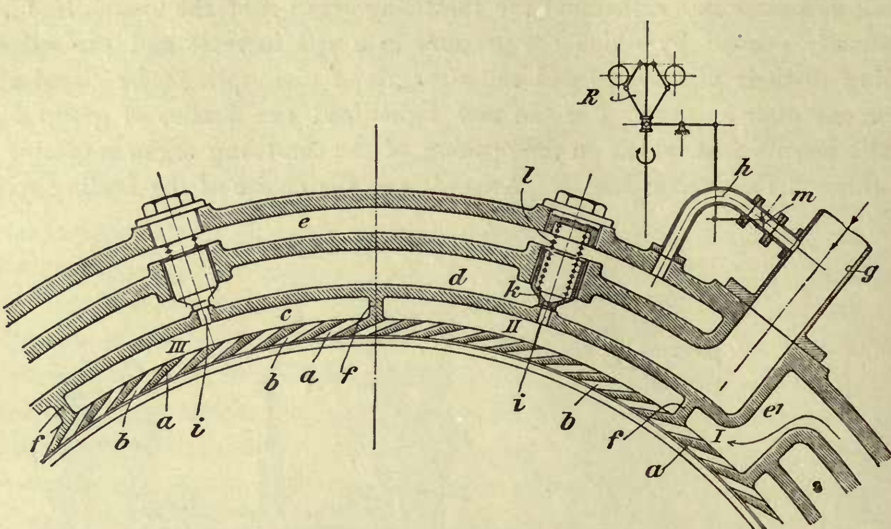
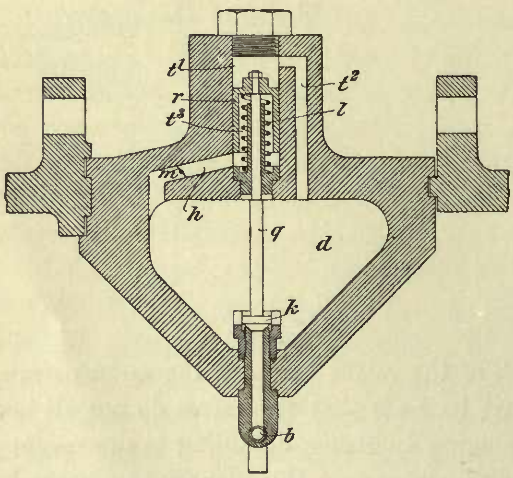


Fig. 336.

circumstance that at the moment of firing of the torpedoes only a limited number of impinging appliances receive the full pressure of the air receptable, and that as the air pressure falls the number gradually increases. The arrangement for turning on the nozzles one after another as the air pressure falls must, of course, be automatic.



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Now an unvarying maximum hydraulic effect in a turbine can only be attained when the velocity of the working steam, even in case of a fall in pressure, can be kept constant, whether such fall be produced by fluctuation of the initial pressure or whether by a change from work with the condenser to work with the exhaust or the reverse. To a certain degree this may also be effected by the admission, to the current of steam, of fresh steam. *Steinle*[2] (1902) applies this to the regulating of the current velocity of a compound turbine which works with expansion nozzles in

[1] A. P. 743,125.

[2] D. R. P. 151,678.

nozzle 24, which is regulated by slide 14. The rod 16 belonging to the latter is linked to a stud 15 which works loosely in a radial groove of the disc *g*.

In case of the maximum steam pressure drop for which the turbine is built, the stud 15 is concentric with the disc *g*, and in this position keeps the auxiliary slide 14 closed for every position of the governor. If the pressure of the fresh steam sink from p_7 to p_6, movement is communicated to the piston 10 (compare also Fig. 342), which is loaded at its upper side by a spring inserted in the case 18 and adjustable by means of the screw 19. The movement thus caused is made to correspond with the pressure drop of the fresh steam, which by means of the branch 20 is in communication with the lower piston surface from the top of the check stud 21 downwards. By this motion of the piston the lever 17 is made to turn about the point 23, which in the first place is assumed to be fixed, and, as shown in the drawing, brings the stud 15 into a position eccentric to the governing disc and dependent on the position of the piston. By this the auxiliary slide is opened and the introduction of fresh steam into the receiver steam is regulated in accordance with the pressure drop of the former and the steam consumption of the turbine. If the pressure of the exhaust rise from p_2 to p_3, the turning-point 23 of the lever 17 will be raised by means of the screw spindle 7. As a result of the lever arrangement, a displacement of the stud 15 from the middle of the governing disc takes place as in the previous case, and, following this, an opening of the auxiliary slide, corresponding with the steam consumption of the turbine and the increase in pressure of the steam. The movements of the points 22 and 23 are communicated by means of wheel gearing (not shown) to an indicator. In accordance with the position assumed by this indicator in relation to a pressure indicator, the appliance is to be observed and adjusted.

In regard to the adjusting appliance itself, the governors of ordinary pattern with weights swinging round with centrifugal force may easily be used for the narrowing or widening of the steam inlets to turbines, provided, in arranging and working them, consideration be paid to the high speeds of rotation of the power engines.

Thus *Rouse* and *Symon*[1] couple to the shaft of the turbine an axial governor, the fly ring of which is acted on by a spring, and imparts a turning impulse to a toothed wheel which screws the spindles of steam-inlet valves up or down.

Seaver[2] (1897), the basis of whose design is a pressure turbine with impingement from within, likewise makes use of an axial governor provided with swinging balls. The adjusting appliance actuates a round slide in the hollow shaft. Again, the radial turbine with impingement from within used by *Weichelt*[3] has on its shaft a sleeve, adjustable in the axial direction by the governor, to guard the first leading ring; while *Pyle*[4] (1895) allows the ring guard of a radial turbine impinged on from within to be moved along the shaft by an axial governor.

Krank[5] (1902) tries to avoid the use of a special governor by making the

[1] E. P. 8832 of the year 1896. [2] A. P. 603,660. [3] E. P. 17,199 of the year 1901.
[4] A. P. 552,396. [5] S. P. 25,914.

hydraulic appliances which serves for making the bearings steam-tight also do duty in actuating the steam-inlet organ. The wheels of the centrifugal which place the water in question under pressure, and which are fitted on the shaft of the turbine, set up a water pressure which increases with the speed of the shaft, and the influence of this pressure on the adjusting gear of the steam-inlet organ is brought to bear in a proportionate degree.

Again, the *pressure turbine* which is impinged upon over the whole working-wheel wreath will only work properly when the inlet pressure of the steam remains the same as that for which the engine has been designed. Since its method of working further constantly requires the turbine to be completely filled and a throttling effected by the narrowing of the impinging apparatus causes loss, bringing expansion in the working wheel, the only means remaining for the regulation of the energy is the introduction of the steam by stages. This is accomplished by means of a suitable distributing apparatus which in ordinary working opens the steam inlet at as short intervals as possible. For the reduction, then, of the work done by the engine the governor produces a more or less frequent stoppage of the opening of the inlet.

As an older method of construction, that of *Nicholl's*[1] (Fig. 343) may here be cited. The steam streams from the arms *a* of the reaction wheels, but not continuously. Chambers are provided which have first to be filled by the steam before the quantity of the latter that is present in them can expand outwards. The expansion, again, follows the filling of the chambers, and so on. For this purpose the arms *a* of the wheel chambers *b* are built on. These receive their steam through openings *c* from the arms *a*, and are provided with the exhaust branches *d*. The slides *e* alternately close the branches *d* until the chambers *b* are filled with steam. The exhausts *d* (position as in Fig. 343) are not opened till this is accomplished. The actuation of the slides *e* is effected by means of the gear *fg* from the lever discs *h*, which rotate with the wheel, and over which a strap *l* resting on the fixed disc *k* is led.

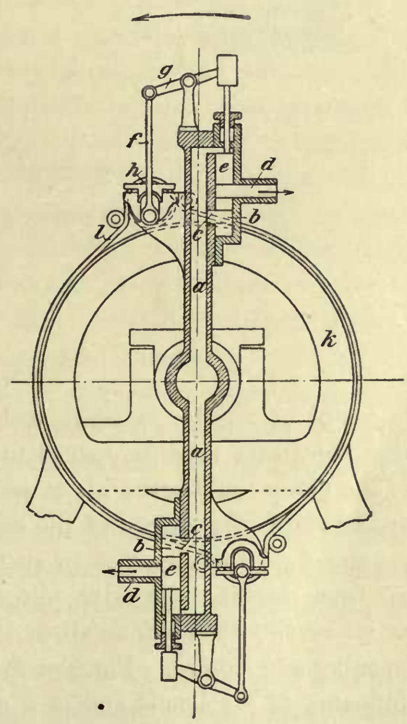


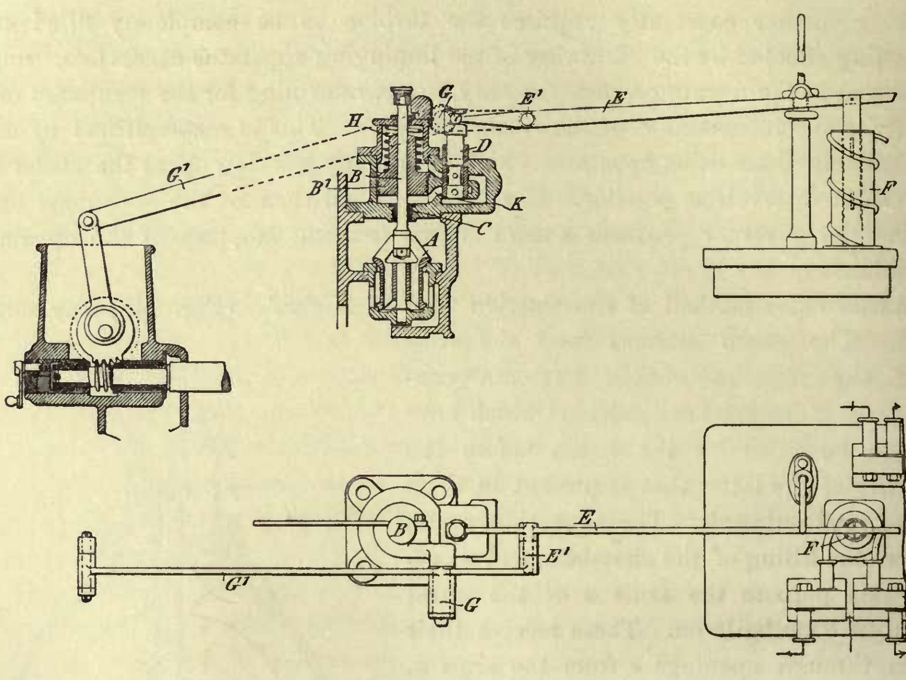
Fig. 343.

In the turbine designed by *Bazin*[2] (1878) also the passage of the steam is governed.

According to the arrangement of *Parsons*[3] (1891) the steam is introduced through a balanced valve (or through several valves at regular intervals). The durations of

[1] E. P. 480 of the year 1853. [2] D. R. P. 5046. [3] A. P. 549,815.

these periods of admission are intended to accommodate themselves automatically to the load on the motor, without the steam pressure being varied, so that a throttling of the working medium negligible in amount can come in question only during the short time in which the valve descends. The spindle of the steam-inlet valve is in consequence connected with a piston which works in a separate case. On the one hand, this is subject to the pressure of a spring tending to close the valve, but on the other it has to sustain the pressure of the compressed air or similar agent which has to open the same. The compressed air



Figs. 344, 345.

is forced at regular intervals below the valve piston through a special pipe, so that the steam valve is opened to a proportionate degree, while the suction period of the pump enables the air to escape through a special outlet, following which the spring effects the closure of the steam valve. This special outlet, however, must be regulated by the governor in such a manner that it first prevents the compressed air from reaching the valve piston, or delays it from doing so in the normal time, or it prevents the air from streaming out again, *i.e.* it prevents the steam valve from opening and closing. Further, the suction valve of the pump also is subject to the influence of a solenoid and of a spring. If too much current be delivered by the dynamo, the suction valve is kept open by the solenoid, and in case of interruption of the current the same office is performed by the spring. The pump then runs empty, and the steam valve remains closed.

The regulation of the steam inlet periods may be accomplished without the help of a pump by means of the arrangement of *Parsons*[1] (1893), shown in Figs. 344 and 345. The rod B of the steam valve A passes with a small amount of play through the stuffing box C, so that steam from the valve chamber can pass through the latter below the piston B^1 and raise it in opposition to the action of a spring H, *i.e.* it can open the valve A. Meanwhile the engine keeps in constant rotation an eccentric, which causes the lever G^1 to swing on its bearing journal G, and thereby also the lever E to make a similar motion. The latter then moves the piston slide D evenly up and down.

If all be in order, the piston slide just as evenly opens a discharge channel K, from which the steam below the piston B^1 escapes more quickly than it can stream in through the stuffing box C, as a result of which the spring H closes the valve A. The lever E, which is linked to the lever G^1 at the point E^1, is also, however, influenced by the solenoid F lying in the dynamo circuit. According as the solenoid exerts a greater or a less force of attraction, the slide D closes the channel K sooner or later than the lever G^1 performs this operation in ordinary working. The inlet period is thus shortened or lengthened accordingly. Since the slide D constantly moves up and down, it is not necessary to overcome the friction due to a state of rest.

Parsons[2] further calls attention to the fact that alternators which are driven by steam engines with irregular turning moments cannot suitably be connected in parallel, because the currents will alter in accordance with the changing speed, *i.e.* they will at the various points of rotation show varying phases and voltages. If a turbine regulation be contemplated in which the steam is introduced by stages, it will be desirable to allow the introduction of the steam into all driving turbines of dynamos connected in parallel to take place simultaneously. In case the turbines work together with ordinary steam-reciprocating engines, the admission of the steam should take place at the moment at which the crank-shaft pressure of these latter engines is the greatest. Accordingly, the influencing of every steam-inlet valve by the shaft of the engine in question does not take place. Instead of this, an (electro-) motor, the shaft of which, by the help of eccentrics and link motion, opens all the inlet valves at once, is provided for these. At the same time, every engine can regulate the quantity of steam flowing to it, in accordance with its greater or smaller requirements, by closing its valves at a later or earlier time by the help of the corresponding solenoid.

In the regulation of *multi-stage* turbines it will clearly be necessary to reckon in increased measure with the effects which follow an alteration of the steam admission above or below for the pressure drop, or alteration of velocity in the individual stages.

Thus, in the arrangement adopted by *Curtis*,[3] shown in Fig. 346, in which velocity turbines are connected in series in three pressure stages, the alteration of the width of the nozzle for the first set of wheels must also be followed by a like alteration in the remaining nozzles, if the conditions of the current are not to

[1] A. P. 549,816. [2] E. P. 19031 of the year 1902.
[3] E. P. 19,248 of the year 1896.

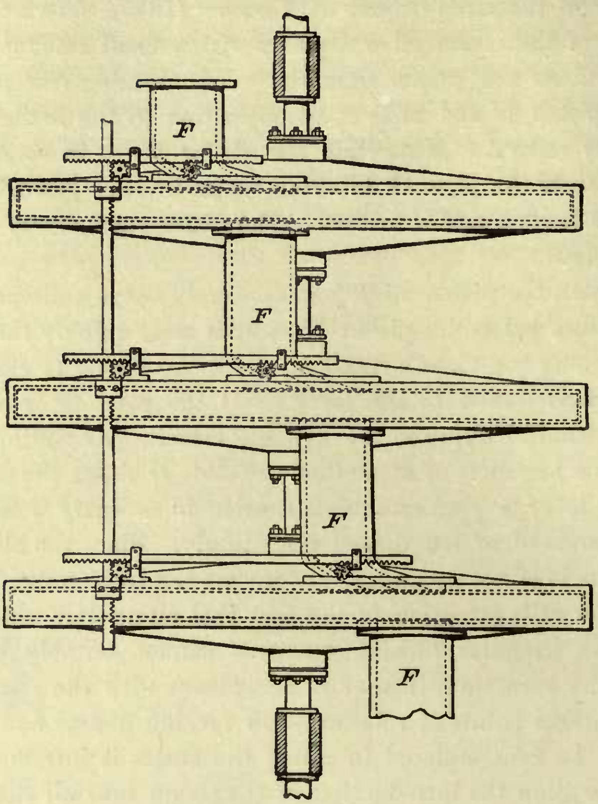


Fig. 346.

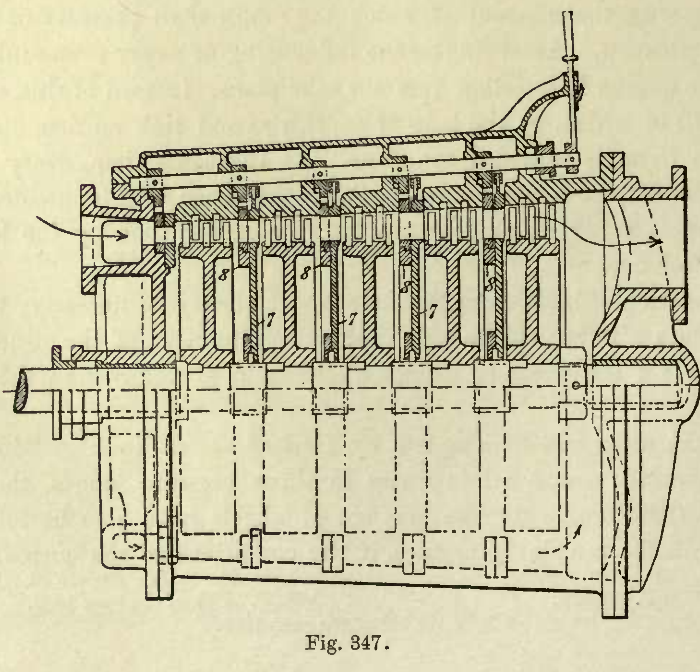


Fig. 347.

be altered. In consequence of this, Curtis connects the regulating gear of the nozzles in the channels F in such a manner that these are forced to act together.

Schulz[1] (1901) rightly observes that the multi-stage turbines can work economically only with a full supply of steam, and that the diminution of the admission of the steam into the first stage, occasioned by a reduction of the load, is accompanied by a relatively poor efficiency of the work of the medium in the turbine. In order, then, to make the most of the expansion of the steam for small loads as well as for large ones, along the whole course of its expansion, he arranges (see Fig. 347 for axial turbines, Fig. 348 for radial ones) adjustable ring slides 8 in front of the inlet openings of all the guide-vane wreaths 7, or of a part of these. The guide-vane wreaths 7 may also themselves be made adjustable, so that a second regulation is rendered possible. Further on, all the ring slides 8 can be made adjustable in common, while the guide-wreath vanes 7 are adjusted singly. Again, all the guide-vane wreaths 7 may be made adjustable in common, while the ring slides 8 are adjusted singly. In the case of a governing appliance in which, in addition to the regulation of the steam consumption so as to correspond with the load, a second regulation, by means of an adjustment of the guide-vane wheel itself or of the seat 7 of the ring slide 8, is rendered possible, a subsequent improvement can be made of the calculated and finished steam sections for any and even

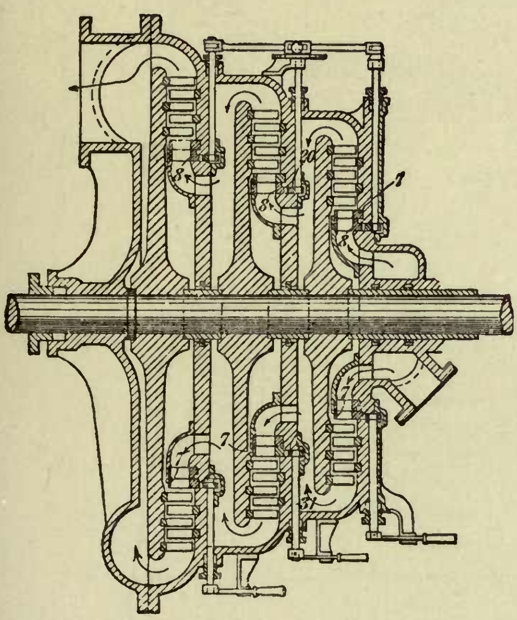


Fig. 348.

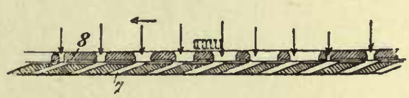


Fig. 349.

for the greatest load. It thus becomes possible gradually to narrow or to widen the inlet openings for each individual turbine section till the measure of pressure assumed for the calculation at each of these points is really attained. This can then be established by means of a manometer. As shown in Fig. 349, the openings in the ring slide 8 are of different sizes, so that the channels of the guide-vane wreaths 7 are closed one after another.

In reference to the above, Schulz makes the following statement: In the multi-stage steam turbines either all the guide-vane wreaths or only a part of them may be provided with the regulating appliance of the leading channels just discussed.

[1] D. R. P. 132,868.

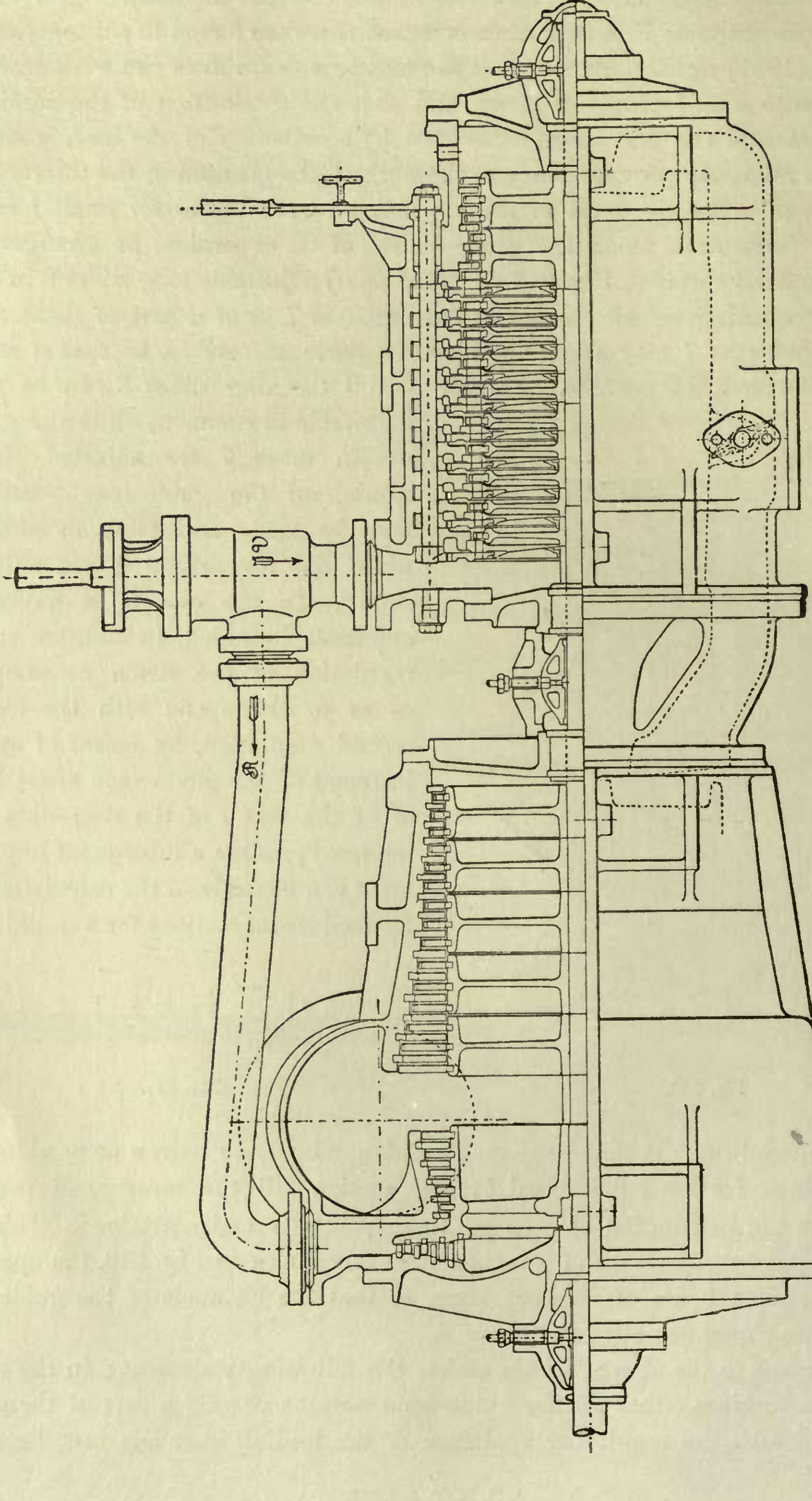


Fig. 351.

PLATE II.

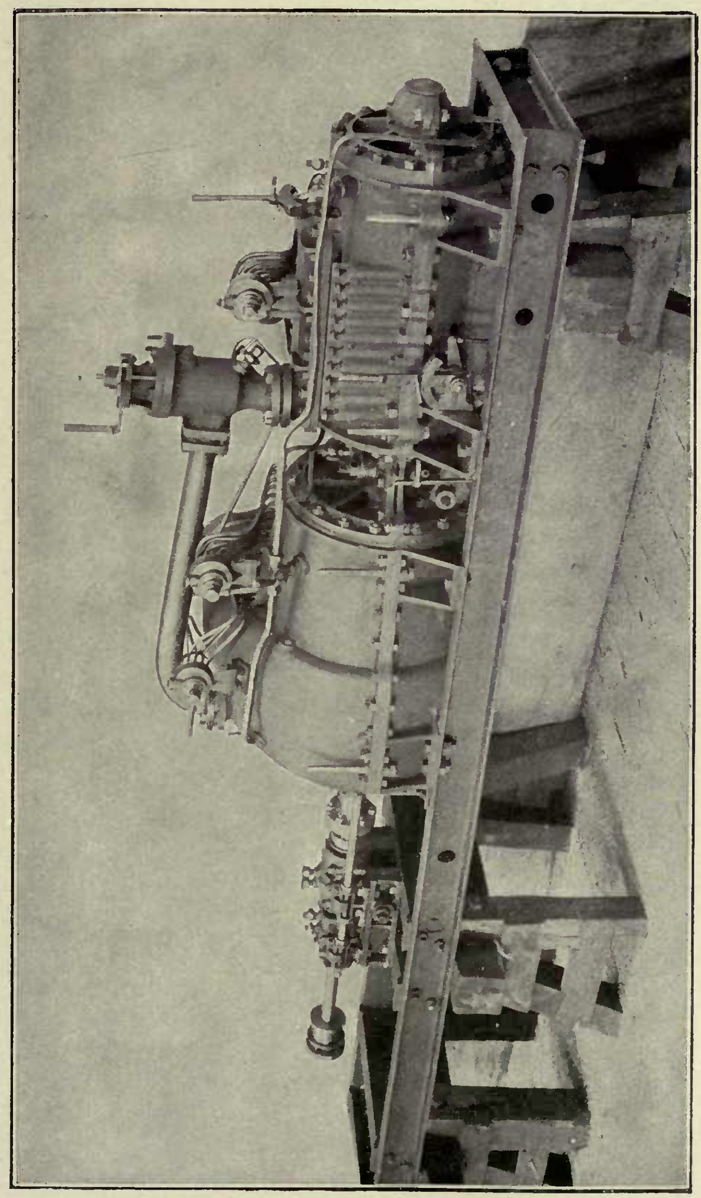


Fig. 350.
Experimental Turbine constructed by Schulz, 1900.

For instance, the first, fourth, seventh, etc., may be so provided, according to the number of pressure stages assumed. The number of these in case of a given total drop in pressure between the inlet and the outlet steam is to be regulated chiefly by the pressure differences chosen for the single pressure stages, and by the speed of rotation or number of revolutions of the turbine. The smaller the number of the pressure stages chosen, the less will the speed of the steam issuing from the leading channels (nozzles), in the case of each pressure stage, be utilized, when only one turbine wheel is made use of for each of these. For this reason it is all the more necessary to allow the steam behind each guide (nozzle) wheel to work upon a certain number of turbine wheels. Schulz accordingly assumes, for instance, for the axial turbine, five adjustable pressure-stage wheels, each with three velocity stages; and for the radial turbine, three pressure stages, each with three velocity stages. With a turbine of this kind some interesting experiments have been made, the details of which, taken from the experimental engine of the year 1900, are shown by Figs. 350 and 351. According to these, the first ten of a total of thirty turbine wheels were constructed on the system described. The experiments showed that it was possible in all cases to maintain the maximum steam pressure of from 176 lbs. to 191 lbs. in the various drops in the impingement of the turbine. As may be seen in Fig. 350, ordinary cocks were made use of for the indicators, from the cases of which communication pipes were led to all the stages, so that each of the latter could be connected with the indicator by the turn of a handle. This turbine was braked to 5000 revolutions per minute when running at the most varying speeds. The maximum of work accomplished by it was 230 E.H.P. Considering the small vacuum in the condenser, the consumption of steam was not unsatisfactory. The boiler had a grate surface of 14 sq. ft., the heating surface being 323 sq. ft., and the superheater surface being 108 sq. ft. With a 7½-fold evaporation, and a superheating of the steam of 220 lbs. per sq. in. to 660° Fahr., the boiler could evaporate 528 gallons of water per hour. The indicator diagrams given in Fig. 352 show all the *steam pressures* in the individual turbine stages for a speed of 1900 revolutions. They were taken by the help of the indicator cocks, in each case, for a turn of the handle. On the occasion of this experimental trial the turbine drove a small boat of 13 tons displacement at a forced speed of 13 knots. In the same figure the steam pressures determined are set off as points in a curve for alteration of volume and pressure. The turbine required from 20 to 25 per cent. less steam when superheating (with a steam temperature of from 600° to 660° Fahr., and an entrance pressure of from 190 to 200 lbs. per sq. in.) was resorted to. The superheating was, in several of the experiments, carried to a steam temperature of more than 750° Fahr. From the indicator diagrams of Fig. 353 the different amounts of work done by the steam in its passage through the various wheels of the high-pressure turbine for three different methods of governing may be read. In the first case all the ten adjustable turbine guide wheels were set at one-sixth passage area; in the second case the same reduction of passage area was made in three only of the turbine wheels, *i.e.* in the first, fifth, and tenth; and in the third case this reduction to one-sixth area was made only in the first turbine wheel. The diagrams clearly show how, when the passage through the guide-vane wreaths is reduced, the partial-impingement turbine wheels in question work as action turbines with high vane pressure and *considerable* drops in pressure, while the rest of the turbine wheels work with weaker vane pressure and *smaller* falls. The following table (on p. 204) gives the performances of the experimental turbine, determined by means of the brake, for speeds of

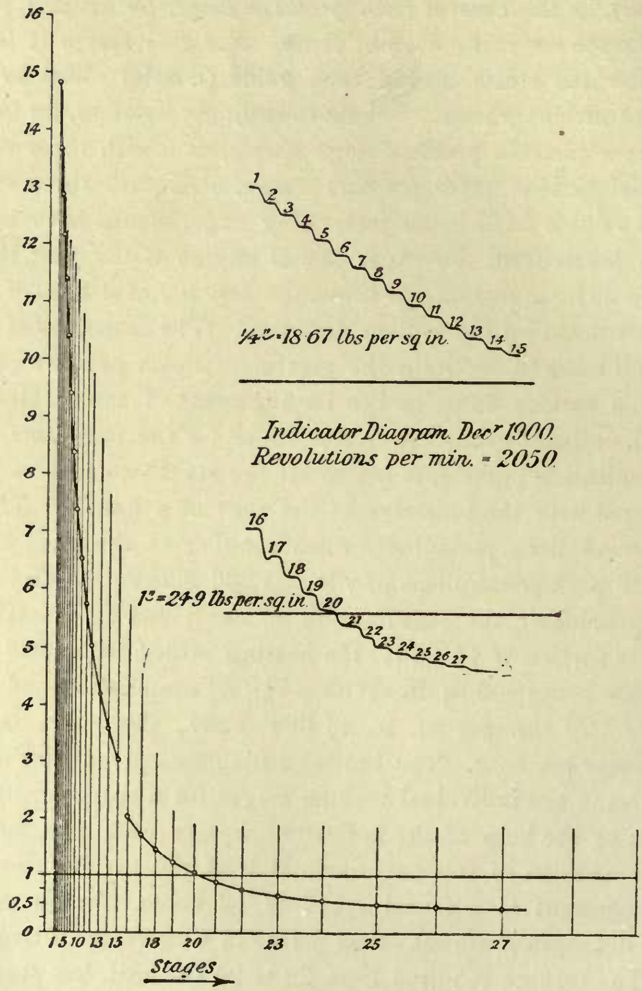


Fig. 352.

from 1600 to 5000 revolutions per minute. The maximum performance was 230 H.P. The weight of the engine, including all fittings, lagging, check valve, and thrust block, amounted to 2443 lbs., *i.e.* not quite 11 lbs. per H.P.

Gross[1] (1902) comes to the conclusion that in order to obtain good regulation in the case of multi-stage steam turbines it is necessary to adjust the passage openings in all leading apparatus in such a manner that the pressure steps in the individual turbines remain constant, and the relations between steam velocity and peripheral

[1] D. R. P. 146,549.

velocity are thus preserved for all loads. For the separate stages arrangements are made by which in each case a flowing pressure medium (steam), actuated by the governor, eventually assisted by springs of different strengths, imparts motion to pistons (Figs. 354 and 355) which are connected with the closing organs of the leading apparatus. Moreover, in this case the pressure chambers *e*, which influence the moving pistons *i*, stand in connection with each other for all turbine wheels. They are themselves

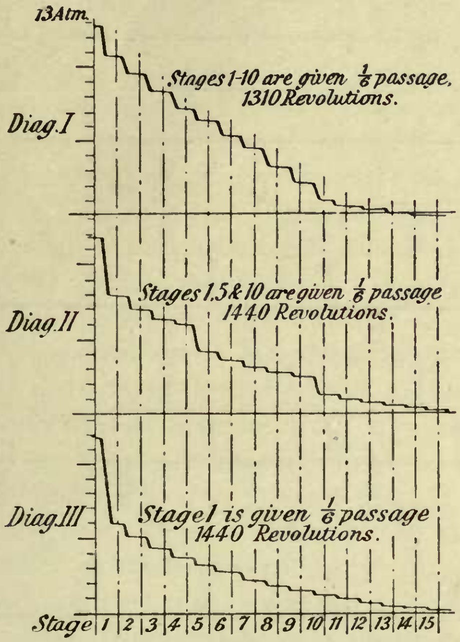


Fig. 353.

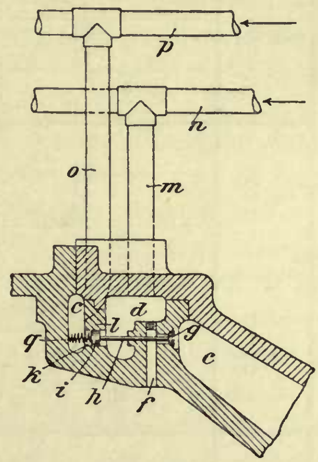


Fig. 354.

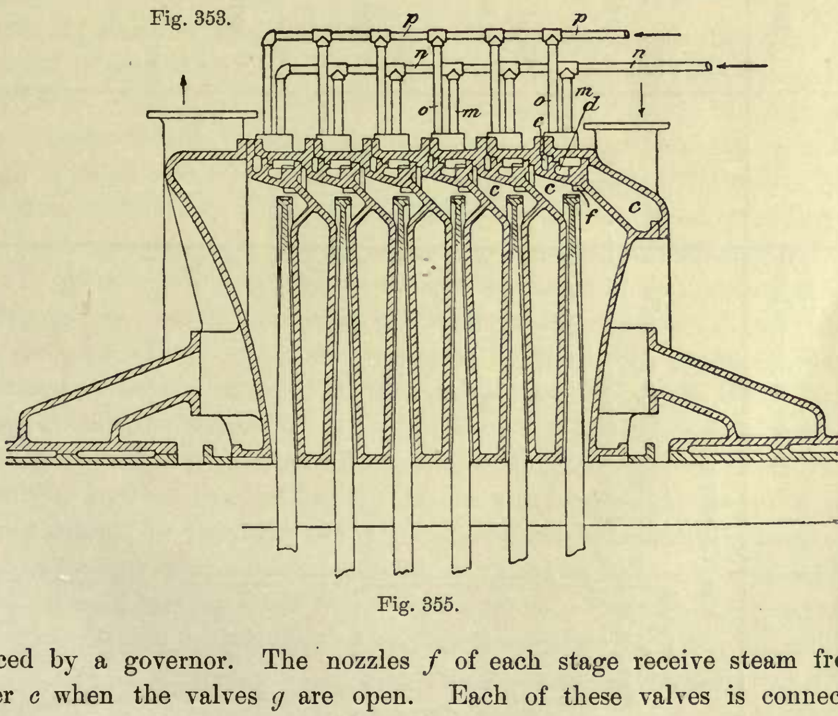


Fig. 355.

influenced by a governor. The nozzles *f* of each stage receive steam from the chamber *c* when the valves *g* are open. Each of these valves is connected by

Schulz Experimental Turbine.

May and June, 1902.

Governing. In front of Turbine lbs. per sq. in.—	Experiment / Revolutions per Minute	1 / 1600	2 / 2000	3 / 2600	4 / 3000	5 / 4000	6 / 4600	7 / 5000
	Steam Pressures (abs.)							
103	,, behind the high pr. turbine a	15·88	15·88	16·25	15·88	16·02		
	,, in the condenser b	10·44	10·73	10·06	10·35	10·25		
	brake H.P.	44·0	53·5	53·5	50·0	62·5		
132	a	16·9	16·17	16·47	16·41	16·17	16·02	
	b	10·73	11·2	10·84	10·74	10·65	11·02	
	H.P.	63·0	78·5	84·0	84·0	86·0	104·0	
162	a	20·42	19·25	19·85	19·85	19·41	19·11	19·11
	b	10·44	11·2	10·52	10·59	10·84	11·02	10·25
	H.P.	78·5	100·0	110·0	112·0	129·0	126·5	121·9
191	a	23·67	22·79	23·53	23·53	7·94		7·35
	b	10·3	10·73	10·65	10·44	10·84		10·44
	H.P.	101·0	118·5	132·0	141·5	1·72		165·8
221	a		26·91	26·62	26·47	26·47		24·99
	b		10·35	10·44	10·64	10·65		10·25
	H.P.		141·5	161·5	177·0	204·0		185·2
235	a	27·58		28·23	27·94	27·94	27·94	27·94
	b	8·9		10·44	10·54	10·44	10·44	10·44
	H.P.	115·0		184·0	200·5	230·0	225·0	226·0

means of a piston rod *h* with a piston *i*. The pistons *i* move in separate borings *k*, which are arranged in the partition wall *l* between the two annular channels *d* and *e*. Each of the annular channels *d* is connected by means of pipe branches *m* with a general supply pipe *n*, and each annular channel *e* by means of pipe branches *o* with a general supply pipe *p*. The pipe *n* leads to the main steam pipe, so that every annular channel constantly contains fresh steam. The pipe *p*, on the other hand, leads indeed into the main steam pipe also, but a throttling organ (not shown) is inserted into it, which is actuated by a governor (also not shown), so that only throttled steam finds its way into the annular chambers *e*. As soon, then, as the pressure of the throttled steam in the chamber *e* (and the spring *q* pressing upon the piston) overcomes the pressure of the fresh steam on the other side of the piston *i*, that is to say, in the channel *d*, the valve *g* will be opened, and the nozzle *f* receive steam. In order that it may be possible at any time to alter the pressure of the throttled steam, there must be a certain current flowing in the ring chambers *e*. In order to attain this, one or two nozzles may be connected with the annular channel *e* direct, so that they receive their steam from the latter, and thus create a current in it. By the choice of springs *q* of various strengths for each group (or corresponding arrangement of the differential piston *i*), it is made possible for the valves *g* of each stage to be successively opened or closed by groups, as the steam pressure varies in the channel *e*. In Fig. 355 is shown a 6-stage turbine of Riedler-Stumpf construction, provided with the governing appliance above described. The working-wheel discs, it may be observed, might with advantage be pretty closely covered in by the chamber walls, so that only small spaces remain for the steam.

If the expansion of the steam through all the stages of a compound turbine, with partial impingement of the first working-wheel wreath, is to proceed in the normal fashion, even when the steam supply streaming towards the high-pressure turbine alters, care must be taken that the steam does not expand either too little or too much in its change from high pressure to low pressure. The impingement of the low-pressure turbine will have to be altered in the same measure as that of the high-pressure turbine. This idea forms the foundation of the method of construction adopted by *Book*[1] (1902), who inserts a governor arranged for a Curtis turbine with several nozzles in front of the nozzles of the low-pressure engine. The piston slide of this governor always has so many nozzles to close, that the pressure in front of the latter is kept constant. In order to displace the piston, the difference in pressure between the steam and the atmosphere can, for instance, be made use of either directly or by the insertion of a hydraulic distributing-valve gear.

Curtis,[2] it may be observed, has worked out a hand-gear for compound turbines of his own design, by which the two ring slides which influence the nozzles of the first and of the second turbine respectively are turned by means of cog-wheel gearing actuated by a governing shaft fitted with a lever. The cog-wheel arrangement for the first turbine is geared in a smaller proportion than that for the second,

[1] A. P. 714,094. [2] E. P. 756 of the year 1902.

so that the ring slide has to move along a shorter path in the first than in the second case, and the nozzles are influenced in correspondingly different degrees.

An alteration in the width of the leading channel is also made by *Kolb*[1] (1902) (Fig. 356) in the regulation of turbines in which the motive medium is led again and again to the wheel by leading channels, the widths of which increase in a degree corresponding with the gradual alteration in the expansion. In the back wall of each channel is a tongue-like spring *d*, one end of which is fixed (at the point *e*). The free end reaches almost to the outlet of the nozzle. The breadth of the spring corresponds with the height of the channel. Behind it, near the free end, a cam is fixed in a recess in the side of the annular channel. This cam is fixed on a shaft, one end of which passes through one of the covers of the channel, while the other end is provided with a lever. The levers actuating all the cams are connected with each other by means of a rod or of a wheel in such a manner that all of them can be worked at once by a single motion. The lengths of the cams are not all the same, but are made to increase gradually in such a manner that the greater the area of section of the channel may be, the longer is the corresponding cam. If, then, the wheel which connects the cams be turned in one direction, the cams will be turned inwards, and the springs will be pressed into the channels. The nozzle-outlet will thus be narrowed. A reversed motion of the wheel produces a corresponding widening of the nozzle openings.

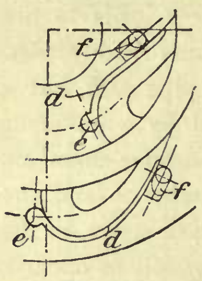


Fig. 356.

At this point an older construction of *Perrigault* and *Farcot*[2] (1864) may be recalled to mind, in which flaps are arranged in the walls of the connecting channels, and can be turned inwards so as to reduce the areas of these.

Turbines with complete impingement may have their power increased by the introduction to their *lower stages* of an auxiliary supply of steam, or of a direct supply of fresh steam, provided they be able to take in a larger quantity of the latter than the higher stages.

Sautter, Harlé, and Co., and *Rateau*[3] (Fig. 357) provide a connection of the steam lead *b* direct with the lower stage *c*, in addition to that with the high-pressure side *a* of the turbine. If the valve *d* be closed, and the valve *e* be opened, steam streams into the second stage *c*, and this in larger quantity than that previously introduced into the first stage, because the leading apparatus of the low-pressure turbine allows more steam to pass through than does that of the high-pressure turbine. In like manner an alteration of the quantity of steam that can be introduced may be made in the case of compound turbines that work on two or more shafts.

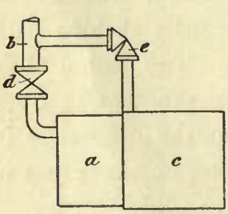


Fig. 357.

The firm of *Brown, Boveri, and Co., Limited*[4] (1902), also lead fresh steam to the lower stages, in order to ensure to the turbine a high degree of efficiency for small

[1] D. R. P. 146,756.
[2] F. P. 65,640.
[3] E. P. 11,701 of the year 1901.
[4] S. P. 25,439.

loads as well as for large ones (Fig. 358). The supply of the fresh steam from the chamber 2 through the pipes 3, 4, and 5 to the various stages is intended to be effected by the adjustment of the piston 7 by the governor, in accordance with the load at any given time. According to the drawing, the regulating piston 7 is in connection with a piston 10, which, on the one hand, is under the influence of a spring 11, and on the other is subject to the pressure obtaining in the regulating cylinder 6. The spring 11 meanwhile endeavours to close the openings of the channels by means of the regulating piston 7. The piston 10, and with it the regulating piston 7, will be more or less raised as soon as there is a sufficient pressure in the regulating cylinder for the purpose. Meanwhile, the pressure in the regulating cylinder depends on the greater or smaller power exerted by the engine. In place of the main turbine governor, the regulating piston could be placed in connection with a separate governor, or, in the case of an electric arrangement, with a magnetic appliance, such as a solenoid, the core 12 of which is connected with the lever 8, and can be placed under the influence of the electric current.

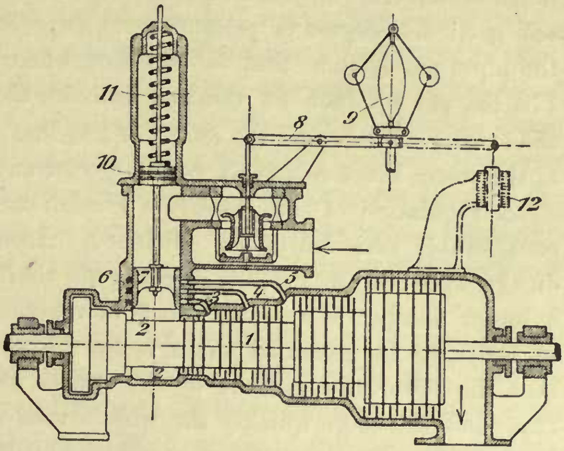


Fig. 358.

Similarly, *Scott* and *Tyzack*[1] lead fresh steam into the lower stages. Before doing so, however, they in each case diminish the pressure of the steam by means of reducing valves to that of the lower stage which is next to be supplied. It would here, then, be a question of the mixture of homogeneous steams, but only of an increase of quantity—not of an increase of the pressure in the lower stage.

In the compound turbines also, in which, for certain reasons, the sets of wheels are placed in separate cases, the same consideration must be exercised, unless the object for which the separation is made alters the conditions.

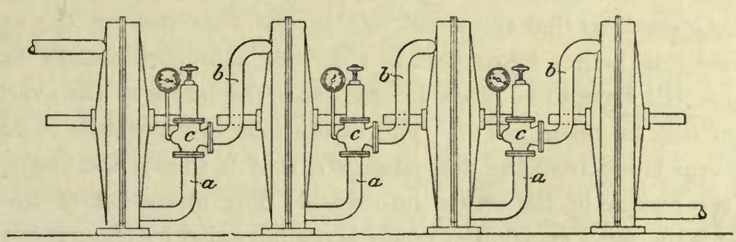


Fig. 359.

Thus *Hörenz*[2] (1901) (Fig. 359) arranges between the cases automatic appliances *c* for regulating the pressure. These provide for a certain drop in pressure between

[1] E. P. 22,740 of the year 1902.

[2] D. R. P. 135,701.

every two turbines, so that an engine is created that runs as slowly as is possible. The pressure-regulating appliances may take the form of the well-known safety or pressure-reducing valves, and must be adjusted in such a manner that only steam of a particular degree of pressure can escape from one turbine to the other through the pipes provided, so that, for instance, when the steam supplied has a pressure of 118 lbs. per sq. inch for the first turbine, the second receives steam of 88 lbs., the third of 59 lbs., and the fourth of 29½ lbs. The result of this is that the pressure of the steam is not relatively very large for any of the turbines.

A regulation of the *number* of *revolutions* is attained by *Curtis*[1] (1897) without considerable variation of the efficiency. According as an increase or a diminution of the speed of rotation of the engine shaft is required, he causes a smaller or a larger number of working wheels of the set driving the shaft, which are placed in series, to be streamed through by the steam. The assumption is here made that the fewer the number of the stages in which the energy of the steam is expended, the more quickly the wheels that are impinged upon must rotate. In Fig. 360, for instance, three working wheels, *a*, *b*, and *c*, are shown in separate cases. The wheel *a* receives fresh steam from the nozzle *d*, and then passes on steam that is deprived of one-third of its energy through the intermediate passage *e* to the nozzle *f* of the second wheel *b*. From this the steam streams through the passage *g* to the nozzle *h*, and through the wheel *c* to the outlet *k*. This and the intermediate passages *e* and *g* are connected with the exhaust, or with the condenser, and the passages *e*, *g* also with pipes *l*, *m*, which can be shut off. If, now, the steam streams through all three turbines, *a*, *b*, and *c*, one after another, the smallest number of revolutions of which the engine is capable will be attained. But if, for instance, the passage *g* towards the exhaust or towards the condenser be opened, so that the steam can escape after passing the second wheel, and the wheel *c* no longer takes part in the work, a proportionately larger number of revolutions will have to be made in order that the whole of the available energy of the steam may be made use of. The revolutions increase still more in number when the steam is led from the first passage *e*, that is to say, the first turbine *a* has to convert the energy of the steam into work. The alteration of the number of revolutions for the same exertion of power is accompanied by the consumption of the same quantity of steam, and since working wheels *a*, *b*, *c* convert equal portions of the energy of the steam into useful work, there will, for instance, when only two wheels are used, be a larger quantity of steam to pass on from the first wheel *a* to

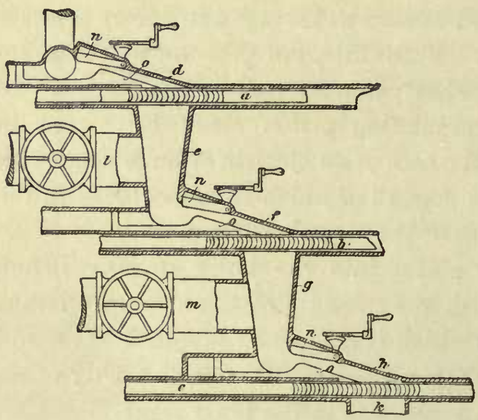


Fig. 360.

[1] A. P. 595,435.

the second one *b*, than when all three wheels are at work together, this being due to the more advanced degree of expansion in the first case of the steam in front of the wheel *a*. A possibility of altering the conditions of supply of the steam must, therefore, be provided. Curtis now provides the nozzles *d, f, h* with flaps *n* that can be actuated singly or together. These form side walls of the nozzle and serve to regulate the angle of its divergence, and therewith the degree of expansion of the steam within it.[1] In addition, each of the nozzles has an adjustable tongue *o*, which at the same time forms one of its sides, and which alters its sectional area only without affecting the expansion. The tongues *o* make the passages through the nozzles narrower or wider, according as the volume of steam to be passed through is larger or smaller, *i.e.* according as the steam has to expand through a smaller or through a larger number of nozzles.

The arrangement can also be so made that, for instance, when three stages are combined the first of these is impinged upon by three nozzles. The steam issuing from one of the nozzles must then expand through all three turbines. On the other hand, the steam from the second nozzles imparts its energy in only *two* stages, while the energy of the steam issuing from the third nozzle must be used up in the first turbine, *i.e.* within *one* stage. Opportunity is thus given, by the opening of the steam inlet, to one or other of the nozzles to establish connection with a larger or smaller number of turbines, and thus to govern the speed of rotation. Incidentally, Curtis calls attention to the circumstance that when only one working wheel is in use it is possible to govern its speed by allowing the steam to work repeatedly upon it, and by a suitable variation in the number of the repetitions.

An automatic stage-by-stage regulation is also aimed at by *Hedlund*[2] (1903) (Fig. 361). He arranges, for instance, three balanced valves *b, c, d*, in the valve chest *a*, the rods of which he leads, with the pistons *e, f, g*, with a certain amount of play, into valve-chamber covers, so that steam can pass from the chamber into the space *h*. The springs *k, l, m* try to press the valves into their "shut" positions, and the width of passage of the outflow pipe is influenced by a governor. The fresh steam proceeds through the branch *o* into the valve chamber. If the pipe *n* be completely open, the steam which flows past the pistons *e, f, g* can produce only a small pressure in the space *h*, and the valve pistons will be raised and the valves held open. If, however, the pressure in *h* increases on account of the throttling of the outflow pipe *n*, it will, with the help of the springs *k, l, m*, finally outweigh the influence of the fresh steam in *a*, and the valves will be closed. The springs *k, l, m* are of different strengths, in order that in the case of a fluctuation of the pressure in *h* the valves will be brought into operation one after another.

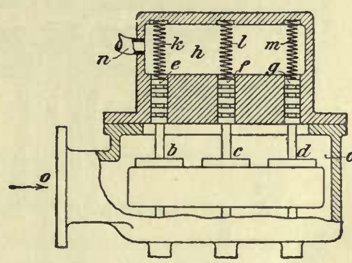


Fig. 361.

This governing appliance can, for instance, be fitted in front of *Lindmark's*

[1] A. P. 589.466. [2] A. P. 742,422.

turbine, in which three working wheels p, q, r (Fig. 362)—corresponding with the number of the valves—are contained in three chambers which are separated from one another by the partitions bearing the leading apparatus. The valve b now opens to the chamber s, the valve c to the chamber t, and the valve d to the chamber u. If all the valves be open, only the wheel r will work, since the full opposing pressure of the steam bears on the two other wheels. But if the valve d be closed, the wheel q will come into operation and the steam will work in the two stages q and r. If the valves d, c be closed, the steam streams through the first wheel p and work is obtained in the three stages p, q, r.

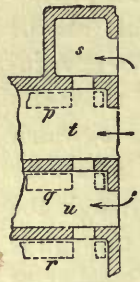


Fig. 362.

A regulation of the number of revolutions of a turbine by the cutting off of the intermediate channel leading from the one stage to the other, so as to cause an alteration of the number of stages, is also contemplated by *Stumpf*.[1] In turbines, indeed, which are divided into velocity stages, the leading apparatus are to be connected and disconnected, while in pressure-stage turbines the separate stages can be put in direct connection with the condenser.

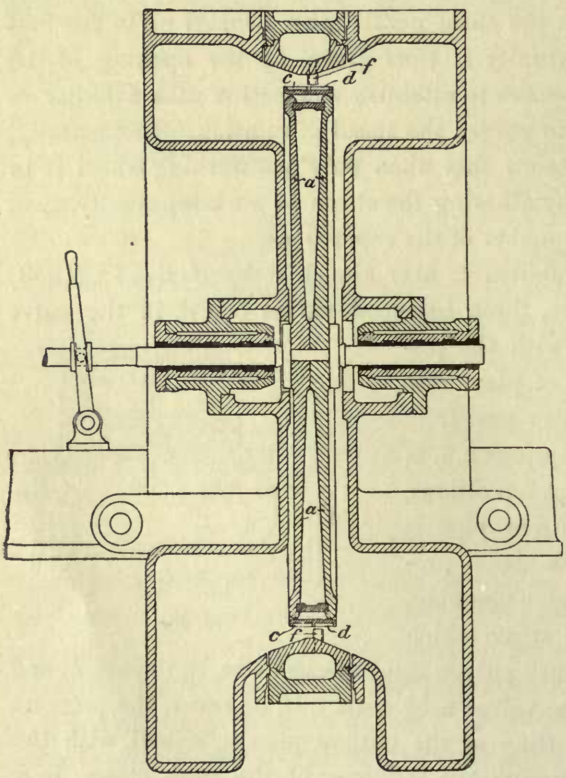


Fig. 363.

For a concussionless entrance of the steam into the working wheel, which must in all cases be aimed at, the speed diagram shows a definite connection between the number of revolutions of the wheel, the vane angle, the angle of entrance of the steam, and the entrance velocity. If the necessity exists—as, for instance, in the case of a marine engine — for varying the number of revolutions of the wheel, one of the other quantities must undergo corresponding alteration.

A partial impingement of the channels would allow of the nozzles being set at different angles. Another arrangement, which would also be suitable for turbines with full impingement, has been proposed by *Gross*[2] (1902). According to this, several vane wreaths with differently formed vanes are arranged in such a manner that by the shifting of the working wheel or working wheels, or of the nozzles, the

[1] E. P. 18,952 of the year 1902.
[2] D. R. P. 147,354. See also Stumpf, E. P. 356 of the year 1903.

combination of nozzles and vanes suitable for any number of revolutions may be brought about. Fig. 363 shows an arrangement according to which the axially movable working wheel *a* bears two groups of vanes *c* and *d*, and in which *f* denotes the fixed nozzles. According to Fig. 364 the wheel *a*, with the groups of vanes *c* and *d*, is immovable in the axial direction, while the nozzle *f* is adjusted axially in accordance with the number of revolutions desired.

As an instance, the vane arrangement of a Laval turbine is illustrated (Fig. 365). The absolute velocity of entrance of the steam W*e* is everywhere equal to 3200 ft. per second. Speeds of 3000, 2200, and 1500 revolutions per minute are contemplated for the turbine. These correspond with peripheral velocities—$V_2 = 1150$ ft.,

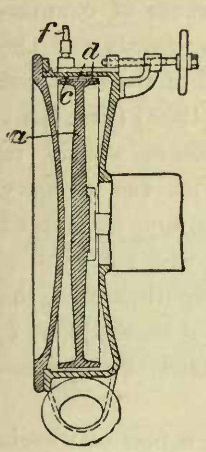


Fig. 364.

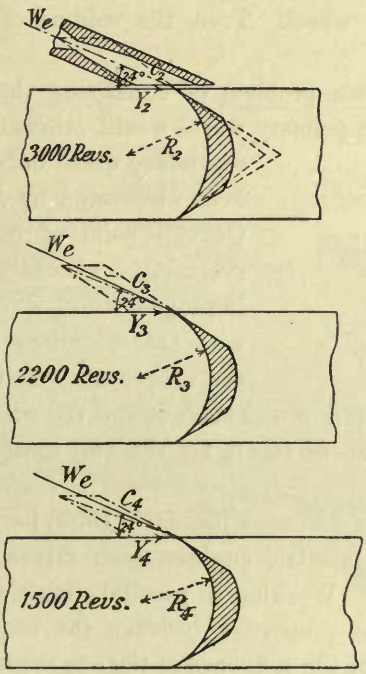


Fig. 365.

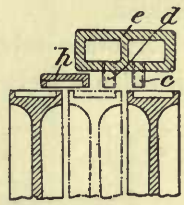


Fig. 366.

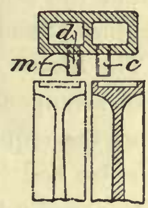


Fig. 367.

$V_3 = 850$ ft., and $V_4 = 575$ ft. The relative entrance velocities thus resulting for the respective wheels are $C_2 = 2215$ ft., $C_3 = 2430$ ft., and $C_4 = 2660$ ft., an angle of inclination of 24° being adopted for the nozzle in each case. The surface of contact of each vane for the steam is circular, and the vanes given are so constructed that the lines *c* form tangents to the vane circles. The resulting radii for the vanes are of different lengths. For the conditions given in the figures they are $R_2 = 2\frac{1}{2}$ ins., $R_3 = 2\frac{3}{8}$ ins., and $R_4 = 2\frac{1}{4}$ ins.

An alteration of this appliance for the adjustment of the revolutions is made by *Gross*[1] (1903), who provides the working wheel with one row of vanes only. On the other hand (Figs. 366 and 367), several nozzle wreaths *c*, *d* are arranged side by side in an axial direction in the fixed guide wheel *e*. These act alternately on the

[1] D. R. P. 147,355.

axially movable working wheel, and in every wreath their nozzles have a different pitch, corresponding with the number of revolutions. With this it is possible to arrange for one or for several systems of reversing or of guide vanes h or m respectively, which, in the wheel-position corresponding with a smaller number of revolutions, either lead the steam back several times into the same working-wheel wreath, or lead it into a second wheel, a third one, etc.

In the case of *counter-direction wheels,* the first of which delivers its steam directly against the channels of the second one, these have been observed to perform different amounts of work, the primary wheel (the first stage) taking up considerably more than half the energy of the steam. In order to equalize the performances of these, *Stumpf*[1] supplies both wheels with fresh steam, so that each of them works both as primary and as secondary wheel. True, the volume of the steam must here be divided.

Another solution of this problem of equalizing the performances of counter-direction wheels, where the primary wheel would otherwise give two-thirds and the secondary wheel only one-third of the total, has been discovered by *Behrisch*[2] (1903) (Fig. 368). Of two pairs of counter-direction wheels, or systems of wheels d^1c^1 and c^2d^2, the primary impinged wheel d^1 or c^2 of the one is coupled with the secondary impinged wheel c^1 or d^2 of the other pair. In the example illustrated, the wheels d^1 and d^2 are mounted on the shaft a, and the wheels c^1 and c^2 on the shaft b, so that the same performances result for the two shafts and for the two dynamo machines f^2 and f^1.

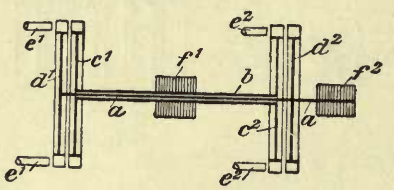


Fig. 368.

The *Siemens and Halske Aktiengesellschaft*[3] (1903) have coupled a pair of wheels by an arrangement of two electric engines, each driven by a wheel, and together giving the required power. Working in parallel circuit, these engines determine by their size and influence the proportion between the respective speeds and powers. By means of a weakening of the influence of the one engine, the speed of the working wheel in connection with it is increased. On the other hand, differing speeds of the two wheels can be equalized by a corresponding adjustment of the magnetic field. It is thus possible within certain limits, by an adjustment of the field-magnet system, to impart different speeds to the wheels, and to distribute the power required between the two as may be required.

If the turbines, thus set one behind another, which we may assume to be counter-direction wheels, be each driven by a separate engine, the speed of each wheel must of course be governed separately in accordance with the amount of energy required from its driving engine. For this reason the *Vereinigte Dampfturbinen-Gesellschaft* (1903)[4] supplies each engine with a governor, which influences the fresh-steam pipe by means of a throttling or closing organ, so that in case the normal steam supply

[1] E. P. 269 of the year 1903.
[2] D. R. P. 149,606.
[3] D. R. P. 150,990.
[4] D. R. P. 153,143.

of either turbine be exceeded, the total steam supply is reduced. If only one governor be made use of, the apparatus can be so arranged that, for instance, by the insertion of differential gear, the quicker-running turbine always acts upon the governor.

Mention may further be made of a special method of *regulating* the current of steam in the turbine, acting in a manner similar to that of the distribution gear in the steam-engine, in which checking appliances are arranged in the case in front of the separate guide-vane wreaths. *A. C. Th. Müller*[1] (1903) arranges this appliance so as to be driven from the turbine itself, with which it is in fixed connection, and at certain regular intervals to check the current of the steam. The designer assumes

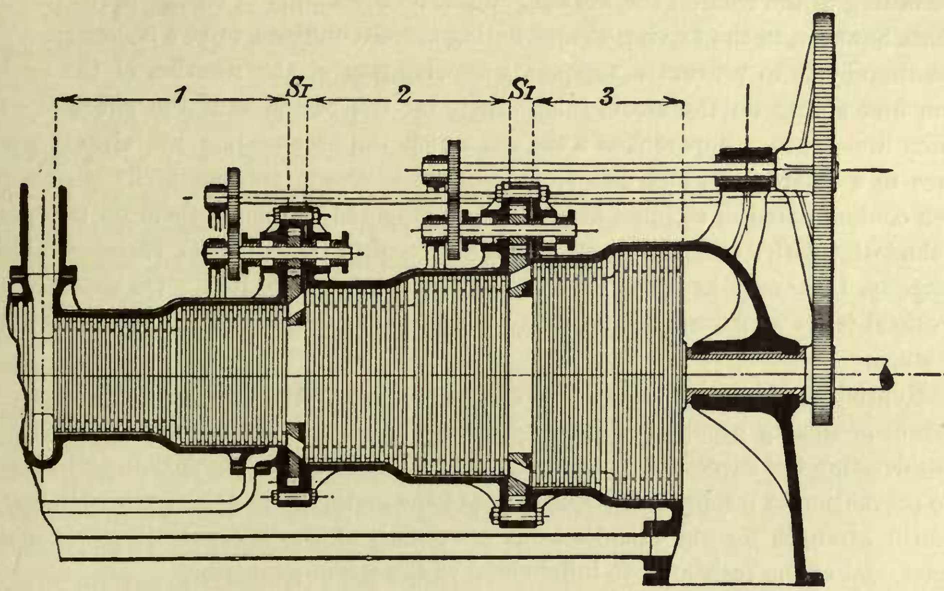


Fig. 369.

that, by the checking of the current flowing in from the inlet side, the steam is periodically made to collect in front of the checking organ, while at the back of the latter it is free to stream away to the condenser or into the atmosphere. When there are several checking appliances, such, for instance, as turning slides (Fig. 369), driven by gearing from the turbine shaft, and coming successively into action, the steam may perhaps first pass through to the checking organ S^1, and there collect, while it is streaming from the groups of wheels 2 and 3. The steam then passes unhindered through to the second checking organ, S^1, before which it again collects while streaming out from group 3. The next collecting together of the steam takes place in front of the succeeding checking organ, and so on, till the process begins over again. Since the passage of the current in single sections takes place to a certain extent in jerks, and the steam comes to rest periodically at the various points, a superheating of the latter can here take place. The object of this arrangement,

[1] D. R. P. 152,474.

which can take the form of an exactly working steam distributer, is the suitable reduction of the revolutions of the turbine, combined with an increase in its efficiency. It may here be observed that the interruptions at the points at which the steam collects clearly cause concussions, which result in destruction of energy.

With an unvarying supply of energy and an unvarying load, the turning moment of the turbine remains constant. It is clear, then, that the action of a *fly-wheel*, which has now and then been borrowed from the reciprocating engine, is, in general, unnecessary. By means of the rotating masses, the speed with which the governing of the engine in accordance with the fluctuating demand for energy takes place is even reduced. A delay of this kind in the reaction effect, exercised by the rising and falling of the load on the working engine on the supply of energy to the turbine, seems, however, in the driving of alternating current engines, to be a necessity. The aim thereby is to prevent a temporary acceleration of the rotation of the anchor from first acting on the steam-inlet organ, for the reason that the effect of such action first becomes apparent at a time at which the acceleration has already given place to a retardation, and *vice versâ*. For this reason the use of fly-wheels has been contemplated in turbines also, and it is proposed to mount them on the shafts of the latter with a certain amount of freedom to turn. The *British Thomson-Houston Company, Lim.*[1] only keys the nave of the wheel fast to the shaft. On the nave the fly-wheel is set with springs, and thus has a certain play, which is, however, limited by stops.

Mention may here suitably be made of the automatic governing appliance for current-producing groups designed by *Routin*[2] (1901–1902), which also takes into consideration the successive phenomena of acceleration and retardation, and aims at the prevention of mechanical disturbances of the motion. For this purpose, however, Routin arranges for the simultaneous governing of the exertion of power of the motor, and of the resistance to influencing of the steam generator.

[1] E. P. 23,393 of the year 1900. Compare also *E. Rosenberg*, Anforderungen an Antriebmotoren beim Parallelbetrieb von Wechselstromdynamos, *Zeitschr. d. Ver. d. Ing.*, Vol. 48, p. 793.

[2] D. R. P. 138,118, 144,051, and 146,525.

XIII

REVERSING GEAR

THE reversibility of steam turbines for conditions in which the motor is used for approximately equal exertions of power in right-hand and in left-hand motion has become a problem of importance, the complete solution of which is still being sought after. Where circumstances admit of it, the *alteration of the direction of rotation* of a working wheel can be effected very simply by a reversal of the direction of the working steam with reference to the parts immediately influenced by it. An expedient of this kind of course presupposes a suitable arrangement of the working wheels and of the steam-supply leads. This again brings us back to the primitive impulse wheels, for the radially set flat vanes of which it, for instance, sufficed to provide nozzles for right-hand and left-hand motion, into which the steam was alternately led by suitable switch arrangements.

MacArthur and Smith[1] lead the steam twice during the revolution of the wheel to the nozzle that just happens to be in action, from a constantly rotating turning-slide which is actuated by means of eccentrics from the shaft. A sleeve inserted between the slide and its chest now opens the way, according to its adjustment, for left-hand or for right-hand motion.

The turbine of *Gray* and *Bass*[2] (1898) (Fig. 370) is said to admit of easy reversal. It contains no vanes, but in place of these are a number of discs or rings D, *d*, set, free to turn, in a cylindrical case A, and fixed rings F, *f*, which are provided with passages E, *e* corresponding with one another and set at an angle one with another. The channels E, *e* have the same section throughout their length, and all the channels E of the rotating discs D work simultaneously with the opposite channels of the fixed rings. The direction of motion is here dependent on the supply of the pressure medium from the one or the other side of the set of discs. The lubricating medium is introduced through the channels N, *n*. The steam pressure can be taken off the side walls of the case A by the insertion of discs M. V is the reversing slide. A practical value can hardly be assigned to this arrangement.

Gardiner[3] has adopted the arrangement illustrated in Fig. 371 for a multi-stage turbine. The steam streaming in at *a* is intended to impinge in a tangential direction successively on wheels mounted on the shaft and contained in separate chambers *b*, *c*, *d*, etc. In the steam inlet *a* and in the communication channels

[1] E. P. 19,823 of the year 1902.
[2] D. R. P. 110,801.
[3] E. P. 15,635 of the year 1902.

between the successive turbine chambers cocks *e* are inserted. These cocks are connected with a single handle, and it is possible by a movement of the latter to turn them all in such a manner that the steam streams to all the impingement nozzles either for right-hand or for left-hand rotation. The middle position of the handle causes the cock in the inlet branch to shut off the steam.

Parsons and *Swinton*[1] (Fig. 372) allowed themselves to be influenced by the consideration that, in the use of a set of nozzles and vanes kept clear of the operation of reversing, marine turbines would be subject to certain disadvantages, in which case the steam from the main engine is first shut off and then led through the reversing nozzles or vanes. By means of this appliance the driving direction can indeed be reversed, but the power acting in the backward direction is insignificant in amount,

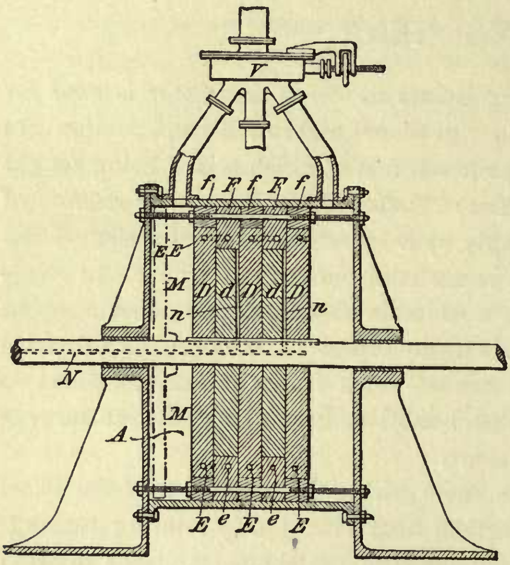


Fig. 370.

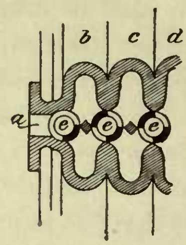


Fig. 371.

and cannot be increased to any considerable extent, because the use of a greater number of reversing nozzles would, on the one hand, make the construction of the turbine too complicated, while on the other hand it would entail too great a reduction in the power for forward motion. Now the Parsons turbine has been shown to change its direction of rotation when steam is introduced from the low-pressure side. Let shafts A, B be assumed which are separately driven by special turbines E, F placed one behind another, the vanes of which, partly fixed and partly movable, are so arranged that they produce the well-known expansive action of the steam as it passes from one set of vanes to the next. The reversal will here be effected in each turbine by the changing of the direction of motion of the steam entering by the branch H and passing through the pipe D from the first turbine to the second one. The auxiliary appliances for changing the direction of the current are set in such a manner that the high-pressure steam enters at the low-pressure end of the high-pressure turbine, and escapes at the high-pressure end of the same, to be passed onward to the low-pressure end of the next turbine, and so on through all the stages. In a reversing arrangement of this kind a backward driving effort of considerable power is attained, in spite of the loss which takes place in the move-

[1] D. R. P. 103,614. E. P. 901 of the year 1897.

ment of the steam from the vanes with small impingement surfaces to those with larger ones. The whole arrangement is simple, and will in many cases prove satisfactory. When four or more turbines are placed one behind another, the reversing arrangement becomes more favourable, because the difference between the conditions of the steam in the successive stages is proportionately smaller. Several turbines thus arranged can be mounted on a single shaft, and either all the expansion stages or single ones of these can be fitted with the reversing arrangement. A difference lies in the circumstance that the reversal is effected by the admission of the steam in the direction in question to each set of vanes. This arrangement increases the reversing power, but it destroys the simplicity in the arrangements of the steam supply.

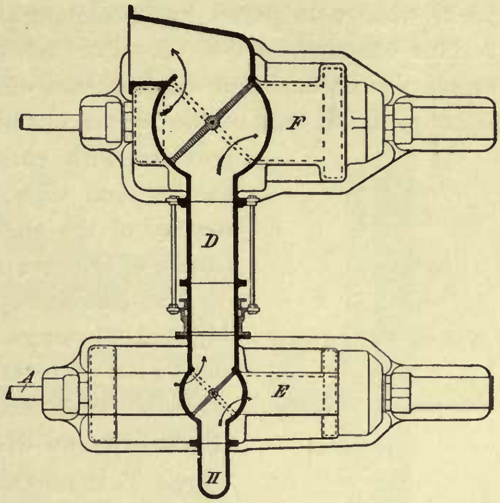


Fig. 372.

To enable the Parsons turbine to be efficiently reversed when running at full

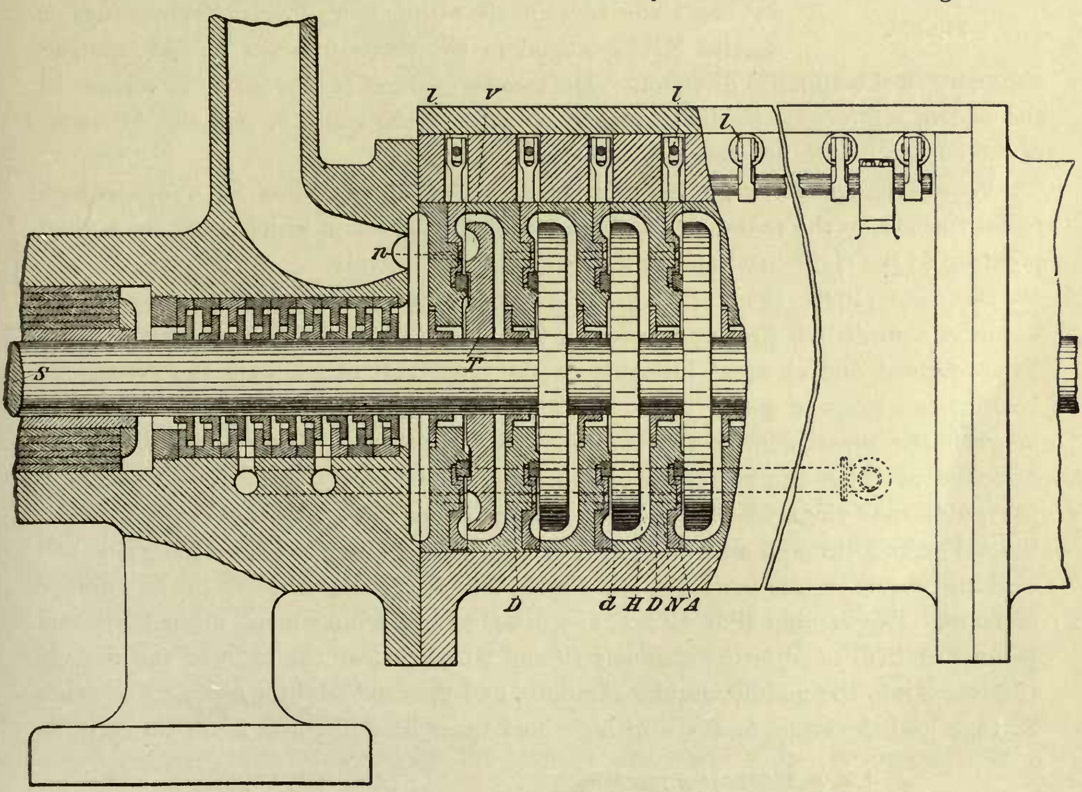


Fig. 373.

speed, *McCollum*[1] also connects the turbine chambers that do not adjoin one another in such a manner that (in the case of motion ahead) the steam from the main pipe is admitted at the fore side of the first chamber, and escapes from it at the back, whence it passes by a communication pipe into the head side of the next turbine chamber, and so on. For backward motion, on the other hand, the main steam pipe is connected with the back of the first chamber, and the head side of the latter with the back of the second chamber, and so on. The connecting leads are provided with check valves which are connected one with another and with the cock in the main steam pipe. The reversal of the engine can thus be effected by a turn of the handle of the said cock.

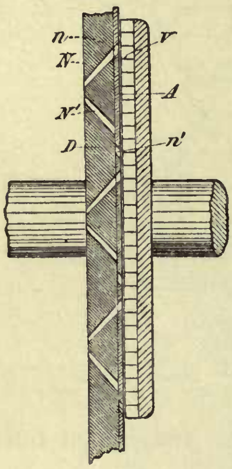


Fig. 374.

The leading appliances, however, which have differently adjusted openings, can also be made movable, so that the engine can be reversed by shifting their positions. Thus the reversible turbine designed by *Clarke* and *Warburton*[2] (1898) (Figs. 373 and 374), with vanes set symmetrically on the wheels T, is provided with sliding pieces A having openings or nozzles $n\ n^1$. These pieces A are arranged in or beside the partition walls D in such a manner that they can be turned by means of suitable levers *l*. By the turn so given, the nozzles nn^1 are made to coincide with one or other of the openings or nozzles NN^1 arranged in the partition walls D and pointing obliquely in the opposite direction. The motive medium is thus made to stream in the one or the other direction opposed to that of the vanes *v*, and the reversing of the engine is thus effected.

McCollum and *Foerster*[3] have produced a similar appliance for a multi-stage radial turbine, in the radially turnable leading appliances of which slits are worked pointing to the right hand and to the left hand respectively.

Burgum[4] (1899) (Figs. 375 to 377) attains the reversibility as follows: by means of equi-distant projecting wreaths 29 on a revolving drum two rows of vanes 24 are formed, and at equal intervals on the inner wall of the case the wreaths 27 bearing two rings of guide vanes 23, 28, set in opposite directions, are arranged between the wreaths 29 of the working-wheel vanes 24. Thus, according to the direction in which the steam streams through the turbine, the one or the other of the guide-vane rings 23, 28 serves as leading apparatus. Fig. 375 shows a longitudinal section through a turbine, Fig. 376 the order of succession of the guide and working wheels on a larger scale, and Fig. 377 cross-sections through the two groups of vanes. The turning slide 10 can, according to one arrangement, allow the steam (as in Fig. 375) to enter the channels 19 and 20, so that at the ends of the turbine it streams into the middle annular channels, and then out of these again. The vanes 23 then lead the steam to the working-wheel vanes 24, while the vanes 28 serve to

[1] E. P. 20,099 of the year 1898.
[2] D. R. P. 109,973.
[3] E. P. 9206 of the year 1902.
[4] D. R. P. 119,818.

carry it off again. By another arrangement the position of the turning slide is such that the steam passes into the middle annular channel 44. In this case the fixed vanes 28 act as supply leads for the steam, and the vanes 23 lead it away. In compound arrangements the turbine has three or more concentric working-wheel vane drums on a common axis in connection with a corresponding number of cylinders having guide vanes, through which in succession the steam streams.

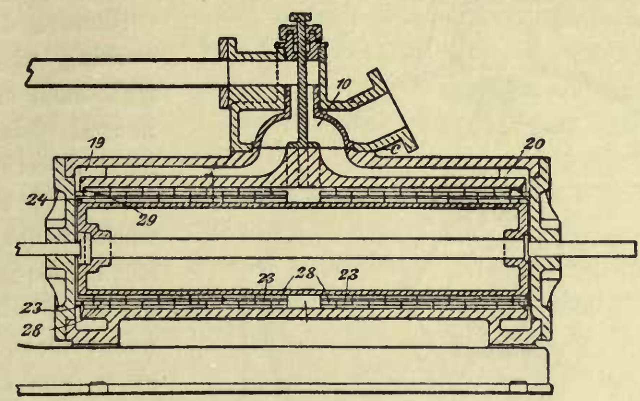


Fig. 375.

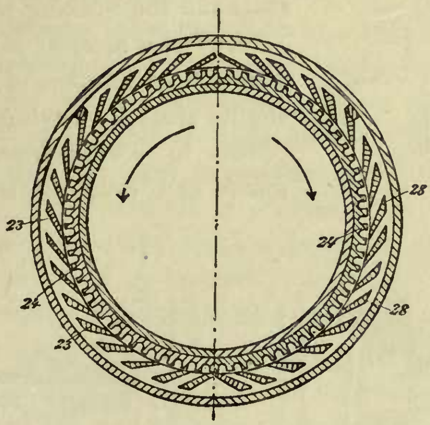


Fig. 376.

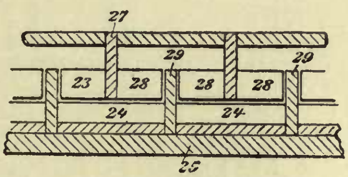


Fig. 377.

In the pressure (reaction) wheels that work without leading apparatus, the turning of the outflow nozzles must clearly alter the direction of rotation of the wheel. Thus, in the design by *Procner*[1] (1901) (Fig. 378 and 379) the nozzles 2 are brought into connection with the shaft 1, which is to be driven in such a manner that the nozzles can be turned through an angle of 180° round the longitudinal axis of the nozzle-pipe O which bears them, so that the steam can issue and act in the one or the other direction, as may be desired. That the steam-nozzles are able to turn presents the further advantage that a regulation of the performance of the engine becomes possible also in the case in which the throttling or partial checking of the steam may be impracticable; for the greater the angle which is formed by the

[1] D. R. P. 132,251. See also E. P. 3937 of the year 1902.

nozzles with the plane in which they turn, the smaller will be the power exercised on the driving shaft. The nozzle-pipes O are free to turn in the body 5. The turning motion is effected by a movement of the rod 8, which works on the pipes by means of screw and nut, and of bevel wheels.

In the cases hitherto treated of it has always been necessary to alter the direction of the steam entering the turbine, and to bring the working wheel over from right-hand motion past the neutral point to the left-hand direction. If the direction of rotation of the wheels is not to be reversed, and consequently the steam not to be admitted by different inlets, it is possible—apart from the insertion of gearing between the turbine shaft and the working shaft—to effect a reversal of the direction of rotation of the engine shaft by having recourse to *counter-direction wheels* and coupling these with the shafts as required. This device effects a quick reversal, but must for this very reason be made use of only with special care.

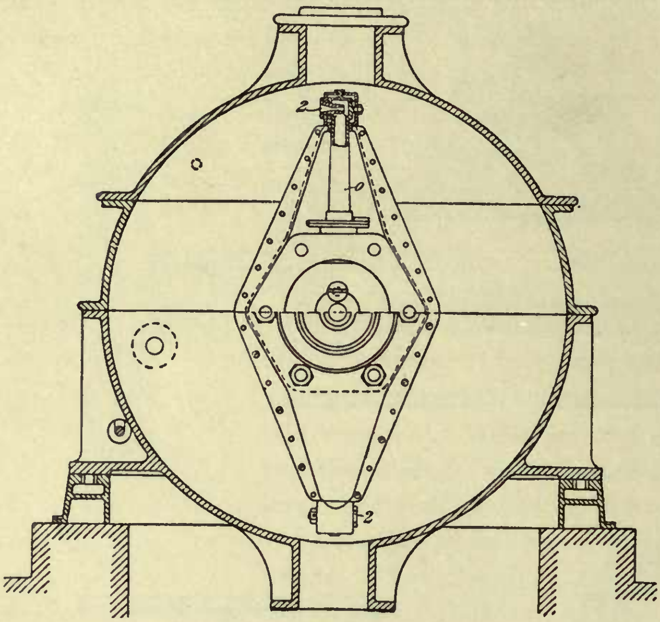


Fig. 378.

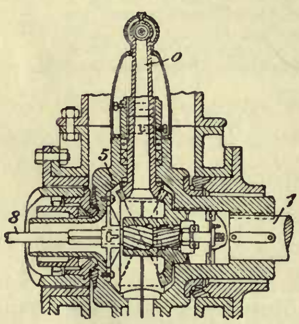


Fig. 379.

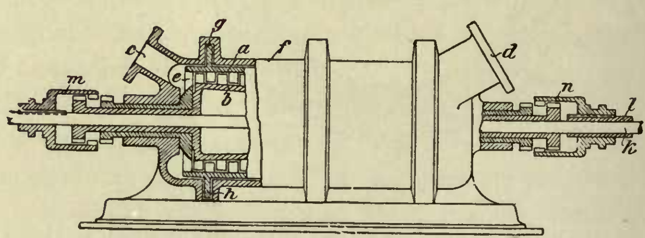


Fig. 380.

The first case here calling for consideration is that in which movable couplings accomplish the reversal. The turbine designed by *Ashton*[1] is provided with such (Fig. 380). Here two (or more) wheels a, b are arranged concentrically with one another, and rotate in opposite directions. The steam entering through the branch c, and escaping through the branch d, is not to act in a tangential direction on the movable vanes direct, and thus set up a back pressure on the fixed case; but it must

[1] E. P. 4295 of the year 1900.

first pass the openings *e* of the wheel *a*, the side walls of which lie parallel with the axis of the turbine before it impinges on the adjoining vanes of the wheel *b*, so that the reaction is taken up by the wheel *a*. Also, the steam inlet of the case *f* must be so placed that the inflowing steam does not exert a reaction tending to turn the case about the axis of rotation of the turbine. The outer turbine wheel *a* must, of course, at its periphery, work against the case *f* without leakage. With this view, annular slots *g* are provided in the case, and suitable flanges *h* on the working wheel; the space between the two is filled by a fluid (eventually by condensed water), which is kept in the slots by centrifugal force. In the drawing it is assumed that two concentric shafts *k*, *l* (for twin-screw steamers, for instance) form part of the arrangement. By means of a simultaneous movement of the couplings *m*, *n* to right hand or to left hand, it will be possible either to couple the shaft *k* with the turbine *b*, and the shaft *l* with the turbine *a*, or to couple *k* with *a* and *l* with *b*, thus setting the shafts for right-hand or for left-hand rotation respectively. If only one shaft be assumed, the arrangement may be so made that the one coupling does not turn, but only holds fast the one or the other of the turbines, so that it simply acts as leading apparatus for the rotating wheel that is for the time being coupled with the shaft. If two pairs of turbines be adopted with axes coinciding with one another, through which the steam entering between the two chambers streams in opposite directions, their axial seating pressure is thereby eliminated.

Henning[1] (1901) makes use of the wheel for the right-hand motion, and of the case for the left-hand motion of the shaft, the case in the first instance and the wheel in the second being held fast and made to serve as leading apparatus. The reversal is effected by means of a coupling, which may eventually be moved by hydraulic means in such manner that an axial movement of this appliance connects either the solid shaft of the wheel or the hollow shaft of the case with the shaft to be driven. Connected with this reversing coupling is a holding-fast appliance, which fixes the shaft of the wheel, or that of the case, to the seat. During the process of reversing the coupling the shaft to be driven is prevented from turning by a strap brake, which may eventually be actuated by hydraulic power.

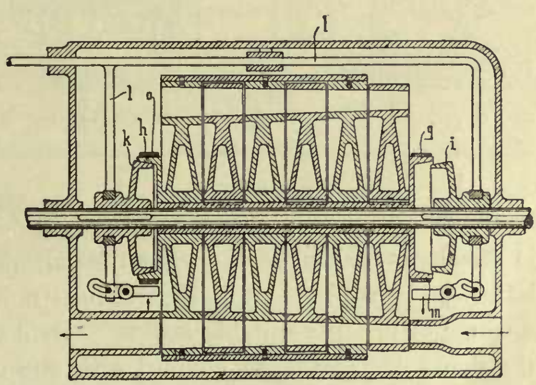


Fig. 381.

Schaeben[2] (1903) simplifies the working of this kind of reversible turbine by the following construction, which also assumes two sets of wheels rotating in opposite directions (Fig. 381). The halves *g* and *h* of two couplings are connected with disc brakes, or are themselves formed as such, and stand in further fixed connection each with one of the sets of wheels. Their disconnectable halves *i* and *k* can, by means

[1] A. P. 713,637. [2] D. R. P. 152,274.

of a pinion *l*, which may work simultaneously on the brakes *m* and *o*, be moved together along the turbine shaft to be driven in such a manner that on the one hand the couplings are thrown alternately into and out of gear and the two sets of wheels thereby alternately put in connection with the shaft to be driven, and on the other the brakes are alternately tightened and eased, and thus hold fast the set of wheels that is not for the time being coupled with the shaft.

The arrangement is less convenient when the wheels themselves undergo an axial shifting movement. This applies to the turbine of *Ashton*[1] (1900) (Fig. 382). This expansion turbine has already been referred to in connection with the multi-stage pressure turbines.[2] In order to produce a reversal of the direction of rotation

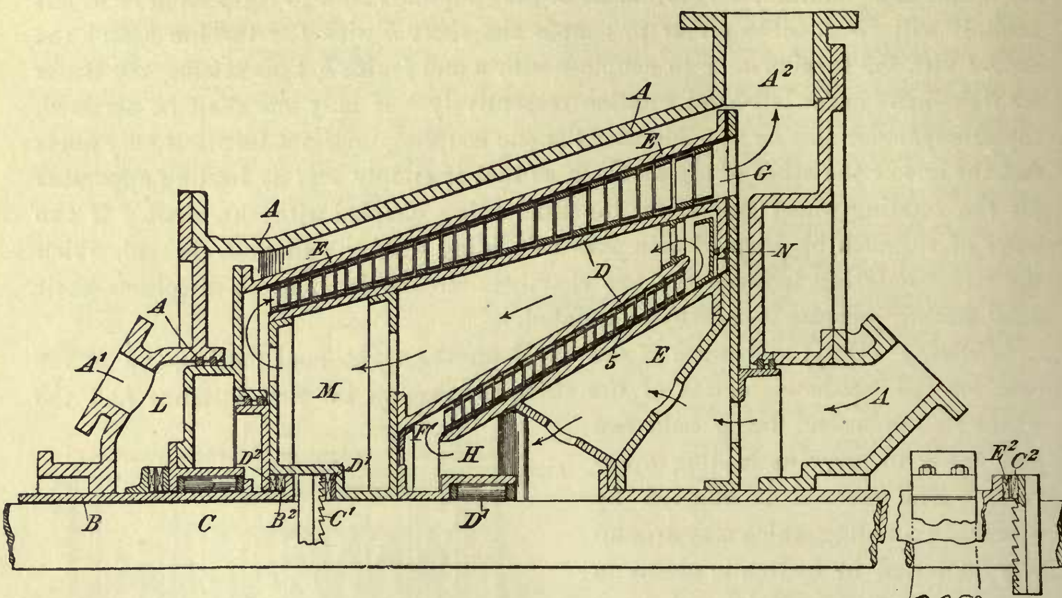


Fig. 382.

of the shafts without at the same time altering that of the vane-bearers D, E, these latter can be subjected to a longitudinal movement by the pressure of the working medium or in other suitable manner. By a suitable arrangement of coupling parts, the shafts B, C are each coupled with one or other of the vane-bearers. For the admission of the power medium, the turbine may, as before mentioned, have two inflow openings lying opposite one another corresponding with the two directions of rotation of the shafts, a two-way valve being arranged in the inflow chamber. For marine purposes in particular, the device is so arranged that the pressure of the power medium on the vane bearers D, E, which effects the movement of these, always acts in a direction opposed to that of the screw propellers. A manner of constructing this turbine is shown in Fig. 382. The motive medium in this case always enters through the inflow A, whether the shafts B and C rotate in the one or in the other direction. The inflow A¹ leads into a chamber L of the case, the

[1] D. R. P. 131,995.

[2] See pages 36 and 122.

boundary walls of which are in part formed by the rotating parts D and E. A ring-formed chamber M is provided between the two annular channels F and G, so that by this arrangement the motive medium does work when it passes through the channel F, but has no active effect when it is passing through the channel G. A direct passage of the motive medium from the end of the channel F to that of the channel G is prevented by the ring-like packing N. For motion ahead, the motive medium is introduced through the inlet A, and acts on the movable members D, E in such a manner that these are moved in the longitudinal direction till the inner of the two, D, becomes fixed to the shaft C by means of the couplings C^1 and D^1, and the outer one, E, to the shaft B. The motive medium then takes its way through the ring-shaped chambers H, the annular channel F, the chamber M, and the channel G, and escapes by the exhaust A^2. In the channels F and G it impinges on the vanes and sets the members D and E in right-hand and left-hand rotation respectively. For a reversal of the motion of the shafts B and C, fresh steam is admitted into the chambers L. The pressure thereby produced moves the front wall of the chamber L and the members D and E in such a manner that the inner member D becomes coupled to the shaft B, and the outer member E to the shaft C. The motive medium, which enters the turbine at A, pursues the same path as before. In case of motion ahead, the pressure in the chamber L becomes reduced, so that the pressure set up from the opposite side acts on the members D and E. The reduction of pressure itself can be produced by simply connecting the chamber L with the outer air or with the condenser. During the reversal of motion the members D and E are subject to a longitudinal pressure set up by the motive medium entering at A; in addition, however, the pressure set up in the chamber L also exercises its power. The latter is the greater of the two, and acts in the opposite direction. This is necessary in order that the couplings B^2, D^2, C^2, E^2 be kept in gear, and a sufficiently large excess of power is ensured by a suitable formation of the chamber L.

Webster,[1] also (Fig. 383), moves the wheels in order to effect reversal. Two working wheels *a* and *b* are set conaxially with one another. The inner wheel *a*, which may be arranged for right-hand motion, is fixed on the shaft *c*, and the guide-vane wreaths corresponding with its vane wreaths project radially inwards from the cylinder *b*. The latter at the same time forms the second working wheel, the vanes

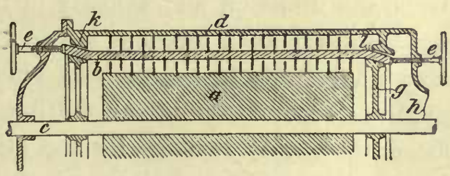


Fig. 383.

of which are set for left-hand motion and work together with those arranged on the case *d*. The cylinder *b* can, by the help of the screws *e*, be moved in an axial direction. According to the illustration, it can be pressed tightly against the friction wheels *g* keyed to the shaft *c*, in which case the steam entering at *h* streams through between the cylinder *b* and the case *d*, imparts left-hand motion to the former and, by means of the wheels *f* and *g*, also to the shaft *c*. The wheel *a* then runs free along with

[1] E. P. 8475 of the year 1903. D. R. P. 154,818.

these. If, on the other hand, the cylinder *b* be so moved that it is made to press against the conical surfaces *k*, *l* of the case, it will there be held fast, and only the way to the wheel *a* is left open to the steam, with the result that right-hand motion is set up. Similar arrangements may be made for turbines with gradually widening working chambers.

The turbine with counter-direction wheels, designed by *Martindale*,[1] consists of two discs with conaxially arranged vane wreaths, which are radially impinged on from without through separate nozzles (four in number) arranged at equal distances apart. The nozzle steam valves are successively opened by projections which revolve with the wheels, so that the steam issuing from the one wheel still has time to expand radially inwards before the next valve is opened. The two shafts of the

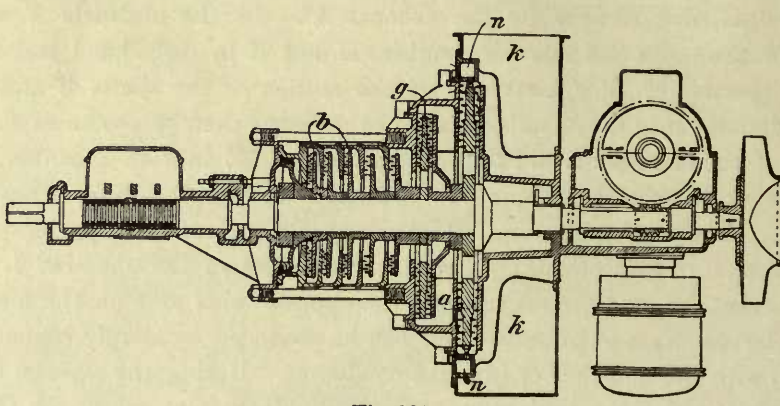


Fig. 384.

wheels now combine in driving a third shaft, by the help of connecting rods and an eccentric, which latter is turned through a certain angle when the shaft is to be driven in the contrary direction of rotation. The wheels continue to revolve in the same direction. In view of the means thus required for the transmission of the work to a third shaft, the solution of the problem does not appear to be a practical one.

If it be considered that, on the one hand, for a concussionless working of the steam, the direction and velocity of the current and the shape of the vanes must bear definitely fixed relations to one another, while, on the other hand, by reason of the high velocity of the working wheels, a reversal of these, whether by means of couplings or by longitudinal shifting of the wheels, cannot in practice be contemplated, there only remains, for the attainment of right-hand and left-hand motion, the expedient of the employment of different wheels or sets of wheels fixed to the same shaft. The circumstance must then be made the best of, that the set of wheels that is not being impinged upon has to revolve along with the set which is doing the work.

Parsons[2] (1895) (Figs. 384 to 386) has connected his multi-stage radial turbine having wheel bodies *b* provided with vanes on one side, and wheel bodies *g* and *a*

[1] E. P. 7979 of the year 1897. See also the *Parsons* arrangement of counter-direction wheels in Section IV.

[2] Ö. P. 13,496. A. P. 553,659.

with vanes on both sides, with a backwards turbine, the last of the wheels *a* being made to take the vanes *m*, which are set to suit the backwards motion arrangement. Around this vane wreath lies the annular channel *n*, with the nozzles for the backwards turbine, which accordingly is impinged on from without. The manner in which the channel *a* is built as part of the turbine case may be seen in the drawing. When the main turbine is at work, the vanes *m* move in exhaust steam, *i.e.* under condenser pressure, so that the ventilation resistance, which is to be looked for, is reduced to a minimum. When the engine is reversed, the steam escapes immediately after the influencing of the vanes *m* in the exhaust *k*. It is true this method of leading away the steam can be completely carried out only on the side of the wheel turned towards the exhaust, while on the other side, turned towards the main

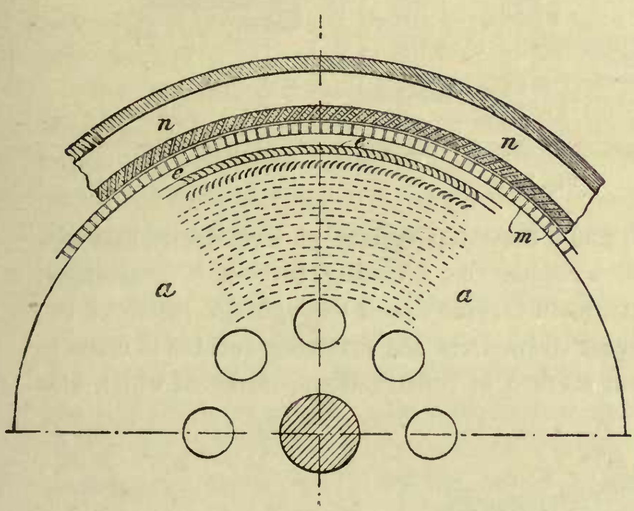


Fig. 385.

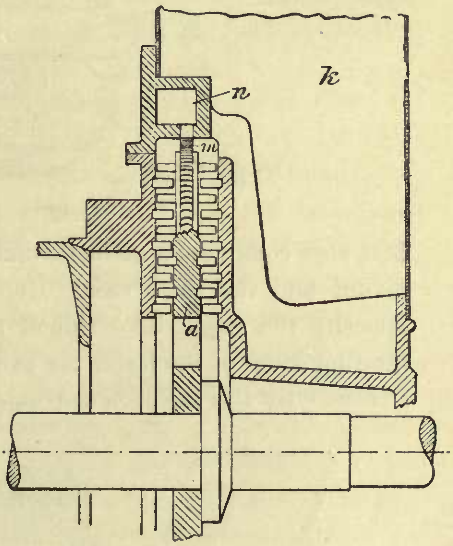


Fig. 386.

turbine, a congestion of the steam is to be expected, which, for a part of the floats of the wheel *a* that is carried round empty, occasions a resistance.

Another design made by *Parsons*[1] is shown in Fig. 387. In this turbine arrangement for the propulsion of vessels, as in that above described, a main and a reversing turbine are mounted on the same shaft and in the same case. Meanwhile, the reversing turbine *b* with its case 7 free to turn and its fixed drum *p* are inserted in telescope fashion into the main turbine *a* in order that the length of the engine may be kept as small as possible. The guide vanes of the main turbine *a* are, together with the case walls *c* and the movable vanes, connected with the drum *o*, which is borne by the shaft *f*. The steam enters the turbine through the channel *n*, and after passing through it is led away to the condenser by the channel *k*. The admission of the steam to the reversing turbine is effected by means of one or more pipes *r* on the inside of the cylinder *p*. The pipes can at the same time serve to

[1] D. R. P. 119,875. E. P. 14,476 of the year 1899.

connect the cylinder p with the case. The steam can also be led from the supply channel n of the main turbine through the shaft f to the reversing turbine. Again, the movable vanes of the reversing turbine may be attached to the inside of the drum o instead of to the special drum 7, so that the latter can be dispensed with.

The turbine arrangement of *Parsons*[1] for the propulsion of vessels, shown in Fig.

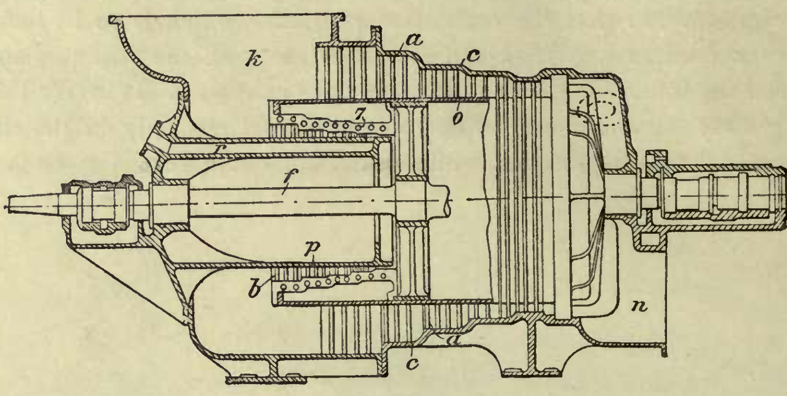


Fig. 387.

388, also comprises a main turbine and a reversing turbine on a common shaft and in one and the same case. Here, however, the reversing turbine b is arranged opposite the steam inlet side of the main turbine a, and through the inside of the rotating drum of the latter the exhaust steam from the reversing turbine is made to stream. By this means a convenient method of construction is attained which also

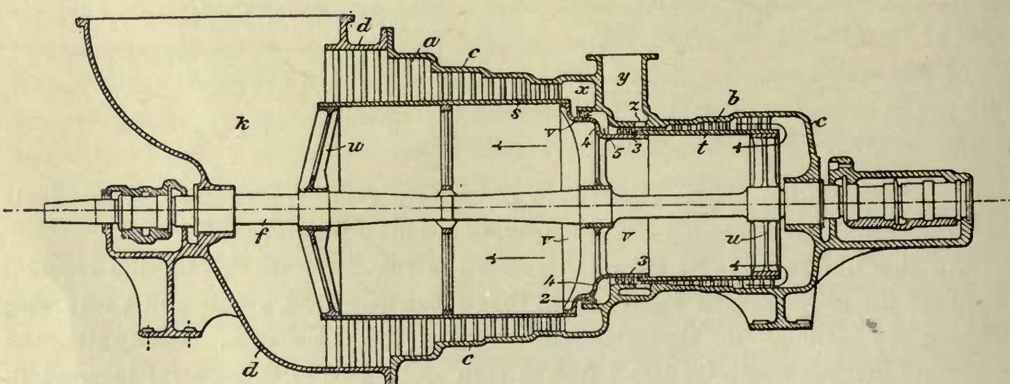


Fig. 388.

enables the reversing operation to be effected with a moderate consumption of steam. The steam inlet sides of the two turbines are, as seen in Fig. 388, arranged next to one another. The movable vanes of this turbine are attached to the drums s, which are connected by means of arms u with the common shaft f and by means of a ring v with each other, while the fixed vanes are arranged on the outer casing c d. The steam streams to the main turbine through a pipe (not visible) and a channel x,

[1] D. R. P. 124,091. E. P. 14,476 of the year 1899.

and to the reversing turbine through the pipe y and the channel z. The exhaust steam passes from the main turbine at once into the case d, which is connected by the channel k with the condenser. From the reversing turbine it streams in the direction of the arrows through the drums t and s to the case d. In order to effect steam-tightness between the inlet channel of the main turbine and that of the reversing turbine, fixed rings are arranged at 2 and 3 alternately against the ring v and at the case c in the form of circular grooves, which produce an effectual throttling of the passing steam. Openings 4 in the ring v serve to carry away the condensed water, which collects between the rings, through a chamber 5 to the condenser. In the design described the reversing turbine can also be arranged with its steam-discharge side in connection with the chamber 5 adjoining the steam-inlet side of the main turbine, in which case, then, the outflowing steam would also escape through the openings 4 and the drum of the main turbine to the condenser.

The reversing of the engine is probably of the greatest importance in the case of ships. This fact is taken into account by *Schulz*[1] (1898) (Figs. 389 to 391). His steam turbine, which for motion ahead is provided with a considerable number of concentric turbine wreaths, is characterized by the circumstance that for the attainment of as powerful a backward motion as possible, one or a small number only of turbine wreaths of the greatest possible diameters are fitted, which make a less efficient use of the steam. Thus in Fig. 389 two radial forwards turbines T are arranged, having working-wheel vanes b to c and the guide vanes contained in the case G, to which the steam is led by the pipes a. The backward-motion turbines T^1 are only single-wreathed, and receive their steam from the pipes e, when it is turned on by the cock X. In each case the exhaust is d. In the modification of the arrangement shown in Fig. 390 the vanes b of the forwards turbine T are on the one side of a disc, the other side of which bears the vanes C of the backwards turbine T^1. In the case illustrated by Fig. 391 the backwards turbine T^1 is connected with a forwards turbine T of conical form. Moreover, the expedient of turning the cock X in such a manner that the steam impinges partly on the forwards turbine and partly on the backwards one, and so governing the speed, has been thought of by the designer.

A turbine of this kind was built and tried by Schulz in 1898. It worked on a ship's propeller in a water tank with a surface of about 43 sq. ft. with forward and sternward motions. More exact data, however, were obtained from an experimental trial made with a turbine which was coupled to a dynamo. The style of construction of the turbine may be seen from Fig. 392. To relieve the axial pressure the current of steam entering at V for the forward motion was divided into two parts, which impinged, one after the other, on two multi-stage radial turbine bodies. For the backwards motion the steam could enter through the branch R. The results of the trial are put together in the following table:—

[1] D. R. P. 103,879.

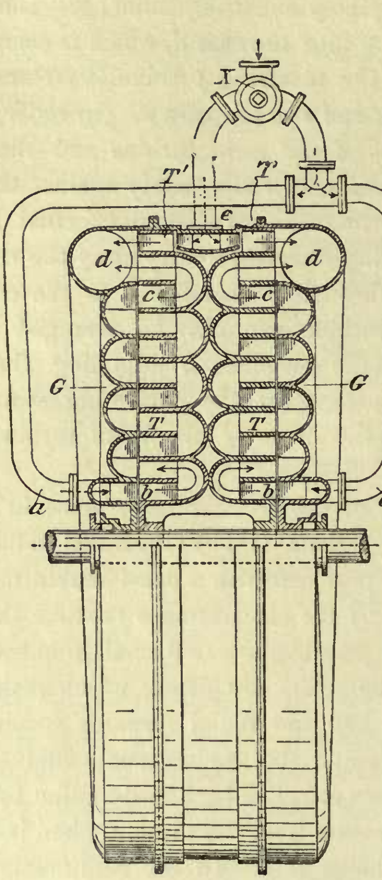


Fig. 389.

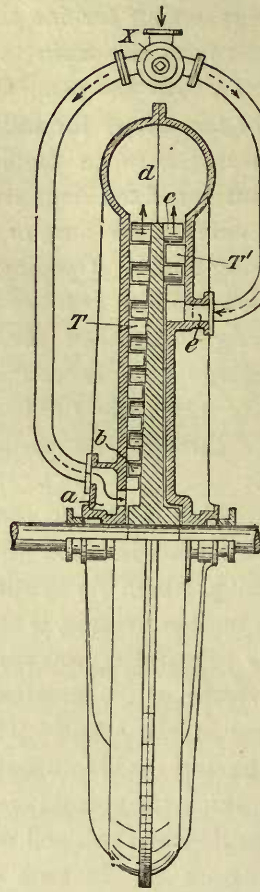


Fig. 390.

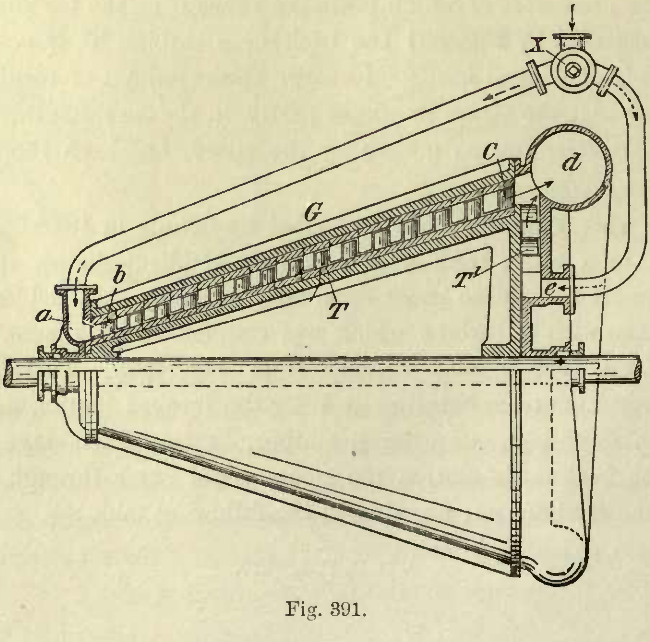


Fig. 391.

Schulz Turbine (1898) connected with a Dynamo.

Experimental trial.	1	2	3	4	5	
Boiler pressure	220	220	162	109	76	lbs. per sq. in.
Steam pressure in entrance chamber	176	176	154	103	74	above atmosphere.
Steam pressure behind the first turbine stage	132	131	115	71	51	"
Steam pressure in passage between first and second turbine bodies	29	29	28	17	10	"
Steam pressure in middle stage of second turbine body	$3\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{1}{4}$	0	0	"
Revolutions per minute	1700	1400	1200	980	800	
Dynamo.—Volts	150	160	100	80	75	
Ampères	250	170	200	110	70	
Power.—Kw.	37·5	27·2	20	8·8	5·25	
Electr. H.P.	51	37	27·2	11·97	7·14	

Curtis,[1] on the other hand, goes to work as follows (Fig. 393): The inlet *a*, which can be regulated by means of a slide (not shown), admits the steam to the first

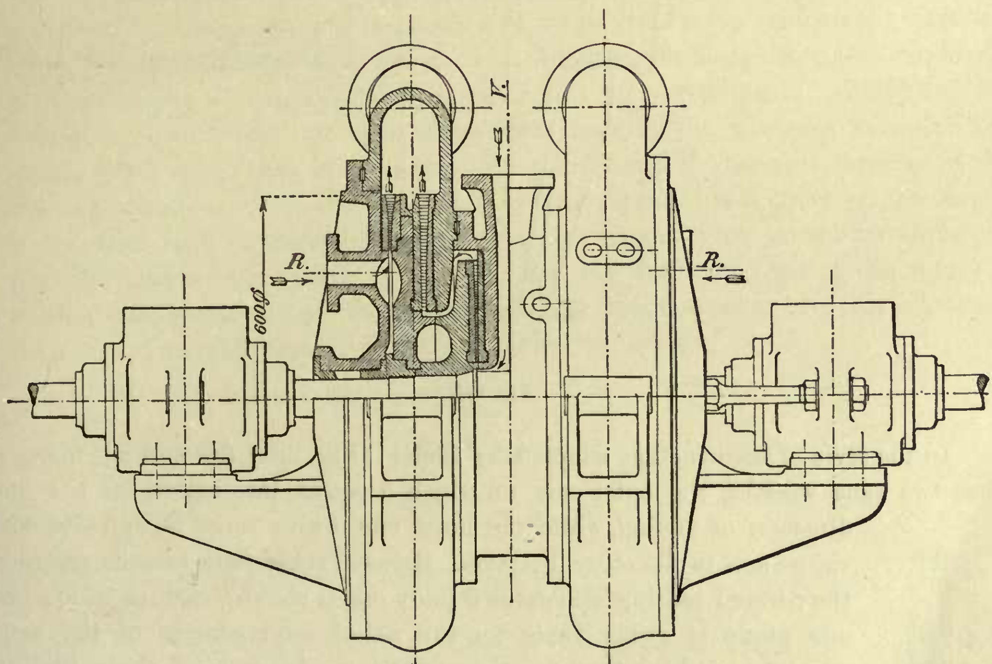


Fig. 392.

turbine *b*, which runs in the case *c*. From the latter the steam, directed by the slide *d*, makes its way to the second turbine *e*, which is contained in the case *f*. This second turbine bears, in addition to the main vane wreath *g* for forward motion, an outer vane wreath *h* for the backward motion of the turbine. The vanes of the two wreaths are of course set for opposite directions of rotation, and the nozzle apparatus at the point *k* below, which can receive its steam directly through the

[1] E. P. 756 of the year 1902.

pipe *l* and is intended to impinge on the back-motion vanes of the turbine, also has nozzles which point in directions opposite to those of the forward-motion ones. In the case illustrated, it will be seen, only the second turbine is applied to the backward motion.

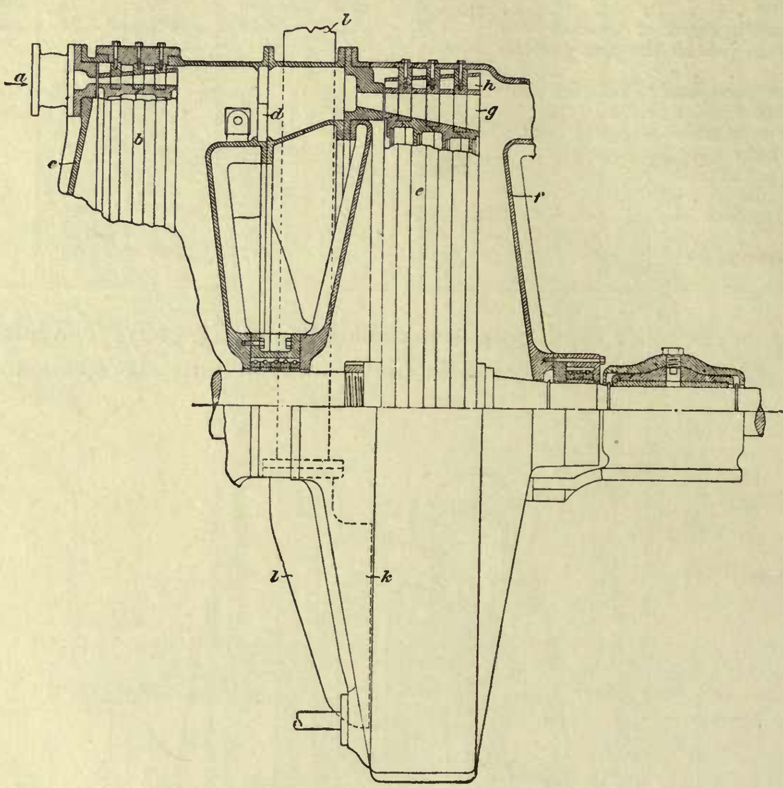


Fig. 393.

In the style of construction adopted by *Boella*[1] (Fig. 394) the working wheel *a* has two vane wreaths, the outer one of which *b* comes into action for the one direction of motion, while the inner one *c* with vanes of opposite curvature acts in the other direction. Between these vane wreaths protrude the covered leading apparatus *d* (only one is shown), each of which has one group of guide vanes for the radial impingement of the vane wreath *b* and another for the wreath *c*. In each of these leading apparatus is inserted a turning slide which can be actuated from the governing shaft *e*, and, according to its position, opens, to the steam streaming into the leading apparatus *d* in the direction *f*, first the one group of guide vanes and then the other.

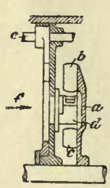


Fig. 394.

A reversible axially impinged velocity turbine with two concentric working wheel vane wreaths has also been designed by *Cassel*[2] (Fig. 395). The vanes *b* set with dovetail-formed feet in the wheel disc *a* have their inner parts *c* curved in

[1] E. P. 6420 of the year 1903. [2] E. P. 21,164 of the year 1901.

such a manner that they may be impinged upon from the nozzle *d*, say, for the left-hand direction of rotation, while the outer parts *b* are by the arrangement of the nozzle *f* made to produce right-hand motion.

Case[1] arranges two pairs of concentric vane wreaths at the side of a disc. The inner wreath on the one side of the disc forms, with the outer wreath on the other, a two-stage turbine, the stages of which stand in connection with one another by means of channels which cross each other diagonally.

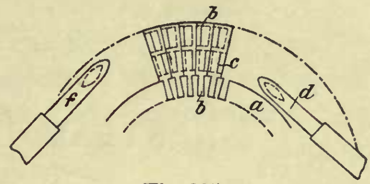


Fig. 395.

In general it will be possible to effect the reversal by the turning off and on of the organs for the admission and discharge of the steam. When, then, the cocks are turned from the positions for forward motion to those for backward motion, the steam in the backwards turbine will in the first place exercise a braking action like that of an elastic buffer, before the reversal of the direction of motion takes place. The duration of this transition period depends in the main upon the magnitudes of the rotating masses the kinetic energy of which has to be overcome. But the steam just cut off continues to act until its expansion is complete, and accordingly to some extent also delays the change of motion. *Windhausen*[2] (1903) had the idea of weakening the influence of the last factor in the following manner: The steam, which at the time of reversing or throwing out of gear still remains in the working chambers of the turbine that is to be reversed or laid off in all the stages, is led from each of these latter direct to the exhaust pipe or to the condenser. For this purpose the working chambers, from the first to the last of the stages of the multi-stage turbine, are connected with the exhaust pipe or with the condenser by means of suitable pipes fitted with checking appliances.

[1] E. P. 8986 of the year 1903.
[2] D. R. P. 151,380.

XIV

STEAM LEADING APPARATUS

On the design of the appliances by which the fresh steam is introduced into the working wheels great care must, of course, be bestowed. In the supply leads of the working medium losses of energy must to the utmost be guarded against. The fulfilment of this condition in the first place presupposes the avoidance of concussions wherever possible. The points which in this connection must have attention are in the main supplied by the theory of the water turbine. It is here also of importance that the steam be supplied without concussion, and that the direction of its current be that suitable for the working-wheel vanes that are to be moved.

Then follow the considerations which the expansibility of the steam bring to notice. Thus in pressure turbines it must be the endeavour completely to fill the cells of the working wheels, since the introduction of too small a quantity of high pressure steam into a working-wheel channel will produce sudden expansion within the latter and consequent loss of energy. This also happens when single channels only are impinged with steam under pressure, in which case the channels just passing in front of the impinging nozzles and those moving forward away from them are only partially filled. The loss will here be the greater, the greater the subdivision of the steam impingement appliances into groups that can be shut off.

The conditions in the velocity turbines that work with pressureless steam are in so far similar that in their case also the endeavour must be made to cause the whole effective surface of the working-wheel vanes, as these pass the supply appliances, to be played upon by the steam, or, in other words, to adapt the breadth of the leading channels to that of the working-wheel channels.

So far as experience hitherto obtained has shown, only nozzles on the *de Laval* principle admitting of a complete conversion of the pressure of the steam into velocity can come under practical consideration as steam-supply appliances for the velocity turbine. This conversion, however, must be effected with as little loss of energy as possible, which in case of unsuitable form and bad condition of the inner walls of the nozzles may assume dimensions such as to exclude the possibility of the economical working of an otherwise good turbine. When it is remembered that the length of an expansion nozzle is very small, and that with the enormous velocity of the steam the whole process of conversion takes place in an infinitesimal period of time, it will be apparent that the impingement nozzles must be the product of very exact and careful work.

In regard to the form of nozzle required for the proper expansion of the steam, attention may be directed to what was adduced in connection with the velocity turbines. It may further be observed that those nozzles only will come in question which widen out in the direction of flow of the steam; a cylindrical form, such as, for instance, *Ferranti*[1] has proposed, will, in spite of the insulating material (plumbago), recommended by the same designer, fail to fulfil its purpose. Just as incorrect seems an older design of *Curtis*[2] (1897) (Fig. 396). Curtis, it may be observed, has also proposed not to make the nozzle gradually widen in accordance with the expansion of the steam, but to let the mouthpiece *a* project into a short tube *b* with parallel walls and so give the tube the diameter which should obtain at

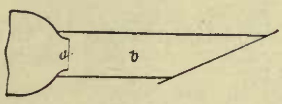


Fig. 396.

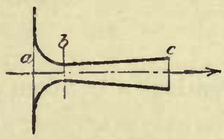


Fig. 397.

the discharge end of a conical nozzle. The idea in this is, that the steam will not be prevented from freely expanding, and that its friction against the walls of the nozzle will be avoided.

As basis for the calculation of the nozzle, the theory of *Zeuner*[3] may be adduced, from which the following may be taken. Let us assume (Fig. 397)—

p_1 the entrance pressure of the steam at the point a in front of the nozzle in lbs. per square inch,

v_1 the specific volume of the steam at the point a in front of the nozzle,

p_m the pressure of the steam at the narrowest point b of the nozzle,

w_m the velocity of the steam at the narrowest point b of the nozzle in feet per second,

F_m the clear sectional area at the narrowest point b of the nozzle in square feet,

p the pressure of the steam in front of the nozzle mouth c,

w the velocity of the steam in the nozzle mouth c,

F the free sectional area „ „ „ „

G the weight of the steam streaming through the nozzle in a second (the mixture of steam and water).

Further, let us assume that p_1 is greater than 1·7318 p, which for the pure velocity turbine should always be the case in order that a multiplicity of wheels may be avoided, especially in the high-pressure region, and that the steam is completely dried before the beginning of the expansion. In case of complete expansion the mouth pressure[4] in the Laval nozzle is proportional to the value p. Then we have—

[1] E. P. 2565 of the year 1895. [2] A. P. 591,822.
[3] Zeuner, Vorlesungen über Theorie der Turbinen, Leipzig, 1899.
[4] Zeuner, Techn. Thermodyn. I., p. 251.

$$p_m = 0{\cdot}5744 \times p_1$$

$$\frac{G}{F_m} = 10{\cdot}8\sqrt{\frac{p_1}{v_1}}$$

$$w_m = 281\sqrt{p_1 v_1}$$

Further—

$$\frac{w}{w_m} = 3{\cdot}3768\sqrt{1 - \left(\frac{p}{p_1}\right)^{\frac{a-1}{a}}}$$

and

$$\frac{F}{F_m} = \frac{0{\cdot}155}{\sqrt{\left(\frac{p}{p_1}\right)^{\frac{2}{a}} - \left(\frac{p}{p_1}\right)^{\frac{a+1}{a}}}}$$

in which equations a is taken as $= 1{\cdot}135$. The proportion borne by the diameters of the nozzles to one another at the points c and b is—

$$\frac{d}{d_m} = \sqrt{\frac{F}{F_m}}$$

and the specific volume of steam in the plane c of the nozzle is—

$$x = \left(\frac{p}{p_1}\right)^{0{\cdot}05826}$$

The current energy then amounts to—

$$H = \frac{w^2}{2g} \text{ in feet per second.}$$

A greater widening out of the nozzles than that provided for in the above formula on the assumption of complete adiabatic expansion and with due regard to the losses by friction, to which reference will afterwards be made, leads to a return conversion of velocity into heat, and thus to a loss of energy. On the other hand, a short prismatic increase of length of the nozzle does not produce an altered state of the steam, since, when the walls are in good condition, losses due to friction are of negligible amount.

Clearly, the conicity of the nozzle will be right when the steam expanding unhindered just fills it. Too small an increase of width leads to incomplete expansion, but too large an increase causes expansion to a pressure smaller than that of the surrounding medium, *i.e.* to compression of the outflowing jet of steam and to a parting of the latter from the wall of the nozzle. Recent experiments with jets of steam have shown the presence of knot formations in them, from which the conclusion has been considered justifiable that the velocity of flow of the steam does not exceed that of sound. These experiments appear, however, to have been made with unsuitable nozzles.[1]

[1] Compare—
Klein, *Zeitschr. d. Ver. d. Ing.*, 1895, p. 1193.
A. Fliegner, Versuche über das Ausströmen von Luft durch konisch divergente Röhren, Schweiz. Bauz., 1898, Vol. 31; Die grösste Ausströmungsgeschwindigkeit elastischer Flüssigkeiten, *loco citate*.

Stodola (Zürich) has also proved experimentally that the pressure of the steam at the smallest section of the nozzle sinks to the so-called "critical pressure," which corresponds with the sound-velocity appertaining to the initial pressure of the steam. A cylindrical continuation of the width of the narrowest point of the nozzle exercises no influence on the condition and velocity of the steam (apart from the friction). On the other hand, in a continuation that widens out in the direction of flow of the steam a greater fall in pressure will be converted into velocity energy without the volume of the steam being increased thereby.

In the formation of the nozzle it will be advisable first to ascertain the smallest sectional area suitable for a given quantity of steam, and on this to base the form of its longitudinal section. At the hinder end the section must be well rounded off, while towards the front it should be straight-lined and diverging in such a manner that the mouth has the width suited to the intended end pressure. A graphic method of ascertaining the section of the nozzle has been given by *Koob*,[1] who connects the heat diagram with the pressure diagram by means of the temperature-pressure curve of saturated steam. If from the latter the sectional area be determined for each point along the axis of the nozzle, it will appear that the straight-lined boundary of the longitudinal section is only an approximation to the theoretically correct one with which the expanding steam will just touch the walls of the nozzle. (Perhaps this circumstance is in part the cause of the knot formation in the steam jet issuing from a faulty nozzle.) Further, as has also been shown by *Stodola*,[2] the friction in the widening nozzle plays a part that must not be under-estimated. Thus in a case in which the initial pressure was 149·3 lbs. per square inch, the initial temperature 388°, and the absolute end pressure 34·7 lbs. per square inch, the loss of current energy due to friction amounted to 0·37, and when the end pressure was 2·84 lbs. per square inch, it amounted to from 0·2 to 0·25 of that attainable with the given fall in temperature. Delaporte reckons with a loss of energy of 0·05 in working with the exhaust and of 0·1 to 0·25 in working with the condenser, and sets the nozzle angle down at from 10° to 12°. It would appear, however, that by the use of carefully smoothed nozzles and superheated steam only about from 10 to 15 per cent. of the

1904; and Ueber den Clausiusschen Entropiesatz, Quarterly publication of the Naturforscher Gesellschaft, Zürich, 1903.

Delaporte, Étude générale du rendement des nouvelles turbines de Laval, *Revue de Mécanique*, 1902, p. 466.

Blaess, Ueber Ausströmversuche mit gesättigtem Wasserdampf, *Physik. Zeitschr.*, 1902, Vol. 4.

Stodola, Die Dampfturbinen und die Aussichten der Wärmekraftmaschinen, Berlin, 1903.

P. Emden, Die Ausströmungserscheinungen des Wasserdampfes, Munich, 1903.

E. Lewicki, *Ztschr. d. Ver. d. Ing.*, 1903, p. 491.

H. Lorenz, Die stationare Strömung von Gasen und Dämpfen durch Rohre mit veränderlichem Querschnitt; *Ztschr. d. Ver. d. Ing.*, 1903, p. 1600.

Prandtl, *l.c.*, 1904, p. 348.

Proell, *l.c.*, 1904, p. 350.

M. F. Gutermuth, *Ztschr. d. Ver. d. Ing.*, 1904, p. 75.

K. Büchner, Zur Frage der Lavalschen Turbinendüse, *l.c.*, 1904, p. 1029.

[1] Die Strömungserscheinungen in den Düsen von Dampfturbinen; *Ztschr. d. Ver. d. Ing.*, 1904, p. 275.

[2] *l.c.*

available heat is lost during its conversion into velocity. If, then, the nozzle mouth is, in spite of the retardation thus taking place, to admit of the passage of the same volume of steam as that for which the smallest section is proportioned, a somewhat greater diameter must be chosen for it than a calculation made without regard to friction would give. On the other hand, however, the overcoming of the influence of the centrifugal forces (compare Section XI.) must also be taken into account, which makes it necessary to leave a certain pressure in the steam issuing from the nozzle.

It is also worthy of note that the losses due to friction are more considerable with higher mouth-pressures than with lower ones. From this the conclusion may be drawn that when a turbine is divided into pressure stages in which in each stage a fall in pressure is converted into velocity energy, greater losses will result from friction in the higher stages than in the lower ones. In special view of these losses the aim will in general be to keep down the number of the pressure stages. With reference, however, to the volume of the steam itself, in case of variable counter-pressure on the mouth of the nozzle, it has been shown by *Blaess* that a reduction of the volume of discharge does not take place until this counter-pressure becomes approximately equal to the initial pressure of the steam. With an initial pressure of 128 lbs. per sq. in. the reduction did not begin till the counter-pressure amounted to 114 lbs. per sq. in.

If—

Gh be the *volume of steam* per hour, in lbs.,

f the sum of the nozzle areas at the narrowest points, in sq. ins.,

p the absolute steam pressure in front of the nozzles, in lbs. per sq. in.,

v the specific volume of the steam of the pressure p, in cubic ft. per lb.,

then, according to *E. Lewicki*,[1]

$$\text{For dry saturated steam } Gh = 303{\cdot}576\sqrt{\frac{p}{v}}$$

$$\text{for superheated steam } Gh = 321{\cdot}793\sqrt{\frac{p}{v}}$$

Meanwhile the proportion "absolute entrance pressure" : "counter-pressure" must

In the first case be $\geqq 1{\cdot}73$
in the second case $\geqq 1{\cdot}85$

In order to reduce the speed of rotation of the velocity turbine it is customary, as before shown, to divide the available fall in pressure so that only a part of the latter is converted, and the steam velocity which is in each case attained is only a corresponding fraction of the whole fall obtainable. In accordance with this, a number of nozzles corresponding with the division of the fall referred to work one behind another. All of these may suitably be so arranged as to convert equal

[1] *Zeitschr. d. Ver. d. Ing.*, Vol. 47, p. 525.

amounts of pressure fall into velocity. If it be taken into consideration, however, that the steam in its performance of work in the higher stages of the turbine is warmed by friction in the nozzle, against the vanes, on the body of the working wheel, &c., its temperature will, before its entrance into the nozzle of the lower stage, have been somewhat increased. As a rule, the chambers of the separate pressure stages are made of such a width that the absolute velocity with which the steam leaves the working wheel is almost equal to nothing. This destruction of the velocity, however, is also combined with a development of heat. The process, as in the case of every conversion from one form of energy to another, is attended by a certain amount of loss, in order to prevent which as much as possible the exhaust of the higher stage is passed from the working wheel, with the least possible reduction of its outflow velocity, directly into a small impingement chamber provided for the nozzles of the lower stage. In this case a concussion due to congestion in front of the nozzles of the lower stage will have to be expected, unless the nozzles be correspondingly increased in width.

Mention may here be made of a proposal made by *Curtis*[1] (1897), according to which the nozzles (as well as the other parts touched by the steam) were to be coated with some insulating material. Curtis starts from the idea that by means of the friction of the steam against the metallic walls heat is developed, which is lost when the walls are cooled from without. In order, then, to render at least a portion of the energy used up in the work of friction available again, the absorption of warmth of the walls that are protected against cooling influences is to be allowed to proceed until a condition of equilibrium is produced in which the metal ceases to take up further heat, and the heat produced by friction is imparted to the steam.

While, now, the conversion, in the nozzle, of the total available fall in pressure—in the present case, by expansion to atmospheric or to condenser pressure, according to the style of working—into velocity gives the greatest current energy, this perfect destruction of pressure is in practice attended by drawbacks which more than counterbalance the advantages. In streaming over the vanes and leading apparatus between these the pressureless steam undergoes changes of direction which cause centrifugal forces to be set up: these tend to alter the condition of the steam mechanically by compressing it. In the case of considerable and repeated alterations in the direction of the steam, the centrifugal force makes its influence apparent in a troublesome manner. The alterations of condition thus leading to losses of energy do not occur if the steam is allowed to keep a certain amount of pressure sufficient to overcome the centrifugal forces. For this reason also *Curtis*[2] allows the steam to expand only to a certain point, so as to leave a residue of pressure for the purpose above mentioned. Moreover, *Rateau*[3] has, from the results of experimental trials, come to the conclusion that it is necessary to leave a certain residue of pressure unconsumed.

Stumpf,[4] in whose wheel the steam undergoes a change of direction of 180°, on

[1] A. P. 566,969.
[2] See page 94.
[3] La Rev. techn. 1903, p. 448.
[4] E. P. 18,572 of the year 1902.

the assumption of an initial pressure of the steam of 147 lbs. per sq. in., and an absolute counter-pressure (in the condenser) of 1·47 lbs. per sq. in., with complete utilization of the fall in pressure, takes for granted that the steam issuing from the expansion nozzle will, after normal losses have been deducted, have a velocity of about 3280 ft. per second. If now the vanes be curved to a radius of 0·0328 ft., the centrifugal acceleration to be reckoned with on account of the centrifugal force amounts to $\frac{3280^2}{0 \cdot 0328} = 32{,}800{,}000$. These considerable forces tend to condense the returning current of steam, the particles of which at a distance from the vanes have the tendency to press in the direction of these latter. Stumpf has observed cases of compression up to 7 lbs. per sq. in., *i.e.* to five times the density of the in-flowing jet of steam, and has estimated the loss in connection therewith at about 20 per cent. In order to avoid this loss he allows the steam in the nozzle to expand, not to the assumed counter-pressure of 1·4 lbs. per sq. in., but to only 7 lbs., *i.e.* to the degree at which condensation occurs again in the working wheel. The still remaining tendency of the steam to expand further while streaming over the vanes acts in opposition to the centrifugal forces. As soon, however, as the action of the latter ceases, namely, at the outflow side of the cell of the working wheel, a final residual expansion of the steam takes place which does work by reaction.

The original *de Laval* nozzle has a circular section throughout, thus giving the mouthpiece, which is cut obliquely to the longitudinal axis, an elliptical form. A mouthpiece of this shape is, however, able fully to impinge only upon a small part of working-wheel channels standing directly in front of it; the other channels are only struck at their middle parts by the jet of steam, which can then spread in a direction at right angles to the direction of its main current, and cause eddies. In order to avoid drawbacks of this kind, the plan has been adopted of fitting the nozzle mouths to the parts of the working wheel to be impinged upon by them, and of giving them a square section. The narrowest section of the nozzle, which is circular, must then be made to merge gradually into the square form of the outflow side.

Amongst other nozzles, that of *Bök*[1] (1898) (Figs. 398 to 401) is formed in this way, and has also some further peculiarities. Within the nozzle *a* also an expansion of the motive medium to the pressure existing in the turbine case takes place. The end *d* of the supply channel directed towards the turbine is, however, curved in such a manner that the expanding motive medium is, by virtue of its kinetic energy, collected by the concave part of the bend, and formed into a jet which is directed against the vanes of the turbine *f*. The curved part *d* is, together with the part of the nozzle containing the expansion channel, so connected as to be capable of being turned (Fig. 398) or shifted (Fig. 399) so that a reversal is made possible. Further, the curved part forms a channel (either open or closed) with a sectional area of passage of unvarying size, which gradually changes from the circular form

[1] D. R. P. 104,805.

to that of a piece of a flat ring or to that of a rectangle (Fig. 400). The steam is intended to fill the channels to their full breadths at its first impingement of any of the vanes. The advantage of a nozzle that fills the cells in an even manner shows itself especially in the regulation of the admission pressure of the steam. The wall of the bend and of the directing channel lying opposite the one along which the motive medium streams and takes its shape can be removed; this part of the channel may therefore be either open or closed. In order that the motive medium may, even with the most considerable reduction of the admission pressure, be able to follow that part of the wall of the expansion channel *a* which merges into the concave part of the bend *b*, the channel may itself be slightly bent, as in Fig. 401. The motive medium then fills the part of the channel marked by crossed hatching in the figure. If the radial dimension of the last section (at *e*) be made large in proportion to the tangential one, its shape may also be made completely rectangular.

The curvature of the expansion nozzle in the *Bök* arrangement is, it is true, the very point which must be looked upon as a defect, and in view of the centrifugal

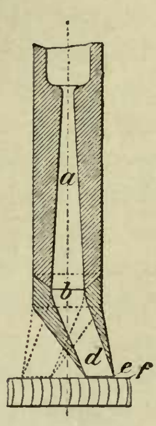


Fig. 398.

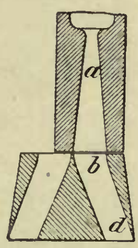


Fig. 399.

Fig. 400.

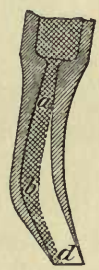


Fig. 401.

forces acting within the nozzle this increases further in importance. Also, it is not likely that an alteration of the pressure of the steam as it enters the nozzles will be made. For the action of the steam, no less than for the arrangement of the nozzles, it is right practice to make the axes of the latter straight, as is done by *Stumpf*[1] (1901) and by *Curtis*.[2]

Moreover, if the nozzle be so dimensioned that the expansion of the steam is exactly completed at its mouth, a want of uniformity will—even though the mouth take a rectangular form—be given the steam jet itself, caused by the circumstance that the plane of the mouth is inclined obliquely to the axis of the nozzle, by reason of which the steam expands at the short side of the latter less than at its long side. This want of uniformity can be got rid of by the fitting of the expansion chamber

[1] S. P. 25,333. [2] E. P. 756 of the year 1902.

proper of the nozzle with a mouthpiece of constant sectional area, in which the jet of steam is not subject to an alteration of velocity.[1]

It will further be of importance that the free expansion of the steam in the nozzle be not disturbed. In view of this, proposals such as that of *de Walden* and *Knudsen*[2] (1902), who proposed to fit spiral insertion pieces in the nozzles so as to give the motive medium a twisting motion, cannot be looked on as specially advantageous.

It is clear that particular care must be taken that the inner surfaces of the nozzles are smooth. In their construction, then, a material must be chosen which can readily be made smooth without the cost of the work being too much increased thereby, and which will neither be injuriously affected by high temperatures nor liable to rust. The metal chosen for this will probably always be steel. It would conduce to exactness of work in the formation of the steam-leading surfaces if the

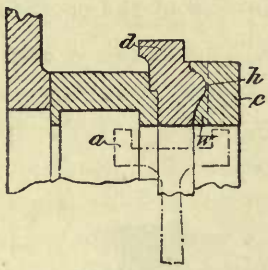


Fig. 402.

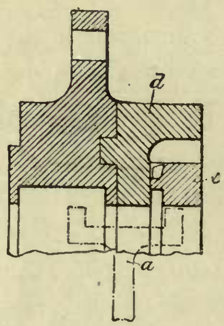


Fig. 403.

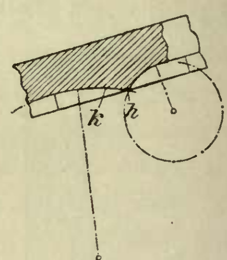


Fig. 404.

nozzles were made in two parts, and their inner surfaces were thereby made readily accessible.

Stumpf[3] (1902) (Figs. 402–404) thus constructs the channels, which, for the attainment of a certain degree of expansion in the leading apparatus, are made to widen out gradually from a narrowest point, the section of which is rectangular, in the following manner. Two rings *c*, *d* fitting together and having a section corresponding with the form of the channel, are made to form a leading apparatus, after channels *w* are cut in one of them *c* by means of a straight-lined slotting process. The narrowest point *h* of each channel *w* is formed by the bulging inward of the section of the ring *d*. Moreover, the channels *w* do not run out in the radial direction, but they are inclined at an acute angle to the vanes of the working wheel *a* in the direction of which the widenings of the channels take place. According to another method, the two rings *c* and *d* can be laid with plane surfaces one above another, and, after grooves corresponding in direction with the form of the channel have been made by a slotting process in one of the rings *c*, they can be put together to form a leading apparatus. The formation of the narrowest point *h* in the leading

[1] Compare Zeuner, Vorlesungen über Theorie der Turbinen, 1899. Delaporte in the *Revue de Mécanique*, 1902, p. 466.

[2] F. P. 327,708.

[3] D. R. P. 140,876.

channel is here effected by the guiding of the slotting tool in a certain curve *k*, instead of in a straight line. For axial turbines, *Terry* [1] (1899) (Fig. 405) arranges expansion nozzles which join together in ring form. He constructs separate ring pieces *a*, into each of which the halves *b*, *c* of two successive nozzles are cast. The ring pieces are then put together in such a manner that the nozzle halves are opposite each other, and these are screwed together to form complete nozzles. The nozzle ring thus formed is then turned in the lathe.

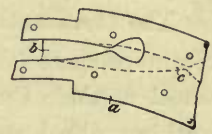


Fig. 405.

This arrangement of a succession of nozzles in such a manner that the steam jets issuing from them join together in the form of a ring is advantageous, because the complete impingement of all the vanes is thereby attained. This is the case, however, only in full working, since the shutting off of some of the nozzles for the purpose of governing the turbine necessarily causes an interruption of the ring of steam.

A nozzle arrangement of this kind, intended to produce a ring of steam, is to be seen in the turbine of *Stumpf* [2] (1901) (Figs. 406 and 407), in which the almost tangentially placed pockets *m* are closed to one another. The nozzles *e* are inserted singly in the leading ring *f*, and are held fast in it by the flanges *d* and the nuts *g*. Each nozzle has a narrowest section, through which the steam passes at full pressure before expanding in the adjacent widened portion. Then follows a guide piece, the section of which is rectangular.

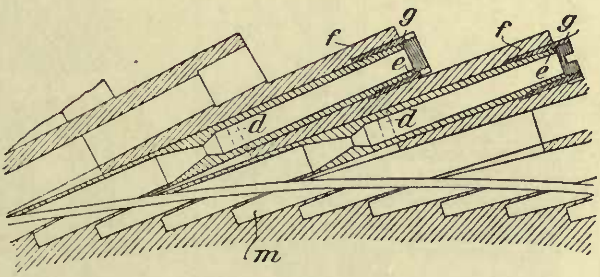


Fig. 406.

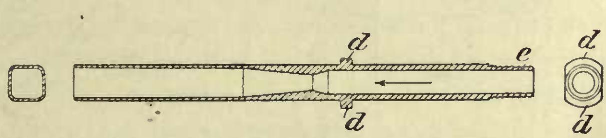


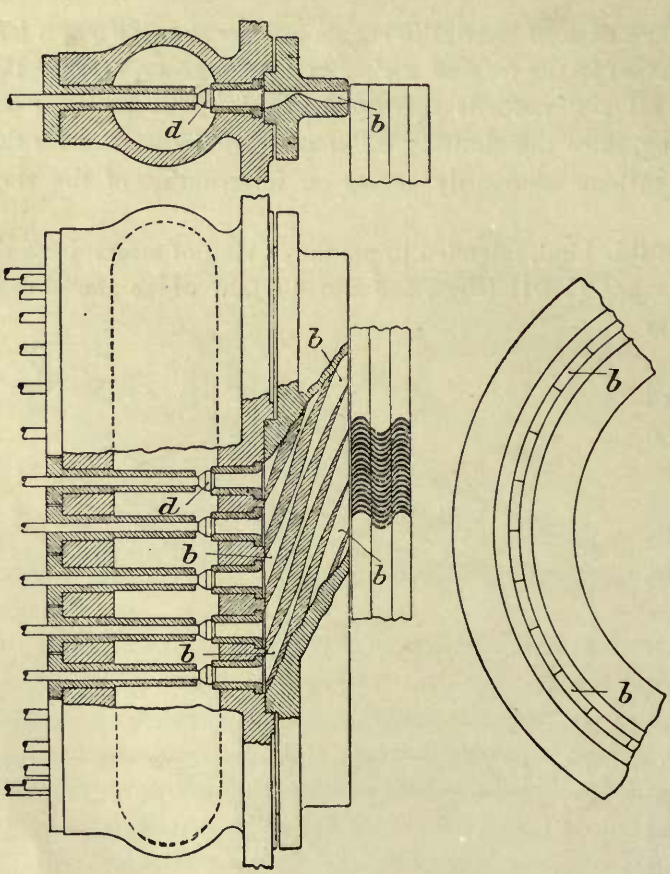
Fig. 407.

On the other hand, *Reuter* [3] (Figs. 408 to 410) has provided for a ring of leading channels *b*. The positions of these alongside each other and the run of their dividing partitions are such that, in case of full impingement, a complete ring of steam is formed. The channels are provided with inlet branches, each of which can be closed off by means of a valve *d*. The valves are closed one after another, as may be required. The leading channels are now carried in a screw-like course round a cylinder. At the first third of the length of each of these is a narrowing which divides it into a fore chamber and an expansion chamber.[4] In this arrangement it is not clear how a proper finishing of the steam passages can be made possible.

[1] A. P. 649,014. [2] S. P. 25,333. E. P. 25,411 of the year 1901.
[3] E. P. 4240 of the year 1902. S. P. 25,441. [4] Compare also E. P. 22,784 of the year 1903.

There is no fair justification for arrangements in which the expanding jets are divided, such as that of *Raworth*,[1] who inserts in the nozzle special partitions for this purpose (Fig. 411).

The *Continental Turbine Company* profess to be able to reckon on a very narrow clearance space (about 0·04 in.) between the impinging apparatus and the working wheel of their Curtis turbine. In this case, however, the varying breadths of the working wheel and of the case against which the impinging apparatus is placed require that it be possible to shift the latter while the engine is at work. The Company fulfil this condition by the method of construction[2] (1903) shown in Figs. 412 and 413. In this an appliance 2 is provided having five nozzles 9, which are separately influenced by means of the valves 7. It is fitted at the sides with eyes 4, by means of which it is hung on to the screw spindle 3, which pierces the lid of the case. The screws 1, which pass through the case with a certain amount of play, serve for fixing the leading apparatus. If these be eased and the counter-



Figs. 408–410.

Fig. 411.

nut 6 be loosened, the appliance 2 can, by means of the hand wheels 5, be moved in an axial direction towards or away from the working wheel 1. Since the spindles 3 pass through the eyes 4 with a certain amount of play, the appliance 2 can also be shifted in a radial direction by means of suitable insertion pieces *o*. Meanwhile this adjustment can be made only while the turbine is being put together. In the same manner it will be possible to shift the intermediate leading apparatus to a certain extent in the axial direction.

[1] E. P. 25,090 of the year 1893.

[2] F. P. 331,540.

Considering that, under the assumption made on pages 233 and 236, the Laval nozzle represents the only form of leading apparatus which is able to convert the pressure of a given volume of steam completely into velocity, every cylindrical or converging nozzle arranged for the same volume of steam must give an imperfect expansion. The steam preserves a certain amount of pressure, which it does not part with till it has left the nozzle. From the interesting experimental trials of *E. Lewicki*,[1] it may, for instance, be observed that this adjustment of temperature (in regard to the medium surrounding the nozzle) takes place within a very short distance. With a proportion of $\frac{\text{absolute entrance pressure}}{\text{counter-pressure}} = 8$, this distance is 0·4 in.; it increases by about 0·04 in. for every additional atmosphere (= 14·7 lbs.

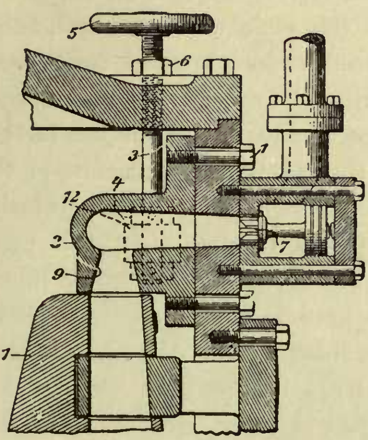


Fig. 412.

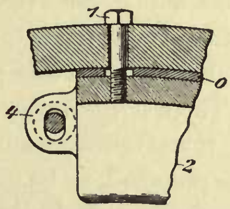


Fig. 413.

per sq. in.). Now, the steam leading channels, the narrowest sections of which are at the mouths, must act in the same manner as these nozzles. Of a given fall in pressure, which determines the pressure before and behind the leading channel, only a part can, when the volume of the steam is constant, be converted into velocity in the channel itself, while the rest of the pressure is carried over into the working wheel. In consequence the turbines that are arranged on the so-called action principle, but which do not make use of the widening nozzles for impingement, must count on the presence of a certain degree of pressure in the working wheel, and this will also have to be used up there.[2]

The leading apparatus, which are inserted between the various working wheels or between the stages of a multi-stage turbine—the *intermediate leading apparatus* —are intended to receive the steam from the one wheel with as little concussion as possible, and deliver it into the next-following wheel in the same manner. Consideration must here be paid to the circumstance that the working steam to a certain degree has the tendency to assume the rotary motion of the wheel that has just been

[1] *Zeitschrift d. Ver. d. Ing.*, Vol. 47, p. 528.
[2] In regard to the axial thrust compare Section XVII.

impinged upon. This tendency, however, is taken from it by the fixed guide vanes, and this the more thoroughly the smaller the parts into which the steam ring is divided by the guide vanes. For a large volume of steam will, by reason of its inertia, continue its movement against the fixed vanes, and thus itself be subjected to a considerable degree of compression, which must be directly followed by an expansion. The guide vanes, however, reverse the direction of the jets of steam issuing from the working-wheel cells in a more or less considerable degree, and then pass them on into the working-wheel channels of the next stage at the proper angles. This alteration of direction of the jets of steam at the same time eliminates the centrifugal forces which tended to compress the steam. These forces were the greater, the greater the change of direction of the steam jets. Thus in the modes of construction in which the steam jets are diverted through an angle of 180°, and brought back to impinge repeatedly upon the same working wheel, the forces are greater than, for instance, in the axial turbines which are each composed of several wheels. The action, however, of the centrifugal forces produced in the leading of the steam from stage to stage by the change of direction of the jets increases, under conditions that are otherwise similar, with the reduction in pressure of the working medium. These irregularities, then, will be more apparent in the velocity turbines driven by pressureless steam than in pressure turbines. In this case also the difficulty is proposed to be got over by a close spacing of the guide vanes, which, it may be observed, must be designed with a gradual curvature. The limits are here fixed by the losses due to concussion of the steam against the edges of the vanes, by practical requirements in regard to simplicity, and other such considerations.

Stumpf,[1] in whose working wheel the jets of steam are caused by the vanes to undergo a change of direction of almost 180°, has found that a compression of the above-mentioned kind, and of considerable amount, takes place within the vanes. Since he leads the steam issuing from the working-wheel pockets through circulating channels back into other pockets of the same working wheel, the compression of the steam is set up by the centrifugal forces in these circulating channels also. Such compressions, which are always immediately followed by expansions, are the cause of explosions, concussions, and losses of energy. In the Stumpf arrangement, the insertion of partitions in the circulating channels becomes even more necessary than in that of *Curtis*.[2] These separate the steam into thin layers and prevent the occurrence of any considerable eddying.

The *Weichelt*[3] (1901) style of construction, shown in Figs. 414 and 415, represents a turbine in which leading chambers h run in snake-like form round the vane wreath, and increase in size either in the direction of motion of the turbine or in that opposed to it. These chambers, which extend right round the wheel, are arranged in the form of annular bodies, and are concentric with the vane wreath b. In radial turbines they are within the inner and without the outer periphery; in axial turbines they are at the sides of the vane wreath. They are made portable and so fitted in the turbine case that their hollow spaces, which form the leading

[1] E. P. 3742 of the year 1903. [2] See p. 97. [3] D. R. P. 142,148.

chambers of the steam, in each case open towards the vane wreath. In order to facilitate an exact construction of the leading chambers, the annular bodies are each made up of several ring pieces. For instance, in the single wreath multi-stage radial turbine here shown *a* is the chamber which regulates the entrance of the motive medium with its discharge channel, *b*, *k* is the outflow chamber, and *c* the turbine wheel with its vanes *e* and their channels *d*, while *f* indicates the inner, and

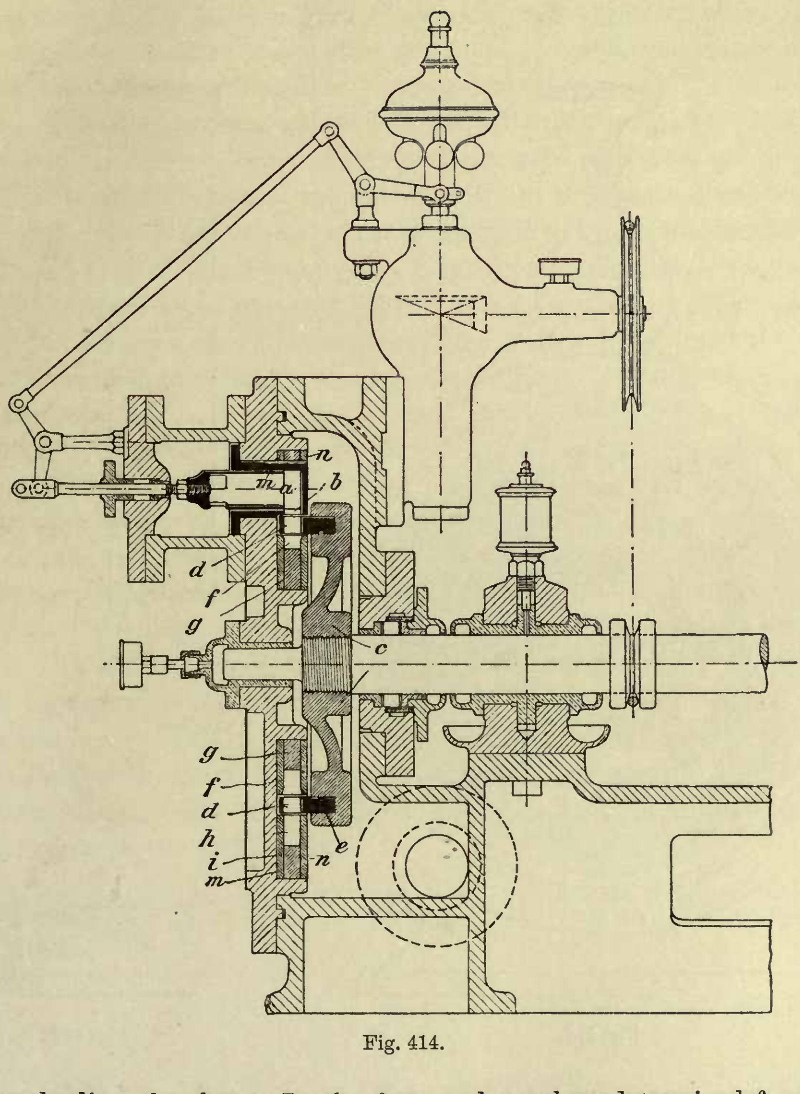


Fig. 414.

h the outer leading chambers. In the form and numbers determined for them by calculation, these are worked out of the side parts of the concentric rings *g* and *i*, which are turned towards the vane wreath. Circular covering plates *m*, *n* are then worked over the rings *g* and *i* at both sides, and where necessary connected with them. The resulting empty spaces, which form the leading chambers, are bounded at their curved leading surfaces by the rings *g* and *i*, and at the sides by the surfaces of the rings *m, n,* which cover the hollows formed in the concentric rings *g* and *i*. The shapes of the rings *g* and *i* may be seen from Figs. 414 and 415 in cross-section and in front view respectively. Other arrangements of the rings may also be chosen. To promote a better leading of the steam, the entrance edges of the rings are rounded off. So far as has been hitherto made known, a similar arrangement underlies the turbine of the *Gesellschaft für Elektrische Industrie* (Karlsruhe).

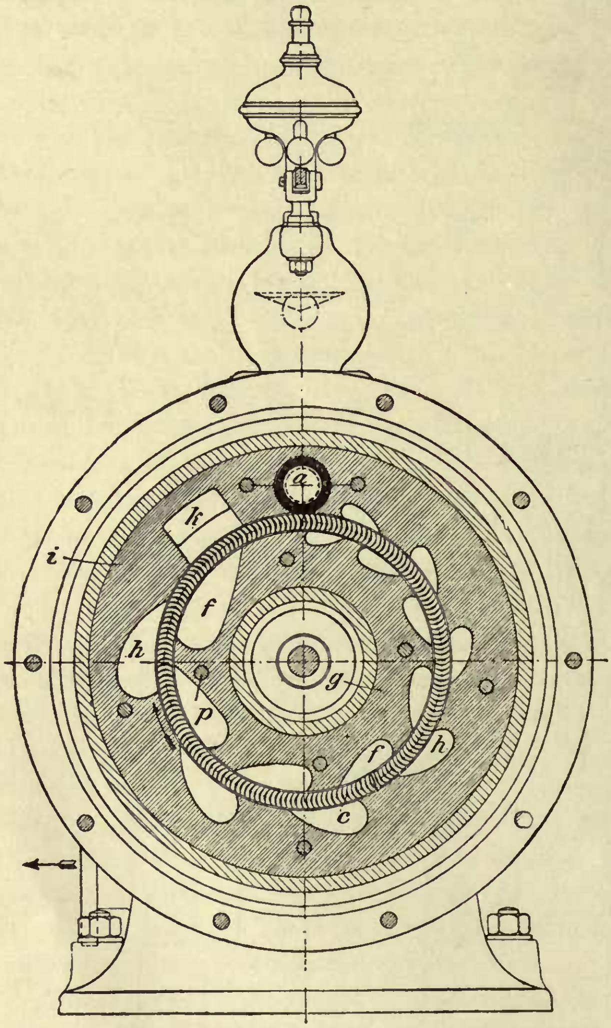
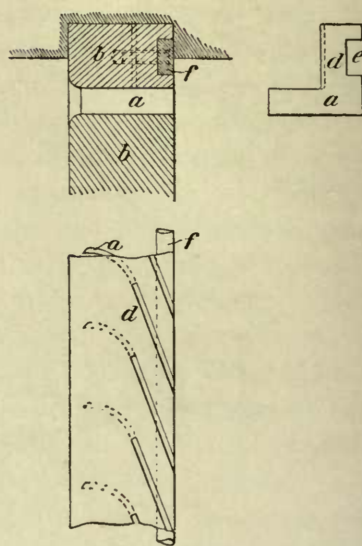


Fig. 415.



Figs. 416–418.

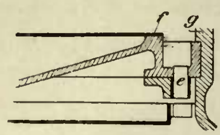


Fig. 419.

In the vane connection for guide wheels of axial turbines (*Aktienges. d. Maschinenfabriken von Escher, Wyss and Co.*[1]) (1903) shown in Figs. 416 to 418, indentations are also provided in the wheels and holding rings. The vanes *a*, however, are furnished with lug projections *d* fitting into the indentations of the guide wheels *b* and held fast by a closing ring *f*. This catches into a slot *e* of the guide

[1] D. R. P. 148,391.

wheels *b*, and into corresponding indentations in the lug projections *d*. This arrangement is said to enable the leading channels to be finished with great precision and to facilitate the renewal of the vanes.

In directing attention to the observations made in regard to leading apparatus during the consideration of the question of the multi-stage turbines, the arrangement of *Fullagar*,[1] which is based upon an axial turbine, may here be referred to. Segments in which the guide vanes are inserted with a feather and groove arrangement from the side are let into fixed rings. For this a suitable widening of each segment must be made at one particular point. After all the vanes have been slipped on, the section must be closed by an insertion piece.

In the case of multi-stage turbines with partial impingement of the first working-wheel wreaths the leading apparatus are not only widened from stage to stage, but their centre lines have a certain amount of shift one with another in the direction of rotation of the working wheels, which is designed to take account of the rotation communicated to the steam by the vanes. Amongst others, *Rateau* (*Société Sautter Harlé & Cie.*[2]) adopts a similar shift of the leading apparatus. His guide vanes *e*. (Fig. 419), which are made of plate material, are in this design fixed to ring pieces which are screwed to the disc *f*. The latter, again, is let into the wall *g* of the case. In the formation of the vanes, on the other hand, account must be taken of the reduction of the velocity of the steam from stage to stage by a suitable alteration of the angles of entrance and discharge.

In the same way as an impingement of one side of a working wheel is unsatisfactory, so also the fitting of the leading apparatus on one side only of a turbine cannot be a practical arrangement (*Othon*[3]). On the other hand, the arrangement of *Dodge*[4] (1903) (General Electric Co.) would seem to be the outcome of practical necessity. According to this, the leading apparatus of the Curtis turbine are put together in segments and let into the walls of the case, which are provided with portable lids in way of them, and are made reversible.

The construction and arrangement of the leading apparatus are for the most part so similar to those usually obtaining in working wheels that it will suffice in regard to them to refer to the next section.

[1] E. P. 14,593 of the year 1901.
[2] E. P. 11,701 of the year 1901.
[3] E. P, 2096 of the year 1901.
[4] A. P. 741,775.

XV

WORKING WHEELS

It is a self-evident matter that the steam spaces of the working wheels, whether in the forms of channels, of cells, or of pockets, must be wide enough to allow of the passage of the quantity of steam per unit of time representing the energy required. For the high velocities of the working medium here in question, these working spaces will, in proportion to the power developed, be very small, which circumstance suffices to explain the small dimensions of steam turbines. In reference to the foregoing, it must here be pointed out that it is necessary to make the steam passages of such width that the volume of steam necessary for the full development of power may be able, before its discharge from the last working wheel, to expand as far as the pressure prevailing in the condenser will allow.

On the formation of the *working-wheel vanes,* as the parts that take up the steam pressures, special care is rightly bestowed. The conditions attendant on the concussionless entrance of the steam into the working-wheel channels and on its further concussionless passage through them, together with the requirement of the best possible utilization of the available energy, for the most part determine the shapes of the steam-supply surfaces, *i.e.* of the working-wheel vanes themselves. This latter requirement, it may be observed, also influences the form and position of the working-wheel channels. In approaching the conditions of the problem more closely, use may be made of the principles well known in the case of water turbines, which must, it is true, be somewhat extended on account of the additional properties of the steam as a working medium. Smooth working surfaces are necessary in consideration of the fact that the steam constantly rubs along the surfaces of the vanes under pressure. Proposals for ribbed and such-like vanes (*Van Gelder*[1]) are of no practical value (compare also Section XI.).

In regard to the forms of the working-wheel vanes, which depend on the designs of the turbines themselves, attention may be drawn to the different systems of the latter already referred to. Attention may also be called to a few statements of *Parsons*[2] in regard to his pressure turbine, the (working-wheel) vane of which has assumed the shape shown in Fig. 420. The back of the vane is provided with a reinforcement a, the strongest point of which lies next to the steam inlet. The reinforcement is tapered off towards the steam outlet more than towards the inlet. By means of these vanes the channel

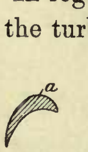


Fig. 420.

[1] E. P. 3565 of the year 1901.

[2] E. P. 8697 of the year 1896.

can be so constructed as to reduce the resistance of the flow of the steam, and consequently also of energy.

In another case, *Parsons* and *Swinton*[1] (1897) have been influenced by the following ideas. In the construction of the vanes it should be considered whether they are to be used only for one direction of rotation of the wheel or for both directions, and in the latter case with what relative degrees of efficiency.

Parsons and Swinton make the following statement:—In Figs. 421–424 *f* denotes the stationary and *r* the moving vanes, which latter, in conjunction with a part of the engine shaft, are for forward motion made to turn in the direction of the arrow 3. According to Fig. 421, the vanes are of a moderately curved shape. If the steam pass through the vanes *r* in the direction opposite to that of the arrow 1, *i.e.* in a forward direction shown by the arrow 2, the steam jets will follow the

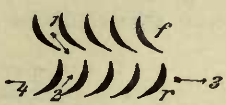


Fig. 421.

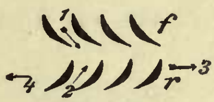


Fig. 422.

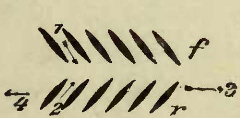


Fig. 423.

Fig. 424.

diverging flanks of the vanes and strike the concave surfaces of the counter vanes, thereby causing a backward motion and repulsion of the latter in the direction of the arrow 4. The power produced during the backward motion is considerably smaller—with regard to the form of the vanes in Fig. 421—than that developed during the forward motion; but may, for all that, for certain purposes be sufficient. If the vanes be less curved, as shown in Fig. 422, and the flanks be less convergent, the divergency for the reversed direction of the steam should be smaller. By this means a reversed motion, accompanied by a greater exertion of power is obtained. The flat or straight-sided form of the vanes to be seen in Fig. 423 gives the same conditions both for the forward and backward motions, and in this case the power exerted in the two directions of motion is the same. In order to avoid the loss of efficiency resulting from flatness of the vane, and to obtain a uniform exertion of power in both directions, the form of the former shown in Fig. 424 may be adopted. The vanes of this form are flat, but are provided with concave grooves, *g*, *h*, so that the steam strikes against these, as against continuations of the vanes, on passing through the vane channels in the one or the other direction. This method of working is shown by the three-headed arrows 1 and 2.

[1] D. R. P. 103,614. E. P. 901 of the year 1897.

The *Société Maison Bréguet*[1] (1903) have devoted their attention to the influence of the centrifugal forces on the steam. This company is of opinion that the injurious pressures within the guide channels can be diminished by the adoption of a special form for the vanes. According to experiments made, the loss of energy can be reduced to a minimum if the direction of the jet in the leading channel be changed at the place where the velocity of the steam is at its lowest, provided the channel be made straight as soon as the velocity begins to increase. In Fig. 425, the radius of curvature of the direction of flow of the steam from A to B increases to the same degree as that in which the channel grows narrower and the velocity of the steam in consequence increases; the piece B, C takes a rectilinear course. As seen in Fig. 426, this can also be attained with the vanes, the convex form of which shows a counter curvature. A certain injurious pressure on the concave surface of the passage AB will not be altogether avoidable; this excess of pressure will, however, disappear on the way from B to C. In order to facilitate this compensation, the

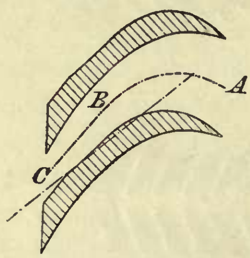


Fig. 425.

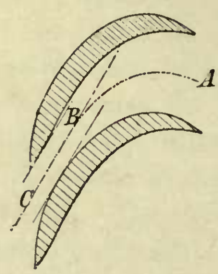


Fig. 426.

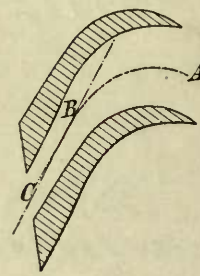


Fig. 427.

passage BC should be provided with a slight curvature, which must be opposite in direction to that of the passage AB, as shown in Fig. 427.

In 1894 the same company has already made the attempt to counteract the effect of the centrifugal forces on the steam during its passage through the working-wheel channels by a special arrangement of the wheels. It was their intention not to set the vanes of the axial Laval wheels radially, but to give them a slight inclination to their radii.[2] The anticipated effect, however, cannot be looked for, and the method has not been adopted.

Davidson[3] causes the steam jet to be diverted through an angle of 180° inside the cell. The working wheel, which is impinged upon radially from the inside, consists of a ring *a* (Figs. 428 and 429) with grooves *b*, *c* running round it. These annular grooves are divided into channels by means of the inserted strips of sheet-iron *d*, which are impinged upon one after another by the nozzles. The steam jet is directed towards the sharp edge between the grooves *b*, *c*. It sweeps over the latter transversely, and escapes in a direction parallel with but opposite to its inlet in such a manner that the axes of the passing jets and those of the steam nozzles are tangential to the same cylinder. The ring *a* is reinforced by being wound round

[1] F. P. 333,700. [2] F. P. 237,267. [3] E. P. 15,502 of the year 1896.

with wire *e* or by means of a ring shrunk on to it. It is riveted to the annular disc *f*, which is connected with the nave of the wheel by means of the discs *g*. In Fig. 428 an additional working wheel *h* has been provided for the opposite direction of rotation. Instead of two grooves *b*, *c* one single groove may be used, so that the steam may enter at the side of the disc *f*, and escape at its edge. According to Fig. 430, a revolving ring *a* having a single groove and a supporting disc *f* arranged at its centre have been adopted. In the arrangement shown in Fig. 431 the smooth

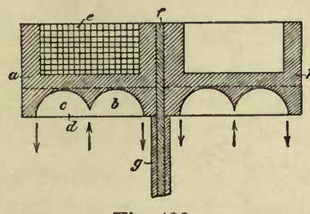


Fig. 428.

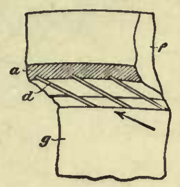


Fig. 429.

groove has been replaced by one, the bottom of which is stepped. By means of the smooth outside surfaces of these wheels, the ventilation resistances are diminished.

The working wheel *a* (Fig. 432), which is impinged upon from the outside, is constructed in a similar manner.[1] This wheel is also provided with sheet-iron strips, which divide the annular grooves into cells, over which the steam sweeps transversely. The drawing shows a double wheel intended for both directions of rotation. In regard to the design, Davidson states that the inside width of the diameters of the grooves should be from four to six times as large as the thickness of the steam jet at its point of entrance into the wheel, but never less than ½ inch

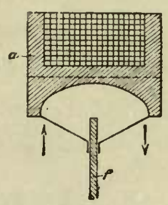


Fig. 430.

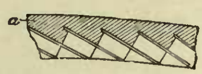


Fig. 431.

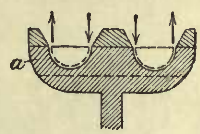


Fig. 432.

for the finest steam jet that is commonly used. The distance between the sheet-iron vanes which separate the cells is given at half the diameter of the inflowing jet of steam, its minimum amount being ⅛th of an inch. *Gelpke*[2] (1901) inserts vanes obliquely to the radii in the annular channels. The steam is supplied and taken up again by a guide ring surrounding the wheel. In order to ensure concussionless passage from the guide ring into the working wheel and *vice versâ*, the vanes of the working wheel are divided in the direction of the axis by the insertion of rings, and are curved down at the steam inlet to an angle differing from that at the outlet.

For designs of turbines in which steam channels are arranged in the coils about the axis, Zahikjanz[3] (1902) gives a method, by means of which core-pieces can be

[1] E. P. 15,503 of the year 1896. [2] S. P. 25,064. [3] D. R. P. 150,230.

dispensed with in casting, and by which thin partitions and smooth steam passages are obtained. The channels are cast into the wreath in the form of hollow bodies of sheet-iron or the like. The side-walls, constructed separately from the vanes, are fixed to the edges of these, and made to serve as stiffeners to them. The burning of the edges of the vanes during the process of casting is thus prevented. Figs. 433 and 434 represent a portion of a turbine wreath with closed U-shaped channels. At

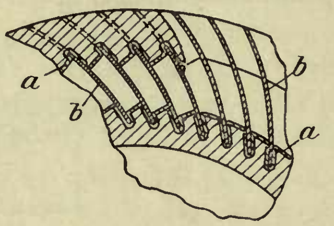


Fig. 433.

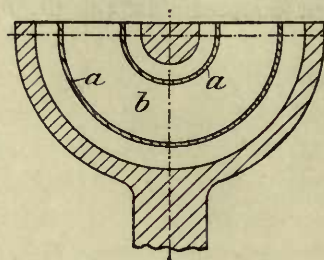


Fig. 434.

the left-hand side of Fig. 433 is a section through the middle of the working wheel, and to the right a section through the flange of the U-shaped working channels lying to the front. In the first section the side-walls *a* are vertical to the vanes *b*, while in the last-mentioned one they are conaxial with these. In both cases the side walls *a* are joggled over the vane-wheels in order to strengthen these, but may also be riveted or soldered to them. In order to protect the vane from being burned or melted during the pouring in of the liquid mass, its sides are re-enforced by small sheet-iron strips.[1]

In order to avoid the use of forms and to obtain smooth-walled coils, *Zahikjanz*[2] (1903) also proceeds in such a manner (Figs. 435 to 437) that the U-shaped vanes *a* of Z-shaped cross section are placed in a row on a centre ring *d*, and by means of lateral projections *f* are fixed in the slits *g* of two lateral rings *e*, which are then to

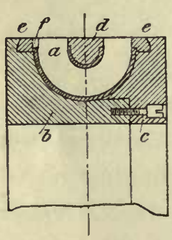


Fig. 435.

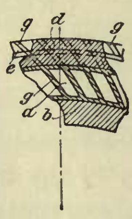


Fig. 436.

Fig. 437.

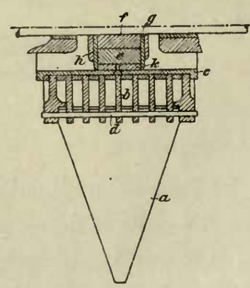


Fig. 438.

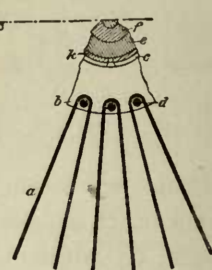


Fig. 439.

be secured together, or in slits in the casing and its cover. The vanes are mounted on a ring *b* of L-shaped cross section, and are pressed tightly into place by means of an annular cover *c*.

The intermediate spaces outside the channels are to be filled with molten metal or other mass which will set solid.

Veith (1897)[3] pursues a different course in the construction of the vanes (Figs. 438

[1] D. R. P. 150,725. [2] D. R. P. 151,497. [3] D. R. P. 97,346.

and 439). His intention is to reduce the number of revolutions of the working wheel by giving it a comparatively large diameter, reducing the weight at its periphery. He attains this by the adoption of vanes consisting of sheet-iron strips, which become gradually narrower towards the periphery. These strips are fixed radially to the body of the wheel, and form bodies of uniform or nearly uniform strength; they are of equal thickness, and their widths increase towards the shafts. The centrifugal force is intended to draw the radiating strips in the outward direction, so that thin sheet-iron can be used without the vanes being bent by the steam pressure during the rotation of the wheel. The sheet-iron strips are arranged on annular discs *b* by means of slits in the latter. The discs rest on the nave *c*, and the strips are fixed to them by means of the bolts *d*. The shaft must not be able to bend out between the bearings, which lie close to each other. In order, however, to enable the working wheel to adjust itself in accordance with its axis of gravity, the nave *c* with the box *k* is pressed tightly against an elastic cylinder *e* (made of indiarubber, or the like), which firmly embraces the box *f*, that is clamped to the shaft. The lateral discs *g*, which are likewise elastic, are pressed against the parts *k*, *e*, and *f* by means of the metal discs *h* and the screw bolts.

A more advantageous step was taken by Veith (1899)[1] in the design shown in Figs. 440 and 441. The vane body arranged in ray-like fashion is intended to be of nearly uniform strength. For this purpose the radial steel rods *a*, the outer ends of which take the forms of action turbine vanes, are each made in one piece roughly prismatic in shape, and of proportionately considerable length, in such a manner that the increase in area of the prismatic cross section is attained principally by the gradual reduction in size of the cavities of the vanes towards the inside of the turbine. Fig. 441 shows a vane of this description viewed in the direction of impingement, whilst Fig. 440 shows a side view, and also cross sections through A A, and through the point B in Fig. 441 respectively. On the omission of the rod 1, the vane can evidently also be used for lateral impingement (axial turbines). The vanes are fixed at their feet between discs.

The *Actiengesellschaft der Maschinenfabriken von Escher, Wyss and Co.* (Zürich, 1901), has designed these vanes in such a manner (Figs. 442 to 444)[2] that the feet K of the radiating rods *d* stand obliquely to the wheel shaft *a*, so as to ensure a better distribution of the material for these, and to facilitate their manufacture. In order to fix the rods, the feet of these are mounted in slot-holes in the disc rings *e*, *f*. The latter are fixed in grooves formed in the nave discs *b*, *c* in such a manner that the slot-holes *g* of the one disc ring shift with the holes *h* of the other.

In the case of the turbine designed by *Richards*[3] (1902) (Figs. 445 to 447), the vanes are fixed to shafts inserted in the disc of the wheel, and held there by pins. In design the vanes are similar to those of the Pelton wheel. For the impingement expansion nozzles are to be used.

Special attention has rightly been paid to the manufacture of the parts for the reception and conveyance of the steam energy. Since the number of the vanes,

[1] D. R. P. 112,724. [2] D. R. P. 135,938. [3] A. P. 740,780.

especially when multi-stage turbines with several sets of wheels are adopted, is very considerable, the attempt has, for obvious reasons, been made to reduce the cost of production of these without affecting the quality of the product.

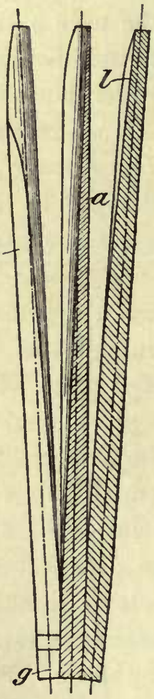


Fig. 440.

Fig. 441.

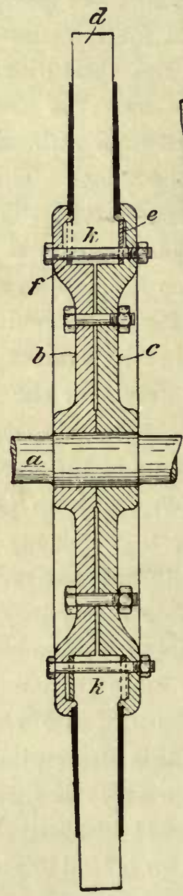


Fig. 442.

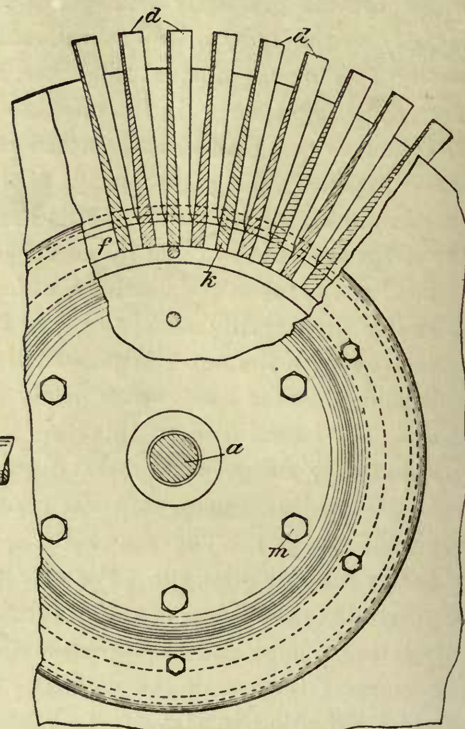


Fig. 443.

The simplest method seems to be to form the working spaces for the steam in the body of the working wheel itself in the manner already carried out in the

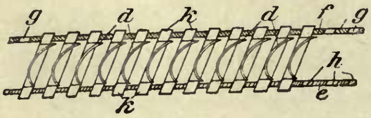
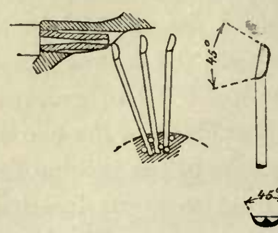


Fig. 444.



Figs. 445–447.

original reaction wheels, the arms of which were made use of for the conveyance of the steam. Thus *Parsons*[1] has made the attempt to improve turbines of this

[1] E. P. 8854 of the year 1893.

description (Fig. 448). He designed arms of sharp oval cross section, polished outside so as to reduce the external resistance to a minimum. The inner walls of the arms were also polished. For this purpose it was necessary to leave the arms open at both sides, so that the inner walls could be worked from the ends. The ends were then closed by means of the stops *a*, which could eventually be used for the purpose of equilibration.

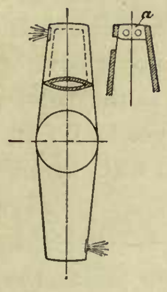


Fig. 448.

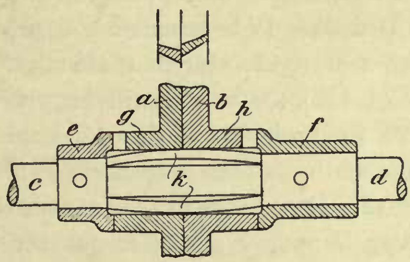
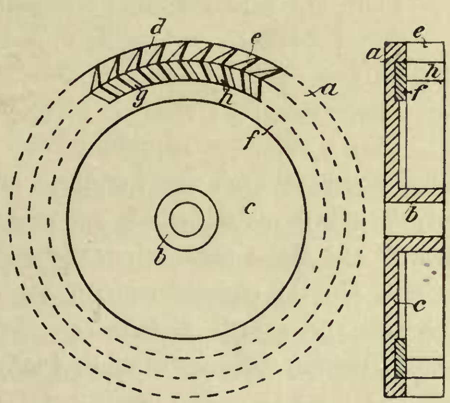


Fig. 449.

In order to be able to slot the vanes out of plain discs, and to replace the single parts of the wheels separately, the working wheel designed by *Nilsson*[1] (Fig. 449) is composed of turbines *a* and *b*, which are screwed together. These are connected, by means of springs, with the ends of the shaft *c*, *d* so as to provide a flexible bearing for the latter. The sockets *e*, *f* are bolted to the strengthened heads of the shaft, and engage the corresponding open ring flanges *g*, *h* of the discs *a*, *b*. The former are



Figs. 450, 451.

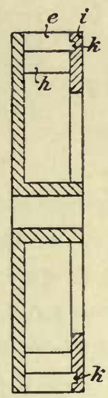


Fig. 452.

provided with longitudinal grooves, in which the greater part of the springs *k* are placed, the ends of which find abutment in the heads of the shaft.

The working wheel adopted by *Parker*,[2] which is designed in a similar manner, consists of two or three discs, into the circumference of which the single vanes are cut.

O. Hörenz[3] (1901) (Figs. 450 to 453) has built wheels with tangential pressure-

[1] E. P. 23,759 of the year 1899. [2] E. P. 16,157 of the year 1902. [3] D. R. P. 132,986.

medium supply in several rings, into which straight-sided portions of vanes are cut. After the rings have been put together, complete vanes of sickle or similar shape result.

Thus in a cylindrical projection *a* (Figs. 450, 451) of a wheel disc *c* provided with the nave *b*, openings *d* are cut in radial direction, so that in the one direction of rotation oblique teeth *e* are produced. Meanwhile in a ring *f* cut into the cylindrical projection *a*, openings *g* are cut in such a manner that oblique teeth *h* are produced trending in the opposite direction. The teeth *h* and *e* will then form the vanes. If a flange *i* be adopted and provided with a ring groove or channel *k*, and the ring *a* or the teeth *e* be made slightly to project so that they engage the channel *k* (Fig. 452), the teeth *e* will be prevented from flying off. In the case of the teeth *k*, a safety device of this kind may be dispensed with, because they rest outside on the teeth *e*, which keep them in place.

Lemoine[1] (1901) proceeds in the manner shown in Figs. 453 and 454. If, in the case of axially impinged steam or gas turbines with large radius, the wreath be constructed independently of the vanes, and worked round the latter, the advantage is obtained that the wreath can be made in one piece. This, however, is coupled with the disadvantage that a fixed connection between the wreath and the other parts of the turbine becomes difficult to make.

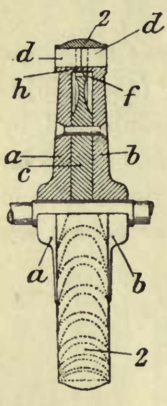


Fig. 453.

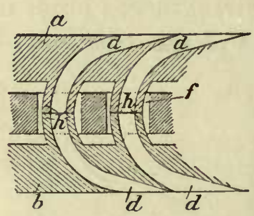


Fig. 454.

If, however, one portion of the wreath be allotted to each vane, so that the complete wreath is composed of as many parts as there are vanes, a close connection is produced between the vanes and the wreath. In this case the wreath is divided into very small parts, so that it does not always offer a sufficient support to the vanes. In order to obviate these defects, Lemoine constructs each wheel of three plates *a*, *b*, *c*, which are connected with each other, the centre plate carrying the wreath 2, below which it is provided with perforations *f*. In these perforations the shoulders *h* of the side plates *a* and *b* face each other. On the circumference of the side plates are half-chambers *d*, which, together with the wreath 2, form the working-wheel channels. Fig. 454 shows a large-scale section passing through the vanes to a conaxial cylinder.

Nadrowski and *v. Knorring*[2] (1901) (Figs. 455 and 456) assume that the working vane, which is clamped in position at one end only, is subjected in an unfavourable manner to a bending strain. The best expedient would be entirely to avoid the fixing of the vanes to the body of the wheel, and to make the two parts in one, so that the material should be subjected during the rotation of the turbine to tensile strain only. In order to ensure this, even in case of a close spacing of the vanes, a number of thin sheet-iron discs S, with perforations *o* at their circumferences, are

[1] D. R. P. 125,959. [2] D. R. P. 136,490.

fitted in layers, one above another, and connected together by means of the ring flanges *l* in such a manner that the perforations are shifted one with another in accordance with the desired vane form. In this case an iron ring *r* holds fast the ends of the discs S.

The construction shown in Fig. 456 (Section XX., below) serves the purpose of presenting a vane with as sharp an edge as possible to the ingoing jet.

Geisenhoner[1] seeks to facilitate manufacture, fixing, and renewal of the vanes by making them of single layers stamped out of thin sheet iron. The layers *a* (Fig. 457), representing the cross sections of the vanes, are laid one above another on the circumference *b* of the wheel *d* until they reach the height of the vane. The ring *c* is then fixed around the sheet-iron vanes, and is drawn tight against the wheel *b* by means of screw-bolts projecting through the sheet-iron plates. If radial widenings of the channels in the direction of flow of the steam be required, these are formed by suitable cuttings *e* in the ring *c*.

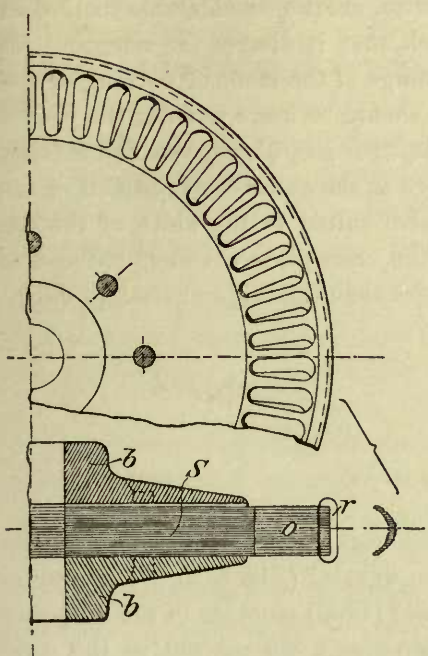


Fig. 455.

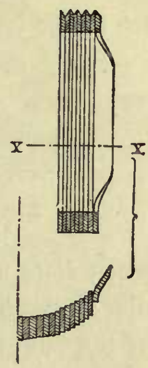


Fig. 456.

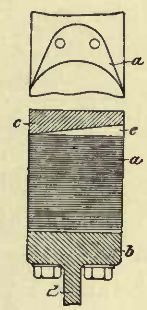


Fig. 457.

On the other hand, *Whitcher* and *Roberts*,[2] who had in view a tangentially impinged turbine, form annular grooves in the periphery of the wheel, and cutting longitudinal grooves across the latter, insert plate discs into these.

It would probably, however, be more advantageous to cut the pocket-shaped tangential cells of the working wheel into the solid wheel wreath, as has, indeed, been proposed by the above-named designers themselves.

For radial turbines, *Hedlund*[3] (1903) also makes use of curved vanes cut from plates of sheet iron, which engage the lateral discs of the wheel by means of pin projections. If necessary, they are passed through radial slits in the discs.

Stumpf[4] (1901) also cuts pocket-shaped channels, which, however, are of special design. Thus if the working-wheel cells take the forms of U-shaped pockets, which

[1] E. P. 18,083 of the year 1903.
[2] E. P. 2815 of the year 1900.
[3] A. P. 741,491.
[4] D. R. P. 131,816.

change the direction of the steam jet by 180°, the latter will, according to Stumpf, likewise pass on as a solid jet of U form (Fig. 458). In this case the central part *c* of the tongue which covers the pocket at the circumference of the wheel may be dispensed with. It thus becomes possible to work a disc rose bit so deeply into the material of the wheel that a complete semicircle *a* of the pocket *b* is covered by the tongue. In order to form the pocket, Stumpf uses a disc rose bit *e f* with two cutting stages (Figs. 459 and 460), the larger cutter *e* producing the pocket *b*, and the smaller cutter *f* a slit *g* in the metal tongue which covers the pocket.

The steam jet entering the pocket will, of course, exert a considerable suction on its surroundings; thus it has proved a drawback that it diverts the outgoing jet from the outer wall of the pocket to the disadvantage of the mode of working of the steam. In order to obviate this influence, the jet should be made to enter the pocket as freely as possible. For this reason the *Gesellschaft zur Einführung von Erfindungen m. b. H.*[1] (1903) makes the width of the cross-piece at the entrance very small or *nil*, and increases the same gradually to the dimension suited to the width of the jet. Moreover, the oblique position of the edge of the cross-piece thereby produced is said to offer less resistance to the ingoing steam jet than the original straight edge.

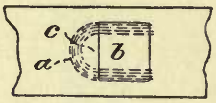


Fig. 458.

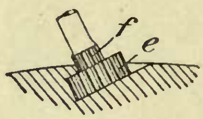


Fig. 459.

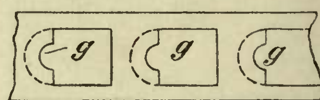


Fig. 460.

In his older experiments, *Rateau* has also used working wheels of this kind, the vanes of which were similar to those of the Pelton wheels[2] (Fig. 445). These vanes are likewise worked out of the solid metal. *Curtis*[3] (1902) proceeds in the following manner. According to Figs. 462 and 463, the vanes *a* are cut out of the solid metal rings *b*, which latter are screwed against sheet-iron discs *c* by means of flanges, the discs *c* being in turn reinforced by means of cross plates *d*. The vanes *e* for the backward motion are also worked out of the rings *f*. These latter close the cells formed by the vanes *a* at their outsides, whilst the cells, which correspond with the vanes *e*, are closed by means of a ring *g*. The connection of the ring *g* with the vanes *e* and *a* is made by means of the pin screws *h*. In the arrangement shown in Fig. 464 solid discs *k*, held apart from each other on the nave by distance pieces, are made use of, in the rims of which the vanes are cut. The latter are provided with projections, to which the rings *l* are riveted. In order to prevent the steam from entering the intermediate space between the discs, the latter are provided with grooves, in which metal cylinders *m* are inserted.

In order from a solid mass to produce steam channels with varying radius of curvature, as in the case of the Curtis turbines, by means of a boring tool, the latter must make a turn about its axis so as to preserve the angle of adjustment. In the

[1] D. R. P. 152,294.

[2] E. P. 18,745 of the year 1894. Rev. techn., Vol. 24, p. 410.

[3] E. P. 756 of the year 1902.

case of the engine designed by the *International Curtis Steam Turbine Company* (New York, 1902),[1] the tool makes a turn of this kind while performing its circular movement. In Fig. 465 a channel of the working wheel is shown; the tool H runs along the sides of the vanes s^2, s^1, s^2 and of the curve 3, while rotating around its axis in the direction of the arrows 2, its holder meanwhile making a rotation in the

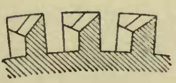


Fig. 461.

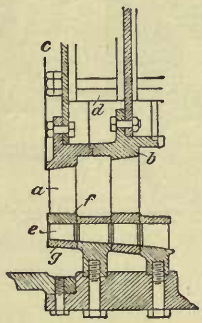


Fig. 462.

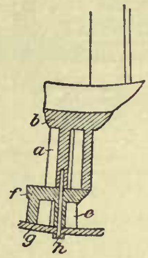


Fig. 463.

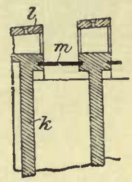


Fig. 464.

direction of the arrows 1. The engine itself is shown in Figs. 466, 467, and 468, and works in such a manner that the tool-holder I makes a circular motion, during the first (working) phase of which it is led in a path of varying radius of curvature and turned about its own axis. During the second (empty) phase the tool-holder, by a reverse turn around its own axis, is withdrawn from the piece of work in an axial direction and again led back to it. The axis of the tool-holder I lies eccentrically to that of its revolving driving pulley G. The circular path of the axis of the tool-holder resulting from the relations borne by the two axes to one another is changed into one of varying radius of curvature. The tool-holder is borne eccentrically to the moving driving pulley G in a slide piece I^2, which follows the rotation of the pulley G, and is displaced lengthwise on the latter to an extent depending on the varying radius of curvature. When its driving pulley G begins to turn, the slide piece I^2 runs along a first stationary curved disc J, the form of which determines the radius of curvature. A toothed rod L^1 is turned in the same manner as the slide piece I^2. This toothed rod is set free to move in the longitudinal direction, and while engaging the toothed tool-holder I is led along a second stationary curved disc L in such a manner that, during the first (working) phase of the rotation of the driving pulley G, it turns the tool-holder I about its own axis. During the second (empty) phase of the driving pulley G it reverses the rotation of the tool-holder about its axis, so that the latter returns to

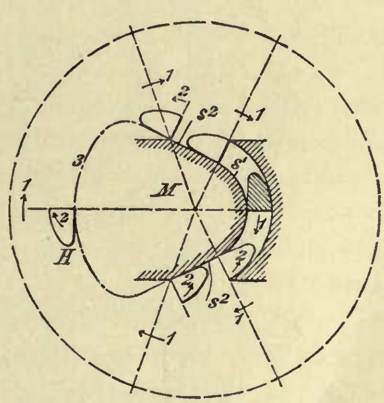


Fig. 465.

[1] D. R. P. 148,633.

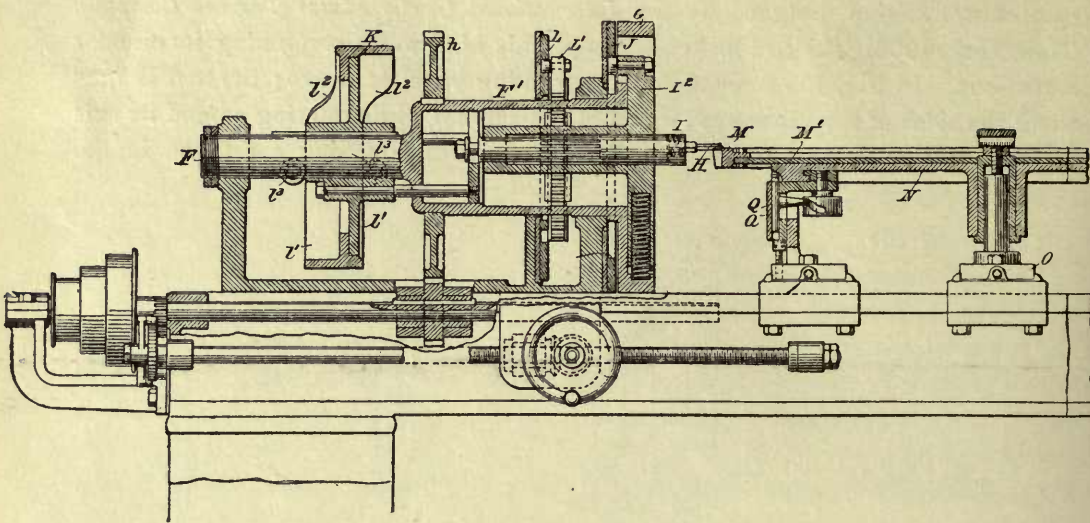


Fig. 466.

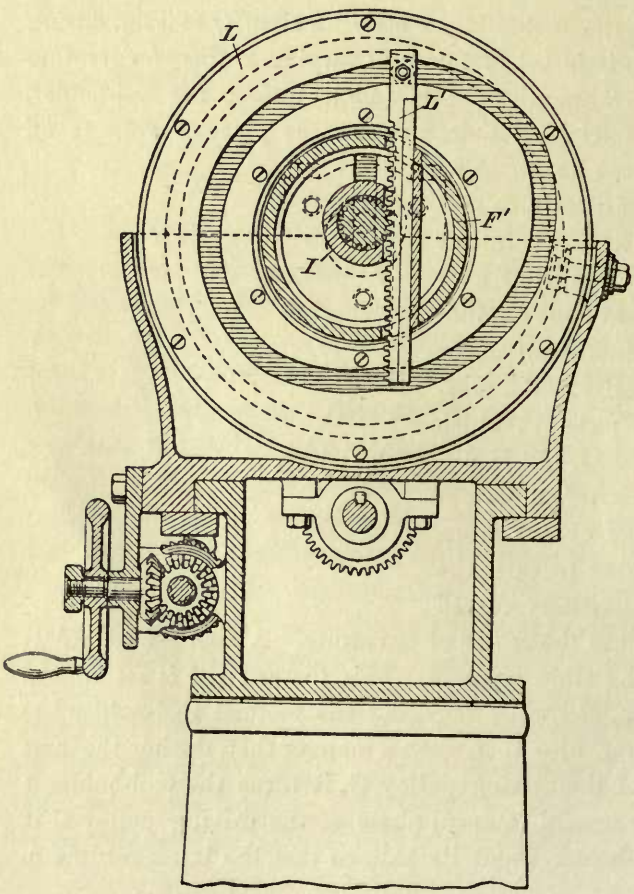


Fig. 467.

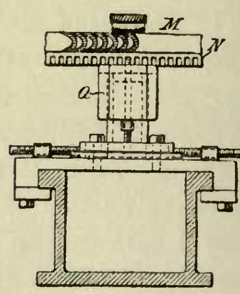


Fig. 468.

its original position. Contemporaneously with the slide piece I^2, a third curved disc K is caused to turn, along which the axially displaceable tool-bearer I is so led that it is withdrawn by the tool during the second (empty) phase of the rotation of the driving disc G, and again brought in contact with it. The driving shaft F is fitted with a cylindrical part F^1, the hollow space of which not only permits of a rotation of the tool-bearer I, but also renders possible its displacement in the axial direction, and together with the slide piece I^2 in the radial direction. The mantle of the cylinder is surrounded by the curved discs L and J, and serves as a guide for the toothed rod L^1.

The front surface of the hollow cylinder F serves to guide the rods connecting the tool-holders I with the curved disc K.

According to the position of the parts shown in Fig. 466, the chisel H begins its cutting course at the moment when the toothed wheel *h* begins to turn the shaft F, the hollow shaft F^1, and the head-plate G. During the rotation the toothed rod L^1 carries the tool I with it, which latter by the help of the slide piece I^2 and the curved disc J makes a motion of varying radius of curvature.

At the same time the toothed rod L^1, guided by the curved disc L, is displaced in a longitudinal direction, thereby bringing about a reversal of the tool-bearer. On the completion of the cut the toothed rod is displaced in the opposite direction, so that the tool-bearer is brought into its original position. By this, however, the surfaces l^2 of the curved disc K are brought between the stationary rollers l^3, so that the tool-bearer is withdrawn and completes its rotation outside the piece of work.

The vane ring M is fixed on an index plate N by means of a disc M^1, the said plate being free to turn on a support O in the longitudinal and transverse directions. The machine board is provided with a fixing appliance O, which is displaceable in the longitudinal and transverse directions.

In case the slits are cut into a piece of work of small radius, or if in a ring of large radius they are to be made of considerable depth, the walls formed between any two slits will be thicker at their inner than at their outer ends (Fig. 469). In order to obtain walls of constant thickness a second cut will have to be made (Fig. 470), in which case the spindle of the index plate undergoes a corresponding displacement in a transverse direction.

For the cutting of the channels of axial wheels, the *Continental Turbine Company* (1903) make use of a steel tool, the blade of which can move freely between the front and the back surfaces for the purpose of cutting, *i.e.* it projects above the tool shaft in both directions. The shaft in cross section is correspondingly bent on one side of the curvation of the working surface of the vane, while the part facing the back of the neighbouring vane is flattened.[1]

The vanes can also, however, be made separately, in which case a means of securely fixing them to the body of the working wheel must be found. Referring to the above-mentioned tangential wheel, attention may be drawn to the construction

[1] E. P. 331,541.

of *Williams*[1] (1896), who employs caps made of sheet-iron, in part inserted one within the other, and arranges them round about the wreath of the working wheel.

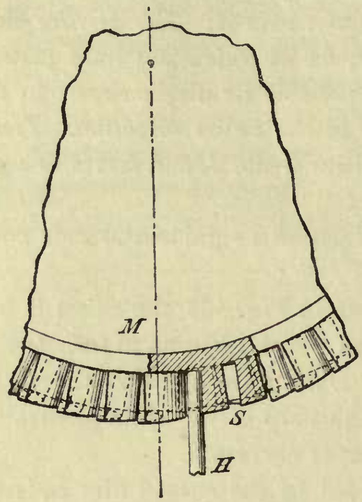


Fig. 469.

Fig. 470.

According to *Imle*[2] (1901) (Fig. 471), the hollow vanes *b* are connected together and made of sheet iron or metal strips *d*. They are arranged on a wheel body *a* provided with teeth, for the purpose of supporting them. *Geisenhoner*[3] (1898) also makes single vanes of sheet-iron in such a manner that their fronts and backs are each formed of a single strip. These front and back strips are riveted to each other at their leading and following edges or otherwise.

By this means a light hollow body is formed, which is made sufficiently stiff by the insertion of two stays. However, the lightness resulting from this method is obtained at the expense of the working cost and of accuracy.

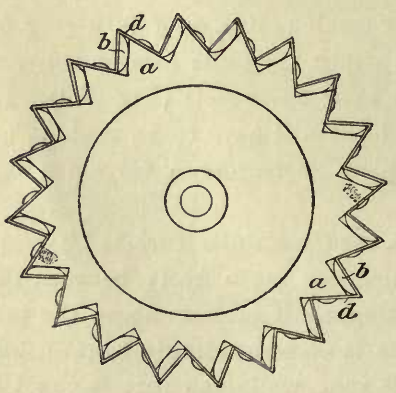


Fig. 471.

The use of metal rods, which have been worked to the cross section of the vane by drawing, rolling, cutting, or the like, and which are cut lengthwise, has proved to be of practical value. This method has been made use of by *Fullagar*,[4] whose older process[5] was more expensive. It consists in first fashioning the front surfaces of the vanes by rolling, or such-like means, and in then carefully working their back surfaces by means of rose bits made to pattern. *Parsons* also cuts the vanes from drawn sectional rods. The process employed by *Weichelt*[6] (Fig. 472), which consist in the slotting out

[1] A. P. 580,100.
[2] D. R. P. 137,126.
[3] A. P. 665,600.
[4] E. P. 14,594 of the year 1901.
[5] E. P. 14,593 of the year 1901.
[6] D. R. P. 136,796.

of the vanes from a previously opened ring a, having a width equal to the height of the former, first by means of a cylindrical rose bit d, and then by a hollow crown rose bit, the object being the attainment of vanes of uniform shape and smooth surface. An annular groove b may be cut into the annular working piece before it is slotted. These grooves serve to fix the individual finished vanes to the turbine or to the guide wheel. By this method it is rendered certain that the grooves b will correspond with one another when the vanes are put together.

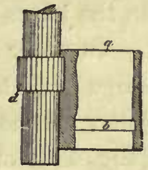


Fig. 472.

In the manufacture of vanes, the *Actiengesellschaft der Maschinenfabriken von Escher, Wyss and Co.*[1] (1902) proceed in the following manner:—Strips cut in forms (Fig. 473) are in the first place rolled conical, that is so as to correspond with the shape of the vanes (Fig. 474), whereupon they are inserted in a plate (Fig. 475), and trimmed, and finally cut to shape (Fig. 476). After the attachment pieces have been formed in them (Fig. 477), and the vanes have received their curvature, the leading edges, etc., are finally trimmed, and the strips worked to a uniform weight (Fig. 478). The latter may be rolled out either in a cold state or in a warm one, and in the first case they must from time to time be annealed. The remaining processes of manufacture are gone through when the material is in a cold state.

The *Parson's Foreign Patents Company, Limited, und A.-G. für Dampfturbinen, System Brown Boveri-Parsons*[2] (1902) (Figs. 479–482), have adopted vanes with

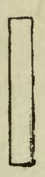


Fig. 473.

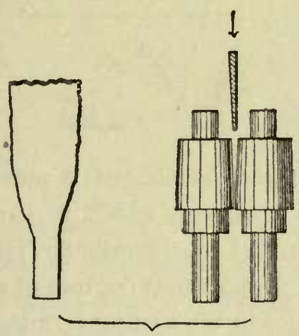


Fig. 474.

Fig. 475.

Fig. 476.

Fig. 477.

Fig. 478.

head plates that are staved up or flanged, and which fit together in the wheel in the form of a ring. The head plates here overlap, and are so formed that a smooth continuous ring is produced which closes the ends of the vanes. The head ends of the vanes can overlap each other in such a manner that a complete wreath is formed either at the entrance side or at the out-flow side of the vane ring. A third style of construction is shown in the figures. On this system the heads c formed on both sides of the vanes a are provided with projections b and d. The head is likewise bent down so that an overlapping can take place, in which the piece d of the one head fits over the piece b of the next one. This

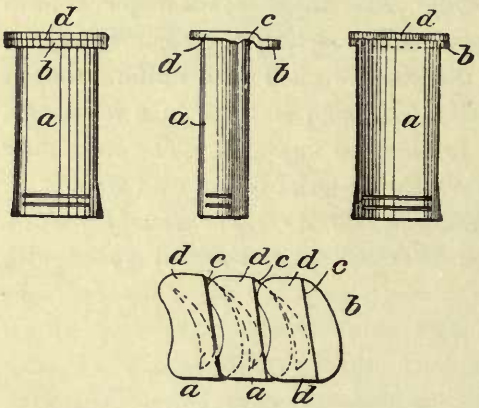
[1] D. R. P. 143,580.

[2] E. P. 12,347 of the year 1901. D. R. P. 144,528.

arrangement admits of local extensions of the cover, in case the latter should become heated by friction. In order to give a greater degree of steam-tightness in the joints of the cover a closure can, in all these cases, be produced by soldering.

In order to make the vane wreaths tight against the neighbouring guide vane rings for the avoidance of the clearance space losses, which will be referred to later, the head plates may project beyond the vanes, either on one of the sides (Fig. 483) or on both. In order to produce a bearing-ring for the glide-ring of the vane-wreath the holding rings or distance pieces *e* are made longer than usual, so that they project over the sides of the grooves and form raised rings. The glide ring *f*, together with the bearing rings, forms a closure of the vane wreaths one against the other.

In regard to the *attachment* of the vanes, which are not in one piece with the body of the wheel, to the latter, these must have a complete protection against



Figs. 479–482.

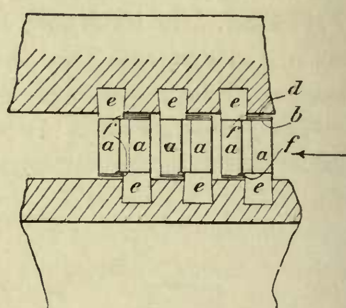


Fig. 483.

loosening even to a slight degree; it must not cost much, but it must admit of an exact adjustment of the vanes. These are conditions which, in particular for the insertion of the vanes, certainly require careful consideration. The problem has been and is being solved in various ways and with various degrees of success.

Thus *Wilson*[1] proposed to insert the plate vanes in the casting forming the body of the wheel in such a manner that their feet are surrounded and held by the cast metal. In order that a close connection may be formed between the feet of the plate vanes and the cast body, the former are covered with borax or similar fusion-promoting material. The heads of the vanes are then held together by a ring of brass or bronze cast on. An alternative arrangement can, according to Wilson, be made, in which the vanes are cast on in groups, with thin rings of brass or bronze at the sides. These segments are put together on the working wheel body to form a ring, and there fixed. A cast metal ring surrounds the head plates. In each case the working wheel is finally turned in the lathe.

In the method of *de Laval*[2] (1892) (Figs. 484 and 485) the vanes F are provided with feet F′, which are either dove-tailed or barbed. The body of the wheel

[1] E. P. 12,026 of the year 1848. [2] D. R. P. 68,359.

consists of two discs A, B, which are provided with annular grooves G, H, to suit the feet, and after the insertion of the latter are pressed together by being tightly fixed between the nut E and the ring-projection C on the sleeve D. The body of the wheel, however, may (Fig. 485) consist of a single piece A′, in which the deep slit K is turned, in order that the discs may be able to spring apart to admit of the insertion of the feet of the vanes. The assemblage of the parts must allow of a close setting together of the vanes, the flangings F^2 of which close the steam channels at the periphery of the wheel.

High-speed free-jet turbines with vanes like those of the Pelton wheels are, according to *Raworth's* plan[1] (Fig. 486), put together in the following manner. Each vane *a* has a foot *b* with dove-tail-shaped section. Grooves are cut into

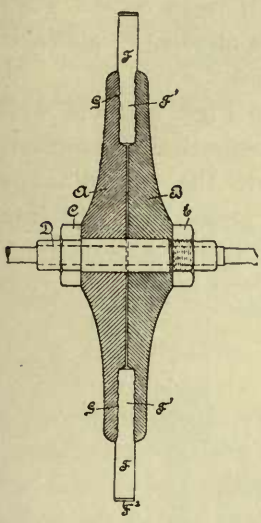


Fig. 484.

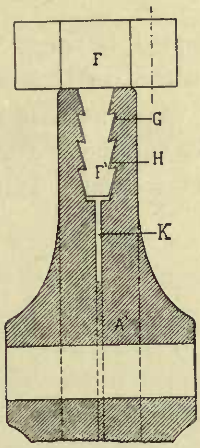


Fig. 485.

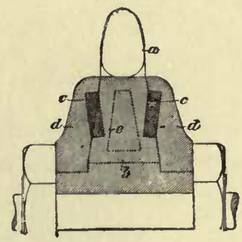


Fig. 486.

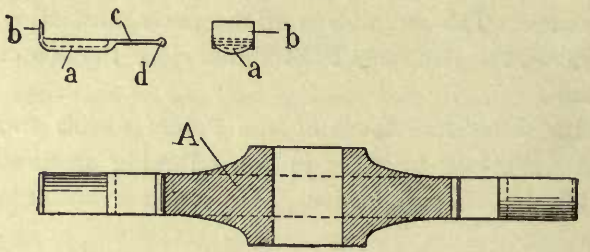
the sides of the feet, and in these are inserted the rings *c*. Between every two rings is fitted a distance-piece, and the whole is enclosed at the sides by the discs *d*. Otherwise the sides of the foot *b* may be left uncut and a wedge-formed opening *e* may be worked out in the middle instead. The feet of the vanes may then be provided with packing pieces and be put together on a suitably formed ring. For this the latter is provided with a slit; the slit is then to be electrically welded together (?). Further, the whole assemblage is finally to be welded or soldered together into one.[2]

Schmidt[3] (1896) (Figs. 487 to 489) goes to work in a different manner. The vanes *a* are provided at their heads with flanges *b*, which fit together to form a ring and close the channels towards the outside. The feet *c* are flat down to the swellings *d* at the bottom. The turbine body A is provided with borings and slits corresponding in number with the vanes *a*, and arranged to take the feet of the latter, which are pushed in from the side. After all the vanes are

[1] E. P. 25,086 of the year 1893. [2] Compare also E. P. 3,506 of the year 1901.
[3] D. R. P. 91,342.

in place the wheel is subjected to a lateral pressure, in order to ensure that the feet of the former fit completely into the holes provided for them.

Fullagar[1] fixes the vanes in a similar manner, but round their feet and heads he arranges concentric thin plate rings, which form the boundaries of the cells in the radial direction. Instead of cutting the slits, which are made in the disc of the wheel to take the feet of the vanes, exactly axially, Fullagar curves them so that they can be worked out by means of a conical borer and become still more curved when the pressure alluded to above is applied to the disc.



Figs. 487–489.

A method of attachment adopted by *Seger*[2] is shown in Figs. 490 and 491. At the lower ends of the vanes *a* the metal is worked out, projections *b*, however, being left standing. After the vanes have been pressed into the curved shape required, they are inserted into the turbine body, so that the projections *b* fit into corresponding holes *c* of I-shaped section. The fillet pieces *d* of the projections

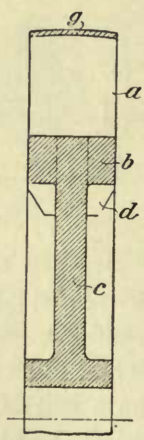


Fig. 490.

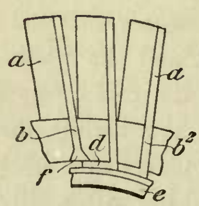
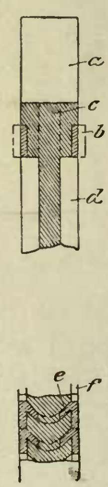


Fig. 491.



Figs. 492, 493.

are bent over in the manner shown in Fig. 491, so as to act as guards in holding the vanes in their places. The security given by the guards is increased by the insertion of the ring *e*. A method can, however, be employed, by which only so much of the material of the projection reaches over the edge of disc *c*, and trends radially inwards, as will admit of the riveting shown at *f*. Round the heads of the vanes also is fitted a thin plate ring *g*, with a section somewhat curved, so that it can take firm hold of the similarly formed vane heads.

[1] E. P. 14,594 of the year 1901.
[2] E. P. 4,611 of the year 1894.

In an alternative arrangement to the above *Seger*[1] (Figs. 492 and 493) punches the vanes *a* out of plate material with the lateral lug-pieces *b* and the lightening hollow *c*. The disc *d* is, on its part, provided with the indentations *e*, and the annular grooves *f*.

The vanes are bent into the required shape, and their feet inserted into the indentations *e*, and the lugs are then bent over till they fill the grooves *f*.

The *Société Sautter, Harlé & Cie.*[2] (*Rateau*) (1900 and 1901) (Fig. 494) rivet the radially set plate vanes 3 to the working-wheel disc 1 when the latter is made of iron plating. The bottom corners 6, which result from the riveted attachment of the vanes to the wreath 1, are filled out with molten metal to enable their thickness to be as far as possible reduced, and their strength to resist the action of the steam still to remain sufficient. The heads of the vanes stand free, or they are held together by means of a ring, to which they are riveted.

On the method of *Montag, Hüter* and *Karb*[3] (1898) (Figs. 495 to 498) the

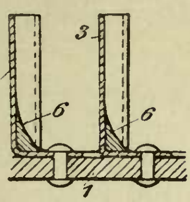


Fig. 494.

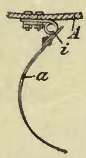


Fig. 495.

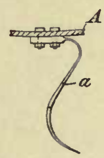


Fig. 496.

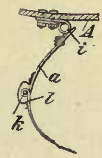


Fig. 497.

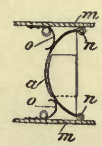


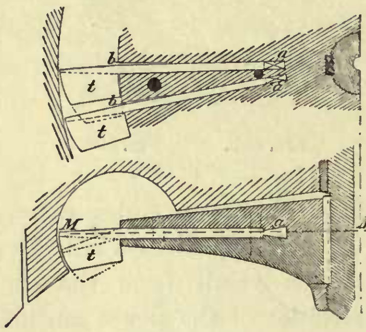
Fig. 498.

vanes *a* are connected with the turbine wheel A with an elastic or spring attachment, so that it can give way to the jet of steam. By this expedient a reduction is said to be effected in the losses in working, which result from the concussions due to the differences in the respective velocities of the power medium and of the vanes. The vanes *a* may be connected with the wheel A by means of springs *i* that are wound round in screw form (Fig. 495). They themselves can, at the part adjacent to the wheel A, be so far weakened that they have a certain amount of spring in the direction of the periphery of the wheel (Fig. 496), or else the vanes *a* are connected together in two or more parts, and likewise all the parts of the vanes one with another, by means of spring connecting pieces *k* (Fig. 497). Finally, the vanes may consist each of two parts, and each of these, supported by suitable springs *o*, is mounted free to move on the side cheeks *m* of the turbine wheel at *n* (Fig. 498).

Another yielding attachment has been proposed by *Buttenstedt* and *Mewes*[4] (1900) (Figs. 499 and 500). The wings or blades set with elastic attachments in the axis of rotation, after the manner of wind-mill vanes, are struck by the steam in a direction at right angles to the direction of motion of the blades. It is thus rendered possible for the pressure of the steam to be transmitted to a number of vanes, fitted one behind another, after the manner of those in the

[1] E. P. 22,842 of the year 1897.
[2] D. R. P. 143,960. E. P. 11,701 of the year 1901.
[3] D. R. P. 105,654.
[4] D. R. P. 135,333.

Parsons turbine, in elastic settings, and for the revolutions per minute to be altered within wide limits to degrees corresponding with the falls in pressure experienced in the individual turbine vanes of the above-named row. The wings, or vanes, can, as shown in Figs. 499 and 500, be impinged upon axially or radially, the steam being allowed to enter the turbine case, and to stream from the axis towards the periphery of the wheel. The elastic wings, which are arranged on the latter in a circle, are then opened by the impact. In the foregoing case the elasticity is not attained by means of spiral springs, the attachment of which would be difficult. Instead of these, a long steel wire or rod, which at one end is worked out to form the wing *t*, and at the other is dove-tailed, is held fast in the driving wheel. From the dove-tail fixing position *a*, the clearance space between the wire and the driving wheel (see double line *b*) widens radially outwards. The elastic wing will then be capable of being turned by the steam, not only about the point *a*, but also in the backward direction. Finally, the wing *t* itself forms an elastic surface, the point of support of which is the steel wire.



Figs. 499, 500.

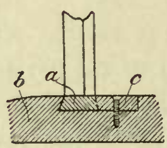


Fig. 501.

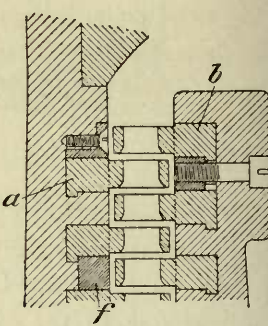


Fig. 502.

These elastic attachments can hardly be said to be of practical value, for the reason that the continuous working of the material at the points of attachment is sure to lead to early fracture.

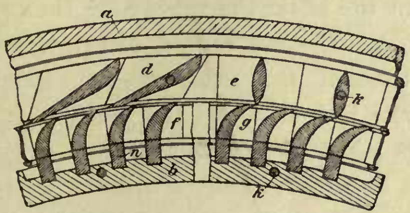
A renewal of faulty vanes is rendered possible by the methods of attachment of *Hodgkinson*[1] (1900) and *Parsons*[2] (Fig. 501). The individual vanes, with feet *a*, are placed in a row, with these set in dovetail-formed annular grooves cut in the drum *b*. To enable the feet *a* to be inserted into the grooves slot-holes are arranged at the sides of these. To fill up the slot-holes plugs *c* are provided, which are screwed to the drum.

In the multi-stage radial steam turbine of *Weichelt*[3] (1901) (Fig. 502), also, removable vanes are fixed in annular grooves. The attachment is here made entirely by means of a one-sided or double-sided projection of rectangular section taking the form of the annular groove and fitted into it. The vanes *a* and *b* are inserted by means of side slot-holes, and tightly pressed together.

[1] A. P. 672,838. [2] E. P. 7,065 of the year 1901. [3] D. R. P. 133,565.

The slot-holes are then closed by means of plugs *f* of trapezium form, or by means of cubes of gradually diminished size.

The vane arrangement of Messrs. *Gelbke* and *Kugel*[1] (1902) has reference to radially impinged multi-stage turbines, in which the motive medium courses through the guide and working wheels in U-shaped coils, and the vane wreaths are made up of separate single vanes. The designers aim at the production



Section AB *Section CD*

Fig. 503.

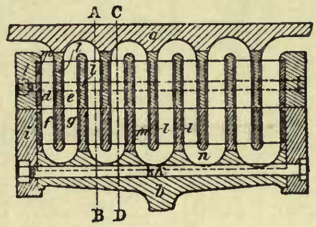


Fig. 504.

of theoretically correct vanes, suitably put together. In view of this, either the vanes arranged side by side in the axial direction are made in one piece, or the entrance vanes and the discharge vanes are made separate from one another. The last-named arrangement is shown in Figs. 503 and 504. Here the guide wheel *a* bears the vanes *d* and *e*, and the working wheel *b* the vanes *f* and *g*. It is here assumed that the guide vane *d* always gives the steam to the working-wheel vane *f*, and that the working-wheel vane *g* passes it on to the guide-wheel vane *e*. The vanes are provided at their ends with lug pieces *m*, which are made to suit corresponding annular groovings *l*, by which they are held in place. The groups of vanes are held together by means of the tightening screws *k* and the side discs *i*. The entrance and discharge vanes communicate with one another by special transmission pieces *n*, each of which may belong either to one cell or to several.

Parsons[2] (Fig. 505) cuts the vanes *a* from rolled or drawn bars that have the same section as these. In the wheel body *b* is worked an annular groove *e*, which may suitably be provided with scorings or flutings. In the groove are inserted the vanes, with distance pieces between them, which also are cut from bars of suitable section. After all the vanes and the distance pieces are in place the latter are caulked or staved up, so that the groove is filled. Vane feet and distance pieces may be roughened to make them hold together better. The separate distance pieces may be avoided if equivalent projections be pressed on to the feet. Long vanes are further made with side indentations, into which a ring *d* is worked. The ring can also be arranged at the periphery. Otherwise, holes are bored in the heads of the vanes, and a ring is drawn through, which holds the latter together.

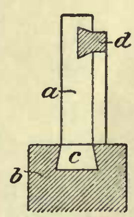


Fig. 505.

Fullagar[3] also recommends that the annular groove *c* be cut somewhat wider

[1] D. R. P. 148,390.
[2] E. P. 8698 of the year 1896.
[3] E. P. 14,593 of the year 1901.

than the breadth of the feet. A strip of metal can then be inserted and caulked in.

Parker[1] (Fig. 506), on the other hand, turns in the shaft, or in the full body of the working wheel *a*, an annular groove *b* of dovetail section. He also planes a longitudinal groove into the same, in which the feet of the vanes are inserted, to be then pushed forward to the annular groove *b*. With their feet in the latter the vanes *c* are then pushed in, one after another; the foot of the last vane *d*, however, is placed in a socket cut in the bottom of the axial groove. Below the feet a strip of metal (copper) *e* is let in, and the piece of it under each of the inserted vane feet is used for caulking.

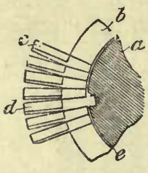


Fig. 506.

An improvement of the foregoing has been sought by *Parsons*, *Stoney*, and *Fullagar*[2] (1901) (Figs. 507 to 509) in the following method. The individual steam turbine vanes are to be combined together in simple manner to form complete rings, half-rings, or ring-sectors. The single vanes are thereby to be securely fixed in such a manner that they can, if necessary, be readily replaced by new ones, and the arrangement is such that the complete ring can easily be connected with the cylinder, or with the working wheel of the turbine. The rings or ring-sectors *a*, *b*, are formed by one or more suitably bent metal bands, into which nicks are cut to take the vanes *c*. The teeth *e*, *f*, formed by the nickings, are then bent over the vane ends inserted in them so as to form a close fit. The vanes are thus held in nicks, cut in one or more rings, by means of the teeth bent round them. The complete rings or ring-sectors *a*, *b*, together with the stronger felloe *a*, which may suitably be somewhat increased in dove-

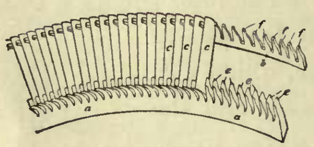


Fig. 507.

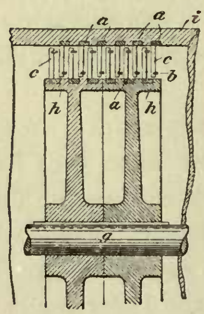


Fig. 508.

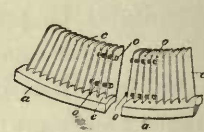


Fig. 509.

tail form towards its lower end, are fixed in suitably formed grooves cut in the fixed cylinder or case *i*, or in the working-wheel drum *h* that is wedged on to the shaft *g*. The purpose served by the stronger felloe *a* is the connecting of the vane-bearing ring with the cylinder, or with the working wheel of the turbine, while the lighter felloes *b* keep the ends of the vanes *c* at their proper distances apart (Fig. 508). For the lower stronger felloe a strip of metal, as broad as possible, may suitably be adopted, in order that (as may be seen

[1] E. P. 324 of the year 1902. [2] D. R. P. 115,228.

in Fig. 508) the working drum, which is generally made of steel or iron, may be covered or protected by the brass ring. After the finished rings have been inserted, the teeth *e* are caulked, one by one, so that the former are held completely fast in the grooves. Strips of soft metal may also be let into the grooves at the sides of the felloes, and the latter may, by the help of these, be caulked tight. For the connection of the last vanes of two adjacent ring-halves, or ring-sectors, pins *o* are provided, passing through borings in the vanes (as may be seen in Fig. 509). These are soldered or welded to the individual vanes. In order to fix the vanes at the point of connection of two halves as firmly as possible, the last of the former may also suitably be soldered or welded to the teeth *e* of the felloe *a*. For radial turbines an arrangement similar to that of Fig. 508, for axial turbines, may be made. The machine, by the help of which the method is carried out, is shown in Fig. 510, and works as follows. In the felloes, or metal bands *a*, which are placed on a table *i* that turns at certain fixed intervals, nicks are cut by the up-and-down motioned cutting disc *l*, and the teeth formed by the nickings are closed by the backwards-and-forwards moving stamp *u* over the inserted vanes.

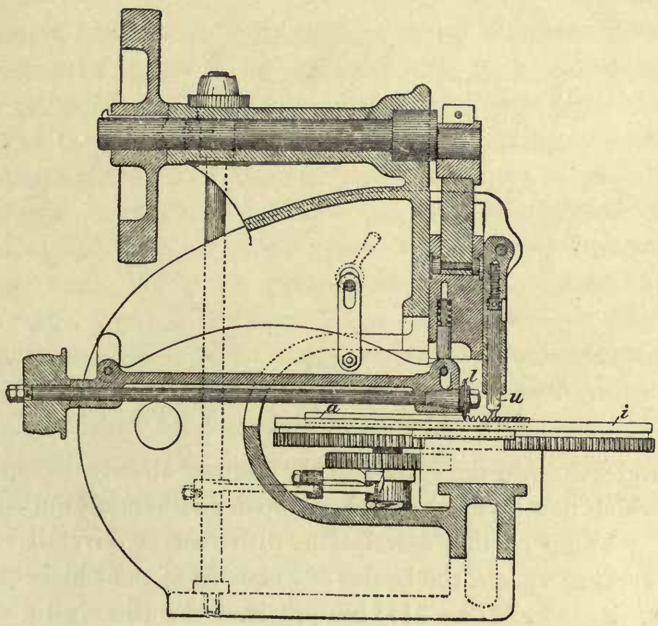


Fig. 510.

Parsons has also entertained the idea of casting the head and foot rings into the vanes.[1] These are, in the first place, stuck through strips of zinc or similar metal, so that they project beyond them at each side. The strips are then bent to form ring pieces, and are imbedded, together with the vanes, in a mould. The metal poured into the latter, which is intended to absorb the greater part of the zinc, flows round the vane head and foot ends, which project above the latter, and, in combination with these, forms the sectors, which are to be put together to form wheels.

The method of attachment by means of lateral holding rings for the vanes and tightening rings for the holding rings can, according to *Fullagar*,[2] be applied as shown in Fig. 511, in which the two holding rings are provided with notches cut into them from their adjacent edges. In these notches the side ends of the vanes are inserted in such a manner that they are half grasped by each of the two holding

[1] E. P. 16,284 of the year 1899.

[2] D. R. P. 152,258. E. P. 7,184 of the year 1901.

rings *a* and *b*. A lateral bracing ring is inserted between the turbine case or turbine shaft and the holding rings, by which the latter are pressed firmly one against the other, and against the vanes which they surround. A completely tight grasp is attained by the arrangement by which the holes in the one holding ring stand at an angle with those in the other, and the cleft ends *c* of the vane feet are bent apart in such a manner that they fit into the holes above mentioned (Figs. 512 to 515). Compare also the arrangement of *Parsons*.[1]

Another method of attachment given by *Parsons*[3] (Fig. 516) is such that out of the ring *a* the teeth *b* are worked, between which open spaces *c* are produced. Into these the vane feet *d* are pushed one after another, and it will then become necessary

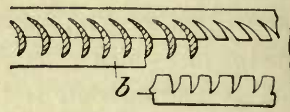
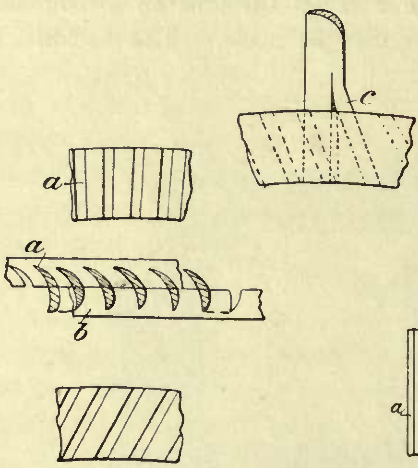


Fig. 511.



Figs. 512–515.

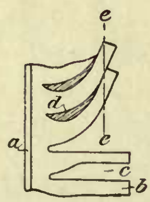


Fig. 516.

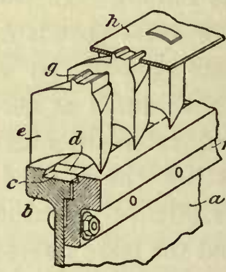


Fig. 517.

to bend down the teeth *b* in the manner already described. The edges of the teeth, which now lie firmly fixed one upon another, are worked away to the line *e*.

Vanes of this description, with feet of dovetail section, may also be used for working wheels, the bodies of which, as in the Curtis turbine, consist of discs. Thus *Geisenhoner*[2] (Fig. 517) has provided the disc *a* with a flange *b*, in which a groove *c* is turned. The feet *d* of the vanes *e* fit into the groove, and are successively pushed into it. The groove *c* is completed by the addition of the ring *f*, which is brought on from the side and screwed against the disc *a* in such a manner that the feet *d* are held fast. The vanes *e*, which may be cast or pressed on, or worked out of the solid, have projections *g*, which extend through the surrounding plate ring *h*, and their flat ends are flanged over the ring so that the latter is firmly connected with the heads of the vanes.

Attention may finally be called to the method of *Wichmann* and *Weller*[3] (1903) for the manufacture of vane wreaths, which is suitable for axial turbines. The steam channels are worked into a ring, the thickness of which is equal to their breadth, and the width equal to their length. The ring is then shipped over the disc of the wheel, and held round by a clasp ring, which closes the channels towards the

[1] E. P. 12,347 of the year 1901. [2] E. P. 18,084 of the year 1903. [3] D. R. P. 153,642.

outside. The projecting edges of the ring containing the channels are finally planed off, so that the inflow and outflow sides of the latter are laid open.

In regard to the *calculation* of the sizes and forms of the steam passages, the principles which rule in the dimensioning of water turbines must, in consideration of the physical properties of steam, of course undergo alterations, more particularly in connection with the conversion of heat into current energy and *vice versâ.* While the theoretical calculation of the quantity of steam required to produce a certain power and the choice of the general dimensions do not in themselves give much trouble, when once the style of construction has been fixed, the determination of the losses, to which the steam is, in its course, subject, and which vary to a considerable degree with the system chosen, is attended with some difficulty, which is the greater that practical experiments are constantly giving evidence of fresh sources of loss.

The hitherto known theories, however, suffice to enable steam turbines, on the systems of construction which have chiefly to be reckoned with, to be calculated with such exactness that the values determined give results differing very little from those of actual practice. The *turbine theory* of *Zeuner* here establishes standard principles. In regard to turbines of the Laval description, mention may be made of the writings of *Delaporte,* "Étude générale du rendement des nouvelles turbines de Laval" (*Revue de Mécanique,* 1902, Vol. 10, p. 466), and *K. Sosnowski,* Turbines à Vapeur Système de Laval (Paris, 1903). A general disquisition is given by *A. Abraham,* in his treatise, "Les Turbines à vapeur" (*La Revue technique,* 1903, Vol. 24), while *M. Hart,* in the pamphlet, "Les Turbines à vapeur et spécialement les Turbines Parsons" (Paris, 1903), goes specially into the question of the Parsons turbines. Rateau has repeatedly published the theory of his method of construction.[1] The minor details, and to some extent the degrees of influence which these are known to exert, have received careful attention at the hands of *Stodola,* in his work, Die Dampfturbinen (Berlin, 1904). Further mention must be made of the work of *H. F. Schmidt,* "The Turbine Problem" (*American Electrician,* 1904, Vol. 16, p. 76). Finally, the Berechnung der Dampfturbinen auf zeichnerischer Grundlage, according to *A. Koob* (*Zeitschr. d. Ver. deutsch. Ing.,* 1904, Vol. 48, p. 660), deserves the special attention of designers.

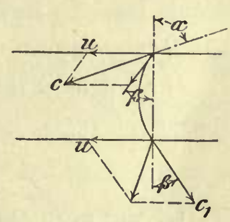


Fig. 518.

Let it be assumed that the jet of steam issuing from the Laval nozzle with the velocity c (Fig. 518), at an angle to a line parallel to the axis, streams against the vanes of the axial, single-stage (Laval) turbine with the velocity u, and leaves the working wheel with the velocity c_1. Let the tangents to the arc-shaped vanes be set at an angle β to the above-assumed parallel line. The formula for the concussionless inflow and outflow of the steam will then be—

[1] Compare also Rey, "La Turbine à vapeur de Système Rateau et ses Applications," *Soc. d'Ing. Civ. de France,* le 18 Mars, 1904.

$$\frac{u}{c} = \frac{\sin(\alpha - \beta)}{\cos \beta}$$

and

$$\frac{c_1}{c} = \frac{\cos \alpha}{\cos \beta}$$

The most advantageous use of the current energy will be obtained when the angle $\alpha = 90°$, as is the case in wheels with tangential impingement. In the axial turbine the endeavour is to place the axis of the nozzles as far as possible parallel to the wheel. In practice, however, smaller angles for the nozzles than $\alpha = 18°–20°$, and greater complementary angles than 72°–70° cannot be taken into account. It will be advisable, it may be observed, to choose the angle of outflow β_1 smaller than the angle of inflow β.[1] The relative velocity of outflow is, by reason of the friction, smaller than the relative velocity of inflow; the former is about 0·7 to 0·9 of the latter, which represents a loss of energy of from 20 to 30 per cent. Also within the full working-wheel channels a conversion of velocity into pressure takes place, which is likewise attended with losses of energy, and is due to the circumstance that the relative inflow velocity is greater than the velocity of sound.

Since, to enable the use of a high number of revolutions of the turbine to be avoided, the energy of the steam is made use of by stages only, it will be necessary in the calculation for every working wheel of a velocity turbine to take into account only a fraction of the velocity of the steam depending on the number of grades made use of. The velocity is then divided into equal parts. The unpractical method of converting the whole pressure into velocity in the set of nozzles of the first stage only leads to the use of velocity stages; in this case the loss of energy during the process in the expansion nozzles, the quantity of energy present in the outflowing steam, the friction against the vanes, the losses due to concussion against opposing surfaces, and the influence of the centrifugal forces will in the main represent the deductions which will have to be made from the total store of work. The centrifugal forces, as already observed, make it necessary that a residue of pressure be left in the working jet of steam. If, however, a division be made into pressure stages, the nozzle loss—so styled for the sake of brevity—will repeat itself; but on the other hand, the congestion of the steam in the channels will no longer be apparent.

If G denote the weight, and H the current energy of the steam, the *available work* is—

$$L = G \times H$$

Of the work thus given, only a portion can be turned to account, which, in the mean time, is determined by the indicated coefficient of efficiency—

$$\eta i = 4 \times \frac{u}{c} \cos \alpha \tan \beta$$

Further, there are the inner resistances W to be deducted, which comprise the losses in the nozzle and in the working wheel, the concussions of the steam against

[1] Delaporte in *Revue de Mécanique*, 1902.

the edges of the vanes, and the influence of the centrifugal forces. Finally, there comes the outflow loss V to be deducted, depending upon the absolute velocity of discharge of the steam and upon the wheel. According to the foregoing, the *indicated work* of the axial, single-stage Laval turbine (for the sake of simplicity here taken as a basis) is—

$$Li = G \times H \times 4\frac{u}{c} \cos a \tan \beta - (W + V)$$

Means of directly indicating the work of the steam in the turbine, as is done in the case of the reciprocating engine, have, up to the present time, not been discovered. The general method adopted is to condense the steam, and from the weight of the condensed product to determine the consumption.

Since the turbines are, in the main, applied in conjunction with working machines, such as dynamos, their *efficiency* is commonly determined by a calculation based on the performance of the working machine, an allowance being made for the working coefficient of the latter. It is also sometimes considered sufficient to give the consumption of steam for the performance attained in the working machine, *i.e.* to set the consumption of energy in proportion to the effective performance of the whole group.

In the multi-stage pressure turbine, as it is built by Parsons, for instance, the steam expands on its working path through the stage wheels, and care must here be taken that a sufficient widening of the passage way is provided. In order that clearance space losses may be avoided, the fall in pressure has to be divided into a great many parts, *i.e.* a large number of stages have to be provided. In view of this the necessity arises of cheapening the cost of production by making the steam channels as uniform as possible, although theory requires them to vary from stage to stage. Parsons takes the theoretical considerations into account by a gradual increase of the working wheels, and therewith of their peripheral velocity, towards the discharge end. In the case of radial turbines with impingement from within, the stage-by-stage increase of the peripheral velocity comes of itself.

From the results of working of the Parsons turbine, it appears that of the available theoretic fall in pressure S of the steam, which lies between the entrance pressure and that of the condenser, along the path through the leading and working wheel channels, from 20 to 30 per cent. is lost, *i.e.* that a loss of energy E must be deducted. Further, the steam passing from the last working wheel to the condenser still possesses a kinetic energy K = 12 to 4 per cent. according to the size of the turbine and the position of the condenser. The loss by leakage U, which varies with the turbine system, may be put down at from 10 to 5 per cent., while the resistance W (inclusive of light load work) may absorb from 12 to 6 per cent. It will accordingly be necessary to reckon with a total loss of

$$V = E + K + U + W = \text{from 64 to 35 per cent.}$$

Of the theoretic fall in pressure an effective portion of only S − E, is obtained, while the actual performance of the quantity of steam acting in the working wheels only amounts to (S − E) − (K + U + W).

In regard to dimensions, it may be further observed that the vanes themselves are very small. Thus those of the Parsons turbine are, on the average, only two inches in length, while the breadth varies between 0·4 and 0·8 of an inch. In spite of this, a very large aggregate vane surface takes up the effective steam pressure, especially in the case of the Parsons turbines. Thus a turbine of from 1000 to 2000 H.P., for instance, has about 30,000 working-wheel vanes which give a pressure surface of about 30 sq. yards. According to this the specific steam pressure on the vanes is small.

In connection with the *working wheel* body itself the high number of revolutions is the matter of greatest interest. Safety in working requires a sufficient degree of strength, which must not, however, be bought at the price of an excess of weight of the revolving masses. This assumes the employment of such material —and this in such form—that want of uniformity in its composition and in the conditions of tension in the finished body is as far as possible non-existent. Especially is this the case with the disc-formed wheel bodies, the diameters of which may, under certain circumstances, be considerable, when, for instance, several stages are contained in a single wheel. In regard to the calculation of the discs, reference may be made to the articles of *Kirsch*, Festigkeit rotirender Scheiben (Zeitschr. d. Ver. deutsch. Ing., 1897, p. 798), to the works of *Stodola*, "Die Dampfturbine" (Berlin, 1904), and to an article of *M. F. Fitzgerald*, "Steam Turbine Discs" (*The Engineer*, 1904, Vol. 97, p. 481).

From the experience of the *Maschinenbauanstalt Humboldt*[1] (1898) may be gathered the following. With the rapidly-revolving discs of the steam turbines, *e.g.* such as have revolving middle bodies, with vanes fixed on their peripheries, the attempt has been made to insure that, in case the speed be exceeded of which the strength will admit, only small pieces can fly off. To this end the disc from its centre to the rings at its periphery which are necessary for the attachment of the vanes, is gradually reduced in thickness. In the course of experiments undertaken with discs of this kind, it however transpired that, by reason of the irregular composition, fracture did not always take place within the ring referred to, and that, accordingly, not only did this break loose, but also larger portions of the disc itself went along with it.

A better solution of the problem is now to be achieved as follows. The disc *a* (Fig. 519) is to be weakened near its periphery by suitable manipulation (the slotting out of grooves and channels, the boring of holes, etc.) in such a manner that, when the admissible rate of speed is exceeded, the fracture of the disc takes place at the weakened point, and thus only the part outside of the latter can be torn off.

Gross[2] (1902) arranges notches in the wreath that bears the vanes. These

[1] D. R. P. 105,073. [2] D. R. P. 144,865.

allow the portions of the wreath to expand when, during the motion of the wheel, the wreath is heated to a higher degree than the disc.

In general, however, the conditions of manufacture are such that discs of unimpeachable quality can be made in single pieces. This will be preferred to those systems in which a wreath is connected with the nave by separate plate discs (*Davidson*[1]). This advantage is also established in the circumstance that the balancing of the weights is, in the first case, such that the axis of rotation of the wheel as far as possible, coincides with its centre of gravity. According to the particulars given by *Riedler-Stumpf* this coincidence is attained to 0·0004 of an inch. A method of balancing the working-wheel on the natural axis of rotation is, according to *Whitcher* and *Roberts*,[2] that in which the wheel is mounted complete on the shaft, and the latter is set in flexible bearings. The wheel can then adapt itself to its axis of gravity. After the wheel has been set rapidly rotating, the position of the shaft is fixed, and the surface of the former finished off.

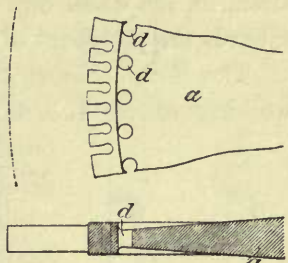


Fig. 519.

An exact balancing of the wheel relieves the designer of the necessity of having to resort to a flexible connection of the wheel body with the shaft (*Lake*[3]), (*Nilsson*[4]), or to make the wheel elastic (de Laval)—all of these means which cannot be made use of in the case of large machinery.

In order to reduce the centrifugal forces the endeavour will be to make the working wheels as light as possible, without detracting from their strength. Indentations and perforations must, however, be applied with caution, because they give rise to eddying, and thus increase the resistances. In order to combat these drawbacks *Gross*[5] (1902) (Figs. 520 and 521) makes use of a turbine wheel

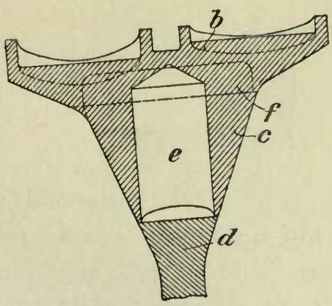


Fig. 520.

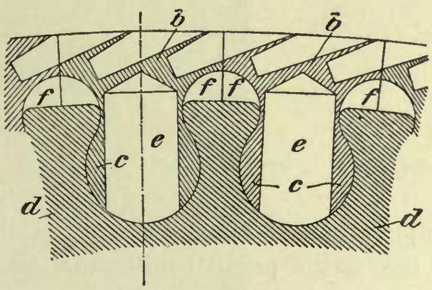


Fig. 521.

with a wreath made in several parts. This is attached to the disc of the wheel by tenons, or other similar method, and in the various parts of the wreath (vane segments b, or disc d) hollow spaces, or lightenings, are formed in such a manner

[1] E. P. 15,503 of the year 1896.
[2] E. P. 2,815 of the year 1900.
[3] E. P. 17,273 of the year 1894.
[4] E. P. 23,759 of the year 1899.
[5] D. R. P. 147,762.

that the wheel has a smooth outer surface. The vane segments *b* are worked into the working wheel *d* by the help of projections *c*. The projections have borings *e* and flutings *f* which are not carried right through. The disc could be honey-combed in similar manner at the periphery. From considerations of strength, the wheel disc *d* and the projections *c* are widened, in the case illustrated, towards the periphery of the wheel.

The *Société Sautter, Harlé & Cie.* (*Rateau*) (1900)[1] (Fig. 522) claims for the working wheels 1, 3, 4, made of flat plating, the advantage over conically formed ones of like material, that centrifugal force does not tend to deform the flat discs 1, nor to bend the peripheries of the wheels in a lateral direction, so that friction could be set up against the fixed guide wheels.

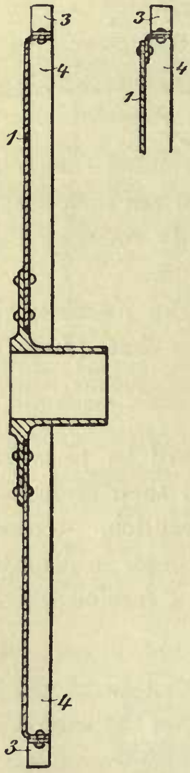


Fig. 522.

Also the irregular warming of the working wheel may be detrimental to the latter. In wheels of proportionately large diameter, temperatures of 150° and more in the wreath, and from 40° to 50° in the disc body proper, have been measured by *Gross*. In view of this the latter[2] (Fig. 523) provides the

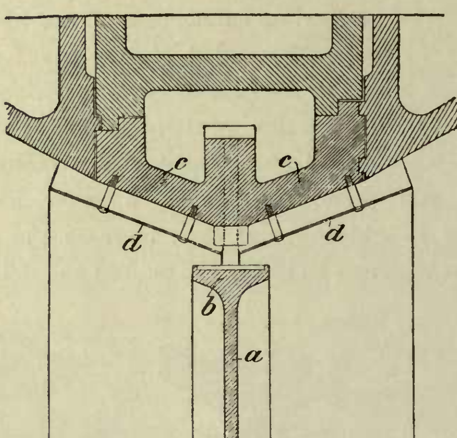


Fig. 523.

steam-supply organ (the annular channel *c*) with an insulating appliance *d*. The object of this is the protection of the wreath *b* of the turbine wheel *a* against heat radiating from the supply organ during the passage of the fresh steam, and against its consequent destruction, brought about by uneven expansion of its parts.

On the other hand, *Morton*[3] wishes to protect the case against the effects of the heat radiating from the turbine wheel, and accordingly coats the wheel with a material which forms an air-tight covering to it, and revolves with it.

Fig. 524 (see Plate III.) shows the working wheel, together with the shaft of a Westinghouse-Parsons Turbine of 3000 H.P. The weight of these rotating parts

[1] D. R. P. 143,960. E. P. 11,701 of the year 1901. [2] D. R. P. 147,600.
[3] E. P. 9158 of the year 1888.

PLATE III.

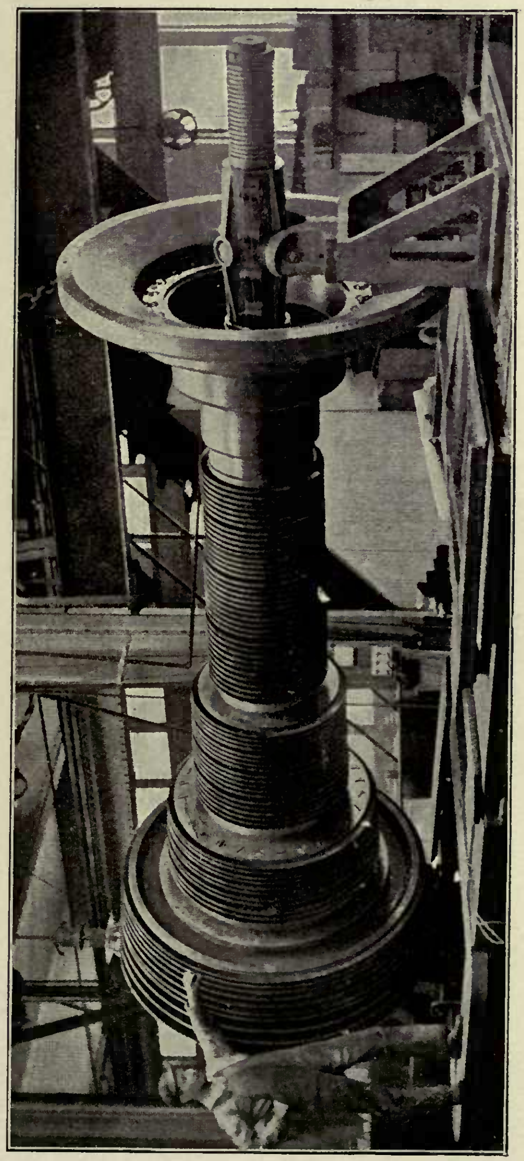


Fig. 524.
Westinghouse-Parsons Turbine of 3000 H.P.

is, in round numbers, $12\frac{1}{2}$ tons; the total length, 19 ft. $8\frac{1}{2}$ ins.; and the distance between the bearings, 12 ft. 3 ins.; the greatest diameter is 6 ft.

It is practically impossible to equilibrate the working wheels so exactly about their turning axis that the latter coincides with their natural axes of weight. It thus becomes possible that at the *critical speed of rotation* of the discs and such-like parts these seek to shift their positions from the axis assigned to them to those of their axes of weight. In what manner this subsequent adjustment will in each case take place cannot, of course, be determined beforehand. The possibilities are discussed (*l.c.*) by Stodola. It is to be observed, however, that this critical speed has, in the construction of large steam turbines, ceased to be a matter of importance, because it is possible, for the number of revolutions that are likely to be adopted, completely to equilibrate the rotating masses.

XVI

CLEARANCE SPACE PACKINGS

BETWEEN the working-wheel vanes and the fixed organs surrounding them there is a *clearance space* at which, when the pressure of the steam at this point exceeds that outside, the so-called "*clearance space loss*" takes place. Steps are accordingly taken as far as possible to remove this source of loss or to neutralize its effect.

An obvious expedient is to move the working wheel close against the leading apparatus, and to provide a special packing for the clearance space. Thus *Henderson*[1] (1897) (Fig. 525) has provided the guide and working wheels with projecting edges, over which a bronze ring is laid with a lining strip of Babbitt's or such-like metal.

Farcot and *Perrigault*,[2] in addition to the labyrinth packing for the closing of the clearance space between the working-wheel disc and the lateral walls of the

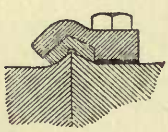


Fig. 525.

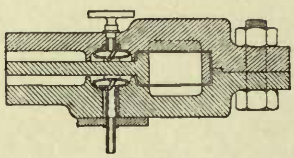


Fig. 526.

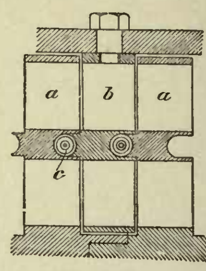


Fig. 527.

case (Fig. 526), make use of elastic packings. These consist of hollow rings *y*, which abut against the disc of the wheel-like spring cushions, and the inner sides of which are subjected to a pressure which is regulable.

A method of packing the working wheels of steam turbines, in which the pressure medium enters the wheel through a slit, is proposed by *Marsh*[3] (1899). The object is that the wheel, which in way of the vanes is not provided with side walls, is packed at the side by a flexible plate fixed outside. The packing plates are provided only at the point of entrance of the steam. A high degree of expansion of the steam is here contemplated.

Scott, Tyzack, and Summerfield[4] (Fig. 527) propose to pack the working wheels *a*, which to admit of reversing have double wreaths, against the fixed guide wheels *b*

[1] A. P. 634,170.
[2] E. P. 1206 of the year 1866.
[3] D. R. P. 127,710. E. P. 20,410 of the year 1899.
[4] E. P. 22,740 of the year 1902.

with feathers and grooves, which leave annular channels between them. Packing rings *c* are then inserted in the latter.

On the other hand, *Hellweg*[1] (1902) arranges the packing place between the working wheel and the case. In the latter he turns annular grooves, in each of which a working wheel runs with its outer ring. Between the wall of the case and the ring, however, a clearance space is left, which is filled with oil under pressure.

By the method of *Dodillet* and *Bergmann*[2] (1900), an oil packing with the least possible friction is to be made between the periphery of the vane wheel and the wall of the case. The vane wheel runs in oil contained in the lower part of the turbine case, and, carrying some of this along with it, throws it against the walls. According to another proposal[3] of the same designer, the working wheel is arranged so that it can shift in the radial direction. In order, then, to readjust the packing of the clearance space, the working wheel can be moved against the leading apparatus, which bears at one side only.

The advice has also been given that packing bars be inserted in the working wheel vanes. These are then to be pressed against the case (*Pooler*[4] (1893), *Davidson & Stacey*[5] (1897)). *Barnes*[6] (1893) even makes the vanes themselves elastic in such a manner that the steam pressure acting on them lengthens them in the radial direction, and thus makes them bear against the casing.

Deutschmann[7] arranges the working wheel to turn on rollers in a fixed frame. Against its periphery cushions of steam containing inlets and outlets are kept bearing. In each of these cushions the impingement nozzle is followed by an expansion chamber in which the steam of the charged working-wheel cells expands during forward motion, and then by the exhaust.

Meanwhile a complete closure of the clearance space, even when it is made on the yielding or elastic principle, is practically impossible because the high speeds which have to be reckoned with must be accompanied by great frictional losses, and even by heating of the packing material to the point of melting. At the same time the alteration in the lengths of the rotating masses while the engine is at work, caused by their setting themselves to their natural axis of weight, and the alteration of their dimensions due to extension produced by heat, must have attention. The result is that a prevention of the unintended passage of the steam by the pressing of the surfaces one upon another has to be given up. Another method is to arrange this packing at a place at which the relatively smallest speed occurs, *i.e.* at the shaft, where *Rateau*, for instance, puts it. The outcome of a method of this kind, however, is that for each stage a chamber, the diameter of which is equal to that of the wheel, has to be provided and placed under the pressure obtaining in the clearance space. For wheels with empty-running channels this brings disadvantages with it by reason of the ventilation resistances.

[1] A. P. 713,065. [2] D. R. P. 129,183 [3] D. R. P. 130,344.
[4] A. P. 516,022. [5] A. P. 608,761. [6] A. P. 517,513.
[7] E. P. 5011 of the year 1902.

Packing which can impart impurities to the steam, such as those in which oil plays a part, cannot be entertained.

Since it is in every case of importance to increase the resistance at the clearance spaces to such a degree that the steam pressure (above) is not sufficient to overcome it, the use of labyrinth packing has been taken into consideration, by which rubbing surfaces may be obviated, and which admit of a certain shift of the working wheel in relation to the fixed parts of the turbine. Such labyrinth packing is, for instance, to be seen in the arrangement of *Obenhain*[1] (1900), who puts it between the working wheel and the leading apparatus, and also between the latter and the case of the turbine. The latter, it may be observed, has radial impingement from without. A similar arrangement is to be met with in a method of construction which was designed by *Hoffbauer* and *Rüdemann;* in this the labyrinth packing with several chambers is fixed beside the clearance space.

An unusual way has been chosen by *Scott, Tyzack,* and *Summerfield*[2] (Fig. 528). In the guide-vane bearers *d* channels *e* are provided, which lead from the guide cells into the clearance space between the working wheel and the guide wheel. The steam passing from the working wheel *f* into the guide wheel, and from the latter into the next working wheel *g*, has the tendency to escapė in the direction of the arrow *h* through the clearance space *k, l.* Here it meets with counter pressure of the steam streaming towards it from the channels *e*.

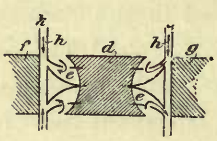


Fig. 528.

Now it is clear that the labyrinth packings also become practical impossibilities when real steam-tightness under high pressures has to be attained. The appliances then become bulky, complicated, and costly, and their value becomes at the best doubtful.

But how is it possible to reduce to a minimum the difference in pressure, which produces the clearance space losses? When, in the steam chamber, no excess of pressure over that in the outer chamber exists beside the clearing space, the motive medium cannot stream out. In a pressure turbine which is specially subject to the clearance space losses a reduction of this kind has to be provided as soon as a pressure is produced about the working wheel, which is approximately equal to that of the steam passing through the clearance space. Such conditions can easily be brought about in the multi-stage pressure turbines; for it is here possible—as in the case of the steam-pressure stages—to connect the chamber encircling one of the stage wheels with the impingement chamber of the next stage, and thereby to make the outer pressure for the first stage equal to the working pressure for the next one. The more pressure stages are created, the more certainly can the adjustment be effected. But the leakage steam also cannot in this case be called sensitive; for the steam that has passed through the clearance space of the first stage mixes with the working steam of the second stage and performs work in the latter, and so on up to the last stage, in which finally only a small excess pressure at the clearance space comes in question. The number of

[1] A. P. 678,811.

[2] E. P. 22,740 of the year 1902.

the pressure stages is, it is true, also of some importance in connection with the further use of the leakage, because the steam that has gone through the clearance space of the one stage is best utilized when it can enter the next stage without any considerable fall in pressure. The Parsons turbine, which works with a large number of pressure stages, has not to contend much with clearance-space losses. While preserving a very satisfactory consumption of steam, Parsons is in a position to make the widths of the clearance spaces 0·04 in. in the radial and from 0·08 to 0·29 in. in the axial direction, and therewith to take into consideration their longitudinal extents, which in the case of the long working-wheel bodies, which are exposed to high-pressure temperatures of steam, become very considerable.

In spite of this, *Parsons*[1] has considered the question of special packing for the clearance space for the case in which part of the channels of the first stage are impinged upon. A packing off of the stages one from another is said to be possible in the following manner: The guide-vane rings *a* (Fig. 529) lie with their head

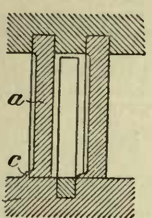


Fig. 529.

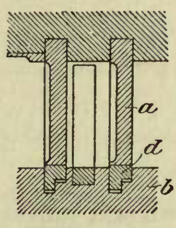


Fig. 530.

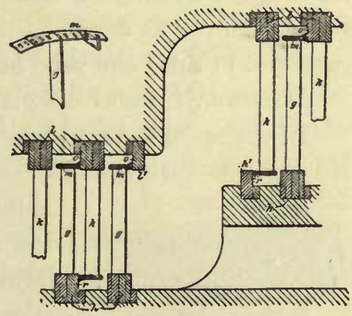


Fig. 531.

surfaces against the rotating drum *b*, and on the steam inflow side run close against the projection *c* of the drum *b*. In order to effect the adjustment of the projection *c* against the guide ring *a* after the erection of the turbine, the shaft, and with it the drum *b*, is mounted axially displaceable. In the arrangement shown in Fig. 530 spring rings *d* are inserted in annular grooves of the drum *b*, and are made to glide over the guide rings *a*. Moreover, the steam which may escape from the ring-shaped chamber before the first guide-vane ring between the working wheel drum and the case is led through an overflow channel to the exhaust.

Another method of packing the clearance spaces of multi-stage turbines with axial steam current has been proposed by *Fullagar*[2] (Fig. 531) as follows: Each vane wreath is provided with one or more packing rings, which, in the case of fixed guide-vane wreaths, make tight the inner circle against the lateral ring surfaces of the case. While sufficient space is left in the radial direction for the vanes, a bending of the shaft is thus rendered possible without the vanes being damaged or the packing injuriously affected. The packing rings are made of several layers *m* worked one above another, by which a labyrinth packing is attained. For these rings thin plating (less than 0·04 in. in thickness) should be chosen, so that, if it happen to

[1] E. P. 7066 of the year 1901.

[2] D. R. P. 152,268. E. P. 7184 of the year 1901.

come in contact with the holding rings, it will admit of a little rubbing off without being heated up to the melting-point, which would render necessary a considerable increase of the clearance space. The packing rings can, however, in each case consist of a single thin plate. The vane wreaths are provided at their peripheries with projecting pins and the packing rings with corresponding holes, into which the pins can be made fast by riveting.

The side-ring surfaces *o*, *r*, against which the edges of the packing rings reach on one side or on both sides (those for the first and last vane wreaths respectively of a set of wreaths excepted), are formed by the lateral surfaces of the holding rings *h*, *l*, to which the movable and fixed vanes *g*, *k* are attached. The ring surfaces for the first and last vane wreath respectively are formed by the side surfaces of separate rings h^1, l^1 attached to the shaft or to the inside of the case. In order to avoid the wear produced by rusting, the holding and packing rings may suitably be made of non-rusting material (brass or delta metal). Instead of having the packing rings made tight against the holding rings, annular grooves may be made in the case, and in these the vane heads and packing rings may be allowed to run.

Moreover, *Parsons* has given his attention to a similar style of packing.[1]

In the arrangement chosen by *Fullagar*[2] (Fig. 532), distance pieces *b* are fitted between the working-wheel rings *a*, while the working-wheel cells are bounded in the radial direction by plate rings *c*, *d*. In the rings *a* and in the guide-wheel rings *e* plate rings *f* are inserted, which form a sort of labyrinth packing. The guide-wheel rings *e*, it may be observed, are provided at their deepest parts with borings *g*, through which any condensed water which would offer resistance to the rotating wheels can flow to the exhaust. Otherwise all the chambers between pairs of

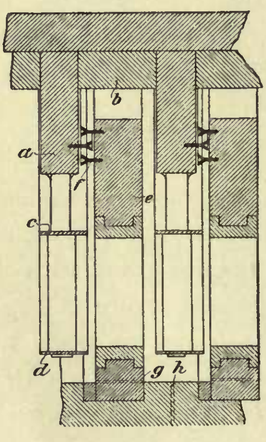


Fig. 532.

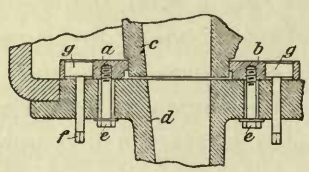


Fig. 533.

guide wheels are for the same purpose connected with a system of condensed-water pipes by means of borings *h*.

Curtis[3] (1903) also, who in his velocity turbine leaves the steam a small pressure sufficient to overcome the inner resistances, has proposed a packing for the clearance space (Fig. 533). The clearance spaces are covered at the outside by ring pieces *a*, *b*, which are screwed to the case or to the leading apparatus, and bear against the walls of the working wheel *c*. The joints, then, are placed in positions which admit of their keeping tight. The illustration shows the packing of the nozzle *d*, situated

[1] E. P. 12,347 of the year 1901. [2] E. P. 8934 of the year 1901. E. P. 5605 of the year 1902.
[3] A. P. 726,032.

in front of the clearance space. The screws *e* of the ring pieces *a*, *b* pass through oblong holes in the case. The ring pieces can then be shifted towards the walls of the working wheel. This is also rendered possible during working by the presence of the adjusting rods *f*, which are inserted with eccentrics *g* in the ring pieces *a*, *b*. The latter may suitably be made of brass or similar soft metal, so that the rotating steel walls can slide with sufficient freedom over the closely lying pieces.

The clearance space excess pressure, however, which in the Curtis turbine can come in question, will in the first stage, in regard to passage of the steam, only play a subordinate part, and in the lower stages become still less important. The excess pressure in question will be eliminated when the steam in the expansion nozzle is allowed to expand to the full and enters the first working wheel of a multi-stage velocity turbine in a pressureless condition. The clearance space could then, for all alterations of the working wheel, be made wide enough without fear of an escape of the steam. Consideration must here, however, be given to the circumstance that the pressureless jet of steam exercises a suction on the clearance space due to the very high velocity of the steam. In such cases air will be sucked up from without, and will pass into the working-wheel. Some appliance will have to be provided to prevent the entrance of the air, the admixture of which with the steam will result in the reduction of the work done by the latter. In reference to the clearance space packings of velocity turbines, the objections in regard to the extension of the materials at high temperatures indeed fall away. It will, however, in each particular case be a matter for consideration whether the anticipated leakage of steam or the sucking up of air is the greater evil.

Westinghouse (1897),[1] in the arrangement of a radial turbine with velocity stages and impingement from within, has endeavoured to lead the steam jet from the guide vane as completely as possible against the effective surface of the next-following guide vane, and not to allow a part of the steam to strike the clearance space betweeen the head of the vane and the guide-wheel body.[2] Meanwhile the lengths of the vanes increase in consequence from stage to stage in such a manner that their heads and feet formed stepped lines, and the head of one working-wheel vane projects over the foot of the preceding guide vane.

[1] A. P. 712,626.

[2] Compare Fig. 328, p. 183.

XVII

RELIEVING ARRANGEMENTS

AXIAL and also radial turbines which, by reason of the supply and discharge of the steam are exposed to a different pressure at each side, are subject to a thrust in the direction of the axis which, according to the purpose served by the turbine, is more or less neutralized, and must, at any rate, be altogether eliminated when only the rotary motion of the shaft of the turbine is to be made use of.

In the case of an engine with a vertical shaft, this one-sided axial pressure of the steam can be applied in counteracting the pressure on the pivot due to the weight of the turbine. To this end the steam is introduced into the working-wheel chamber from below (*Hoehl, Brakell,* and *Günther*[1]).

If the general conditions do not admit of an introduction of the steam in the manner described, a special means of relieving the pressure on the pivot must be found, such as the introduction below it of a pressure fluid, forming a cushion on which it rests. This expedient is now generally adopted. Reference may here be made to a pivot relief appliance based on this idea by *Mallyna*[2] (1898) (Fig. 534). To take the weight of the turbine shaft a a case filled with pressure fluid is arranged, consisting of three parts. The lid c rotates with the turbine-shaft, and the bottom g is fixed at the foot f of the case. The ring h_1 h_2, which is in two parts, acts as packing for the case, being set free to move up and down on the fixed part g. On the one hand it is pressed by the weights K, or by springs, against the rotating part c, while, on the other hand, an indiarubber or leather ring l fixed between h_1 and h_2 makes all tight between h_1 and the saucer-formed part g. Through the pipe m is introduced the pressure fluid, which in steam turbines is usually oil.

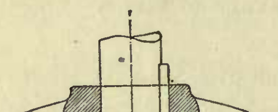


Fig. 534.

In the case of turbines with horizontal axes, the weight of the rotating masses is not available as a medium of relief. Relief can, however, easily be attained by a suitable lead of the steam. In connection with the old reaction wheels with axial supply of the steam, *Tetley*[3] causes the steam to enter at both ends of the hollow shaft, or he closes both its ends and provides it with radially arranged slits, through which

[1] E. P. 2429 of the year 1863. [2] D. R. P. 105,537.
[3] E. P. 1706 of the year 1854.

the steam gains admission to the shaft. *Teulon,*[1] on the other hand, diverts the axes of the steam outlet nozzles from the plane of rotation and sets them at such an angle to the latter that, in working, in addition to the back pressure tending to produce rotation, an axial pressure remains over, which, it is true, is here intended to counteract the thrust of the propeller.

A convenient expedient also consists in the dividing of the turbine into parts in such a manner that the steam has to stream through the one set of wheels in the one axial direction, and the other set in the opposite one. This method has been chosen by *Parsons*[2] (1884) (Fig. 535) in one of his older arrangements. On the turbine body *a*, which is in one piece with the shaft *s*, are shipped the rings *b*, divided

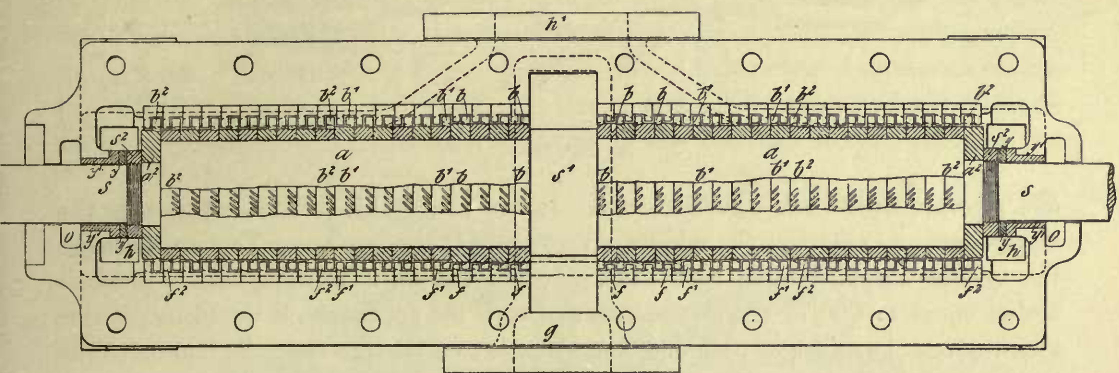


Fig. 535.

equally, as regards numbers, at the two sides of the ring projection s^1. Round the latter extends an annular chamber, which is fed by fresh steam through the steam-inlet *g*. The vanes b^1, b^2 on the rings *b* have only about half the breadth of the latter, so that the guide vanes f, f^1, f^2 find room between them, which is worked out of the rings fixed in the case. From the above-mentioned annular chamber fed from *g* the steam streams through the turbine in both the axial directions to the chambers *h*. From these latter, channels lead to the exhaust h^1. The increase of volume of the steam is allowed for by an increase in the radial vane breadths given in the direction of flow of the steam. The last working-wheel rings a^2, which serve as end pieces, are pressed by means of nuts s^2 against the body *a* and its rings b^1 b^2. The shaft *s* is packed in the end walls of the case by means of brass rings *y* and brass coupling-boxes y^1. Outside of these packings there are also the chambers *o*, from which any steam that may get through is drawn by an injector.

Parsons[3] (1887) has tried to improve the alteration given in Fig. 536. The turbine is mounted on a single shaft and its diameters are increased by steps in the direction of flow of the steam. The arrangement is so made that as the pressure decreases a greater peripheral velocity can be attained. There are several (*e.g.* three) pairs of working wheels *c, e, g,* so that the steam streaming in the one axial direction

[1] E. P. 706 of the year 1874.

[2] D. R. P. 33,066; *Proc. of the Instit. of Mech. Eng.*, 1888.

[3] D. R. P. 41,479.

exerts the same pressure as that streaming in the opposite direction. The outflow sides of each pair of working wheels are interconnected by means of circulating channels *d, f, h*. The object of this connection is the ensuring of an even steam pressure on the ring surfaces which are produced at the point of junction of two wheels of different diameters, and also to ensure those of the turbine shaft against an

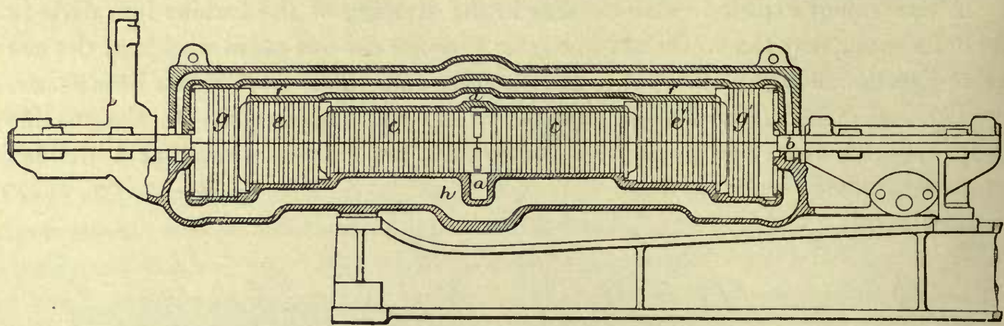


Fig. 536.

end pressure in the one direction only. In the case of higher steam pressures the motive medium enters at the middle of the engine at the point *a* and streams through the turbine stages towards both ends. If the engine is to work with a pressure almost equal to that of the atmosphere and with the condenser, it would be possible to introduce the steam at each end, and allow it to discharge from the middle. The turbine wheels then increase from the ends towards the middle. However simple and self-evident a relieving method of this kind may be, in practice it is of only partial advantage, because, by the division of the steam over two small sets of wheels, the working economy suffers.

The arrangement of *Schulz*[1] (1901) may be considered a very practical solution of the problem. In order to effect the total or partial neutralization of the axial pressure, an axial turbine is combined with a radial one, In Fig. 537 this arrangement is shown with a backwards-motion turbine on the left-hand side mounted on the same shaft with a larger forwards turbine (on the right). To take the forwards turbine first, it consists of the high-pressure turbine 2, and the low-pressure engine 1, through which the steam streams, one after the other, in contrary directions. The axial thrust is thus avoided. The Schulz speciality consists in the circumstance that the axial turbine body *a* and the radial turbine body *b* are so connected that both are

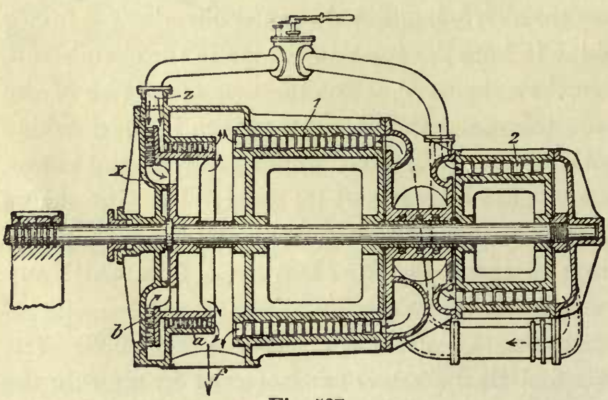


Fig. 537.

[1] D. R. P. 135,937.

subject to axial thrust on the part of the steam or other motive medium in one and the same direction, while the radial turbine body *b*, before its entrance into the axial turbine body *a*, experiences a thrust in the opposite direction at its hinder end, the force of which can, at will, be so regulated as to wholly or partially neutralize the thrust of the two turbine bodies first mentioned. For reversing turbines an engine of this kind for forward motion and one for backward motion may be arranged to work on a common shaft. The drum bearing the pressure vanes of the axial turbine body *a* is enclosed by a case containing the guide vanes, and on the end presented towards the case-lid *x* bears a ring flange (in the illustration pointing both inwards and outwards) projecting beyond the wall of the drum. On the side of the ring flange presented towards the lid *x* the pressure vanes of the radial turbine *b* are attached, while the inner side of the lid *x* bears the corresponding guide vanes. Encircling the ring flange is a suitable ring-shaped extension of the case to which the entrance branch *z* for the motive medium is attached. *f* is the discharge branch. The steam entering through the branch *z* streams through the

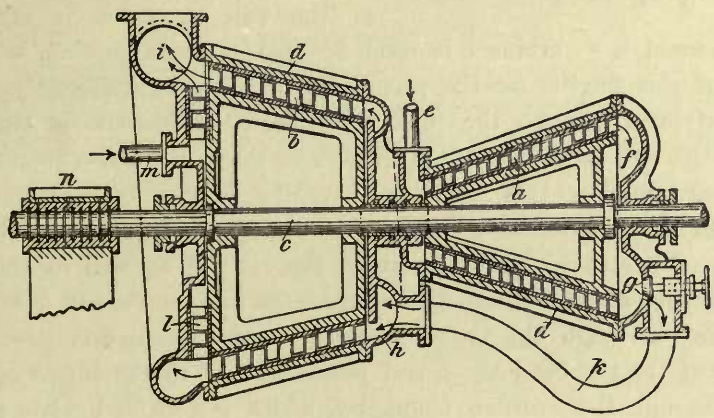


Fig. 538.

radial turbine body *b* from without, but does not reach the axial body *a* until it has acted upon the rearward side of the radial turbine disc bearing the vanes. It is not necessary that the two turbine bodies *a* and *b*, which act together, should do even approximately the same amount of work. For instance, the axial turbine body *a* can be arranged for the main work, while the radial body is, for small work, made of such dimensions that the steam acting from *b* to the left with full pressure on the back of the disc neutralizes the thrust forces acting in the opposite direction in the two turbines *a* and *b*, as far as may be desirable. In the relieving arrangement the size of the inner ring flange projecting over the wall of the drum of course exerts an influence.

In the design of *Schulz* it is of importance that the high and low pressure turbines be arranged, in relation to one another, in such a manner that their axial pressures act in opposite directions and wholly or partially neutralize each other.[1] In Fig. 538 a steam turbine with axially arranged wreaths is illustrated, which

[1] D. R. P. 137,792. E. P. 8378 of the year 1901.

in the body *a*, working with higher pressure has a smaller, and in the body *b*, working with lower pressure, a larger diameter. The two turbine bodies *a* and *b* are attached to a shaft *c* in the case *d*. The thrust-bearing of this shaft is denoted by *n*. The steam enters through a branch *e*, streams through the turbine section *a*, and leaves it by the overflow channel *f* to pass through the pipe *k*, which is provided with a governing valve *g*, into the entrance channel *h* of the body *b*. The latter is traversed in a direction opposite to that of the body *a* by steam of low pressure, which leaves the compound turbine by the exhaust *i*. By means of pressure vanes on the one end wall of the body *b*, and leading channels on the end wall of the case *d*, which is arranged as a discharge channel *i*, a turbine *l* is formed for backwards motion, which, in the application of this engine; for the propulsion of ships, is a necessity. The fresh steam here streams through the inlet pipe *m* to the smallest of the concentric turbine wreaths, and passes from the largest of these into the exhaust channel *i*. During forward motion this backwards turbine *i* runs free without injurious action in the reduced pressure of the outflowing steam. Fig. 539 shows the arrangement of a steam turbine in which the body *a* as well as the body *b* is provided with several discs fitted in the case *d* on the shaft *c*, and bearing wreaths on both sides. Through the entrance branch *e* the steam first proceeds to the smallest disc of the turbine body *a*, and passes onward to the larger disc. Thence it streams through the overflow channel *f*, which is provided with a governing valve *g*, and through the pipe *k* into the entrance channel *h* of the low pressure turbine body *b*, and leaves the latter through the outflow channel *i*. Here also a backwards motion turbine forms part of the arrangement. It would, of course, be feasible in like manner to combine more than two turbine bodies for the same purpose, and to make use of separate cases. In the particular case of a turbine employed as a marine engine, it will be of advantage so to chose the diameter of the turbine body of the compound engine that when the latter is in motion a pressure opposed to the axial one of the propeller acts in the direction of the shaft, so that the thrust-bearing *n* is strained as little as possible. This axial pressure is, in the application of the turbine alluded to, directed in the backwards direction, and its amount must be so chosen that it is approximately equal to the forward thrust of the propeller. The governing valve *g* in the overflow channel *f* is intended, if necessary, to detain a part of the steam issuing from the turbine body *a*, so that behind this latter the pressure of the steam increases, while the pressure of the steam that streams through the free opening of the valve to the turbine body *b* is reduced. By means of an adjustment of this governing valve *g* the resultant axial

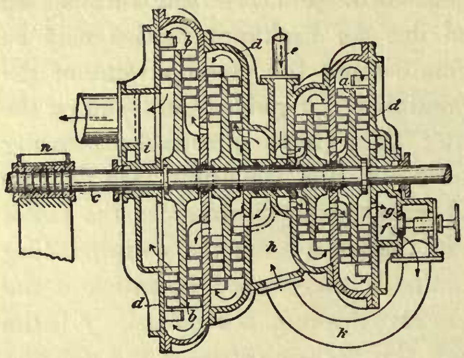


Fig. 539.

pressure of the shaft c on the thrust-bearing n can, within certain limits, be altered. Manometers attached to the turbine bodies enable the steam pressures to be read off at any time.

In the design of *Fullagar*[1] (Fig. 540) the working wheel a is provided on the steam discharge side with a counter-pressure disc b in order to relieve the pressure of the steam streaming in the axial direction through the turbine. Between this disc and the wall of the case a small space is left, which is divided by labyrinth packings d into concentric rings e. The pressure stages of the turbine are collected into groups of different diameters, and the annular chamber e is connected, by means of the longitudinal channels arranged symmetrically in the shaft, with the steam chamber in front of the group in question. This is done in such a manner that the innermost annular chamber c receives steam from the chamber in front of the

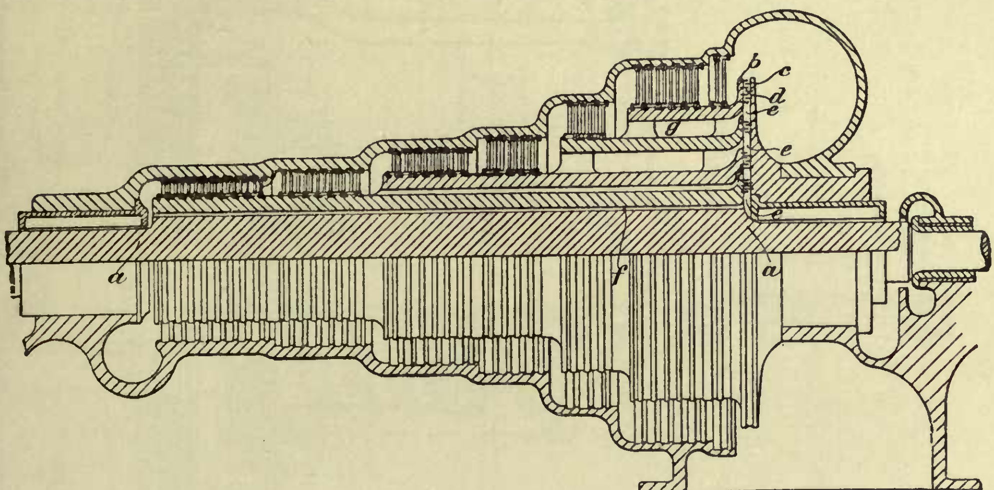


Fig. 540.

first group through the channel f, while the outermost annular chamber e, on the other hand, receives it from the chamber in front of the last group through the channel g. By this arrangement it clearly becomes possible to attain effectual relief without losses of any considerable importance being brought about at the packings, since one of the salient points of the labyrinth packing is the prevention of the passage of steam of a certain pressure into a neighbouring chamber in which there is a pressure only slightly less in degree. If the shaft—as assumed in Fig. 540—be made up of separate sleeves, the longitudinal channels for leading the steam are hollowed out of the inner surfaces of these.

For his multi-stage radial turbine[2] *Lindmark* (1902)[3] has, for the relief of the axial pressure which is produced by reason of the increase of the sectional areas of the central inlets for the motive medium, also made use of discs loaded by steam and arranged to act in opposition to the axial pressure. The relieving discs or

[1] E. P. 7184 of the year 1901. D. R. P. 152,259. S. P. 24,039.
[2] Compare p. 39. [3] D. R. P. 152,981.

parts of the naves of the turbine bodies, together with the fixed walls of the case, form chambers which are interconnected, and in which a mean between the high and low pressures is present. Further, for each turbine body requiring relief there is a chamber between two walls of the case which encircles the former. One of these walls abuts against a section of the nave, which has the same outer diameter as the central inlet for the corresponding turbine body. The other wall abuts against a section of the nave which has the same outer diameter as the central inlet for the next-following turbine body. In the case of the last of the turbine bodies, however, the above-named wall abuts against a portion of the nave, the outer diameter of which is equal to that of the central inlet for the first turbine body. The chambers

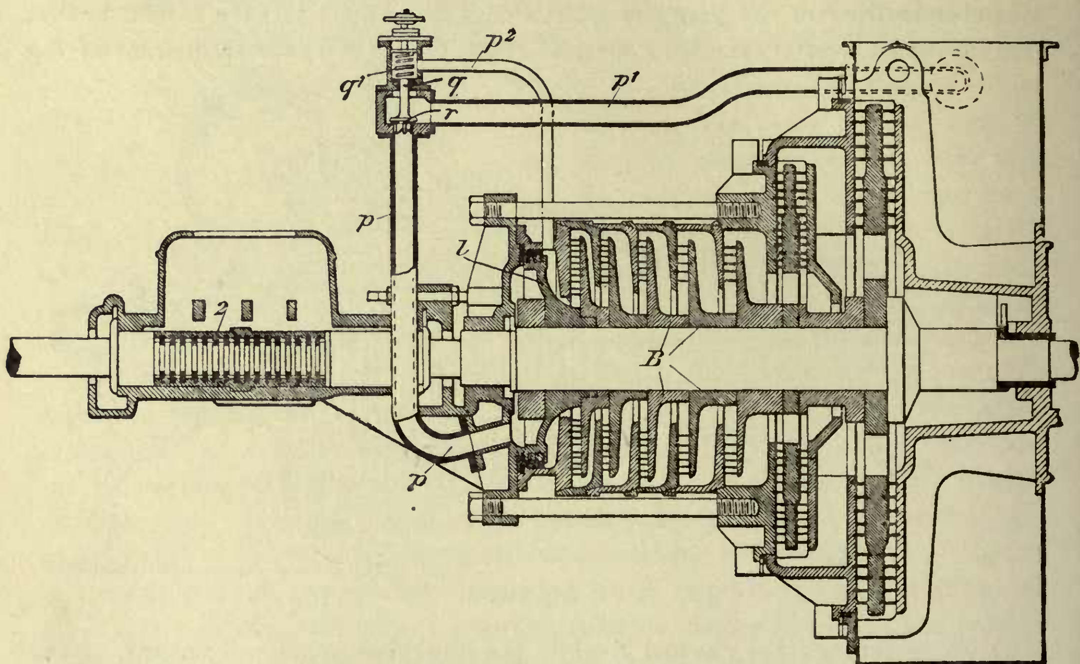


Fig. 541.

referred to are connected with one another by a channel. When in the case of marine engines the axial thrust of the working steam is only so far neutralized at the relief places that an amount equal to the back pressure of the ship propellers working on the turbine shaft remains over, the relieving appliance will work suitably when its relieving force varies with the thrust pressure of the propeller.

Parsons (1896) endeavoured to produce an automatic alteration of the relieving pressure by the arrangement shown in Fig. 541,[1] which is here fitted to a radial turbine, but may also be applied to an axial one. The turbine shaft is mounted by means of yielding rings (2)[2] in a collar bearing, and is to a small extent free to move longitudinally. The relieving piston *l*, together with the working wheels *B*, is fixed firmly on the shaft and moves with the latter in the axial direction. It is

[1] A. P. 553,932. [2] Compare p. 302.

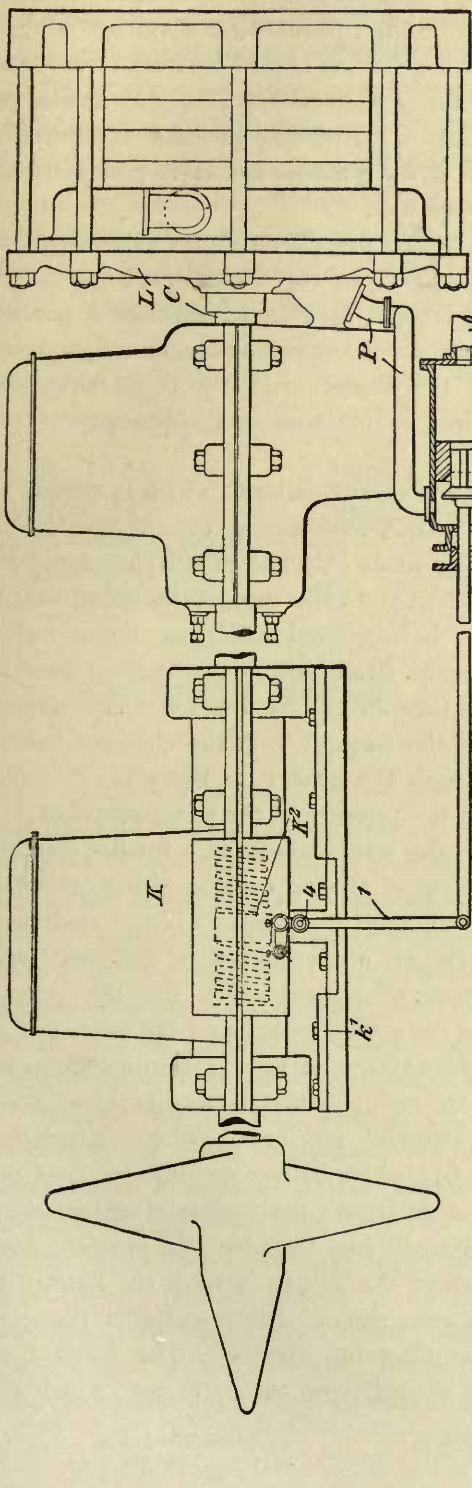


Fig. 542.

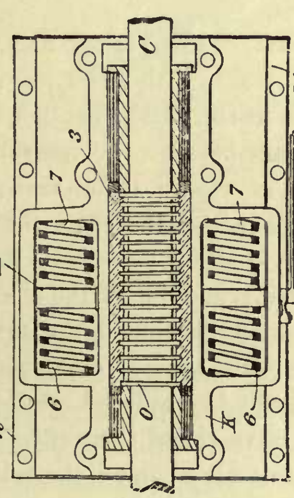


Fig. 543.

influenced by the working medium from the side of the fresh-steam chamber. The rearward chamber, bounded by the piston and by the cover of the case, is connected by means of the pipes p, p^1 with the exhaust of the turbine when a valve r is open. The rod q of the latter, which is formed as a piston, is loaded by fresh steam from the high-pressure chamber of the turbine, entering through the pipe p^2; the amount of this load can be altered by hand by means of the spring q^1, which is adjustable from without. The effective surfaces of the piston q and of the valve r respectively must of course be in suitable relation to one another.

The alteration of the pressure of the screw propeller can be so arranged as automatically to cause a corresponding alteration in the counter-pressure of the relieving appliance. The propeller shaft is then capable of making a limited horizontal movement and releasing a governor. An unavoidable accompaniment of such an arrangement, however, is that the alteration of the counter-pressure of the relieving appliance lags more or less behind that of the pressure of the propeller.

Parsons[1] (1895) (Figs. 542 and 543) mounts the propeller C, which is turned by the turbine L, with the fixed rings O in a box 3, the bearing case K of which is movable in the axial direction on the bearing stool k^1. The case K has arms K^2, which pass between the pressure springs 6, 7 fixed to the case; these latter enable the use of elastic collar-bearing rings[2] to be dispensed with; in the case of a displacement of the shaft they act as a buffer. Coupled to the bearing case K, however, is also a two-armed lever 1 free to turn on the fixed pin 4. This adjusts the valve c, and varies the clear opening of the passage from the chamber behind the relieving piston in the turbine L^3 through the pipe P to the pipe P′ which leads into the air. In case of an increase in the pressure of the screw propeller, the shaft C will be displaced somewhat to the right, which results in a further opening of the valve c, and therewith in a setting clear of the air cushion which otherwise comes into play behind the relieving piston in the turbine case. If the shaft acts so as to close the valve c, the pressure of the air cushion must be deducted from that of the relieving steam on the relief piston. This means, then, that the decreasing pressure of the propeller also diminishes the actual pressure on the relief piston.

It is also possible, however, to effect the adjustment of a governing organ by means of the axial movement of the shaft, the result of this being the introduction of additional pressure to the back of the relieving piston. Another construction adopted by *Parsons*[4] (1896) is shown in Fig. 544, which is in so far different from the foregoing that the relieving piston W is not built into the turbine chamber, but is keyed on the turbine shaft without it and made free to move in a separate fixed cylinder W^1. The movable bearing also swings the double lever 4, the lengths of the arms of which are so dimensioned that even a small displacement of the shaft produces a considerable movement of the governing slide 8. The latter then admits the pressure medium to that side of the relieving piston towards which the

[1] D. R. P. 98,493. E. P. 394 of the year 1894.
[2] Compare p. 302.
[3] Compare p. 292.
[4] A. P. 553,932.

displacement of the shaft caused by the altered pressure of the propeller has taken place. This arrangement is thus applicable to motion either ahead or astern.

For the relief of pressure, again, *Le Sueur*[1] (1894) (Fig. 545) makes use of a source of power which is independent of the steam. The end *a* of the shaft is bored through in the axial direction, and in the boring is set a stuffing box *b*, which with its flange *c* is fixed in the cover *d* of the case. Within the box *c*, in which the end of the shaft must then turn, is a semi-solid mass of graphite lubricating material, which is kept under suitable pressure by a spindle *e* screwed into the cover *d*. Round the end of the shaft *a*, however, the nave of a disc *f* is firmly fixed, and it fits closely into a boring *g* made in the cover *d*. In the space left between the disc *f* and the cover *d* a cold fluid (water) is introduced through the pipe *h*. The relief of the

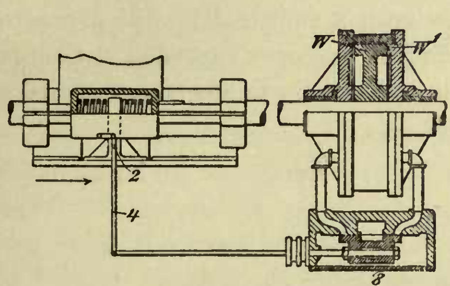


Fig. 544.

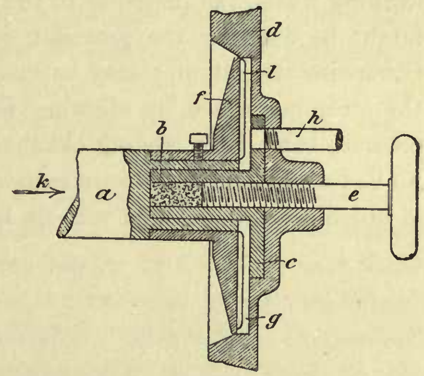


Fig. 545.

shaft from steam pressure in the direction *k* can now take place in such manner that the fluid admitted under a certain pressure bears against the disc *f*. Otherwise this fluid is introduced without pressure: in this case it accompanies the rotation of the radially arranged ribs *l* of the disc *f*, so that a pressure of the fluid corresponding with the speed of rotation of the shaft is occasioned by the centrifugal forces. In each of these cases, when the neutralizing axial steam pressure is exceeded by the counter-pressure of the (cooling) fluid on the disc *f*, both the latter and the shaft are so displaced that fluid can escape between the disc and the cover of the case and a diminution of the counter-pressure be produced.

Hitherto the relieving arrangement was made to depend on an axial thrust exerted by the steam in a direction coinciding with that of the general flow of the latter. The occurrence of this thrust depends upon the existence of an excess pressure on the inflow side over that on the outflow side of the working-wheel channel of an axial turbine, as is the case in the pressure (reaction) turbine. If the pure velocity (action) turbine be adopted, in which the steam is intended to flow through the working wheel without difference of pressure, the above-named excess pressure, and with it the axial thrust, will fall away. The expansion nozzles of a velocity turbine may now be assumed to be arranged in a disc which coincides with

[1] A. P. 545,238.

the plane through the ends of the mouthpieces of the nozzles and covers the working wheel, so that only a clearance space remains between it and the latter. Then the jets of steam streaming at high velocity from the nozzles will exercise a certain amount of suction on the clearance space and tend to produce a reduced pressure in the space between the disc (leading apparatus) and the working wheel. In this case then, *e.g.* in velocity turbines having pressure stages the working wheels belonging to which are placed in separate chambers, an axial thrust will result, which is opposed to the direction of flow of the steam. When the nozzles are collected together in loose groups, this phenomenon is of course not to be apprehended. As soon, however, as the impingement channels form portions of complete disc bodies, its occurrence becomes possible. In order to equalize the pressures existing before and behind the working wheel, the pressures in the chambers would have to be made alike, and this might be done by the provision in the wheel bodies of sufficiently large holes. Otherwise the attempt may be made to prevent the suction of the jets in passing the clearance space by allowing the steam to keep a sufficiently high degree of pressure behind the leading channels. According to the views expressed in Chapter XIV., this residue of pressure remains behind in turbines the impingement apparatus of which are not provided with de Laval nozzles.

XVIII

BEARINGS

In regard to bearings, the difficulties experienced in the attempt to make the centre of the working wheel coincide with the axis of weight pointed to a device which consisted in mounting the rigid shaft in such a manner that its axis could, during the rotation of the turbine, adjust itself in the direction of the natural axis of the system.

A vertical shaft can be hung in a ball bearing so that it is free to swing (*Laidlaw* and *Matthey*[1] (1896)).

Trenta[2] also (1896) endeavours, by means of a swinging suspension of the shaft, to obtain an easy adjustment of the rotating masses and thus to avoid concussions and shocks. At its lower end the shaft is completely free, and it can yield slightly throughout its length. At its upper end it terminates in a strengthened head which is worked into a bell F and a spur wheel H. The latter engages into the toothed wheels J, J′, which are mounted symmetrically to it. These wheels bear wreaths M, M′ with exactly similar conical surfaces on the edges of which the

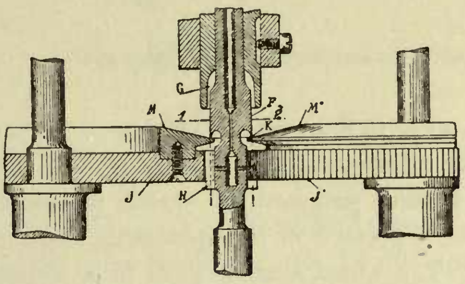


Fig. 546.

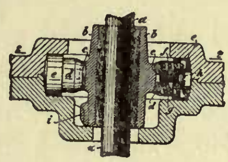


Fig. 547.

bell F rests. The diameter of the bell is the same as that of the sector of the spur wheel H. The weight of the turbine with its shaft is borne by the two wreaths M, M′ at the points at which they touch the edge of the bell, so that only rolling friction comes in question. The upper part of the shaft runs free in the bearing G.

Whitcher and *Roberts*,[3] indeed, let the shaft run in packed bearings. The latter, however, are aligned by means of radially placed springs, *i.e.* they are themselves able to yield in the bearing. In this the arrangement is such that the steam leading nozzles can adjust themselves to the working wheel; to this end the steam supply pipe is also enabled to yield.

[1] A. P. 592,147. [2] D. R. P. 91,006. [3] E. P. 2815 of the year 1900.

In similar manner *Krank*[1] (1901) guides the shaft bearing by means of radially set springs (Fig. 547). His arrangement, however, is characterized by the circumstance that the amount of spring in the direction of the centre of the shaft is so limited that the latter is not pressed beyond the central position by it. To this end suitably formed buffer cases *d*, which preserve the bearing sleeve *b* and the shaft *a* in the axis of rotation, are held in the caps *e* by the springs *f*. Adjustable bolts *g* with their heads *h* bearing against the cap *e* are screwed into the buffer cases *d*. These latter press against spherically shaped ring projections *c* of the bearing sleeve *b*, which is prevented from turning by means of a projecting piece *i*.

In the case of horizontal shafts the one-sided pressure exerted by the weight of the rotating masses comes in to add to the difficulty, but in this case also yielding bearings are made use of.

In the arrangement made by *Dow*[2] the end of the shaft is borne in a stuffing box which is provided with a ring at about the middle of its length. The latter lies in

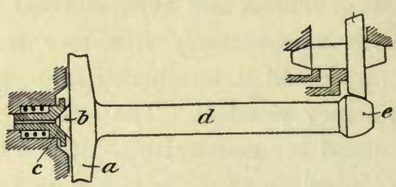


Fig. 548.

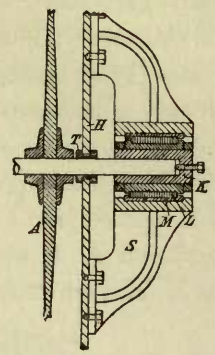


Fig. 549.

a case the walls of which are cut out so as to form tongued strings. The condensed water of the turbine is made use of for lubrication.

Nordenfeldt and *Christophe*[3] (1894) have proposed the system of bearing shown in Fig. 548. The wheel *a* runs with its point *b* in a displaceable bearing *c*, which is pressed against it by a spring. The wheel shaft *d* ends in a conical arched piston *e*, which turns between the wheels *f*. This system of bearing allows the wheel, while rotating, to alter its position to such an extent that it is always in equilibrium.

On the other hand, *Krank*[4] (1902) holds each shaft projection by means of a ring made spherical at its outer side, which does not rotate, and which lies between spring buffers. Meanwhile the springing of these buffers is limited in the direction of the axis of rotation, so that the buffer cases lying behind it during the rotation of the shaft, are not able to increase the wavering of the latter which naturally takes place.

[1] D. R. P. 132,549. (Compare Chap. XXII.)
[2] E. P. 16,072 of the year 1888.
[3] D. R. P. 84,853.
[4] S. P. 25,914.

Maardt[1] (1895) proposes to make the engine run smoothly by inserting a thick gutta-percha belt L between the sleeve K surrounding the shaft, and a second sleeve M in the bearing case. It is said, however, that the belt may be replaced by four spiral springs V (Fig. 550), the pressure of which may be regulated by plugs X, which can be screwed in or out. According to the arrangement in Fig. 551, rings R are cast on both side walls of the turbine chamber with the object of keeping the steam from the shaft. The working-wheel disc is intended to act as a ventilator and to draw air through the bearing in order to cool it.

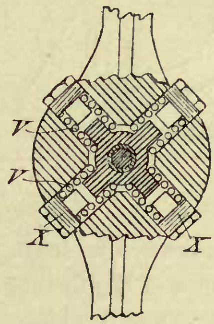


Fig. 550.

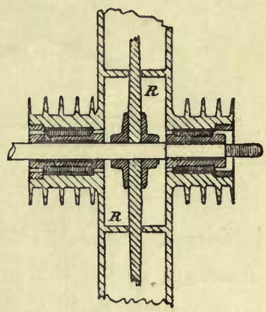


Fig. 551.

Such makeshifts, however, must be applied with caution, since a turbine is of value for practical purposes only when the energy developed by it can be passed on, and for this the fixed points of support are awanting. The measure adopted by *de Laval* of mounting the shaft in fixed bearings, and then allowing it in itself to yield, must therefore be regarded as a step in advance, which rendered his turbine a practically useful engine.[2]

The highest degree of perfection that is here possible has probably been attained by *de Laval*. Fig. 552 shows a turbine spring-shaft 4 with vane wheel 5

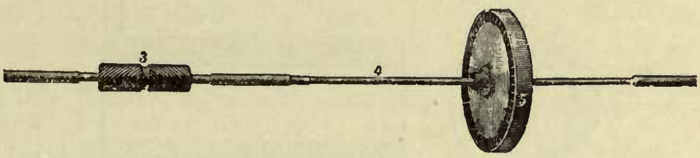


Fig. 552.

and pinion 3. The yield of the shaft, however, diminishes with increase of size, and from a certain point onwards it will, for the attainment of greater capacity for work, be necessary to take refuge in a higher speed of rotation of the working wheels—an expedient which is not desirable. But the increase of the distance between the bearings in order to lengthen the freely borne shaft and enable it to yield when larger in diameter becomes impracticable beyond a certain limit which has already been reached by de Laval. It appears, in fact, that the Laval turbine of 350 H.P., as exhibited in 1900 in Paris in the steam dynamo group of the exhibition, represents the higher limit of capacity of this method of construction.

The system of construction for large steam turbines must make the best of rigid shafts, and it is able to do this as soon as special care is bestowed on the question of

[1] D. R. P. 87,519.

[2] In regard to de Laval's shaft, compare *Civilingenieur*, Vol. 41, pp. 333 and 519, and Vol. 42, p. 249; also *Revue industrielle des Mines*, Vol. 33, p. 141.

balancing, and because the considerable diminution of the number of revolutions per minute does much to lessen the difficulties of the case.

De Ferranti[1] (Fig. 553) gives the shaft *a* with its stuffing box a long bearing in the case *b*. The nave *c* of the working wheel is shipped on the conical end *d* of the shaft; it is pressed against the bearing by the nut *e*, while on the other side of the former the ring-projection *f* of the shaft offers a support in the other direction.

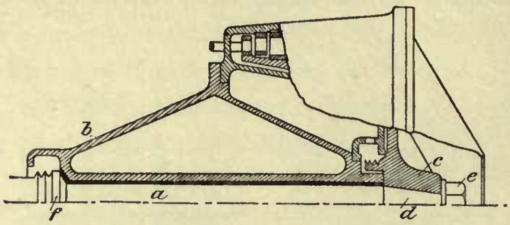
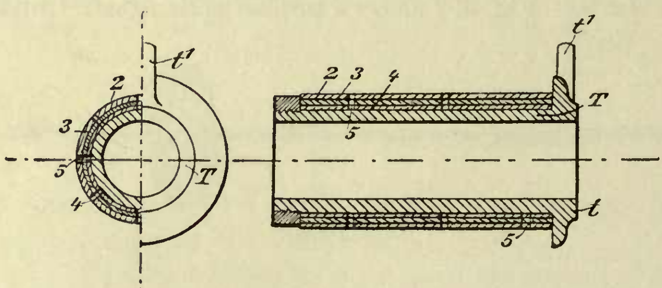


Fig. 553.

It will in any case be necessary to give the shaft a certain amount of play in the radial direction in its bearings. *Parsons*[2] surrounds the shaft with a sleeve T (Figs. 554 and 555) by means of which the pressure of the former is transmitted to the bearing. The sleeve is provided with a head flange *t*, which, in order to prevent its turning with the shaft, is provided with a projection t^1. Over the sleeve T are shipped three tubes 2, 3, and 4, between which small amounts of play are left; into these oil is pressed through the holes 5, and this offers a considerable hydraulic resistance to vibrations of the shaft. The middle tube 3 can also be replaced by three longitudinal strips of plate which have a somewhat smaller radius than the tubes 2 and 4, so that they exercise a spring pressure in the radial direction.



Figs. 554, 555.

Again, the danger of the heating of the turbine shaft can of course be diminished by the expedient of giving different speeds of motion to the friction surfaces. If, for instance, a vertical shaft rest with a ring projection on the bearing plate of the case, loose friction plates that are free to turn are inserted between these so as to encircle the shaft. These are carried round by the revolving shaft by means of the friction at the ring projection, while on the other hand the bearing plate tends to hold them back. The revolutions of the friction plates thus grow gradually quicker from the bearing point to the shaft projection. In order, now, to provide a certain regularity in the rotations of these inserted friction plates 4, *Hedlund*[3]

[1] E. P. 2565 of the year 1895.
[2] E. P. 1120 of the year 1890. E. P. 14,944 of the year 1890. D. R. P. 98,493.
[3] D. R. P. 152,475.

(1903) (Fig. 556) gives these latter and the ring projection 3 of the shaft the forms of toothed wheels. In order to compel the plates to preserve a certain speed of rotation, the toothed wheels engage into a suitable number of other toothed wheels 6 fixed to a shaft 5 mounted in the case. The proportion in which the toothed wheels are geared to one another is so chosen that the relative motion between two neighbouring plates is small, and that it is approximately the same between all the plates.

A subsequent readjustment of the bearings will in most cases be unavoidable. It will in part be rendered necessary by the wear of the bearing surfaces themselves in the course of working, while the possibility of giving the working wheels a certain amount of shift with reference to the leading apparatus after the erection of the turbine, is desirable.

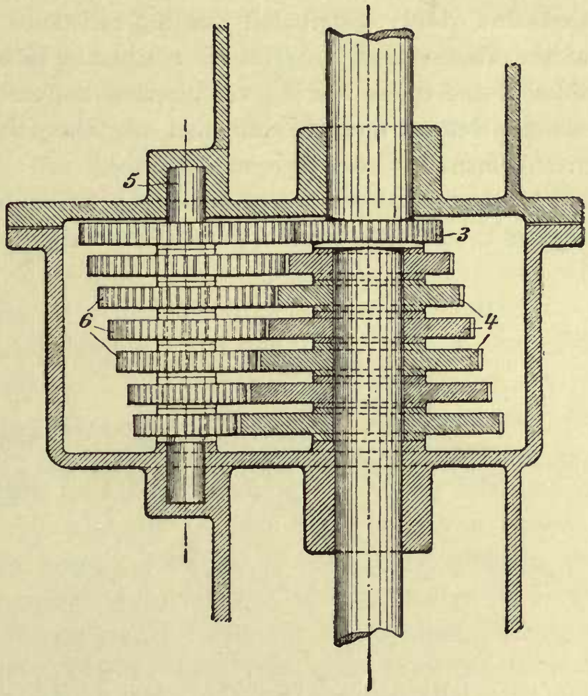


Fig. 556.

If a vertical shaft with pivot work in a socket, it may of course be made displaceable in its case (*Girard*[1]). Even in small turbines it will be disadvantageous to allow the pivot of the vertical shaft to rest on a horizontally placed cone, which is displaceable in the direction of its own axis and thus offers to the pivot a higher or lower lying bearing surface (properly speaking a point) (*Lolmann*[2]).

In the *Parsons* axial turbine we find the shaft resting with rings in a collar bearing which is divided horizontally at the height of the axis. While one of the bearing brasses is fixed, the other can be displaced in the axial direction till the shaft rings are clasped by both.[3]

For screw propeller shafts *Parsons*[4] replaces the usual collar bearing by an elastic one, the object of which arrangement is the neutralizing and preventing of the concussions which occur in working and which set up longitudinal strains at the point of support. In this, help is given by the relieving piston of the turbine itself (Fig. 557). The shaft C which bears the screw S is guided in a bearing seat *k*,

[1] E. P. 30 of the year 1855.
[2] E. P. 16,635 of the year 1897.
[3] E. P. 1120 and 14,944 of the year 1890. Compare also Chap. XXI.
[4] E. P. 394 of the year 1894. D. R. P. 98,493.

which is made in two parts. In this the bearing brasses k^1, which are also in two parts, are fixed, and their ring-grooves are taken by the disc-rings 2, which are provided with radial slits, and are therefore within certain limits pliable. The disc-rings fit into ring-grooves C^2 of the shaft, which allow the rings 2 a certain amount of play. On account of this during the forward thrust of the shaft in the vessel, occasioned by the pressure of the screw S, the ring-groove surfaces of the shaft find evenly distributed yielding resistance against the corresponding surfaces of the elastic discs 2. The oil receptacle K is open at its lower side. At both sides of the collar bearing are inserted the casings T, which are each made up of several sleeves, and are intended to take the radially directed shocks. Their arrangement has already been described.[1]

At the second position the shaft C is mounted in the bracket bearer Q, which adjoins the turbine case. It comprises casing q^1, which is in two parts, and which

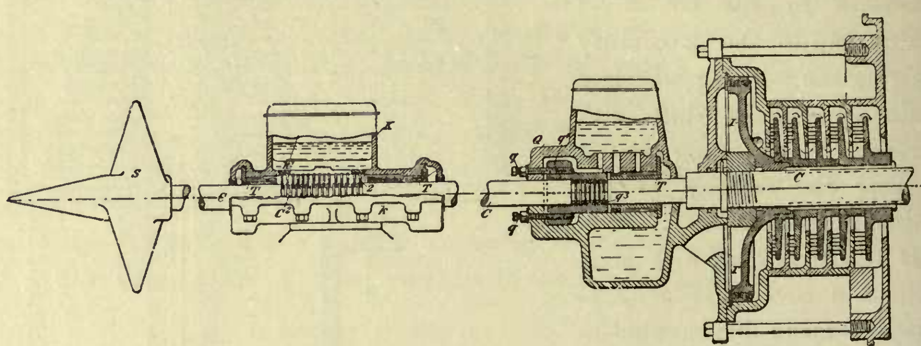


Fig. 557.

can be axially displaced by the screw q. Its rings q^3 engage in ring-grooves of the shaft C in such a manner that, while a backward motion of the shaft is prevented, a slight movement in the forward direction is rendered possible. This bearing also has a casing T for the purpose above alluded to, and a receptacle for oil.

For driving the shaft there is a multi-stage radial turbine, which, however, might be replaced by an axial one. The relieving piston J is made tight by labyrinth packing against the wall of the turbine case. On its after side rests the pressure of the atmosphere, and on its fore side the steam pressure, which now has to balance not only the forward thrust of the working steam, but also that of the screw S. In case of an additional concussive action of the screw S, the yielding disc rings of the collar bearing come into action.[2]

For the diminution of the *journal friction* in the bearings the well-known engineering expedients are of course available. Further, the uniform torsional moment possessed by turbines facilitates the employment of these.

In regard to the *lubrication* of the bearings of quickly revolving shafts, the simple leading of the medium to its work is not sufficient. It is necessary to use

[1] Compare p. 300. [2] Compare also descriptions of relieving appliances.

pressure-oil, which may most advantageously be produced by the turbine itself. Thus, for instance, *Girard*[1] leads oil, which has been acted upon by the pressure of the working steam, from a separate receptacle to the bearings. In working, then, the motive medium presses the oil into the bearings.

It is also customary to allow the heel pivot of a large turbine with vertical shafts to run in a thin film of oil which is pressed between the pivot and the bearing plate.

Lubricating appliances, especially for the inner-lying bearings, in which separate oil-pumps produce a closed oil-circuit, must be so disposed that the entrance of the medium into the turbine case is prevented, whether the engine is at work or at rest. For this reason the *Société Sautter, Harlé & Cie.*[2] (1901) insert in the return lead between each bearing and the main oil holder an intermediate receptacle which, by means of a pipe, stands in connection with the inside of the turbine case behind the respective bearing, or with the condenser, and is either closed against the main oil holder by a cock which serves to let off the oil, or stands in connection with a pump which draws the latter off. A simplified arrangement of the appliance is shown in Fig. 558.[3] The simplification consists in the circumstance that the bearings *g*, *h*, *i*, which stand under pressure, are lubricated one after another by means of a single circulating pump through a pipe which goes from bearing to bearing and is provided with only one intermediate receptacle. The oil is led through a pump *n*, which may be driven as found convenient, and pipes 1, 2, to the bearing *g* subject to the highest pressure, and instead of being led back from the point of discharge of this latter directly to the intermediate receptacle *p*, passes through pipe 3 into the intermediate bearing *h*, in which the pressure is the least. The oil flows, then, only from the point of discharge *h*, through pipes 8, 9, back to the intermediate receptacle *p*. In the position shown in the figure the receptacle *p* on the one hand stands in connection with the return lead from the bearing *h* through 8 and 9, and on the other it is kept under condenser pressure by means of the lead 19 and the cock *r*. When the intermediate receptacle *p* is filled, a turn of the handle *z* suffices to adjust the exactly similar cocks *q*, *r*, *s* in such a manner that the oil passes from the receptacle *p* back into the main holder *m*. In this adjustment the cocks *q*, *r* are closed and the cock *s* is opened. The air-pressure is produced in the vessel *p*, and the oil runs, by reason of the difference in level, through pipe 5 back to the holder *m*. The non-return valve *o* is intended to prevent the oil from the receptacle *p* from being sucked through 9 and 8 into the turbine when the latter comes to rest and condenser pressure is present from one end of it to the other. The oiling of the bearing *i* is effected in a somewhat different manner from that of the bearings *g* and *h*. In the first place the end of this bearing is in a chamber the pressure in which is approximately that of the condenser, which is always below the pressure of the atmosphere. The lead 13, 14, for the oil in which the oil-pump N is inserted, is arranged in the same manner as that for the two other bearings, but

[1] E. P. 30 of the year 1855. [2] D. R. P. 125,115. [3] D. R. P. 131,155.

the abstraction of the oil from the bearing is effected from both ends of the latter. The pressure of the oil as it enters the bearing *i* is greater than the atmospheric pressure. The oil, then, resists the entrance of air through the end of the bearing *i*, by forming a fluid air-stopper. If the oil has made its way to the outer end of the bearing, it passes out through the pipe 15 and flows directly to the holder *m*. The greater part of the oil disperses itself in the bearing up to its inner end. It would, then, be apt to get into the turbine case, since the pressure in the chamber into which the end of the bearing projects is less than that of the atmosphere. 16 is the return lead into the receptacle *O*, in which a lower pressure is maintained

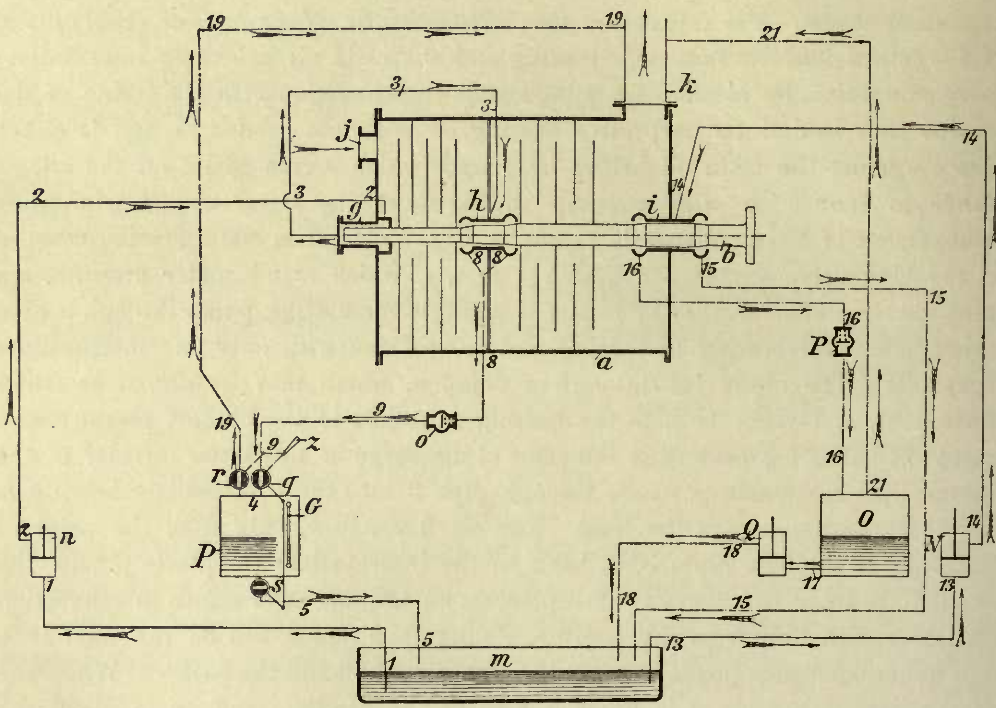


Fig. 558.

than that of the atmosphere by means of the pump Q, which draws off the back-flowing oil. The oil collects and is led back by the pump and by the pipe 18 into the holder *m*. To ensure the return of the oil from the end of the bearing *i* into the receptacle O, the upper part of the latter is, by means of lead 21, placed in connection with a point of the discharge pipe leading to the condenser; since this point lies nearer to the condenser than the bearing *i*, the pressure at this point is slightly lower than in the chamber which directly encloses the bearing, and in consequence the oil has the tendency to flow off through pipe 16, but not to spread itself through the engine.

Meanwhile the press-oil necessary for the lubrication of a bearing can be produced in the bearing itself by means of the centrifugal action of a fly-wheel.

This the *Gesellschaft zur Einführung von Erfindungen m. b. H.* (1903)[1] intend to do. With this view they form the bearing plate *c* (Fig. 559) as a centrifugal pump. The disc is provided with vanes *d*, which draw in the oil from the suction chamber *e* through the holes *h*, and place the chamber *f* under a certain degree of pressure. From the collecting channel *g* the oil can be led through a cooling-coil *i* before it passes into the chamber *e* again.

Worthy of remark was the lubricating appliance (apparently not found practical) of the original *Parsons* turbines,[2] which, in spite of the parting of the current of steam into opposite axial directions, ran at from 6500 to 18,000 revolutions per minute. The oil serving for the lubrication and cooling of the bearing was there drawn by means of an exhauster mounted on the turbine shaft from a

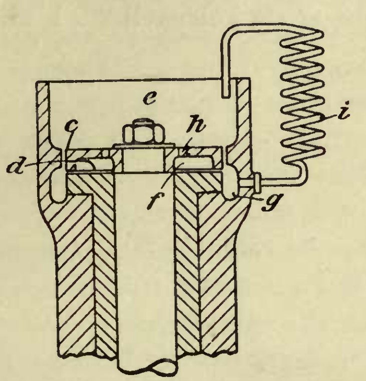


Fig. 559.

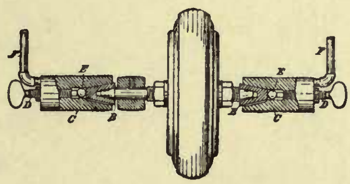


Fig. 560.

collecting receptacle below into a stand pipe, so that it could flow with a certain amount of pressure to a screw which formed the continuation of the shaft. This screw then pressed the lubricating medium into the bearing from which it flowed back into the collecting receptacle.

According to the plan adopted by *Woods*[3] (1898) (Fig. 560) bearing cones B are mounted on the ends of the shaft, and sit in corresponding conically formed seats of the bearing body C. The latter, again, are provided with press-screws D, which, by means of the inserted balls E, exert a powerful longitudinal pressure on the shaft, with the object of relieving the journal surfaces of the bearing cones as much as possible. The lubricating medium is pressed into the bearing through the pipe F.

When the conditions of working of the engine otherwise admit of it, the idea of the replacement of the friction of gliding by that of rolling surfaces is not to be summarily dismissed. In particular the conditions have here received attention in which the driving and driven shafts respectively are not directly coupled, so that some medium of transmission becomes necessary.

An arrangement of *Ericsson's*[4] may here be recalled, who mounts the working

[1] D. R. P. 150,746.
[2] E. P. 6734 of the year 1884. *Proc. of the Instit. of Mech. Eng.*, 1888.
[3] D. R. P. 105,153.
[4] E. P. 5961 of the year 1830.

wheel on discs, which, in order to cool the former, are made to run in water or oil. Resting on one end of the shaft there is also a friction roller, which takes up work from the turbine shaft.

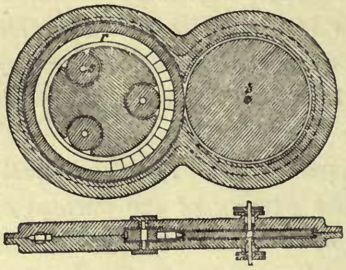


Fig. 561.

An older style of construction, which does not admit of practical application and which is given by Fig. 561, may be mentioned in passing. It was proposed by *Farcot* and *Perrigault*,[1] and its characteristic is that the working wheel is formed of only one ring *r*, which works on three guide-rollers *s*. The ring *r* engages into a toothed wheel *t*, which transmits the motion; or else one of the guide-rollers *s* is made use of as a driving wheel.

[1] E. P. 1206 of the year 1866.

XIX

TRANSMISSION OF POWER

For quick-running turbines, friction wheels for the transmission of power have for obvious reasons often been under consideration. Thus *de Laval*[1] (1883) (Fig. 562) has arranged the reaction tubes *b* between the discs *a*, the outer ends of the former being bent over tangentially and their inner ones axially. The wheel is mounted so that its shaft *c* and the friction driving wheel *d* are axially displaceable, and so that the steam entering at *e* also causes the driving wheel *d* to press against the large friction wheel *f*.

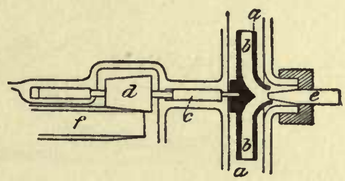


Fig. 562.

If slipping is not to take place, friction wheels require considerable pressure, which again places a heavy load on the bearings. In order to remove this drawback, *Riegel*[2] has placed the driving disc on the turbine shaft, which is horizontal, and made capable of yielding. The driven disc is then formed as an electro-magnet, so that it holds the first-mentioned one against itself.

Nordenfeldt and *Christophe*[3] (1894) made use of friction wheels to guide the motion of two turbine wheels rotating in the same direction at different speeds on a common shaft.

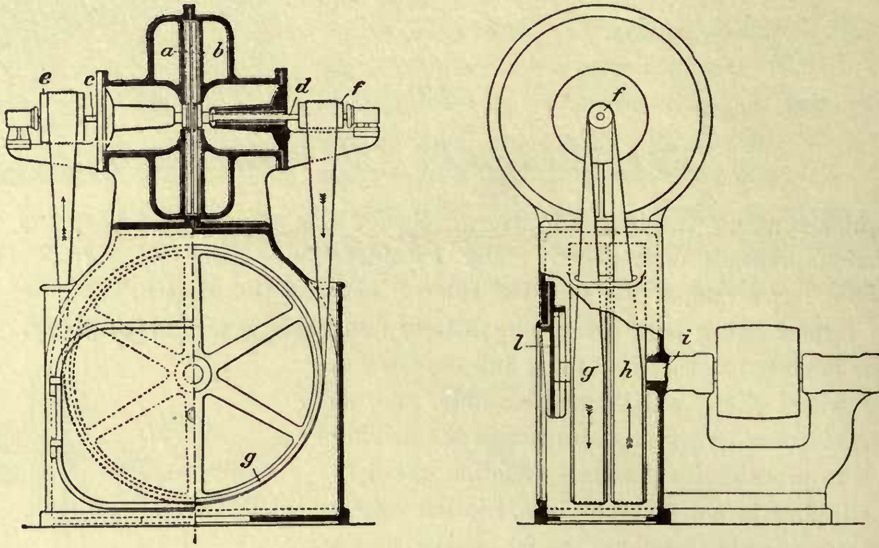
For the transmission of the work taken up by two turbine wheels *a, b*, rotating in opposite directions on a common shaft *i*, *Seger*[4] (1897) (Figs. 563 and 564) has adopted the following arrangement: The two turbine wheels are fixed on the shafts *c, d*, lying at opposite sides of the engine in line with one another, each of which bears a belt wheel *e, f*. The diameters of the latter are made to suit the different relative speeds of the turbine wheels. These wheels *e, f* have a belt in common, which on its path between the belt wheels runs on the one side over a disc *g*, and on the other over another belt wheel *h*, mounted on the before-mentioned shaft *i*. The free disc *g*, which is mounted on the displaceable slide *l*, here serves to stretch the belt.

Thiele[5] (1903) connects a relieving appliance (Figs. 565 and 566) with the apparatus for the transmission of power. The shaft *g*, driven by means of a belt *f* or similar appliance from the quick-running driving shaft *b*, is to be mounted hanging, and influenced in such a manner by, say a weighted lever *h l*, that it

[1] D. R. P. 24,346. [2] E. P. 17,182 of the year 1900. [3] D. R. P. 84,853.
[4] D. R. P. 100,797. [5] D. R. P 146,891.

endeavours by means of the said belting to remove the driving shaft from the bearing surfaces.

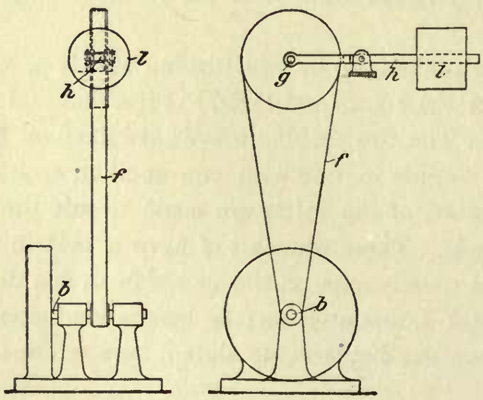
If in order to attain a free self-adjustment the turbine shaft be mounted only in



Figs. 563, 564.

a ball bearing, or in portions of such, the middle point of the appliance which transmits the power can, according to *Hörenz*[1] (1902), be arranged like that of a belting disc in the middle or turning point of the bearing ball.

Reference may here be made, in passing, to the peculiar proposal made by



Figs. 565, 566.

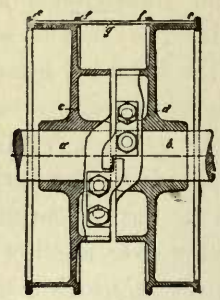


Fig. 567.

Vojácek[2] that two shafts be coupled together in such a manner that discs which rotate with a certain amount of play in opposite directions are mounted on both of them. Between the two discs a jet of compressed water is directed, which plays against both disc wreaths.

[1] D. R. P. 153,373.

[2] D. R. P. 92,372.

The great importance to which the steam turbine as now built has attained, is due, however, in no small degree to the circumstance that—thanks to its reduced speed of rotation—it does not require special means of transmission of power; that, on the contrary, the turbine shaft is enabled directly to drive the shaft of the working engine as well. It is the custom, then, to mount the power and working engines on a common shaft. If special circumstances require the use of separate pieces of shafting, these can generally be directly coupled together. In such case it appears advisable that the coupling be so made as to yield somewhat, and thus allow of the adjustment of the rigid pieces of shafting in their bearings so as to suit the positions in which the axes of their weights may lie.

Rateau and the *Société Sautter, Harlé & Cie.*[1] (1903) (Fig. 567) construct a spring coupling of this kind in such a manner that discs *c*, *d* are mounted on the shafts *a*, *b*. The projecting edges *e*, *f* of these discs are provided with holes, into which the pins *g* are inserted.

[1] D. R. P. 150,005 and 150,890.

XX

STUFFING BOXES

Where the shaft passes through the turbine case, and also in cases in which chambers situated inside the latter have to be packed against each other, stuffing boxes are used, which keep tight, entail only small losses by friction, and while allowing sufficient play, promote coolness rather than heat at the surfaces of contact.

Ulffers[1] (1883) has proposed the following stuffing box, especially for vertical shafts: thick discs, consisting of parchment and metal, are arranged in alternate layers one above another; the latter are provided with radial bores. The wall of the stuffing box, which is enclosed in a cover, is provided with uniform bores and vertical channels. The mantle space contains water under pressure, which enters the packing and causes the parchment to swell. By this means an exceedingly slippery mass of great toughness is produced, which causes but little friction.

For use in working with the condenser, *Maardt*[2] (1895) proposes the stuffing box shown in Fig. 568, which is made tight on the shaft. It is connected with the wall H of the case by means of a flexible plate, so that it can adjust itself independently of the latter. Further, Maardt also places a ring on the shaft, which has sufficient play in the opening of the wall of the case, and is to a small extent displaceable on the shaft in the direction of the axis. The ring loosely grasps the wall of the case with two flanges arranged inside and outside respectively. According as excess of pressure is set up outside or inside, either the outside or the inside flange is pressed tightly against the wall of the case.[3]

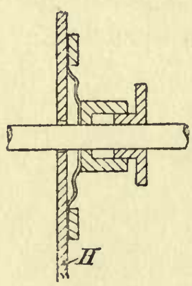


Fig. 568.

Webster[4] ships metal cones over the shaft on the low-pressure side of the casing; these are slit in the longitudinal direction and held at the bottom by means of a ring, and their points are pressed together by means of a spring packing ring, which bears against them.

If condenser pressure exists in the turbine case, a water stop for the shaft, such as is used by *Parsons*[5] (1887) (Figs. 569 and 570), is made in the following manner: first of all, packing rings *a* are attached. The annular channel *b*, contained in the cover of the case, is supplied through the tube *c* with water, which is sucked into the casing by the under pressure of the condenser, and at the same time prevents

[1] D. R. P. 25,383. [2] D. R. P. 87,519.
[3] See also p. 298, Fig. 549, where T represents the above-mentioned ring.
[4] E. P. 16,232 of the year 1901. [5] D. R. P. 41,479.

the entrance of air. In order to retard the passage of water, the shaft is provided with screw-shaped grooves *d* (Fig. 569), which act with effect on the former during the rotation of the shaft. If, for packing purposes, rings *e* (Fig. 570) are to be inserted in separate grooves cut into the cover of the case, attention must be paid to the circumstance that such rings do not offer a sufficient resistance to a strong hydraulic pressure. For this reason a number of rings should be inserted towards the end of the turbine in such a manner as to distribute the pressure between them. It is, however, preferable to introduce into the annular chamber *b* small quantities of steam which possesses a little more than the atmospheric pressure, and is taken from the steam pipe or from some point in the motor at which the pressure is low, eventually by means of a regulating valve. This arrangement ensures a better packing than an older proposal of *Parsons* (1884),[1] according to which any steam that may enter the chamber *b*, which in this case is arranged outside the stuffing box proper, is drawn off by an ejector fed with fresh steam.

In order to permit of oscillating or eccentric movements of the shaft *s*, and to

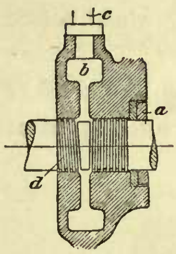


Fig. 569.

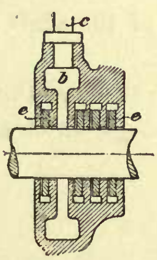


Fig. 570.

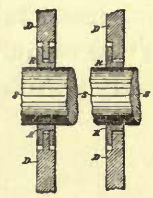


Fig. 571.

prevent the escape of the motive medium at the points at which it is packed, *Clarke* and *Warburton* (1898)[2] (Fig. 571) have arranged suspension rings R of L, U, or ⊥ shaped cross section on the shaft, either separately or in rows.[3] These rings form a packing for the partition walls D.

The above-named designers have proposed a special stuffing-box arrangement[4] for balanced axial turbines, in which the steam streams from the centre towards both ends. The hollow bearer of the working wheel is made tight in the covers of the case by means of labyrinth packing or conical packings, whilst the other two projecting ends of the shaft are provided only with labyrinth packing. Between the former and the latter, spaces are formed at both ends of the case, which are connected with each other by means of a special pipe system, so that, in case unequal amounts of steam should leak into the two spaces, the pressures may be equalized. By this means one-sided axial steam pressures, which might be set up by steam leaking through the packing between the bearer of the working wheel and the casing, are prevented. Besides, the compensating pipe for the pressure between the above-mentioned spaces may also be connected with a lower turbine stage, so that steam

[1] D. R. P. 33,066.
[2] D. R. P. 112,438.
[3] See also under reversing gear.
[4] E. P. 25,135 of the year 1901.

escaping from the turbine chamber through the packing may be led back to be used over again.

Curtis has likewise proposed a stuffing box with several rings, which is flexible in the radial direction.[1]

In the construction of *Terry*[2] (1889) (Fig. 572) the water box *a* surrounding the shaft is provided with an opening *b* towards an annular space *c*. In the latter an annular disc *d* with radial ribs, fixed on the shaft, rotates. The opening *b* is closed by means of a clack in such a manner that the water can only pass into the box *a* from the space *c*. If water passes from the box into the space *c* at the end of the shaft, *i.e.* towards the turbine chamber, it will be thrown about by the disc *d* until the centrifugal forces increase to such a degree that it is pressed from the space *c* through the opening *b* back into the box *a*.

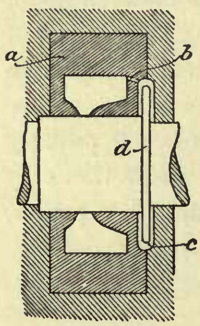


Fig. 572.

Huggins and *M'Callum*[3] arrange the surface condenser at one end of the turbine chamber, and lead the cooling water through the stuffing box of the shaft of the working wheel before passing it to the condenser tubes.

In the case of multi-stage steam turbines the stuffing boxes 15 and 16 (Fig. 573) are preferably provided with fluid rings, which are connected with a pressure-equal-

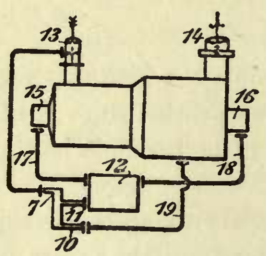


Fig. 573.

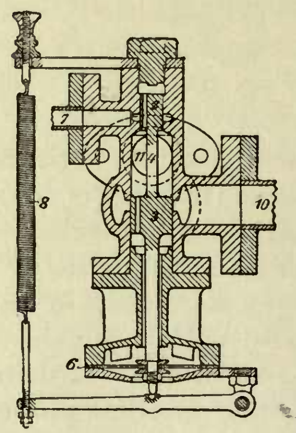


Fig. 574.

izing vessel 12 by means of the pipes 17 and 18. According to *Rateau* and the *Société Sautter, Harlé & Cie.* (1902)[4] (Figs. 573 and 574), a regulator is connected with the pressure equalizer 12, to enable a constant pressure in the fluid rings inside the stuffing boxes to be maintained. This regulator is connected with the vessel 12 by means of the branch 11, with the fresh steam pipe 13 by means of the tube 7, and with the low-pressure end by means of the tubes 10. The pistons 2 and 3,

[1] E. P. 756 of the year 1902.
[2] A. P. 649,014.
[3] E. P. 23,832 of the year 1897.
[4] D. R. P. 142,788.

fixed on the rod 4, close or open the pipes 7 and 10; the displacement of the same is effected by the steam pressure bearing on the diaphragm 6, or by the spring 8. The steam, escaping from the stuffing box 15 or its fluid ring, passes through the tube 17 into the vessel 12, and thence to the fluid ring in the stuffing box 16, and finally enters the interior of the turbine on the condenser side. If the pressure, acting on the diaphragm 6 through the bore, decreases in the vessel 12, the piston 2 is moved upwards by the action of the screw spring 8, thereby liberating the channel 7. The high-pressure steam will then pass through the connecting tube 7 into the regulator, and increase the pressure in the vessel 12 to its normal height. On the other hand, the excess steam leaves the vessel 12 through the branch 10 and tube 19, and enters the low-pressure turbine at a suitable point. If, however, the pressure in the interior of the vessel 12 increases, the piston 3 is moved downwards by means of the diaphragm 6, thereby releasing the opening in the branch 10. The vessel is then subject to the pressure of the steam introduced through the tube 19. The regulator can also be provided with valves instead of piston slides.

XXI

STEAM TURBINES FOR DYNAMOS

On land the machines for the production of electric energy are the ones on which the quickly and uniformly running steam turbine reacts most advantageously. In electrical engineering the turbine has proved to be a power engine, which adapts itself well to the peculiarities of the dynamo, besides opening new ways for its perfection and enabling large amounts of energy to be produced within a small space. The steam turbine has herewith found a large field of action for itself, and the constructions which have heretofore been brought on the market are specially

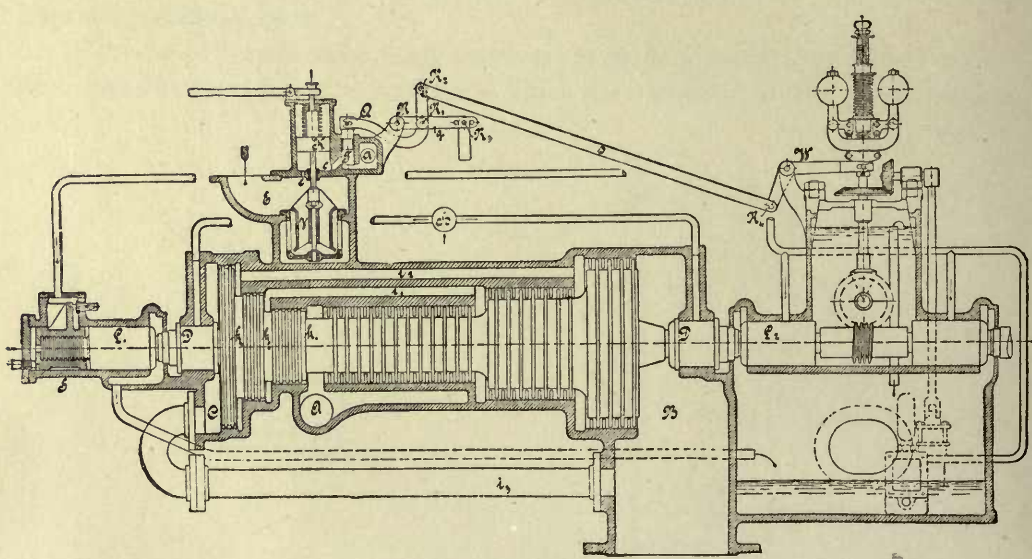


Fig. 575.

intended for the driving of dynamos. It seems advisable, therefore, first to discuss the steam turbine systems used in practice as motors for turbine dynamos.

The equipment and descriptions of methods of working of the present *Parsons turbine* are derived from the statements of Brown, Boveri & Cie. The complete arrangement is shown in the outline representation of the Parsons turbine given in Fig. 575. E represents an outer chamber, into which the steam first passes. A is the inlet of the steam into the cast-iron cylinder, B the steam outlet. The steel rotating body extends from C to B, the wreaths of vanes being fixed on the part AB on the right-hand side in the figure, whilst on the left-hand part AC three pistons, k_1, k_2,

PLATE IV.

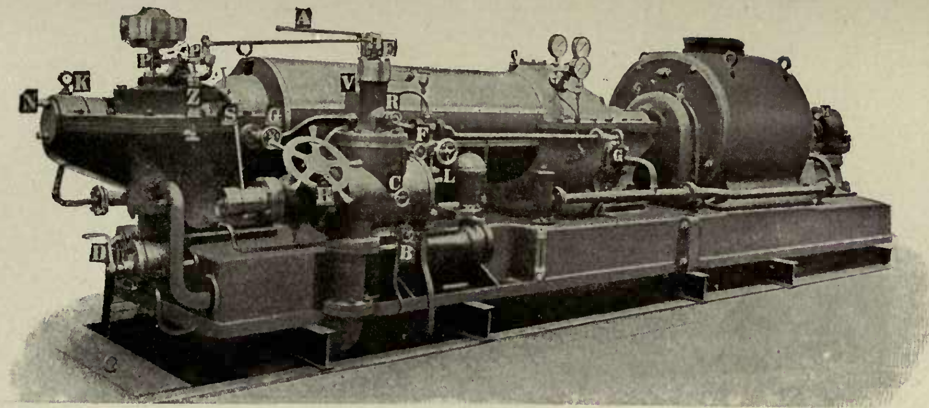


Fig. 576.
Turbine coupled to a Dynamo.

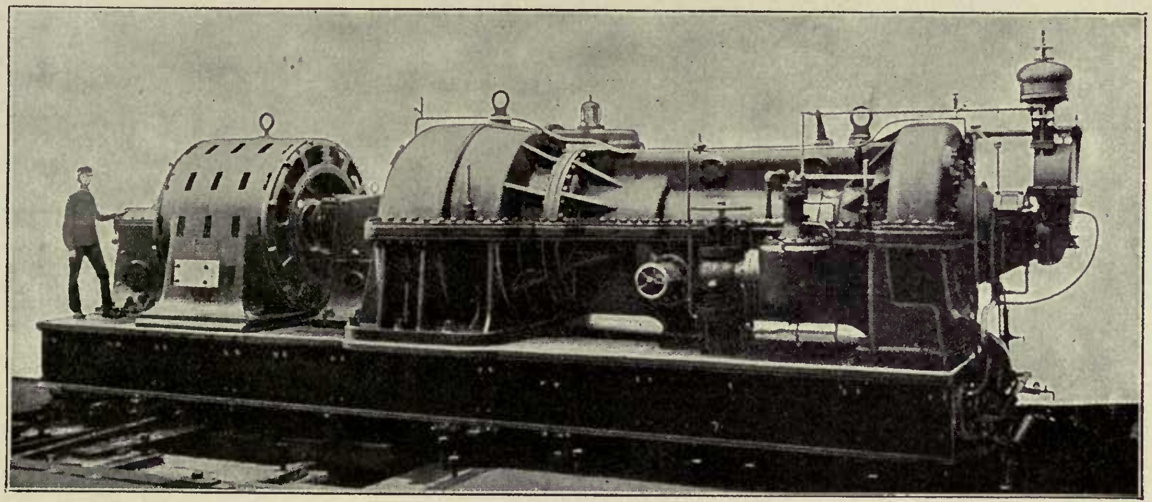


Fig. 578.
Engine on the Westinghouse-Parsons Dynamo System.

PLATE V.

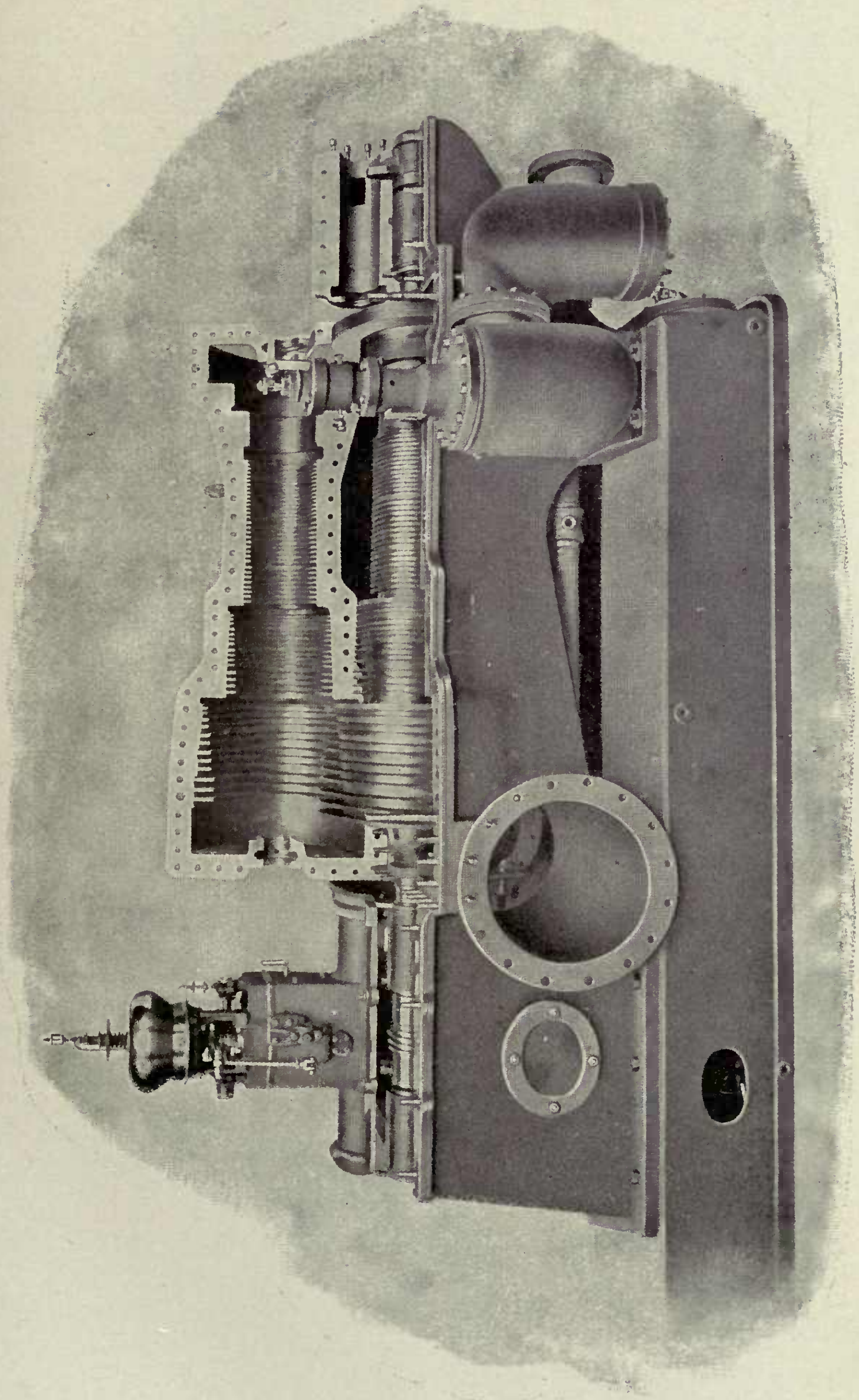


Fig. 577.
Parsons Turbine of 600 H.P., by the Westinghouse Machine Company.

and k_3, are arranged. The steam expands from the inlet A to the outlet B from one vane wreath to the other. Meanwhile its volume at the same time increases from wreath to wreath. The cross section of the flow of the steam inside the turbine increases in a degree corresponding with its constantly increasing volume. In the Parsons turbine this is attained by the arrangement that the vane-bearing part AC of the steel rotating body is subdivided into three stages, the diameter of which is smallest at the steam inlet A and largest at the outlet B, where the volume of steam is greatest. By this means the complete expansion of the steam is divided into three principal stages, as is the case in the triple-expansion steam engine. Owing to the movement of the steam from A to B, an axial pressure is exerted on the steel body in the direction of flow AB of the steam. In order to neutralize this pressure, the three pistons k_1, k_2, k_3 above mentioned, the diameters of which correspond with those of the three divisions of the vanes, and on which a steam pressure is exerted in the direction AC opposed to AB, are arranged on the left-hand part of the body AC. The spaces between these pistons and that behind the last piston are connected with the corresponding stages of the part of the body AB and the steam outlet by means of the channels i_1, i_2, i_3, so that the steam pressures in the directions of AC and AB are exactly the same. The shaft of the steam turbine is therefore completely relieved from axial pressure. The collar bearing S permits of an accurate adjustment of the rotating vane wreaths with reference to the stationary ones. The relieving pistons k_1, k_2, k_3 run practically without friction, being provided with labyrinth packing against the wall of the case. A small quantity of steam is supplied to the stuffing boxes D, so as to prevent the entrance of air when the condenser is in use. For this purpose the steam operating the inlet valve is made use of. The shaft is enabled to yield in the bearings L_1, L_2, by an arrangement by which lubricating oil with a pressure of 21 lbs. is forced into interstices between a number of boxes shipped one over another. The oil is kept in circulation by means of a pump. Worm gearing drives the oil pump, and the shaft of the governor, which latter influences the movement of the steam inlet V (in the manner described under governing) by displacing the piston slide T, which is kept moving up and down by the turbine shaft, by means of worm gear, eccentric, and lever q. The governor merely alters the stroke of a constantly moving organ. The strokes of the valve per minute amount to from 150 to 250, according to the size of the turbine. The steam passes from the governing appliance to the stuffing boxes.

Fig. 576 (see Plate IV.) represents the turbine directly connected with a dynamo. In this illustration, M is the exciting machine, H the hand wheel for the main steam valve, C the inlet valve box, A the hand lever for the first raising of the valve on starting the turbine, PPE the governor and the steering arrangement, Z the counter spring for adjusting the number of revolutions, NK the collar bearing, DBL the oil pump, with the air chamber L, and D the crank for the starting of the turbine.

For the sake of comparison, Fig. 577 (see Plate V.) shows a 600 H.P. turbine of the Parsons type, as constructed by the *Westinghouse Machine Company*, of East Pittsburg. The upper part of the case and the covers of the bearing are removed so as to show the rotating parts. It is to be noted that the radial widths of the vanes in each series gradually increase.

As regards the dimensions of the Parsons turbine, it may first be observed that the steam pressure acts on a very large surface. Thus, a turbine of medium size (of 1000 to 2000 H.P.) is fitted with about 30,000 working-wheel vanes. Each of these has an average length of 2 ins., so that the total length of the vanes is nearly 1 mile. The widths of the vanes vary between $\frac{3}{8}$ in. and $\frac{3}{4}$ in., so that the total vane surface is about 270 sq. ft. An opening of from $\frac{1}{32}$ to $\frac{1}{8}$ in., according to the diameter of the wheel, is left between the vanes of the working wheel and the cylinder wall, while the openings between the vanes of the working and guide wheels are even from $\frac{1}{8}$ to $\frac{5}{32}$ in. wide. There is thus sufficient play for the variation of the length produced by the heating and cooling of the parts of the machine. The equilibration of the working wheel is completely achieved, both statically and dynamically, so that a critical speed no longer exists. The small dimensions of the wheels offer great advantages. The body of the working wheel, which has the shape of a roller, has, in a turbine of 850 H.P. running at 3000 revolutions per minute, including the relieving piston, a length of 8 ft. 2½ ins. and an approximate diameter of 1 ft. 6 ins. In a turbine of 1000 H.P., running at 1000 revolutions per minute, the roller body has a length of 10 ft. 6 ins., and a mean diameter of 3 ft. 11¼ ins. In order to reduce the number of revolutions to 750 per minute, the diameter of the working wheel alone is slightly increased.

According to the measurements of engines that are in use, the *consumption of steam* in the case of Parsons turbines corresponds with that of the best steam reciprocating engines. The possibility of working with high degrees of superheating offers a prospect of an increase in the working economy. Out of the large number of experimental trial results available, the following table gives a few figures in reference to turbine dynamos:—

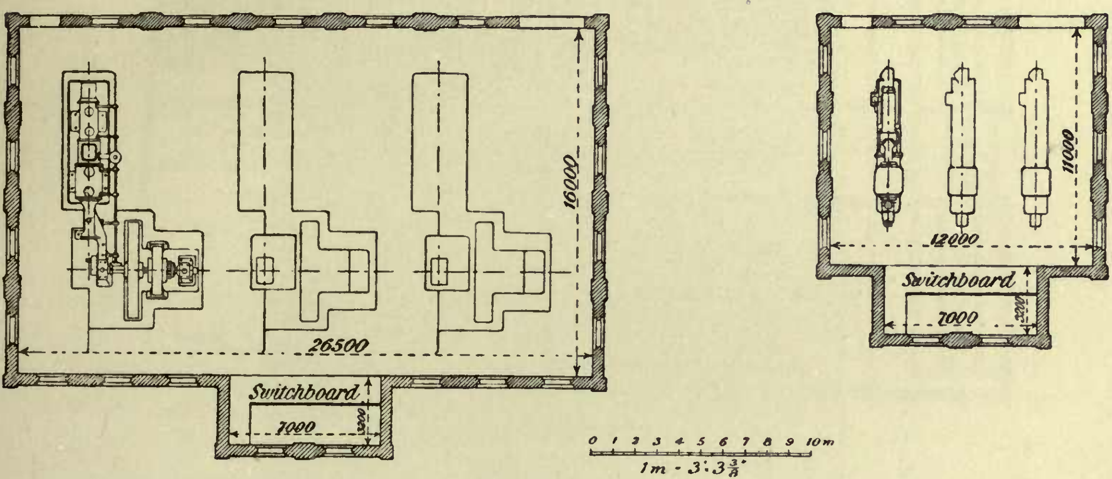
Performance (kilowatts).	Revolutions per minute.	Steam pressure (lbs. per sq. inch).	Steam temperature (°).	Vacuum (per cent.).	Consumption of steam measured in lbs. per kilowatt.	Converted into		Consumption of steam in lbs. per kilowatt per hour.
						Steam temperature.	Vacuum.	
382	2500	113·24	343°	92·5	21·54	572°	95	15·69
445	3000	122·06	450°	90·0	17·64	,,	,,	15·48
2995 (Frankfort o. M.)	1350	155·88	585°	90·0	14·77	,,	,,	13·67
3367 (Milan)	1260	182·35	450°	92·0	16·16	,,	,,	13·31

The economical advantages which the turbine possesses over steam reciprocating engines explain the rapid introduction of the Parsons construction, which, up till the beginning of 1904, *i.e.* within a period of six years after its extensive introduction into practice, has shown a total output of about 700,000 H.P., amongst which single installations of 10,000 H.P. are found.

Fig. 578 (see Plate IV.) represents a turbine dynamo of the Westinghouse-Parsons type (the Westinghouse Machine Co. of Pittsburg, Pa.) mounted in the power house of the Hartford Electric Light Co. The turbine is directly connected with a generator of 1500 kilowatts. The plant comprises an area of 35 ft. 6 ins. in length by 8 ft. $9\frac{1}{8}$ ins. in width. The total weight, inclusive of the current generator, amounts to about 78 tons.

Since the Parsons steam turbine system is at present the one most generally adopted, and therefore furnishes the largest number of results of working, the special properties of this type of engine and its advantages over steam reciprocating engines may here be mentioned.

1. *Small dimensions* result on the one hand from the possibility of introducing large amounts of energy into the engine in a unit of time, and on the other from the ability to dispense with the arrangement for converting the reciprocating motion into a circular one. For turbines up to 400 H.P., Parsons reckons on only half of



Figs. 579, 580.

the space required by steam reciprocating engines of equal power; for such as are above 400 H.P., on only one-third of it.

The Parsons turbine requires a base of about one-third of the size of that for a horizontal steam reciprocating engine, and of half the size of that for a vertical one of equal power. The dimensions of the working machine, with which the turbine is connected, may be diminished in consideration of the high number of the revolutions of the latter, which is a great advantage in the case of electric current generators. In connection with the statements of the Akt.-Gesellschaft für Dampfturbinen on the Brown, Boveri-Parsons system, mention may be made of Figs. 579 and 580, which illustrate the proportion borne by the space required for a reciprocating engine to that for a turbine of equal power. A general plan of the engine house and of the foundations of the power station Porta Volta of the Edison Company in Milan, is shown in Fig. 581. At this station, steam dynamos for a total power of 3950 kilowatts are used in connection with turbo-dynamos of 5600 kilowatts total power. Of these two turbines, that for 3000 H.P. was built by C. A. Parsons of Newcastle, and the other, for 5000 H.P., by Brown, Boveri & Co., who also supplied the polyphase current generators for both turbines.

As regards the space required, mention may be made of the Parsons turbine for 5000 H.P., which is directly connected with an alternating current machine and serves for the extension of the Städtisches Elektrizitätswerk, Frankfort o. M., shown in Fig. 582 (see Plate VI.). The complete set of engines has a length of 54 ft. 1 in., and a height and width each of 8 ft. 2½ ins. A graphic comparison between the space required by a reciprocating engine and its foundation, and that for a turbine, is also shown in Fig. 583 (see Plate VII.).

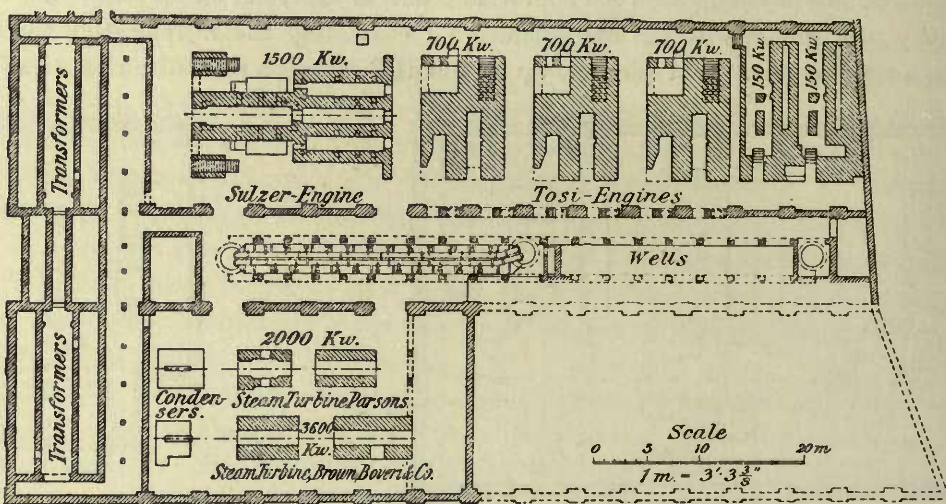


Fig. 581.

The plant assumed is one of 5000 kilowatts. In the one case, the energy is produced by a twin Corliss engine with horizontal high-pressure and vertical low-pressure cylinders, whilst in the other case, a Westinghouse-Parsons turbine dynamo is used for the same purpose.

2. The *small weight* of the turbines results from the same causes as do their small dimensions. In addition to this, the means required for maintaining uniformity of the movement (fly wheels) in the case of common reciprocating engines may be dispensed with. According to the size of the engine, Parsons recommends a weight of 55 to 33 lbs. per H.P. per hour, against 220 to 132 lbs. in the case of reciprocating engines. The 5000 H.P. turbine of the "Frankforter Elektrizitätswerk" weighs about 65 tons, whilst a Sülzer engine of equal power, without dynamo, weighs 394 tons.

It is obvious that the small dimensions and the reduced weight also lead to convenient transport and render quick and easy erection possible. Besides, the foundations may be small and light, especially as they need not take up the

PLATE VI.

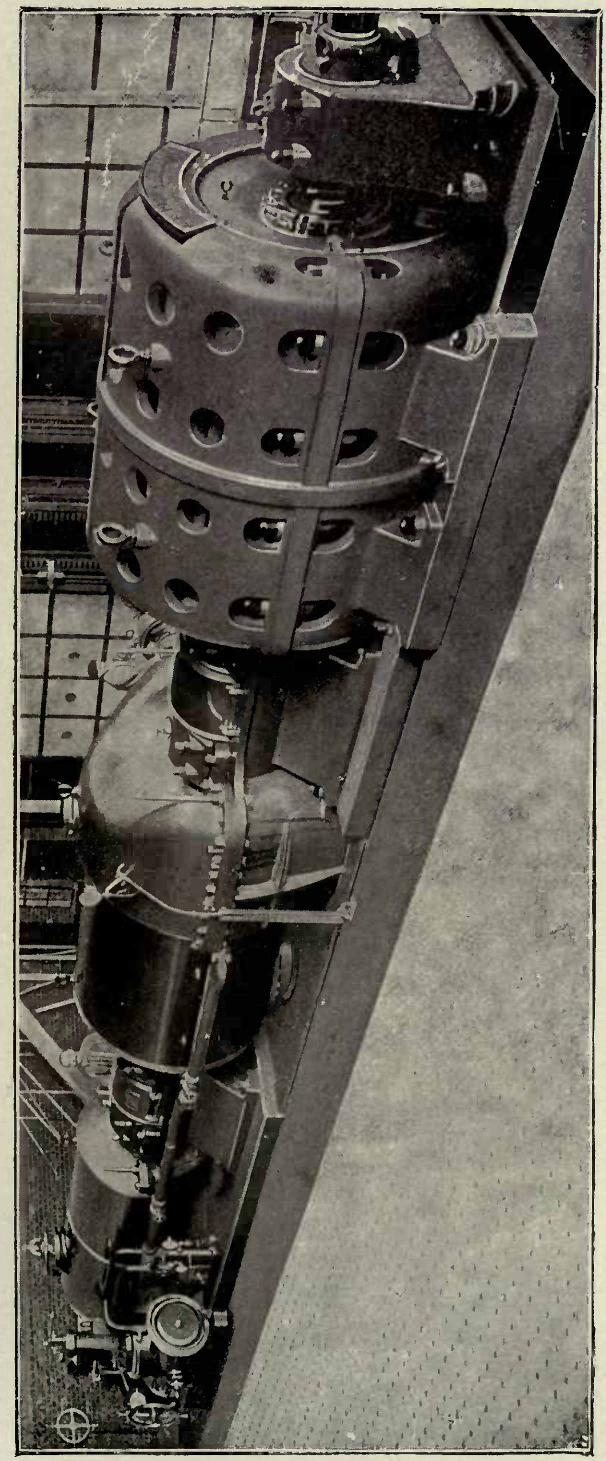


Fig. 582.
Turbine in the Frankfort-on-the-Main Electric Power House.

PLATE VII.

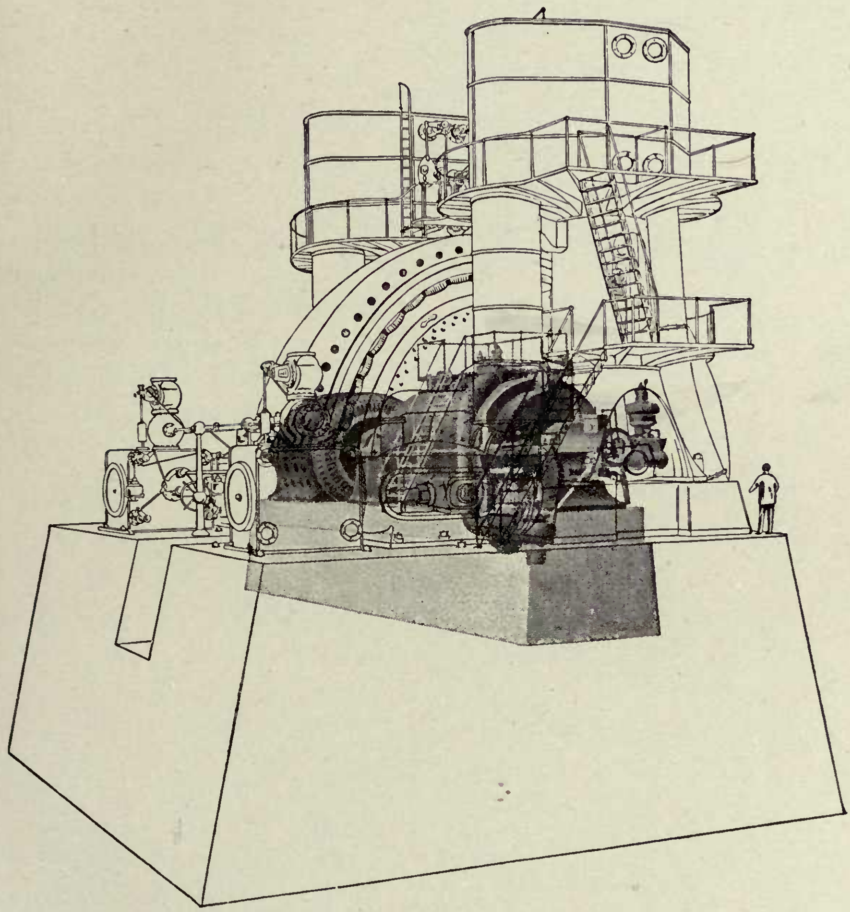


Fig. 583.
Comparison between Turbine and Piston Engine as regards Room.

PLATE VIII.

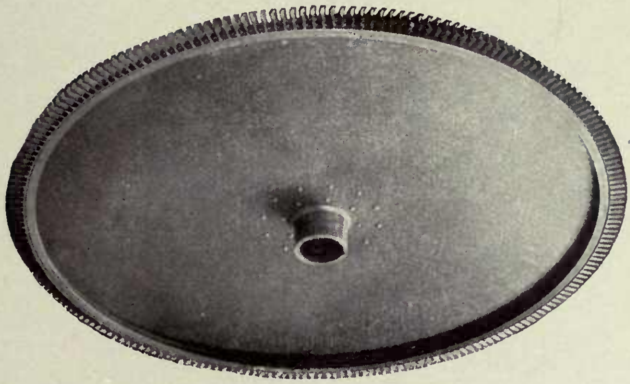


Fig. 585.
Working Wheel of a Rateau Turbine.

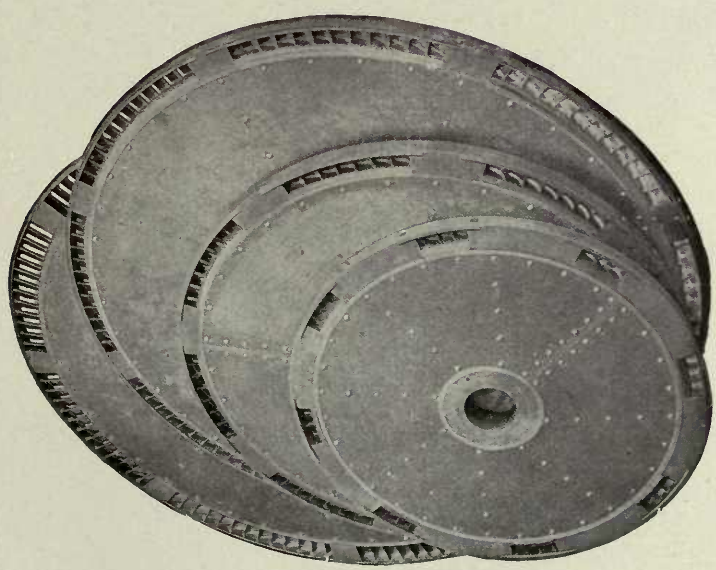


Fig. 584.
Guide Wheels of a Rateau Turbine.

PLATE IX.

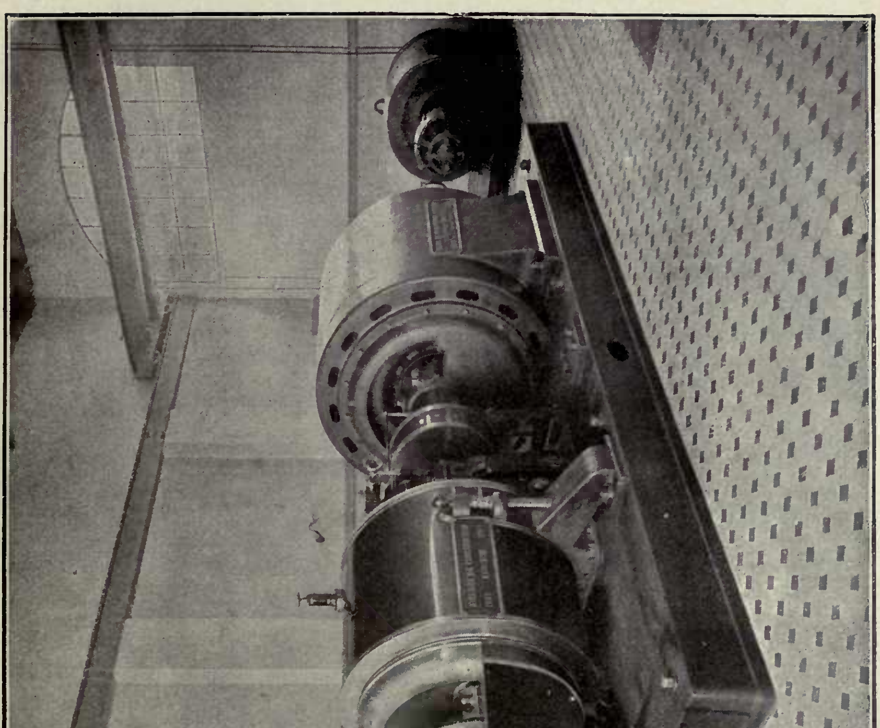


Fig. 586.
Rateau Turbine with Polyphase-current Dynamo of 1000 Kw.

PLATE X.

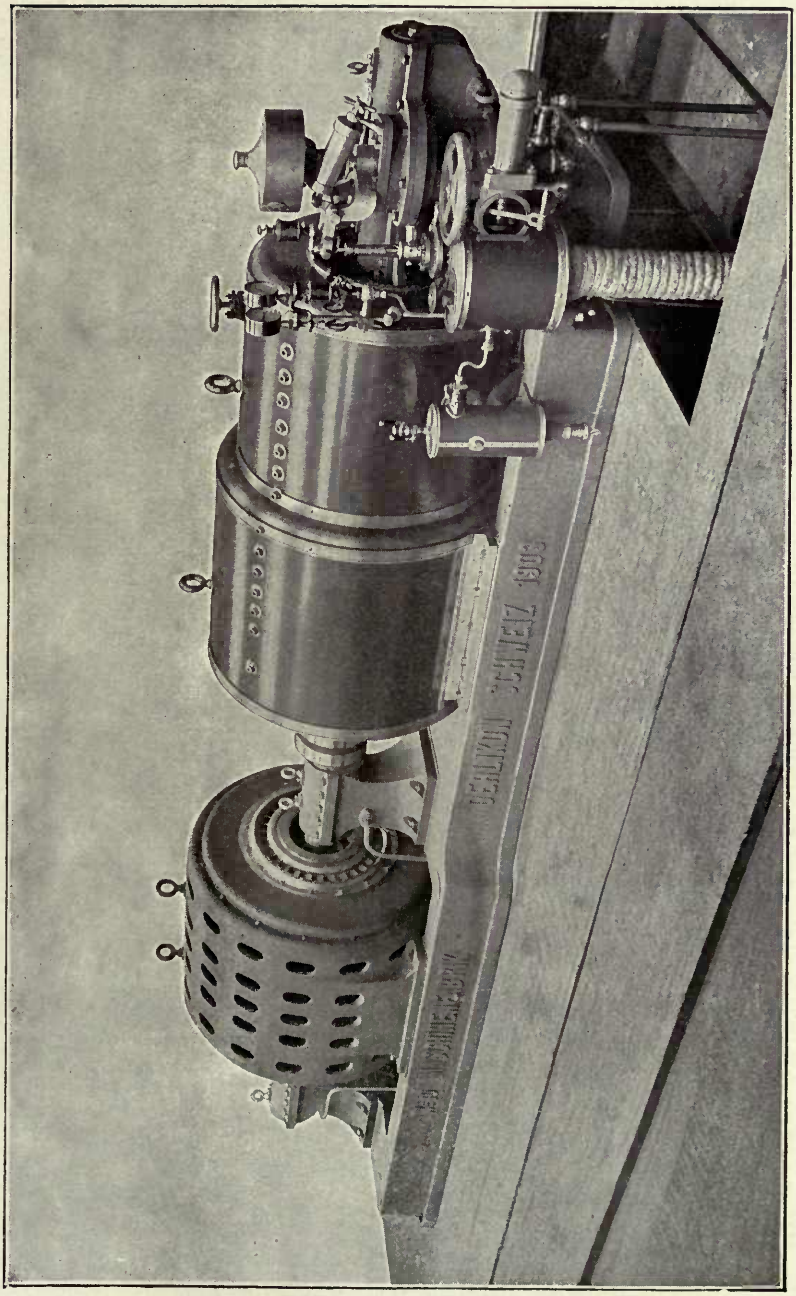


Fig. 587.
Rateau Turbine with Continuous Current Dynamo of 150 Kw.

oscillations of the moving masses. The approximate volume of the seating of a steam turbine is said to be about 15 to 30 per cent. of that of a steam reciprocating engine of the same power. These facts offer great advantages in shipbuilding. It is stated that the weight of the engine of an ocean steamer of considerable size can be reduced to from one-third to one-fourth of that at present necessary.

3. The fact that the individual parts of the turbine are but *slightly subject to wear and tear* is explained by the non-existence of metallic rubbing in the turbine itself, and in the reduction of the same at the main bearing and the much simplified valve gear. It has been established by irreproachable reports that after a working period of 7000 hours, neither the vanes nor the bearings have suffered from the steam. If, however, the working parts do not change during a period of several years, the consumption of steam must also remain the same.

4. The *consumption of oil* is very small. In order to lubricate the bearings, the oil is kept in circulation; the loss of lubricating material is due to evaporation, etc. Experience has shown that the consumption of oil in a Parsons turbine amounts to from 0·013 to 0·015 lb. per H.P. per hour.

5. A further advantage possessed by the turbine over reciprocating engines is that the *exhaust steam* and the *waste water* are *free from oil,* owing to the circumstance that the parts of the engines over which the steam sweeps need not be lubricated. For this reason the waste water can be led into the boiler in a warm condition without having to be purified, provided it has not come in contact with the cooling water.

6. For the use of *superheated steam* the turbine offers a wide field. As decomposition of the lubricating material is not to be feared, there is no limit to the degree of superheating until temperatures are attained such as affect the metal parts.

7. The *governing* has reached a degree of perfection that cannot be attained by steam reciprocating engines. The Parsons turbine possesses only one steam inlet, which causes the working medium to enter at intervals of one-third to a quarter of a second. Every change in the period of admission brought about by the governor is almost immediately observable in the whole turbine, as the steam streams through the working chambers in an exceedingly short space of time. The movement of the rotating masses, which are comparatively small, can be quickly adapted to the effective energy.

In this connection mention may be made of the trial results with the 450 H.P. turbine of the Elektrizitätswerk Linz-Urfahr. The number of revolutions of a turbine at empty and at full loads respectively varies by 2 per cent. only. In case of sudden variations of the load by 100 per cent., the number of revolutions only varied by 1½ per cent. in each direction, measured from the centre line of the diagram, whilst the condition of steady motion was attained within 3½ seconds after the variation had taken place.

The 250 H.P. turbo-dynamo of the Heidelberg Städtische Gas-, Wasser- und Elektrizitätswerk is able completely to adapt itself to the continuously varying amount of energy required by the tramway without the use of a buffer battery, and is not subject to a dangerous increase of the number of revolutions in case of sudden relief.

The facility of governing the turbine offers an additional advantage, which is of great service in the case of working in parallel. It has been found that a turbo-alternator can be easily connected in parallel with another turbo-alternator, or with current generators driven by reciprocating engines. In this respect mention may be made of the installation of the Tramway and Elektrizitätsgesellschaft Linz-Urfahr.

8. The consumption of steam can be made independent of the *operator,* so that the economy of working does not depend on the care of the staff. The lubrication is indeed effected automatically, and a subsequent tightening of the stuffing box is unnecessary.

9. The *stuffing* and *packing material* is also dispensed with.

10. The *starting* of the engine can be quickly and easily effected. The turbine can be set in motion from any position, and it does not require to be slowly heated up by feed water.

11. As regards the *waste water,* experience has shown that its injurious results are confined to a reduction in the number of the revolutions. When, for instance, the 450 H.P. turbine dynamo of the Tramway and Elektrizitätsgesellschaft Linz-Urfahr was dismounted after running for 2000 working hours, in order to determine the wear and tear suffered by it, boiler mud was discovered in the central expansion stages, which was easily removed.

In regard to the parallel working of Parsons turbines and steam reciprocating engines, the installation of the *Tramway and Elektrizitätsgesellschaft Linz-Urfahr* [1] gives some interesting information in a case in which a turbo-alternator of 300 kilowatts (2000 volts 45 periods per second, and 2700 revolutions per minute), constructed by Brown, Boveri & Co. in Baden, together with three single-phase alternating current machines of 100 kilowatts each (2000 volts. 45 periods per second, and 270 revolutions per minute), and one of 200 kilowatts (2000 volts 45 periods per second, and 169 revolutions), has been at work since January 1, 1902, nearly every other day.

Variations such as accompany the connection of reciprocating engines in parallel with one another do not show themselves in this case, in which a turbine is so connected with one or more of these. The convenient adjustment of the number of revolutions permits of the connection of the turbine in parallel, irrespective of the load on the other engines, whilst the governability of the turbine equalizes the variations of load in the system, including those on the reciprocating engines.

At the Elektrizitätswerk of the city of Heidelberg, a dynamo group of the Brown-Boveri-Parsons type of 180 kilowatts, in connection with two steam reciprocating engine dynamos of 300 kilowatts each, produces direct current of 550 volts for tramway work. In this case the usual buffer battery, connected in parallel, has

[1] Schw. Bztg. 1903, p. 10.

proved to be superfluous, because the turbine itself performs the equalizing work of the battery. Even if variations of the load on the turbine from about zero up to full load should take place in connection with this automatic regulation, the variations of the pressure will only amount to from 1 to 3 per cent.

The curves of steam consumption of the Parsons turbine prove that, in contradistinction to that of reciprocating engines, the commercial efficiency of the latter improves with increasing excess of load. Since, owing to the possibility of the provision of rational ventilation in a boiler plant of sufficient dimensions, the high-speed Parsons turbines are able to bear especially high excess loads, turbines of approximately small size are considered satisfactory for this purpose.

The *Westinghouse Company* [1] runs turbine dynamos of—

5000 kilowatts	with	750	revolutions per minute
2000	„	1200–1560	„ „
1000	„	1500–1800	„ „

The Pennsylvania Railroad Company are now erecting engines of 5000 kilowatts (7500 H.P. each). These types of engine are of comparatively short length, coupled with large diameters of the working wheels. These turbines cover an area of 27 ft. 7 ins. by 13 ft. 1½ ins. without dynamos, or of 47 ft. 3 ins. by 13 ft. 1½ ins. with dynamos.

The total height (inclusive of a rail over the case of the engine) is 12 ft. It may be pointed out that the case is enclosed by a heat-protecting cover. The inner bearing is stationary, whilst the outer bearing slides in grooves in the framework of the seating. The bearing has a bore of 1 ft. 3 ins. only. For lubricating the bearings, oil circulates at a low pressure. The hot oil is led through a serpentine arranged in the seating frame, in order to cool it. A valve which, in case of overloading, is to supply fresh steam to the lower stages, the capacity of which latter, as compared with the others, is increased by 50 per cent., is of great importance.

Attention may be called to a recent proposal of *Parsons*,[2] in which the counter-direction wheels are used. Either both of these are arranged as flying wheels, in which case the shaft of each is connected with a dynamo and the two dynamos are connected in parallel, or one of the shafts is inserted into the other. In the latter case only one dynamo is used, the armature of which is fixed on the one shaft and the field magnets on the other. (A similar arrangement was also planned by *Lohmann*.[3])

To enable alternating current generators to be directly coupled with quickly running steam turbines, *Parsons* [4] (1903) proposed a method of starting, the object of which was the generation of currents of a frequency lower than had till then been possible and than those to which the speed of the turbine was adapted. According to this, the speed of the driving motor was utilized by stages in two or more generators in which the relative speed between the two effective elements of each

[1] See *Western Electrician*, Vol. 34, p. 38.
[2] E. P. 6142 of the year 1902. A. P. 729,215. See also p. 119.
[3] E. P. 16,635 of the year 1897.
[4] D. R. P. 151,152.

generator only formed a portion of that of the motor. In the higher stages the electro-magnetic influences are thus divided between current generation and the driving round of the one element of the following stage.

The *Rateau turbine*, the arrangement and mode of working of which have already been sufficiently discussed, is constructed by the Swiss engineering works, "Oerlikon," and by *Sautter, Harlé & Cie.*, of Paris. I am indebted for the illustrations to the first-named firm. Fig. 584 (Plate VIII.) shows the guide wheels belonging to several stages. The leading channels are collected together in groups, which from stage to stage cover increasing portions of a circle. The working wheels (Fig. 585) (Plate VIII.) are made of steel plates, the edges of which are flanged; the vanes are riveted into the rings formed by the latter. Fig. 586 (see Plate IX.) shows a Rateau turbine with a polyphase of 1000 kilowatts, and Fig. 587 (see Plate X.) a Rateau turbine in connection with a continuous-current dynamo of 150 kilowatts. The following table shows the sizes of the Rateau turbine dynamos for various powers, built by the "Oerlikon" works:—

Power in kilowatts	100	200	300	400	500	600	800	1000	1250	1500	1750	2000	2500	3000	3500	4000
Revolutions per minute approx. .	3000	3000	3000	3000	3000	3000	1500	1500	1500	1500	1500	1500	1000	1000	1000	1000
Length, extreme, in feet and inches, approx.	16 4$\frac{3}{4}$	18 0$\frac{1}{2}$	19 4$\frac{1}{4}$	20 8	21 3$\frac{7}{8}$	21 11$\frac{3}{4}$	22 11$\frac{5}{8}$	23 7$\frac{1}{2}$	25 7$\frac{1}{8}$	27 6$\frac{3}{4}$	29 6$\frac{1}{4}$	31 2	34 5$\frac{3}{8}$	37 8$\frac{3}{4}$	41 0$\frac{1}{8}$	44 3$\frac{1}{2}$
Breadth, extreme, in feet and inches, approx.	3 7$\frac{1}{4}$	4 5$\frac{1}{8}$	5 1	5 6$\frac{7}{8}$	5 10$\frac{3}{4}$	6 2$\frac{3}{4}$	6 6$\frac{3}{4}$	6 10$\frac{3}{4}$	7 2$\frac{5}{8}$	7 6$\frac{1}{2}$	7 10$\frac{1}{2}$	8 2$\frac{3}{8}$	9 0$\frac{1}{4}$	9 10$\frac{1}{8}$	10 7$\frac{7}{8}$	11 5$\frac{3}{4}$
Height, extreme, above the floor, in feet and inches, approx.	2 11$\frac{3}{8}$	3 7$\frac{1}{4}$	4 1$\frac{1}{4}$	4 5$\frac{1}{8}$	4 9$\frac{3}{8}$	4 11$\frac{3}{4}$	5 3	5 6$\frac{7}{8}$	5 10	6 0$\frac{3}{4}$	6 4	6 6$\frac{3}{4}$	7 2$\frac{5}{8}$	7 10$\frac{1}{2}$	8 6$\frac{1}{4}$	9 2$\frac{1}{4}$

In regard to the efficiency, it may be observed from a design of Rateau's (dating from the year 1900) that the hydraulic coefficient of performance up to that time determined was more than 70 per cent., and that the losses due to friction of the discs, as well as those at the clearance space, did not exceed from 10 to 12 per cent. at full load. About 6·5 per cent. of the available energy of the steam is said to be transmitted to the shaft. In the case of large engines with pressures of more than 190 lbs. per sq. inch, with condenser pressures of 1$\frac{1}{2}$ lbs. per sq. inch, Rateau expects to be able to reduce the consumption of steam per E.H.P. to 11·46 lbs.

The weight of a 1200 H.P. turbine is hardly 3$\frac{1}{2}$ tons. It is said to be possible, without exceeding this weight, to increase the power of the turbine to 2500 H.P., which would give 3·08 lbs. per H.P.

The *Zoelly turbine*, which is being constructed and developed by the *Actiengesellschaft der Maschinenfabriken von Escher, Wyss & Cie.*, in Zurich (Switzerland) and Ravensburg (Würtemberg), seems also to have got beyond the experimental stage. The aims of Zoelly were evidently directed towards the construction of as large working wheels as possible, which with high peripheral velocities could, of course, have low speeds of rotation, and enable the number of the stages to be reduced. In order here to avoid difficulties in the manufacture and to ensure safety in working, the designer has attached to a disc vanes in the form of radially projecting comparatively long bars, which represent bodies of approximately equal strength.[1] In the original method of construction we find a division into pressure stages, and the expansion nozzles are made use of in every stage. In a more recent arrangement radial wheels are combined with axial ones, and in the latter, indeed, the expansion nozzles are dispensed with.[2]

The style of construction of the Zoelly turbine which has come into practical use is different from the above. It is illustrated in Figs. 588 to 593. The turbine is divided into a high-pressure section and a low-pressure one, and these are placed in separate cases. Between the latter is a shaft bearing. Two other bearings are likewise arranged outside the cases, and these latter are connected by means of a pipe. The working wheels are forged of Siemens-Martin steel, and in each case the disc and the nave are in one piece. The greater the diameters of the discs, the shorter the ray-formed vanes attached to them are made. The latter are made of finely polished nickel steel, and their feet are set in the groove of the disc flange with spaces between which are occupied by distance pieces; a ring-shaped counter-flange completes the attachment. The guide vanes are severally inserted in cast-steel walls,[3] which closely adjoin the walls of the case and the naves of the wheels, and thus form chambers, one for each working wheel. Since these partition walls lie closely against the outer rings and are slightly let into the case, the latter takes up the axial thrust resulting from the excess pressure of the steam which bears against the one side. The partition walls are made tight against the naves of the wheels by means of labyrinth packing. Expansion nozzles are not resorted to, but in their stead the steam is led from a small number of leading channels into the first stage of the first working wheel, so that the latter is impinged upon only over a part of its periphery. In the succeeding stages the area of impingement becomes greater.

By the firm of *Escher, Wyss & Cie.*, who manufacture it, the turbine is called a multi-stage axial action turbine. Now, on the assumption that the arrangement shown for the low-pressure turbine is adopted for the high-pressure one also, the steam clearly passes from the impinging leading channels of the first working wheel and fills the inflow section of the latter. While the breadth of the (radial) vanes remains constant, that of the working-wheel vanes increases in the direction of flow of the steam. If only the volume of the steam be in question, this increase of breadth of course enables a corresponding diminution of the angle of discharge to be made.[4] By the increase of breadth an acceleration of the steam due to expansion is also, apparently, attained. In each stage, then, a portion of the pressure in the working wheel will clearly be converted into velocity, and this be consumed by the

[1] Compare p. 254.
[2] Compare p. 99.
[3] Compare p. 246.
[4] Compare the Curtis turbine.

wheel. The result is an axial velocity turbine working in pressure stages with acceleration of the steam in the working wheel.

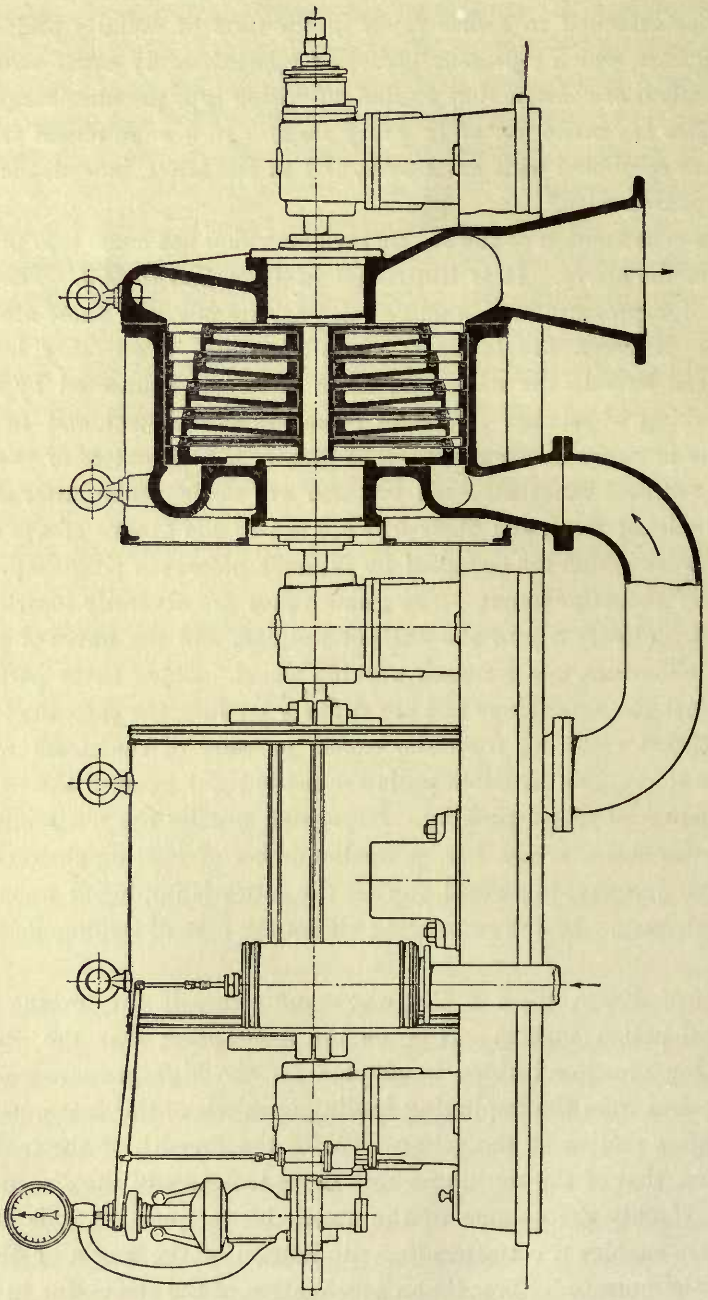
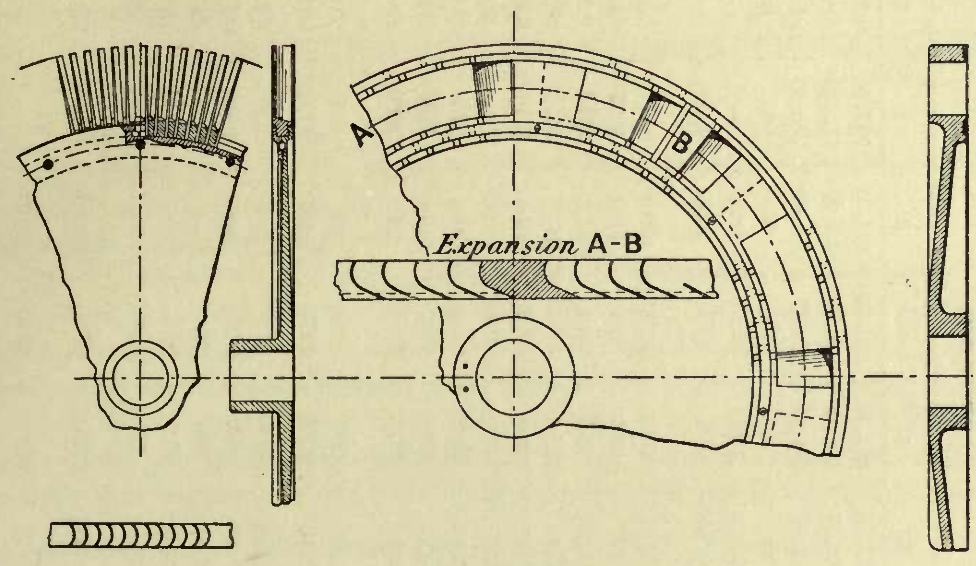


Fig. 588.

The cases mounted on the engine frame are parted in the horizontal plane of the axes, and the flanges of the parts are smoothed so that these fit one upon another.

The regulation of the turbine takes place, like that of the water turbines of the firm, by means of a sensitive spring governor with an auxiliary engine, the entrance pressure of the steam being altered in accordance with the varying load. The governor thus takes the form of a throttling arrangement, which is, in itself, a simple one. Whether it does not disadvantageously influence the consumption of steam of the turbine may be a matter for separate consideration. In addition, a safety governor is provided, which, after a certain number of revolutions is reached, releases a spring that closes the governing valve.

Experiments have been made with 147 lbs. excess pressure and 3000 revolutions per minute on a normal Zoelly turbine of 500 H.P., which is directly coupled with



Figs. 589–593.

an alternating current generator supplied by the Siemens-Schuckert Works. The turbine has ten working wheels. The total length of the group of turbine dynamos is about 18 ft. 9 ins., the extreme breadth (measured outside) 4 ft. 11 ins. The results of the experimental trials made by Messrs. Escher, Wyss & Cie. are put together in the table given on next page.

For the sake of comparison, the results of the steam consumption trials made by Prof. Stodola on a 600 H.P. steam dynamo in February, 1894, may also be appended. In connection with these it must be observed that the trials were carried out with a boiler excess-pressure of about 154 lbs. per sq. inch, and with a vacuum of about 93·5 per cent. The temperatures given for superheating are measured in front of the valve (not at the engine). The figures for steam consumption are those per kilowatt, and are measured at the terminals of the dynamo. The consumption of power for the condensation, which is worked by itself, is not, however, included in these figures; an addition of from 2·5 to 3 per cent. must therefore be made for it.

	Dry saturated steam.								Superheated steam.		
1. Number of the experiment	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.
2. Date	21 XII. 03	25 I. 04	25 I. 04	25 I. 04	25 I. 04	18 I. 04	25 I. 04	25 I. 04	5 II. 04	5 I. 04	5 I. 04
3. Beginning	3 h. 10	3 h. 15	3 h. 55	2 h. 45	1 h. 30	4 h. 00	11 h. 25	10 h. 35	3 h. 50	3 h. 50	11 h. 15
4. End	6 h. 10	4 h. 35	4 h. 45	3 h. 35	2 h. 20	5 h. 00	12 h. 25	11 h. 10	5 h. 00	4 h. 10	12 h. 35
5. Duration in minutes	180	80	50	50	50	60	60	35	70	20	80
6. Gross power developed	363·78	388·47	335·31	240·78	182·85	80·62	—	—	392·5	390·41	391·2
7. Exciter voltampère	0·72	0·82	0·80	0·68	0·63	0·49	0·479	—	0·81	0·806	0·816
8. Useful work done (that of exciter deducted)	363·06	387·65	334·51	240·1	182·22	80·13	—	—	391·66	389·6	390·4
9. Number of revolutions	2967	2967	2977	2983	2984	2995	2995	3000	2972	2973	2968
10. Pressure (abs.) in front of the water separater	164·1	164·1	160·3	161·9	161·3	162·3	162·2	163·6	188·4	193·1	165·6
11. Temperature °F. in front of the water separater	369·0	369·7	364·4	365·5	365·2	364·8	364·8	366·3	476·8	497·3	439·2
12. Saturation temp. ,, in front of the water separater	362·7	362·7	360·7	361·6	361·2	361·8	361·7	362·9	373·9	375·8	363·4
13. Superheating, Pos. 11–Pos. 12 °F.	6·3	7·0	3·7	3·9	4·0	3·0	3·1	3·4	102·9	121·5	75·8
14. Pressure (abs.) in front of the guide wheel	(148·5) ?	148·68	132·8	101·76	80·43	45·14	17·9	10·97	142·9	142·9	144·1
15. Temperature °F. in front of the guide wheel	355·8	356·0	347·2	328·8	313·9	276·8	227·8	217·2	421·7	426·2	421·7
16. Saturation temp. ,, in front of the guide wheel	354·0	354·9	346·1	326·5	309·9	272·5	220·5	196·2	351·7	351·7	352·4
17. Superheating, Pos. 15–Pos. 16 ,,	1·8	1·1	1·1	2·3	4·0	4·3	7·3	21·0	70·0	74·5	69·3
18. Pressure in exhaust . . . . . abs.	1·051	1·060	0·998	0·966	0·972	0·766	0·750	0·756	0·956	0·976	1·017
19. Temperature in exhaust . . . . °F.	102·4	103·8	102·0	98·8	97·9	90·9	90·0	107·8	100·4	101·8	100·4
20. Pressure in condenser . . abs.	—	0·676	0·693	0·750	0·779	0·647	0·647	0·676	0·588	0·617	0·617
21. Temperature at condenser: Pipe . . °F.	72·5	72·3	72·0	73·0	75·4	—	61·7	61·7	68·4	68·9	68·7
Temperature at condenser: Receptacle ,,	75·1	75·1	76·6	79·2	80·2	74·5	79·2	80·8	72·3	72·3	74·7
22. Height of barometer in inches	29	$28\frac{3}{4}$	$28\frac{3}{4}$	$28\frac{3}{4}$	$28\frac{3}{4}$	$28\frac{7}{8}$	$28\frac{3}{4}$	$28\frac{3}{4}$	$28\frac{1}{8}$	$28\frac{1}{8}$	$28\frac{1}{8}$
23. Consumption of steam in lbs. per hour	7903·6	8326	7426·3	5778·3	4683	2649·9	1025·1	651·2	7453·8	7334·5	7728·7
24. Consumption per useful kw.-hour in lbs.	21·77	21·48	22·20	24·06	25·70	33·07	—	—	19·03	18·82	19·79

The results are—

I. *Saturated Steam:*

Empty load without excitation:	consumption of steam	per hour			651·2	lbs.
„ „ with „	„	„	„		1026·2	„
80·1 kw. developed	„	per kw. per hour			33·1	„
182·2 „ „	„	„	„		25·8	„
240·1 „ „	„	„	„		24·0	„
334·5 „ „	„	„	„		22·2	„
387·6 „ „	„	„	„		21·47	„

II. *Superheated Steam:*

390·4 kw. developed	428° temp.,	consumption of steam per kw. per hour	19·79	lbs.
391·7 „ „	464° „	„ „ „ „	19·02	„

The de Laval turbine, the great speed of rotation of which has hitherto limited it to an utmost power of 350 horses, is in Germany manufactured by the *Maschinenbau-Anstalt Humboldt,* of Kalk, near Cologne am Rhein. Thanks are due to these works for the following information.

The dimensions of a 100 H.P. turbine coupled to a Siemens continuous-current dynamo are to be seen from Table I., in which also one of the impingement nozzles, which is closable from the outside, is shown. The diameter of the working wheel from centre to centre of the breadths of the vanes is 1 ft. $7\frac{1}{16}$ ins. The driving wheel mounted on the working-wheel shaft measures $1\frac{13}{16}$ ins. over the pitch circle, and engages with two bevel wheels on the Laval system, $22\frac{3}{8}$ ins. in diameter. When in this size of engine the shafts directly driven make 1050 revolutions per minute, as intended, those of the working wheel become $1050 \times \frac{22\frac{3}{8}}{1\frac{13}{16}} =$ about 12971. An outside view of this turbine dynamo is given in Fig. 594 (see Plate XI.).

It is customary to allow the smaller Laval turbines to work on a single-gearing shaft arranged at one side. Larger engines of 30 H.P. or more impart their rotary motion to pairs of symmetrically arranged shafts. The first of the following tables (p. 328) contains the general dimensions of a Laval turbine that is to be driven by belting, and the second those of another such to be driven by continuous-current turbine dynamos.

In regard to the consumption of steam of the Laval turbine, the experimental trials which were made on December 20, 1899, with the 300 H.P. turbine of the Pabianicer-Baumwollmanufakturen, Krusche & Ender, among others, give particulars.

Effective horse-power.		3	5	10	15	20	30	50	75	100	150	200	300
Room required	Length in feet and inches . . .	2 7½	3 0¼	3 4⅞	3 6⅛	4 5½	4 9¾	7 6½	8 10¼	10 5⅞	11 1¾	13 9¼	14 9⅞
	Breadth " " . . .	1 2⅝	1 7¾	1 9⅝	1 9⅝	2 1¼	2 1⅝	3 2⅝	3 5¾	4 3⅞	4 3⅞	5 2⅞	6 5⅞
	Height " " . . .	1 1⅜	2 7½	3 3⅜	3 3⅜	3 7¼	3 7¼	4 5½	4 3⅞	4 9½	4 9½	4 11	5 0¼
Belt wheel on the motor	Revolutions per minute . . .	3000	3000	2400	2400	2000	2000	1500	1250	1050	1050	750	750
	Diameter in feet and inches . .	6¼	6¼	7⅞	7⅞	9⅜	9⅜	1 1¾	1 3⅜	1 5¼	1 6⅛	1 8½	2 1⅝
	Breadth " " . .	3⅛	3⅛	3⅞	4½	5⅛	6⅛	9⅛	9¾	11¾	9¾	1 3¾	1 7¾

Effective horse-power of the turbines		1½	3	5	10	15	20	30	50	75	100	150	200	300
Power developed by the turbines in watts .		800	1650	3000	6110	9420	12580	19140	32250	48500	66000	100000	132000	200000
Revolutions of the dynamos per minute .		3000	3000	3000	2400	2400	2000	2000	1500	1250	1050	1050	750	750
Room required	Length in feet and inches	2 6¼	3 9¼	5 6⅞	6 0¾	6 6¾	7 3¾	7 7¼	8 8¼	9 10⅞	11 4⅝	13 1½	14 7¼	16 0⅞
	Breadth " "	11¼	1 3⅜	1 7¾	2 4¾	2 5½	2 11⅜	2 11⅜	3 3⅜	3 7¼	4 3⅞	4 11	5 3	6 6¾
	Height " "	11¾	1 4¼	2 7½	3 3⅜	3 3⅜	3 7¼	3 7¼	4 5½	4 3⅞	4 9½	4 9½	4 11	5 0½

PLATE XI.

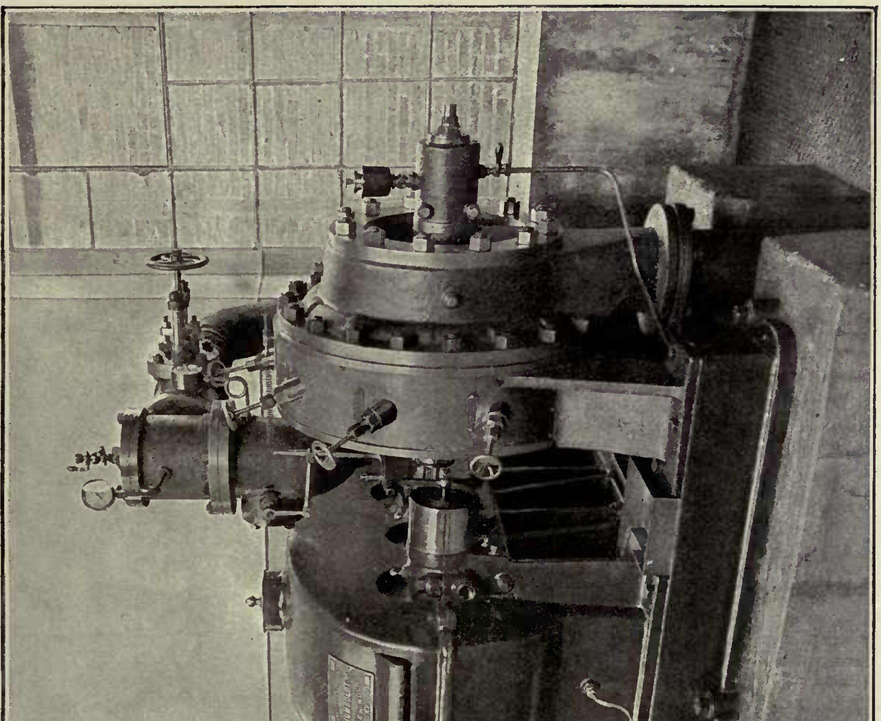


Fig. 594.
Side View of Turbine-Dynamo of 100 H.P.

Steam pressure in front of the nozzles in lbs. per sq. inch.	Temperature of the steam at the inlet valve. Fahrenheit.	Vacuum in the discharge from the turbine case in inches of mercury.	Number of steam nozzles opened.	Number of revolutions per minute of the driving shaft.	Brake horse-power.	Consumption of steam per E.H.P. per hour.
192·7	453·7	$27\frac{1}{4}$	7	772	307·8	13·96
196·3	437·0	$27\frac{5}{8}$	6	762	259·0	14·46
196·3	440·6	$27\frac{1}{2}$	5	767	219·9	14·20
196·3	437·0	$27\frac{5}{8}$	4	775	175·0	14·28
190·6	426·2	$27\frac{7}{8}$	3	777	123·3	14·72
196·3	390·2	$28\frac{1}{8}$	2	775	75·2	17·02
213·3	388·4	$28\frac{1}{2}$	1	773	31·9	21·29

In order to test the speed-governing arrangements of this turbine, the latter was suddenly relieved from the full load of 307·8 H.P. The number of revolutions per minute of the tachometer thereupon rose from 750 to 780, which is equivalent to an increase of speed of 4 per cent. The vacuum in the turbine case fell from $27\frac{1}{2}$ ins. to $24\frac{1}{4}$ ins., *i.e.* by 12 per cent. Finally, the steam pressure at the nozzles fell from 198·5 lbs. to $88\frac{1}{4}$ lbs. per sq. inch, *i.e.* by 55·5 per cent.

From the report of October 13, 1903, on the trials of the turbine of 200 H.P. installed in Lille, the following may be taken. The steam which is generated in a Belville boiler was first dried; its pressure was measured from a manometer fixed above the nozzle channel and the vacuum at the discharge branch of the turbine. The turbine drove two "Breguet" dynamos which were directly coupled to it, the electric work of which was exactly determined by ampère and voltmeters. The consumption of steam, which varies with the number of nozzles that are opened, depended, further, on the pressure. The proportion borne by consumption to pressure of steam had been determined at a preliminary trial.

For the coefficients of performance of the dynamos the figures were taken that were given by Messrs. Breguet, *i.e.* for more than 130 kw., up to 0·88; for 130 kw., 0·885; for 104 kw., 0·875; and for 65 kw., 0·850. All losses are here given for the installation up to the switchboard. There were measured:—227·23 volts, 557·5 ampères, 126·680 kw., 194·48 E.H.P.; number of nozzles opened, 6; nozzle entrance pressure, 104·79 lbs. per sq. inch; vacuum, $25\frac{1}{4}$ ins.

Total consumption of steam per hour . . .	3201·5 lbs.
Consumption of steam per E.H.P. per hour . .	16·46 lbs.
Revolutions per minute	900

Under the above conditions the turbine worked well, and the consumption reached the guaranteed minimum exactly. The following table shows how the engine worked under different loads:—

Load.	Volts.	Ampères.	Kilowatts.	E.H.P.	Nozzles opened.	Nozzle entrance pressure.	Vacuum.	Total consumption of steam per hour.	Revolutions per minute.	Consumption of steam per E.H.P. per hour.
Overloaded	232·4	684·0	158·961	245·53	7	111·18	$24\frac{3}{4}$	3926	900	16·00
With $\frac{5}{6}$ load	208·8	531·9	111·019	172·20	5	109·94	$25\frac{5}{8}$	2778	900	16·12
„ $\frac{2}{3}$ „	217·8	436·7	95·126	148·56	4	118·76	26	2372	903	15·98
„ $\frac{1}{2}$ „	230·5	274·7	63·318	101·21	3	108·80	$26\frac{3}{8}$	1649	904	16·29

From these figures it appears that the consumption rose but little as the load was decreased. In order to determine the variation in the revolutions per minute when the load was suddenly altered, several experiments were made. When the turbine was worked at successively decreasing powers from 130 kw. to 0, the revolutions per minute rose from 900 to 930; the difference in speed between the full and empty loads thus amounted to 3 per cent.

Laval turbines in connection with dynamos have also been made use of in railway work in a peculiar manner. In Fig. 595 (see Plate XII.) an engine combination of this kind is shown mounted on the boiler of the locomotive of an express train; it receives steam directly from the boiler and provides the lighting installation of the train with electric energy. The turbine in this case develops 20 H.P. There are 22 arrangements of this kind at work.

Up till the end of 1901, 3000 Laval turbines of from 3 to 300 H.P. have been constructed with a total horse-power of 85,000.

That attempts are made to apply the Laval turbine to the practical development of greater powers is not surprising. To the attainment of this end the attempt will be made to reduce the number of revolutions per minute of the working wheel to such an extent that the insertion of gearing between the motor and the working machine may be dispensed with. This problem will clearly be made possible by the introduction of a means of withdrawing the energy stage by stage.

The *Curtis turbine* is an American invention. It makes use of the velocity of the steam, which is relieved of its pressure in the expansion nozzles, by stages.[1] It is manufactured in the United States by the *General Electric Co.* of Schenectady, in England by the *British Thomson-Houston Co.*, and in Germany by the *Allgemeine Elektrizitäts-Gesellschaft*, for the direct driving of dynamos. As far as can be ascertained, it is made in sizes of 500 kw. and upwards, with the shaft vertical. The expansion nozzles are worked into a casting in groups, and cover only about one-sixth of the periphery of the first working wheel. They are closed one by one, according as energy is required, by valves which are adjusted separately by steam steering engines; the entrance of the steam to the latter again is governed by electrically influenced

[1] Compare p. 90.

PLATE XII.

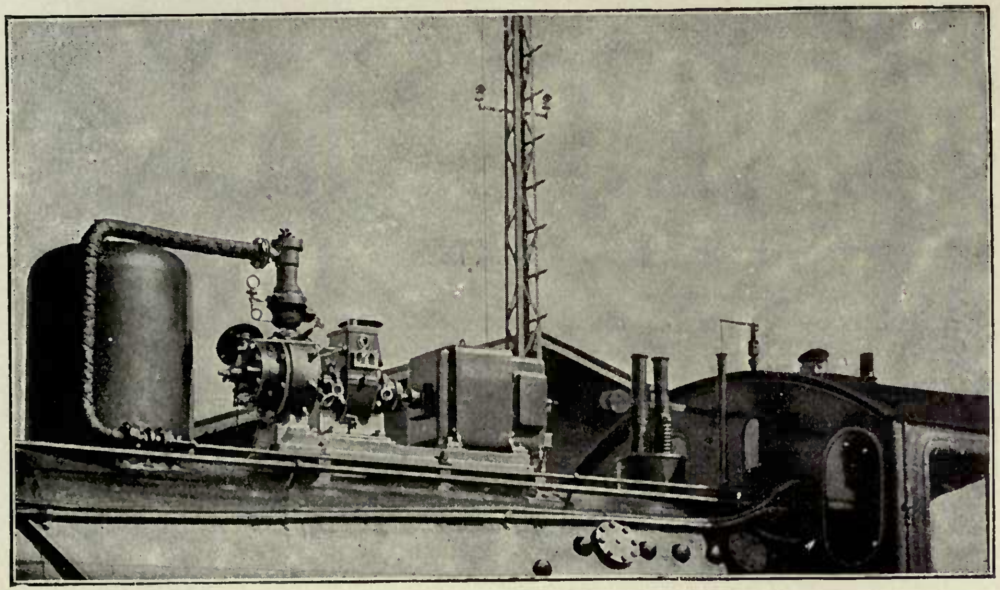


Fig. 595.
Laval Turbine with Dynamo on Express Locomotive.

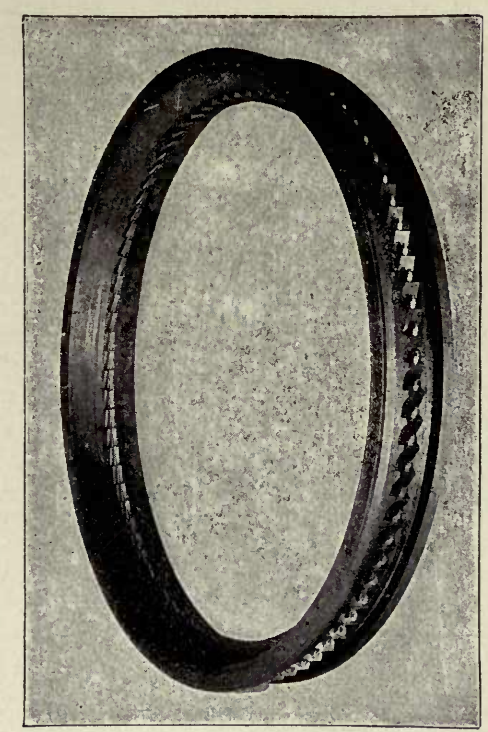


Fig. 608.
Vane Wreath of the Moabit Turbine of 2000 H.P.

spindle valves. The intermediate leading appliances are screwed into annular sections of the case. Between the guide and working wheels a clearance space remains of

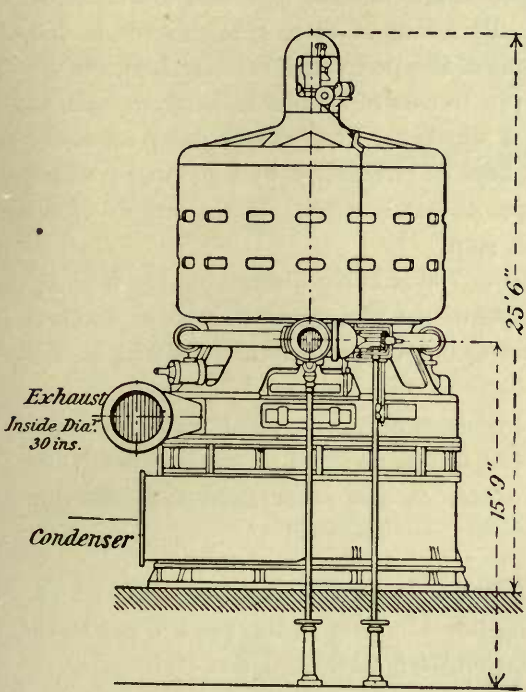


Fig. 596.

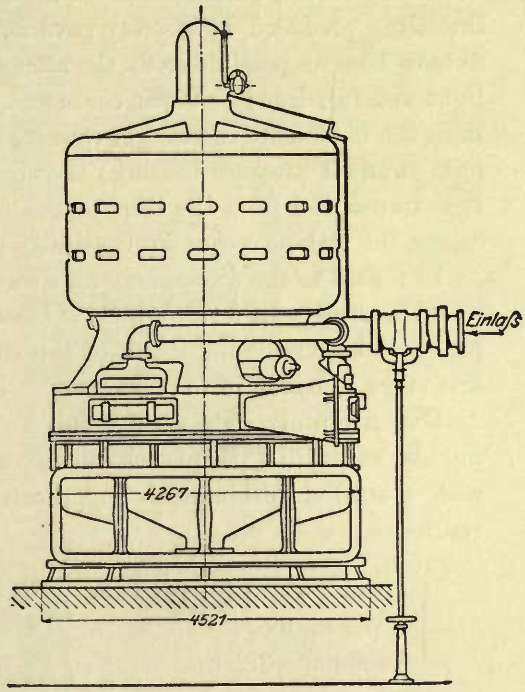


Fig. 597.

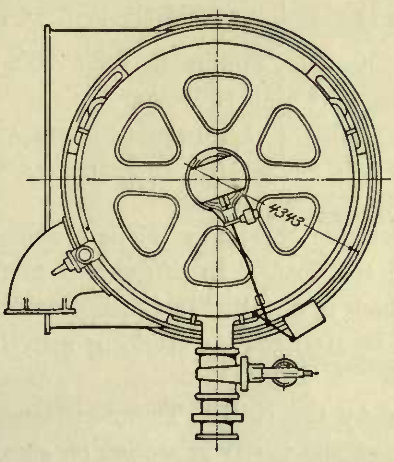


Fig. 598.

from 0·04 to 0·1 of an inch. Each working wheel is a disc of about 1 inch in thickness, in the periphery of which steam channels are cut by means of special grooving machines. According to size, a wheel is provided with from 200 to 280 vanes, which the machine produces in one day.[1] A ring subsequently shipped on closes the channels, in the radial direction, at the outside. Besides the influencing of the nozzles by valves, the regulation of the engine is also effected by means of additional throttling produced by a safety governor. By means of these appliances it is said to have become possible to fix the alterations of the peripheral velocity between the light and full loads to 2 per cent., and, even in case of sudden relief from load, to limit the immediate increase of velocity to 4 per cent. The lower footstep takes the form of an oil pressure bearing; the oil is kept in circulation by a pump, by which it is transferred from the step to the upper throat bearing. If the supply of oil ceases, the turbine comes automatically to a stop.

In regard to the dimensions, for a turbine of 700 H.P. the diameter of the working wheel is about 4 ft. 6 ins., which is associated with a peripheral velocity of 420 feet per second. The radial depth of the steam channels increases from $\frac{3}{4}$ inch in the first stage to 1 inch in the last one.

Experimental trials with a 600 kw. engine running at 1500 revolutions per minute, and with a peripheral velocity of 420 feet per second on the one hand, and with a smaller turbine with superheated steam on the other, gave the following results:—

I.	Steam pressure = 144·1 lbs. per sq. in. Vacuum = 38½ ins.	Powers in kw.: 800, 550, 425, 250, 100, and 75. Consumption of steam in lbs. per kw. per hour: 19·0, 19·3, 19·5, 21·0, 24·5, and 26·5.
II.	Steam pressure = 144·1 lbs. per sq. in. Vacuum = 38½ ins. Superheating about 149° F.	Powers in kw.: 800, 550, 425, 250, 100, and 75. Consumption of steam in lbs. per kw. per hour: 17·3, 16·8, 17·1, 18·4, 21·5, and 22·7.
III.	Steam pressure = 206 lbs. per sq. in. Vacuum = 38½ ins. Superheating about 149° F.	Powers in kw.: 700, 500, 400, 200, 150, and 100. Consumption of steam in lbs. per kw. per hour: 16·0, 16·2, 16·5, 18·0, 19·0, and 21·5.

These values show a certain constancy of the consumption of steam in case of considerable variation of the load. In Figs. 596 to 598 a turbine dynamo of 5000 kw. is illustrated, which is installed in Chicago. The dynamo is placed above the turbine. The latter is arranged for working with the exhaust and with the condenser.

An arrangement made by the British Thomson-Houston Co. is shown by Fig. 599 partly as an outside view and partly in section on a scale of 1 : 30. This turbine dynamo is arranged for 1500 kw., 1000 revolutions per minute, 1100 volts, and a frequency of 50 periods per second. It contains four working wheels, each with two velocity stages, and each working in a separate chamber. The partition walls, which

[1] Emmet, in the *Electrical World* of April 11, 1903; *Zeitschrift d. Vereins deutscher Ingenieure*, Vol. 47, p. 1120.

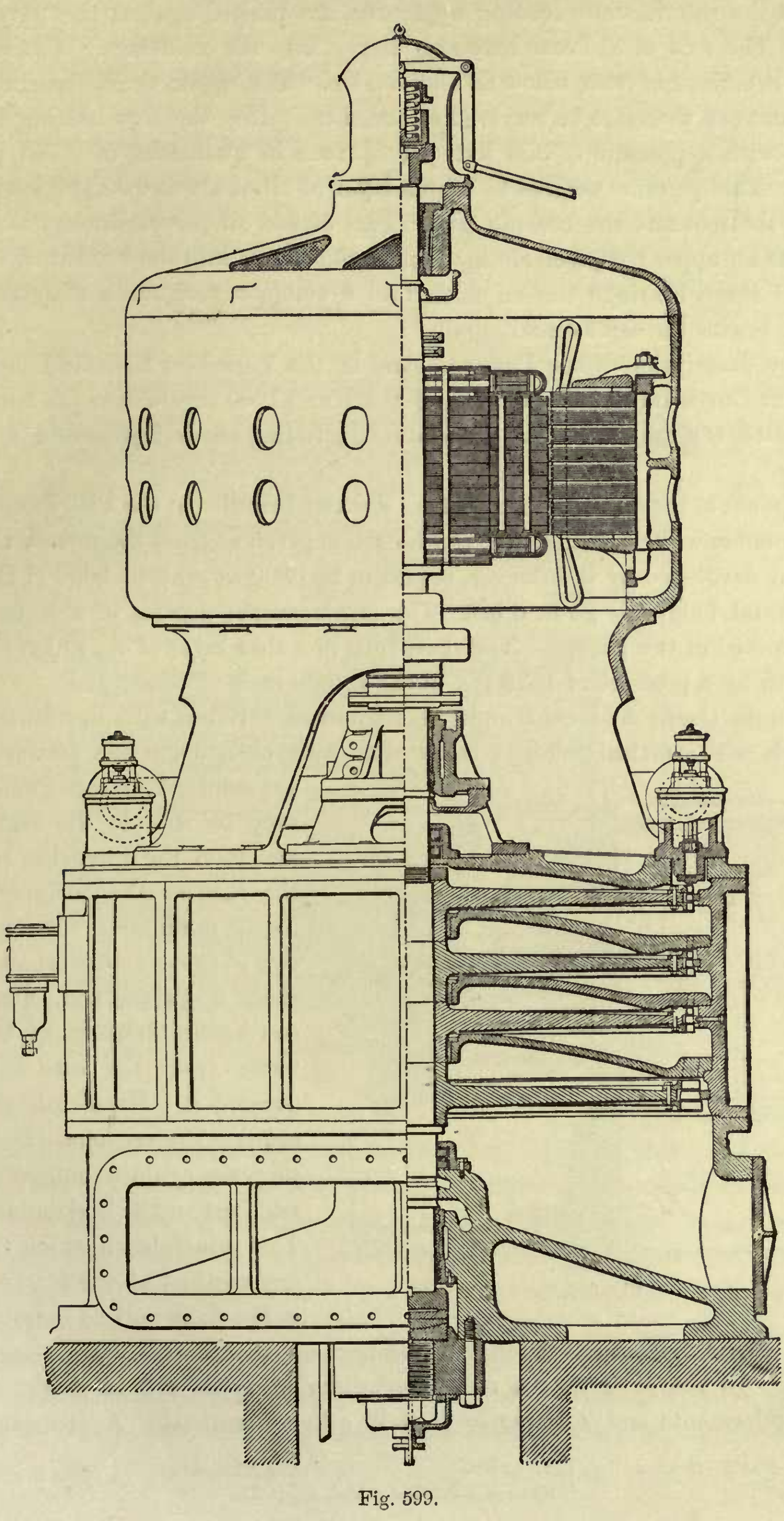


Fig. 599.

also bear the intermediate leading apparatus, are packed against the naves of the wheels. The foot of the case may also merge into the condenser. The lower foot bearing is accessible from below and adjustable. The upper throat bearings are in halves, and can therefore be removed at the sides. Into the foot bearing the oil is pumped with a pressure of 350 lbs. per sq. inch in quantities of 0·775 pints per minute. This pressure suffices to keep a layer of oil of about 0·005 in. between the pivot at the foot and the bearing disc. Part of the oil passes through a reducing valve into an upper receptacle, from which it runs down into the two throat bearings. From all three bearings the oil flows into a common receptacle, and after being cooled it begins the same circuit again.[1]

In the description of the Power Central of the Yorkshire Electric Power Co., in which four Curtis turbine dynamos of 1500 kw. and 1000 revolutions per minute each are installed, the weight of a set of turbines including current generator is given at 57·6 tons.[2]

The power of the second Curtis turbine dynamo installed at the Fisk Street Station of the Commonwealth Electric Co. in Chicago is given at 5000 kw. nom.[3]; the power at present developed by it, however, is said to be 6800 kw. (about 9000 H.P.).

The total height is 25 ft. 6 ins. The impingement nozzles of this turbine are also composed of two groups. The shaft runs in a thin layer of oil, which is kept in circulation by a pressure of 1320 lbs. to the square inch.

Although *Curtis* first contemplated the use of turbines with horizontal shafts, only such with vertical ones in connection with electric-current producers have apparently been constructed. This may be due to the fact that it has been the intention to reduce the base of the turbine dynamo to a minimum. However, the ease of fixing vertical shafts favours their adoption, and, finally, the convenient delivery of the waste water from the case may have decided it. Meanwhile the *Allgemeine Elektrizitäts Gesellschaft* proposes again to construct Curtis turbines with horizontal shafts. The principle on which the working method of the Curtis turbine is based, as well as details having for their object improvements in its operation or construction, have already been discussed. In course of time a number of different types will no doubt be built, when requirements and experience lead in other directions. As an example of

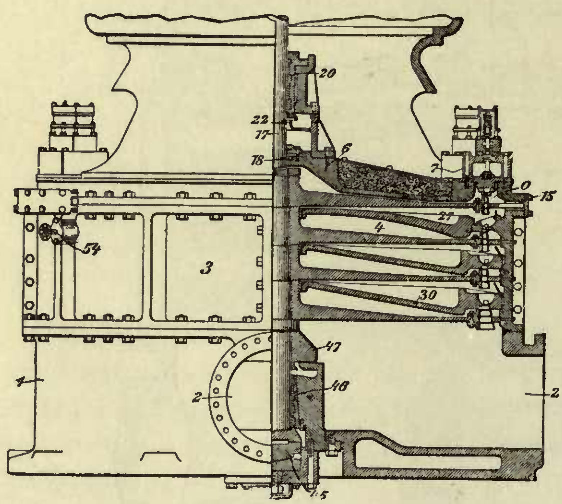


Fig. 600.

[1] *The Electrician*, Vol. LII., p. 160. [2] *The Electrician*, Vol. LII., p. 242.
[3] *Western Electrician*, Vol. 34, p. 87.

a recent construction (1903), that of the *American Continental Turbine Company of Jersey City* (New Jersey, U.S.A.) may be mentioned.[1]

Fig. 600 shows a general arrangement of the turbine chamber, whilst Fig. 601 shows details of the same. The turbine chamber is fixed to the hollow machine frame 1, with the branch 2 leading to the condenser (or exhaust). The wall of the chamber consists of segments 3, which are screwed together. The chamber is closed at the top by means of a cover 6, above which the dynamo casing is fixed, and which is provided with two annular clearance spaces 0 lying symmetrically to the turbine shaft. These annular spaces are closed by means of the valve box 7, from which the

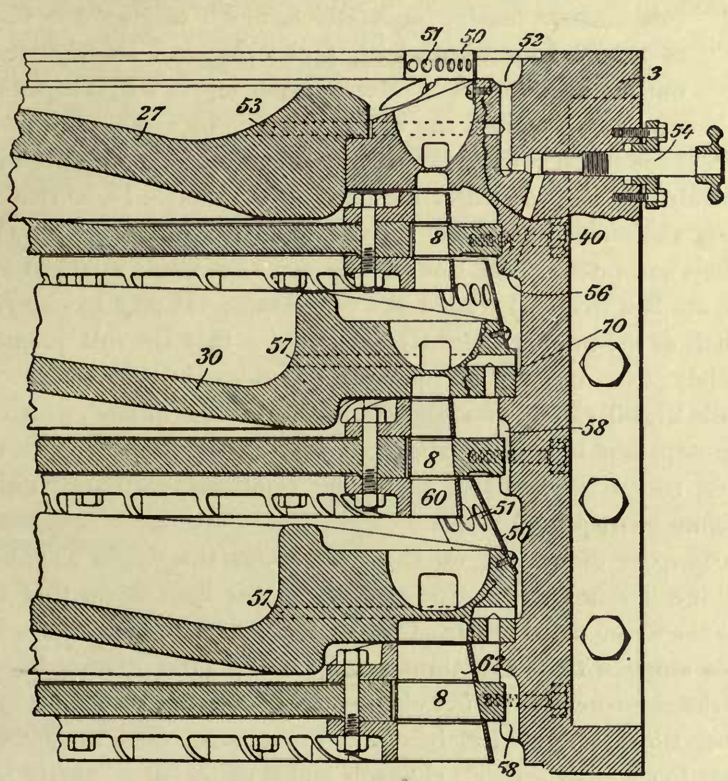


Fig. 601.

steam-supply nozzles lead into the turbine chamber. The turbine itself shows four stages of pressure, each with two velocity stages, besides four working wheels 4, each with two vane segments 60 covering each other, and fixed below the corresponding valve boxes. The size of the segments, *i.e.* the number and size of the steam channels, decreases from stage to stage in a degree corresponding with the expansion, but also, of course, in due relation to the retardation of the steam at the passage ways. The naves of the working wheels rest on collars forged on the shaft 17. Owing to the subdivision of the stages of pressure, each working wheel must run in a separate chamber. For this purpose, partition walls are inserted, the uppermost of

[1] F. P. 331,539.

which (27) is thicker, but smaller in diameter than the lower ones (30). On the other hand, the group of nozzles of the second pressure stage is carried by the wall of the case, while the groups of nozzles belonging to the lower stages are fixed to the edges of the partition walls. The latter are made tight against the naves of the working wheels, with their edges lying in the intrusions of the wall of the case, where they are held in place by their own weight. In the case of turbine dynamos with horizontal shafts, the bolts 71, pressing the partition walls against the shoulders of the wall of the case, should be provided with safety nuts (Fig. 602). Annular grooves in this wall take up the segments of the intermediate leading apparatus 8, which connect the velocity stages of the working wheels with each other, and are tightened from the outside by means of countersunk pin screws 40. The screw heads are made tight against the wall of the case by means of conical shoulders, so as to prevent the steam at these points from escaping. Moreover, the intermediate leading apparatus of the first wheel are fixed to the plates 15, so that they can be adjusted from the outside after the turbine has been mounted. The vanes of the working wheels and intermediate leading apparatus are firmly attached to the bodies, whilst strips are laid (riveted) around the vane heads. It may be observed that the steam channels at the inlet are slightly enlarged, so that the full quantity of steam can pass safely from channel to channel. Besides, by the construction of the partition walls 27, 30, suitable annular spaces have been formed, which to a certain degree form separate impingement spaces for the nozzles. Moreover, the radial strengthening ribs on the rotating parts offer injurious ventilating resistances, and should therefore be dispensed with.

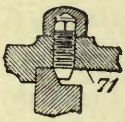


Fig. 602.

For the purpose of leading off the waste water, the single chambers are connected with the hollow foot 1. The assumption has been made that the drops of water which the steam delivered from the second velocity stage of the first working wheel carries along with it are thrown into the annular groove 52 through the holes 51 of the sheet-iron plates 50, whence they enter the collecting channel 70 of next chamber through a vertical bore and the channel 56. Thus the water flows from chamber to chamber through channels and slots 58 till it finally arrives at the exhaust space. The water accumulating on the walls 27 and 30 is likewise delivered into the collecting channels through the channels 53 and 57. With the exception of those of the last stage, all impingement spaces are protected by the perforated sheet-iron plates 50. The width of the passage of the delivery channel from the first chamber can be regulated by means of a valve spindle 54. It is also of advantage to keep the channel filled to a certain degree with water, in order to prevent the free passage of the steam into the exhaust.

The cover 6 of the case is fitted with a stuffing box, consisting of two carbon rings 18, held apart by means of a fixed plate. Above the stuffing box there is a simple collar bearing 20, below which there is a vessel 22 to catch the lubricating material for the bearing. The seating frame 1 is likewise provided with a collar bearing 46, whilst the footstep 45 carries the shaft. On the latter there is a ring 47, by means of which the shaft is supported on the lower collar-bearing body, in case the footstep bearing has to be removed.

The nozzles of each pressure stage are divided into two groups, each of which forms a separate casting (see Fig. 603). The vertical inlet for each nozzle is of circular section, which merges into a rectangular one.

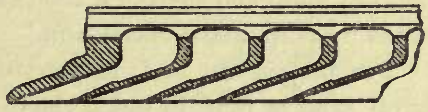
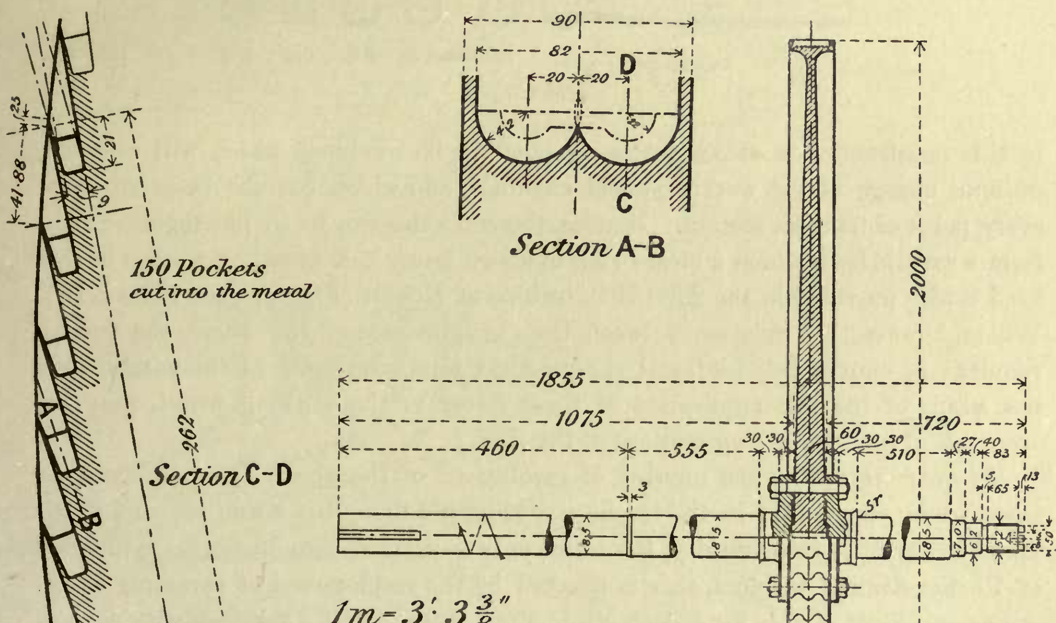


Fig. 603.

In the case of *turbines* of the *Riedler-Stumpf type,* a solid steel disc of large diameter (6 to 10 ft.) is used for the working wheel. With 1500 to 3000 revolutions per minute (which can be practically utilized without the use of gearing), this disc possesses a high circumferential velocity. The latter is aimed at in order as far as possible to utilize the conditions of the flow of steam in an effective manner, if possible in a single wheel. In designing the cross section of the disc, special attention has been paid to the effects of the centrifugal forces, these being strongest in the centre, and diminishing towards the circumference. Wherever admissible, the



Figs. 604–606.

disc is made without a bore for the shaft. The attachment is made by flanges, with which the parts of the shaft abut against the centre of the disc. If the insertion of the shaft proves to be a necessity, the disc is provided with a long nave. The material used consists of nickel-plated steel with a factor of safety of 5 ; but ingot steel of a tensile strength of 32 tons per square inch may also be used for this purpose.

The designers have succeeded in reducing the inaccuracy of the centre of gravity down to 0·0004 in., so that the exceeding of the critical speed does not become a danger. In order to ensure an automatic adjustment to the centre of gravity when the dynamic equilibrium is destroyed by an excess of speed over the normal one, the shaft is made flexible in the long bearings. For the reception of the steam jets, pockets are slotted out of the circumference of the wheel.[1] Figs. 604 to 606 show working wheel and shaft of a 2000 H.P. turbine of the Riedler-Stumpf type, adopted in the "Elektrizitätswerk Moabit."

For the purpose of impingement, an expansion nozzle, similar to that of the de Laval type, is used, in which the conversion of the steam pressure into velocity is effected. The initial circular cross section of the nozzle merges into a rectangular one, which is followed by a piece with a uniform section (see Fig. 607). By means

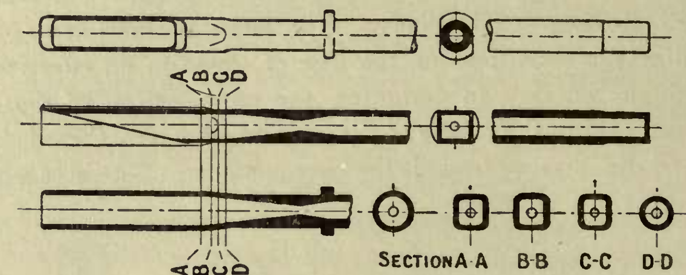


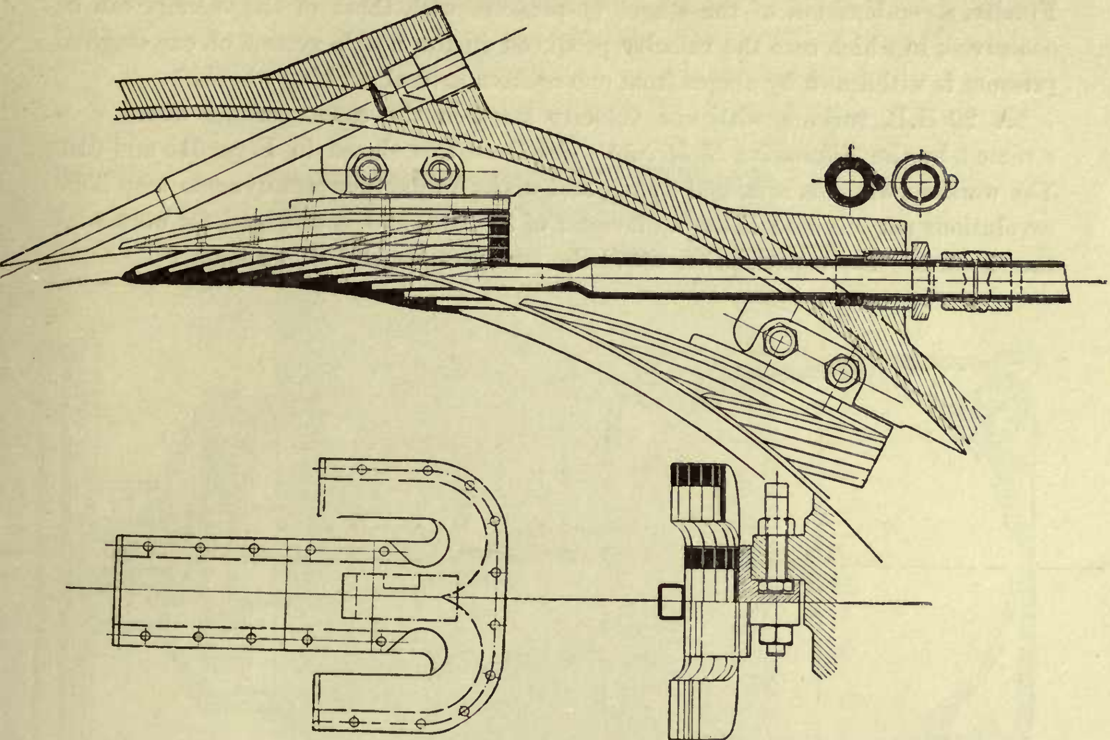
Fig. 607.

of this construction a steam jet is intended to be produced which will ensure a uniform charge of the working-wheel channels, and which has the same speed at every point of its cross section. Besides, these nozzles can be so put together as to form a wreath, from which a closed ring of steam issues.[2] A wreath of nozzles of this kind is also provided in the 2000 H.P. turbine at Moabit (Fig. 608, see Plate XII.). Where, however, the relation between the circumference of the wheel, the energy required in case of full load, and the practical minimum width of the nozzles does not admit of the full application of these latter to the working wheel, they are arranged in groups forming sections of the ring.

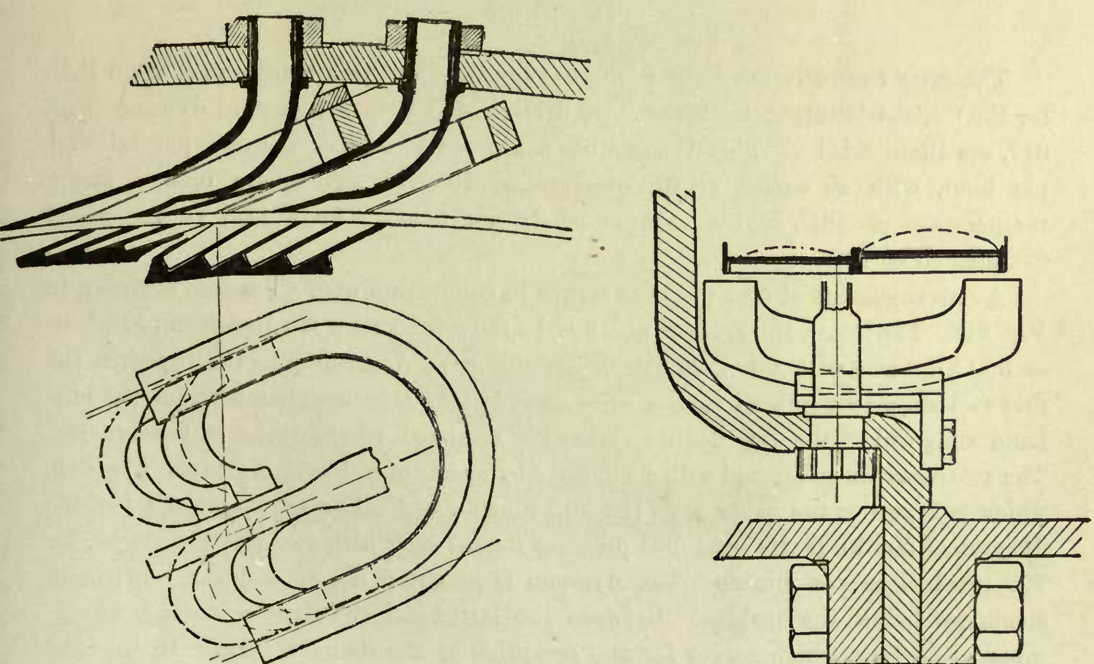
In order to reduce the number of revolutions of the wheel, the energy can be supplied by stages. If in the impingement nozzle the entire steam pressure available be converted into velocity, the latter only is divided into stages. In the case of Riedler-Stumpf turbines, this is effected by the employment of reversing vanes. An arrangement of this kind is shown in Figs. 609 to 611 for turbines with double-vane wreaths, for the case in which the stages are to be arranged in one working wheel only. If two working wheels are at disposal, a loop reversal is adopted, by means of which the steam jet is reversed while preserving the same direction of rotation[3] (Figs. 612 to 614).

Meanwhile, a division, according to pressure stages, can also be effected by the placing of a number of working wheels in separate chambers, corresponding with the number of stages, each of which is fitted with its own impingement nozzles. In this case only part of the steam pressure in each wheel is converted into velocity.

[1] See p. 258. [2] See p. 251. [3] See p. 244.



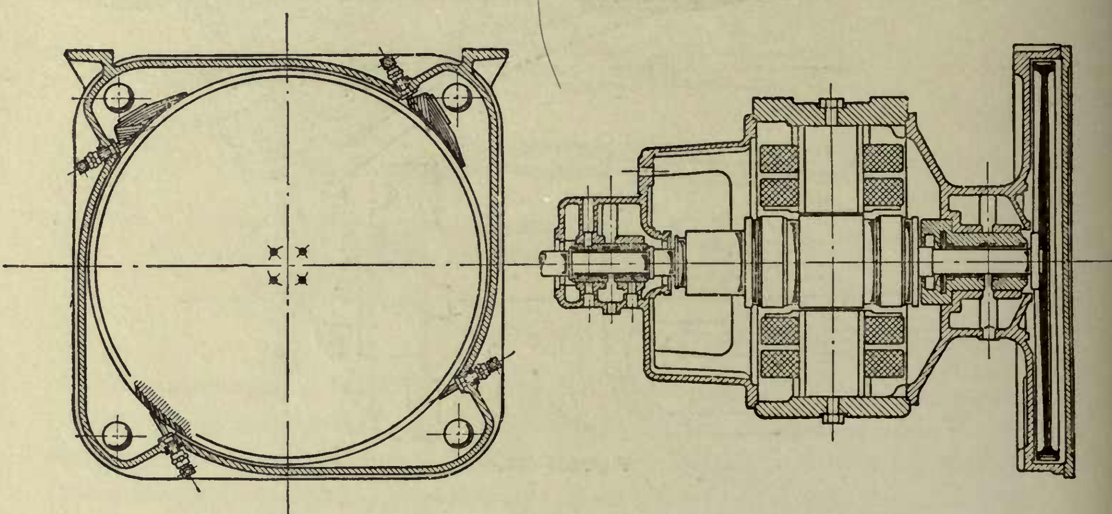
Figs. 609–611.



Figs. 612–614.

Finally, a combination of the stages of pressure with those of the velocity can be conceived, in which case the velocity produced in the nozzle system of one stage of pressure is withdrawn by stages from one or from several working wheels.

A 20 H.P. turbine, with one velocity stage in the same working wheel, constructed by the *Allgemeine Elektrizitäts-Gesellschaft*, is shown in Figs. 615 and 616. The working wheel is arranged flying, and works a direct-current dynamo with 3500 revolutions per minute; it has a diameter of $31\frac{1}{2}$ ins., and is arranged for work with the exhaust. The consumption is $37\frac{1}{2}$ lbs. of steam per E.H.P. per hour.



Figs. 615, 616.

The same company has built a single-stage Riedler-Stumpf turbine of 2000 H.P. for the "Elektrizitätswerk Moabit," for driving a three-phase current dynamo (Fig. 617, see Plate XIII.). This dynamo has consumed 17·64 lbs. of steam per kilowatt per hour, with an excess steam pressure of 195 lbs. per square inch, a steam temperature of 561°, and a vacuum of 82 per cent. The weight of the wheel is stated to be 1874 lbs.

An arrangement of four pressure stages in connection with a dynamo is shown in Fig. 618. The latter, the scale of which is 1 : 30, represents a Riedler-Stumpf turbine of 500 kilowatts with 500 revolutions per minute. According to the drawing, the first two stages are placed in the same case, but in separate chambers, on the left-hand side, whilst the two deeper stages are arranged to the right of the dynamo. The two cases are connected with each other by means of a circulating pipe. Fig. 619, which is drawn to the same scale (1 : 30), shows a vertical arrangement of a turbine dynamo of 500 kilowatts with four pressure stages, each with two velocity stages, for 750 revolutions per minute. The dynamo is mounted above, and the centrifugal condenser below, the turbine. Between the latter and the dynamo there is an oil-pressure bearing, which serves for the reception of the rotating parts. In the case of working machines (dynamos) placed below the turbine, the *Vereinigte Dampf-*

PLATE XIII.

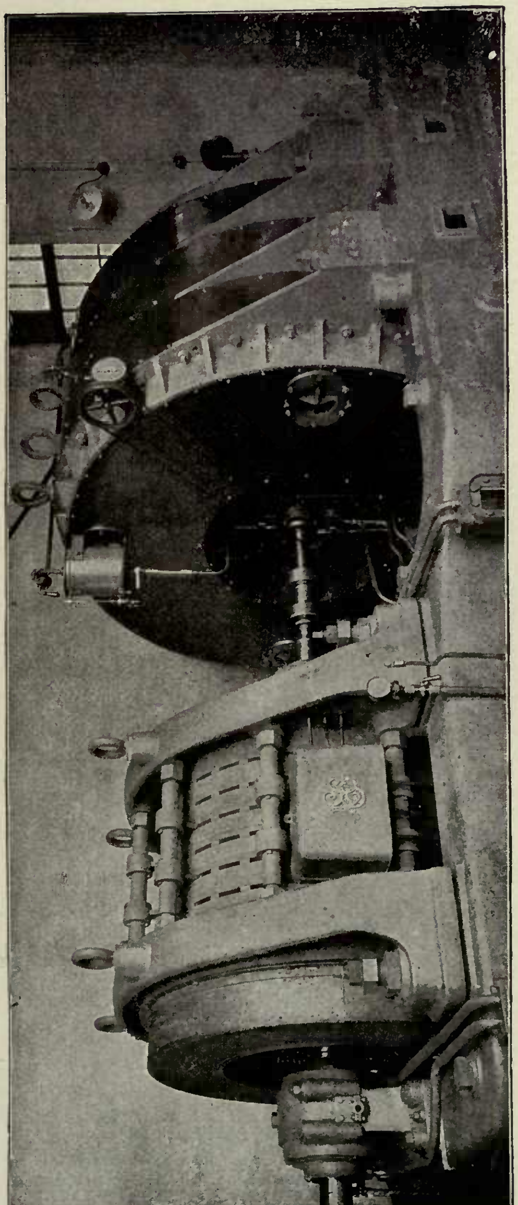


Fig. 617.
Riedler-Stumpf Turbine of 2000 H.P.

turbinen-Gesellschaft m. b. H. arrange the condenser between the driving and the working engines, so as to protect the latter from the heat which is given off by the former.[1]

The "Elektra" steam turbine, constructed by the *Gesellschaft für Elektrische*

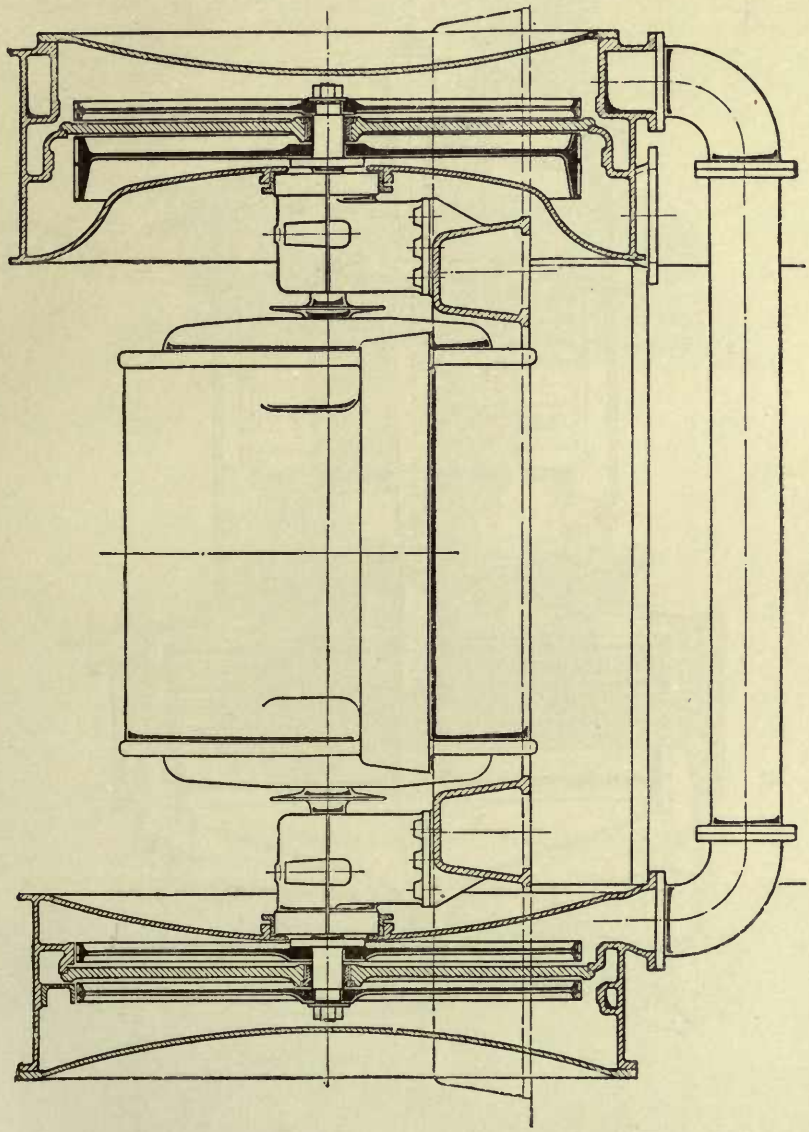


Fig. 618.

Industrie, is, as far as is known, a radial velocity turbine, the (two) expansion nozzles of which impinge a working wheel from without. The velocity stages which follow are arranged in the working wheel itself, the latter being repeatedly impinged upon by the steam both from within and from without.

[1] D. R. P. 153,252.

Amongst other proposals, that of *Johansson*[1] may be mentioned. According to the latter, the turbine is to be connected with the dynamo in such a manner that the exhaust steam of the former can, provided its density be lower than that of the outer air, be utilized for conveying the heat and reducing the work at empty load of the current producer. The latter is either fixed in a separate case, connected

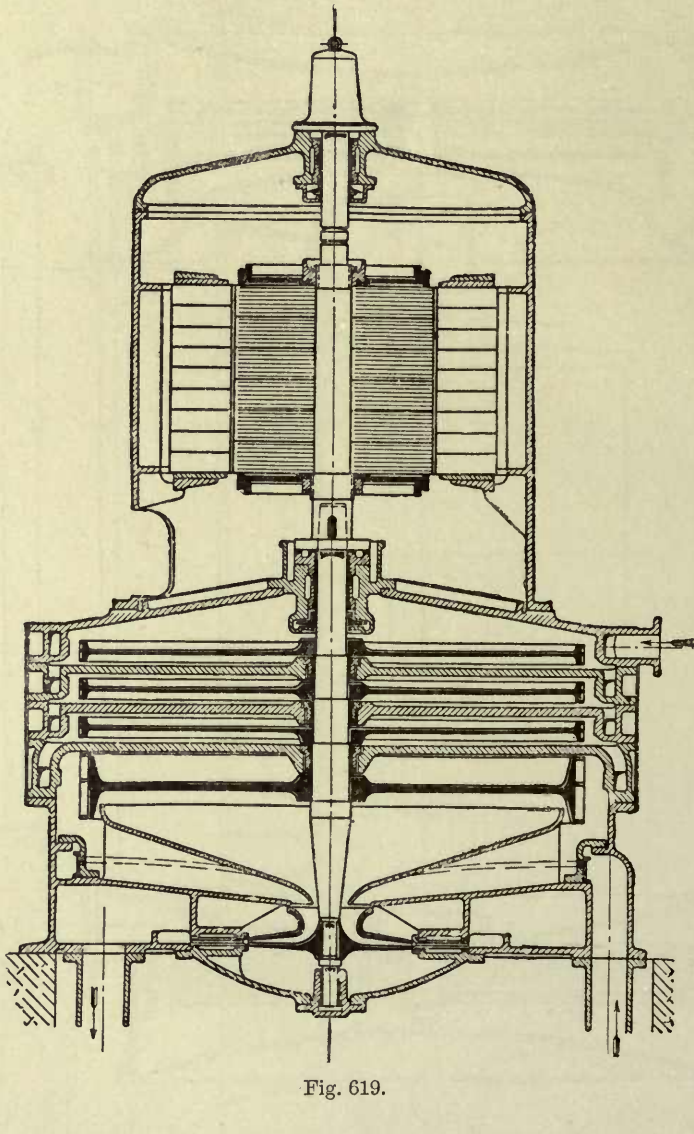


Fig. 619.

with the turbine chamber by means of a channel, or chamber and casing form a continuous space. In both cases the exhaust steam of the turbine sweeps over the working space of the dynamo. In the case of unipolar machines, simplicity is attainable, in so far as the working wheel of the turbine can be so constructed as to form the armature of the single-pole machine.

[1] D. R. P. 133,041.

PLATE XIV.

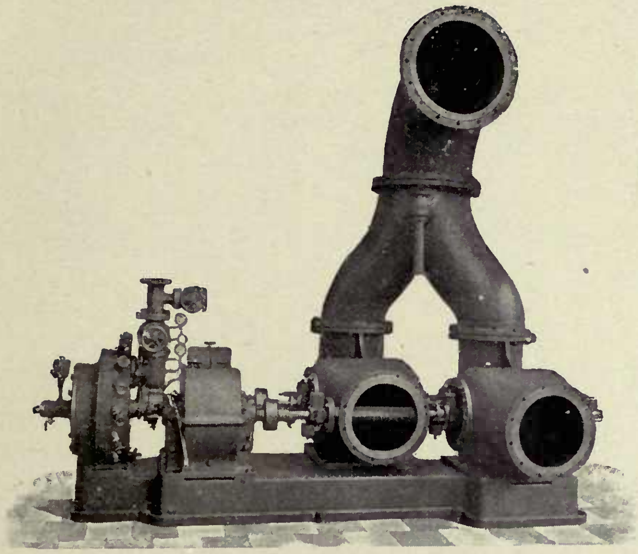


Fig. 620.
Laval Turbine connected with a Centrifugal Pump for Water.

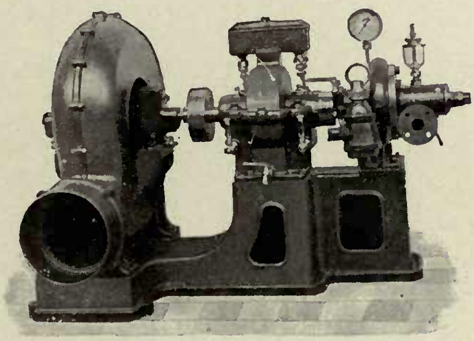


Fig. 621.
Laval Turbine connected with an Air-pump.

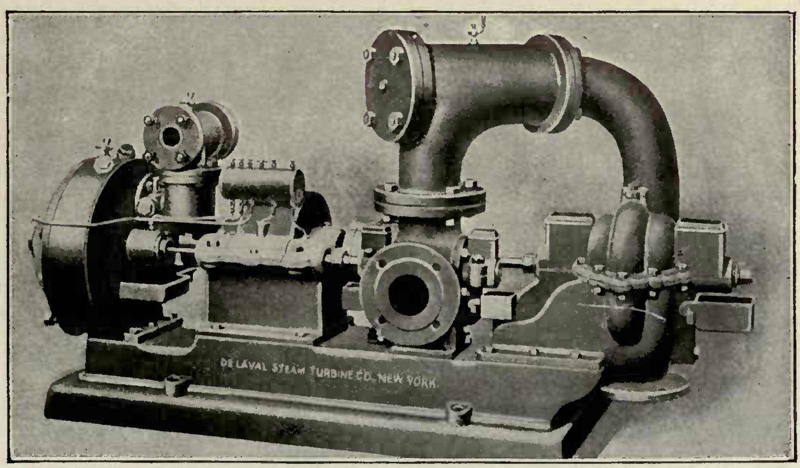


Fig. 622.
Laval Turbine to raise 224 gallons of water to a height of 197 feet.

XXII

TURBINE PUMPS AND TURBINE BLOWERS

Recent efforts to replace piston pumps by centrifugal pumps may to a great extent be attributed to the success obtained with the steam turbine. Just as the latter ensures the development of a large power in a small space, so does also the high-speed centrifugal pump perform a large amount of work on small dimensions. The uniformness of the movements of the dynamo electric motor and of the working engine enables transmission appliances to be dispensed with. It will, of course, have to be considered whether liquid or gaseous substances are to be dealt with, and whether a simple transmission of power or a greater or less increase of pressure is to be effected.

A discussion on the general question of centrifugal pumps will not here be entered into, but mention may be made of a few combinations of pumps with turbines.

Thus Fig. 620 (see Plate XIV.) represents a de Laval turbine with centrifugal pump for water, while Fig. 621 (see Plate XIV.) shows the same turbine connected with an air-pump. These types are made by the *Maschinenbau-Anstalt Humboldt.*

Lindmark[1] (1900) has connected the Laval turbine with a centrifugal compound pump for great heights of throw by following the principle of drawing water by means of a pump of slow speed and subjecting it to a medium pressure. It is then conveyed to a high-speed pump, which completes the increase of pressure required. For this purpose Lindmark has connected the shaft of the small high-pressure pump with that of the working wheel of the turbine. The shaft of the larger low-pressure pump is then connected with another shaft, which, by the help of ordinary gearing, is made to run at a slower speed. According to this arrangement, the high-pressure pump must make 14,000 revolutions and the low-pressure pump 1500 revolutions per minute, when the turbine wheel has a diameter of 19¾ ins. At a practical trial the low-pressure pump drew water without difficulty, the set of pumps raising 770 gallons of water to a height of 492 ft. in a minute.

For great heights of delivery, the construction shown in Fig. 622 (see Plate XIV.) has proved to be very efficient. In the example given the high-pressure turbine pump delivers 35·32 cub. ft. per minute to a height of 197 ft. Its low-pressure pump wheel is driven by the Laval gearing; it makes 1000 revolutions per minute, and conveys the water to the high-pressure pump wheel, which is

[1] A. P. 706,187.

mounted directly on the turbine shaft, and therefore makes 20,000 revolutions per minute.

In the mines at Lens (France) a similar compound pump, driven by a Laval turbine of 150 E.H.P., serves to draw water from a mine. The high-pressure pump mounted on the shaft of the working wheel makes 13,000 revolutions per minute, and the low-pressure pump, driven by the connecting gear, 650 revolutions per minute. The latter draws water from a depth of about 10 ft., and conveys it to the high-pressure pump with 32·8 ft. pressure, the installation having been made at the bottom of the pit. Thus in ordinary work the pump delivers 3532 cub. ft. of water per hour to a height of 853 ft.[1]

If the turbine shaft does not make an excessive number of revolutions in the unit of time, the pump wheel may be fixed directly to it, where it enters the pump casing. Such an arrangement is, for instance, found in the turbine blower of *Hoehl, Brakell,* and *Günther,*[2] who have fixed the blower wheel below a two-stage radial turbine with vertical shaft.

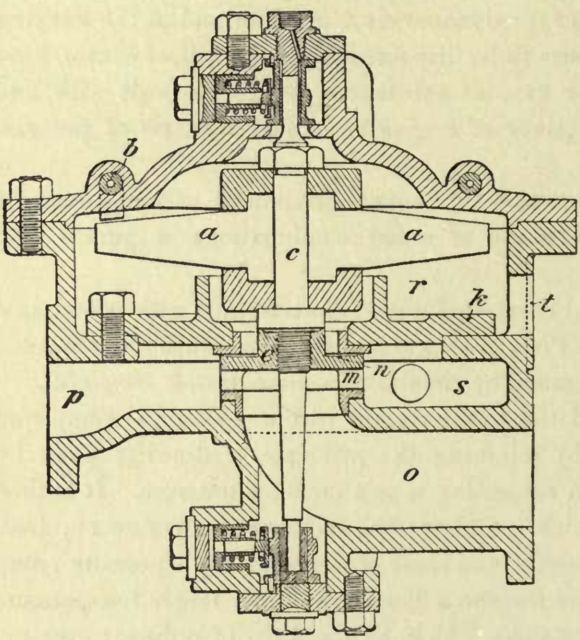


Fig. 623.

Nedden[3] constructs the blower wheel, the shaft of which is likewise vertical, as a turbine (undershot paddle wheel), the vanes of which rotate in an annular chamber.

On the other hand, in view of the high speed of the turbine shaft, *Parsons*[4] has proposed screws of small diameter for the pump. In this case the shaft has two sets of bearings in the long pump casing. Amongst other fields of application for his turbine blowers, Parsons contemplated that of fire engines, in which they were to supply the draught.

The turbine pump of *Krank*[5] (1901) is intended to supply water under pressure (Fig. 623). The turbine *a*, which is axially impinged from the nozzles *b*, is mounted on the shaft *c*. On the latter, *i.e.* in the space *s*, which is separated from the steam space *r* by means of the partition wall *k*, a centrifugal wheel *l* is fixed, to the lower side of which vanes *m* curved obliquely backwards are attached. The fluid, to be set under pressure, is conveyed through the tube *o* into the pressure space *s* by

[1] Sosnowsky, "Pompe centrifuge à haute pression syst. de Laval," *Mém. d. l. Soc. d. Ing. Civ. de France*, 1904, p. 233.

[2] E. P. 2429 of the year 1863.

[3] E. P. 2609 of the year 1880.

[4] E. P. 3024 of the year 1895.

[5] D. R. P. 135,555.

means of the vanes *m* of the quickly running centrifugal wheel *l*. The fluid is then subjected to a high pressure in consequence of the high speed of the wheel. From the pressure space *s*, to which a safety valve may be attached, the fluid under pressure is conveyed through the opening *p* of a secondary engine or to a water wheel fixed to the shaft of the latter. (The arrangement can also be used for fire engines.) As the shaft *c* is led through the separating wall *k*, but on account of its lateral deviations cannot be made tight in it by a stuffing box, the centrifugal wheel is so constructed as to ensure the separation of the steam and water spaces and to prevent the passage of the press water into the steam space. For this purpose the wheel *l* is provided at its upper side with radial vanes *n*. The pressure produced by these is higher than that produced by the oblique vanes, and the passage of the water from the pressure space *s* into the steam space *r* is thereby prevented. This stuffing arrangement—the centrifugal wheel—might also be so constructed that vanes set either radially or obliquely were attached to its upper and lower sides. In every case, however, the upper vanes must develop a greater degree of centrifugal force than those attached to the lower side of the wheel.

Rateau,[1] whose contributions are carried out by the Société Sautter, Harlé & Cie. of Paris, has devoted his special attention to the direct driving of blowers and pumps by means of steam turbines, and has obtained apparently good results. In this connection mention may be made of the experiments with a *turbine blower*, in which the turbine made in the form of a Pelton wheel was fixed on the same shaft, with a steel centrifugal wheel. The turbine wheel was $11\frac{13}{16}$ ins. and the blower wheel 10 ins. in diameter. The latter ran in a cast-iron casing, into which the air was drawn axially at both ends. The machine made from 8000 to 20,200 revolutions per minute, so that the blower wheel attained a circumferential velocity of nearly 870 ft. per second, with a pressure of more than 7 lbs.

The relation of the turbine work to the power developed by the blower is here regulated by means of a governor inserted between the blast pipe of the blower and the steam valve of the turbine. The governor consists of a piston moving in a cylinder, its rod working a double lever. The latter is loaded at the arm of the piston rod by means of a spring, and acts upon the steam valve with its other arm. The space above the rod is connected with the blast pipe of the blower by means of a tube. The tube leading from the lower cylinder space, projects with its curved part into the blast pipe in such a manner that the compressed air sweeps along the mouth of the tube. As the pressure below the governing piston varies with the velocity of the compressed air, this organ exercises influence upon the steam valve in proportion to the quantity of the latter. If the regulation is to take place in degree corresponding with the variations of pressure in the blower, *i.e.* if it be directed to the attainment of a constant pressure rather than of a constant volume, the upper part of the cylinder must be connected with the atmosphere, and the lower part with the blast pipe. The governing piston may also be replaced by a spring plate or similar appliance. The arrangement for the

[1] Bulletin de la Soc. de l'industrie minérale, 1892; Bulletin de la Soc. d'Encouragement, 1901.

lubrication of the bearings is likewise interesting. For the reception of the lubricating material a high-placed reservoir is used, from which the former runs into the bearings. The dropping oil accumulates below, and is again carried upwards by means of an injector worked by compressed air.

Rateau calculates that, by the use of a single centrifugal wheel, the pressure can be increased up to 22 lbs. With two wheels working in series an increase up to 33 lbs. can be effected, while with three wheels an increase of 50 lbs., and with the addition of a fourth one an air-pressure of $73\frac{1}{2}$ lbs. per square inch can be produced. In the case of a turbine blower for a furnace dealing with 157 tons of cast-iron per day and requiring 320 cub. ft. of air of atmospheric pressure per second, the blower wheel has a diameter of 2 ft. $7\frac{1}{2}$ ins., and, if the air is to be compressed to 7 lbs. per square inch, will make 6000 revolutions per minute.

As regards *turbine pumps*, the experimental engine made use of by Rateau consisted of a two-stage turbine (with Pelton wheels $11\frac{13}{16}$ ins. in diameter), which, with the centrifugal wheel ($3\frac{1}{8}$ ins. in diameter) of the pump, were mounted on the same shaft. The water was supplied in an axial direction and discharged in the tangential one.

A regulating device (Fig. 624) influences both the pump outlet and the steam valve. A lever A, B, C with two arms works on a pin B, which is connected with the water piston F, one of its arms C being connected with the air-piston J. The water piston F moves in the cylinder M, the lower part of which is connected with the suction pipe of the pump by means of a pipe H, and the upper part by means of a bend G leading into the interior in a direction opposite to that of the flow of the water. When the water enters the pump, a one-sided excess pressure is exerted on the piston F proportional to the square of the velocity, and thus also to the quantity of the water, and displaces the piston F in opposition to the regulating action of the spring R. By this means a regulation of the inlet of the steam is effected, so that a uniform quantity of water is maintained. The lower part of the air-cylinder N receives compressed air by means of a tube K from a ventilator mounted at the end of the engine shaft. The compressed air has the tendency to lift the air-piston J, in opposition to its own weight and to the strain of the spring r engaging at D near the lever, and thus to move the double lever, thereby adjusting the steam valve, and keeping the velocity of the turbine constant and preventing an excess velocity when the pump is freed of its load.

Rateau in his experiments worked with revolutions of from 9000 to 18,000 per minute. He was able to raise limited quantities of water up to 997 ft., and the normal quantity of 2·64 gallons per second, suited to the pump, to about 863 ft. per second with 18,000 revolutions per minute, which was equal to an exertion of 42 H.P. with a total efficiency of 31·5 per cent. According to the results obtained, a pump for 500 H.P. (at the pump), coupled to a multi-stage turbine of new design, is capable of raising 97 gallons to a height of 279 ft. in a second while running at 1800 revolutions per minute. With its 500 H.P., then, it has an efficiency of 70 per cent. In this case the consumption of steam would be 19 lbs. per E.H.P.

per hour (in raised water). The plant covers an area of about 13 ft. 11 ins. by 4 ft. 1 in. only.

The experiments of Rateau have sufficiently shown the utility of turbine pumps —for instance, for the water supply of towns.

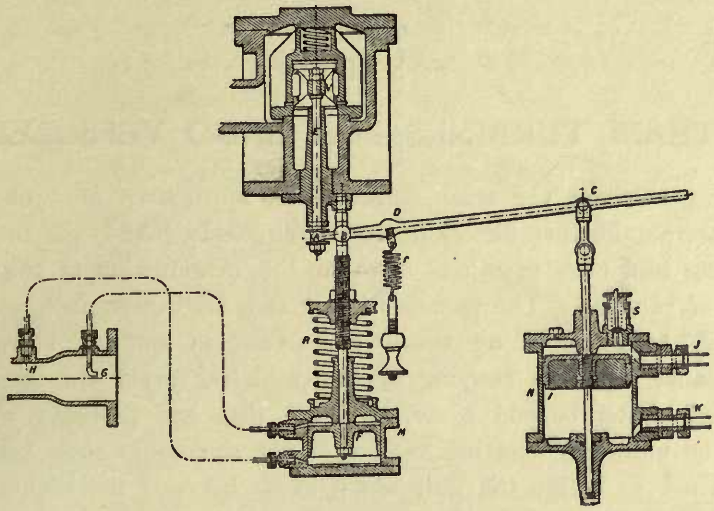


Fig. 624.

An older American arrangement (*Babbitt's*[1]) is characterized by the circumstance that the centrifugal pumps are mounted flying on the shaft at each side of the power engine. In this case they work in parallel; they draw from a single pipe, and also have a delivery in common.

This flying arrangement of the pumps is met with again in the blowing engines, amongst others, of Riedler-Stumpf. Here, however, the two pumps work in parallel so long as moderate air-pressures are in question. For the attainment of high degrees of air-pressure they are arranged in series. A turbine blower of this kind of 1500 H.P. is to be tried by the Berliner Elektrizitätswerk (Moabit) in conjunction with the 2000 H.P. Riedler-Stumpf turbine.

[1] A. P. 220,107.

XXIII

STEAM TURBINES FOR LAND VEHICLES

For travelling on land the steam turbine has apparently not yet been used, although attempts in this direction have been made. And yet the conditions now prevailing lead ever more and more to the introduction of this engine for the driving of vehicles. The peculiarities of the driving engines at present in use, in which the reciprocating masses and forces, in spite of clever balancing expedients, cause injurious swaying of the vehicles, forbid the increase of the speed of these latter beyond a certain point that has probably already been reached. The uniformly rotating turbine is not subject to such variations, and it appears fitted to bridge the gulf which exists between the highest attainable speeds of steam engines and electro-motors respectively.

Meanwhile it must not be left out of account that against the above-mentioned advantages of the turbine considerable difficulties must be set which arise in the main from the varying style of working and the small number of revolutions per minute made by the axles of the vehicles. Also the fitting of condensers on these latter will, in consideration of the influence exercised by them on the economy of the turbine, probably become a matter of special interest. The spirit of invention may here find a rich and promising field for its activity.

For the driving of vehicles the turbine assumes importance only when it can turn the axles without intermediate gearing, *i.e.* when it can be made to work directly upon these. Also the various power and velocity stages must be produced without such gearing, and all this must be attainable without appreciable loss in an economical point of view.

Attention may here be directed to the proposal made by *Pilbrow*,[1] to use reversible velocity turbines for the driving of locomotives.

These were to turn air-screws, which were also to be formed as working-wheel bodies.

The turbine designed by *Sayer*[2] (1901) may be adduced as an attempt to solve this combination of difficult problems, although, on account of the employment of impact wheels with repeated impingement of the motive medium, it cannot claim to work economically.

The turbine case consists of the two half-globes B, C (Fig. 625), which enclose the fixed hollow axle D and bear the hollow tyre Z. The hollow axle widens

[1] E. P. 9658 of the year 1843. [2] D. R. P. 149,193.

out into the chamber A; lateral slits in the axle, however, lead from the chamber A into the chamber E. The steam entering through the axle D, then, fills the two chambers A and E. The working-wheel wreaths *g*, having different diameters

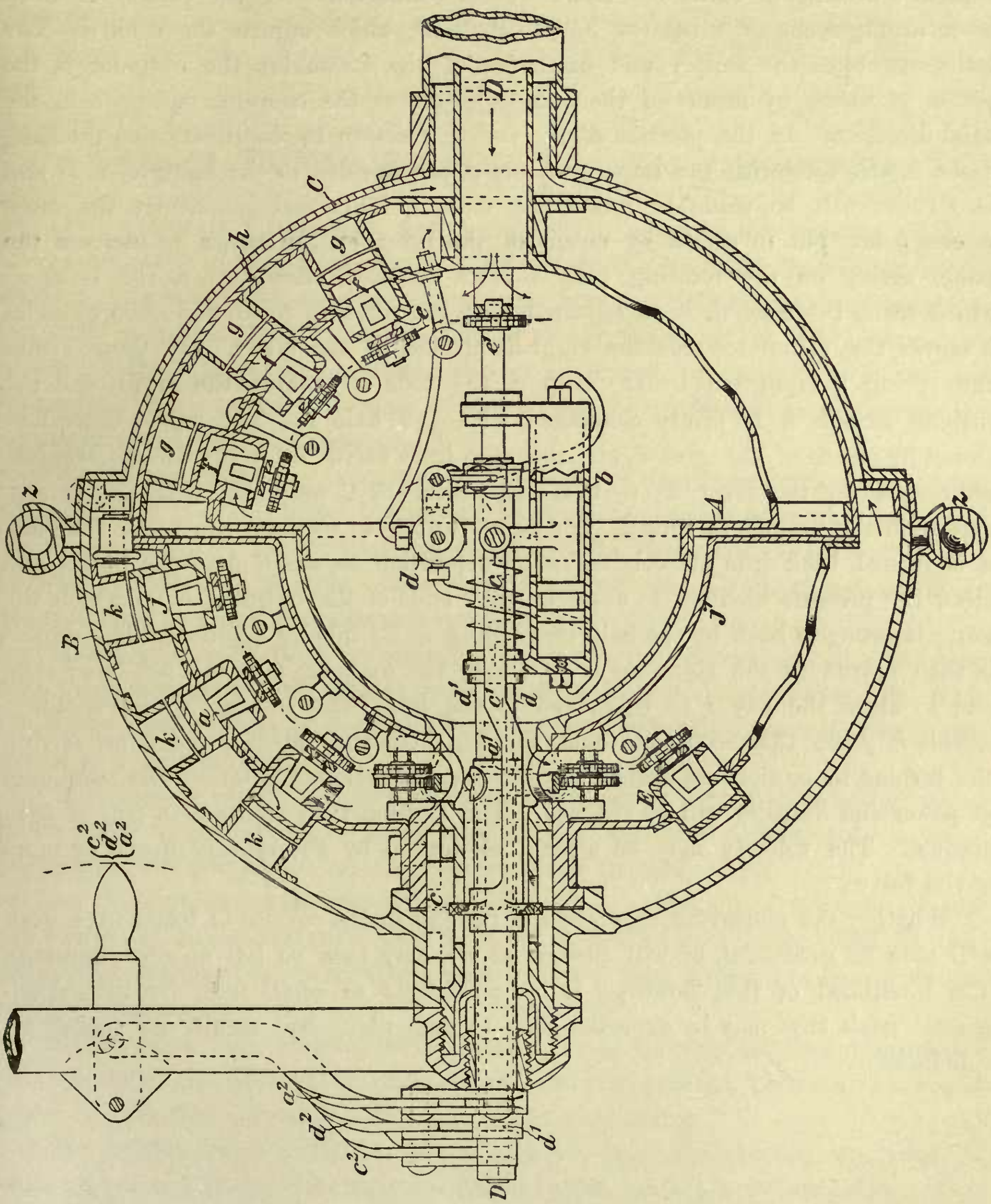


Fig. 625.

the axes of rotation of which coincide with that of the axle D, are attached to a separate half-globe *h*; they are impinged from the nozzles *f*, which are influenced by cocks *e*. These latter are opened and closed in succession by the lever arrangement c^2, which is actuated by the shaft e^1 in combination with chain wheels. The working-wheel wreaths k are arranged close to the half-globe B. The corresponding impingement nozzles j are likewise opened or closed in succession by cocks a when the lever arrangement a^2 is actuated and the hollow shaft a^1 is turned, which again turns the chain wheels connected with the cocks. A third lever arrangement d^2 turns the hollow shaft d^1, which adjusts the cock d. The latter regulates the supply and discharge of the steam for the cylinder b, the piston of which, by means of the rods c^1, displaces the coupling spring c in the axial direction. In the position of the spring c shown in the illustration the half-globe h, and therewith the impact wheels g, are coupled to the half-globes B and C. These will be made to rotate the more quickly and powerfully the more wheels g are put in action by means of the lever e^2. In order to increase the power acting on the rotating body BC, *i.e.* on the hollow tyre z, the lever a^2, which turns the cock d, is so adjusted that the pressure medium in the cylinder b moves the piston towards the right-hand side of the latter. By these means the spring c is moved by the help of the rods c^1 towards the right, and the turbine section h is firmly connected with the axle D. The valves e are now closed by means of the lever e^2, and then the inlet valves a for the motive medium are opened by the lever d^2, so that the bodies B, C only are, with the tyre z, driven in the desired direction. In order further to increase the power the cock d is turned back into its original middle position so as to close the outlet and allow the pressure medium to enter at both ends of the cylinder b, on which the key c is brought back by the help of a spring to its middle position. The cock d is then moved to the right so as to empty the cylinder b at the left-hand side, and to allow the key c to be moved to the left. The two groups of working-wheels k, g are then coupled together. If, then, the valves e and a for setting the turbine in motion are opened for the same direction of rotation, the maximum of power and velocity will be reached. The turbine then works as a pair of twin engines. The velocity may be altered as desired by the closing of one or more of the valves.

Whether the endeavour to place the turbine in the service of locomotive work will soon be successful, or will succeed at all, may here be left an open question. The likelihood of this, however, is by no means so small that, for the experimental trials that may be expected soon to take place, bad results are a foregone conclusion.

XXIV

STEAM TURBINES FOR USE ON SHIPBOARD

In 1903 a special commission was appointed to consider the question whether it was desirable to introduce steam turbines into the Navy of the United States of North America. The conclusions drawn by Admiral Melville from his investigations and submitted by him in a report to the Secretary of the Navy are worthy of note.[1] He says, "With a turbine little or no care or precaution is required in starting or reversing, and these operations can be performed as rapidly as the necessary valves can be manipulated.

"As the propellers used with turbines are smaller than those with reciprocating engines, the tips of the upper blades will be more deeply immersed and less likely to be uncovered by the pitching of the vessel, so there will be less possibility of racing. But even if racing does occur, there will be no risk of serious injury or breakdown as with a reciprocating engine.

"There will be little or no vibration caused by the turbine, and the vibration from the propellers will be greatly decreased by reason of their smaller size. This absence of vibration and perfect balance of the parts will allow much lighter engine platforms to be used.

"The absence of all interior rubbing parts will allow highly superheated steam to be used, which cannot be done satisfactorily with the reciprocating engine; this in turn will add greatly to the economy of the turbine.

"As the turbine would have but two bearings, any working of the vessel would not disturb it, and could only tend to throw it out of adjustment with the line shaft; this tendency, however, would be no greater than is now the case with the reciprocating engine.

"The absence of all working parts except a few of the very simplest description, and especially the absence of linkages and interior rubbing parts in the steam spaces, reduces the cost and labour of upkeep to a minimum.

"The lubrication of a turbine is almost ideal, since the absence of adjustable bearings permits forced lubrication without appreciable loss of oil or any of it being carried into the condenser by the exhaust. The same oil can be pumped through the bearings over and over again, being cooled in passage by a water coil."

According to the experience obtained with Parsons machines, the circumstances

[1] *The Mech. World*, 1903, p. 173.

are such that, in comparison with the ordinary reciprocating engines of merchant vessels, in which, in the main, only rapid motion ahead is of importance, these turbines offer considerable economic advantages. As regards war vessels, in which provision must also be made for cheap voyages at marching speeds and for rapid motion astern, it would seem that favourable solutions for these problems for turbines are about to be achieved.

Parsons' first experimental vessel, the *Turbinia*, was built in 1894.[1] Her principal dimensions are—

Length between perpendiculars	100 ft.
Breadth	9 ft.
Depth	6 ft. 10½ ins.

A Yarrow watertube boiler of 42 sq. ft. grate surface and 1100 sq. ft. heating surface supplied steam at an excess pressure of 224 lbs. per square inch. The turbine had also to drive the furnace ventilator. The first experiment made with a single turbine which, with a speed of rotation of 2500 revolutions per minute, developed 1500 H.P., failed on account of the inefficiency of the too rapidly revolving propeller. The distribution of weights in this first arrangement will, however, be of interest. It was as follows :—

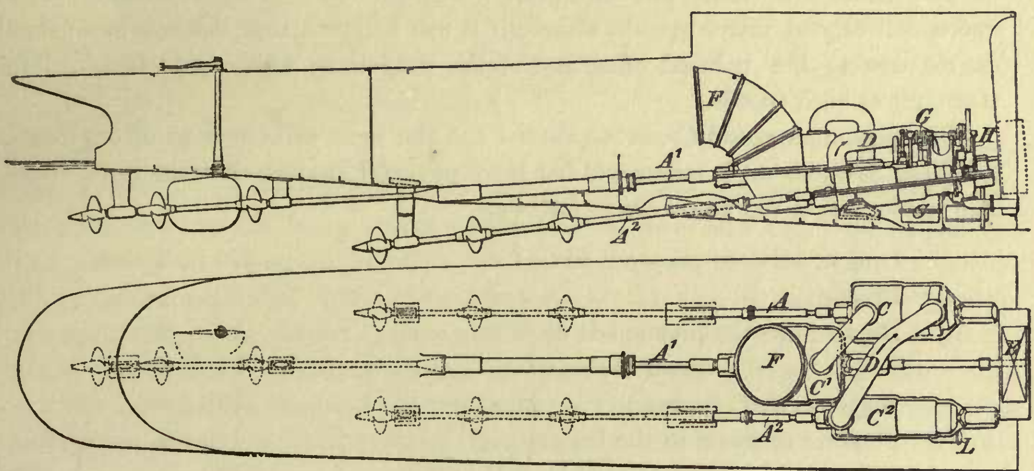
Turbine	3·65 tons.
Condenser, water, shafting, boilers, propellers, etc., . .	18·34 „
Hull of the vessel	15·00 „
Coals and spare water	7·49 „
Total . .	44·48 „

The trials showed that the high number of revolutions of the shafting entailed the use of propeller blades with a too small effective surface. The small rapidly revolving screw in the main worked in producing a vacuum behind the blades, and finally in the evaporation of the water. Parsons accordingly reduced the amount of work required of a unit of surface of the screw by the arrangement of two or more propeller shafts, each driven by a turbine.[2] The turbines are in connection with one another, and thus work together. If, for instance, three shafts A^2, A, and A^1 be provided, there are also three separate turbines. (See Figs. 626 and 627.) The high-pressure turbine C^2 takes its fresh steam from the pipe L and passes it on by the pipe D to the intermediate turbine C. Between these two lies the low-pressure turbine C^1, from which the steam passes through the pipe F to the condenser. On the shaft of this latter turbine also is placed the ventilator E for the furnace. On the other hand, the double-acting air-pump G of the condenser is driven by a small steam engine H. The distribution of the

[1] Bull. de l'Assoc., Tech. Maritime, 1898, p. 31.
[2] D. R. P. 99,108. E.P. 11,086 of the year 1896.

work to be performed over several turbines produces a more rigid construction than can be attained with a single large turbine, and admits of a smaller angular velocity without the disadvantage of uneconomical use of the steam.

In this manner Parsons divided the single turbine into three such of equal power in the *Turbinia*. He thus in the aggregate attained a 150-fold expansion of the steam. In this new arrangement the high-pressure turbine lies in the longitudinal axis of the vessel; somewhat farther aft on the starboard side is the intermediate, and on the port side the low-pressure turbine, the exhaust steam from which flows to the condenser. The latter, which has a cooling surface of 4200 sq. ft., is in two parts, and lies somewhat farther aft. The propeller shafts are inclined at a slight angle towards the stern; their bearings are lined with lignum vitæ. Since the respective pressures of the steam and of the propellers balance each other, the turbines are placed directly upon the keelsons. For their lubrication oil under a



Figs. 626, 627.

pressure of 15½ lbs. is kept in circulation by a pump which is connected with the air-pump. The oil is kept under pressure by means of a hand-pump only, so long as the turbine is not at work. In regard to the turbines themselves, the only point which required careful attention during a voyage was the keeping of the stuffing boxes steam-tight. With the shafts making 2500 revolutions per minute a speed of 34½ knots was attained. An indicated power of the turbines equal to 1576 was estimated for a speed of 31 knots on the assumption of a coefficient of performance of 60 per cent. It may be mentioned that for speeds of less than 15 knots the efficiency of the turbines became less. Moreover, a go-astern turbine is arranged on the middle shaft; this gives the vessel a speed of 6½ knots, and is capable of stopping her within 36 seconds, *i.e.* within about 600 yards, when she is moving at a speed of 30 knots. To bring the vessel from a state of rest to a speed of 30 knots the forwards turbines require 40 seconds.

The two lost torpedo-boat destroyers *Viper* and *Cobra*, which had displacements of 368·3 and 394 tons respectively, had each two separate sets of machinery, viz. a high-pressure turbine combined with a low-pressure one on each side of the vessel, and four shafts in all. The two low-pressure turbines, to the shafts belonging to which the backwards turbines were also fitted, lay next the middle-line plane of the vessel, and the two high-pressure turbines with the condensers at the sides. At a speed of 36 knots the engines of the *Viper* developed 12,300 I.H.P., with an average number of revolutions of 1180. In steaming astern a speed of 15 knots was attained. The weight of the engines and steam-generating plant was

$$= 30 \text{ lbs. per I.H.P.}$$

The weight of the engines, including auxiliary machinery and propellers, came to

$$= 11{\cdot}7 \text{ lbs. per I.H.P.}$$

The trials with these torpedo-catchers also showed that the turbines worked uneconomically at march speeds, although it was believed that this was in no small degree due to the reduced efficiency of the propellers, which were designed for steaming at high speeds.

The passenger vessel *Queen Alexandra* has the same arrangement of engines as the *King Edward*, but is larger and has more powerful engines than the latter vessel. The particulars of the vessel are as follows:—

Length between perpendiculars	270 ft. 2½ ins.
Breadth	30 „ ¼ „
Depth from the promenade deck	18 „ 9 „
„ „ main deck	11 „ 6 „
Displacement (approx.)	738 tons
Pressure of steam in the boilers	150 lbs. per sq. in.
„ „ delivered to H.P. turbine . . .	135 „ „ „
„ „ delivered to L.P. turbine . . .	82 „ „ „
Power developed (approx.)	4500 I.H.P.
Speed attained	21·68 knots.

In ordinary steaming the high-pressure turbines made 700 revolutions, the low-pressure ones 1000 revolutions per minute. It must here be observed that the propellers must not have a greater velocity at the tips of the blades than from 165 to 200 feet per second.

It may in general be said that the Parsons turbines with their appurtenances take up approximately the same ground area as the modern reciprocating engines, but are considerably less than the latter in height. In comparison with the quickly-running reciprocating engines, a saving of weight (for the turbines alone) of from 20 to 22 per cent. is attained. For a passenger vessel of 18,000 tons to steam at 26 knots Parsons estimates the engines required at 38,000 H.P., and their weight at 134 lbs. per I.H.P. It must here, however, be pointed out that for war vessels, which require powerful go-astern turbines, the saving in weight over the reciprocating engines is not very great. The saving will here in the main be due to the lighter seatings and to the abolition of the armoured domes over the engine-rooms.

In contrast with the state of affairs in the reversing of the reciprocating engine, it is here possible, when changing from motion ahead to motion astern, to lead the steam into the go-astern turbines immediately after the delivery to the go-ahead turbines has been shut off. The former will, like elastic brakes, first overcome the kinetic energy of the rotating parts. The more powerfully the brakes act, the more rapid will clearly be the reversal of the direction of motion. The power of the go-astern turbine must then be made the more nearly to approach that of the go-ahead one in power the more quickly the reversal of direction is to be effected.

The *King Edward* was built in 1901 as a sister ship to the *Duchess of Hamilton* on the occasion of the Glasgow Exhibition. The particulars of the vessel are:—

Length	250 ft.
Breadth	32 „
Depth from the main deck	10 „ 6 ins.
„ „ promenade deck	17 „ $8\frac{1}{2}$ „
Displacement	640 tons.

The vessel can accommodate 2000 passengers. The machinery consists of a high-pressure turbine lying at the centre line of the vessel for 5-fold expansion, and of the two low-pressure turbines for 25-fold expansion at her sides, so that in all a 125-fold expansion of the steam takes place. In each of the cases of the low-pressure turbines a go-astern turbine also is placed. In ordinary work the three go-ahead turbines are worked in series; the high-pressure turbines, then, run at 700 and the low-pressure ones at 1100 revolutions per minute. By a simple adjustment of the valve the steam may, however, be directed to the low-pressure turbines alone. The main air-pumps are driven from the shafts of the low-pressure turbines by means of the toothed-wheel gearing. The set of turbines will in ordinary work indicate 3000 H.P., and it weighs 66 tons, which gives about $48\frac{1}{2}$ lbs. per I.H.P. On the other hand, the modern machinery of the *Duchess of Hamilton* is about twice as heavy. The latter vessel, on a voyage of 12,106 nautical miles, which was accomplished at a mean speed of $18\frac{1}{2}$ knots, used about 1967 tons of coal, while the *King Edward*, at the same speed, only used 1406 tons on a voyage of 12,116 nautical miles, thus showing a saving of 562 tons. If reciprocating engines of the same power were fitted, their greater weight would entail greater dimensions of the vessel, so that only about 19·7 knots, instead of 20·48 knots, would be attained. That is to say, an advantage of about 0·8 of a knot can be calculated in favour of the turbines, about 0·2 of a knot being due to the smaller displacement, and 0·6 of a knot to the greater efficiency of the turbines. Under these circumstances, then, there would be a gain in power of about 20 per cent. in favour of the turbines.

In the *Velox*, a new torpedo-catcher for the British Admiralty, Parsons has, in order to reduce the coal consumption for long voyages at slow speed, combined the turbines with a reciprocating engine. The shafts of the low-pressure turbines are prolonged in the forward direction and coupled with independent reciprocating engines, which work on voyages made at slow speed (15 knots). It may be observed that similar combinations were also designed by Parsons for the German war vessels under construction, but were not accepted. The German war vessels are probably the first in which economy of working of Parsons turbines fitted specially for voyages at slow speed will undergo a trial.

Soliani[1] also (Gio. Arnoldo & Company in Genoa) combines with the turbines a reciprocating engine which drives a separate propeller shaft. The latter then acts as the main engine on slow-speed voyages and when going astern, and for forced speeds the turbines are coupled on. When three shafts are fitted, the reciprocating engine drives the middle one, and the high and low-pressure turbines are distributed over the other two. For high speed, fresh steam is supplied to the reciprocating engine as well as to the turbines. On slow-speed voyages only the first-named receives fresh steam. In this case, then, the side propellers are carried along inactive. In order, then, to overcome the resistance thus arising, steam is led from the low-pressure receiver into the high-pressure one in such quantity that these are turned round at empty load or do a small amount of propelling work. A valve inserted in the circulating pipe is so connected with the reversing gear of the reciprocating engine that it is closed when the engine is reversed for going astern. Another valve, which is to be adjusted by hand, serves to regulate the passage of the steam from the receiver according to the speed of the vessel, and to shut it off altogether when the turbines are to be supplied with fresh steam.

The requirements of the war vessels in particular which require a low coal consumption on long voyages as well as on short-distance spirts have stimulated the search for means by which the turbine can be turned into an economical engine for low speeds as well as for high ones. *Parsons*,[2] for instance, proposes a development of the marine steam-engine, the object of which is that the turbines shall work as economically as possible under all conditions of power and speed. The consumption of coal per H.P. is thereby to be made as small as possible at low powers as well as at high ones. If four shafts be arranged for, each of which is driven by a special compound steam turbine, these latter are inter-connected by valves and pipes in such a manner that at a small exertion of power the steam streams through the four compound engines one after the other and passes from the last of these into the condenser. For this purpose the four turbines are so arranged that the quantity of steam they can take, or their capacity, gradually increases. The steam then expands continuously from its entrance into the first turbine case till it finally passes from the fourth low-pressure one into the condenser. The variation in the capacity for the reception of steam may be attained either by an increase of the dimensions in the proportion of one to four, or by the adoption of greater velocity of rotation, or by both of these means; also it is possible to increase the capacity for steam of each separate turbine so that it is smaller at the inlet than at the discharge end.

[1] E. P. 17,941 of the year 1901.
[2] E. P. 11,223 of the year 1897. D. R. P. 103,559.

For more considerable powers the steam-pipes are so arranged that the high-pressure steam streams from the boiler to the inlets of the first and second turbines, and from these latter to the inlets of the third and fourth; it then passes from the discharge openings of the last turbines into the condenser. In this case, then, the turbines would be arranged in compound-parallel, and the combination would for most purposes suffice. It is possible, however, in exceptional cases so to arrange the valves that high-pressure steam is led to all four turbines at once, from which it then passes directly into the condenser; this is called the simple-parallel arrangement. In the case of large powers it is advisable that the turbines be placed in such a manner that the first-named style of parallel arrangement is sufficient. It is also possible, when it is desired to reduce the boiler pressure, to arrange the first and second turbines in parallel and the third and fourth ones in series. In this case the two last-named receive their steam from the first and second ones together. This arrangement may also be employed when the pressure in the boiler is to be reduced to that of the atmosphere. In order that the expansion of the steam may be evenly distributed in all the turbines, whether in the compound-parallel or in the series arrangement, it is necessary either to make the capacity for steam constant from the first to the last of them or to let it increase by stages. When turbines on the compound-parallel system are working at full power, the pressure exercised on the various propeller shafts is an irregular one. This drawback is met by the expedient of making either the screws on the shafts or the cubic contents of the turbines somewhat different from those most suitable for the series arrangement. This latter method has no very injurious effect in any of the styles of arrangement discussed. The turbines which are not at work, but are allowed to run free, are connected with the condenser, so that they offer as little resistance as possible.

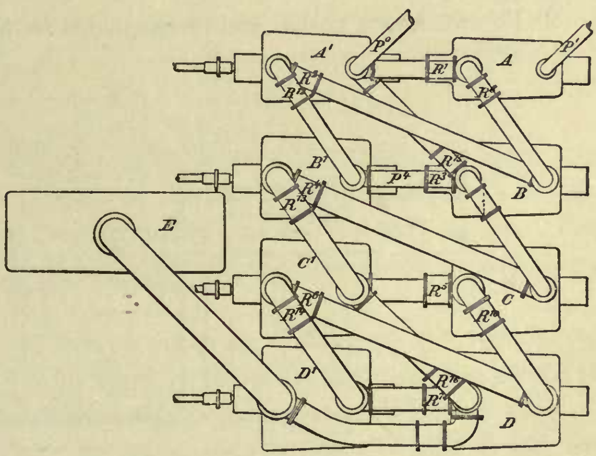


Fig. 628.

In Fig. 628 four shafts are arranged, on each of which two turbines are placed. The two compound sets ABCD and A'B'C'D' can be arranged in parallel or in series. Each turbine of the latter set is larger than the corresponding one of the first-named set on the same shaft. P^1 and P^0 are steam-inlets, and E is the condenser. The following methods can be adopted:—

1. All the turbines are arranged in series, and the pipe P^1 is used for the feed alone.

(*a*) If the valves $R^1, R^2, R^3, R^4, R^5, R^6, R^7$ be open, the steam streams successively through

$$A \to A' \to B \to B' \to C \to C' \to D \to D' \to E.$$

(*b*) If only the valves R^8, R^{15}, R^{12}, R^4, R^{10}, R^{16}, R^{14} be open, the path of the steam is

$$A \to B \to A' \to B' \to C \to D \to C' \to D' \to E.$$

2. In the case of the compound-parallel arrangement, the steam streams in through the pipes P^1, P^0. Its paths are: ABCDE and A'B'C'D'E'. In this case it is possible to make use of only one set when small powers are required.

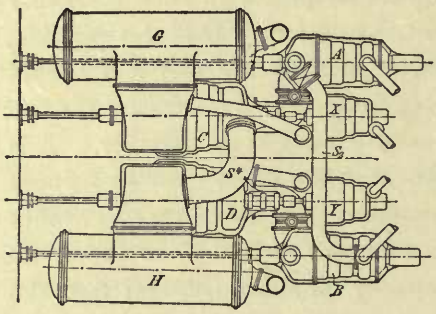


Fig. 629.

In Fig. 629[1] an arrangement is illustrated in which one each of two high- and two low-pressure turbines AB or CD turns a shaft. The turbines are connected by pipes and valves in such a manner that they can be arranged either in parallel or in series. The condensers are denoted by GH, the go-astern turbines by XY.

The following groupings may here be made :—

1. For full power the two compound turbines AC and BD are arranged in parallel.

2. For moderate speed and consequent reduced work the steam delivery is

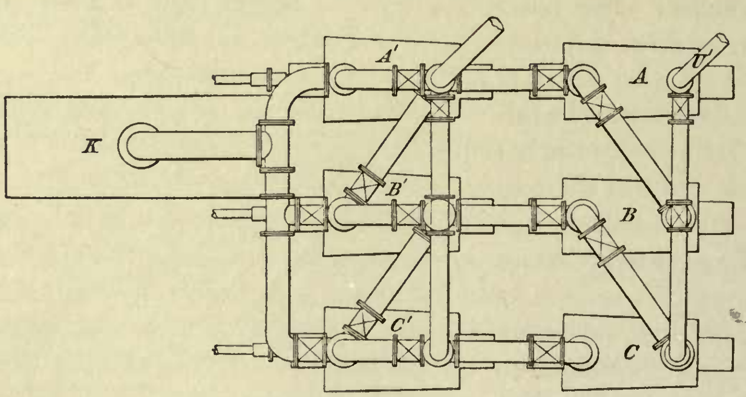


Fig. 630.

opened only to A, and the circulating pipes S^2 and S^4 are made use of; on the other hand, the condenser H is shut off. The path of the steam is then

$$A \to B \to D \to C \to G.$$

If only three shafts be adopted, six turbines will be employed, two of which,

[1] Compare also E. P. 26,553 of the year 1897.

AA′, BB′, CC′, act on each shaft, and are connected in parallel and in series respectively by means of pipes and valves (see Fig. 630).

The following paths of the steam are then obtained:—

1. When all the turbines are arranged in series, in which case the steam is supplied by the pipe U^1 only,

$$A \rightarrow B \rightarrow C \rightarrow C^1 \rightarrow B^1 \rightarrow A^1 \rightarrow K \text{ (condenser).}$$

2. In case of compound parallel arrangement: $A \rightarrow A^1 \rightarrow K$
$B \rightarrow B^1 \rightarrow K$
$C \rightarrow C^1 \rightarrow K$

3. In case of simple parallel arrangement all the turbines receive fresh steam, which they discharge into the condenser.

According to *Parsons*,[1] the four-shaft triple-expansion turbine arrangement given in Fig. 631 is compact and economical in steam, and manœuvres very well. The high-pressure turbine A gives over its steam to the intermediate turbine B, and

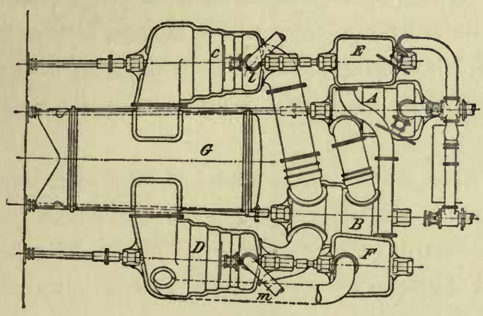


Fig. 631.

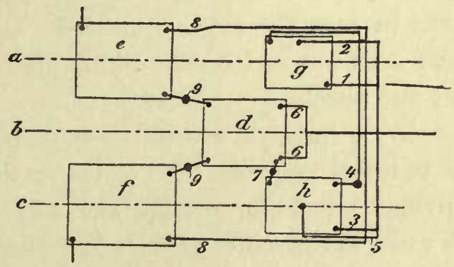


Fig. 632.

the latter passes it on to the two low-pressure cylinders C, D, which discharge into the condenser. By means of the pipes *l, m,* the low-pressure turbines C, D can, however, be fed directly, so that the two outer shafts can be separately driven. The go-astern turbines E, F, on the outer shafts, work together, the steam from E being passed on to F.

Finally, *Parsons*[2] proposes to ensure economy in working at the reduced or marching speeds, as well as at the high ones, by means of the grouping shown in Fig. 632. If three screw-shafts *a, b, c* be adopted, the arrangement is the following:—The middle shaft *b* is connected with the main high-pressure turbine *d*, and the corresponding low-pressure turbines *e, f* turn the shafts *a, c*. Here, however, there are two additional turbines *g, h,* of which the first-mentioned is the smaller, the second one being smaller than turbine *d*, and both being of such small capacity that they can only take sufficient steam for cruising. At the slowest speed the turbine *g* receives steam from the pipe 1. The steam then streams through the turbines

$$g, h, d\ (e, f)$$

[1] E. P. 26,553 of the year 1897.

[2] E. P. 17,391 of the year 1901.

in succession and escapes to the condensers. For a small increase of the speed additional steam is supplied to the turbine *g* through the pipe 2, and for a further increase a supply is delivered through the pipe 3 to the turbine *h*, any back pressure that is set up towards the turbine *g* being taken up by the non-return valve 4. The turbine *h* also can, in like manner, receive additional fresh steam through the pipe 5.

To produce full speed fresh steam is led through the pipes 6 into the high-pressure turbine *d*, the non-return valve 7 affording security against back pressure towards the turbine *h*. Finally, fresh steam may be allowed to stream through the low-pressure turbines *e*, *f*, in which case the non-return valves 9 come into action. When the turbine *d* is directly impinged, the turbines *g* and *h* can be put out of gear.

The general idea, then, is that to a set of turbines intended for the development of full power a further turbine is added, which comes into use when full speed gives place to that of marching. The coupling on of the latter has the effect of an increase in the number of the stages, and thus of a reduction in the revolutions per minute made by the shafts. This increase of the number of the stages is, it is true, attained only in case the expanding steam is, in the last stage, still able to perform work, and does not arrive there in the form of a lifeless mass that is only carried round by the wheel.

If, for instance, a vessel is to be propelled at a maximum speed of 35 knots, but is to make economical use of the steam at other speeds as well, the engine may be divided into high, intermediate, and low-pressure turbines (denoted by H, M, and N), and to the shafts of the two latter small additional turbines (A and B) may be added. Then

for 30 to 35	knots	the turbines	H, M, N	receive	full	steam.
„ 25 „ 30	„	„	H, M, N	„	normal	„
„ 20 „ 25	„	„	A, H, M, N	„	full	„
„ 15 „ 20	„	„	A, H, M, N	„	normal	„
„ 10 „ 15	„	„	B, A, H, M	„	full	„
„ 5 „ 10	„	„	B, A, H, M	„	normal	„

Plates XVI., XVII., XVIII., and XIX., which are intended to show the arrangement of the Parsons turbines on shipboard, have been supplied by the Aktiengesellschaft Brown, Boveri & Cie., of Baden, Switzerland. Plate XVI. shows the arrangement of the turbines on a passenger steamer, Plate XVII. that on a torpedo-boat, and Plate XVIII. that on a cruiser. The manner of working of the sets of machinery themselves will, after the foregoing explanations, at once be clear from the drawings. The accompanying table, in which the vessels are given, that up to the completion of this work have been fitted for propulsion by Parsons turbines, and are now either at work or under construction, shows to what importance the Parsons turbines have attained for shipbuilding.

PLATE XV.

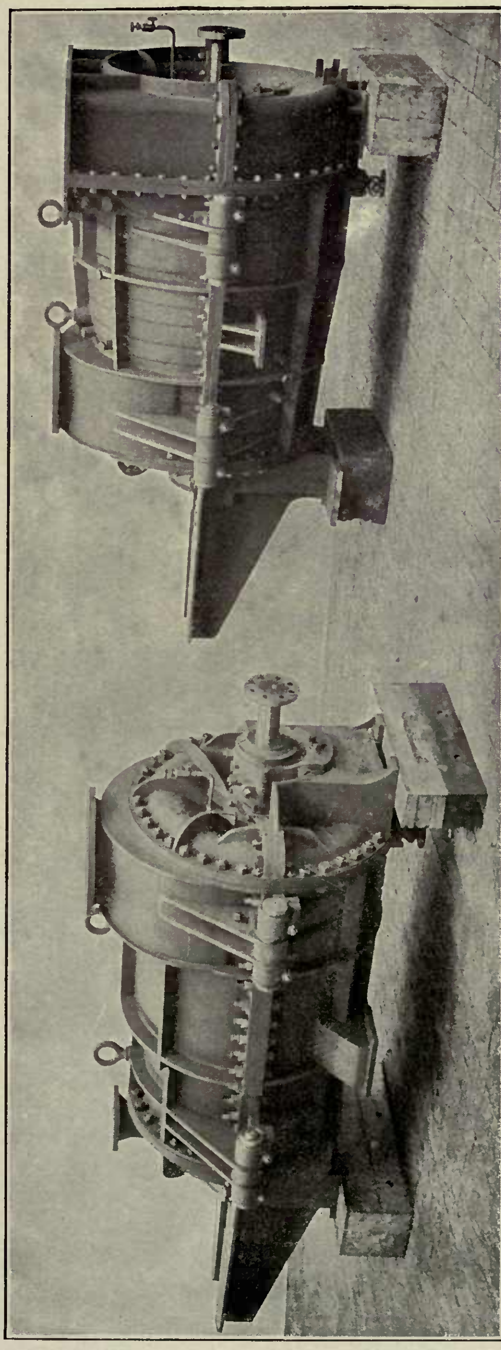


Fig. 633.
The Rateau Turbine as Marine Engine. H.P. and L.P. Turbine of 850 H.P. each.

List of Vessels under Construction or Completed, the Engines of which are Parsons Turbines.

Name.	Type.	Year.	Length. ft.	Length. ins.	Breadth. ft.	Breadth. ins.	Displacement. tons.	I.H.P.	Number. Shafts.	Number. Screws.	Speed attained. Knots.	
Turbinia	Experimental vessel	1900	100	1	9	0	43	2300	3	9	$34\frac{1}{2}$	3 shafts, 3 propellers.
Viper	Torpedo-catcher	1900	210	0	21	0	366	12300	4	8	37	Lost.
Cobra	,, ,,	1900	223	1	20	5	394	12000	4	8	35	,,
King Edward	Passenger vessel	1901	249	5	30	0	640	3500	3	5	$20\frac{1}{2}$	Two seasons on service.
Queen Alexandra	,, ,,	1902	269	0	32	0	738	4200	3	5	$21\frac{1}{2}$	,, ,, ,,
Velox	Torpedo-catcher	1902	210	0	21	0	364	9000	4	8	33	Has recipr. marching engine.
Tarantula	Yacht	1902	152	7	15	$3\frac{1}{2}$	138	2200	3	6	$25\frac{1}{2}$	Has marching turbines.
Loreno	,,	1903	252	7	32	11	1370	3500	3	5	16	
Emerald	,,	1903	198	6	28	6	787	1500	3	5	16	
Newhaven-Dieppe (Brighton)	Channel steamer	1903	275	7	—		—	6300	3	5	21	First vessels for Channel service.
Calais-Dover (Queen)	,, ,,	1903	310	0	—		—	7300	3	5	21	
H.M.S. Eden	Torpedo-catcher	1903	—		—		—	7000	3	6	$25\frac{1}{2}$	New British type.
H.M.S. Amethyst	3rd class cruiser	1903	—		—		2950	9800	3	5	$21\frac{3}{4}$	Launched. First turbine cruiser.
French T.B. 293	1st class torp.-boat	1904	129	7	13	11	93	1800	3	6	24	Launched on March 16.
German T.B. S.125	Torpedo-boat	1904	—		—		367	6300	3	6	27	Under construction by Schichau.[1]
S.M.S. Lübeck	Small cruiser	1904	—		—		3150	10000	4	8	22	First vessel in which all shafts reversible: under construction by the Vulcan, Stettin.[1]
2 vessels of the "Cunard Line"	—	—	—		—		Each 28000	Each 65000 to 70000	—	—	25	Under construction.
2 Transatlantic vessels of the "Allan Line" for service between Great Britain and Canada	—	—	—		—		—	—	—	—	—	,, ,,

[1] The steam turbines for these vessels are being built in the works of Messrs. Brown, Boveri & Cie., in Baden.

The Rateau turbine, the style of construction of which has already been discussed, has also been arranged by the Maschinenfabrik *Oerlikon* for the propulsion of vessels. In an older arrangement a marine turbine of 1200 H.P. was placed in a case to act directly on a single propeller shaft. A new style of construction shows the distribution of the power over two shafts (see Fig. 633, Plate XV.). A high-pressure and a low-pressure turbine are fitted, each of which is to develop 850 H.P.

So far as at present known, besides the installation on the French torpedo-boat No. 243, which has been undergoing experimental trials since 1902, the Rateau turbine has also been fitted on the small vessels *Vedette* and *La Libellulle*.[1] Another torpedo-boat built in the English style by the yard of Yarrow & Co., and fitted with Rateau turbines made by the Oerlikon works, was in the autumn of 1903 and in January of 1904 submitted to searching experimental trials, the results of which are as follows:—The vessel has a length of 150 ft. 11 ins., a breadth of 15 ft. 1 in., and a displacement of 138 tons. The set of turbines consists of a high-pressure turbine on the port side, and a low-pressure one lying further aft on the starboard side. The condenser lies on the port side abaft the high-pressure turbine. Alongside of the latter amidships stands a triple expansion reciprocating engine of 250 I.H.P., which turns a separate propeller shaft, and on voyages made at slow marching speed works alone. The turbines, which together weigh 7·62 tons, develop about 2000 I.H.P.; they come into action in the case of voyages at high speed. In this case, then, the turbine is combined with the reciprocating engine. The whole set of engines takes up about 3 ft. 3 ins. more room in the longitudinal direction of the vessel than would reciprocating engines of equal power, and the mean speed attained was only 26·39 knots. In the mean time, then, compared with

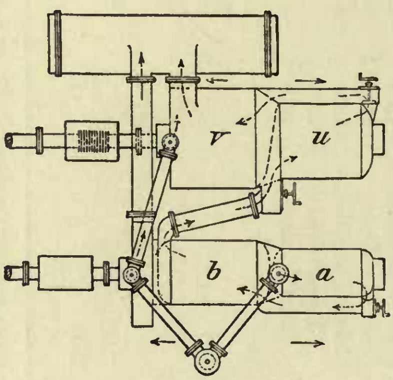


Fig. 634.

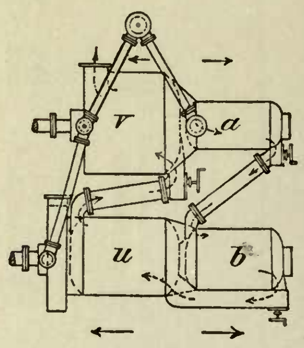


Fig. 635.

the reciprocating engine, economical advantages in favour of the Rateau turbines have, at least in the type of vessel experimented with, not been attained.

The compound arrangement of *Schulz*[2] for two (screw) shafts, in which the

[1] "Progrès de l'app. d. turbines à vapeur à la prop. d. navires," *Genie civil* of January 2, 1904; Rateau, "Prop. des navires," *Rěvue industr.* of April 23, 1904; and Rateau, "Steam Turb. prop. f. marine purp.," in *Marine Eng.* of April 1, 1904, and *Eng.* of April 8, 1904.

[2] D. R. P. 137,792. E. P. 8378 of the year 1901.

attempt is made to relieve the former from axial thrust by the combination of axial with radial turbines, may here be given (see Chap. XVII.). In the example given in Fig. 634 the arrangement is such that the smaller turbine bodies *a* and *b* are on the one shaft, and the larger ones, *u*, *v*, on the other, which is parallel to it. The two turbine bodies *a* and *u* exercise an axial pressure towards the right, and the two bodies *b* and *v* a like pressure towards the left, as shown by the arrows. In the example in Fig. 635, on the other hand, the smallest turbine body *a*, together with the largest body *v*, is arranged on the one shaft, while the second smallest

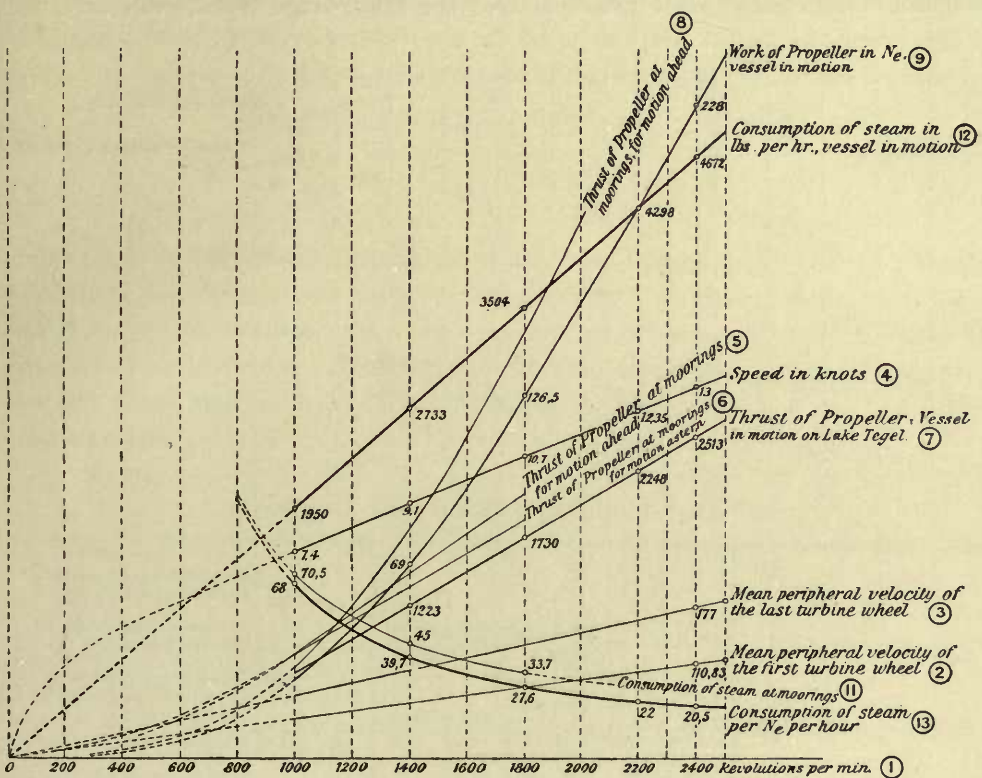


Fig. 636.

body *b* and the second largest one *u* are placed on the other shaft parallel to the first.

In general, however, the grouping of the compound turbines is the outcome of the desire to provide economical methods of working, in particular for war vessels.

Schulz, it may be said, as the result of his experiences, comes to the conclusion that a good steam turbine, at any rate in a war vessel, must be expected to work as economically as possible, not only at maximum and minimum powers, but also at all the various intermediate ones. With this view, in addition to the engine construction required for the maximum power, the Schulz turbine is provided with a considerable number of wheels that are gradually reduced in size and combined in separate groups or bodies, which have inlet and outlet check-valves in connection with the steam supply pipes or with the inlet to the next largest turbine body in each case. If the smallest turbine body be of such dimensions that it can serve for the smallest amount of work at the highest possible pressure of steam, the turbine bodies lying between it and the main turbine are used for the intermediate powers, so that here also the steam may be used at its highest possible pressure. For smaller work the turbine wheels can, it is true, not all be brought into use. Meanwhile, the largest and the last turbine wreaths that are not performing work are carried round *in vacuo*, and use up only a small amount of power in overcoming friction. Schulz goes on the principle that the multi-stage turbine can work well

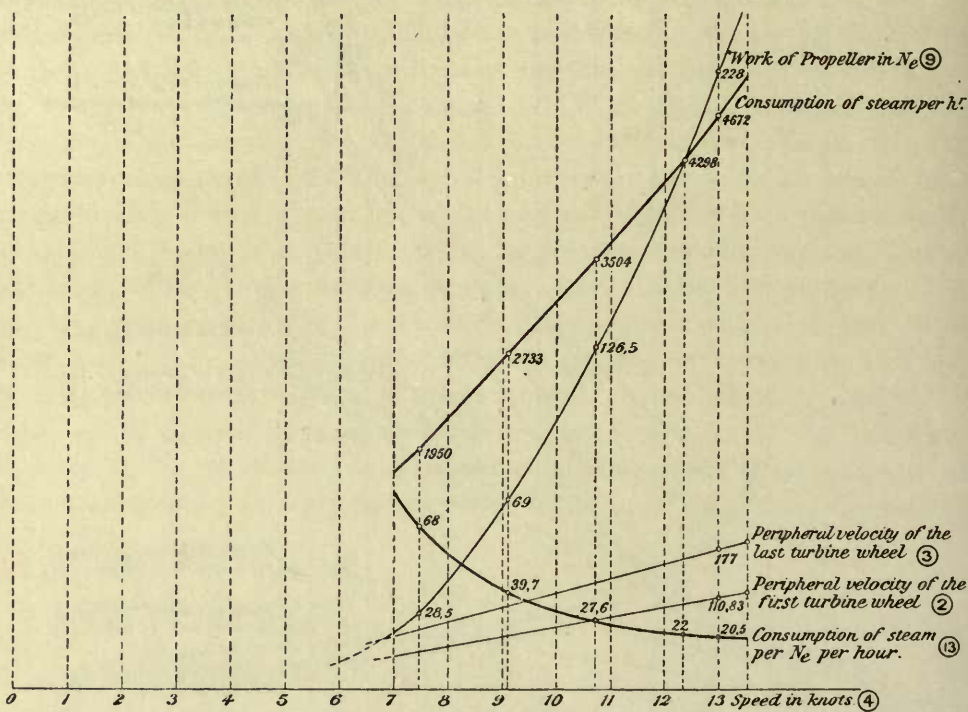


Fig. 637.

with *the smaller* peripheral velocity, the *greater the number* of the stages. As a matter of fact, in his turbines, the fewer the revolutions (corresponding with smaller power and speed of the vessel) made by the turbines and propellers, the more stages come into action.

The "Germania" works in Tegel have recently (see p. 200) conducted a number of experiments with the Schulz turbine, the results of which may well afford encouragement to experiments on a larger scale. The turbine was fitted in a small boat which had a length of 59 ft., a breadth of 9ft. $2\frac{1}{2}$ ins., a mean draught of 4ft. $8\frac{1}{2}$ ins., and a displacement of from 15 to 16 tons. From the time when it was set to work (December 3, 1901) the engine worked to perfection, with propellers of from $11\frac{3}{4}$ ins. to 1 ft. $4\frac{3}{4}$ ins. in diameter; even at a high speed of rotation (more than 2000 revolutions per minute) it produced no vibration in the hull. As final results of the trials, the relations between the revolutions of the turbine, their consumption of steam, and their powers, are shown in Figs. 636 and 637. In Fig. 636 the revolutions of the turbine per minute (between 1000 and 2500) are set off along the base line, while in Fig. 637 the speeds of the vessel (between 7 and 13 knots) form the abscissæ instead. It may be observed that the steam was superheated by from 158° to 176° Fahr., and at the highest development entered the turbine with an excess pressure of from 206 to 220 lbs. per square inch. The vacuum, it is true, amounted only to 9·7 lbs. per square inch, because the air-pump was too small. The principal results are put together in the following table. Meanwhile, braking experiments made at Tegel in June, 1901, had shown that for every 1·47 lbs. per square inch less pressure in the condenser the power increased

with 2000 revolutions per minute by 18·5 H.P.
„ 2200 „ „ 19·5 „
„ 2600 „ „ 21·5 „

From this the conclusion may be drawn that with a vacuum of 13·2 lbs. and 2200 revolutions per minute the power of the Schulz turbine increases from 195 to 240 H.P., and at the same time the consumption of steam per H.P. per hour falls from 22·05 to 17·86 lbs. Moreover, at 2200 revolutions the peripheral velocity of the first wheel amounted only to 101·7 ft. per second, and that of the last one did not reach 164 ft.

Working Results of the Schulz Turbine.

1	2	3	4	5	6	7	8	9	10	11	12	13
Revolutions per minute.	Mean peripheral velocity of the First turbine wheel.	Mean peripheral velocity of the Last turbine wheel.	Speed of vessel.	Thrust of propeller at moorings. Ahead.	Thrust of propeller at moorings. Astern.	Thrust of propeller. Vessel in motion.	Work of propeller. Vessel at moorings.	Work of propeller. Vessel in motion.	Consumption of steam per hour with vacuum = 9·7 per sq. in. Vessel at moorings.		Vessel in motion.	
	Ft. per sec.	Ft. per sec.	Knots.	lbs.	lbs.	lbs.	H.P.	H.P.	Total.	Per H.P.	Total.	Per H.P.
1000	46·2	73·8	7·4	838	838	683	33·7	28·5	2380	70·5	1951	68·3
1400	64·7	103·4	9·1	1455	915	1223	82·1	69·0	2703	44·9	2733	39·7
1800	83·1	132·9	10·7	2072	1896	1730	150·4	126·5	5070	33·7	3485	27·6
2200	101·6	162·4	12·35	—	—	2248	—	195·0	—	—	4299	22·0
2300	—	—	12·6	—	—	2380	—	211·0	—	—	4497	21·31
2400	110·9	177·2	13·0	—	—	2513	—	228·0	—	—	4674	20·5

The velocity turbines which work with expansion nozzles have not been thoroughly tried for driving the propellers of vessels. It is said that the *Bréguet* turbine, which may be looked on as a multi-stage Laval turbine, is to be fitted in a French torpedo-boat.[1] The Navy Department intends to build two vessels of 4000 tons each, one of which is to have Parsons turbines of the Westinghouse build

[1] *Génie civ.* of January 2, 1904.

and the other *Curtis* turbines,[1] the latter to be supplied by the General Electric Co. Meanwhile, the *Allis-Chalmers Company*[2] have come forward as additional competitors with an altered arrangement of the Parsons turbine. The American yacht *Revolution*[3] also is driven by Curtis turbines.

The rapidly rotating turbine wheels produce a sort of gyroscopic effect whenever the vessel alters the direction of her longitudinal axis, and at the same time the axis of rotation of the turbines. The more decidedly and quickly these directions of motion take place, the more does the gyroscopic action make itself felt. On the occasion of the losses of the torpedo-boats *Cobra* and *Viper*, which were fitted with Parsons turbines, this peculiarity was a matter of special interest, and the questions were raised as to its effect on the shaft bearings and their connections with the hull of the vessel. The steering qualities of the vessels were probably not affected by it to any considerable degree. If two turbines completely alike in form and speed, but rotating in opposite directions, be arranged one at each side of the longitudinal axis of the vessel, the effects of their gyroscopic actions on the hull itself completely balance each other, unless—on account of a break-down or similar occurrence—only one of the engines be working. In the arrangement with three turbines, in which the high-pressure one is at the middle line, the comparatively weak gyroscopic action of the latter remains unbalanced. In general, however, especially in large steamers, the changes of direction due to the winds and waves and intentional changes of course occur so slowly that the gyroscopic action of the turbines is of no great importance. Its amount is hardly as great as that which has to be reckoned with in the case of the rotating masses of the ordinary reciprocating engines.

If p denote the weight of the revolving parts in lbs.,
ω the angular velocity of the same,
n the number of revolutions per minute,
r the amount of change of direction of the rotating parts,
w the angular velocity when the axis of rotation is displaced,
g the acceleration of gravity,
the gyroscopic work amounts to—

$$Gy = \frac{p \times \omega \times n \times r \times w}{g}$$

If a pair of *counter-direction wheels* are so arranged that they are of equal power, their velocities of rotation will vary when they meet with different degrees of resistance. Amongst others, *Carter*[4] has paid attention to this circumstance in the driving of his ship propellers, which are set one behind the other, and turn in opposite directions. He acts upon the assumption that the foremost propeller has to overcome a greater resistance than the after one, to which water that has already

[1] Compare also "Curtis Turbine for Marine Propulsion" in *Engineering* of December 11, 1903.
[2] *Iron Age*, vol. 73, p. 29.
[3] *Sc. Am.*, vol. lxxxi. No. 10.
[4] E. P. 30,932 of the year 1897.

been set in motion is flowing. If, now, the after propeller and one of the turbine wheels be fixed to the solid shaft, and the foremost propeller and the other wheel be fixed to a hollow shaft that is conaxial with the solid one, the resistances springing, amongst other causes, from the friction of the stuffing boxes of the two groups may be balanced to such a degree that only the unequal resistances offered by the water to the screws remain for consideration. In working, then, the working wheel which drives the after propeller will continue to increase the relation borne by its speed to that of the one which is coupled to the foremost propeller, until the amounts of work performed respectively by the two wheels, and therewith the performances of the propellers, have become alike. (In the case of a pump which is driven by two screws, the one of these which lies the farther aft in the direction of flow of the water will have the greater velocity.)

Attention may be called to the circumstance that *Parsons* also has given attention to the question of the employment of counter-direction wheels for marine work.[1] The shafts of a pair of wheels are passed the one through the other and each of them has a separate thrust bearing with oil lubrication. The turbine is made reversible. In reference to this arrangement it must, however, be observed that the difficulties caused by the conaxial shafts in construction and in working are such as to confine the counter-direction wheels also to the domain of the smaller vessels.

[1] E. P. 6142 of the year 1902. A. P. 729,215. Compare p. 119.

LIST OF PATENT SPECIFICATIONS MADE USE OF

German Patent Specifications (D.R.P.).

Austrian Patent Specifications (Ö.P.).

Swiss Patent Specifications (S.P.)

French Patent Specifications (F.P.)

English Patent Specifications (E.P.)

Patent Specifications of the United States of America.

INDEX

Abbreviations

H.P.	= Horse-power.
I.H.P.	= Indicated horse-power.
A.P.	= American patent specification.
D.R.P.	= German " "
E.P.	= English " "
F.P.	= French " "
Ö.P.	= Austrian " "
S.P.	= Swiss " "

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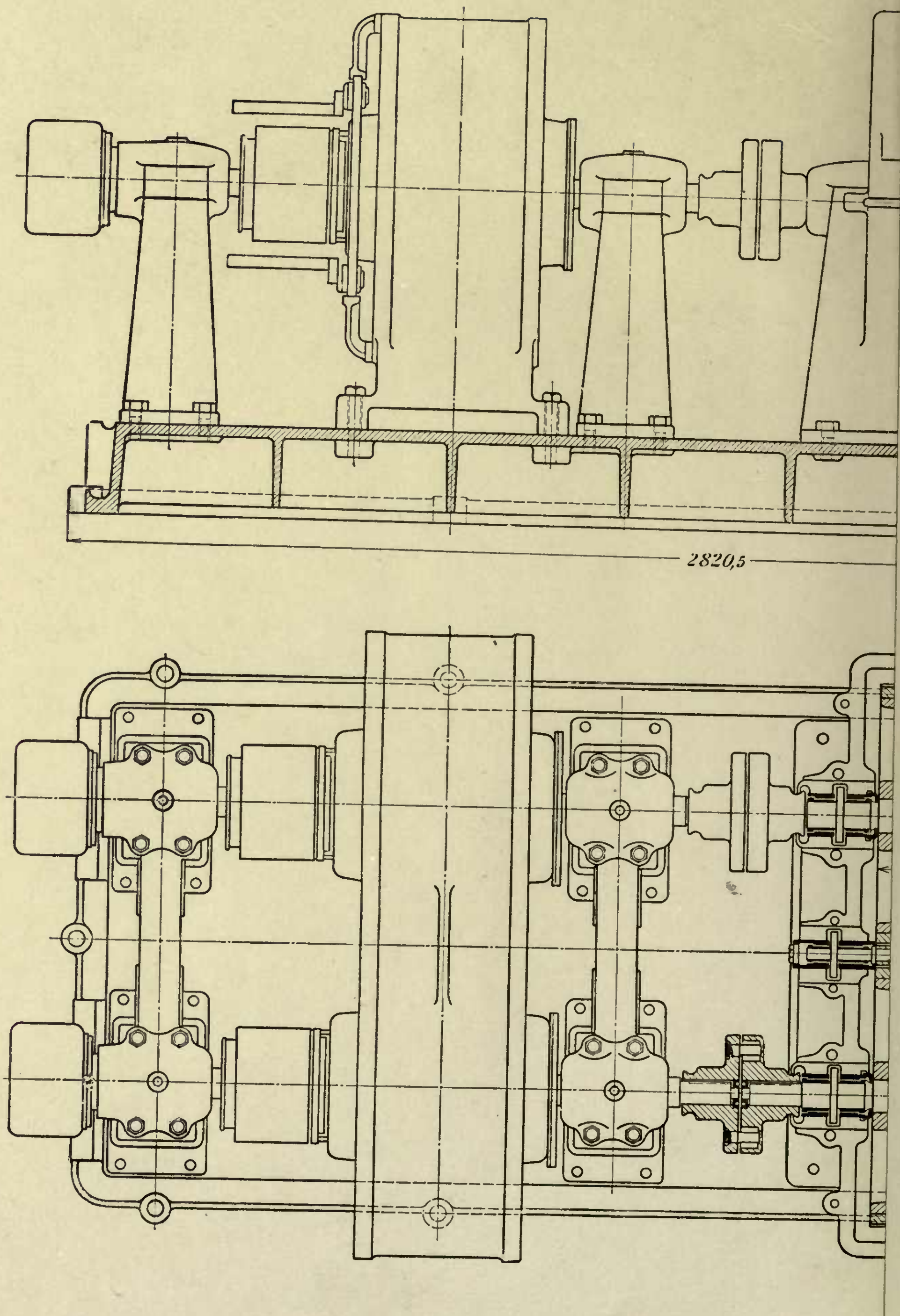
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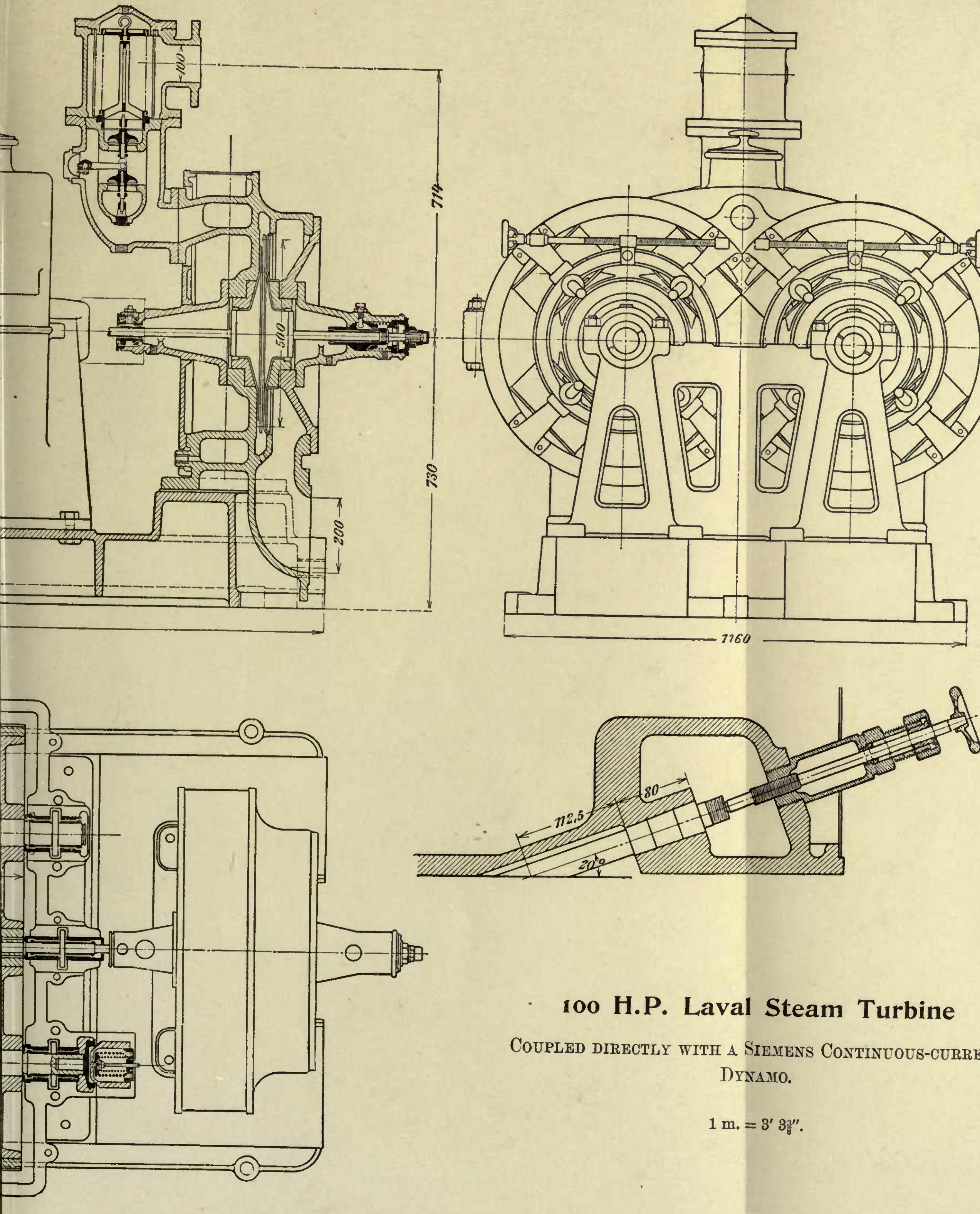
PLATE XVI.



100 H.P. Laval Steam Turbine

Coupled directly with a Siemens Continuous-current Dynamo.

1 m. = 3′ 3⅜″.

Arrangem

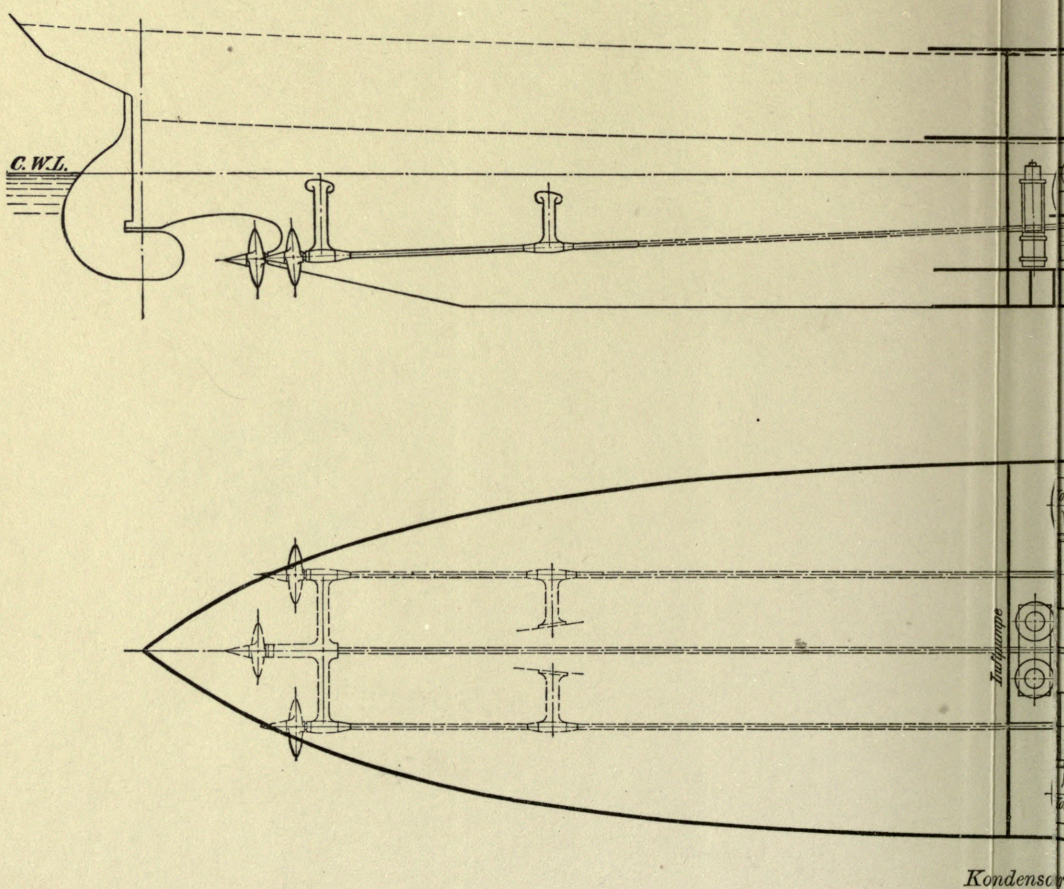
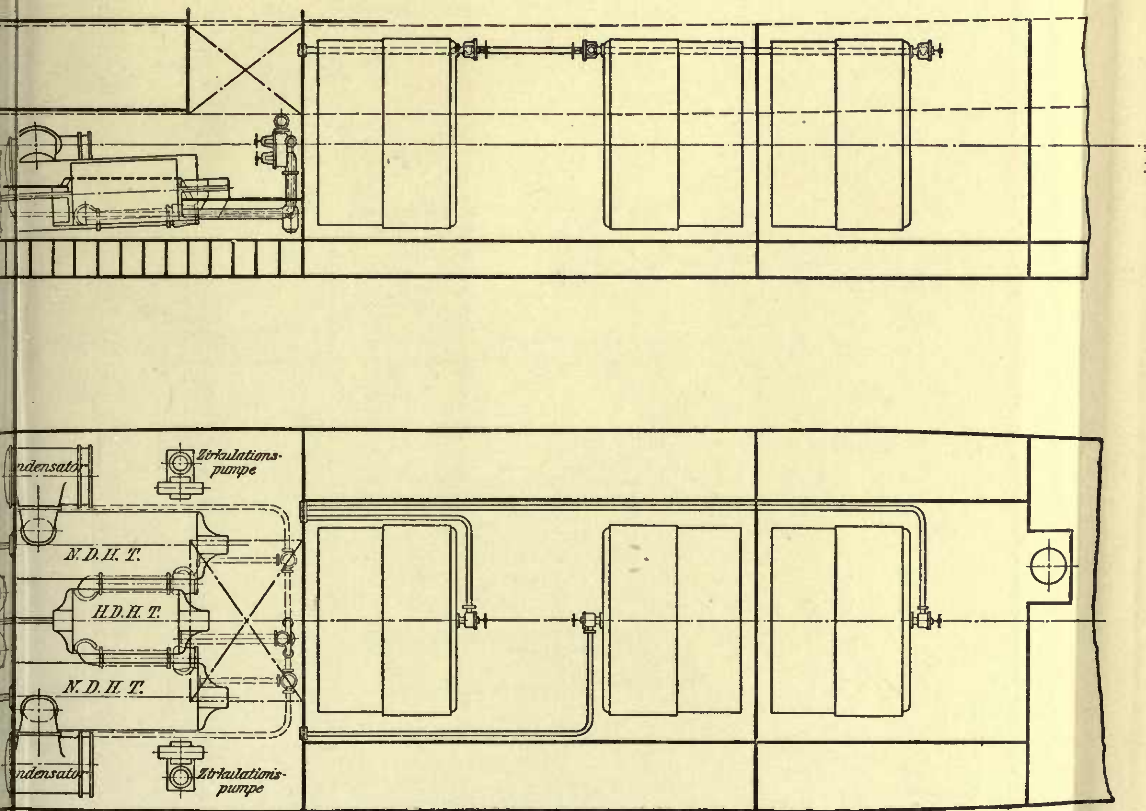


PLATE XVII.

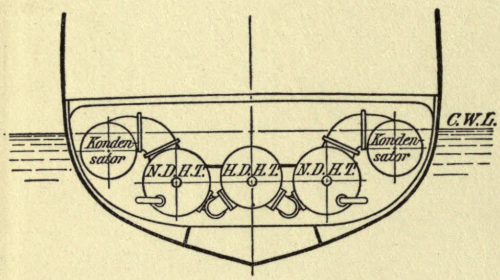
nt of the Parsons Turbines in a Passenger Steamer.

Scale 1 : 200.



at = Condenser.
io *pumpe* = Circulating pump.

N.D.H.T. = Low pressure main turbine.
H.D.H.T. = High pressure main turbine.



	Ft.	Ins.
Length between perps. on the W.L.	269	0
Breadth	34	6
Draught	11	6
Displacement	1500 tons.	
Power (indicated)	3600 H.P.	
Speed	18 knots.	

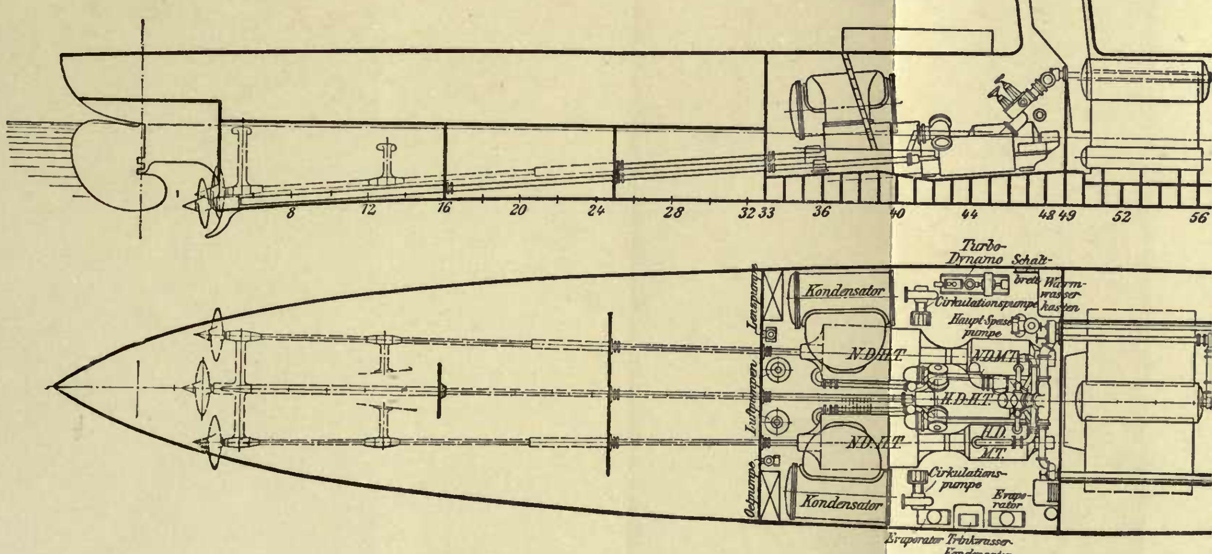
PLATE XVII

Arrangement of the Parsons Turb

Scale 1 : 200.

	Ft.	Ins.
Length	213	3
Breadth	21	4
Draught	6	1

Displacement . . .	360 tons.
Power.	6600 H.P.
Speed	28 knots.



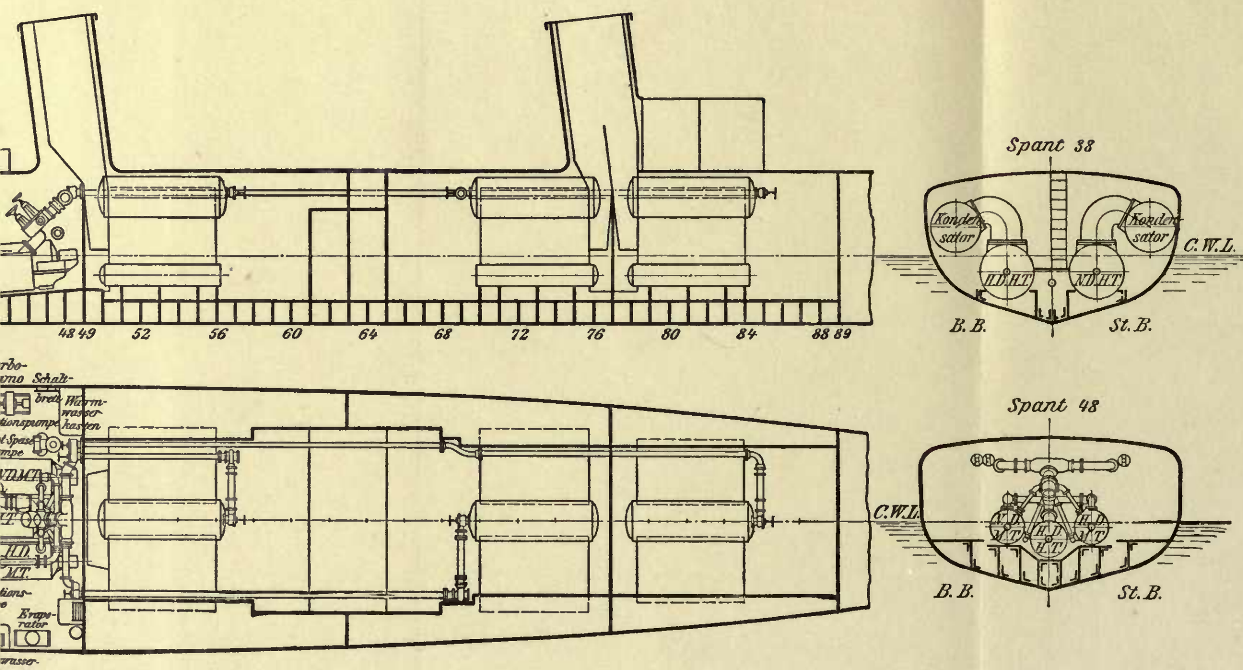
Spant = Frame.
Kondensator = Condenser.
N.D.H.T. = Low pressure main turbine.
H.D.H.T. = High pressure main turbine.
N.D.M.T. = Low pressure marching speed.
C.W.L. = Designed water line.

B.B. = Port (side).
St. B. = Starboard (side).
Turbo-Dynamo = Turbine dyn
Schaltbrett = Switchboard.
Zirkulationspumpe = Circulati
Haupt Speise-pumpe = Main f

PLATE XVIII.

Parsons Turbines in a Torpedo Boat.

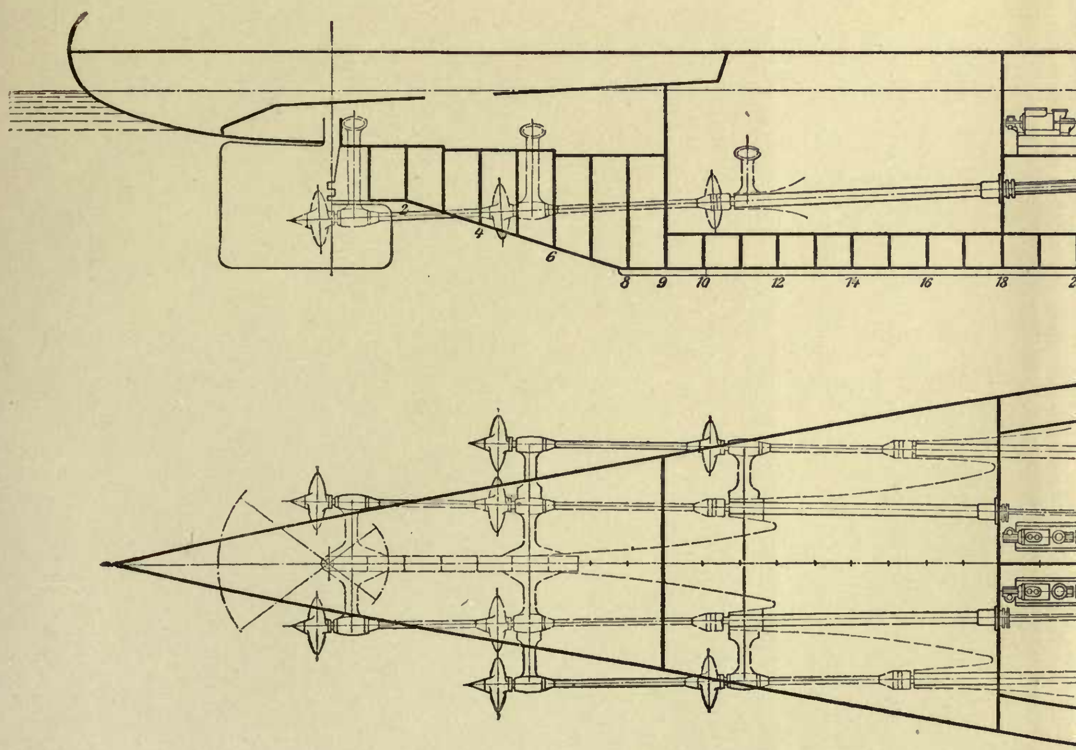
Scale 1 : 200.



= Port (side).
= Starboard (side).
-*Dynamo* = Turbine dynamo.
brett = Switchboard.
lationspumpe = Circulating pump.
t Speise-pumpe = Main feed pump.

Trinkwasser = Drinking water.
Warmwasserkasten = Warm-water tank.
Oelpumpe = Oil pump.
Luftpumpe = Air pump.
Lenzpumpen = Bilge pumps.

	Ft.	Ins.		
Length	328	0	Displacement . .	
Breadth	43	4	Power (indicated).	
Draught	25	5	Speed	



Spa
C.W
B.B
St.
Hoc

PLATE XIX.

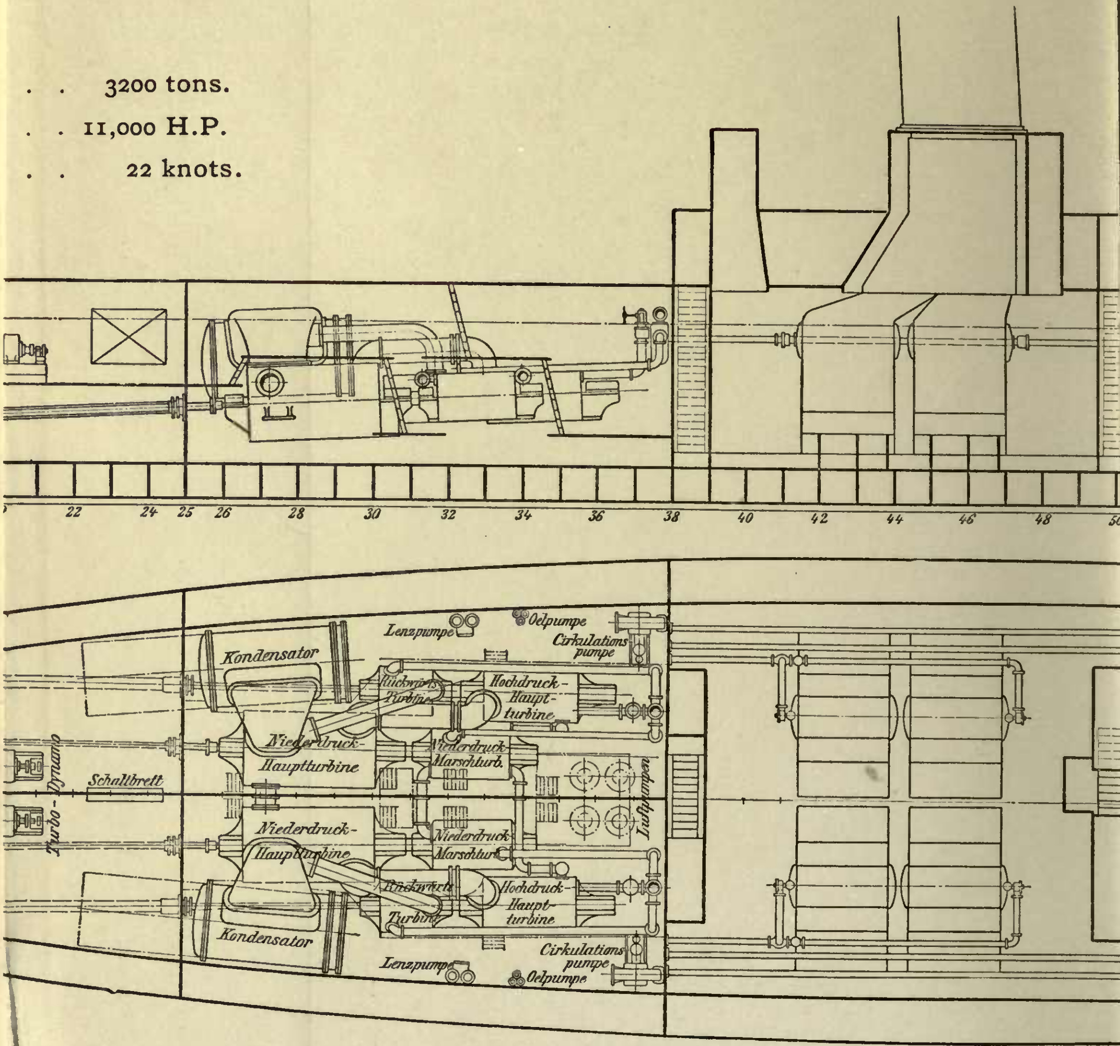
Arrangement of the Parsons Turbines in a

Scale 1 : 200.

. . 3200 tons.

. . 11,000 H.P.

. . 22 knots.



nt = Frame.

VL. = Designed water line.

B. = Port (side).

B. = Starboard (side).

hdruck Marscht. = High pressure march turbine.

Hochdruck Hauptt. = High pressure main turbine.

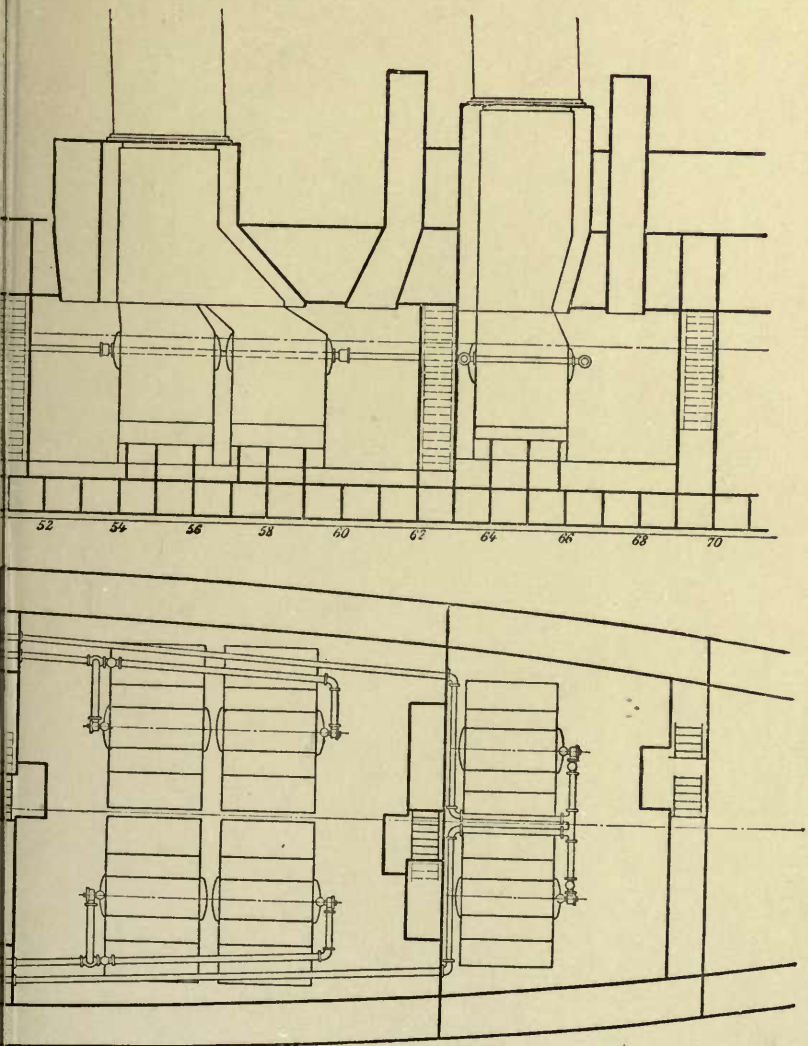
Niederdruck Marscht. = Low pressure march turbine.

Kondensator = Condenser.

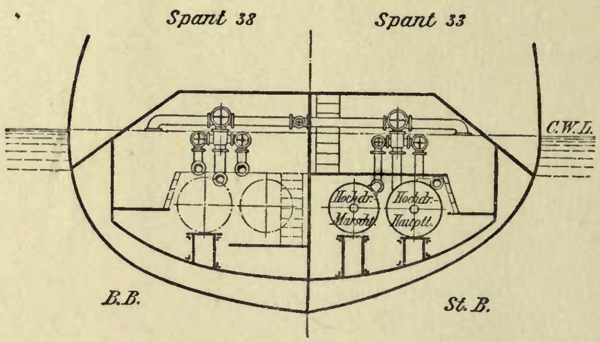
Niederdruck Hauptt. = Low pressure main turbine.

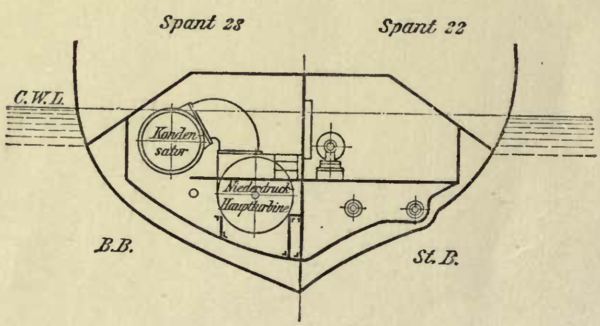
Turbo-Dynamo = Turbine dynamo.

ruiser.



Schaltbrett = Switchboard.
Lenzpumpe = Bilge pump.
Oelpumpe = Oil pump.
Cirkulations pumpe = Circulating pump.
Rückwärts Turbine = Go-astern turbine.


Spant 38
Spant 33
C.W.L.
Hochdr. Marscht.
Hochdr. Hauptt.
B.B.
St.B.


Spant 28
Spant 22
C.W.L.
Konden-sator
Niederdruck-Hauptturbine
B.B.
St.B.

$ m. 91/c a
ms o net