THE AIRPLANE ENGINE

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To

MY WIFE

JOSEPHINE PRESTON PEABODY

PREFACE

This volume attempts two things: to formulate existing knowledge of the functioning of the airplane engine and its auxiliaries; and to present and discuss the essential constructive details of those engines whose excellence has resulted in their survival.

The material here collected is largely new; very little of it could have been written before the war and only a small fraction was available for publication before 1919. It is based mainly on the researches and engine developments originating during the war and resulting from the war's urgencies. The researches have been carried out almost exclusively under governmental auspices; in the United States at the Bureau of Standards and at the Air Service experimental plant at McCook Field; in Great Britain at the Royal Aircraft Factory and the National Physical Laboratory; in France and Germany at equivalent institutions. Many of the results of these investigations were published confidentially during the war in Reports of the Bureau of Standards; in Bulletins and Technical Orders of the Airplane Engineering Division of the U. S. Army; in Reports of the (British) Advisory Committee for Aeronautics; in Bulletins de la Section Technique de l'Aéronautique Militaire; and in Technische Berichte. This material has now become available and much of it has been published in the Reports of the (U. S.) National Advisory Committee for Aeronautics and in the technical press.

Similarly, the constructive details of most of the existing airplane engines are now available, chiefly from descriptions of captured machines. The German and Austrian engines captured by the British were subjected to a technical analysis which has set a new standard in such matters. Not only were the engines and their auxiliaries tested exhaustively for performance but all the parts were minutely measured, loads and stresses calculated, and the metal analyzed for chemical composition. The French carried out similar analyses of German engines. The Germans published corresponding, though less detailed, analyses of

French, English and American engines. Since the war, the U. S. Air Service has also analyzed American and foreign engines and has published its findings in Technical Orders and Information Circulars.

With all this material, the designer of the airplane engine has at hand more detailed precedent from which to depart than is available for other types of engine.

The writer desires to acknowledge his indebtedness to Professor E. B. Warner and to Lieut. E. E. Aldrin for assistance in obtaining information; and to Mr. R. H. Taylor for assistance in reading proofs and in preparing the index.

L. S. M.

Cambridge, Mass.
January, 1922.

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THE AIRPLANE ENGINE

CHAPTER I

POWER REQUIRED AND POWER AVAILABLE

Power Required for Flight.—An airplane in flight is sustained by the lift of the wings. Consider the wing as a thin flat plate, Fig. 1. Four forces are acting:

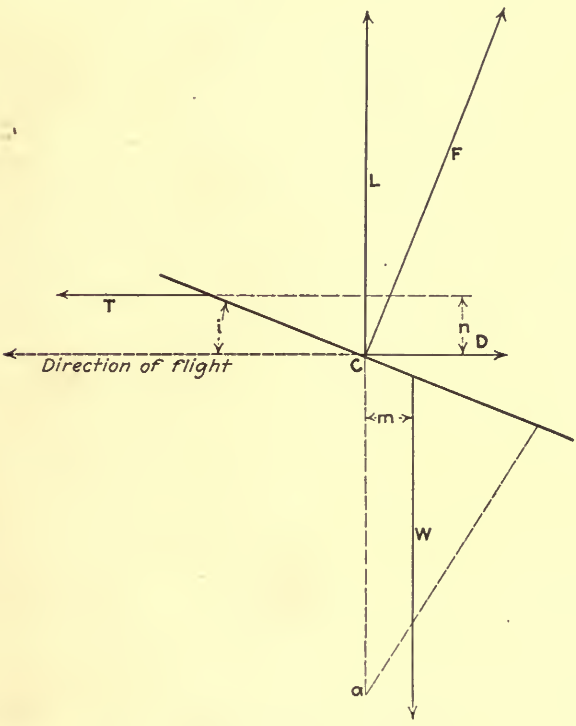


FIG. 1.—Forces acting on a flat plate.

1. The **weight,** W, of wing and parts supported thereby, downward.
2. The **thrust,** T, or forward impulse due to the propeller.
3. The **lift,** L, of the air which acts in a direction perpendicular

to that of the plane with respect to the air, and produces sustentation.

4. The **wing resistance** or **drag,** D, measured perpendicularly to L, the component of the total force on the wing which opposes forward motion.

At constant horizontal speed, $L = W$ and $D = T$. As these pairs of forces are not acting in the same lines they give rise to turning moments and it is necessary for stability that $L \times m = T \times n$. Both L and D are created by the velocity of the plane; the direction of the latter is concurrent with (but opposite to) the path of flight. The former is directed at right angles with the path of flight. In horizontal flight L and D are vertical and horizontal forces respectively.

Wing Characteristics.—*Flat Plates.*—A plate moving with respect to the air undergoes an approximately normal pressure F which is proportional to the *density* of the air and (within limits) to the square of the *relative velocity.* This pressure may be resolved into components L and D perpendicular and parallel to the path of flight. $F^2 = L^2 + D^2$. The component L is useful, while D is objectionable. The pressure F depends on the *area* of the plate, but varies somewhat with its shape.

The incidence, i, or angle between the plate and the flight path has a dominating influence on the resulting pressure; and particularly on the relation between L and D. Since $L = F \cos i$ and $D = F \sin i$, L/F decreases and D/F increases, as i is increased from 0 deg.

For a given angle of incidence the following relations hold:

$$L = K_L d A V^2$$
$$D = K_D d A V^2$$

where K_L and K_D are experimentally determined lift and drag coefficients respectively; d is the air density; A is the wing area; and V the relative air velocity. The usual units are L and D in pounds; d, relative air density in terms of normal density; A in square feet; and V in miles per hour. From the above equations it is seen that $\frac{L}{D} = \frac{K_L}{K_D}$. That is, lift is obtained with minimum drag when $\frac{K_L}{K_D}$ is a maximum.

It is found that more favorable ratios of K_L/K_D are obtained with **curved wings,** as in Fig. 2, than with flat plates. The angle of incidence of such a wing is arbitrarily defined as the acute angle between the wind direction and the lower chord of the wing. Figure 3 gives the values of K_L, K_D, and K_L/K_D or L/D for the wing or aerofoil section shown in Fig. 2, and shows a maximum value of L/D of 17 at an angle of incidence of 3 deg.

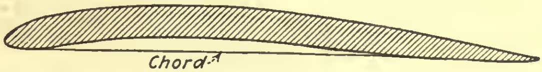


FIG. 2.—Cross section of wing.

An **airplane** consists not only of the wings which give sustentation, but also of other members such as the fuselage, radiator, landing gear, and wing bracing. These give no aid in sustaining the plane but offer a resistance, the **parasite resistance,** which

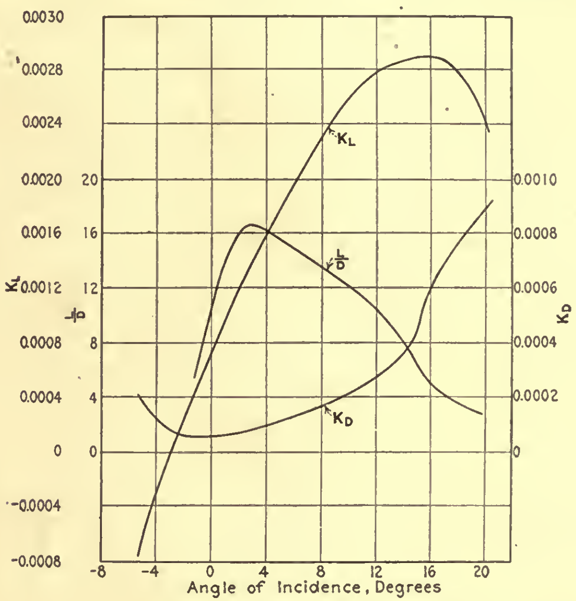


FIG. 3.—Lift and drag coefficients of a wing.

must be overcome in flight. If P is the parasite resistance in pounds

$$P = K_N d V^2$$

where K_N is a coefficient with practically constant value in any given airplane.

The **total resistance,** R, to the motion of the plane in the direction of flight is the sum of the wing and parasite resistances, or

$$R = D + P = d(K_D A + K_N)V^2$$

This resistance must equal the propeller thrust, T, in uniform flight. The horse power required to overcome the resistance is

$$H_R = \frac{1.467\,RV}{550} = \frac{RV}{375}$$

where 1.467 is the constant to convert miles per hour to feet per second and 550 is the equivalent of 1 hp. in foot-pounds per second. If the efficiency of the propeller is e, the brake **horse power,** H_B, **required** at the engine for horizontal flight at speed V is:

$$H_B = \frac{RV}{375e} = \frac{d(K_D\,A + K_N)V^3}{375e}$$

The lift, L, must equal the weight of the plane, W, in horizontal flight, and the equation $W = K_L dA\,V^2$ must be satisfied simultaneously with the b.h.p. equation, with values of K_L and K_D corresponding to some one angle of incidence. In a design W, A, V, K_N are known by assumption or calculation and it is required to find H_R, K_L and K_D for a series of values of V. Since K_L and K_D are functions of i there are only two independent variables H_R and i to be found.

The power required in an airplane of 200 sq. ft. wing area with a total weight of 1,200 lb. using the wing with the properties shown in Fig. 3 and with parasite resistance $P = 0.0025\ V^2$ is shown in Fig. 4. It will be seen that the plane has a minimum possible velocity of about 44 miles per hour but that the power required to drive it starts to increase very rapidly if the speed gets below 46 miles per hour. The flying conditions are for speeds of 46 miles per hour or higher. At lower speeds there are reversed controls, that is, more power is required to go slower, and the angle of incidence is so high that the plane will be in danger of stalling. The *power required* curve of Fig. 4 is of a form which may be regarded as typical.

If the propeller does more work than is required the plane will climb and its rate of climbing will depend only on the amount of the excess power; all the additional work goes into raising the plane. If, on the other hand, power is deficient and the propeller does less work than is required to keep the plane in level flight at the existing speed, the plane will glide down and the work done by the falling plane (weight × fall) will be exactly equal to the difference between the work required to keep the plane moving with its existing speed in the direction of flight and the work done by the propeller. If the power available is at any time in excess of that required for level flight, it can be reduced by throttling the engine. There is only one speed possible in level flight for any given angle of incidence. Opening the throttle does not, as with automobiles, increase the speed in level flight, but will start the airplane climbing unless the incidence is also

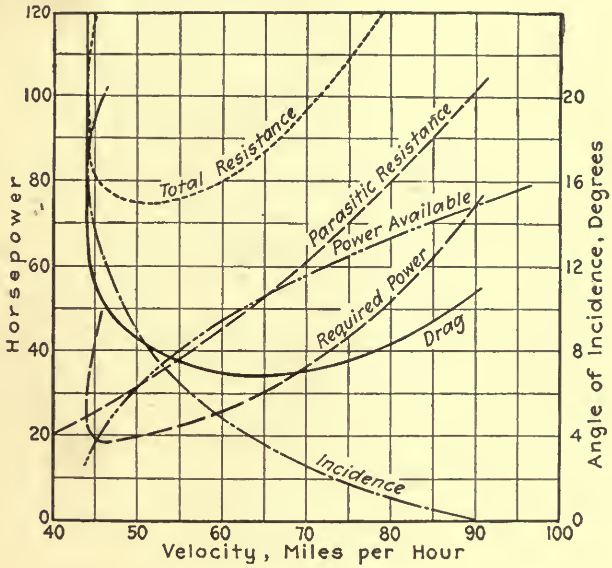


FIG. 4.—Characteristics of an airplane in level flight.

changed by operating the elevator. The power supplied by the engine with a given incidence cannot affect the velocity of the plane but will determine whether the plane climbs, flies level, or glides.

The horse power available for driving the plane is given by $\frac{TV}{375}$, that is, at any plane speed it depends only on the propeller thrust. As the thrust depends on the engine speed both the engine and the propeller characteristics must be examined before the thrust can be determined.

Air Propellers.—The propeller converts the torque at the engine crankshaft into a thrust along the axis of the propeller shaft. Its action is remotely similar, to the turning of a solid screw in a solid nut and the same general terminology is employed. The axial distance that would be travelled by the propeller for one revolution if the air were incompressible is called its **nominal pitch**; the nominal pitch divided by the outside diameter is the **nominal pitch ratio.** An air propeller does not, however, advance a constant amount for one revolution; it is advancing in a medium which is readily displaced, and it is found that it must advance each revolution a distance greater than the nominal pitch if it is not to displace the air backward. This greater distance is called the **dynamic pitch.** The difference between the dynamic pitch and the actual or **effective pitch** is called the **slip.** Positive thrust can be obtained only when there is slip and the greater the slip the greater will be the thrust; maximum thrust is obtained when the plane is on the ground in which case the effective pitch is zero.

If a propeller makes N revolutions per minute and the plane moves through the air with a velocity of V feet per minute the effective pitch is V/N and the **effective pitch ratio** is V/ND.

The characteristics of propellers are most easily determined by tests on models. It is found that geometrically similar propellers have the same characteristics for the same values of V/ND. The important characteristics are the **thrust,** T, lb.; **torque,** Q, lb.-ft.; **torque horse power,** H_B; **thrust horse power,** H_R; and **efficiency,** e. The torque horse power is necessarily the same as the engine brake horse power. The efficiency, $e = H_R/H_B$.

Torque and thrust are given by the following equations:

$$Q = \frac{w\ V^2\ D^3 \cdot a}{1000}$$

$$T = \frac{w\ V^2\ D^2 \cdot b}{100}$$

where a and b are the torque and thrust coefficients respectively and w is the air density in pounds per cubic foot.

Efficiency is obtainable directly from these coefficients.

Efficiency =

$$\frac{\text{Thrust horse power}}{\text{Torque horse power}} = \frac{TV}{2\pi QN} = \frac{10b}{2\pi a}\ \frac{V}{ND} = 1.59\frac{b}{a} \cdot \frac{V}{ND}$$

Values of these coefficients and of e have been determined by Durand[1] for propellers of many types and proportions. The

[1] *Nat. Adv. Comm. on Aeronautics,* 1917.

curves of Fig. 5 give typical values obtained from three propellers which differ only in nominal pitch ratio; the values of the pitch ratio are 0.5, 0.7 and 0.9 respectively. By the use of such curves the horse power required can be easily obtained. For example, assume a propeller 8 ft. in diameter (Fig. 5) with pitch ratio 0.9, making 1,200 r.p.m. and with a speed of 72 miles per hour (105.6 ft. per second) at an elevation of 3,000 ft. The value

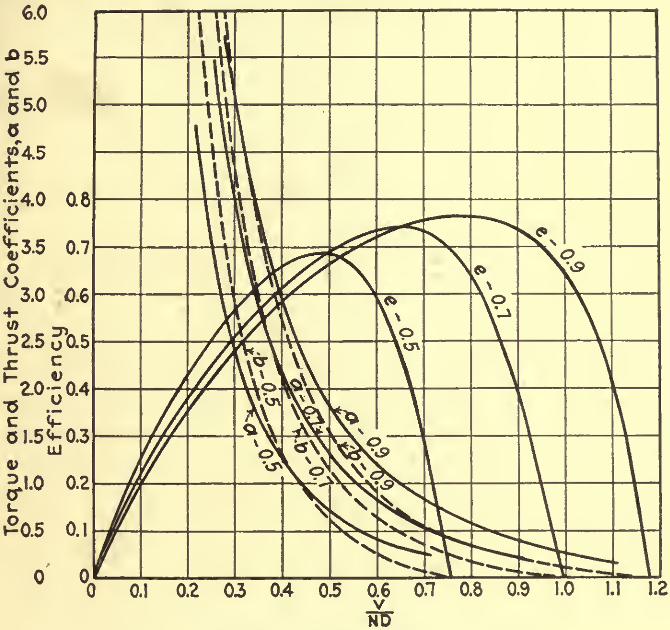


FIG. 5.—Propeller coefficients and efficiencies.

of $V/ND = 0.66$. From Fig. 5, $b = 0.685$, and $a = 0.970$. Also $w = 0.071$ lb. per cubic foot. Then

$$T = \frac{0.071 \times (105.6)^2 \times 8^2 \times 0.685}{100} = 347 \text{ lb.}$$

$$Q = \frac{0.071 \times (105.6)^2 \times 8^3 \times 0.970}{1,000} = 393 \text{ lb.-ft.}$$

Torque horse power, H_B, =

$$\frac{2\pi QN}{33,000} = \frac{2\pi \times 393 \times 1,200}{33,000} = 89.5$$

$$\text{Efficiency} = 1.59 \times \frac{0.685}{0.970} \times 0.66 = 0.742$$

The efficiency can also be read directly from Fig. 5.

Power Available for Flight.—The horse power available with a given engine and given plane speed, V, can now be determined. Throughout a considerable range of revolutions per minute the mean effective pressure in the engine is practically constant but falls off at highest speeds. If it is assumed constant the propeller torque will be constant. The torque equation can be written

$$\frac{a}{1{,}000} \cdot \left(\frac{V}{ND}\right)^2 = \frac{Q}{N^2D^5}$$

or since Q and D are constant

$$N = \frac{\text{Constant}}{\left(\frac{V}{ND}\right) \times \sqrt{a}}$$

Assuming various values of $\frac{V}{ND}$ and finding in Fig. 5 the corresponding values of a, a series of values of N can be obtained and substituting these values in V/ND the corresponding values of V are determined. That is, the change of revolutions per minute with flying speed, V, or with V/ND is obtained. In Fig. 6 there are plotted curves[1] showing this variation for propellers of nominal pitch ratios of 0.5, 0.7 and 0.9 respectively. These curves do not correspond exactly to the coefficients of Fig. 5; they average the results of Durand's tests and apply fairly well to standard forms of propeller. The unit values of V/ND and N are those corresponding to maximum efficiency. The product of the ordinates and abscissae for any point is $(V/ND) \times N = V/D$, and since D is constant, the ratio of this product at two points is also the ratio of the corresponding plane velocities.

As an example suppose a propeller with nominal pitch ratio 0.7 designed for an airplane flying normally at 100 miles per hour and that the full engine power is absorbed at 1,600 r.p.m. If the speed of the machine increases so that $V/ND = 1.1$, then N becomes 1.052 and V increases in the ratio $1.1 \times 1.052 = 1.157$. The propeller will then turn $1.052 \times 1{,}600 = 1{,}683$ r.p.m. on full throttle at a plane speed of 115.7 miles per hour.

The torque horse power is proportional to the engine speed; the efficiency can be obtained from Fig. 5, and multiplied by

[1] Supplied by E. P. Warner.

the torque horse power will give the available horse power. In the example just given, assume that the propeller has the characteristics shown in Fig. 5 and that the torque horse power at maximum efficiency is 100. Maximum efficiency is seen to be for $V/ND = 0.64$, which therefore corresponds to the unit of Fig. 6. For $V/ND = 1.1$ on Fig. 6 we have $V/ND = 0.64 \times 1.1 = 0.704$ in Fig. 5; the corresponding efficiency is seen to be 0.74. The torque horse power is 100×1.052 and the available horse power is $105.2 \times 0.74 = 78$. By finding the available

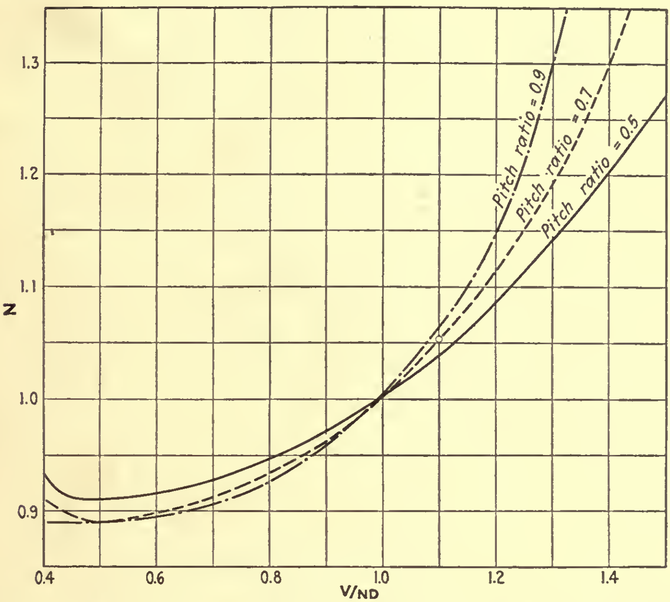


Fig. 6.—Variation of revolutions per minute N, with speed of airplane, V; engine torque constant.

horse power for a number of other values of V/ND and plotting them against V the available horse power curve of Fig. 4 is obtained.

If engine torque varies with the engine speed, N, another procedure must be followed in finding the values of N corresponding to different values of V. Taking the propeller torque equation $Q = \dfrac{wV^2D^3a}{1{,}000}$ a series of curves may be drawn, as in Fig. 7, each giving the change in propeller torque with speed at constant V. If the engine torque is now drawn on the same figure its intersection with the constant V curves will give the values of N at which engine and propeller torques are equal, that is, it will give the operating speeds. The available horse power can then be found by the use of the efficiency curve of Fig. 5.

Returning to Fig. 4 it is seen that with wide-open throttle the available horse power is equal to the required horse power at 45 miles per hour and at 90 miles per hour; between these limits the available horse power is in excess of the power required for level

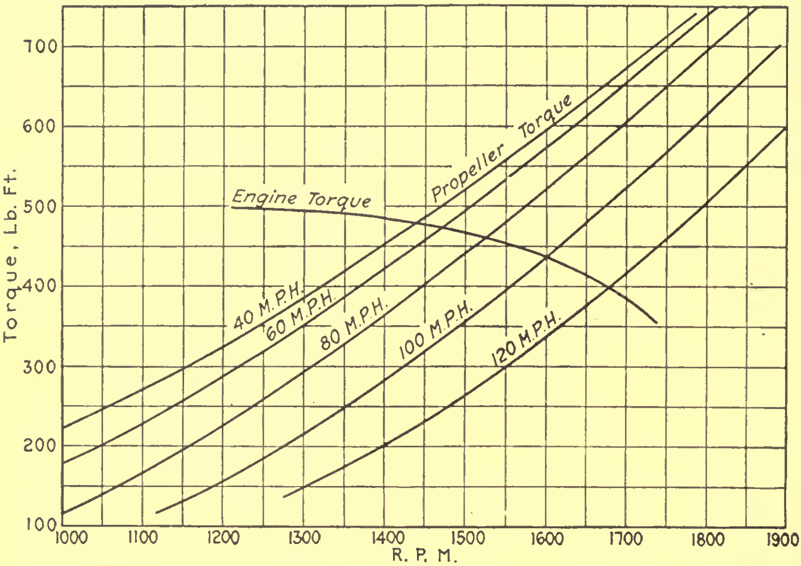


Fig. 7.—Variation of engine and propeller torque with revolutions per minute at various airplane speeds.

flight; outside these limits the engine power is insufficient and the plane will glide. Between 45 and 90 miles per hour level flight can be maintained only by closing the throttle and the speed of flight will depend on the angle of incidence. With the throttle wide open the plane will climb and its rate of climb will be greatest at that speed and corresponding angle of incidence at which the difference between the available power and required power is a maximum. In the case of Fig. 4 this will be at about 65 miles per hour and an angle of incidence of 3.6 deg.

CHAPTER II

ENGINE EFFICIENCIES AND CAPACITIES

A typical airplane engine is shown in transverse section in Fig. 8 and in longitudinal section in Fig. 9. It has six vertical, single-acting, water-cooled cylinders in a row, driving the crankshaft, *b*, through pistons, *p*, piston pins, *g*, connecting rods, *r*, and crankpins, *k*. The crankshaft is supported on seven bearings, carries the propeller hub, *d*, at its forward end with a thrust bearing, *e*, behind it and has a bevel wheel, *f*, at its rear end meshing with another bevel wheel on the vertical shaft, *l*. The shaft *l* drives the camshaft, *c*, through bevel gearing at its upper end and carries at its lower end the centrifugal water pump, *n*, and the two gear oil pumps, *o;* it also drives the magneto, *m*, through helical gears, *q*. The camshaft, *c*, carries inlet cams which act on the rocker levers, *t*, and open the inlet valves, *v*, against the compression of the valve springs, *h;* there is one inlet valve to each cylinder. The camshaft also carries cams which act directly on the tappets of the exhaust valves, *u*, of which there are two per cylinder. The water jackets, *w*, surround the cylinders and the valve cages. Air enters the carburetor, *a*, and goes through the inlet manifold, *i*, and past the inlet valve, *v*, to

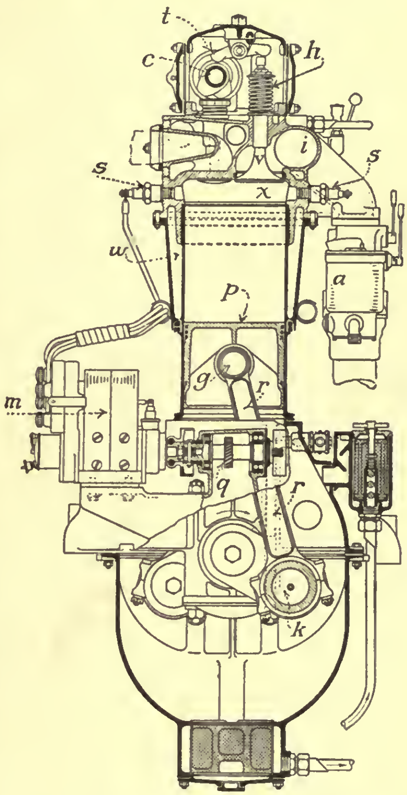


Fig. 8.—Transverse section of Siddeley "Puma" airplane engine.

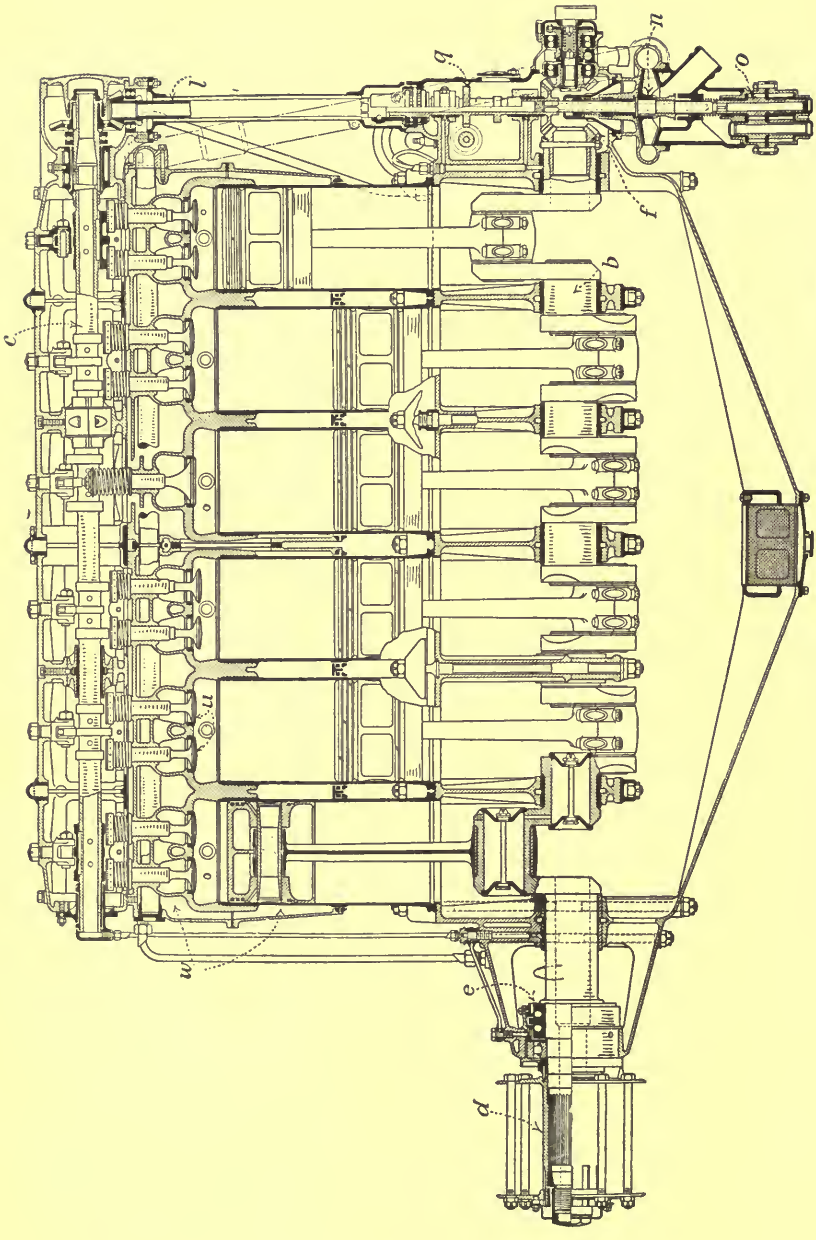


FIG. 9.—Longitudinal section of Siddeley "Puma" airplane engine.

the cylinder, x. After compression the charge is ignited by the spark plugs, s, which get their current from the magneto, m.

Actions Occurring in the Cylinder.—Figure 10 is a **theoretical indicator card** for an engine using the Otto cycle, which is always used in aviation engines. The indicator card shows (vertically) the gas pressure inside the cylinder at each position of the piston. At the position a the piston is at the end of its stroke most remote from the crankshaft and the volume of is the volume of the clearance or combustion space; the total volume displaced by the piston is represented by fg and is the product of the piston area by its stroke; the maximum volume of gas in the cylinder is og.

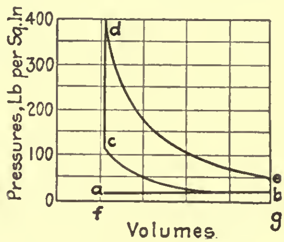


FIG. 10.—Ideal indicator card of Otto-cycle engine.

The cycle of operations inside the cylinder begins with the piston at a and with the gas inlet valve open, establishing free communication between the cylinder and the external air. The piston makes its stroke from a to b with the inlet valve kept open; the pressure will remain atmospheric. This is the **suction stroke.** The inlet valve now closes and the piston returns. Since both valves are closed, the mixture in the cylinder is compressed (curve bc, **compression stroke**) the final pressure depending mainly upon the volume of the clearance space into which the gas is crowded at the inner end of the stroke.

When the piston is at c (actually a little sooner than this) the compressed mixture is ignited and the pressure suddenly increases (line cd). This pressure drives the piston back, the burnt mixture expanding (curve de) while the valves remain closed. This is the **power** or **expansion stroke.**

At the end, e, of the expansion stroke, the exhaust valve opens. The pressure drops to atmospheric (line eb). The piston returns and the burned gas is driven out at the exhaust valve at atmospheric pressure. At the end of this **exhaust stroke,** ba, the exhaust valve closes, the inlet valve opens, and the cycle recommences.

SUMMARY

STROKE	ACTION	INLET VALVE	EXHAUST VALVE
First out	Suction	Open	Closed
First in	Compression	Closed	Closed
Second out	Explosion and expansion	Closed	Closed
Second in	Exhaust	Closed	Open

Pressures and Temperatures in the Ideal Cycle.—Let p be the absolute gas pressure in pounds per square inch, v the volume in cubic feet, t the temperature Fahrenheit and T the absolute temperature. The suffixes a, b, c, etc., indicate the points on the cycle, shown in Fig. 10, at which these quantities are being considered. In the ideal cycle, the volume $v_b - v_a$ of explosive mixture is taken into the cylinder during the admission period and is still at atmospheric pressure and temperature at the end of the stroke b. The mixture or "charge" is now compressed adiabatically, that is, without addition or abstraction of heat. As a result of the compression the pressure and temperature increase; the *compression pressure* is given by the equation

$$\frac{p_c}{p_b} = \left(\frac{v_b}{v_c}\right)^{1.4} = r^{1.4}$$

where $r = \frac{v_b}{v_c}$ is the **ratio of compression.** The temperature is given by

$$\frac{T_c}{T_b} = r^{0.4}$$

If the atmospheric pressure, p_b, is 14.7 lb. per square inch and the temperature t_b is 60°F., or, $T_b = 460 + 60 = 520°$ absolute, then, with $r = 4.8$, $p_c = 9 \times 14.7 = 132.3$ lb. per square inch and $T_c = 1.873 \times 520 = 973°$ absolute, or, $t_c = 973 - 460 = 513°$F.

As a result of the explosion at c the heat of combustion is liberated and is used in heating up the charge. The heat of combustion may be assumed to be 80 B.t.u. per cubic foot of charge admitted, or $80\ (v_b - v_a)$ B.t.u. The weight, w, of gas in the cylinder is given by the perfect-gas equation

$$144\ pv = WRT,$$

where R is the gas constant, which may be assumed to have the value 52 for the usual explosive mixture. For the conditions assumed, the weight of gas is

$$w = \frac{144\ pv}{RT} = \frac{144 \times 14.7}{52 \times 520} v_b = 0.0784\ v_b.$$

The heat required to raise 1 lb. of this gas 1°F. while the volume remains unchanged is called the *specific heat at constant volume*, C_v, and may be taken as 0.171 B.t.u. The rise in temperature during explosion is given by the equation

Heat of explosion, H, = weight of gas × specific heat × rise of temperature, or

$$H = wC_v(T_d - T_c)$$

For the conditions assumed,

$$80(v_b - v_a) = 0.0784\, v_b C_v(T_d - T_c)$$

and since

$$\frac{v_b}{v_a} = 4.8$$

$$T_d - T_c = 80 \times \frac{3.8}{4.8} \times \frac{1}{0.0784 \times .171} = 4{,}740°$$

and the *explosion temperature*, T_d, = 4,740 + 973 = 5,713°.

The *explosion pressure*, p_d, is given by the equation

$$\frac{p_d}{p_c} = \frac{T_d}{T_c}$$

$$\text{or, } p_d = \frac{5{,}713}{973} \times 132.3 = 778 \text{ lb. per square inch.}$$

The expansion curve in the ideal cycle is adiabatic, so that the equations are the same as for the compression. The pressure p_e at *release* or beginning of exhaust, e, is given by

$$\frac{p_d}{p_e} = \left(\frac{v_e}{v_d}\right)^{1.4} = r^{1.4}$$

and

$$\frac{T_d}{T_e} = r^{0.4}$$

$$\text{Consequently, } p_e = \frac{778}{9} = 86.4 \text{ lb. per square inch}$$

and

$$T_e = \frac{5{,}713}{1.873} = 3{,}060°$$

The **actual indicator card** is shown in Fig. 11. At the end of the exhaust stroke the pressure in the cylinder is above that of the atmosphere; usually, p_a = 15.4 to 17 lb. per square inch. This pressure falls as the piston moves down and the fresh charge is drawn in. Mixing with the residual burned gas this charge is somewhat heated, the temperature, t_b, being usually from 170 to 260° and the pressure, p_b, 12 to 14 lb. absolute. This pressure depends on the engine speed and on the resistance

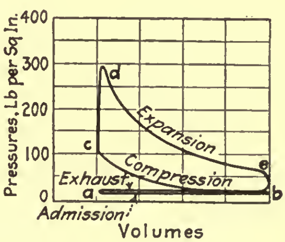


Fig. 11.—Actual indicator card of Otto-cycle engine.

encountered by the fresh charge in the carburetor, manifold, and inlet valve.

The compression curve *bc* is not adiabatic, but may be represented by the expression $p_c v_c^n = p_b v_b^n$, where n has a value between 1.23 and 1.35. The compression pressure, p_c, is determined by the clearance volume: 110 to 120 lb. gage is near the average of good practice for water-cooled engines. The compression pressure tends to fall with increase of engine speed in consequence of the increase of vacuum in the cylinder at the moment of closure of the inlet valve. The curve for the Liberty engine (Fig. 12) shows this effect. The maximum temperature, t_d, following ignition, is usually from 2,500 to 3,200°F., and the maximum pressure, p_d between 300 and 400 lb. When p_c is increased, p_d also increases. The expansion curve *de* also is not adiabatic but may be represented by $p_d v_d^n = p_e v_e^n$, in which n = 1.27 to 1.5; the low values are realized in cases where combustion continues after expansion has been well started. The terminal pressure, p_e, is from 38 to 75 lb., and t_e from 1,200 to 2,000°F. The pressure during exhaust varies from 17 to 15.4 lb. per square inch.

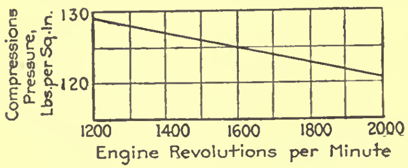


Fig. 12.—Variation of compression pressure with engine speed.

The compression ratio, $\frac{v_b}{v_c}$, is fixed by the clearance and may be varied in a given engine by the use of pistons with heads of different shapes. It is the chief factor in determining p_c. The accompanying table is based on p_b = 12.5. A high compression

Table of Compression Pressures, Pounds per Square Inch

n	Compression ratios					
	3.5	4.0	4.5	5.0	6.0	7.0
1.25	59.9	70.6	81.9	93.2	116.5	143.0
1.30	63.7	75.8	88.3	101.3	128.4	156.9
1.35	67.8	81.2	95.2	109.8	140.4	172.9
1.41	73.1	88.3	104.2	120.9	156.4	194.3

ratio increases power output and efficiency. It also increases the temperature at the end of compression since

$$\frac{T_c}{T_b} = \frac{t_c + 460}{t_b + 460} = \left(\frac{p_c}{p_b}\right)^{\frac{n-1}{n}}$$

A high value of t_c may cause preignition and thus fixes a limit of compression ratio which must not be exceeded. Usual values are from 4.5 in hydroplanes to 5.6 for high-altitude land machines. Values of compression ratio are given for various engines on pages 66 to 70, where it is seen that high values result in increased power output per unit of cylinder volume. Values of six or higher may be used for high altitude work, but the engines will develop preignition if operated at full throttle near the ground. When operated on partial throttle the entering charge is of reduced weight, both because of lower pressure and because of dilution of the fresh mixture with a relatively greater weight of burnt gas remaining over from the previous cycle. Consequently the heat developed per cycle is less and the mean temperatures in the cylinders are reduced.

Efficiency.—If the working substance in the cylinder followed the laws of a perfect gas

$$pv = RT$$

and $$C_v = \text{constant}$$

and if the combustion were instantaneous and complete, the efficiency of the cycle would be equal to

$$e = 1 - \left(\frac{1}{r}\right)^{n-1} \tag{1}$$

where r is the compression ratio. It is here assumed further that the cycle takes place without any heat exchange between the working charge and the cylinder. The efficiency so found is the highest possible efficiency for an engine operating on the Otto cycle and could be attained only under the conditions stated above. It is sometimes called the **air-cycle efficiency.** Its value for various compression ratios is given in the following table:

r = 3.50	4.00	4.50	5.00	5.50	6.00	6.50	7.00
e = 0.40	0.43	0.45	0.48	0.50	0.51	0.53	0.54

The efficiencies actually obtained in airplane engines are seldom greater than 60 per cent of these values. For instance, with $r = 5.5$, the actual efficiency will not exceed $0.6 \times 0.5 = 0.3$

and will in general be between 0.25 and 0.3. This considerable discrepancy between the actual performance of the engine and the air cycle efficiency is due to a variety of causes, the principal of which are as follows:

I. The theoretical cycle assumes that the total heat (lower heat value) of combustion of the fuel taken into the cylinder is utilized during explosion in heating up the working mixture. This is not actually the case for two reasons:

(*a*) The whole of the heat of combustion is not evolved during explosion because combustion is not instantaneous, so that combustion will continue for part (or with incorrect mixture, for the whole) of the expansion stroke, thereby reducing the amount of heat available for conversion into work. Furthermore, complete combustion at the end of explosion is not attainable because chemical equilibrium requires the presence of a certain amount of hydrogen and carbon monoxide. Their existence is commonly ascribed to dissociation. The amount of heat suppression from these causes is not considerable in a high-grade engine operating with gasoline and with a good mixture.

(*b*) Some of the heat actually evolved goes to the cylinder walls by radiation and conduction. The total heat so going to the walls in airplane engines is from 25 to 30 per cent of the total heat of combustion. If the heat were abstracted from the burning mixture during the explosion it would result in a loss of efficiency of the same magnitude. The actual passage of heat from the mixture to the walls continues from the middle of compression to the end of exhaust. Throughout the whole of this time the gases are hotter than the walls. The heat flows in the opposite direction during the admission period and the first part of the compression, but the amount of heat thus flowing is small, as the temperature difference between the walls and the gases is small. Such heat as passes off during the explosion and the first part of the expansion stroke may be regarded as entirely lost to the engine; the heat flow to the walls near the end of expansion and during exhaust is no loss at all, as it is necessary to discharge the hot gases and it is immaterial, from the point of view of efficiency, whether the heat is carried away by the jacket water or in the exhaust gases.

General experience would indicate that more than one-half of the total heat given to the jacket may be regarded as abstracted by radiation or conduction from the working substance during explosion and the early part of expansion. It should be noted in this connection that the heat of the jacket water includes most of the friction work between the piston and cylinder, which is a considerable fraction of the total friction of the engine. As the jacket heat is 25 to 30 per cent, the heat lost during explosion by radiation and conduction may be taken as not more than 12 to 15 per cent of the heat of complete combustion.

II. The working substance is not a perfect gas and, in particular, it is not true that the specific heat at constant volume is a constant. It is found on investigation that the gases (CO_2, N_2, H_2O etc.) which are present in the cylinder after explosion have specific heats which increase considerably with increase of temperature. These specific heats follow the equation

$$C_v = a + bt$$

where a and b are constants.

The efficiency of the cycle is diminished as a result of this increase of specific heat. The immediate result is that the rise of temperature during explosion, for a given amount of fuel burned, is diminished and consequently the pressure, p_d (Fig. 10), is lower than would be realized with constant specific heat. The expansion curve *de* is consequently lowered and the work of the cycle diminished. The efficiency with adiabatic expansion and compression but with variable specific heat is given by the expression,[1]

$$E = e\left\{1 - \frac{b}{2a}\left[(1 - e)T_d + T_b\right]\right\}$$

where e is the air-cycle efficiency and T_d and T_b are the absolute temperatures at the end of explosion and beginning of compression respectively. The constants a and b have values of about 0.194 and 0.051×10^{-3} for the average working mixture. For the conditions customarily met in airplane engines the ratio of the two efficiencies $\frac{E}{e}$ is about 0.80; in other words, the theoretical efficiency with the actual working substance is only 80 per cent of that which would be attainable if these substances were perfect gases.

The actual working substance consists almost exclusively of nitrogen, water vapor and carbon dioxide. All three of these substances show a considerable increase in specific heat with rise

[1] WIMPERIS, *The Internal Combustion Engine*, p. 85.

in temperature and the last two dissociate at high temperatures, especially at low pressures. The following tables[1] give mean specific heats at constant volume, and percentage dissociation.

Mean Specific Heats at Constant Volume (in B.t.u. per degree Fahrenheit) between 200°F. and the Stated Temperature

Temperature, degrees Fahrenheit	930	1,830	2,730	3,630	4,530	5,430
Nitrogen	0.185	0.188	0.196	0.205	0.214	0.225
Water vapor	0.350	0.385	0.425	0.468	0.540	0.623
Carbon dioxide	0.187	0.217	0.229	0.238	0.247	0.249

Dissociation, Per Cent

Temperature, degrees Fahrenheit	Pressure in atmospheres			
	0.1	1.0	10	100
	H_2O			
2,730	0.043	0.02	0.009	0.004
3,630	1.25	0.58	0.27	0.125
4,530	8.84	4.21	1.98	0.927
5,430	28.4	14.4	7.04	3.33
	CO_2			
2,730	0.104	0.048	0.0224	0.01
3,630	4.35	2.05	0.96	0.445
4,530	33.5	17.6	8.63	4.09
5,430	77.1	54.8	32.2	16.9

Calculation of the theoretical efficiency, taking into account both the variable specific heats and dissociation, shows that this

[1] Tizard and Pye, *The Automobile Engineer*, Feb., 1921.

efficiency, with a mixture giving maximum efficiency, is represented very closely by the equation

$$E = 1 - \left(\frac{1}{r}\right)^{0.295} \tag{2}$$

The heat loss to the walls reduces the actual efficiency below the theoretical values. With the very best design of cylinder and optimum operating conditions the highest attainable indicated thermal efficiency is given fairly accurately by the equation

$$E = 1 - \left(\frac{1}{r}\right)^{0.25} \tag{3}$$

A comparison of these efficiency values is given in the following table. There are added the best results obtained by Ricardo[1] on a special engine in which every known refinement was employed with a view to raising the thermal efficiency.

CYCLE AND ENGINE EFFICIENCIES

Compression ratio r	Efficiency			
	Air cycle	From equation (2)	From equation (3)	Ricardo's observed values
4.0	0.426	0.336	0.296	0.277
4.5	0.452	0.359	0.314	0.297
5.0	0.475	0.378	0.332	0.316
5.5	0.494	0.396	0.348	0.332
6.0	0.512	0.411	0.361	0.346
6.5	0.527	0.424	0.375	0.360
7.0	0.540	0.437	0.386	0.372
7.5	0.553	0.449	0.396	0.383
8.0	0.565	0.460	0.406	

The difference between the air-cycle efficiency (constant specific heat and no dissociation) and the theoretical efficiency of the cycle using imperfect gases, with the properties given in the preceding tables, diminishes as the explosion temperature diminishes. In the limiting case in which there is no fuel in the charge and consequently no rise of temperature at explosion the two efficiencies become equal. The less the fuel in the charge, or,

[1] *Proc. Royal Aeronautical Society,* 1920.

the weaker the mixture, the more nearly does the cycle efficiency approach the air-cycle efficiency.

Calculations by Tizard and Pye show the cycle efficiency to vary with the mixture strength as in Fig. 13. The curve for correct mixture shows the efficiency when the air-fuel ratio is chemically correct; the equation to the curve is $E = 1 - \left(\frac{1}{r}\right)^{0.258}$

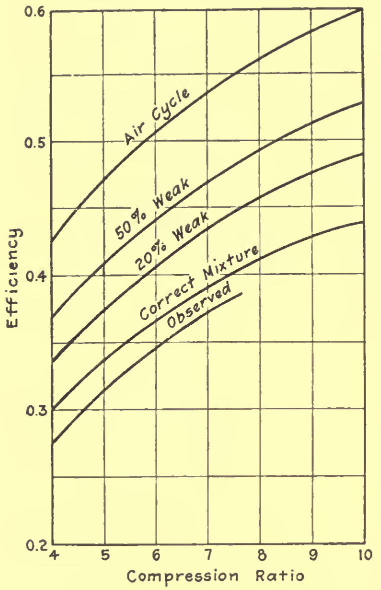


FIG. 13.—Calculated and observed thermal efficiencies with various strengths of mixture and compression ratios.

The 20 per cent weak curve is calculated for 20 per cent excess of air which is usually about the limit of explodibility; the improved cycle efficiency in this case is verified by engine tests which generally show maximum indicated thermal efficiency with about 20 per cent excess of air. The 50 per cent weak curve represents a condition which cannot be attained in the normal Otto-cycle engine as it gives a non-explosive mixture; it can be realized by an injection of the fuel into the compressed air as in the Diesel cycle, or by having a stratified charge in the cylinder with an explosive mixture surrounding the igniter. In any case the realization of the higher efficiency of the weak mixture will be attended by reduced engine capacity.

Another point of importance is brought out by the curves of Fig. 13. The ratio of cycle efficiency to air-cycle efficiency increases with the ratio of compression; that is, we may expect to realize a larger percentage of the air-cycle efficiency as the compression ratio increases. The ratio of the efficiencies for a compression ratio of 4 is 0.685; with a compression ratio of 10 it is 0.735. This improvement is shown also in actual engines. The ratio of observed efficiency (Fig. 13) to the air-cycle efficiency rises from 0.65 at a ratio of compression of 4 to 0.685 at a ratio of compression of 7.

III. The theoretical indicator diagram is not realized for still another reason. The admission and exhaust of the charge are attended by frictional resistance to the passage of the gas through the carburetor, inlet manifold, inlet valve, and exhaust valve. Moreover, as the flow of the gas is at high velocity, there must be a pressure drop to bring about this flow; with an inlet velocity of 250 ft. per second, this would amount to about 0.6 lb. per square inch. The frictional resistance and velocity head cause a lowering of the admission pressure, *a*, a raising of the exhaust pressure, and the forming of the "loop" (Fig. 11) at the bottom of the indicator diagram. This loop represents the negative **pumping** or **fluid friction work** which the engine has to perform. Engine tests indicate that the pumping work increases rather more rapidly than the square of the engine speed; the actual amount of the work depends on the dimensions and arrangement of the engine. At 1,000 r.p.m. it will probably average about 4 per cent of the indicated work of an aviation engine. This means in an engine with 120 lb. per square inch. brake m.e.p. that the mean height of the loop is 5 lb. per square inch at 1,000 r.p.m. The pumping loss is a function of the gas velocity in the manifolds and through the valve ports. Its magnitude will vary from about 2 lb. per square inch with a gas velocity of 100 ft. per second to 8 lb. per square inch with a gas velocity of 200 ft. per second. The indicated work may properly be considered as being only the positive loop of the indicator card; the suction-exhaust loop is one of the engine friction losses.

Minor factors affecting the area of the indicator card are the rounding of the "toe" of the diagram which results from the opening of the exhaust valve before the end of the expansion stroke in order to facilitate exhaust, and the departure of the expansion and compression curves from the theoretical adiabatic curves.

IV. It is not usually practicable to determine directly the work done by the working substance in the engine cylinder or the **indicated work.** In all tests the power measured is the **useful work** or **brake horse power** which is what remains of the indicated work after some of it has been used up in overcoming the friction of the engine and in driving the water and oil pumps.

$$\text{Indicated work} - \text{friction work} = \text{useful work}$$

$$\frac{\text{Useful work}}{\text{Indicated work}} = \text{Mechanical efficiency}$$

Friction Losses.—The mechanical losses in an engine may be divided into two groups:

1. The losses due to bearing friction and the driving of such auxiliaries as valve gears, oil and water pumps, magnetos, etc.
2. Piston friction.

Tests by Ricardo show that to overcome the first group a mean effective pressure of from 1.5 to 3 lb. per square inch is usually required—the lowest figure applying to a large multi-cylinder engine. The distribution of these losses is about as follows:

Bearings	0.75 to 1.00 lb. per square inch
Valve gear	0.75 to 0.80 lb. per square inch
Magnetos	0.05 to 0.01 lb. per square inch
Oil pumps	0.15 to 0.25 lb. per square inch
Water pump	0.30 to 0.50 lb. per square inch
Total	2.00 to 2.65 lb. per square inch

Piston friction is the largest item of loss; its magnitude probably results from the fact that the motion is reciprocating and

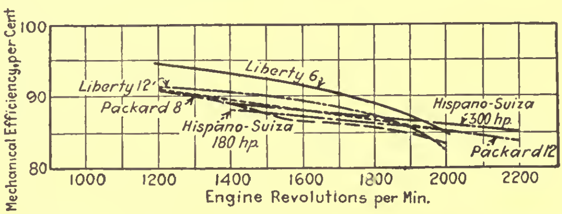


FIG. 14.—Mechanical efficiencies of airplane engines.

that the film of oil on the walls is more or less carbonized by the high temperatures and consequently has a high viscosity. The magnitude is probably about 7 lb. per square inch of piston area.

The total loss from bearing friction, piston friction and fluid friction in the best ungeared engines is from about 10.5 to 14 lb. per square inch of piston area, the lower figure referring to radial air-cooled engines and the higher to water-cooled engines. Taking 120 lb. per square inch as the brake m.e.p., these values correspond to mechanical efficiencies of 92.0 and 89.5. Tests (Fig. 14) indicate that friction work increases more rapidly than the engine speed but not so rapidly as the square of the speed.

Taking into account the losses enumerated, it is possible to arrive at a fair approximation to the actual efficiency of an airplane engine. Consider for example a high-grade engine with a compression ratio of 5.5, using gasoline as fuel. For every 100 B.t.u. (lower heat value) of heat of combustion we may expect a heat suppression (Item I.*a*) of 4 B.t.u., leaving 96 B.t.u. developed. Of this quantity, 13 B.t.u. will go to the walls by radiation and conduction (Item I.*b*f)eore it ca n be utilized, leaving 83 B.t.u. The theoretical efficiency for a compression ratio of 5.5 is 0.396 (equation (2) p. 21). The theoretical work of the cycle is $0.396 \times 83 = 32.9$ B.t.u. The actual indicated work is thus 32.9 per cent of the heat of perfect combustion of the fuel and this quantity is usually spoken of as the **indicated thermal efficiency,** or the **thermodynamic efficiency,** E_t, of the engine. It measures the efficiency of the engine in converting heat into work.

As previously stated, this indicated work is not readily measurable. The useful or brake work may be taken as 85 per cent (Item IV) of the indicated work, or in this case, $0.85 \times 32.9 = 28.0$ B.t.u. The thermal efficiency referred to b.h.p. is then 28.0 per cent. This quantity may be compared with the results of tests on a high-grade engine. Such tests may be expected to show the consumption of about 0.50 lb. of gasoline per brake horse power hour. The fuel has a lower heat value of almost 18,500 B.t.u. per pound. The thermal efficiency referred to b.h.p. is $\dfrac{\text{Work of 1 b.h.p. hour, B.t.u.}}{\text{Heat of combustion of the fuel, B.t.u.}} = \dfrac{2{,}545}{0.50 \times 18{,}500}$ $= 0.275$, which agrees very closely with the calculated efficiency. A reduction in any of the itemized losses will increase the final efficiency.

Mean Effective Pressures.—The mean effective pressure (m.e.p.) of a gas engine is that gas pressure on the piston which, if maintained constant for one stroke of the engine, would do as much work as is actually done in the two revolutions of the cycle.

In aviation engines the m.e.p. is practically always obtained from the brake horse power and is called the brake m.e.p. It is given by the equation

$$\text{brake m.e.p.} = \frac{b.h.p. \times 33{,}000}{\frac{\pi}{4}d^2 \times \frac{s}{12} \times \frac{N}{2} \times n} = 1{,}083{,}000 \frac{b.h.p.}{d^2 \times s \times N \times n}$$

where d is the cylinder diameter in inches, s is the stroke in inches, N is the revolutions per minute, and n is the number of cylinders.

The brake mean effective pressures usually given are computed from the b.h.p.; values range from 70 to 135 lb. per square inch. The true m.e.p. in the cylinder is this value divided by the mechanical efficiency, E_m.

Torque and Power.—If p = actual brake m.e.p., the average useful force exerted in the cylinders of a four-cycle engine is $\frac{\pi}{4}nd^2 \times \frac{p}{4} = 0.1964\ pnd^2$ lb. This force is maintained while the piston moves during each revolution $2s$ in., or $s \div 6$ ft. and the work done in foot-pounds is $0.1964\ pnd^2 \times \frac{s}{6} = 0.03273\ pnd^2s$. *Torque* is the average turning moment and is numerically equal to the force continuously exerted at the propeller at 1 ft. radius. This is exerted, during each revolution, over a distance of $2\pi = 6.2832$ ft. The work being equal to that already computed, the torque in pound-feet is $Q = 0.03273\ pnd^2s \div 6.2832 = pnd^2s \div 192$.

For $s = 7$, $d = 5$, $n = 12$, $p = 120$; $Q = 1{,}315$ lb-ft. The actual torque varies, but has this *average* value.

If S = piston speed, feet per minute $= \frac{2sN}{12}$, the b.h.p. is $H_B = \frac{\pi}{4}d^2\frac{p}{4}Sn \div 33{,}000 = pd^2Sn \div 168{,}000 = Q \times N \div 5{,}250$. Thus for 1,600 r.p.m., in the preceding example, $H_B = (1{,}315 \times 1{,}600) \div 5{,}250 = 401$.

Capacity and Volumetric Efficiency.—The weight of fuel mixture taken into the ideal engine (Fig. 10) is given by the gas equation

$$w_a = \frac{144p\ (v_b - v_a)}{R\ T_m}$$

where $v_b - v_a = \frac{\pi}{4}d^2s$ is the volume of the mixture admitted and T_m is the absolute temperature of the external air.

Actual engines do not draw in weights equal to that expressed by the above equation. The weight of mixture actually admitted is the difference between the weight present at the points b and a, Fig. 10. The weight present at any point is $w = \frac{144pv}{RT}$: therefore the weight admitted is $\frac{144p_bv_b}{RT_b} - \frac{144p_av_a}{RT_a}$. The ratio of the weight actually admitted to that which would be admitted to the ideal engine is called the **volumetric efficiency,** E_v.

$$E_v = \frac{w}{w_a} = \left(144\ \frac{p_b v_b}{R\ T_b} - 144\ \frac{p_a v_a}{R\ T_a}\right) \div \frac{144 \times 14.7\ (v_b - v_a)}{R\ T_m}$$

Writing $\frac{v_b}{v_a} = r$, the above reduces to

$$E_v = \frac{T_m}{14.7(r - 1)}\left(\frac{p_b r}{T_b} - \frac{p_a}{T_a}\right)$$

Taking $t_m = 100$, $r = 5$, $p_b = 12$, $t_b = 200$, $p_a = 16$, $t_a = 900$ we find $E_v = 0.76$.

The volumetric efficiency is determined mainly by two factors, the temperature, T_b, and the pressure, p_b, at the end of admission.

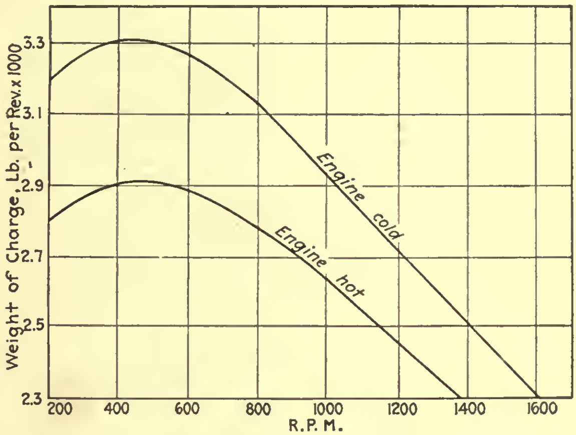


FIG. 15.—Volumetric efficiencies of hot and cold engines.

The temperature of the mixture rises during admission as a result of the addition of heat from the hot interior surfaces of the cylinder. Comparative tests of the volumetric capacity of an engine (1) when being motored over cold, and (2) in ordinary operation, show that the heating effect decreases slightly as the speed of the engine increases in consequence of the shorter time available for the transmission of heat. The pressure drop, however, increases continuously with increase of engine speed. Figure 15[1] gives curves of weight of charge taken into the cylinder per revolution for an engine of high valve resistance. The effect of the temperature rise in reducing the volumetric efficiency is from 12 to 15 per cent and would be appreciably greater but for the evaporation of the fuel, which, through the abstraction of the latent heat of evap-

[1] JUDGE, *High-speed Internal-combustion Engine*, p. 161.

oration, reduces the rise of temperature of the charge by 30 to 40°F. The variation of volumetric efficiency of the Liberty-12 with engine speed is shown in Fig. 16. In the same figure there is also shown the volumetric efficiency of the Hispano-Suiza 300 engine, but in this case the volumetric efficiency is given as the ratio of the weight of air actually admitted to the weight of a volume of air equal to piston displacement and of the density of the air in the inlet manifold. Measured in this way the volumetric efficiency is 95 per cent at 1,600 r.p.m.; this corresponds to 93 per cent when compared with air at room density. The broken line shows the volumetric efficiencies compared with air at room density.

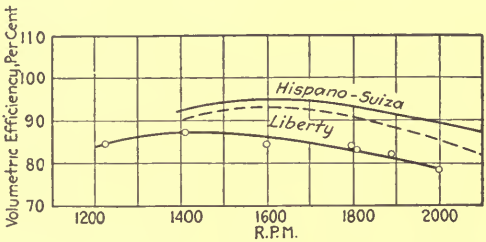


FIG. 16.—Volumetric efficiencies of airplane engines.

The **pressure drop during admission** is much more variable in different engines than is the temperature rise. Its magnitude depends on the pressure drop through the carburetor and the size and arrangement of the manifolds and the inlet valve. In every case it will increase rapidly with increased engine speed; its actual magnitude will usually be small for low speeds.

The pressure drops in the manifolds of two engines are given in Fig. 17. It will be seen that with wide-open throttle the pressure drop is nearly proportional to the engine speed. With an engine loaded with a propeller, change of speed is obtained only by varying the opening of the throttle valve; the manifold vacuum increases rapidly as the throttle valve is closed. The pressure in the cylinder is considerably less than that in the manifold because of the valve resistances.

The volumetric efficiency in engines of good design will be from 80 to 85 per cent. With low speeds and other exceptionally favorable conditions, values as high as 90 to 92 per cent have been recorded.

The maximum possible volume of charge admitted per cycle in the ordinary engine is the volume enclosed at the instant of valve closure less the clearance volume. The admission valve always closes past the dead center. If the closing angle is 45 deg. late the piston will have returned about 12 per cent of its stroke and the maximum possible volumetric efficiency will be 88 per cent. Occasionally the operating conditions and induction pipe length may be such as to give more than atmospheric pressure in the cylinder at the instant of closure which would result in increased volumetric efficiency.

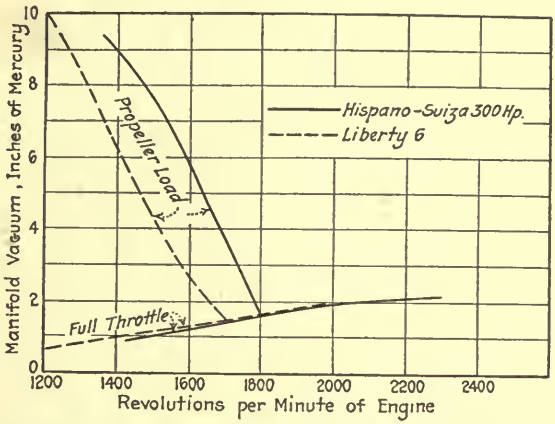


FIG. 17.—Intake manifold depressions with full throttle and with propeller load.

It should be noted that the *capacity* (as affected by volumetric efficiency) and *thermal efficiency* of an engine are not necessarily related to one another. The diminution in capacity of an engine resulting from heating of the entering charge, from a high carburetor or inlet-valve resistance, or from diminishing speed, may or may not result in a change in efficiency, and the change, if it takes place, may be either an increase or a decrease. A given engine may at one time be developing 200 h.p.; at another time 250 h.p.; the efficiency may, however, be the same in both cases, although it tends to be lower for the lower h.p. because of the approximate constancy of engine friction, which makes the efficiency referred to the b.h.p. less at light loads.

Units of Capacity.—In determining the size of a projected engine, or in comparing the performance of existing engines, it is desirable to have some standard unit for measuring the specific capacity. The most common unit is the piston displacement in cubic inches per brake horse power, or the brake horse power developed per cubic foot of piston displacement. The piston displacement is the displacement per stroke of one cylinder multiplied by the number of cylinders. As the horse power varies almost directly as the engine speed, the above units do not really lead to a satisfactory comparison of engines operating at different speeds. For this purpose it is better to state the capacity at 1,000 r.p.m., deducing this capacity from the actual performance by the use of the assumption that horse power is proportional to engine speed. For example, the Liberty-12 engine, 5 by 7 in., develops 400 h.p. at 1,700 r.p.m. The piston displacement per cylinder is $\frac{\pi}{4} \times 5^2 \times 7 = 137.4$ cu. in. per stroke; the total piston displacement is $12 \times 137.4 = 1{,}648.8$ cu. in.; the piston displacement per brake horse power is $1{,}648.8 \div 400 = 4.12$ cu. in. The brake horse power per cubic foot of piston displacement is $12^3 \div 4.12 = 420$ h.p.; the piston displacement per brake horse power at 1,000 r.p.m. is $4.12 \times \frac{1{,}700}{1{,}000} = 7.0$ cu. in.; the b.h.p. per cubic foot of piston displacement at 1,000 r.p.m. is $420 \times \frac{1{,}000}{1{,}700} = 247$.

The Hispano-Suiza engine, with 718.9 cu. in. displacement, develops 150 h.p. at 1,450 r.p.m. This corresponds to 4.8 cu. in. per horse power; or 6.96 cu. in. per horse power at 1,000 r.p.m. This last figure shows that the Hispano-Suiza and Liberty engines have practically the same capacities per cubic inch of piston displacement per minute.

The fixed-cylinder radial air-cooled A B C Dragonfly nine-cylinder engine, with 1,389.3 cu. in. displacement, develops 310 h.p. at 1,650 r.p.m. This corresponds to 4.48 cu. in. per horse power, or 7.38 cu. in. per horse power at 1,000 r.p.m.

The lower specific capacity of rotating-cylinder engines is illustrated by the nine-cylinder Gnome, with a piston displacement of 770 cu. in., which develops 104 b.h.p. at 1,200 r.p.m. This corresponds to 7.41 cu. in. per horse power, or 8.89 cu. in. per horse power at 1,000 r.p.m.

The above figures may be regarded as characteristic of the different types.

Tests of Performance.—The results obtained on the test of an engine will vary greatly with a number of factors such as the air pressure and temperature, kind of fuel, type and dimensions of carburetor, temperature of jacket water and of lubricating oil, and condition of engine. For example the Liberty 12 has shown a brake horse power at 1700 r.p.m. which varies from 380 to 480 b.h.p.

The following tests are reported under sea-level conditions:

Engine	Revolutions per minute	Brake m.e.p., lb. per sq. in.	Brake horse power	Friction horse power	Mechanical efficiency
Hispano-Suiza-180........	1,200	114.0	121.8	16.8	0.88
	1,500	119.1	159.0	23.5	0.87
	1,700	117.7	178.0	28.4	0.86
	1,900	111.0	187.0	34.2	0.84
Liberty-12..............	1,200	118.0	295.0	27.6	0.91
	1,400	119.5	348.0	38.3	0.90
	1,600	119.5	398.0	49.1	0.89
	1,800	117.6	442.0	65.4	0.87
	2,000	104.0	433.0	88.0	0.83

A plotting of test results on the Liberty 12 is shown in Fig. 18.

These figures illustrate the usual laws of performance. The

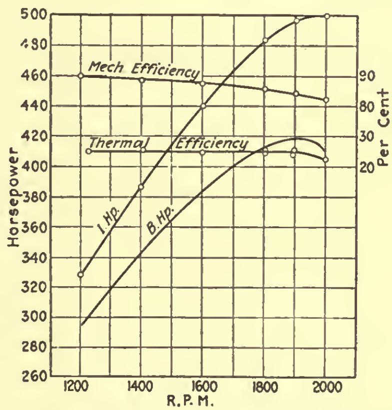


FIG. 18.—Performance curves of Liberty-12 engine.

mean effective pressure, p, and consequently the torque, reach maximum values at some moderate speed. The power increases with increasing speed, but at a rate which diminishes after the maximum value of p has been reached. If p were constant the power would vary directly with the speed. Cylinder cooling reduces p at low speeds; high resistance through ports and passages reduces volumetric efficiency and p at high speeds (Fig. 16). Maximum power is reached when the rate of decrease of p with engine speed is equal to the rate of increase of engine speed.

Figure 79 gives results of trials on a 230 h.p., six-cylinder Benz engine. Here the failure of the power to increase proportionately to speed is clearly shown. Maximum mean effective pressure occurs at 1,050 r.p.m. and maximum power at 1,650 r.p.m. The fuel consumption rate is also shown. The economy is practically constant over the speed range 900 to 1,200 r.p.m.

It will be observed that throttling the engine increases the fuel consumption per horse power. The full-line curves show the performance when the throttle is wide open and the engine is loaded until it assumes the desired speed. The broken lines show the performance when the engine had the propeller load only; in this case the engine speed can be reduced below 1,400 r.p.m. only by partly closing the throttle; speeds above 1,400 r.p.m. are not possible with the propeller load.

Figure 143 is for a rotary-cylinder engine. In this case the effective horse power is the indicated horse power minus the engine friction loss and minus the windage loss; the rapid increase of windage loss when the engine speed increases makes the net horse power a maximum at about 1,250 r.p.m.

Correction to Standard Atmospheric Conditions.—The published results of engine tests may give either the actual horse powers observed or these horse powers corrected to some standard atmospheric condition. The latter is much preferable as it will permit an immediate comparison of engine performances. The best measure of capacity is the brake m.e.p. That engine which has the maximum m.e.p. is developing a horse power on the smallest piston displacement. But to make such direct comparison the operating conditions must be the same or else the results must be corrected to allow for differences. Such corrections can be readily applied to differences in atmospheric pressure and temperature. It has been commonly assumed that the horse power developed is proportional to the density of the atmosphere. The density is proportional to the atmospheric pressure, and inversely proportional to the absolute temperature. If the standard conditions are 14.7 lb. pressure (29.92 in. of mercury) and 32°F., and the observed horse power is P at pressure p and temperature t, the corrected horse power is

$$P_c = P \frac{14.7}{p} \frac{460 + t}{492}$$

In most of the published tests the correction to standard conditions has been made by use of this equation. Tests at the Bureau of Standards indicate, however, that the temperature correction in this equation is excessive and that more accurate results are obtained from the equation

$$P_c = P \frac{14.7}{p} \frac{920+t}{952}$$

The m.e.p. is corrected in exactly the same manner.

There is no method for directly comparing two engines which are using different fuels or mixtures of different strengths

Influence of Strength of Mixture on Capacity and Efficiency.—The effect of strength of mixture has been investigated by Berry[1] on automobile engines; his results are supported by the investigation of others on both automobile and aviation engines (see p. 260). For any constant engine speed and constant throttle opening, they show (Fig. 19) that the maximum power is obtained with a comparatively rich mixture, and that for maximum efficiency a weaker mixture must be used. As the throttle is closed the mixture for maximum efficiency (Fig. 21) becomes richer and at the lowest loads coincides with that for maximum power. The speed of the engine has no appreciable influence on the variation of engine power with strength of mixture (Fig. 22). Maximum power (Fig. 20) is obtained with 0.08 lb. of gasoline per pound of air, or 12½ lb. of air per pound of gasoline. Maximum efficiency is obtained with a mixture of 15 to 16 lb. of air per pound of gasoline at full load, but this

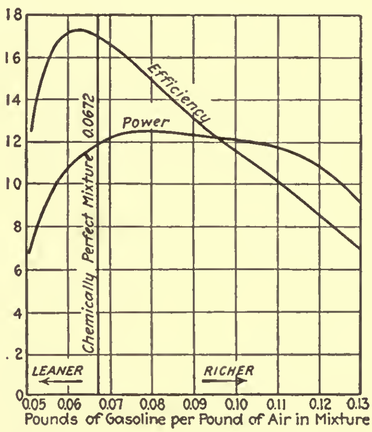


FIG. 19.—Variation of power and thermal efficiency with strength of mixture, at full throttle.

[1] *Trans. Am. Soc. Mech. Eng.*, 1919.

mixture must be made richer as the load diminishes and becomes 12½ lb. of air per pound of gasoline at lowest loads.

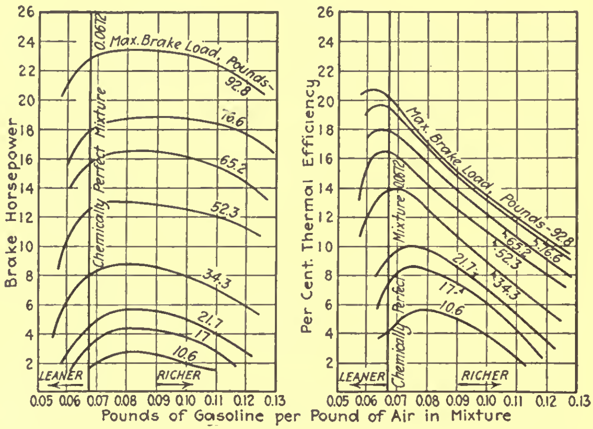
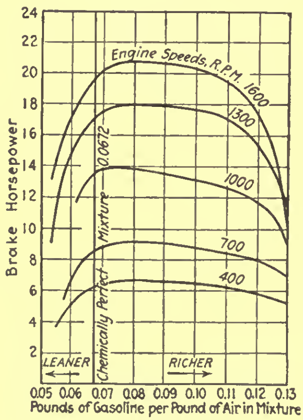


Fig. 20. Fig. 21.

Fig. 20.—Variation of engine power with strength of mixture at constant engine torque and varying speed.

Fig. 21.—Variation of thermal efficiency with strength of mixture at constant engine torque and varying speed.



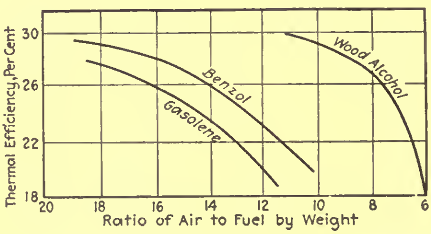


Fig. 22. Fig. 23.

Fig. 22.—Variation of engine power with strength of mixture at constant engine speed and varying torque.

Fig. 23.—Maximum thermal efficiencies of certain fuels with varying strength of mixture.

Tests by Watson[1] on an automobile engine with gasoline, benzol, and wood alcohol as fuel show (Fig. 23) the variation in

[1] *Proc. Inst. Aut. Eng.*, 1914.

efficiency of these fuels with strength of mixture. The comparison given by these curves is not, however, complete, since the same compression ratio was used for all three fuels. With alcohol it is possible to increase the compression pressure considerably without danger of preignition and without producing excessive explosive pressures; with benzol the compression ratio can similarly be increased slightly. The efficiency with alcohol could probably be raised to at least 35 per cent by the use of a higher compression ratio.

Influence of Air Temperature on Capacity and Efficiency.—As already pointed out in the discussion of volumetric efficiency (p. 27) the temperature of the air admitted to the cylinder has

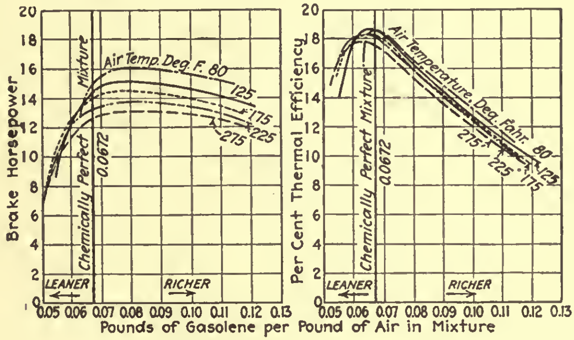


FIG. 24. FIG. 25.

FIG. 24.—Variation of engine power with strength of mixture at various air temperatures and full throttle.

FIG. 25.—Variation of thermal efficiency with strength of mixture at various air temperatures and full throttle.

a considerable influence on the power developed. The tests at the Bureau of Standards show the diminution in power with increase in temperature to be proportional to half the increase in absolute temperatures, for the range of temperature from 4 to 120°F. For example, if the absolute temperature of the air increases from 500 to 580°, or 16 per cent, the power of the engine will decrease 8 per cent. Berry's tests on automobile engines (Figs. 24 and 25) show that this law does not hold for a higher temperature range. The decrease in power is roughly proportional to one-third the increase in temperature (Fig. 24). The efficiency (Fig. 25) is also seen to diminish with increase of air temperature but through a much smaller range.

Tests by Berry with air at a temperature lower than 80°F.

showed a rapid falling off in capacity and efficiency. These tests, however, were carried out with a commercial gasoline of low volatility as compared with the gasolines specified for airplane engines. In all cases maximum power is obtained with the lowest air temperature which will permit satisfactory distribution and vaporization of the fuel. This temperature depends not only on the volatility of the fuel but also upon the manifold design. A "hot spot" between the carburetor and manifold (heated by the exhaust gases), on which the liquid spray from the carburetor impinges, causes vaporization of part of the fuel without heating up the air appreciably and is found to result in a better distribution of the mixture to the different cylinders and in improved engine operation when the air supply is cold. With this device it is possible to lower the temperature range of the entering air somewhat without a falling off in capacity or efficiency.

Influence of Throttling on Efficiency.—It is generally found that thermal efficiency tends to increase as the power is cut down

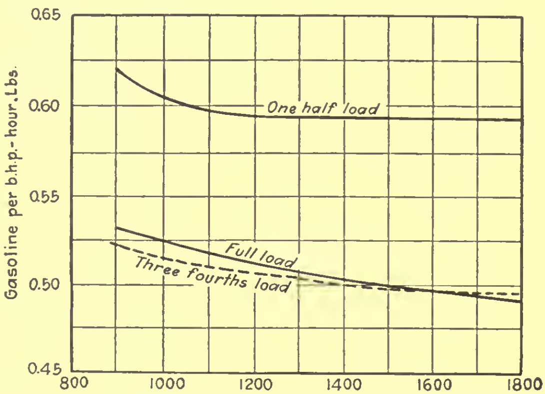


FIG. 26.—Variation of specific fuel consumption with engine speed at various throttle positions.

from maximum to three-fourths load. This phenomenon is probably due mainly to improvement in the mixture at partial load. The carburetor is set to give maximum power for full throttle, which, as just shown, is obtained with an over-rich and consequently inefficient mixture. If the mixture becomes leaner at partial throttle, the economy will improve. What actually happens will depend primarily on the characteristics of the carburetor used.

In Fig. 26 are given the fuel consumptions per brake-horse-power hour of a Liberty 12, at full, three-fourths and one-half loads. It will be seen that the efficiencies at full and three-fourths loads are substantially the same, but that there is a marked falling off at half load. The efficiency is again seen to increase with engine speed.

Influence of Compression Ratio on Capacity.—Tests of a 150-h.p. Hispano-Suiza engine at 1,500 r.p.m. with various compression ratios show the maximum attainable brake horse power to have been as follows:

Ratio of compression.........	4.7	5.3	6.2
Maximum brake horse power..	160.0	165.0	169.0

The percentage increase of power with increase of compression ratio is about the same as in the tests of a Liberty 12 engine, which at 1,600 r.p.m. gives the following results:

Ratio of compression....................	4.9	5.5
Maximum brake horse power.............	380.0	398.0

The Influence of Revolutions per Minute on Capacity is shown in all performance curves (see Figs. 48, 50, 53, etc.). The fall in

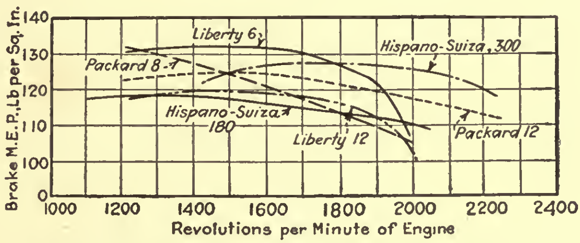


FIG. 27.—Variation of mean effective pressure with engine speed for airplane engines.

brake m.e.p. (Fig. 27) causes the b.h.p. to go through a maximum; the efficiency practically always increases with speed, but becomes a maximum before the engine reaches maximum horse power.

Influence of Jacket-water Temperature on Capacity.—Figure 28 shows the effect on the capacity of a Liberty 12 engine of varying the temperature of the jacket water. The amount of water circulated was constant at any given speed but the inlet temperature was varied, thereby giving the series of outlet temperatures indicated. It will be seen that the power increases as the cooling-water temperature decreases to about 100°F. At 200°F. and 1,800 r.p.m. the power is only 417 h.p. while at 90° it is 436 h.p.

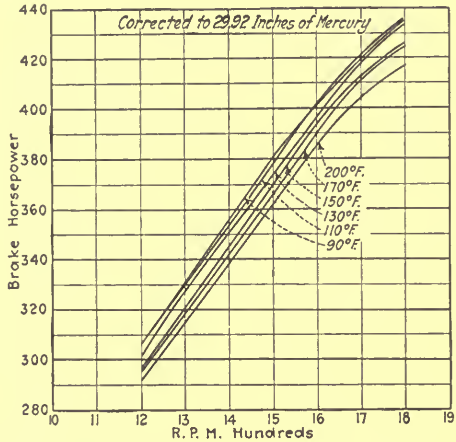


Fig. 28.—Variation of engine power with speed for various jacket-water discharge temperatures.

Various tests have demonstrated that the friction horse power decreases with increase of jacket-water temperature, so that the above increase of brake horse power with lower jacket-water temperature must have resulted from a still greater increase in in-

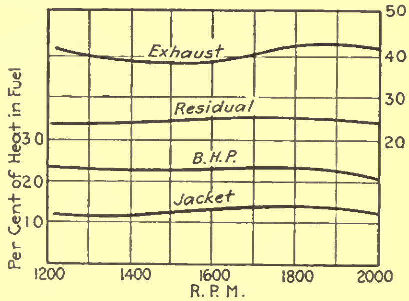


Fig. 29.—Typical heat balance of airplane engine.

dicated horse power. The increase is primarily due to improvement in volumetric efficiency resulting from less heating up of the charge as it enters. This phenomenon is shown in Fig. 15.

Heat Balance.—Of the total heat of combustion of the fuel admitted to the engine cylinder, part is converted into brake horse power, part goes to the water jacket, part escapes as heat in the exhaust gases, and the rest is lost in various ways such as incomplete combustion and radiation and conduction from the engine. A typical heat balance for the Liberty 12 is shown in Fig. 29 in which the percentages are in terms of the higher heat value of the fuel. It will be observed that the heat distribution does not change notably with engine speed.

CHAPTER III

ENGINE DYNAMICS

Turning Moment.—The pressure on any single piston of a four-cycle engine is varying continuously throughout the cycle.

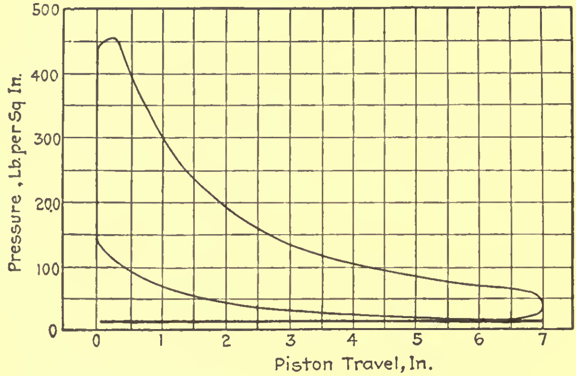


Fig. 30.—Indicator card of the Liberty-12 engine.

If the indicator card is as in Fig. 30, the resulting total gas pressure on the piston of a 5-in. diameter, 7-in. stroke engine for various

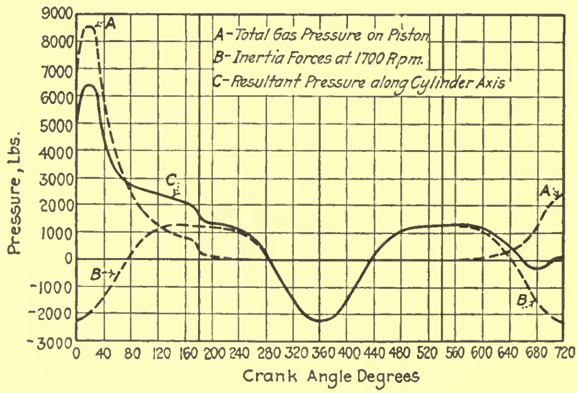


Fig. 31.—Forces acting in the Liberty-12 engine.

successive crank positions is represented by the curve A, Fig. 31. The pressure transmitted to the crankpin is modified, however, by the inertia of the reciprocating masses of the piston and connecting rod. During the first part of each stroke these masses are being accelerated; during the second part they are retarded. Hence the net useful force acting on the crankpin in a direction parallel to the cylinder axis is alternately less or greater than that shown by the curve A.

Let W = Weight of reciprocating parts, pounds (complete piston and half of rod).
n = Revolutions per minute.
a = Angle turned through by crank, starting from its uppermost position, degrees.
r = Crank radius, feet (half the stroke of the engine).
l = Length of connecting rod (center to center of pins), feet.

Then the **accelerating force** at any moment, in pounds, is $P_a = 0.00034\ Wn^2r(\cos a \pm \frac{r}{l}\cos 2a)$, approximately, the + sign being used for the down stroke and the − sign for the up stroke. The quantity $(\cos a \pm \frac{r}{l}\cos 2a)$ is an approximation; a more correct expression is $\cos a \pm \dfrac{\frac{r^2}{l^2}\cos 2a + \sin^4 a}{\left(\frac{l^2}{r^2} - \sin^2 a\right)^{3/2}}$. Values of this quantity for the range of ratios of l to r common in airplane engines are given in Table 1. The minus sign indicates negative acceleration from 0 deg. to 180 deg., and positive acceleration from 180 deg. to 360 deg. Calculations made for the Liberty-12 engine give the results shown in Fig. 31. The indicator card (Fig. 30) is plotted for 18 per cent clearance and a brake m.e.p. of 123 lb. per square inch, the exponents of the compression and expansion curves being taken as 1.32. The cylinder is 5 by 7 in., the connecting rod 12 in. long and the weights of reciprocating parts are: piston complete with pin, 4.838 lb.; upper half of connecting rod, 1.225 lb.; total, 6.063 lb. The engine is assumed to make 1,700 r.p.m. The inertia forces, P_a, calculated from the preceding equation, are plotted as curve B, Fig. 31. The algebraic sum of gas pressures A and inertia pressures B is shown by the resultant pressure curve C.

TABLE 1.—INERTIA FACTORS

$$\cos a \pm \frac{\frac{l^2}{r^2}\cos 2a + \sin^4 a}{\left(\frac{l^2}{r^2} - \sin^2 a\right)^{3/2}}$$

Crank angle, degrees	$\frac{l}{r} = 4$	$\frac{l}{r} = 3.75$	$\frac{l}{r} = 3.5$	$\frac{l}{r} = 3.25$	$\frac{l}{r} = 3.0$	Crank angle, degrees
0	1.2500	1.2667	1.2857	1.3077	1.3333	360
5	1.2426	1.2590	1.2778	1.2995	1.3249	355
10	1.2204	1.2362	1.2543	1.2752	1.2997	350
15	1.1839	1.1986	1.2155	1.2351	1.2580	345
20	1.1335	1.1468	1.1621	1.1798	1.2006	340
25	1.0702	1.0817	1.0948	1.1102	1.1283	335
30	0.9950	1.0042	1.0149	1.0274	1.0423	330
35	0.9091	0.9158	0.9236	0.9328	0.9440	325
40	0.8140	0.8179	0.8225	0.8281	0.8349	320
45	0.7112	0.7121	0.7133	0.7149	0.7172	315
50	0.6026	0.6004	0.5980	0.5955	0.5929	310
55	0.4899	0.4846	0.4787	0.4719	0.4643	305
60	0.3751	0.3668	0.3573	0.3465	0.3338	300
65	0.2601	0.2490	0.2363	0.2215	0.2041	295
70	0.1468	0.1332	0.1175	0.0992	0.0776	290
75	0.0368	0.0211	0.0030	−0.0182	−0.0434	285
80	−0.0682	−0.0854	−0.1055	−0.1288	−0.1567	280
85	−0.1669	−0.1851	−0.2062	−0.2309	−0.2605	275
90	−0.2582	−0.2767	−0.2981	−0.3234	−0.3536	270
95	−0.3412	−0.3594	−0.3805	−0.4052	−0.4348	265
100	−0.4155	−0.4327	−0.4528	−0.4761	−0.5040	260
105	−0.4809	−0.4965	−0.5146	−0.5358	−0.5610	255
110	−0.5373	−0.5509	−0.5665	−0.5848	−0.6064	250
115	−0.5851	−0.5962	−0.6090	−0.6237	−0.6411	245
120	−0.6249	−0.6332	−0.6427	−0.6535	−0.6662	240
125	−0.6573	−0.6625	−0.6685	−0.6752	−0.6829	235
130	−0.6830	−0.6852	−0.6875	−0.6901	−0.6927	230
135	−0.7030	−0.7021	−0.7009	−0.6993	−0.6970	225
140	−0.7181	−0.7142	−0.7096	−0.7040	−0.6972	220
145	−0.7292	−0.7225	−0.7137	−0.7055	−0.6944	215
150	−0.7370	−0.7279	−0.7172	−0.7047	−0.6898	210
155	−0.7423	−0.7310	−0.7178	−0.7025	−0.6843	205
160	−0.7459	−0.7326	−0.7173	−0.6997	−0.6788	200
165	−0.7480	−0.7333	−0.7163	−0.6968	−0.6738	195
170	−0.7492	−0.7334	−0.7153	−0.6944	−0.6700	190
175	−0.7498	−0.7334	−0.7146	−0.6929	−0.6675	185
180	−0.7500	−0.7333	−0.7143	−0.6923	−0.6667	180

The **resultant pressure,** P, curve C, acts along the axis of the cylinder. The force acting along the connecting rod, Fig. 32, is

$$P_E = P \div \cos b.$$

The component acting tangentially to the crankpin circle is

$$P_Q = P_E \sin (a + b) = P \sec b \sin (a + b)\cdot$$

Table 2 gives values of the tangential factor [sec b sin $(a + b)$].

The angles a and b are connected by the equation

$$\sin b = \frac{r}{l} \sin a.$$

Consequently,

$$P_Q = P \sin a \left(1 + \frac{r \cos a}{\sqrt{l^2 - r^2 \sin^2 a}}\right)$$

The **torque** or **turning moment** applied to the crank at any crank angle a is

$$T = P_Q r$$

Figure 33 shows the torque variation for a single cylinder of the Liberty 12 engine. Since the brake m.e.p. is 123 lb., the engine horse power per cylinder is

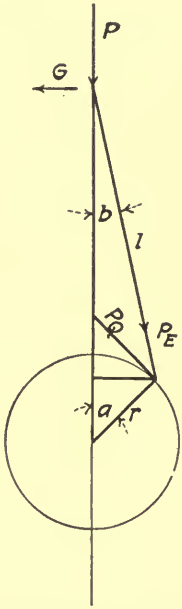


Fig. 32.—Diagram showing effect of obliquity of connecting rod.

$$\frac{123 \times \frac{7}{12} \times \left(\frac{\pi}{4} \times 5^2\right) \times 850}{33{,}000} = 36.3$$

The mean torque at the propeller must lead to the same result: $2\pi n T = 36.3 \times 33{,}000$, or, $T = 112$; that is, the mean torque per cylinder is 112 lb.-ft. The mean torque at the crankpin as determined from Fig. 33 is greater than this by the torque required to overcome the frictional resistance of one cylinder, or one-twelfth of the total frictional torque of the engine. If the mechanical efficiency of the engine is 85 per cent, the indicated mean effective pressure is $^{100}/_{85} \times 123$ lb. per square inch and the mean torque per cylinder is $^{100}/_{85} \times 112$ lb.-ft. The total horse power for the 12-cylinder engine is $12 \times 36.3 = 436$ and the mean total crankshaft torque is $12 \times 112 = 1{,}345$ lb.-ft.

TABLE 2.—TANGENTIAL FACTORS

$$\frac{\sin (a + b)}{\cos b}$$

Crank angle, degrees	$\frac{l}{r} = 4$	$\frac{l}{r} = 3.75$	$\frac{l}{r} = 3.5$	$\frac{l}{r} = 3.25$	$\frac{l}{r} = 3.$	Crank angle, degrees
0	0.0000	0.0000	0.0000	0.0000	0.0000	360
5	0.1089	0.1103	0.1119	0.1139	0.1161	355
10	0.2164	0.2193	0.2226	0.2264	0.2307	350
15	0.3214	0.3257	0.3305	0.3360	0.3425	345
20	0.4227	0.4281	0.4343	0.4415	0.4499	340
25	0.5189	0.5254	0.5329	0.5415	0.5515	335
30	0.6091	0.6165	0.6250	0.6314	0.6464	330
35	0.6923	0.7003	0.7098	0.7206	0.7333	325
40	0.7675	0.7761	0.7860	0.7974	0.8108	320
45	0.8340	0.8429	0.8529	0.8647	0.8786	315
50	0.8914	0.9001	0.9101	0.9219	0.9358	310
55	0.9391	0.9475	0.9572	0.9685	0.9819	305
60	0.9770	0.9847	0.9938	1.0041	1.0167	300
65	1.0046	1.0116	1.0195	1.0290	1.0401	295
70	1.0223	1.0282	1.0352	1.0430	1.0524	290
75	1.0303	1.0349	1.0401	1.0464	1.0539	285
80	1.0290	1.0320	1.0356	1.0399	1.0452	280
85	1.0186	1.0202	1.0221	1.0242	1.0268	275
90	1.0000	1.0000	1.0000	1.0000	1.0000	270
95	0.9739	0.9723	0.9703	0.9680	0.9656	265
100	0.9408	0.9376	0.9339	0.9296	0.9245	260
105	0.9016	0.8970	0.8916	0.8853	0.8780	255
110	0.8570	0.8511	0.8443	0.8364	0.8268	250
115	0.8082	0.8011	0.7930	0.7836	0.7723	245
120	0.7551	0.7473	0.7384	0.7278	0.7153	240
125	0.6989	0.6907	0.6811	0.6697	0.6563	235
130	0.6406	0.6320	0.6219	0.6102	0.5962	230
135	0.5801	0.5713	0.5613	0.5495	0.5356	225
140	0.5181	0.5094	0.4997	0.4882	0.4748	220
145	0.4549	0.4468	0.4375	0.4267	0.4140	215
150	0.3908	0.3835	0.3750	0.3652	0.3536	210
155	0.3263	0.3198	0.3124	0.3038	0.2936	205
160	0.2614	0.2559	0.2498	0.2425	0.2339	200
165	0.1962	0.1920	0.1872	0.1817	0.1751	195
170	0.1309	0.1280	0.1247	0.1209	0.1166	190
175	0.0654	0.0640	0.0624	0.0604	0.0582	185
180	0.0000	0.0000	0.0000	0.0000	0.0000	180

With the firing order used on the Liberty engine and with a Vee angle of 45 deg., the firing intervals between two cylinders on any one crank are 315 and 405 deg. of crank revolution. The turning moment on any one crank is obtained by superimposing

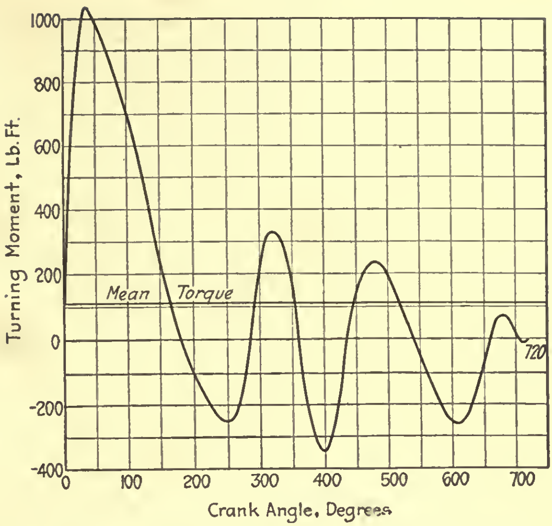


FIG. 33.—Turning moment for a single cylinder of the Liberty-12 engine.

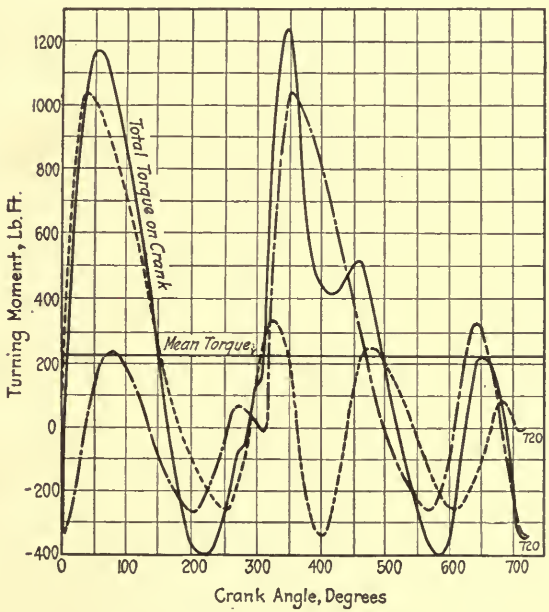


FIG. 34.—Turning moment on each crank of the Liberty-12 engine.

two torque curves like those of Fig. 33 with a phase lag of 315 deg. Adding the ordinates of two such curves, as in Fig. 34, gives the total torque on one crank.

The six cranks of this engine are spaced at angular intervals of 120 deg. The total torque on the crankshaft is obtained by superimposing six curves like the resultant curve of Fig. 34, with angular intervals of 120 deg., and taking the algebraic sums of ordinates at the various crank positions. This process gives the curve of Fig. 35. The torques and torque ratios are as follows:

	One cylinder	One crank	Whole engine
Maximum crankshaft torque, pound-feet	1,030	1,240	1,670
Ratio of maximum to mean torque	9.2	5.54	1.24

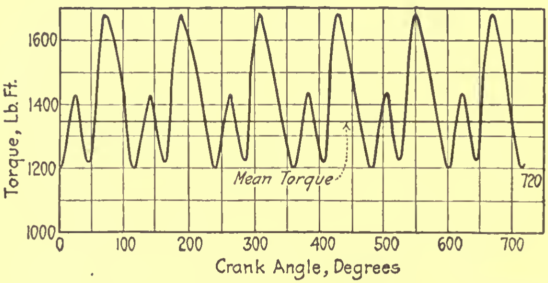


Fig. 35.—Torque at propeller end of the crankshaft of the Liberty-12 engine.

The ratio of maximum to mean torque varies with the angle of the Vee. For 5- by 7-in. cylinders at 120 lb. mean effective pressure, the following ratios hold for torques *on one crank:*

Vee angle	45 Deg.	60 Deg.	75 Deg.	90 Deg.
T_1	5.2	5.1	5.6	4.8
T_2	−2.7	−2.3	−1.7	−1.5
T_3	7.9	7.4	7.4	6.3

Here T_1 = maximum torque ÷ mean torque,
T_2 = minimum torque ÷ mean torque,
T_3 = range of torque ÷ mean torque.

For *the whole engine,* ratios are as follows:

	Vee angle	T_1	T_2
8-cylinder	90	1.40	0.66
	75	1.42	0.18
	60	1.70	−0.13
	45	2.14	−0.26
12-cylinder	60	1.13	0.86
	45	1.25	0.89

In both tables negative signs indicate reversal of direction of torque. The minimum torque variation is seen to occur with equal firing intervals (eight-cylinder, 90-deg., and 12-cylinder, 60-deg. Vee engines). The great increase in this variation as the Vee angle of the eight-cylinder engine is diminished is very marked.

A smooth curve of crankshaft turning moment, approximating as closely as possible to the mean torque line, is in every way desirable. This can best be obtained by the use of a plurality of cylinders with equal firing intervals. The greater the number of cylinders, the more uniform is the torque. The firing *order* of the cylinders is unimportant from this standpoint, as long as the firing *interval* is constant. The firing order is of the utmost importance, however, in relation to the balancing of forces and the stresses in the engine. Smoothness of running depends on the magnitude of the areas enclosed between the total torque curve and the mean torque line (Fig. 35). Areas above the line represent work done by the engine in excess of the resisting propeller torque and lead to acceleration; areas below the line represent a deficiency in engine work and a consequent slowing down. In ordinary engine practice a flywheel is used to absorb the excess and make up the deficiency without permitting excessive change in engine speed. In an airplane engine the propeller takes the place of a flywheel; its large radius of gyration enables it to absorb a considerable amount of excess work with only a very small increase in speed of rotation. Moreover, the resisting torque varies as the square of the angular velocity and hence increases notably with small increases in rotative speed. The influence of the number of cylinders on the variation of crankshaft torque is clearly shown in the following table.[1] The firing interval is constant in all cases.

TABLE OF TORQUE VARIATION

Number of cylinders	1	2	3	4	5	6	7	8	9	10	12	16	18
Ratio of maximum instantaneous torque to mean torque	7.70	5.20	2.74	2.94	1.64	1.17	1.45	1.40	1.22	1.12	1.13	1.06	1.03
Relative values of the maximum torque at propeller	1.0	1.35	1.07	1.53	1.06	0.91	1.32	1.45	1.43	1.46	1.76	2.20	2.41

The relative values of the maximum torque are also given in the table, the value for a single cylinder being taken as unity.

[1] From article by G. D. ANGLE, *Aviation*, Oct. 1, 1919.

It will be seen that the maximum torque at the propeller end of a 6-cylinder engine is less than the maximum torque exerted by a

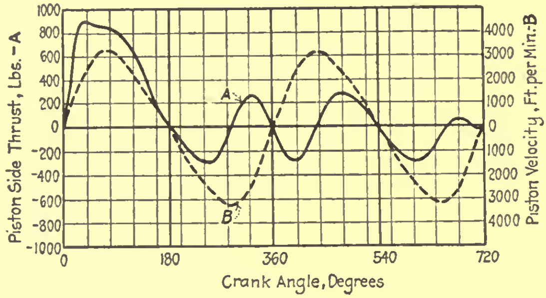


FIG. 36.—Side thrust against the cylinder walls of the Liberty-12 engine.

single cylinder and consequently the crankshaft must be as strong at the free end as at the propeller end.

Side Thrust.—Side thrust of the piston against the cylinder wall exists in consequence of the obliquity of the connecting rod; it disappears at the two dead centers. Its magnitude, G (Fig. 32), is given by $G = P \tan b$. Since $r \sin a = l \sin b$,

$$\frac{G}{P} = \frac{r \sin a}{\sqrt{l^2 - r^2 \sin^2 a}}$$

For the Liberty engine with indicator diagram, as in Fig. 30, the side thrust is as shown in Fig. 36, the maximum value reaching nearly 1,000 lb. Change of sign indicates change of thrust from one side of the cylinder to the other. In stationary cylinder engines side thrust is important only in relation to frictional wear.

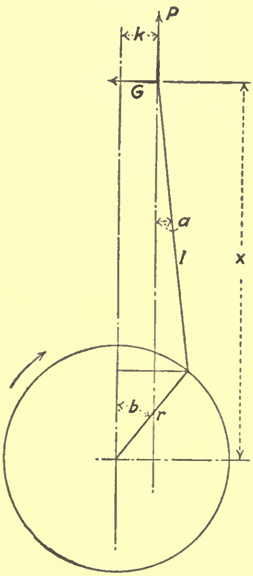


FIG. 37.—Diagram showing the effects of the obliquity of the connecting rod in an offset cylinder.

Offset Cylinders.—The side thrust during the expansion stroke, and the friction loss and cylinder wear, may be reduced by *offsetting* the cylinder: that is, by so locating it that its center line does not pass through the axis of the crankshaft. Minor effects of this arrangement are that the piston stroke is slightly greater than twice the crank throw and the mean speed of the piston is greater during the down stroke than the up stroke. This last point has the advantage of reducing the heat loss to the cylinder walls during the expansion period.

In Fig. 37, if k = offset and x = distance from a point on the piston to a horizontal plane through the crankshaft axis (in a vertical engine), then—

$$x = r\cos b + l\cos a$$

$$\frac{dx}{db} = -r\sin b - l\sin a\frac{da}{bd}$$

$$\frac{d^2x}{db^2} = -r\cos b - l\sin a\frac{d^2a}{db^2} - l\left(\frac{da}{db}\right)^2\cos a$$

$$a = \sin^{-1}\left(\frac{r\sin b - k}{l}\right)$$

$$\frac{da}{db} = \frac{r\cos b}{l\sin a}$$

$$\frac{d^2a}{db^2} = \frac{-l\cos a\cdot r\sin b + r\cos b\cdot l\sin a\frac{da}{db}}{l^2\cos^2 a}$$

$$\frac{d^2x}{db^2} = -r\cos b - l\cos a\frac{r^2\cos^2 b}{l^2\cos^2 a}$$

$$+ l\sin a\frac{l\cos a\cdot r\sin b + r\cos b\cdot l\sin a\cdot\frac{da}{db}}{l^2\cos^2 a}$$

$$= -r\cos b - \frac{r^2\cos^2 b}{l\cos a} + r\sin b\cdot\tan a - \frac{r^2\cos^2 b\cdot\tan^2 a}{l\cos a}$$

$$= -r\cos b - \frac{r^2\cos^2 b}{l\cos a}(1 + \tan^2 a) + r\sin b\cdot\tan a.$$

The acceleration of the reciprocating parts is equal to

$$\left(\frac{2\pi n}{60}\right)^2\frac{d^2x}{db^2} = 0.011n^2\frac{d^2x}{db^2}$$

and the accelerating force is

$$P_a = \frac{W}{g}\left(0.011n^2\frac{d^2x}{db^2}\right) = 0.00034Wn^2\frac{d^2x}{db^2}$$

The side thrust is given by $G = P\tan a$.

In using the above equations it should be noted that the angle a is positive when the connecting rod swings away from the crankshaft, as in the position shown in Fig. 37, and becomes negative on the other side of the vertical. The angle a is found for any value of b from the equation

$$l \sin a = r \sin b - k$$

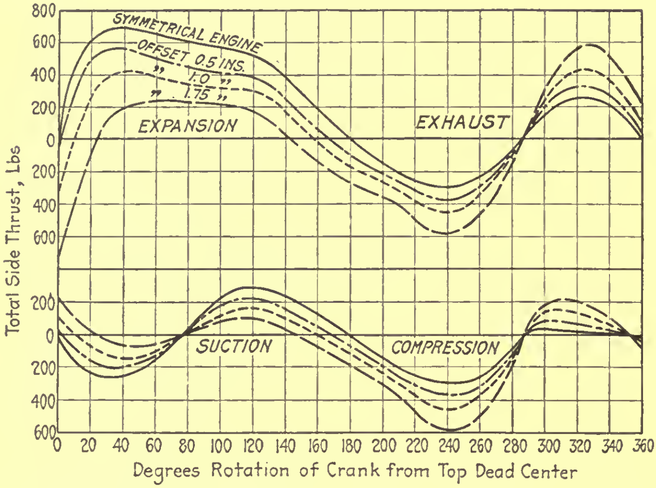


FIG. 38.—Effects of different degrees of offset on the side thrust in a single cylinder of the dimensions of the Liberty-12 engine.

The results of an analysis of the Liberty engine with offsets of 0.5, 1.0 and 1.75 in. gives side thrusts as shown in Fig. 38. It will be seen that with an offset of half the crank throw (1.75 in.) the side thrusts during the exhaust and compression strokes are nearly as high as the maximum value reached in an engine without offset. The lowest maximum is obtained with an offset of 1 in.

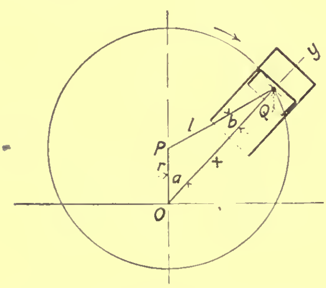


FIG. 39.—Diagram of rotary engine.

Rotary Engines.—The turning moment in a rotary engine results entirely from side thrust on the cylinder walls. This thrust is due not only to the obliquity of the connecting rod, as with stationary-cylinder engines, but also to thrust resulting from tangential acceleration of the reciprocating parts. The radial and tangential accelerations of these parts and the inertia forces resulting from them may be determined as follows.

Take a single cylinder, as in Fig. 39, rotating about the shaft O, while the connecting rod rotates about the fixed crankpin P.

The piston pin Q will move along the axis OX as that axis rotates about Q. If the length $OQ = x$, the point Q will undergo radial acceleration α_R along OQ and also tangential acceleration α_T at right angles to OQ. The magnitudes of these accelerations are given by the general theorems:

$$\alpha_R = \frac{d^2x}{dt^2} - x\left(\frac{da}{dt}\right)^2$$

and

$$\alpha_T = 2\frac{dx}{dt}\cdot\frac{da}{dt} + x\frac{d^2a}{dt^2}$$

With rotary engines the angular velocity ω of the cylinders is constant; or

$$\frac{da}{dt} = \omega.$$

and

$$\frac{d^2a}{dt^2} = 0.$$

Consequently

$$\alpha_R = \frac{d^2x}{dt^2} - x\omega^2$$

and

$$\alpha_T = 2\omega\frac{dx}{dt}$$

To find values of $\frac{dx}{dt}$ and $\frac{d^2x}{dt^2}$, the relations, $r \sin a = l \sin b$,

and, $x = r \cos a + l \cos b$, are used. Combining these,

$$x = r \cos a + l\sqrt{1 - \frac{r^2 \sin^2 a}{l^2}}$$

$$= r \cos a + l\left(1 - \frac{r^2 \sin^2 a}{2l^2}\right) \text{ approximately. Then}$$

$$\frac{x}{r} = \cos a + \frac{l}{r} - \frac{r}{2l}\sin^2 a$$

But $a = \omega t$

therefore

$$\frac{x}{r} = \frac{l}{r} + \cos \omega t - \frac{r}{2l}\sin^2 \omega t$$

$$\frac{1}{r}\frac{dx}{dt} = -\omega \sin \omega t - \omega\frac{r}{l}\sin \omega t \cos \omega t$$

$$\frac{1}{r}\frac{d^2x}{dt^2} = -\omega^2 \cos \omega t + \omega^2\frac{r}{l}(\sin^2 \omega t - \cos^2 \omega t).$$

Then,

$$\frac{\alpha_R}{r} = \frac{l}{r}\frac{d^2x}{dt^2} - \frac{x}{r}\omega^2$$

$$= -\frac{l}{r}\omega^2 - 2\omega^2\cos\omega t + \frac{3}{2}\omega^2\frac{r}{l}\sin^2\omega t - \omega^2\frac{r}{l}\cdot\cos^2\omega t$$

and

$$\alpha_T = 2\omega\frac{dx}{dt}$$

$$= -2\omega^2 r\sin\omega t\left(1 + \frac{r}{l}\cos\omega t\right)$$

The values of α_R, multiplied by the mass of the reciprocating parts, give the forces necessary to overcome inertia in the radial direction. Combining these forces with the total gas pressures (as in Fig. 31) gives the axial force P from which the side thrust G (Fig. 32) resulting from obliquity of the connecting rod is obtained, as on page 48. To find the total side thrust there must be added to this the thrust due to the tangential acceleration α_T, which is the product of the acceleration by the mass of the reciprocating parts. The product of the total side thrust by the distance of the piston pin from the crankshaft is the turning moment.

For several cylinders the turning moments are additive. As the cylinders are spaced at equal angular intervals, the total turning moment at any angular displacement a of one cylinder is

$$T = T_{(a)} + T_{\left(a + \frac{1}{n}\cdot 720\right)} + T_{\left(a + \frac{2}{n}\cdot 720\right)} + \cdots + T_{\left(a + \frac{n-1}{n}\cdot 720\right)}$$

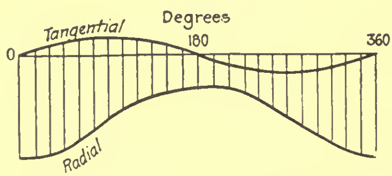


Fig. 40.—Variation of radial and tangential accelerations of the reciprocating parts of a rotary engine.

The quantities in brackets are the angular displacements of the various cylinders: n is the total number of cylinders.

Mayer[1] has calculated the turning moments for various arrangements of rotating-cylinder engines. For a seven-cylinder engine, 110 mm. diameter, 120 mm. stroke, length of connecting rod 213 mm., weight of reciprocating parts 1.3 kg., making 1,600 r.p.m., the radial and tangential accelerations α_R and α_T vary throughout the revolution as indicated in Fig. 40. It will be seen that the radial acceleration is always negative,

[1] "*Etude dynamique des moteurs à cylindres rotatifs.*"

instead of changing sign as in the acceleration of the reciprocating parts of stationary-cylinder engines. Combining the inertia force required to give the reciprocating parts this radial acceleration with the gas pressure force, the resultant force acting along the cylinder axis varies as shown in the solid black line in Fig. 41. It will be seen that this force is positive only for a very short period at the beginning of expansion. If the speed of the engine is increased the condition is soon reached when the resultant force on the piston pin is negative throughout the cycle. In that case the connecting rod would be under tension all the time (a very favorable condition, permitting considerable lightening of the rod) and the connecting rod might be replaced by a chain except that it is under compression before the engine has attained its full speed.

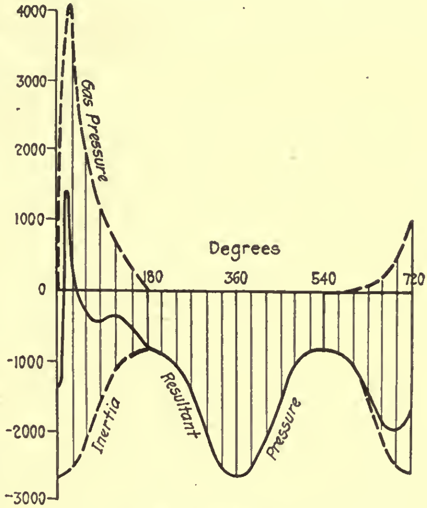


FIG. 41.—Forces acting along the cylinder axis of a rotary engine.

The turning moment for a seven-cylinder Gnome engine of the eimensions given above is shown in Fig. 42. The maximum dxcess of power developed over the mean resistance (shaded

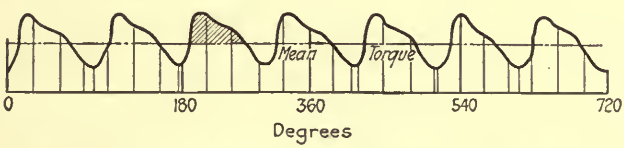


FIG. 42.—Turning moment of a 7-cylinder Gnome engine.

area) is 60 ft.-lb., or about $\frac{1}{120}$ of the total kinetic energy of the engine, and would be a much smaller fraction of the kinetic energy of engine and propeller combined. The earlier Gnome engines governed by cutting out the explosion on one or more of the cylinders according to the power requirements. The cutting out of a cylinder has a serious effect on the uniformity of turning moment and results in a maximum deficiency of work done during the cycle of 280 ft.-lb. as compared with the maximum excess (or deficiency) of 60 ft.-lb. when all cylinders are functioning.

The resultant force on the crankpin of the above engine at full load varies from 2,640 to 800 lb.

BALANCING

The forces acting in an engine are of two kinds, in so far as their effect on stresses is concerned. The gas pressure in the cylinder, being exerted both on the cylinder head and on the piston, subjects the engine to equal and opposite forces which are inherently balanced, or, in other words, has no effect in displacing the engine as a whole. The inertia forces of the moving part in each cylinder are not inherently balanced. If the engine is to operate smoothly (without vibration) it must be so designed as to make these unbalanced forces counteract one another as far as possible.

There are two kinds of moving parts to be considered: (*a*) the rotating parts, and (*b*) the reciprocating parts.

Rotating Parts.—A body of mass m (weight $W = mg$) revolving with angular velocity ω radians per second (n r.p.m.) in a circle of radius r feet has an acceleration of $\omega^2 r$ which gives rise to a centrifugal force $F = m\omega^2 r = 0.00034\ Wn^2 r$ lb. acting radially. The rotating body is usually composed of the crank, crankpin, and the large end of the connecting rod. The resulting force F is an unbalanced force acting on the crankshaft. In multicrank engines there will always be a number of these forces, one acting at each crank. If there is more than one connecting rod attached to the crank (as in Vee and fixed radial engines) the revolving mass includes the large ends of all the connecting rods attached to the crank. In rotating cylinder engines the revolving mass includes the cylinders themselves but not the crank and crankpin.

It is easy to balance the rotating parts. To accomplish this it is necessary (1) that the vector sum of these unbalanced forces should be zero and (2) that the sum of their couples about any (arbitrarily selected) plane should be zero. As the centrifugal forces on all the cranks are equal, the condition (1) is met, with any number of cylinders greater than one, by making the crank angle intervals equal. Condition (2) can be met, with any number of cylinders greater than two, if the cranks are spaced at equal distances apart along the shaft.

There is an unbalanced centrifugal pressure on the crankpin due to the inertia of the big ends of the connecting rods, and on the main bearings due to the inertia of all the rotating parts.

Reciprocating Parts.—The inertia force of the reciprocating parts has already been seen to be

$$P_a = 0.00034Wn^2r\left(\cos a \pm \frac{r}{l}\cdot\cos 2a\right)$$

If the connecting rod were infinitely long the expression would be

$$P_{aI} = 0.00034\ Wn^2r\cdot\cos a.$$

The second term in the brackets is due to the obliquity of the connecting rod and increases in value as l becomes shorter. It is customary to separate the two terms in a discussion of balancing, the quantity P_{aI} being called the **primary inertia force,** while

$$P_{aII} = \pm\ 0.00034Wn^2\frac{r^2}{l}\cdot\cos 2a$$

is called the **secondary inertia force.** It is found that the balance of the primary forces is much more easily achieved than that of the secondary forces. Since $\frac{r}{l}$ is usually about ¼, the magnitude of the secondary forces is only about one-quarter that of the primary forces so that secondary balance is not as important as primary balance.

The conditions for balance of primary and secondary inertia forces are exactly the same as those for centrifugal forces. There is, however, this difference, that these forces act always in the direction of the line of stroke. The practice of balancing the primary forces by the use of counterbalancing masses attached to the crankshaft is inadmissible (where avoidable) because of the necessity of keeping the total weight as low as possible and also because the same result can be obtained by multicylinder construction.

The following general results of an analysis of various possible cylinder arrangements may be stated, it being assumed that the reciprocating masses are the same in all cylinders:

With **cylinders in a line** and equally spaced at intervals of L ft.:

For *three-crank engines* with cranks at 120 deg., the primary and secondary forces are balanced, but the primary and secondary couples are not. The maximum unbalanced primary couple is $\sqrt{3}\cdot m\omega^2 rL$.

For *four-crank engines* with cranks at 180 deg., the primary forces are balanced but the secondary forces are not. The secondary forces in a vertical engine have a vertical resultant which is equal to $4m\omega^2 \frac{r^2}{l}$ at each quarter turn (0 deg., 90 deg., 180 deg., etc.) and become zero at each eighth turn (45 deg., 135 deg., etc.).

For *five-crank engines* with cranks at 72 deg., the primary and secondary forces are balanced, but the couples are not. The maximum unbalanced primary couple is 2.5 $m\omega^2 rL$; the maximum unbalanced secondary couple is $5m\omega^2 \frac{r^2}{l}\cdot L$.

For *six-crank engines* with cranks at 120 deg., the primary and secondary forces and couples are all balanced.

With **opposed cylinder engines,** with two cylinders and cranks at 180 deg., the primary and secondary forces are balanced; the primary couple is unbalanced with a maximum value of $m\omega^2 rL$.

With **Vee engines:**

For *six cylinders,* angle of Vee 120 deg., three cranks at 120 deg., the primary and secondary forces are balanced; the couples are unbalanced; maximum unbalanced primary couple is $\frac{\sqrt{3}}{2}\cdot m\omega^2 r\cdot L$.

For *eight cylinders,* angle of Vee 90 deg., four cranks at 180°, primary forces balanced, secondary forces unbalanced; primary and secondary couples balanced. Unbalanced secondary force acts always at right angles to the longitudinal plane of symmetry and has a maximum value of 5.6 $m\omega^2\frac{r^2}{l}$.

For *twelve cylinders,* angle of Vee 60 deg., six cranks at 120 deg., primary and secondary forces and couples all balanced.

With **fixed radial engine** of k cylinders with all the connecting rods on one crank, the inertia forces have a constant resultant of $\frac{k}{2}m\omega^2 r$, approximately, along the crank. It can be balanced completely by a counterbalance mass of $\frac{km}{2}$ at radius r, opposite the crankpin. If there are two banks of cylinders, each with k cylinders and with cranks at 180 deg., the primary forces will balance but there will remain an unbalanced primary couple of $\frac{k}{2} \cdot m\omega^2 rL$.

Rotating-cylinder engines have two sets of rotating masses:

1. The cylinders, which are perfectly balanced, and
2. The pistons and connecting rods, which have primary balance but not secondary balance.

PERIODIC UNBALANCED FORCES

The results of the existence of periodic unbalanced forces in an airplane engine may be two-fold: (1) to move the engine as a whole, and (2) to distort the engine.

The engine in an airplane is supported on wooden members which are flexible and are in no way the equivalent of a rigid foundation. If the engine moves as a whole it will flex its supports instead of moving the airplane as a whole. The maximum possible duration of the periodic variation of any unbalanced force is one engine revolution or about $1/25$ sec.; ordinarily it will be not more than one-quarter of a revolution or $1/100$ sec. This time is too short to permit the unbalanced force to produce appreciable deviation of the airplane as a whole although it may set up vibration in it.

The crankcase of the airplane engine is always very thin and consequently flexible. Much of the unbalanced force may be taken up in producing distortion of the crankcase. This is very undesirable since it necessarily results in varying alignments of the crankshaft bearings and consequent increase in shaft friction.

Both the engine supports and the crankcase have their own natural periods of vibration. If the frequency of the disturbances due to unbalanced forces in the engine is the same as the natural frequency of the vibration of either of these members, the amplitude of vibration will be greatly increased beyond the very small amount due to a single application of the unbalanced force. It is most important that the critical speed at which the vibration of the engine on its supports becomes a maximum should be avoided, as well as simple multiples of those speeds. The natural periods of vibration of such complex structures as engines, on their supports in an airplane, can not be calculated. If excessive vibration is found at or near the designed speed of the engine the structures must be altered to change the natural period.

Various devices have been used for **neutralizing the unbalanced secondary forces** in four- and eight-cylinder engines. A good example of these is the Lanchester balancer (Fig. 43), which has had considerable application to automobile engines. This consists of two exactly similar unbalanced cylinders located under the center main bearing and driven by a gear on the crankshaft. These cylinders revolve in opposite directions and their unbalanced weights are located so that their common center of gravity travels up and down in a vertical plane and so balances the displacement of the common center of gravity of the pistons, which falls in a plane at the middle of the piston travel when the pistons are on

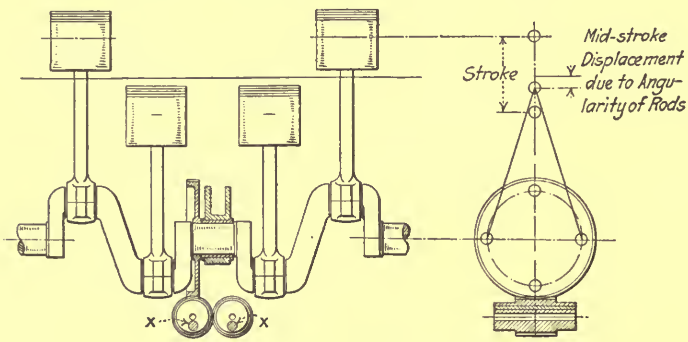


Fig. 43.—Lanchester balancer.

their dead centers, but falls below it when the cranks have revolved 90 deg. past that position.

It should be clearly recognized that complete balance of primary and secondary forces and couples does not ensure absence of vibration. Such complete balance means that the engine as a whole has no tendency to move, but there are always internal stresses, particularly those imposed by the opposing couples on the engine structure. If the periodicity of the application of these stresses coincides with the natural period of the structure (or some fraction of it) severe vibrations may be set up. In any case heavy bearing loads are likely to be imposed, especially in the center main bearing, through the action of opposing couples.

For torsional oscillations of the crankshaft see p. 146.

The following table gives values of inertia and centrifugal forces, resulting bearing pressures and other calculated quantities for the six-cylinder 200-h.p. Austro-Daimler engine, the six-cylinder 270-h.p. Bassé-Selve engine and the 12-cylinder 400-h.p. Liberty engine:

INERTIA FORCES, BEARING LOADS, ETC.

	Austro-Daimler engine	Bassé-Selve engine	Liberty-12 engine
Weight of piston complete with rings and piston pin, lb.	4.18	6.187	3.838
Weight per sq. in. of piston area, lb.	0.188	0.211	0.1955
Weight of connecting rod complete, lb.	4.84	9.00	2.9 and 6.35
Weight of reciprocating part of connecting rod, lb.	1.66	2.25	1.225
Total reciprocating weight per cylinder, lb.	5.84	8.437	5.063
Weight per sq. in. of piston area, lb.	0.263	0.288	0.258
Length of connecting rod (centers), in.	12.40	14.17	12.0
Ratio connecting rod length to crank throw	3.6:1	3.6:1	3.44:1
Inertia, lb. per sq. in. piston area (top center)	63.8	80.7	117.0
Inertia, lb. per sq. in. piston area (bottom center)	36.2	45.7	65.2
Inertia, lb. per sq. in. piston area (mean)	25.0	31.6	
Weight of rotating mass of connecting rod, lb.	3.18	6.75	1.675 and 5.125
Total centrifugal pressure, lb.	610.0	1,480.0	1,469.0
Centrifugal pressure, lb. per sq. in. piston area	27.5	50.7	74.8
Mean average loading on crankpin bearing, total from all sources, lb. per sq. in. piston area	91.0	115.7	175.0
Diameter of crankpin, in.	2.20	2.75	2.375
Rubbing velocity, ft. per sec.	13.42	16.8	17.7
Effective projected area of big-end bearing, sq. in.	5.02	5.39	5.34
Ratio piston area to projected area of big-end bearing	4.42:1	3.5:1	3.68:1
Mean average loading on big-end bearing, lb. per sq. in.	402.0	405.0	642.0

CHAPTER IV

ENGINE DIMENSIONS AND ARRANGEMENTS

Certain special requirements control the selection of the dimensions and arrangements of airplane engines. These are:

1. Minimum total weight of the engine and its accessories per brake horse power developed, or maximum power output per pound of weight.
2. Maximum fuel economy.
3. Compactness.
4. Freedom from unbalanced forces and from vibration.
5. Reliability.

To obtain **minimum weight per horse power,** it is necessary that the engine should have minimum weight per cubic foot of piston displacement per revolution, and that it should be operated with maximum power per cubic foot of cylinder volume. The latter demands a combination of the maximum obtainable mean effective pressure with high speed of revolution. The mean effective pressure is constant at moderate engine speeds but falls off at high speeds in consequence of falling volumetric efficiency. Beyond a certain limiting speed the mean effective pressure will fall off more rapidly than the increase in engine revolutions per minute (see p. 37) and the engine power will decrease. This limiting speed should be made as high as possible by making the valve openings large and the inlet and exhaust manifolds short and of ample cross-section.

All airplane engines must be multicylindered in order to give the necessary uniformity of turning moment and freedom from unbalanced forces. The weight of the engine per cubic foot of piston displacement per revolution will depend on the unit size of cylinder selected. Comparing two unit cylinders of exactly the same form but of different sizes, it will be found that the thickness of the cylinder walls will not have to be increased as rapidly as the cylinder diameter because the wall (for structural reasons) is always made thicker than the stresses demand, by an amount which does not vary much with the diameter. Consequently the weight of the cylinder per cubic foot of piston displacement diminishes with increased size, and the same is true of most of the other engine parts.

On the other hand, the weight of the engine per cylinder diminishes with increase in the number of cylinders in line. The engine consists of a number of exactly similar units (cylinder, running parts, section of crankcase, etc.) and certain approximately constant weights such as ends of crankcase, pumps, magnetos, propeller hub and so forth. The addition of more cylinders will diminish the weight of the engine per cylinder.

A further diminution in weight of the engine per cylinder can be obtained by using the Vee or W arrangement of cylinders. A substantial saving in the weight of crankshaft and crankcase, per cylinder, results from these arrangements. Still greater saving results from the adoption of the radial arrangement.

Considerations of torque and balance (see p. 56) indicate that the number of cylinders should not be less than six for an engine of moderate power (100 to 150 h.p.). For higher powers the choice is between more cylinders and larger cylinders. Engines have been built with as many as 24 cylinders, but it does not seem likely that this number will be much used or exceeded.

Cylinder size can be increased by increasing either diameter or stroke or both. The diameter is at present limited to from 6 to 7 in. by the difficulty of keeping the piston cool. The heat given to the center of the piston has to travel radially to the walls, and it is necessary to increase the thickness of the piston as the diameter increases in order to give sufficient section of metal to carry the heat away. This results in a heavy piston and excessive inertia forces in the reciprocating parts. The engine stroke is also limited at present to 8 in.; increase in stroke beyond that limit increases the over-all height of the engine to dimensions which are difficult to accommodate without increasing the size of the fuselage. Furthermore, increase in stroke increases the weight of the engine more than does a corresponding increase in diameter. The small ratio of stroke to diameter which characterizes airplane engines is not as objectionable as it would be in other engines; it increases the ratio of water-jacketed surface to cylinder volume, but the percentage of heat lost to the jacket is nevertheless smaller than in other engines in consequence of the high engine speed and high mean effective pressure.

The weight of the engine per cubic foot of piston displacement per revolution is also a function of the ratio of connecting rod length to stroke. The smaller this ratio the less is the over-all height and the weight of the engine. The objection to a small ratio is the increase in magnitude of the secondary inertia forces which result from the obliquity of the connecting rod. These secondary forces can be perfectly balanced with certain arrangements of cylinders (see p. 56) and the objection eliminated. The ratio usually ranges from 1.5 to 1.7.

High **fuel economy** is important primarily in its effect on the weight to be carried by the plane. For a five-hour flight, an engine weighing 2.5 lb. per horse power and using 0.5 lb. of fuel per horse-power hour will have the same total weight of engine and fuel as an engine weighing 2 lb. per horse power and using 0.6 lb. of fuel per horse-power hour. The heavier and more efficient engine would ordinarily be the better of the two in respect to reliability and durability. To obtain high fuel economy the most important factor is the ratio of compression which should be as great as can be used without detonation or preignition.

Compactness is important both in respect to frontal area and over-all length. The frontal area of large engines should, if possible, be of such form and dimensions as not to require any increase in the cross-section of the containing fuselage. Short over-all length is distinctly advantageous and permits the fuel tanks to be located close to the center of gravity of the plane.

Freedom from vibration is necessary because the mounting of the engine is on elastic supports—usually wooden longerons—which leave the crankcase free to distort under internal forces. Such arrangements of cylinders as will eliminate or minimize unbalanced forces are desirable but they cannot be relied upon to prevent vibration. Even with the most perfect balancing, torsional vibration of the crankshaft may produce excessive vibrations at certain engine speeds; such speeds must be avoided.

Engine Arrangements.—Airplane engines have been built in a great variety of arrangements of which a number have survived. These may be classified as (1) radial, (2) vertical, (3) Vee, (4) **W**, (5) **X**. Radial engines (both fixed and rotary) are discussed in Chapter VIII. They have minimum weight per horse power and shortest over-all length, but they have maximum frontal area and until very lately have shown economy inferior to that of the other types. They are generally air cooled.

The vertical engine (Figs. 8 and 9) has one row of cylinders. The **Vee** engine (Figs. 46 and 47) has two linear rows of cylinders;

the "angle of the Vee" is the acute angle between the axial planes of the two rows. The **W** or Ψ engine (Figs. 72 and 73) has three rows of cylinders of which the central one is vertical and the other two form equal angles with the vertical. The **X** engine has four rows of cylinders arranged symmetrically about the vertical and horizontal planes but not necessarily with equal angles between the planes of the cylinder axes. In the radial engine (Fig. 136) three or more cylinders, constituting a group, have their axes intersecting at a point on a common shaft. The *radial* or *fixed* engine should not be confused with the *rotary* engine, in which the cylinders revolve about a stationary crankshaft. A radial engine may consist of more than one group or bank of cylinders, each group having its own common plane of cylinder axes, perpendicular to the axis of the shaft.

Each cylinder of a four-cycle engine requires two revolutions or 720 deg. of rotation of the crankshaft to complete its cycle.

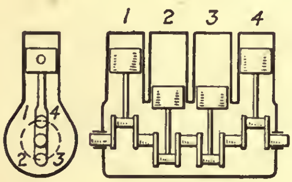


FIG. 44.
Four-cylinder engine

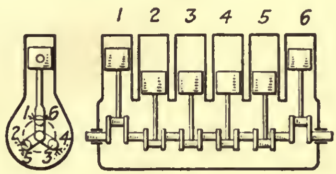


FIG. 45.
Six-cylinder engine

The explosions in a cylinder occur every other revolution. If explosions are to occur in n cylinders at equal intervals, the interval expressed in degrees of crankshaft rotation must be $720 \div n$. Explosion always occurs when the piston is closest to the cylinder head. With a four-cylinder engine, the interval between explosions is 180 deg.; the crankpins lie all in one plane passing through the shaft axis, one possible arrangement being shown in Fig. 44. A six-cylinder vertical engine has the cranks $720 \div 6 = 120$ deg. apart: the crankpins lie in three planes intersecting at the shaft axis. One arrangement is shown by Fig. 45. Constancy of interval between impulses may be obtained with crank dispositions other than those shown in Figs. 44 and 45. For example, in Fig. 44, cranks 2 and 3 may be either in line or opposed. If the former, cranks 1 and 4 will be in line and 180 deg. away from 2 and 3. If the latter, 1 and 4 will be opposed, and either may be in line with 2. If equal intervals of time between explosions were the sole requisite, the number of possible crank arrangements with a 6-cylinder engine would be large. The actual crank arrangements are determined mainly by considerations of engine balance.

In a Vee engine each pair of cylinders (in one plane) acts on a common crankpin. If there are n cylinders in $\frac{n}{2}$ pairs, the crank interval is $720 \div \frac{n}{2}$. The interval between explosions of the cylinders is $720 \div n$. If θ is the angle of the Vee, any one crank moves through the angle θ in the interval necessary for the two pistons actuating it to reach respectively their highest positions. As the explosions occur with the pistons in their highest positions, the explosion in a leading cylinder will precede that of its following cylinder by an angle of θ, or $360 + \theta$, deg. according to whether both cylinders explode during the same revolution of the crank or the explosions occur in succeeding revolutions. The latter is always the case, because, if the second explosion occurred after the crank angle θ, the explosion pressure would be transmitted to a crankpin which was already subjected to the pressure of the gas expanding in the other cylinder of the pair and the crankpin would be unduly loaded. For equal explosion intervals the angle θ is equal to $720 \div n$. This leads to a 90-deg. angle for 8-cylinder and a 60-deg. angle for 12-cylinder Vee engines. These angles give the maximum uniformity of turning movement; other angles may be employed for special reasons but always at some sacrifice of uniformity of turning movement.

The choice as between vertical, Vee and W arrangement is largely determined by the number of cylinders. It has been found that much trouble is experienced with a crankshaft having more than six cranks and the over-all length of the engine becomes excessive. Eight-crank engines have been built but they have not survived. The perfect balancing of the six-crank engine has made it a favorite. With six cylinders the vertical arrangement is almost universally adopted; it has the advantage of having minimum frontal area and consequently of being most easily accommodated in the fuselage. With eight cylinders the 90-deg. Vee engine with four cranks is generally accepted as the best arrangement although the wide Vee angle results in considerable over-all engine width. With 12 cylinders the usual arrangement is a 60-deg. Vee and six cranks. The 45-deg. Vee adopted in the Liberty engine results in decreased width of engine but increased height; it also results in a less uniform turning moment on the crankshaft.

The W arrangement for 12 cylinders is with three rows of cylinders and four cranks; this shortens the over-all length and decreases the weight but greatly increases the engine width, especially if a 60-deg. angle (which gives most uniform torque) is used between the rows. With 18 cylinders the W arrangement with three rows of cylinders and six cranks is used; the angle between the rows for most uniform torque should be 40 deg., which diminishes the over-all width as compared with the 12-cylinder W engine.

Vertical, Vee and W engines are nearly always water-cooled in airplane practice. Air cooling has been successful only with low compression ratios.

Engine Dimensions.—Table 3 gives the general dimensions and arrangement of the principal American and foreign airplane engines. The horse powers given are generally the maker's rating, but they are naturally variable with the fuel, the carburetor, the manifolding, the ratio of compression, and the engine speed. The ratio of compression is readily varied by changing the dimensions of the piston and for certain engines different pistons are supplied, according to whether the engine is to be flown at low altitude (as in seaplanes) or at higher altitudes. The dry weight includes carburetors, magnetos, and propeller hub. The weight per horse power naturally varies with the horse power and is given for the rated horse power. It is the product of two factors, the weight per cubic inch of piston displacement and the piston displacement per horse power. The piston displacement per horse power is the aggregate displacement volume of the cylinders per stroke divided by the horse power developed. It is an excellent measure of the degree to which the piston displacement is utilized. The aggregate displacement volume is equal to $l \times a \times n$ cu. in. where l is the stroke in inches, a the piston area in square inches, and n is the number of cylinders. The horse power $= \dfrac{p \times l \times a \times n \times N}{2 \times 12 \times 33{,}000}$, where p is the mean effective pressure, and N is the revolutions per minute. The piston displacement per horse power $= 33{,}000 \times 24 \div p \times N$; that is, it depends only on the mean effective pressure and the revolutions per minute.

TABLE 3.—GENERAL DIMENSIONS OF AIRPLANE ENGINES

Engine	Nationality	No. of cylinders	Rated horse power	Bore		Stroke		Ratio stroke to bore	Revolutions per minute		Compression ratio	Piston speed, ft. per min.
				In.	Mm.	In.	Mm.		Engine	Propeller		
Rotary Engines												
Gnome	French	7	50	4.33	110	4.72	120	1.09	1,200	1,200	....	945
Gnome	French	7	80	4.33	110	5.90	150	1.36	1,200	1,200	....	1,180
Gnome (A. I.)	French	9	103	4.33	110	5.90	150	1.36	1,200	1,200	4.95	1,180
Gnome (M_2)	French	14	100	4.33	110	4.72	120	1.09	1,200	1,200	4.00	945
Gnome	French	14	160	4.88	124	5.52	140	1.13	1,200	1,200	....	1,102
Gnome	French	18	200	4.88	124	5.90	150	1.21	1,200	1,200	....	1,180
Le Rhone	Amer.	9	80	4.13	105	5.52	140	1.33	1,200	1,200	4.8	1,100
Le Rhone	Amer.	9	100	4.50	114	6.00	152	1.33	1,200	1,200	....	1,200
Le Rhone	French	14	120	4.13	105	5.52	140	1.33	1,200	1,200	....	1,102
Clerget	French	7	80	4.72	120	5.90	150	1.25	1,180	1,180	....	1,160
Clerget	French	9	122	4.72	120	6.30	160	1.33	1,250	1,250	4.36	1,260
Double Rotary												
Siemens-Halske	German	11	200	....	...	....	...	......	(1,800)	900	....	
Radial Air-cooled												
Anzani	French	6	60	4.13	105	4.72	120	1.14	1,300	1,300	....	1,022
Anzani	French	10	100	4.13	105	5.52	140	1.33	1,200	1,200	....	1,102
Anzani	French	20	200	4.13	105	5.52	140	1.33	1,300	1,300	....	1,196
A. B. C., Gnat	British	2	45	4.75	121	5.50	140	1.16	1,800	1,800	....	1,650
A. B. C., Wasp	British	7	170	4.50	114	5.90	150	1.31	1,750	1,750	4.02	1,720
A. B. C., Dragonfly	British	9	320	5.50	140	6.50	165	1.18	1,650	1,650	4.42	1,785
Cosmos, Lucifer	British	3	100	5.75	146	6.25	159	1.09	1,600	1,600	....	1,670
Cosmos, Jupiter	British	9	400	5.75	146	7.50	200	1.31	1,650	1,650	5.00	2,165
Cosmos, Jupiter geared	British	9	450	5.75	146	7.50	200	1.31	1,850	1,200	5.00	2,310
Cosmos, Mercury	British	14	315	4.37	111	5.81	148	1.33	1,800	1,800	....	1,940
Cosmos, Hercules	British	18	1,000	6.25	159	6.50	165	1.04	1,750	1,150	....	1,900
Lawrance	Amer.	3-sym.	60	4.25	108	5.25	133	1.23	1,900	1,900	....	1,660

TABLE 3.—*Continued*

Engine	Brake m.e.p., lb. per sq. in.	Torque, lb.-ft.	Valves per cylinder: Inlet, No.	Inlet, Diam., in.	Inlet, Lift, in.	Exhaust, No.	Exhaust, Diam., in.	Exhaust, Lift, in.	Weight of engine dry, lb.	Weight per cu. in. piston displacement, lb.	Piston displacement per h.p., cu. in.	Weight of engine per h.p., lb.
Rotary Engines												
Gnome	59.3	219	0	.....	.....	1	.....	.....	172	0.354	9.72	3.44
Gnome	86.8	350	0	.....	.....	1	.....	.....	210	0.346	7.60	2.63
Gnome (A. I.)	88.7	450	0	.....	.....	1	2.360	0.413	270	0.346	7.58	2.62
Gnome (M_2)	67.9	438	0	.....	.....	1	.....	.....	220	0.226	9.72	2.20
Gnome	73.2	700	0	.....	.....	1	.....	.....	396	0.274	9.03	2.48
Gnome	66.5	876	0	.....	.....	1	.....	.....	540	0.272	9.92	2.70
Le Rhone	81.0	350	1	1.80	0.562	1	1.800	0.562	260	0.390	8.32	3.25
Le Rhone	76.9	438	..	.....	.....	..	.....	.....	285	0.332	8.58	2.85
Le Rhone	76.3	525	..	.....	.....	..	.....	.....	375	0.362	8.64	3.13
Clerget	74.4	363	..	.....	.....	..	.....	.....	216	0.299	9.03	2.70
Clerget	78.0	512	1	1.575	0.394	1	1.968	0.453	367	0.370	8.12	3.00
Double Rotary												
Siemens-Halske	.....	.....	..	.....	.....	..	.....	.....	427	......	....	2.14
Radial Air-cooled												
Anzani	94.0	247	..	.....	.....	..	.....	.....	176	0.452	6.48	2.93
Anzani	89.2	446	..	.....	.....	..	.....	.....	305	0.412	7.40	3.05
Anzani	82.4	892	..	.....	.....	..	.....	.....	680	0.459	7.40	3.40
A. B. C., Gnat	101.0	131		.....	.....	....	.....	.....	115	......	....	2.30
A. B. C., Wasp	115.0	510	1	2.17	0.470	2	1.338	0.295	260	0.390	3.87	1.53
A. B. C., Dragonfly	110.0	1,080	1	2.656	0.653	2	1.693	0.433	635	0.456	4.48	1.98
Cosmos, Lucifer	102.0	328	..	.....	.....	..	.....	.....	.....	.....	4.86	....
Cosmos, Jupiter	113.0	1,270	2	.....	.....	2	.....	.....	636	0.364	4.38	1.59
Cosmos, Jupiter geared	113.0	1,275	2	.....	.....	2	.....	.....	757	0.433	3.89	1.68
Cosmos, Mercury	112.0	920	..	.....	.....	..	.....	.....	587	0.482	3.87	1.86
Cosmos, Hercules	126.0	3,000	..	.....	.....	..	.....	.....	1,400	0.390	3.59	1.40
Lawrance	112.0	169	1	2.06	.....	1	2.06	.....	130	0.582	3.72	2.17

TABLE 3.—*Continued*

Engine	Nationality	No. of cylinders	Rated horse power	Bore		Stroke		Ratio stroke to bore	Revolutions per minute		Compression ratio	Piston speed, ft. per min.
				In.	Mm.	In.	Mm.		Engine	Propeller		
Radial Water-Cooled												
Salmson	French	9	110	4.72	120	5.52	140	1.17	1,250	1,250	....	1,150
Salmson	French	14	200	4.72	120	5.52	140	1.17	1,250	1,250	....	1,150
Salmson	French	9	250	4.92	125	6.70	170	1.36	1,500	1,500	5.3	1,670
Salmson	French	9	300	5.90	150	8.27	210	1.40	1,200	1,200	....	1,650
Salmson	French	18	600	5.90	150	8.27	210	1.40	1,200	1,200	....	1,650
Vertical												
Aeromarine	Amer.	6	85	4.32	110	5.63	143	1.30	1,400	1,400	....	1,310
Curtis, K6	Amer.	6	150	4.50	114	6.00	152	1.33	1,700	1,700	....	1,700
Hall-Scott, A5	Amer.	6	128	5.00	127	7.00	178	1.40	1,200	1,200	....	1,400
Hall-Scott, A5a	Amer.	6	152	5.25	133	7.00	178	1.33	1,200	1,200	....	1,400
Hall-Scott, L6	Amer.	6	200	5.00	127	7.00	178	1.40	1,700	1,700	5.50	1,985
Liberty	Amer.	6	231	5.00	127	7.00	178	1.40	1,700	1,700	5.42	1,985
Bassé-Selve	German	6	270	6.11	155	7.88	200	1.29	1,400	1,400	4.34	1,840
Benz, B	German	6	95	4.17	106	5.90	150	1.41	1,200	1,200	....	1,180
Benz, FF	German	6	160	5.12	130	7.10	180	1.38	1,400	1,400	....	1,655
Benz	German	6	230	5.72	145	7.50	190	1.31	1,400	1,400	4.93	1,750
Austro-Daimler	Austrian	6	90	4.72	120	5.52	140	1.17	1,300	1,300	....	1,195
Austro-Daimler	Austrian	6	120	5.11	130	6.90	175	1.35	1,200	1,200	....	1,380
Austro-Daimler	Austrian	6	200	5.31	135	6.90	175	1.30	1,400	1,400	5.03	1,610
Maybach	German	6	180	6.30	160	6.69	170	1.06	1,200	1,200	....	1,340
Maybach	German	6	294	6.50	165	7.09	180	1.09	1,400	1,400	5.95	1,654
Mercedes	German	6	150	5.52	140	6.30	160	1.14	1,250	1,250	4.80	1,310
Mercedes	German	6	252	6.30	160	7.09	180	1.12	1,400	1,400	4.94	1,655
Mercedes	German	6	174	5.52	140	6.30	160	1.14	1,400	1,400	4.64	1,468
Beardmore	British	6	178	5.60	142	6.90	175	1.23	1,350	1,350	4.56	1,550
Galloway	British	6	236	5.71	145	7.48	190	1.31	1,400	1,400	4.96	1,745
Siddeley-Puma	British	6	240	5.71	145	7.48	190	1.31	1,400	1,400	5.00	1,745
Isotta-Franchini, V5,	Italian	8	250	5.11	130	7.48	190	1.46	1,400	1,400	5.00	1,745
S. P. A., 6A,	Italian	6	210	5.31	135	6.70	170	1.26	1,600	1,600	5.00	1,785
Fiat, A12,	Italian	6	300	6.30	160	7.09	180	1.12	1,600	1,600	4.31	1,890
Twin Vertical												
Bugatti	Amer.	16	410	4.33	110	6.30	160	1.45	2,000	1,333	5.00	2,100

TABLE 3.—*Continued*

Engine	Brake m.e.p., lb. per sq. in.	Torque, lb.-ft.	Valves per cylinder						Weight of engine dry, lb.	Weight per cu. in. piston displacement, lb.	Piston displacement per h.p., cu. in.	Weight of engine per h.p., lb.
			Inlet			Exhaust						
			No.	Diam., in.	Lift, in.	No.	Diam., in.	Lift, in.				
Radial Water-cooled												
Salmson	80.2	462	..	.....	.....	..	.....	.....	552	0.635	7.90	5.02
Salmson	93.8	840	..	.....	.....	..	.....	.....	900	0.667	6.75	4.50
Salmson	115.0	874	..	.....	.....	..	.....	.....	472	0.412	4.58	1.89
Salmson	97.4	1,312	..	.....	.....	..	.....	.....	990	0.487	6.77	3.30
Salmson	97.2	2,620	..	.....	.....	..	.....	.....	2,460	0.605	6.8	4.10
Vertical												
Aeromarine	97.0	319	..	.....	.....	..	.....	.....	440	0.886	5.83	5.18
Curtiss, K6	122.0	463	2	1.625	.....	2	1.625	.....	400	0.700	3.81	2.67
Hall-Scott, A5	102.2	560	..	.....	.....	..	.....	.....	592	0.718	6.45	4.62
Hall-Scott, A5a	110.0	665	..	.....	.....	..	.....	.....	562	0.618	6.00	3.70
Hall-Scott, L6	113.0	618	..	.....	.....	..	.....	.....	500	0.607	4.12	2.50
Liberty	131.0	714	1	2.50	0.438	1	2.50	0.375	568	0.683	3.57	2.45
Bassé Selve	110.0*	1,020	2	2.20	0.39	2	2.20	0.39	885	0.640	5.12	3.28
Benz, B	130.0	415	..	.....	.....	..	.....	.....	540	1.12	5.08	5.71
Benz, FF	103.5	600	..	.....	.....	..	.....	.....	592	0.676	5.47	3.70
Benz	113.0	862	1	2.244	0.433	1	2.244	0.433	829	0.750	5.23	3.68
Austro-Daimler	95.0	364	..	.....	.....	..	.....	.....	316	0.546	6.43	3.51
Austro-Daimler	93.5	525	..	.....	.....	..	.....	.....	575	0.678	7.06	4.79
Austro-Daimler	123.5	750	2	1.73	0.390	2	1.73	0.40	729	0.798	4.58	3.65
Maybach	95.0	787	..	.....	.....	..	.....	.....	990	0.792	6.95	5.50
Maybach	117.7	750	2	1.89	0.372	2	1.89	0.368	872	0.619	4.80	2.97
Mercedes	105.0	630	..	.....	....	..	.....	.....	618	0.685	5.64	4.12
Mercedes	107.5	945	..	2.362	.....	1	2.362	.....	935	0.824	4.50	3.71
Mercedes	109.0	653	1	2.68	0.453	1	2.68	0.453	660	0.709	5.21	3.79
Beardmore	98.7	692	1	2.559	.....	1	2.559	.....	592	0.540	6.60	3.33
Galloway	116.0	885	1	2.716	0.472	2	1.811	0.472	690	0.601	4.91	2.97
Siddeley-Puma	118.0	900	1	2.598	0.472	2	1.732	0.472	625	0.544	4.78	2.61
Isotta-Franchini, V5,	115.2	938	1	.....	.....	..	.....	....	825	0.672	4.90	3.30
S. P. A., 6A,	116.8	689	..	.....	.....	..	.....	.....	539	0.606	4.24	2.57
Fiat, A12,	112.0	983	2	2.00	0.430	2	2.008	0.438	928	0.701	4.41	3.09
Twin Vertical												
Bugatti	109.2	1,075	2	1.535	0.653	1	2.263	0.700	1,170	0.790	3.62	2.85

* M.e.p. and r.p.m. assumed as engine was damaged and could not be tested.

TABLE 3.—*Continued*

Engine	Nationality	No. of cylinders and angle of Vee	Normal h.p.	Bore In.	Bore Mm.	Stroke In.	Stroke Mm.	Ratio stroke to bore	Revolutions per minute Engine	Revolutions per minute Propeller	Compression ratio	Piston speed, ft. per min.
Vee												
Curtiss, OX	Amer.	8–90°	93	4.00	102	5.00	127	1.25	1,400	1,400	4.50	1,165
Curtiss, VX	Amer.	8–90°	206	5.00	127	7.00	178	1.40	1,350	1,350	4.71	1,575
Curtiss, K12	Amer.	12–60°	375	4.50	114	6.00	152	1.33	2,250	1,350		2,250
Hall-Scott, A8	Amer.	12–60°	450	5.00	127	7.00	178	1.40				
Hispano-Suiza, I	Fr.-Amer.	8–90°	150	4.72	120	5.12	130	1.08	1,450	1,450	4.80	1,230
Hispano-Suiza, E	Fr.-Amer.	8–90°	190	4.72	120	5.12	130	1.08	1,800	1,800	5.32	1,535
Hispano-Suiza, II	Amer.	8–90°	325	5.52	140	5.90	150	1.07	1,800	1,800	5.32	1,578
Liberty	Amer.	12–45°	421	5.00	127	7.00	178	1.40	1,700	1,700	5.42	1,985
Packard	Amer.	8–60°	180	4.75	121	5.25	133	1.10	1,600	1,600	5.02	1,400
Packard	Amer.	12–60°	282	4.75	121	5.25	133	1.10	1,600	1,600	5.56	1,400
Sturtevant	Amer.	8–90°	140	4.00	102	5.50	140	1.38	2,000	1,200	....	1,830
Thomas-Morse, 88	Amer.	8–90°	150	4.13	105	5.50	140	1.33	2,000	.....	....	1,835
Duesenberg, V12	Amer.	12–60°	250	4.87	124	7.00	178	1.43	1,800	.....	....	2,100
Duesenberg (geared), H16	Amer.	16–45°	800	6.00	153	7.50	190	1.25	1,800	1,350	4.75	2,250
Rolls Royce, Falcon III	British	12–60°	270	4.00	102	5.75	146	1.44	2,200	1,300	5.30	2,110
Rolls Royce, Eagle VIII	British	12–60°	359	4.50	114	6.50	165	1.44	1,800	1,080	5.30	1,950
Rolls Royce, Condor	British	12–60°	600	5.50	140	7.50	190	1.36	1,700	935	5.10	2,120
Sunbeam, Arab 2	British	8–90°	212	4.72	120	5.11	130	1.082	2,000	1,200	5.30	1,710
Sunbeam, Maori 4	British	12–60°	275	4.33	110	5.31	135	1.23	2,100	2,100	....	1,860
Sunbeam, Cossack	British	12–60°	350	4.33	110	6.30	160	1.45	2,000	1,000	5.00	2,100
Sunbeam, Manitou	British	12–60°	300	5.32	135	5.60	142	1.05	2,000	1,272	....	1,860
Sunbeam, Matabele	British	12–60°	400	4.80	122	6.46	164	1.34	2,000	1,222	....	2,155
Sunbeam, Sikh	British	12–60°	800	7.08	180	8.26	210	1.17	1,400	.....	....	1,930
A. B. C.	British	8–90°	100	5.00	127	4.50	114	0.90	1,300	.....	....	975
Galloway-Atlantic	British	12–60°	550	5.72	145	7.48	190	1.31	1,500	1,500	5.40	1,872
Renault	French	12–47½°	400	4.92	125	6.30	160	1.28				
Peugeot	French	8–90°	200	3.93	100	7.09	180	1.80	1,850	.....	....	2,185
Panhard	French	8–90°	100	4.33	110	5.52	140	1.27	1,500	.....	....	1,380
Panhard	French	12–60°	340	4.52	115	6.69	170	1.48	1,750	....	....	1,950
Fiat	Italian	12–60°	400	4.72	120	5.91	150	1.25	2,300	1,500	....	2,260
Fiat	Italian	12–60°	650	6.69	170	8.27	210	1.24	1,550	1,550	4.95	2,140
W												
Napier lion	British	12–60°	450	5.50	140	5.13	130	0.933	2,000	1 317	5.55	1,708

TABLE 3.—*Continued*

Engine	Brake m.e.p., lb. per sq. in.	Torque, lb.-ft.	Valves per cylinder						Wt. of engine dry, lb.	Weight per cu. in. piston displacement, lb.	Piston displacement per h.p., cu. in.	Weight of engine per h.p., lb.
			Inlet			Exhaust						
			No.	Diam., in.	Lift, in.	No.	Diam., in.	Lift, in.				
Vee												
Curtiss, OX	105.0	349	1	1.909	0.344	1	1.909	0.388	385	0.778	5.40	4.2
Curtiss, VX	110.0	801	1	2.756	0.374	1	2.756	0.374	645	0.588	5.32	3.13
Curtiss, K12	115.0	875	2	1.625	.....	2	1.625	.....	680	0.591	3.06	1.81
Hall-Scott, A8	.....	.....	..	.....	.....	..	.....	.....	800	0.486	3.66	1.78
Hispano-Suiza, I	114.5	543	1	1.968	0.394	1	1.968	0.394	440	0.615	4.76	2.93
Hispano-Suiza, E	116.0	554	1	1.968	0.394	1	1.968	0.394	476	0.660	3.79	2.50
Hispano-Suiza, H	127.0	948	1	2.205	0.511	1	2.205	0.511	632	0.561	3.46	1.94
Liberty	119.0	1,300	1	2.500	0.438	1	2.500	0.375	845	0.512	3.91	2.00
Packard	120.0	590	1	1.750	0.438	1	1.750	0.375	542	0.730	4.14	3.02
Packard	125.0	925	1	1.750	0.438	1	1.750	0.375	733	0.655	3.97	2.60
Sturtevant	100.0	367	..	.....	.....	..	.....	.....	500	0.905	3.95	3.57
Thomas-Morse, 88	107.0	394	..	.....	.....	..	.....	.....	525	0.890	3.93	3.50
Duesenberg, V12	70.6	733	..	.....	.....	..	.....	.....	923	0.592	6.23	3.69
Duesenberg (geared) H16	104.0	2,330	1	2.72	0.562	2	1.97	0.500	1,575	0.464	4.23	1.97
Rolls Royce, Falcon III	112.0	645	1	1.750	0.444	1	1.750	0.440	723	0.835	3.21	2.68
Rolls Royce, Eagle VIII	124.0	1,040	1	2.000	0.485	1	2.000	0.480	933	0.755	3.54	2.60
Rolls Royce, Condor	130.0	1,900	2	.....	.....	2	.....	.....	1,350	0.632	3.56	2.25
Sunbeam, Arab 2	117.0	556	1	.....	0.354	2	.....	0.354	550	0.770	3.40	2.60
Sunbeam, Maori 4	110.0	686	2	.....	.....	2	.....	.....	.....	.....	3.41	....
Sunbeam, Cossack	125.0	919	2	1.614	0.354	2	1.614	0.350	1,118	1.000	3.18	3.19
Sunbeam, Manitou	79.5	788	..	.....	.....	..	.....	.....	845	0.565	4.97	2.82
Sunbeam, Matabele	113.0	1,050	..	.....	.....	..	.....	.....	1,000	0.715	3.50	2.25
Sunbeam, Sikh	115.0	3,000	..	.....	.....	..	.....	.....	.....	....	4.89	....
A. B. C	86.0	404	..	.....	.....	..	.....	.....	375	0.531	7.06	....
Galloway-Atlantic	126.0	1,925	..	.....	.....	..	.....	.....	1,150	0.500	4.18	2.09
Renault	.....	.....	..	.....	.....	..	.....	.....	800	0.558	3.59	2.00
Peugeot	124.0	568	..	.....	.....	..	.....	.....	.....	.....	3.44	....
Panhard	81.0	357	..	.....	.....	..	.....	.....	440	0.675	6.51	4.40
Panhard	120.0	1,020	..	.....	.....	..	.....	.....	.....	.....	3.78	....
Fiat	111.0	913	2	.....	.....	2	.....	.....	800	0.646	3.10	2.00
Fiat	94.4	2,200	2	2.62	0.468	2	2.62	0.442	1,739	0.495	5.40	2.67
W												
Napier Lion	122.0	1,180	2	.....	0.344	2	.....	0.354	850	0.582	3.25	1.89

TABLE 4.—DETAILED DIMENSIONS OF

Engine	Liberty		Packard	
Approximate horse power	200	400	180	270
Bore, inches	5.00	5.00	4.75	4.75
Stroke, inches	7.00	7.00	5.25	5.25
Number of cylinders	6	12	8	12
Stroke-bore ratio	1.40	1.40	1.105	1.105
Piston displacement per cylinder, cu. in.	137.4	137.4	93.0	93.0
Total piston displacement, cu. in.	824.5	1650.0	744.0	1116.4
Compression volume, cu. in.	31.1	31.1	23.17	20.4
Total volume of cylinder, cu. in.	168.5	168.5	116.2	113.4
Compression ratio	5.42	5.42	5.02	5.56
Per cent compression	18.46	18.46	24.9	18.0
Camshaft bearings, number on shaft	7	7	5	7
Diameter, inches	6–0.998 1–1.1235	6–0.998 1–1.1235	4–0.998 1–1.124	6–0.998 1–1.124
Inlet valves:				
Number per cylinder	1	1	1	1
Port diameter, inches	2.50	2.50	2.00	2.00
Lift, inches	7/16	7/16	7/16	7/16
Angle of seat	30°	30°	30°	30°
Total area of opening, square inches	3.202	3.202	2.61	2.61
Tappet clearance, inches	0.013–0.016	0.013–0.016	0.013–0.016	0.013–0.016
Exhaust valves:				
Number per cylinder	1	1	1	1
Port diameter, inches	2.50	2.50	2.00	2.00
Lift, inches	0.375	0.375	0.375	0.375
Angle of seat	30°	30°	30°	30°
Total area of opening, square inches	2.748	2.748	2.20	2.20
Tappet clearance, inches	0.019–0.021	0.019–0.021	0.019–0.021	0.019–0.021
Valve springs:				
Number per valve	2	2	2	2
Tension inlet (both springs):				
Valve open, pounds	62.2	62.2	63.5	63.5
Valve closed, pounds	50.0	50.0	47.5	47.5
Tension exhaust (both springs):				
Valve open, pounds	85.7	85.7	81.5	81.5
Valve closed, pounds	71.5	71.5	61.5	61.5
Internal spring tension:				
Valve closed, pounds	26.5	26.5	24.0	24.0
Valve open (inlet), pounds	32.9	32.9	35.0	35.0
Valve open (exhaust), pounds	31.8	31.8	35.0	35.0
Valve timing:				
Inlet:				
Opens, past top center	10°	10°	10°	10°
Closes, past bottom center	45°	45°	45°	45°
Exhaust:				
Opens, before bottom center	48°	48°	48°	48°
Closes, past top center	8°	8°	8°	8°
Piston:				
Type	Ribless	Ribless	Ribless	Ribless
Area of head, square inches	19.63	19.63	17.72	17.72
Material	Aluminum	Aluminum	Al. alloy	Al. alloy
Distance from center of pin to top of piston, inches	3.469	3.469	2.25	2.67
Length over all, inches	5.469	5.469	4.25	4.665
Length of bearing in cylinder	3.75	3.75	3.094	2.99
Clearance in cylinder:				
Top, inches	0.0395	0.0395	0.037	0.034
Bottom, inches	0.020	0.020	0.017	0.017

American and German Engines

Hispano-Suiza		Benz	Maybach	Austro-Daimler	Bassé-Selve	Mercedes
180	300	200	300	200	270	180
4.72	5.511	5.512	6.50	5.31	6.10	5.51
5.11	5.905	7.480	7.09	6.89	7.87	6.30
8	8	6	6	6	6	6
1.08	1.07	1.357	1.09	1.296	1.29	1.14
89.9	140.8	178.4	235.3	152.8	230.2	150.3
719.0	1126.0	1070.4	1410.0	916.8	1381.0	901.7
20.7	32.6	37.2	47.6	38.0	69.0	41.3
110.6	173.4	215.6	282.9	190.8	299.2	191.6
5.33	5.32	5.8	5.95	5.02	4.34	4.64
18.76	18.8	17.25	16.8	19.9	23.0	21.5
3	3	4	5	4		
1.342	1.344	0.984				
1	1	2	2	2	2	1
1.968	2.205	1.693	1.89	1.73	2.20	2.677
0.393	0.511	0.433	0.372	0.390	0.390	0.453
45°	45°	30°	30°	45°	45°	
1.894	2.79	2.214	2.33	2.12	2.72	3.81
0.078	0.030	0.016	0.012	0.01	....	0.017
1	1	2	2	2	2	7
1.968	2.205	1.693	1.89	1.73	2.20	2.677
0.393	0.511	0.433	0.368	0.40	0.390	0.453
45°	45°	30°	30°	45°	45°	
1.894	2.79	2.214	2.33	....	2.72	3.81
0.078	0.030	0.016	0.016	0.012	....	0.014
2	2	1	1	1	1	1
75.0	80.0	43.0	133.0			
44.5	42.0	25.5	101.8			
75.0	80.0	44.0	133.0			
44.5	42.0	24.0	101.0			
19.0						
30.0						
30.0						
10°	−10°	5°	−8°	−10°	....	0°
50°	62°	45°	35°	30°	....	40°
45°	62°	55°	33°	45°	....	40°
10°	29°	18°	7°	7°	....	10°
Ribless	Ribless	Ribbed	Ribless flat-top	Ribbed	Convex crown	Steel concave crown
17.53	23.82	23.82	33.15	22.2	29.2	23.84
Aluminum	Aluminum	Al. alloy	Cast iron	Aluminum	Aluminum	Cast iron
1.765	3.00	2.480	3.19	2.09		
4.375	5.12	4.843	5.944	4.35	5.11	
....	....	3.071				
0.040	0.040	0.025	0.029	0.039	0.020	
0.0165	0.020	0.008	0.009	0.016	0.01	

TABLE 4.—

Engine	Liberty		Packard	
Approximate horse power	200	400	180	270
Rings:				
Number per piston:				
Top	3	3	3	3
Bottom	0	0	0	0
Tension, pounds	11–15	11–15	9.0	9.0
Width, inches	0.248	0.248	0.217	0.217
Width of gap (ring in cylinder), inches	0.031	0.031	0.031	0.031
Pin:				
Length, inches	4.234	4.234	4.141	4.138
Diameter, inches	1.25	1.25	1.25	1.25
Connecting rods, *plain:*				
Length c. to c	12.0	12.0	9.0	9.0
Number of bolts	4	2	2	2
Diameter of bolts, inches	0.311	0.4365	0.437	0.437
Thread	5/16–24	7/16–20	7/16–20	7/16–20
Connecting rods, *forked:*				
Length c. to c	....	12.0	9.0	9.0
Number of bolts	....	4	4	4
Diameter, inches	....	0.3115	0.312	0.312
Thread	....	5/16–24	5/16–24	5/16–24
Rod-stroke ratio	1.714	1.714	1.714	1.714
Wristpin bearing:				
Length, inches	2.00	2.00	2.00	2.00
Diameter, inches	1.25	1.25	1.25	1.25
Material	Bronze babbitt lined	Bronze	Bronze	Bronze
Big end bearing, *forked:*				
Length, inches	....	2.25	2.492	2.492
Diameter, inches	....	2.375	2.125	2.125
Material	....	Bronze babbitt lined	Bronze babbitt lined	Bronze babbitt lined
Big end bearing, *plain:*				
Length, inches	2.25	....	1.125	1.125
Diameter, inches	2.375	....	2.631	2.631
Material	Bronze shell babbitt lined	....	Steel	Steel
Crankshaft bearings:				
Number	7	7	5	7
Length, inches	6–1.625 1–4.25	6–1.625 1–4.25	1–1.625 2–1.750 1–2.625 1–4.438	1–1.625 4–1.750 1–3.000 1–4.438
Diameter, inches	2.625	2.625	2.375	2.375
Carburetor:				
Name	Zenith	Zenith	Packard	Packard
Model	55 AS	U. S., No. 52		
Type	Single nozzle	Double annular nozzles	Double Venturi	Double Venturi
Diameter of flange, inches	2.165	2.047	2.0	2.0
Choke, inches	1.575	1.417	1.34	1.42
Jets, diameter:				
Main, inches	1.80 mm.	1.65 mm.	0.0730	0.1285
Compensating, inches	1.55 mm.	1.70 mm.	0.0700	0.0995
Idling well or pilot jet, inches	1.00 mm.	1.00 mm.	0.0785	0.1040

Continued

Hispano-Suiza		Benz	Maybach	Austro-Daimler	Bassé-Selve	Mercedes
180	300	200	300	200	270	180
			3 and one			
3	3	3	scraper	3	3	3
1	1	1	0	0	1	1
4.0	14.0	7.0				
0.0984	0.25	0.118	0.255	0.275	0.230	
0.011	0.018	0.072	0.055	0.019		
4.125	4.781	4.764	6.26	5.10		
1.181	1.375	1.299	1.496	1.100	1.37	
8.858	10.375	12.992	12.205	12.40	14.17	
2	2	4	4	4	4	
8.0 mm.	0.500	0.394	0.551	0.39	0.47	
Pitch,	....	Pitch,	....	Pitch,	Pitch,	
1.52 mm.		0.9 mm.		1.5 mm.	1.75 mm.	
8.858	10.375					
4	4					
12 mm.	0.375					
Pitch,						
1.25 mm.						
1.73	1.76	1.736	1.72	1.80	1.80	
2.203	2.250	2.795	3.66	2.64	3.14	
1.183	1.375	1.299	1.496	1.10	1.37	
Bronze	Bronze	Bronze	Cast iron	Phosphor-	Phosphor-	
			bush	bronze	bronze	
2.508	2.50					
1.97	2.125					
Bronze	Bronze					
babbitt	babbitt					
lined	lined					
1.25	1.125	2.441	2.893	2.63	3.03	
2.50	2.75	2.362	2.598	2.20	2.75	
Steel on	Steel on	Bronze	Bronze,	Bronze,	Bronze,	
bronze	bronze	babbitt lined	white metal	white metal	white metal	
5	5	7	7	7	7	7
1–3.86	1–4.34	1–1.772	2.638	1–2.20	1–2.36	
3–1.578	3–2.00	5–1.693	....	1–1.71	5–2.48	
1–Steel ball	1–Steel ball	1–2.402	....	5–1.97	1–3.74	
2.282	2.50	(1) –1.890	2.598	2.28	2.75	
		(2–7)–2.441				
Stromberg	Stromberg	Benz	Maybach	Austro-	Bassé-	Mercedes
NAD 4	NAD 6	BZ 3A–137		Daimler	Selve	
Double	Double	Barrel				
Venturi	Venturi	throttle	....	Dual	....	Twin jet dual
2.18	2.375	1.89				
1.50	1.812	1.654	....	0.945	1.96	0.945
0.0935	....	0.039	Variable	....	0.0102	1.473 mm.
0.1286	0.116	0.020	....	....	0.0046	0.558 mm.

TABLE 4.—

Engine	Liberty		Packard	
Approximate horse power	200	400	180	270
Ignition:				
Type	Battery	Battery	Battery	Battery
Manufactured by	Delco	Delco	Delco	Delco
Maximum spark advance, before top center	30°	30°	45°	45°
Maximum retard, after top center	10°	10°	10°	10°
Speed ignition drives (CS = crankshaft speed):				
Generator or magneto	1.0×CS	1.5×CS	2.0×CS	2.0×CS
Distributors	0.5×CS	0.5×CS	0.5×CS	0.5×CS
Firing order	1-5-3-6-2-4	1L-6R—5L-2R-3L-4R-6L-1R-2L-5R-4L-3R	1L-4R-3L-2R-4L-1R-2L-3R	1L-6R-5L-2R-3L-4R-6L-1R-2L-5R-4L-3R
Rotation, direction of (facing propeller)	Counter clock	Counter clock	Counter clock	Counter clock
Spark plugs:				
Number per cylinder	2	2	2	2
Location	Cyl. head	Cyl. head	Side of head	Side of head
Size, millimeters	18.0	18.0	18.0	18.0
Pitch, millimeters	1.5	1.5	1.5	1.5
Gap, inches	0.015–0.018	0.015–0.018	0.015	0.015
Oil pump speed	1.5 × CS	1.5 × CS	1.5 × CS	1.5 × CS
Water pump speed	1.5 × CS	1.5 × CS	1.5 × CS	1.5 × CS
Reciprocating and centrifugal weights:				
Piston, complete with rings and pin, pounds	4.7	4.9	3.9	9.2
Upper end connecting rod, pounds	1.1	1.3	1.1	1.1
Lower end forked connecting rod, pounds	....	4.4	3.3	3.3
Lower end plain connecting rod, pounds	3.7	1.9	1.4	1.4
Total weight of connecting rods complete, pounds:				
Forked	0.0	5.7	4.4	4.4
Plain	4.8	3.7	2.5	2.5
Total	4.8	8.9	6.9	6.9
Valve (assembled)	0.8	0.8	0.93	0.93
Loads from maximum explosion pressure, pounds per square inch:				
Assumed maximum explosion pressure	510	510	465	527
Total load on piston head, pounds	10,000	10,000	8,240	9,320
Load on wristpin	4,000	4,000	3,290	3,730
Load on crankpin	1,872	1,872	1,548	1,751
Load on main bearings	(1–5)1,173	(1–5)1,173	(1, 2, 4)1,066	(1, 2, 3, 5, 6) 1,207
	(6) 1,393	(6)1,393	(3)694	(4)682
	(7)364	(7)364	(5)402	(7)455

Continued

Hispano-Suiza		Benz	Maybach	Austro-Daimler	Bassé-Selve	Mercedes
180	300	200	300	200	270	180
Magneto Splitdorf	Magneto Splitdorf	Magneto Bosch	Magneto Bosch	Magneto Bosch	Magneto Bosch	Magneto Bosch
26°	26°	32°	38°	40°	....	30°
−4°	−4°	−5°				
CS	CS	1.5 × CS	1.5 × CS	1.5 × CS	1.5 × CS	1.5 × CS
CS	CS					
1L–4R–3L–2R–4L–1R–2L–3R	1L–4R–3L–2R–4L–1R–2L–3R	1–5–3–6–2–4	1–5–3–6–2–4	1–5–3–6–2–4	1–5–3–6–2–4	1–5–3–6–2–4
Counter Clock	Counter clock	Counter clock	Counter clock	Counter clock	....	Counter clock
2	2	2	2	2	2	2
Side of head	Side of head	Side of head	Cyl. head	Side of head	Side of head	Side of head
18.0	18.0	18.0				
1.5	1.5	1.5				
0.021	0.021	0.012				
1.2 × CS	0.933 × CS	0.737 × CS	0.5 × CS	0.667 × CS		
1.2 × CS	1.2 × CS	1.5 × CS	2.0 × CS	1.894 × CS	....	1.5 × CS
3.6	5.95	5.0	14.05	4.1	6.19	
1.0	1.25	1.4	3.30	1.6	2.25	
3.1	4.40					
1.5	1.80	4.4	5.62	3.18	6.75	
4.1	5.65					
2.5	3.05	5.8	8.92	4.84	9.00	
6.6	8.70	5.8	8.92	4.84	9.00	
0.9	0.95	0.60	0.881	0.50	0.75	
500	500	550				
8,768	11,920	13,120				
3,360	3,860	3,612				
1,878	2,250	2,273				
(1)498	(1)588	(1)1,959				
(2, 3, 4)745	(2, 3, 4) 1,192	(2, 3, 4, 5, 6) 1,586				
(5)4,384	(5)5,960	(7)1,118				

Table 5.—Weights in Pounds of Engine Parts

	Liberty		Packard		Hispano-Suiza		Austro-Daimler	Bassé-Selve	Benz	Maybach	Mercedes
Number of cylinders	6	12	8	12	8	8	6	6	6	6	6
Rated horse power	200	400	180	280	180	300	200	270	200	300	180
Cylinders complete with valves	115.2	241.2	124.6	189.9	152.3	210.0	131.3	232.7	185.5	217.2	131.2
Crankshaft assembly including thrust bearing	104.4	105.8	64.8	86.5	53.7	73.4	101.8	138.5	87.0	117.6	70.0
Upper crankcase assembly with bearings, studs, etc.	77.2	92.0	60.3	80.7	33.8	52.6	133.4	209.5	85.6	94.3	172.3
Lower crankcase assembly	69.3	70.0	48.3	61.7	44.3	52.8	73.5		75.0	96.6	
Valve gear assembly including drive shafts, housings and accessory drives	44.6	90.2	73.8	95.5	26.7	31.9	74.8	82.3	39.0	74.2	70.4
Pistons complete with pins and rings	28.1	58.8	31.1	50.4	28.2	48.2	25.1	37.1	30.1	84.3	41.1
Connecting rod assembly including bearings:											
Forked	.....	34.2	17.6	26.2	16.0	22.9					
Plain	28.8	19.2	8.7	15.0	9.7	12.6	29.0	54.0	35.0	49.1	30.0
Propeller hub assembly	22.1	26.6	18.6	18.6	18.7	21.7	11.3	29.4	17.3	21.0	12.5
Ignition assembly:											
Generator	12.5	11.6	12.1	12.1							
Storage battery	10.5	10.5	10.5	10.5							
Distributor, wires, conduit	13.5	19.8	15.0	18.2	5.2	6.3					
Spark plugs	2.2	4.4	2.9	4.1	2.9	2.9					
Switch and voltage regulator	2.4	2.4	2.1	2.1							
Magnetos, drive gear and housing	.....	.....	.....	.....	43.1	40.7					
Total	41.1	48.7	42.6	47.0	51.2	49.9	35.6	33.1	29.3	26.2	32.6
Carburetor system:											
Carburetor	11.7	19.0	8.7	9.5	7.1	10.7	24.1	4.0	18.7	32.0	16.7
Intake header assembly	11.2	26.4	17.3	25.0	7.5	11.7	8.5	4.0	6.0	11.1	10.0
Water pump and connections	11.5	16.9	10.5	13.2	5.6	10.3	14.2	8.0	8.4	8.5	11.0
Oil pump and connections	9.0	7.5	5.4	5.4	3.6	10.5	16.4	9.6	5.7	13.0	13.2
Fuel pump	.....	.....	.....	.....	1.3	.....	1.6	.....	3.9	3.5	4.8
Miscellaneous	6.2	.....	8.2	8.3	.....	.....	28.1	27.8	11.7	7.0	31.8
Self starter	.....	.....	.....	.....	.....	.....	.....	.....	.....	6.0	
Total dry weight	580.0	856.0	542.0	933.0	476.0	632.0	708.5	870.0	638.2	871.7	647.0
Weight of water in engine	21.0	45.0	25.7	41.0	44.0	58.0	.....	.....	19.0		

The weight per cubic inch of piston displacement per stroke is an excellent measure of the success of the designer in keeping down weight. The actual horse power developed depends on mean effective pressure and revolutions per minute and is affected by factors outside the engine proper: the mean effective pressure is determined largely by the fuel used and the carburetor and manifold resistance; the revolutions per minute are limited by the desirable propeller speed in ungeared engines.

Detailed dimensions of some of the most successful American and German engines are given in Table 4. Weights of engine parts are given in Table 5.

External over-all dimensions for selected engines are given in Table 6. The maximum width occurs at the crankcase level

Table 6.—Over-all Dimensions of Engines

Type	Name	Horse power	No. of cylinders	Dimensions, inches		
				Maximum		
				Length	Width	Height
Rotary	Clerget	130	9	36.22	40.15	40.15
	Gnome	100	9	24.75	37.5	37.5
	Le Rhone	80	9	36.1	37.25	37.25
Radial air-cooled	ABC Wasp	170	7	35.7	42.2	42.2
	ABC Dragon Fly	320	9	42.1	48.5	47.7
Vertical	Curtiss K6	200	6	63.0	22.37	39.25
	Liberty 6	200	6	67.25	19.0	43.0
	Beardmore	160	6	57.08	19.92	31.88
	Galloway	240	6	67.32	20.07	42.91
	Hall-Scott A5	125	6	63.38	18.5	41.25
	Hall-Scott A7	95	4	57.0	18.5	39.5
	Siddeley Puma	240	6	69.88	24.09	43.62
	Austro-Daimler	200	6	68.9	22.4	45.3
45-deg. Vee	Liberty	400	12	69.1	26.8	43.0
60-deg. Vee	Sunbeam Cossack	320	12	61.81	37.79	38.89
	Rolls-Royce, Eagle 8	360	12	75.98	42.52	48.03
	Rolls-Royce, Falcon 3	220	12	72.04	37.24	42.0
	Sunbeam Maori	250	12	55.41	35.46	33.85
	Packard	270	12	62.25	27.125	34.75
	Curtiss K12	400	12	68.3	27.9	40.1
90-deg. Vee	Sunbeam Arab L	200	8	43.5	31.7	35.5
	Curtiss OX	90	8	50.0	30.0	27.0
	Curtiss VX	160	8	67.38	45.5	35.92
	Hispano-Suiza	180	8	51.3	33.5	32.7

in vertical engines and at the cylinder tops in Vee engines. As the tops of the cylinders are often above the fuselage, the width of the fuselage is not necessarily controlled by the maximum width of the engine.

AMERICAN ENGINES

Liberty Engine.—The Liberty engine is constructed either as a six-cylinder vertical, or a 12-cylinder Vee with an included angle of 45 deg. It has built-up steel cylinders, overhead valves and camshaft, and battery ignition. The cylinder units are the same in both constructions. Detailed dimensions are given in Table 4; weights are given in Table 5. Longitudinal and transverse sections of the 12-cylinder engine are shown in Figs. 46 and 47. The performance of this engine is shown in Fig. 48; the full throttle curves are from tests with a dynamometer load and are carried up to 2,000 r.p.m.; the propeller load curves are from tests of the engine equipped with its proper propeller and mounted on a torque stand. With the propeller used the engine runs at about 1,700 r.p.m. at full throttle at ground level. Maximum power is obtained at about 1,850 r.p.m. and maximum economy at about 1,800 r.p.m. The brake mean effective pressure, mechanical efficiency, and manifold depression are shown in Figs. 27, 14, and 17 respectively. The gear trains for driving the camshafts and various accessories are shown in Figs. 49 and 50 for six- and 12-cylinder constructions respectively. Some of the details of this engine are discussed under the appropriate headings in Chapters VI and VII.

At the rated speed of 1,700 r.p.m. the six-cylinder engine develops 232 h.p., and the 12-cylinder engine 425 h.p. The diminution in horse power per cylinder results from the lower mean effective pressure (see Fig. 27), which apparently results from lower volumetric efficiency. The weight per horse power falls from 2.45 lb. for the six-cylinder to 1.99 lb. for the 12-cylinder engine.

Packard.—This engine is built both as an eight- and 12-cylinder engine, with an included Vee angle in both cases of 60 deg. It is very similar to the Liberty engine in its general features but has smaller bore and stroke, a different method of cylinder construction (see Fig. 92) and an underneath carburetor with induction pipes through the crankcase. Detailed dimensions are given in Table 4; weights are given in Table 5. The per-

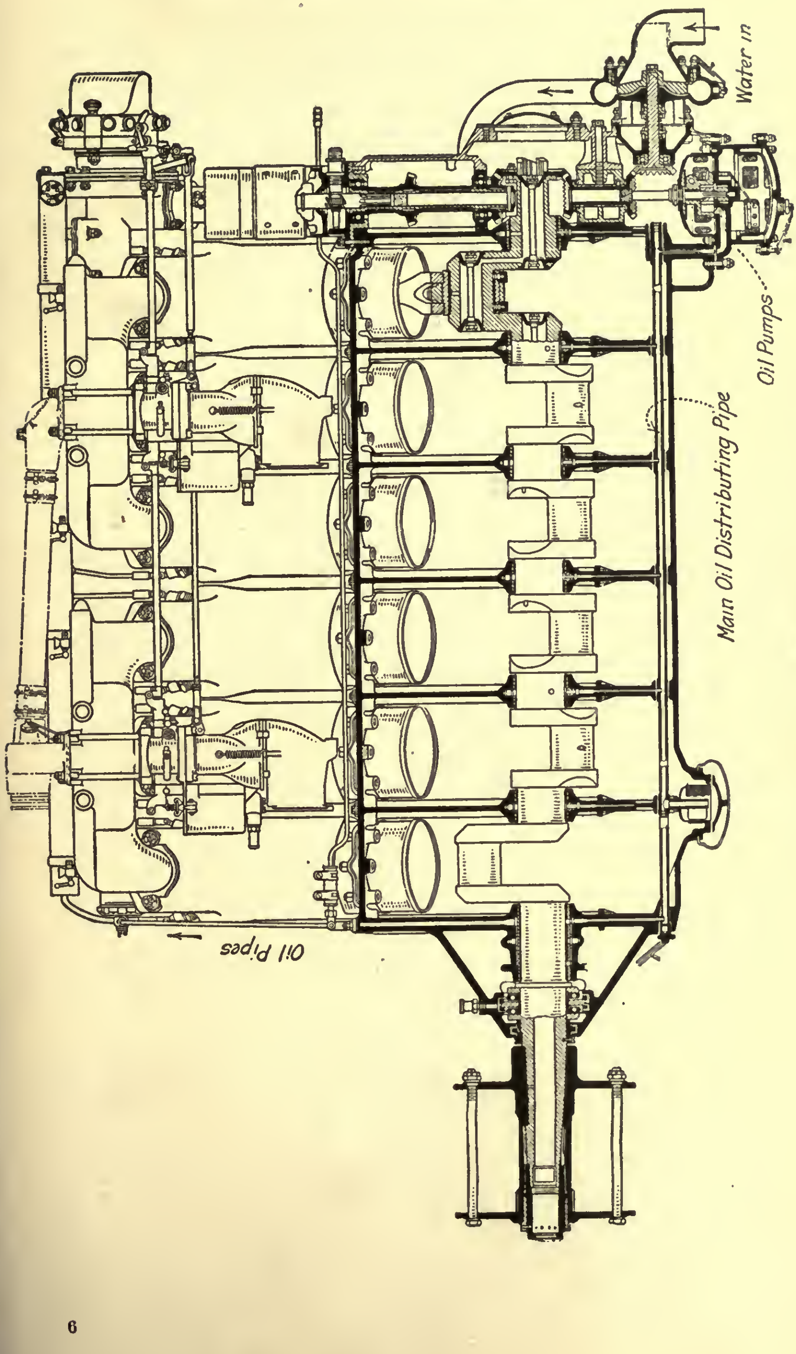


FIG. 46.—Longitudinal section of Liberty-12 engine.

formance of the 12-cylinder engine is shown in Fig. 50. With the propeller used in the tests the engine speed is 1,600 r.p.m. at full throttle; maximum power is at about 2,400 r.p.m., and maximum economy at about 1,800 r.p.m. The brake mean effective pressure and mechanical efficiency are shown in Figs. 27 and 14 respectively.

At the rated speed of 1,600 r.p.m. the eight-cylinder engine develops 192 h.p., and the 12-cylinder engine 280 h.p. The

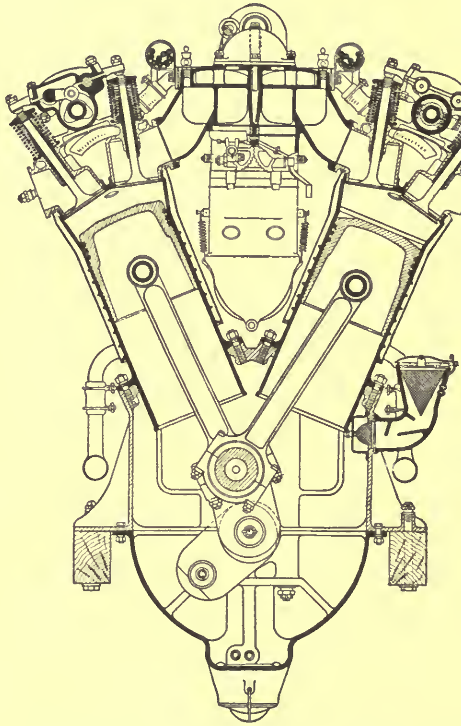


Fig. 47.—Transverse section of Liberty-12 engine.

horse power per cylinder remains constant. The weight per horse power falls from 2.82 lb. for the eight-cylinder engine to 2.62 lb. for the 12-cylinder engine.

Hispano-Suiza.—This engine is built in several sizes as an eight-cylinder Vee with included angle of 90 deg. In this country two cylinder sizes are built. The design is characterized by steel cylinder sleeves screwed into aluminum water jackets cast in blocks of four, by overhead valves and camshafts, and by

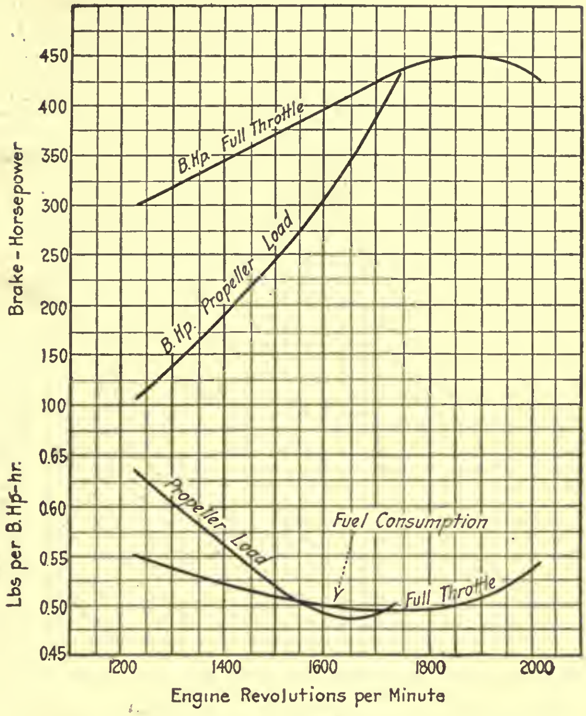


FIG. 48.—Performance curves of Liberty-12 engine.

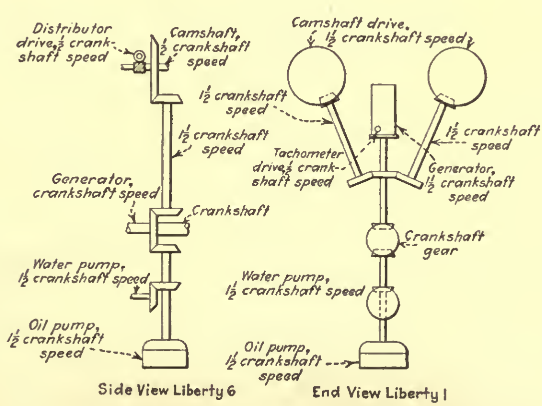


FIG. 49. FIG. 50.
Gear trains of Liberty engines.

the marine type of connecting rods. Magneto ignition is used. The models are known as E (180 h.p.) and H (300 h.p.). The E engine with a lower compression ratio (4.72) and running at 1,450 r.p.m. develops 150 h.p. and is known as model I.

Detailed dimensions of models E and H are given in Table 4; weights are given in Table 5. Transverse and longitudinal sections of model E are shown in Figs. 51 and 52. The performance of the 300-h.p. engine is shown in Fig. 53. With the propeller used in the tests the engine speed is 1,800 r.p.m. at full throttle; the maximum power is at about 2,300 r.p.m. and maxi-

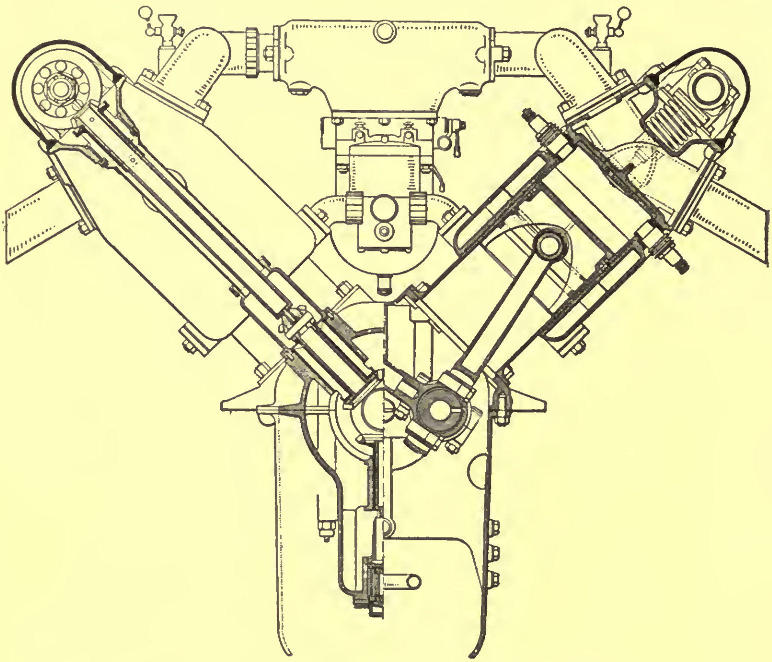


FIG. 51.—Transverse section of Hispano-Suiza 180.

mum economy at about 1,900 r.p.m. The brake mean effective pressure, mechanical efficiency, and manifold depression are shown in Figs. 27, 14 and 17, respectively. The gear trains for driving the camshafts and various accessories are shown in Fig. 54. Details of this engine are described later.

At the rated speed of 1,800 r.p.m. the smaller engine develops 187 h.p., the larger engine 327 h.p. The weight per horse power falls from 2.57 lb. for the smaller to 1.94 lb. for the larger engine.

Wright Engine.—This engine is a slightly modified Hispano-Suiza. The cylinder jacket is a little lower, cylinder heads

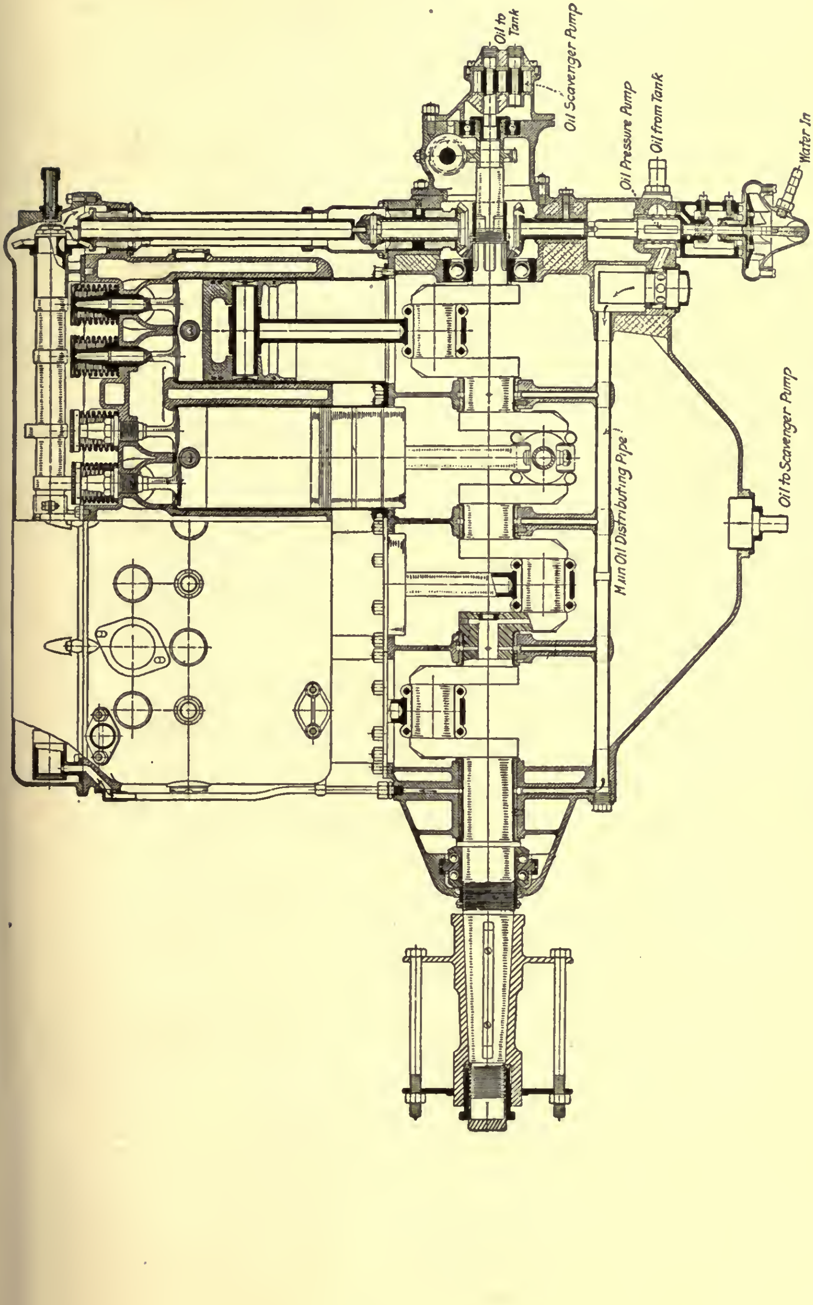


Fig. 52.—Longitudinal section of Hispano-Suiza 180.

thicker and the lubrication system is altered. These modifications make for greater durability.

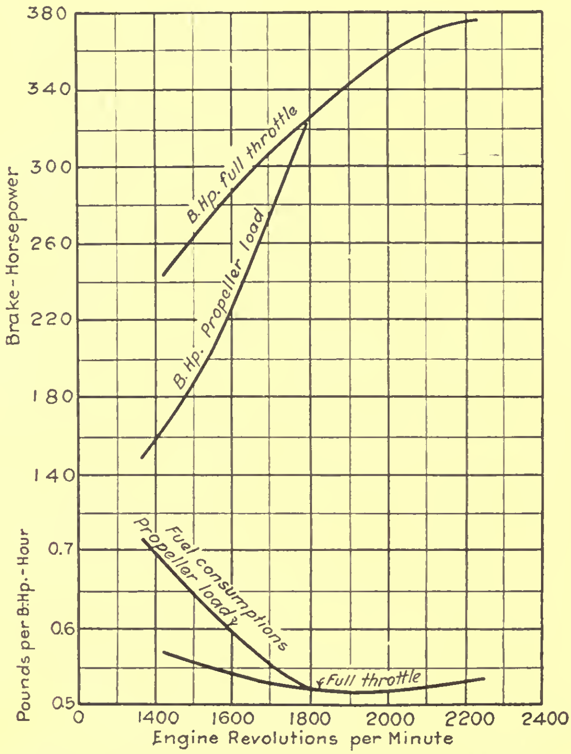


FIG. 53.—Performance curves of Hispano-Suiza 300.

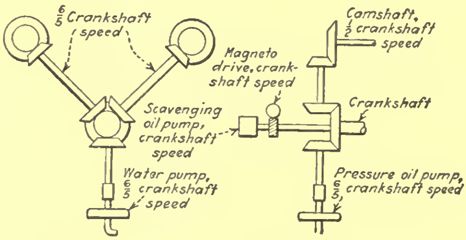


FIG. 54.—Gear train of Hispano-Suiza 180.

Curtiss.—The most recent Curtiss models are the K-6 and K-12; earlier models include the OX, VX, V-2. General data on these engines are given in Table 3. The K-6 is a six-cylinder

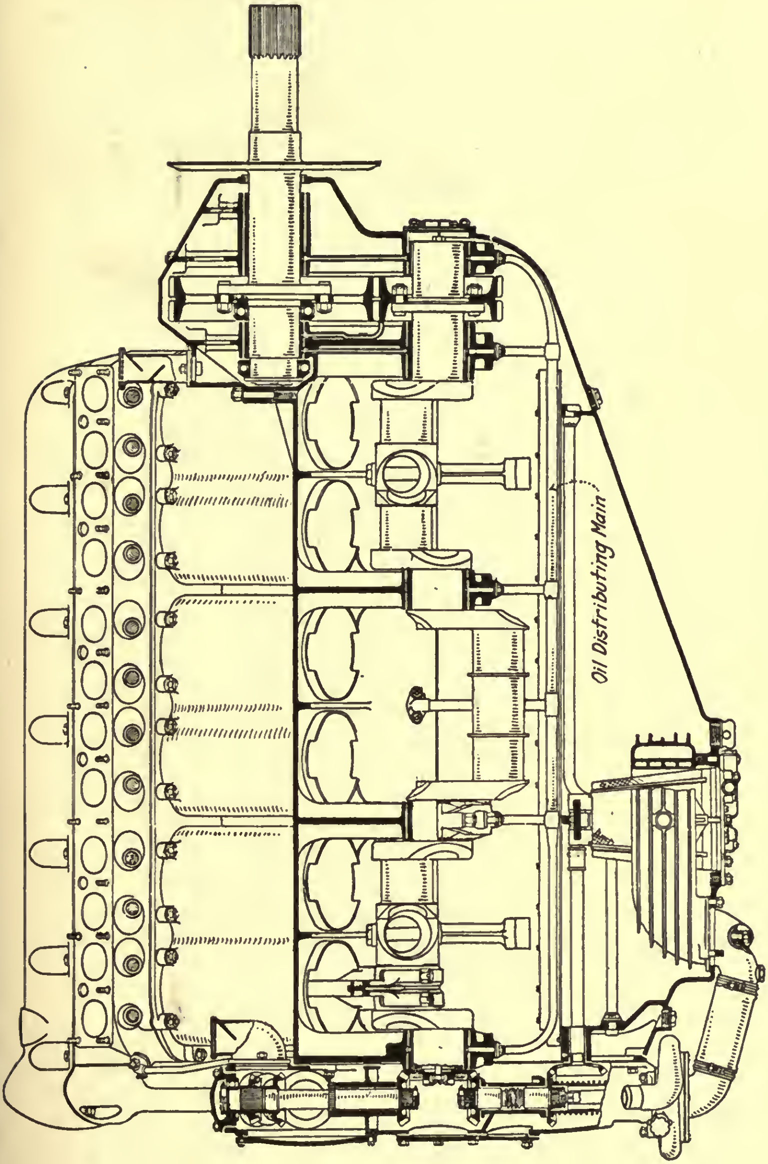


FIG. 55.—Longitudinal section of Curtiss K-12.

vertical engine; the K-12 is a 12-cylinder 60-deg. Vee engine with the same unit cylinder. These engines are characterized by the following features: cylinder blocks and top half of crankcase in one aluminum casting; cylinder heads in a separate aluminum casting bolted to the cylinder block; steel cylinder liners screwed into the cylinder heads; packed watertight joint between cylinder liners and aluminum blocks; separate overhead camshafts for the

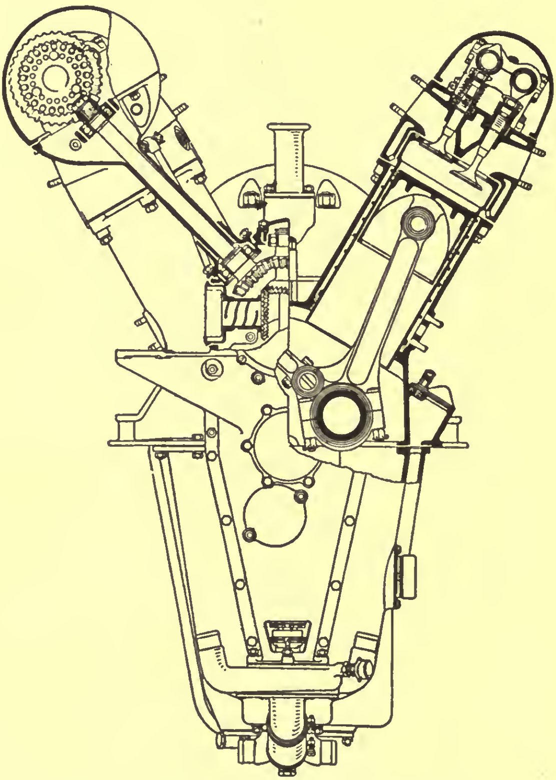


Fig. 56.—Transverse section of Curtiss K-12.

inlet and the exhaust valves; two inlet and two exhaust valves per cylinder; sliding cam follower; four-bearing crankshaft with counterweights; crankshaft bearings supported by partition walls of upper half of crankcase; ribbed pistons. The K-6 is direct drive; the K-12 has a herring-bone reduction gear with a ratio of 5:3, and an articulated connecting rod.

Longitudinal and transverse sections of the K-12 are shown in Figs. 55 and 56 respectively; a transverse section of K-6 is shown in Fig. 57.

At the rated speed of 2,250 r.p.m. (propeller speed 1,350 r.p.m.) the K-12 develops 385 h.p., corresponding to a mean effective pressure of 119 lb. per square inch; at 2,550 r.p.m. the mean effective pressure is 113 lb. per square inch and the engine develops 415 h.p. The weight dry without exhaust manifold is 665 lb., giving a weight of 1.73 lb. per horse power at rated speed. The K-6, weighing 420 lb., develops approximately 200 h.p., which gives a weight of 2.1 lb. per horse power. Performance curves of the K-12 are given in Fig. 58.

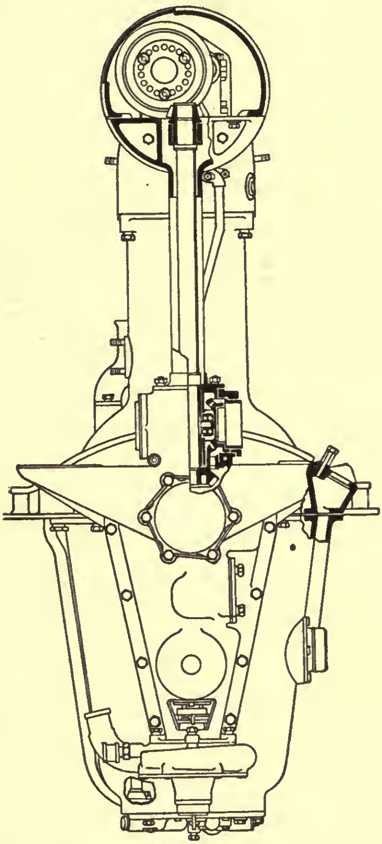


Fig. 57.—Transverse section of Curtiss K-6.

The OX model is an eight-cylinder 90-deg. Vee. It has separate cast-iron cylinders with brazed monel-metal jackets; the cylinders in the two rows are staggered with reference to one another so as to permit the use of side-by-side connecting rods on each of the four crankpins. The valves are inclined to the cylinder axis and are operated from a camshaft inside the crankcase. A push rod operates the exhaust valve through a rocker arm; a pull rod operates the inlet valve. The piston is of aluminum. The single carburetor is below the crankcase and has long intake pipes leading to the inlet manifold.

Longitudinal and transverse sections of this engine are shown in Figs. 59 and 60. Performance curves of the OX-5, as published by the builder, are given in Fig. 61.

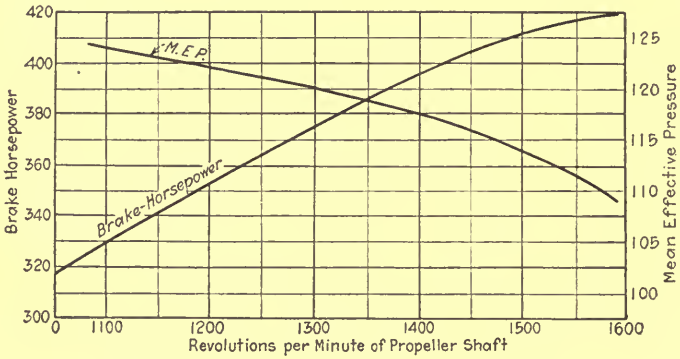


FIG. 58.—Performance curves of Curtiss K-12.

The V-2 model is similar in general arrangement to the OX but is of larger size. The cylinders are of steel with welded monel-metal jackets; the valve stems are parallel to the cylinder axes. Two carburetors are used and are located at the level of the bottom of the crankcase. When used for low level flying (up to 6,000 ft.) an aluminum liner is used between the cylinders and the crankcase; for high flights these shims may be taken out and the compression ratio thereby increased. A transverse view of this engine is shown in Fig. 62.

Hall-Scott.—General data on several models built by this company are given in Table 3. The latest model is the L-6; it has the same bore and stroke as the A-5, A-7 and A-8 models and also the Liberty engine. Longitudinal and transverse sections of

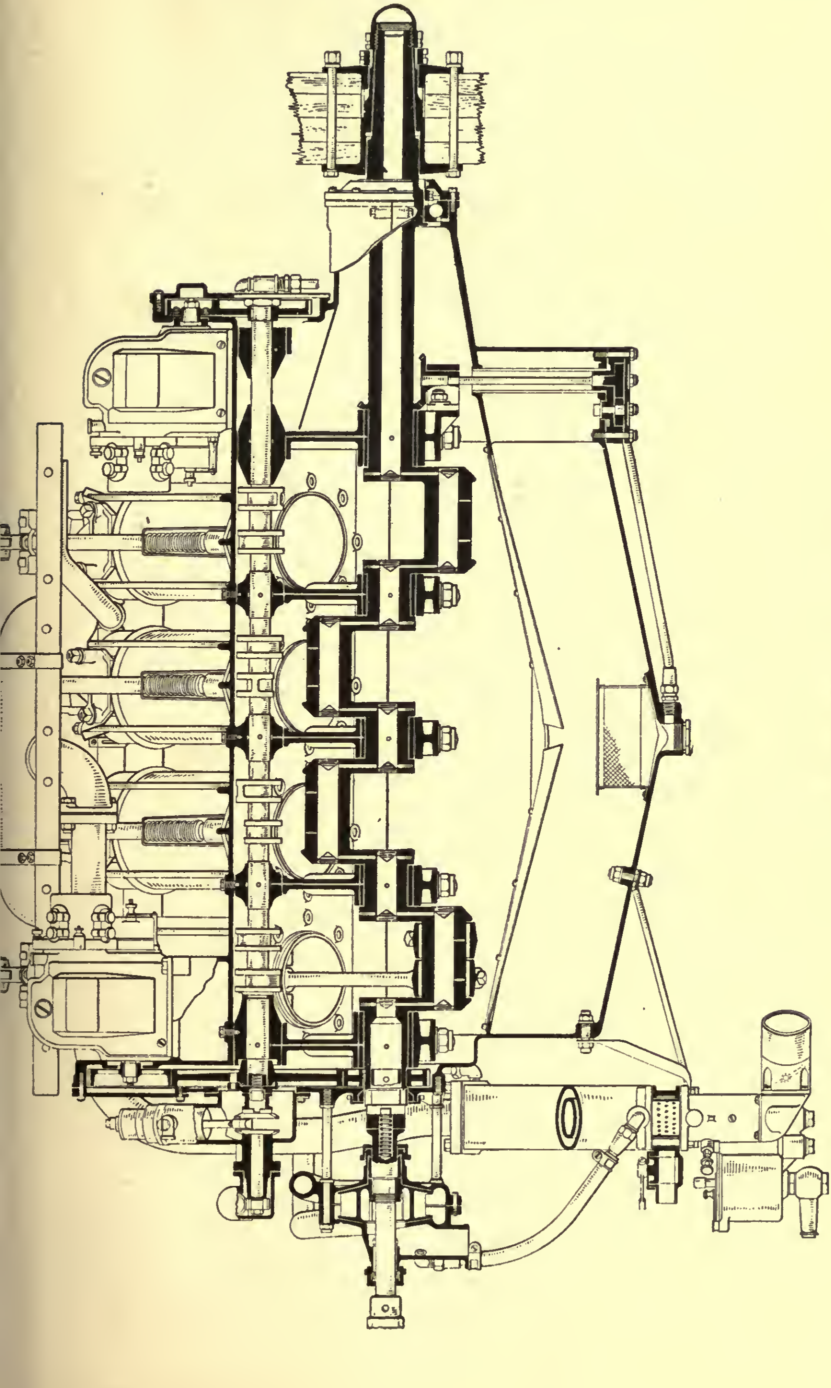


FIG. 59.—Longitudinal section of Curtiss OXX-3.

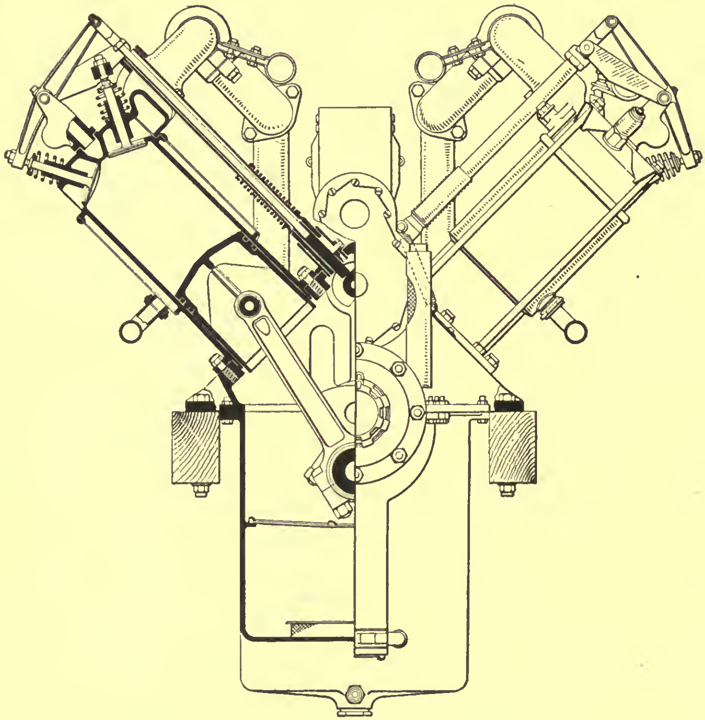


FIG. 60.—Transverse section of Curtiss OXX-3.

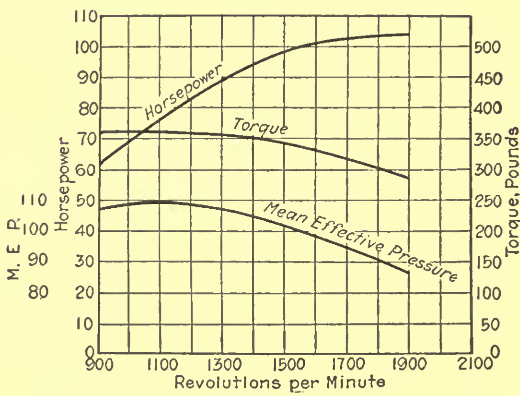


FIG. 61.—Performance curves of Curtiss OX-5.

this engine are shown in Figs. 63 and 64. It is very similar to the Liberty 6 in its cylinders, valves, pistons, camshaft, connecting rod and crankshaft. The crankshaft is supported entirely by the upper half of the crankcase, the bearing caps being bolted through the crankcase by through bolts which on the upper end act as cylinder hold-down bolts. The piston pin floats freely in

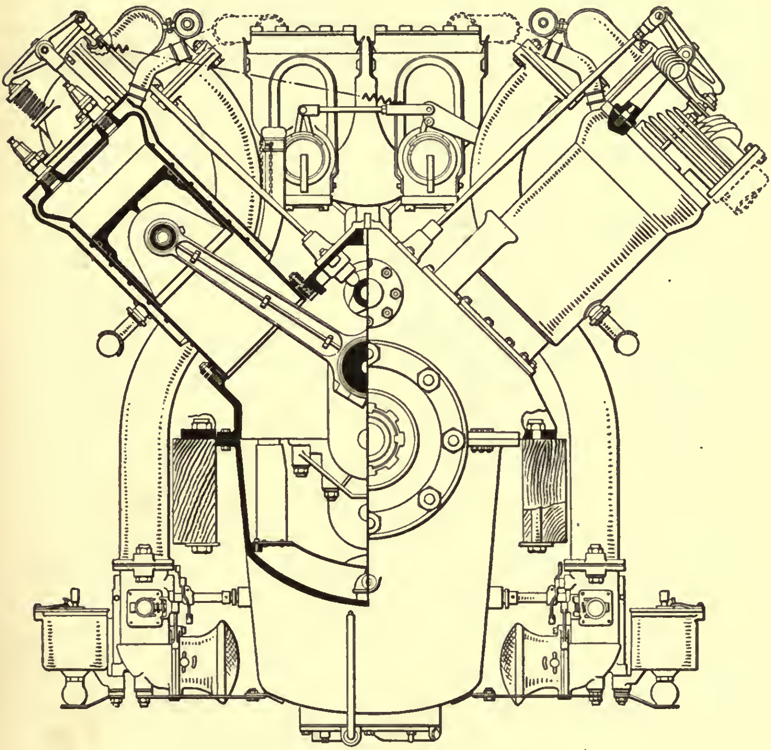


Fig. 62.—Transverse section of Curtiss V-2.

both the rod and piston. Carburetion is through two carburetors and hot-spot water-jacketed manifolds.

Performance curves of the four cylinder L-4, as published by the builder, are given in Fig. 65.

Bugatti.—The King-Bugatti engine, built in the United States, is a twin-vertical 16-cylinder engine with the two crankshafts geared to drive a common propeller shaft. It is a modifi-

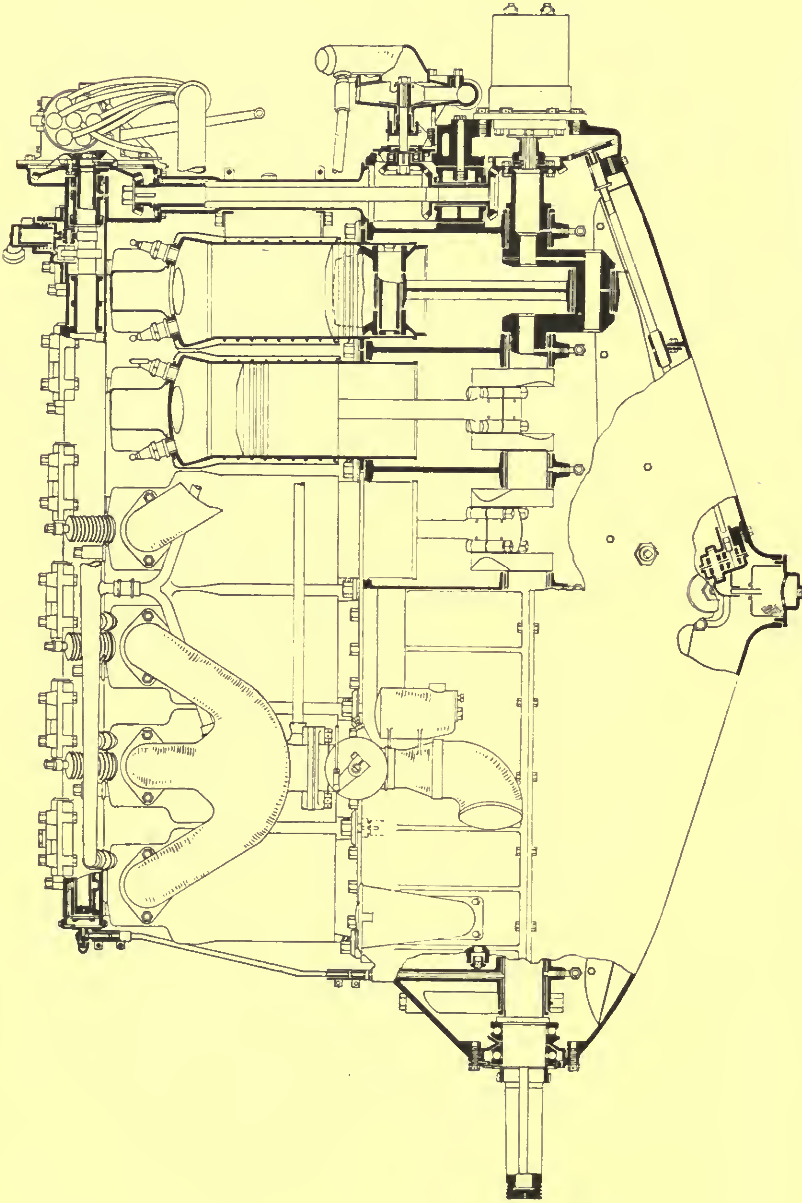


Fig. 63.—Longitudinal section of Hall-Scott L-6.

cation of the French Bugatti engine. It has a number of special features which may be seen in the sections, Figs. 66 and 67. The cylinders are of iron, cast in blocks of four, with integral water jackets except at the sides, which are covered with cast aluminum plates bolted to the cast iron. There are two inlet valves and one exhaust valve per cylinder. The crankshaft bearings are supported from the upper part of the cast aluminum crankcase; the caps, each of which supports two bearings, extend nearly the whole width of the crankcase. The eight-throw, nine-bearing crankshaft is in two halves connected at the center by a taper and key, shrunk and drawn up by a nut. Each half of the shaft forms a four-cylinder shaft with the throws all in one plane; the throws of the two sections are assembled at right angles. In

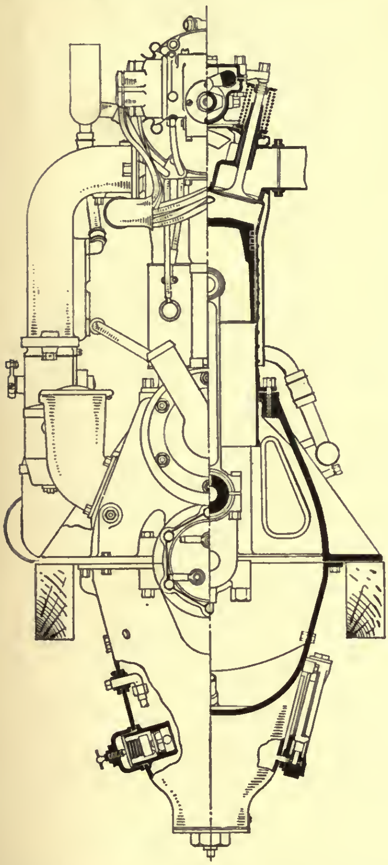


Fig. 64.—Transverse section of Hall-Scott L-6.

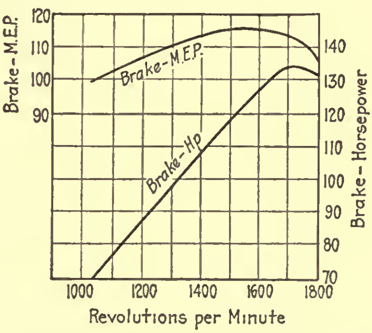


Fig. 65.—Performance curves of Hall-Scott L-4.

assembling the completed crankshafts in the crankcase they are placed in such relation to each other that if No. 8 throw left is on top dead center, No. 8 throw right will be 45 deg. past bottom dead center. All bearings, including the connecting rod bearings, are undercut and thereby shorten the over-all length of the engine by about 5 in. The valves are operated by a single camshaft for

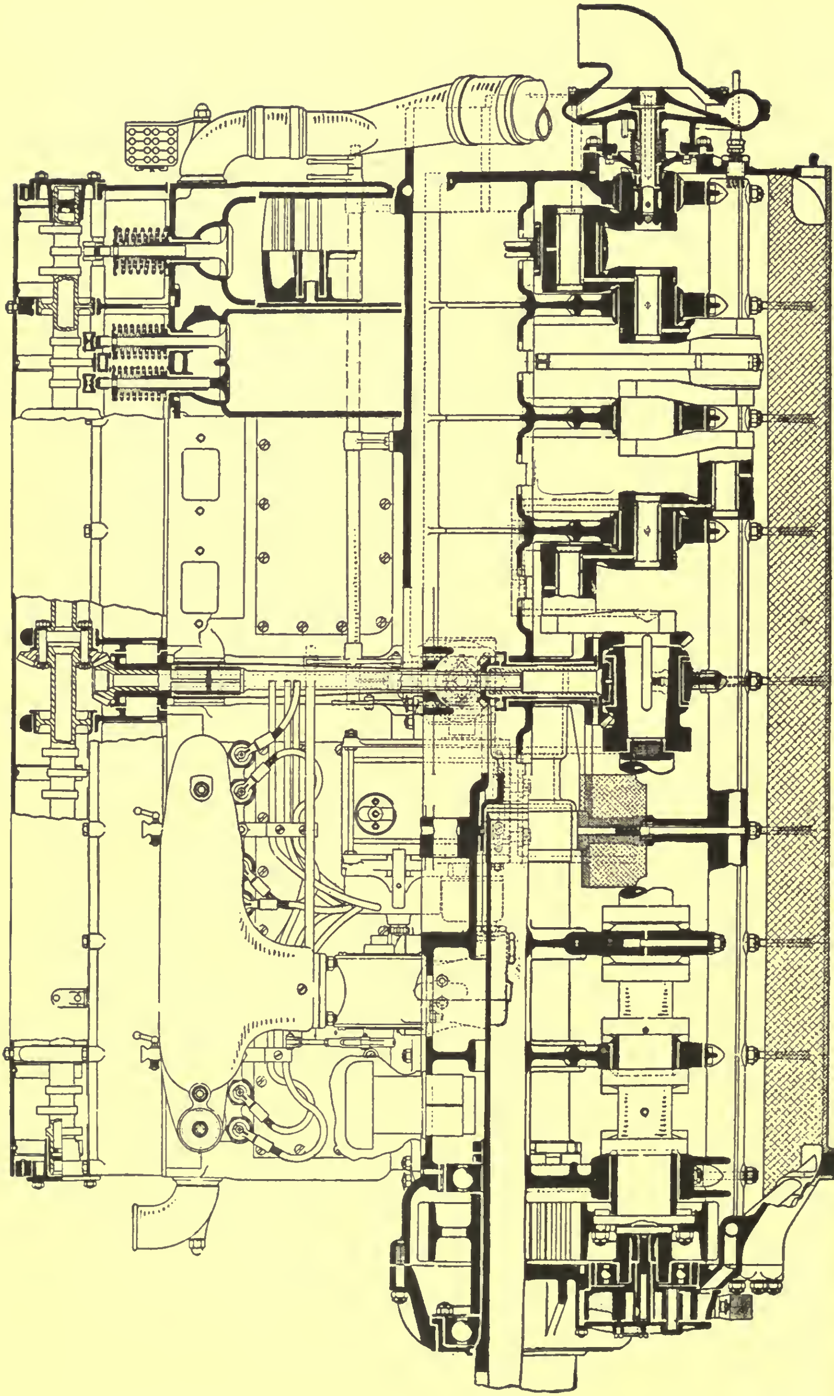


FIG. 66.—Longitudinal section of Bugatti engine.

each half of the engine. This camshaft is operated through a bevel gear, driven by a vertical shaft between the two cylinder blocks, which in turn is driven by a bevel gear on the crankshaft.

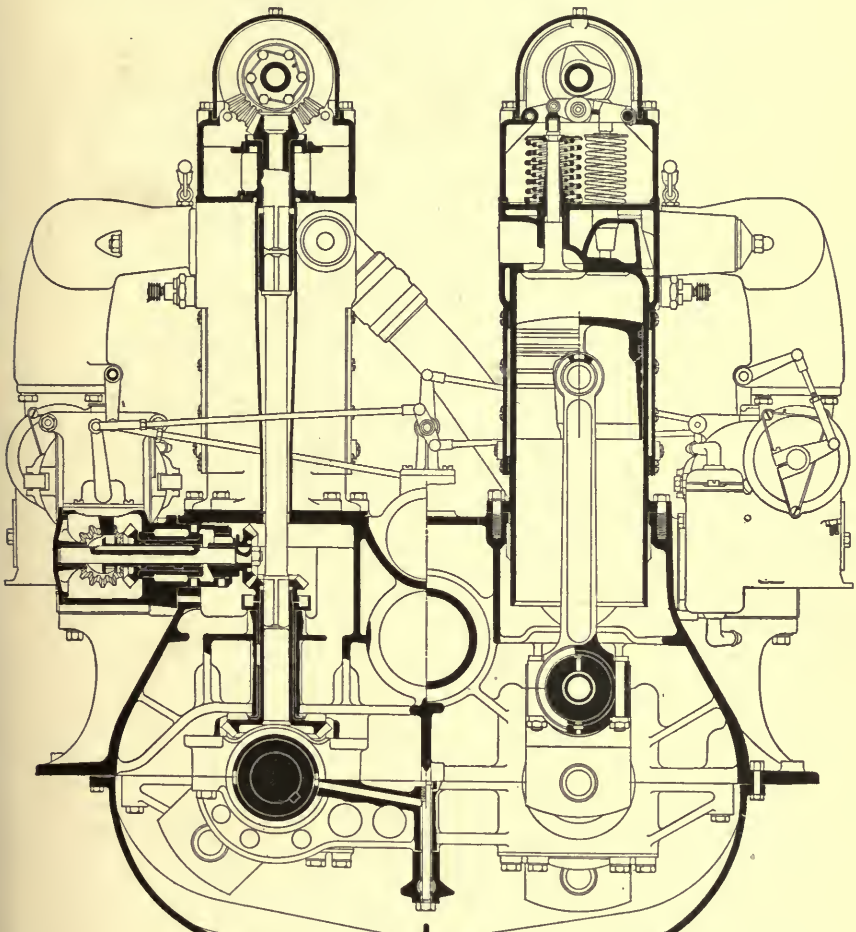


FIG. 67.—Transverse section of Bugatti engine.

The magneto drive (see Fig. 245) is from a bevel which is on this vertical shaft. The rocker arms are pivoted on rods running along each side of each camshaft housing. The cam follower is a hardened-steel roller; a smaller roller operates direct on a cap on top of the valve stem. The hollow propeller shaft runs at two-thirds engine shaft speed.

A performance curve of this engine both with dynamometer and propeller load is shown in Fig. 68. Maximum horse power is at an engine speed in excess of 2,400 r.p.m.

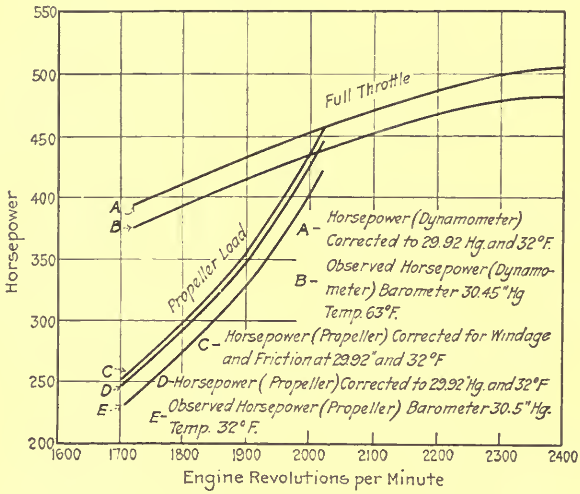


Fig. 68.—Performance curves of Bugatti engine.

ENGLISH ENGINES

Rolls-Royce.—The Rolls-Royce engines are interesting as probably representing the highest grade of design and manufacture in any country. General dimensions of the 12-cylinder, 60-deg. Vee "Eagle" and "Falcon" models are given in the following table, which shows how the power and efficiency of these engines have been improved by increasing the revolutions per minute and the compression ratio. Sectional views are given in Figs. 69 and 70. These engines have an epicyclic reduction gear concentric with the crankshaft (see p. 381); this gear is contained in a housing bolted to the front of the crankcase and is not shown in Fig. 69. The housing for the driving gears of the valve motion, magnetos, and other accessories is bolted to the rear of the crankcase. The upper part of the crankcase carries the bearings of the crankshaft; the lower part is merely a deep oil well. Each cylinder is fastened to the crankcase by four bolts. The engine is supported by arms which

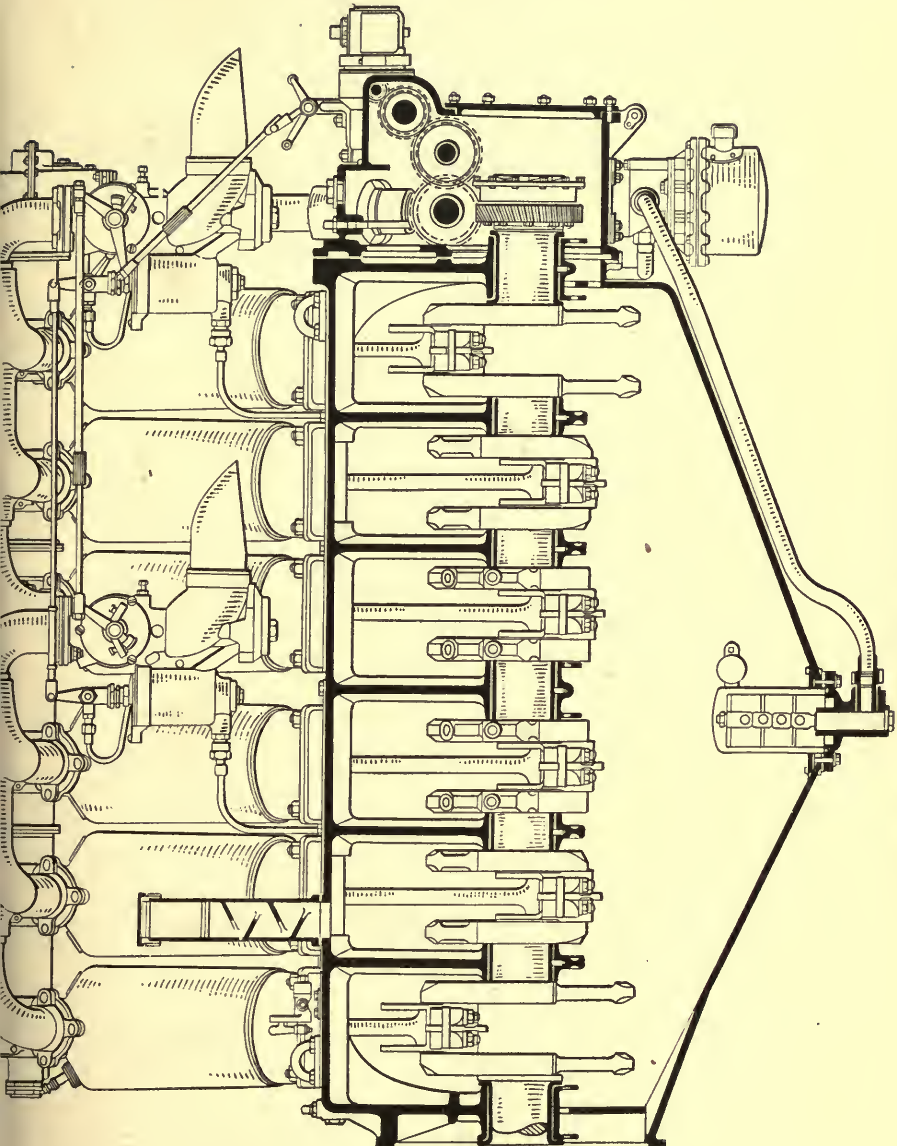


FIG. 69.—Longitudinal section of Rolls-Royce "Eagle."

are bolted to vertical surfaces at the sides of the crankcase, thus permitting the ready adaptation of the engine to the airplane without the need of special engine bearers in the fuselage.

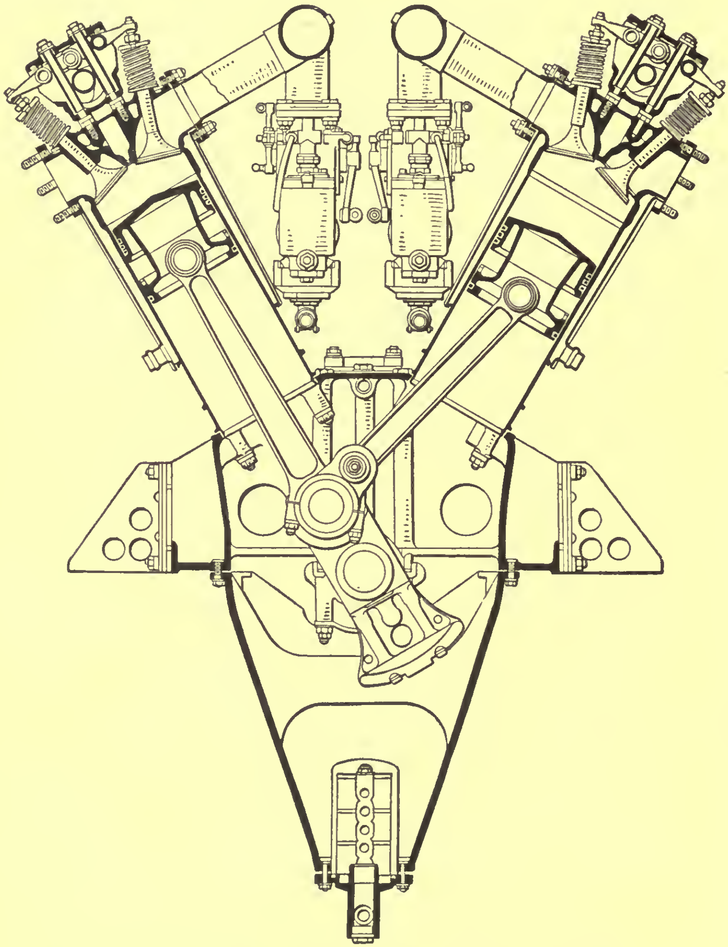


Fig. 70.—Transverse section of Rolls-Royce "Eagle."

The cylinders are of steel with valve fittings welded to short nipples in the cylinder head and with welded jackets. The aluminum piston is of special design without a skirt at the middle third of its length (see Fig. 98, p. 135); this design transfers the gas pressures directly to the piston bosses and reduces the

PERFORMANCE OF ROLLS-ROYCE ENGINES

Engine	B.h.p. at normal r.p.m.	Manufacturer's data			Bore, in.	Stroke, in.	Compression ratio	Piston speed, feet per minute	Brake m.e.p., lb. per sq. in.	Weight dry, lb.	Weight per b.h.p., lb.
		Engine r.p.m.	Propeller r.p.m.	Fuel consumption, lb. per b.h.p., hour							
	260	1,600	1,024	0.567	4.5	6.5	4.84	1,731	103.1	847	3.25
	(287)	(1,591)	(1,017)	(0.556)	4.5	6.5	(4.66)	1,724	(114.5)	847	(295)
	300	1,650	990	0.543	4.5	6.5	5.00	1,791	114.8	847	2.88
	(307)	(1,635)	(981)	(0.505)	4.5	6.5	(4.92)	1,771	(118.9)	847	(276)
	(309)	(1,616)	(969)	(0.503)	4.5	6.5	(4.80)	1,751	(121.0)	847	(274)
Eagle	320	1,800	1,080	0.485	4.5	6.5	5.10	1,948	113.1	882	2.76
Eagle	352	1,800	1,080	0.503	4.5	6.5	5.43	1,948	123.8	948	2.69
	200	1,700	1,005	0.565	4.00	5.75	4.50	1,633	106.0	694	3.47
Falcon	250	2,000	1,179	0.497	4.00	5.75	4.80	1,910	113.1	717	2.87
Falcon	264	2,000	1,179	0.485	4.00	5.75	5.00	1,910	119.5	728	2.76

bending stress in the piston head. The piston pin is close to the lower edge of the piston.

The valves are of tulip style (see Fig. 115, page 157), with bored stems. Inlet and exhaust valves are interchangeable. A hardened contact button is set in the end of the valve stem. The crankshaft is provided with balance weights of cast steel bolted to each crankarm; these have increased the weight of the engine but have reduced the pressures on the main bearings. The forward end of the crankshaft carries a flange to which is secured

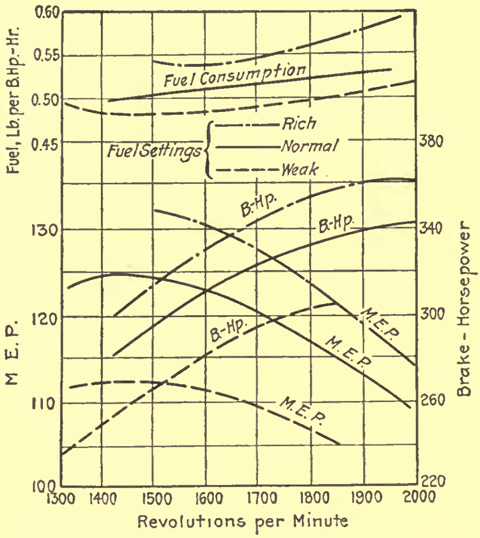


FIG. 71.—Performance curves of Rolls-Royce "Eagle."

the internally-toothed reduction-gear wheel (see Fig. 296). A stud pressed into the bore of the crankshaft at the forward end serves as pilot for the spider of the planetary pinions. The connecting rods are of the articulated type, the secondary rods working on pins which are clamped into lugs formed on both sides of the main connecting rod.

The camshaft drive is from a worm on the crankshaft, which is connected to the shaft through a serrated hub joint, and a spring coupling clamped by means of a disc brake; this device protects the drive against rotary vibration. The worm drives a cross shaft which has bevel wheels at its ends driving the inclined intermediate shafts going to the camshafts. The magnetos are driven from the cross shaft.

Performance curves for the "Eagle" with various fuel valve settings are given in Fig. 71.

Napier "Lion."—The Napier "Lion" is the only W or "arrow" type of airplane engine that is definitely successful. Details of

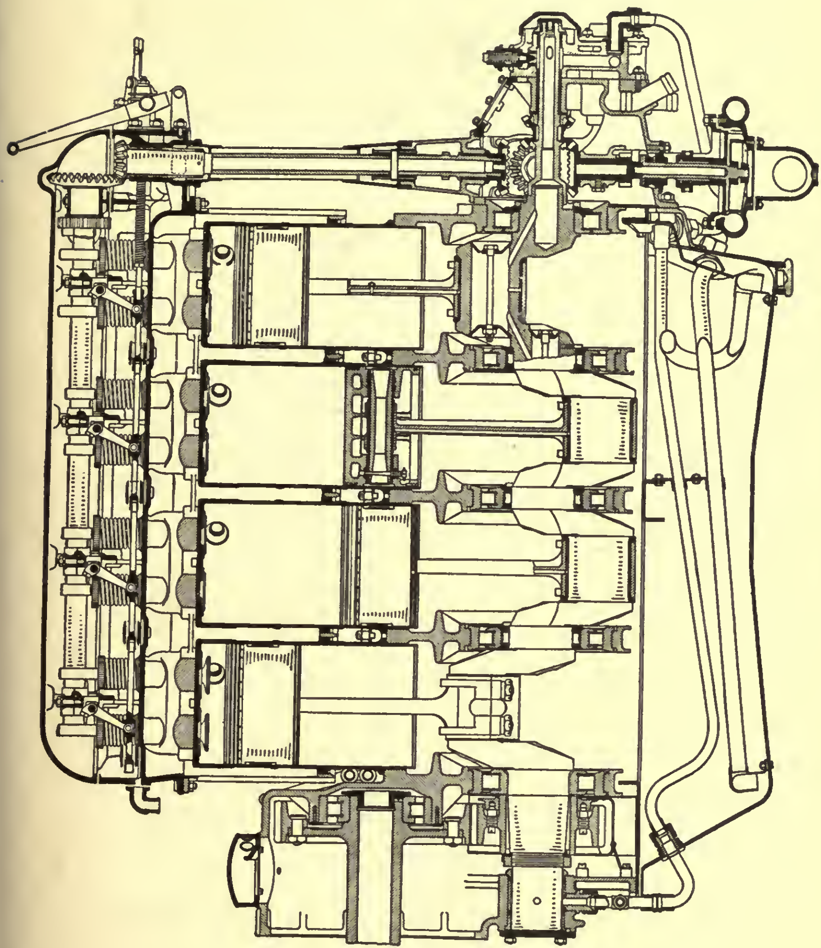


Fig. 72.—Longitudinal section of Napier "Lion."

this engine are shown in Figs. 72 and 73; dimensions in Table 3. It has three blocks, of four cylinders each, mounted on a single crank case, with an angle of 60 deg. between the rows. This engine is probably the lightest of all the successful water-cooled engines;

it weighs 1.86 lb. per horse power, dry, and 2.51 lb. per horse power with its jackets full of water.

The cylinders are $5\frac{1}{2}$-in. bore and $5\frac{1}{8}$-in. stroke which is an unusually low stroke-bore ratio and makes for small over-all height and width. Each block of cylinders is built up and consists of

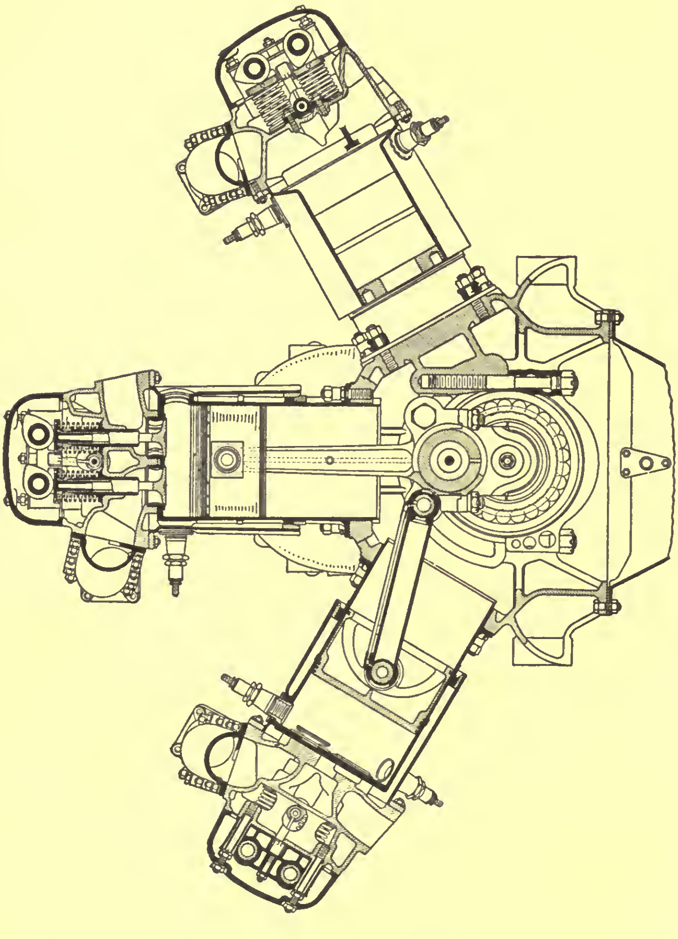


Fig. 73.—Transverse section of Napier "Lion."

four steel liners with sheet-steel water jackets. The cylinders of each block are secured to an aluminum head casting to which they are fastened by valve seats, which pass through the crown of each cylinder and screw into the head casting. The head casting carries camshafts and their bearings and drives, and contains the inlet and exhaust ports and passages. Each cylinder has two inlet and two exhaust valves, the latter being on the outside of the inclined blocks. The valve guides are bronze and a tight fit in the cylinder head. The inlet and exhaust camshafts operate the valves directly without intermediate rockers or plungers; each valve has an adjustable tappet head. One of each pair of camshafts is driven by bevel gears on the inclined and vertical shafts leading from the distribution gearing; the other camshaft is driven by spur gearing from the first shaft. The crankshaft has four throws and is supported on five roller bearings; the roller bearings at the front and rear are fitted direct to the shaft; the three intermediate ones have large inner races, which permit the bearings as a whole to be threaded over the crank webs and which are mounted on split bushings keyed to the shaft. The connecting rod assembly consists of a master rod and two side rods carried on lugs integral with the big end of the master rod. The pistons are unusually shallow, having a depth of only $3\frac{3}{8}$ in. The firing order is:

Propeller End

7	2	9
4	11	6
10	5	12
1	8	3

The propeller shaft is mounted on roller bearings with double ball-bearing thrust block. The ratio of propeller to engine speed is 29 to 44. The performance curve for this engine is given in Fig. 74. The brake horse power is still increasing rapidly at 2,100 r.p.m.; the mean effective pressure is a maximum at 1,800 r.p.m.

Siddeley "Puma."—This engine is a typical six-cylinder vertical engine of about 250 h.p. at 1,400 r.p.m. Cross-section views are given in Figs. 8 and 9; general dimensions in Table 3. The cylinder construction consists of steel liners in aluminum heads and jackets. The heads are cast in sets of three and incorporate the valve ports and head jackets. The barrel jackets are also cast in sets of three and are bolted to the heads. The steel liners are screwed cold into the heads which are heated to about 300°C. The water joint at the lower end of each jacket is made by a screwed gland compressing a rubber ring. The three liners have to be trued up by surface grinding after assembly into a unit. The inlet and exhaust valve seats are of phosphor bronze expanded into the aluminum. The difficulties which others have met in using this material at very high temperatures have been overcome by using an exceptionally hard-chilled alloy. There are two exhaust valves and one inlet valve per cylinder. The cooling water enters the cylinder head at two places; one is in direct communication with the water space; the other connects with an aluminum tube which runs inside the full length of the jacket and directs comparatively cool water on to the hottest places. The pistons, connecting rods and crankshaft are of

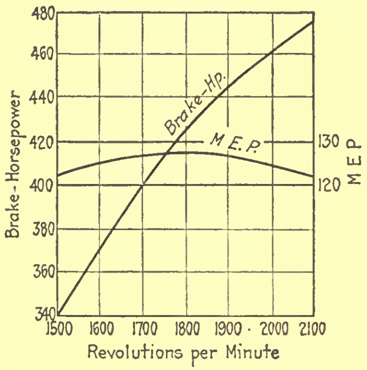


FIG. 74.—Performance curves of Napier "Lion."

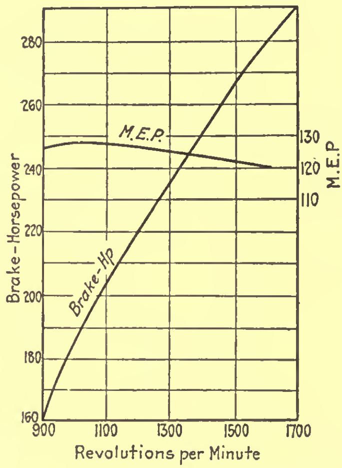


FIG. 75.—Performance curves of Siddeley "Puma."

conventional form. A roller bearing and double-thrust ball bearing are at the propeller end of the crankshaft; the five intermediate shaft bearings are supported by the upper half of the crankcase. The bearing caps are of aluminum reinforced with webbed plates of steel; the caps of the two outer journals are integral with the lower half of the crank case.

The camshaft is placed off-center and directly over the exhaust valves. It is made of steel tubing with cams pinned in position, an arrangement making for easy and cheap replacement of worn cams. The exhaust cams act directly on adjustable tappet heads on the valve stems; the inlet cams act on short drop-forged rockers without rollers. The valves are trumpet-shaped and give good flow lines.

A performance curve for this engine is shown in Fig. 75; maximum mean effective pressure is developed at about 1,000 r.p.m.; maximum horse power well above 1,700 r.p.m.

ITALIAN ENGINES

Fiat.—The Fiat 650-h.p. is one of the most powerful aircraft engines in use at present. It is a 12-cylinder 60-deg. Vee engine of unusually large cylinder dimensions, 170 by 210 mm. A longitudinal section of this engine is given in Fig. 76; general dimensions are given in Table 3.

The cylinders are of built-up steel construction with the cylinder heads integral with the barrels and with welded water jackets. Two inlet and two exhaust valves are located in the head of each cylinder at an angle of 25 deg. to the cylinder axis. The valve stems work in phosphor-bronze bushings. Each pair of valves is closed by duplex springs located between the valve stems (Fig. 134). The camshaft for each row of cylinders is in two parts which are driven by bevel gears from the center of the length of the engine. A tubular lay shaft, on ball bearings, is mounted in the center of the crankcase in the top of the Vee and is driven by spur gearing from the rear end of the crankshaft; it extends only to the center of the engine where it is fitted with a bevel gear driving the two inclined shafts which operate the camshafts. The connecting rods are forked type, the forked rods being fitted with bronze bearing shells with white-metal liners; the center rod has a case-hardened steel liner running on the outside of the bronze shell of the forked rod. The crankshaft is of conventional design; the main bearings are held between the two halves of the crankcase, that is, the lower half of the crankcase has the bottom halves of the bearing housings cast integrally with it. The water pump is below the engine at the middle of its length; the oil pumps are below the two ends of the lower crankcase. Water and oil pumps are driven from a lay shaft, at the bottom of the lower crankcase, which receives its motion from the crankshaft through spur gears at the rear of the engine. The induction manifolds are of sheet copper with steel flanges brazed at the joints. All distribution gears and driving shafts are mounted on ball bearings. There are four spark plugs per cylinder receiving current from four 12-cylinder magnetos. The normal output of the engine is 650 h.p. at 1,500 r.p.m.; the maximum is 720 b.h.p. at 1,700 r.p.m.

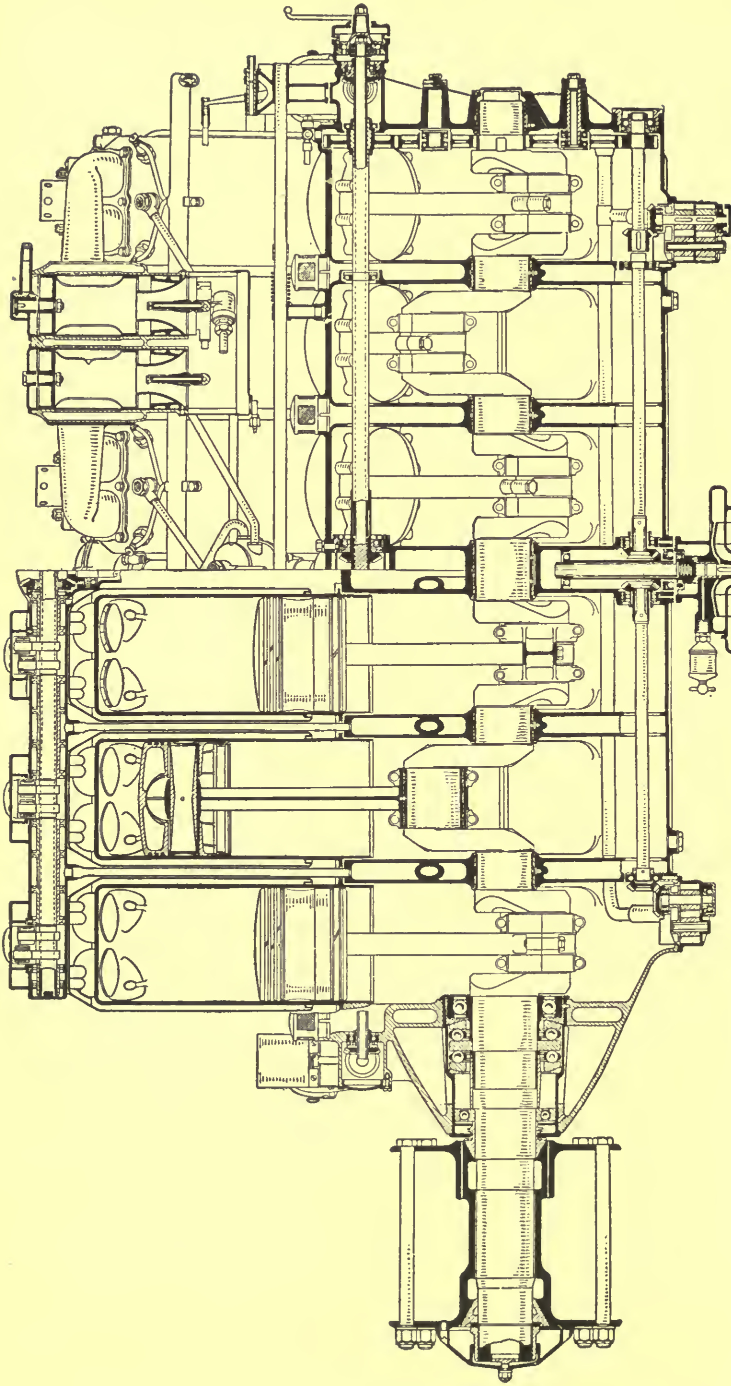


FIG. 76.—Longitudinal section of Fiat 650.

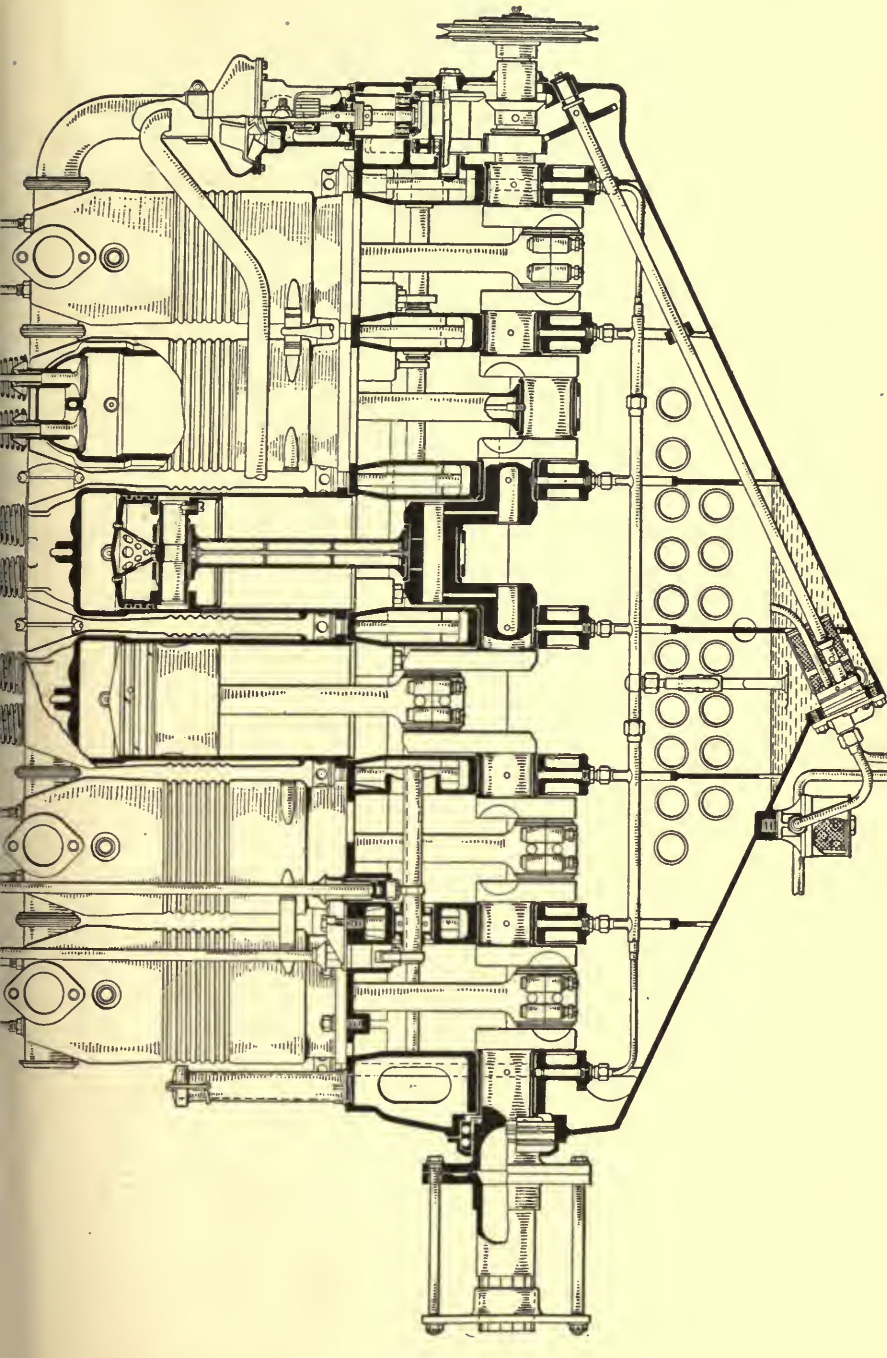


Fig. 77.—Longitudinal section of Benz 230.

GERMAN ENGINES

Benz.—The Benz 230-h.p. engine has six vertical water-cooled cylinders 140 by 190 mm. Longitudinal and transverse sections of this engine are given in Figs. 77 and 78; dimensions in Table 4.

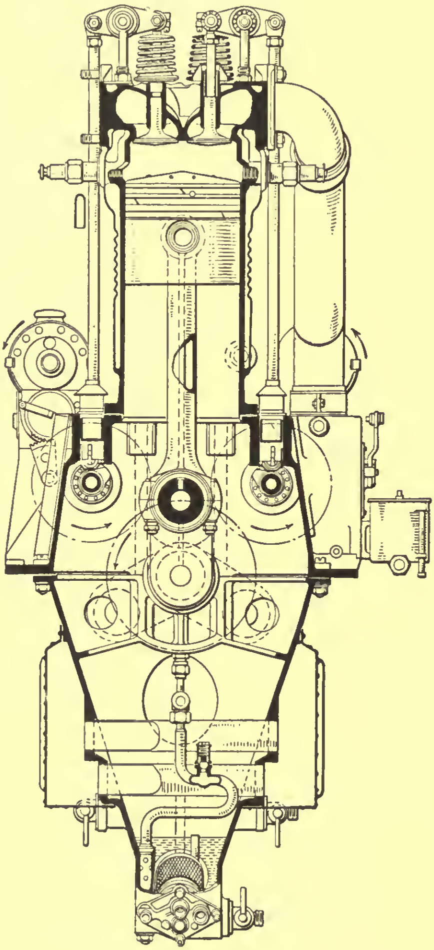


Fig. 78.—Transverse section of Benz 230.

The cylinders are of cast iron with pressed-steel water jackets around the barrels; they are bolted to the crankcase by long studs which pass through the upper crankcase and are screwed into the bottom halves of the crank bearing housings which are cast integral with the lower half of the crankcase. The pistons are of cast iron with the heads supported by conical steel forgings riveted and welded to the piston crown and bearing on the piston pins through slots cut in the small ends of the connecting rods; by this construction the greater part of the force of the explosion is transmitted directly to the connecting rod. The connecting rods are tubular with internal oil pipes.

Two inlet and two exhaust valves are arranged vertically on each cylinder head and are operated through rockers mounted on ball bearings. The rockers actuate the valves through rollers mounted on eccentric bolts which permit a fine adjustment for the

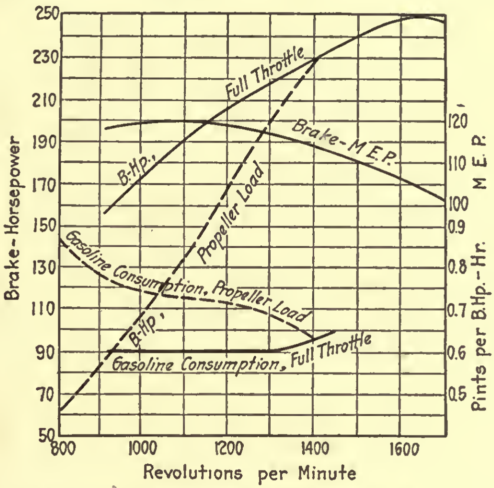


FIG. 79.—Performance curves of Benz 230.

tappet clearance. There are separate camshafts for the two sets of valves; these shafts are of steel tubing and are inside the crankcase. They are driven by spur gears from the crankshaft through an intermediate gear. The push rods have hemispherical ends at the bottoms which work in steel cups inside the hollow tappets. The tappets are slightly offset from the camshaft centers and carry steel rollers.

The upper half of the crankcase has transverse air passages cast in the webs which serve to cool the crankcase and to supply warm air to the carburetors. The lower half is cooled by transverse aluminum tubes, the air being led into one side by a large louvered cowl.

Performance data on this engine are given in Fig. 79.

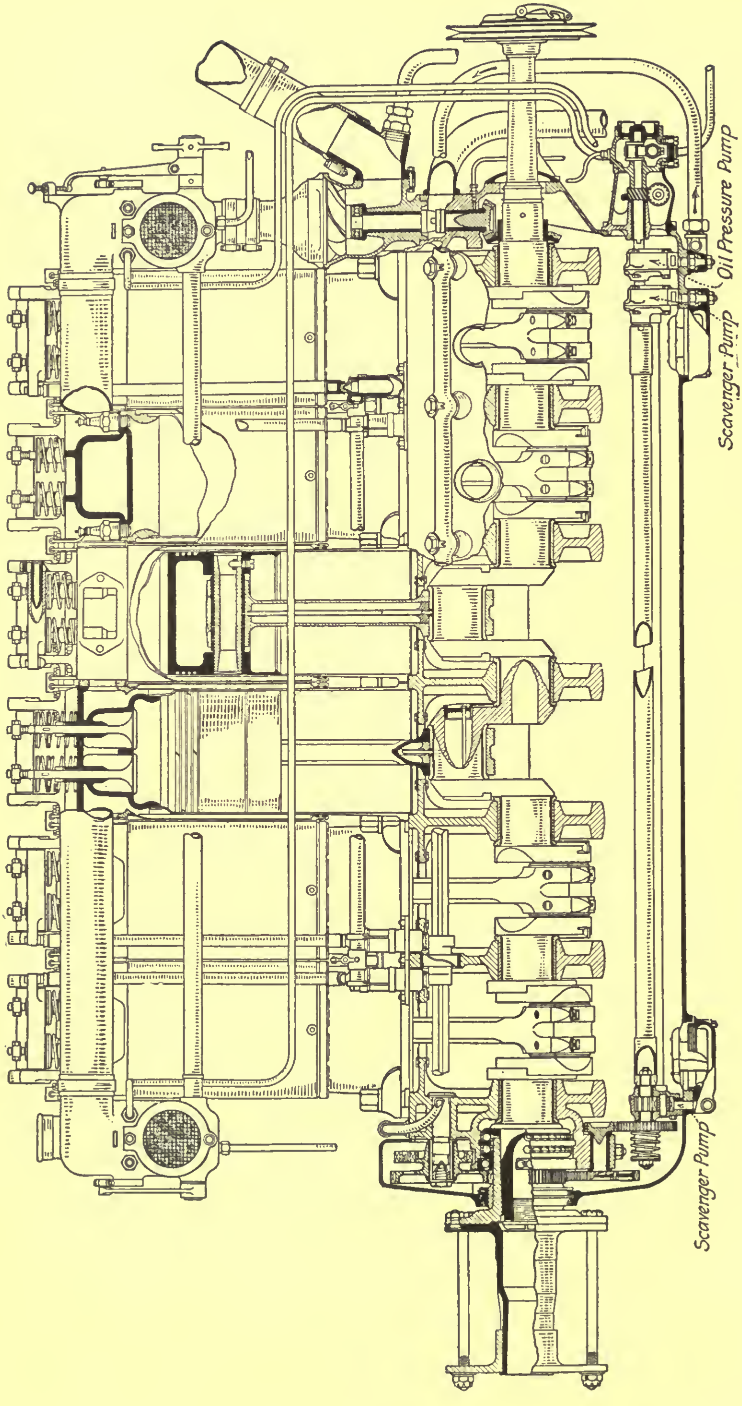


FIG. 80.—Longitudinal section of Maybach 300.

Maybach.—The Maybach 300-h.p. engine has six vertical cylinders built up of steel liners screwed into cast-iron cylinder heads and with forged steel jackets screwed to the cylinder heads. Sectional views of the engine are given in Figs. 80 and 81; general dimensions in Table 4. The compression ratio of this engine, 5.94:1, is unusually high. The pistons are of cast iron. Connecting rods are of square section, bored out; the small end has a cast-iron floating bushing.

There are two inlet and two exhaust valves working vertically in cast-iron guides in each cylinder head and operated by rocker levers mounted on roller bearings, each pair of valves being operated by a single tappet rod

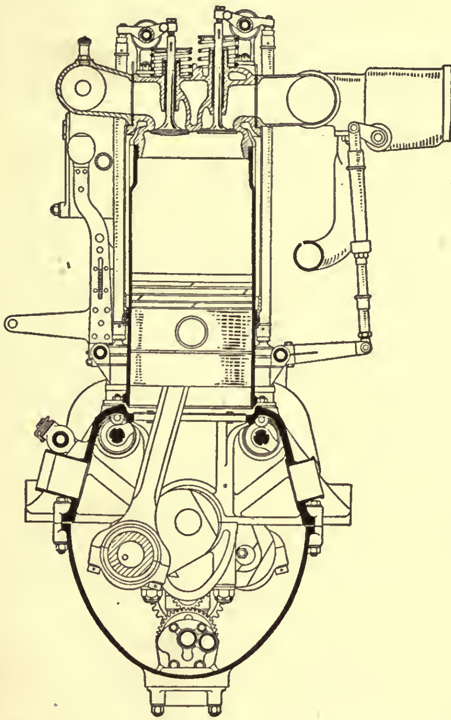


FIG. 81.—Transverse section of Maybach 300.

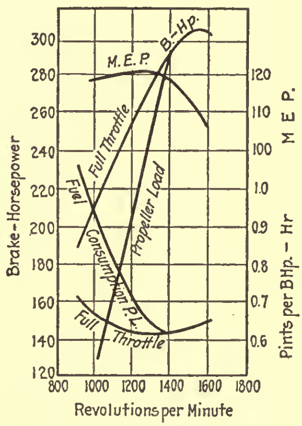


FIG. 82.—Performance curve of Maybach 300.

from one of the camshafts in the crankcase. The bottom halves of the crankshaft bearings are very deep and are bolted to the top half of the crankcase. The lower half of the crankcase supports the oil pumps and carries detachable oil pumps.

The performance curves, Fig. 82, show maximum mean effective pressure at 1,300 r.p.m., maximum power at 1,550 r.p.m., and maximum economy at 1,300 r.p.m.

8

CHAPTER V

MATERIALS

The structural materials available for airplane engine parts are in five classes, *steels,* carbon and alloy and either forged or cast, *cast iron, aluminum alloys, bronze,* and *bearing metals* or *babbitts.* The choice of metal for any given part is determined principally by (1) the suitability of the metal for meeting the stresses and general operating conditions to which the part is to be subjected, (2) the weight, and (3) the machinability of the material. It is obvious that condition (1) must always be met, but it is often possible, especially with lightly stressed members, to meet that condition with a number of different materials; for example, the crankcase may be of cast iron or aluminum. The selection as between these materials will usually be on the basis of weight although machinability or cost may determine the final choice. From the point of view of tensile strength alone the important quantity is not strength per unit of cross-section area but per unit of weight. On this basis the metals have the following properties:[1]

	Weight per cubic foot	Tensile strength, pounds per square inch	Tensile strength ÷ weight per cubic foot
Aluminum alloy, cast	170	27,000	159
Aluminum alloy, forged	170	60,000	350
Cast iron	480	20,000	42
Gun metal (bronze)	500	31,000	62
Malleable iron	480	40,000	83
Cast steel	480	60,000	125
Mild steel	480	60,000	125
High tensile steel	480	100,000	208
Nickel-chrome steel	480	135,000	280

The advantage of aluminum over cast iron for castings is not only in the great reduction in weight but also in the much greater

[1] Pomeroy, *Jour. Soc. Aut. Eng.,* Jan., 1920.

facility for machining. The substitution of a lighter material in a stressed member may result in considerable saving in weight even if the strength-weight ratio of the lighter material is not so favorable as in the heavier material. With castings this results from the fact that they cannot be reduced below certain thicknesses, which are often in excess of strength requirements, because of foundry considerations. For machined members, such as connecting rods, it is also undesirable to go below certain thicknesses, and additional material has to be left to avoid high intensity of stress at fillets where the cross-section is changing rapidly. By using a lighter metal it may be possible to load all parts of the member up to the allowable working stress, and thereby to diminish the weight.

An important design factor affecting the choice of metal is the fact that the strength of members subjected to bending or torsion is proportional to the cube of the cross-sectional linear dimensions, while the weight is proportional to the square of the linear dimensions. Thus, comparing two beams or shafts of the same material and length and with similar cross-sections but with linear dimensions of 1 and 1.41 respectively, the second will be twice as heavy as the first, and $(1.41)^3 = 2.8$ times as strong. Stiffness is also an important consideration, and as this varies as the fourth power of the linear cross-section dimensions, the second beam or shaft in the above example would be $(1.41)^4 = 4$ times as strong as the first one. If, in the second case, the allowable stress on the material is only half of that permissible in the first case, but the strength-weight ratio is unchanged, the weight will be unchanged, the strength increased 1.4 times and the stiffness doubled. For the same strength for a beam or shaft, with a stress in the second case one-half that in the first case, and constant strength-weight ratio, the relative linear dimensions would be 1 to $\sqrt[3]{2}$, or 1 to 1.26, and the relative weights 1 to $(1.26)^2$, or 1 to 1.588.

Materials for Special Parts.—*Cylinder liners* are nearly always of steel. It is not necessary to use a steel of high tensile strength since the liners cannot be machined with safety down to the dimension which will stress the material fully. The use of an aluminum alloy may be desirable where short life is expected (as in military use), but its resistance to abrasion is low although its heat conduction is much superior to that of steel.

Crankshafts are dimensioned for stiffness as well as for strength.

Stiffness for given dimensions depends only on the modulus of elasticity and not at all upon tensile strength. As the modulus of elasticity is practically constant for all steels, there is no advantage, so far as stiffness is concerned, in using a steel of very high tensile strength for shafts.

Connecting rods must be made as light as possible to keep down the unbalanced inertia forces. As they have to be machined all over and reduced to very thin sections it is important that the material should be readily machined and free from flaws. Forged aluminum would be excellent for this purpose when available.

Piston materials are considered on p. 132. *Valve* materials are selected on other than strength considerations (see p. 159).

Steels.—Steels for airplane engine use should have the following fundamental properties in as great a degree as possible: (1) high strength, (2) high toughness, (3) great durability, (4) soundness, (5) ease of machining, (6) ease of heat treatment, (7) constancy of properties, (8) homogeneity. The steels which are available are endless in number but divide themselves into certain types. Those which are of most importance for airplane engine construction are listed in Table 7,[1] which gives their composition, heat treatment and physical properties. Other special steels for valves are discussed on page 160. The 0.45 C steel is an excellent general-utility steel with good mechanical strength, high ductility and easily machined. The 0.9 – 1.0 C steel gives good service in resisting abrasion, is readily produced in sheet form and does not need hardening and tempering. The case-hardening steels are of importance for those parts which require local surface hardening combined with general toughness; the surface hardness of the 0.1 C and the 5 per cent Ni are about the same as shown by the Brinell test but the strength of the nickel steel is much higher. The air-hardening Ni-Cr steel shows the very high Brinell number of 477; its use is principally for gears. The 3 per cent Ni steel combines high tensile strength and high ductility; the addition of chromium increases the hardness. The chrome-vanadium steel has shown fewer failures in practice than most of the alloy steels, probably as a result of absence of air-hardening characteristics. The high-chromium or "stainless" steel has not only its non-rusting qualities but its high mechanical properties to recommend it. At present its cost is high.

[1] Selected from HATFIELD, *The Automobile Engineer*, June, 1920.

TABLE 7.—AIRPLANE ENGINE STEELS

Type of steel	0.45 per cent C	0.9 to 1.0 per cent C	0.1 per cent case hardening	5 per cent Ni case hardening	Ni–Cr air hardening		3 per cent Ni	3 per cent Ni Cr	Chromium vanadium	High chromium	
Composition:											
Carbon, per cent	0.45	0.90	0.11	0.14	0.30		0.39	0.32	0.46	0.28	
Manganese, per cent	0.70	0.55	0.73	0.29	0.40		0.55	0.60	0.64	0.22	
Silicon, per cent	0.15	0.11	0.10	0.16	0.105		0.11	0.12	0.19	0.102	
Sulphur, per cent	0.044	0.036	0.018	0.018	0.027		0.039	0.03	0.028	0.03	
Phosphorus, per cent	0.036	0.031	0.02	0.009	0.033		0.031	0.03	0.031	0.026	
Chromium, per cent	.....	.....	.....	.....	1.38		.....	0.76	1.48	12.58	
Nickel, per cent	.....	.....	0.12	4.84	4.07		2.96	3.41	0.19	0.20	
Vanadium, per cent	.....	.....	.....	.....	.....		.....	.....	0.33		
Condition	Normalized 820°C.	Normalized 850°C.	Normalized 900°C. Water quenched 760°C.	Normalized 860°C. Water quenched 760°C.	Oil quenched 830°C. Tempered 620°C.	Air hardened 820°C.	Oil quenched 850°C. Water quenched 620°C.	Oil quenched 850°C. Water quenched 600°C.	Oil quenched 850°C. Air cooled 650°C.	Oil quenched 950°C. Water quenched 650°C.	Oil quenched 950°C. Tempered 250°C.
Physical properties:											
Yield point, pounds per square inch	51,000	92,000	38,000	94,000	105,000	202,000	93,000	125,000	109,000	89,000	212,000
Ultimate strength, pounds per square inch	92,000	133,000	67,000	135,000	134,000	245,000	110,000	140,000	126,000	110,000	240,000
Elongation, per cent	25.0	11.0	33.0	17.5	21.0	9	24	21	22	22	7
Reduction of area, per cent	47.0	15.5	63.3	46.9	53.8	21	63.6	62	60	55	29.5
Brinell No.	170	286	146	286	269	477	228	277	255	217	444
Shore No.	29	40	25	43	44	69	35	45	41	35	64

The materials that have been employed for the various parts of airplane engines are given in Table 8,[1] which also states the nature of the stresses to which the part is subjected and the intensity of the service which it has to perform. In addition, there is given in the last column the material recommended by Doctor Hatfield on the basis of an unusually wide experience in investigating the physical properties of the steels and the causes of failure of the parts of airplane engines. There is naturally much difference of opinion among engineers as to the best material to use for several of these parts, and practice is by no means standardized.

Aluminum Alloys are largely used in airplane engines on account of their light weight. The specific gravity of pure alumi-

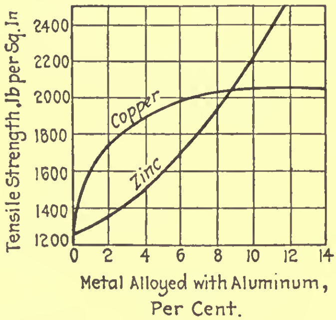


FIG. 83.—Tensile strength of aluminum alloys.

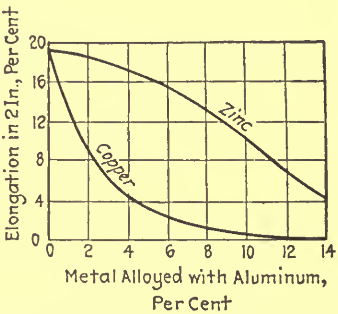


FIG. 84.—Ductility of aluminum alloys.

num is 2.56; of the common alloys about 2.8; of steel about 7.8. The important alloys are those with copper and with zinc. The effect of the presence of these substances on the tensile strength

TABLE 8.—MATERIALS FOR ENGINE PARTS

Glossary of Terms used.

A. H. Ni. Cr.	= Air-hardening Nickel-chromium Steel.	M. C. I.	= Malleable Cast Iron.
Al.	= Aluminium Alloy.	M. C. Steel	= Medium-carbon Steel.
C. H. C.	= Case-hardening Carbon Steel.	M. S.	= Mild Steel (0.15 carbon).
C. I.	= Cast Iron.	Ni.	= Nickel Steel (3 per cent).
C. Steel	= Carbon Steel.	Ni. C. H.	= Nickel Case-hardening Steel (5 per cent Ni.).
Cr. Van.	= Chromium Vanadium Steel.	Ni. Cr.	= Nickel-chromium steel (3 per cent).
H. C. Steel	= High-carbon Steel.	Ph. Br.	= Phosphor Bronze.
H. S.	= 12 to 14 per cent tungsten High-speed Steel.	Si. Cr.	= Silicon Chromium Steel.
H. T.	= High-tensile Steel.	Steel C.	= Steel Casting.
		T. Steel	= Tungsten Steel.

[1] HATFIELD, *loc. cit.*

TABLE 8 (*Continued*)

Part	Nature of chief stresses in service	Intensity of service	Materials which have been or are used	Materials recommended
Cylinder..........	High temperature, tension and abrasion.	Heavy or medium.	Al.; C. I.; Steel C.; 100,000 lb. C. Steel; Ni.; 3 per cent Ni. Cr.; Steel Mixture.	Al.; 80,000 lb. Steel.
Cylinder holding-down bolts.	Tension and bending.	Medium.	Medium or M. S.; Ni.	3 per cent Ni.
Cylinder liners.....	High temperature and abrasion.	Heavy.	C. I.; Steel C.; Forged Steel.	100,000 lb. C. Steel.
Spark-plug body...	High temperature.	Medium.	Brass; C. I.; M. S.; Stainless.	Stainless.
Spark-plug electrode.	Very high temperature.	Heavy.	T. Steel; Ni.	Nichrome Alloy or Stainless.
Valve cages........	High temperature, various slight stresses.	Medium.	C. I.	M. C. I.
Valve-rocker rollers.	Abrasion.	Light to medium.	C. H. C.; Ni. Cr.	5 per cent Ni. C. H. C.
Valves............	High temperature, tension, shock and abrasion.	Heavy.	H. S. Steels; 25 per cent Ni.; Stainless Steels; Ni. Cr.; 3 per cent Ni.; H. C. Steel.	Stainless.
Valve guides.......	High temperature and abrasion.	Medium.	Al.; C. I.; C. Steel; T. Steel.	Stainless.
Valve seats........	Abrasion, shock and high temperature.	Heavy.	C. I.; M. C. Steel; Ni. Cr.; Ph. Br.	80,000 lb. C. Steel.
Valve springs......	Torsion and bending.	Light.	Cr. Van. Steel; Si. Cr. Steel; H. C. Steel (oil hardened).	Cr. Van. Steel.
Valve rockers......	Bending, shock and abrasion.	Medium.	Ni. Cr.; C. H.; Ni.; Bronze.	5 per cent Ni. C. H.; 3 per cent Ni.
Valve-rocker bearing.	Abrasion and compression.	Light.	Ph. Br. and White Metal.	Ph. Br.
Water jacket......	Very slight.	Light.	Al.; Copper; Pressed Steel; Sheet Steel.	Steel Sheet.

TABLE 8 (*Continued*)

Part	Nature of chief stresses in service	Intensity of service	Materials which have been or are used	Materials recommended
Connecting rod....	Compression, tension, bending and shock.	Heavy.	A. H. Ni. Cr.; Cr. Van. Steel; Ni. Cr.; Ni.	Ni. Cr.; A. H. Ni. Cr.
Connecting-rod, big-end bearing...	Abrasion and compression.	Heavy.	White Metal; Ph. Br.	
Connecting-rod, little-end bearing..	Abrasion and compression.	Medium.	Ph. Br.; Gun Metal.	
Connecting-rod bolts.	Tension and bending.	Medium.	Ni.; Ni. Cr.; Cr. Van. Steel.	3 per cent Ni.
Piston pin.........	Shear, bending and abrasion.	Heavy.	Ni.; 160,000 to 180,000 lb. C.; C. H. C. Steel; C. Steel; Ni.; Ni. Cr.; T. Steel.	5 per cent Ni. C. H.
Piston............	Temperature, bending and other stresses.	Heavy.	Al.; C. I.; Steel Drawn or Pressed; Ni. Cr.	Al.; 80,000 lb. C. Steel.
Piston ring........	High temperature, bending and abrasion.	Heavy.	C. I.; H. S. Steel.	C.I.
Bevel-gearshaft for overhead camshaft.	Torsion and bending.	Medium.	C. Steel; Ni.; Cr. Van. Steel.	5 per cent Ni. C. H.
Crankcase.........	Bending and various slight stresses.	Light.	C. Steel; Al.; Ni.	Al.
Crankshaft........	Torsion, bending, and shock.	Heavy.	Cr. Van. Steel; Ni. Cr.; Ni.	Ni. Cr.
Crankshaft journal bearing.	Abrasion and compression.	Medium.	Ph. Br.; White Metal.	
Propeller reducing gears.	Shear and bending, abrasion and shock.	Medium.	Ni. C. H.; Ni.; Ni. Cr.; Cr. Van. Steel; A. H. Ni. Cr.; C. H. C.	A. H. Ni. Cr.
Cams.............	Compression and abrasion.	Heavy.	Ni. C. H.; C. H. Cr.	5 per cent Ni. C. H.
Cam housings......	Negligible.	Light.	Al.	Al.
Camshaft bearing..	Abrasion and compression.	Medium.	White Metal; Ph. Br.	White Metal; Ph. Br.; Gun Metal.
Camshaft.........	Torsion and abrasion.	Heavy.	Ni. Cr. C. H.; Ni.; H. C. Cr. Steel.	5 per cent Ni. C. H.
Tappets...........	Compression and abrasion.	Heavy.	C. H. C. hardened on wearing surface; Ni. Cr.	5 per cent Ni. C. H.

of cast-aluminum alloy is shown in Fig. 83. It will be seen that copper increases the strength up to about 9 per cent; with 7 to 8 per cent of copper there is obtained a tough alloy of a tensile strength of 20,000 lb. per square inch. With zinc the tensile strength increases up to about 35 per cent; with this alloy the tensile strength reaches 50,000 lb. per square inch but it is very brittle and has a specific gravity of 3.3. The ductilities of the two types of alloy are shown in Fig. 84. With both copper and zinc present, higher tensile strength combined with fair ductility can be obtained; for example, with 2.75 per cent Cu and 7 to 8 per cent Zn a tensile strength of 28,000 lb. per square inch and a ductility of 8 per cent. This alloy falls off rapidly in tensile strength with increase in temperature; at 570°F. the strength is 9,500 lb. per square inch. A small addition of manganese to a copper aluminum alloy increases the strength and maintains it better with increase of temperature.

Forged-aluminum alloy is best represented by **Duralumin,** whose composition is 93.2 to 95.5 per cent aluminum, 0.5 per cent magnesium, 3.5 to 5.5 per cent copper and 0.5 to 0.8 per cent manganese. This material can be made into plates and tubes. The tensile strength is about 60,000 lb. per square inch but can be increased by rolling to about 75,000 lb. per square inch though with loss of ductility; the elongation of the alloy is 15 to 20 per cent. At 460°F. the tensile strength is halved. Forged-aluminum is a good bearing metal.

Both cast- and forged-aluminum alloy have a modulus of elasticity of about 10,000,000 lb. per square inch or about one-third that of steel. The stiffness of a plate structure is proportional to the modulus of elasticity and to the cube of its thickness. An aluminum plate of the same weight as a steel plate would be nearly three times as thick; its stiffness would be about eight times as great as that of the steel plate.

CHAPTER VI

ENGINE DETAILS

Cylinders.—There is considerable variety in the design of modern water-cooled airplane-engine cylinders. In one important respect they are all in accord, namely, in the adoption of overhead valves, in which they differ from the common automobile engine. They differ further from the automobile engine in that the cast-iron block construction is seldom used. Both of these changes have been made, primarily, in order to reduce the weight of the engine.

Airplane-engine cylinders are formed either singly, or in blocks of two, three or four. The single cylinder is flexible and permits freedom of movement of the cylinder without putting strains on other parts of the engine. As the engine is not held rigidly it is desirable to give all its parts a maximum of freedom. The single-cylinder construction is used in the majority of existing engines and is always employed in those engines whose cylinders are all steel. Examples of this construction are the Liberty, Packard, Hall-Scott L, Curtiss K, Rolls-Royce, Napier, Renault, Lorraine-Dietrich, Fiat, Mercedes and Austro-Daimler engines. It is necessarily employed also in radial and rotary engines (see Chapter VIII). The block arrangement has a common jacket around the cylinders which may result in some slight decrease in weight of the jacket itself but will usually increase the weight of water in the jackets and give inferior water circulation. The major advantage of employing the block construction is that it permits the cylinders to be more closely spaced and thereby diminishes the over-all length (and weight) of the engine. In the Thomas-Morse and Sturtevant engines the cylinders are in pairs; in the Siddeley "Puma" in threes; in the Hispano-Suiza and Bugatti in fours.

Another variation among airplane-engine cylinders is in the form of the cylinder head. In some engines (Hispano-Suiza, Napier) the cylinder head is flat and of the same diameter as the cylinder barrel. With two equal valves in the head, the maximum possible external valve diameter is half the cylinder diameter minus half the thickness of the bridge between the valves.

It is frequently found that this diameter is insufficient to give the desired opening for the admission of the mixture and results in a low volumetric efficiency. To remedy this the head may be made with sloping sides either without enlargement of diameter (Curtiss OX, Mercedes) or with enlargement (Liberty, Lorraine Dietrich, Austro-Daimler, Fiat). Another common method of meeting this difficulty is by the use of multiple valves. The two devices may be used together.

The most important factor in determining the design of cylinders is the material employed; the following constructions are in use:

(*a*) All cast iron, with cylinders either single (Curtiss OX) or in blocks. This construction leads to excessive weight.

(*b*) Cast-iron barrel, head, and part of jackets, but with aluminum sides to the jacket (Bugatti). This is for block construction only. It reduces weight and makes the inside of the jacket accessible for machining and cleaning.

(*c*) Cast-iron barrel and head, but with sheet metal jackets, either copper (Beardmore), steel (Benz) or monel-metal (Curtiss).

(*d*) Steel barrels, aluminum heads and jackets. The steel barrel may be integral with its head (Hispano-Suiza, Siddeley "Puma") or only a cylindrical shell (Sturtevant). The aluminum jacket may be complete (Hispano-Suiza, Sturtevant) or may use the steel barrel as the inner wall (Siddeley "Puma").

(*e*) Steel barrel, cast-iron head and steel jacket (Maybach, Benz).

(*f*) All steel (Liberty, Packard, Hall-Scott L, Curtiss K, Rolls-Royce, Napier, Renault, Lorraine-Dietrich, Fiat, Austro-Daimler, Bassé-Selve, Maybach). This construction is used more than any other and appears to be displacing other constructions. The cylinder is either machined out of a solid forging or may be made from drawn steel tubing. The jackets are commonly stamped to shape in two halves along a plane through the cylinder axis, and are welded together and to the cylinder. The valve ports and guides offer some difficulties as compared with cast cylinder heads.

Thickness of Cylinder Walls.—If p = maximum gas pressure in the cylinder, pounds per square inch, d = cylinder diameter, inches, t = wall thickness, inches, and s = allowable tensile stress, pounds per square inch, then $t = pd/2s$, gives the thickness of metal necessary to withstand the gas pressures. Taking s as 5,000 and 14,000 lb. for cast iron and forged steel, the respective wall thicknesses for a 5-in. diameter cylinder and for p = 500 lb. are 0.25 and 0.09 in. With cylinders of small diameter the thickness may have to be increased over the calculated values to ensure sufficient stiffness and, in the case of cast iron, to offset a possible lack of homogeneity. No allowance need be made for wear or reboring as the engines are essentially short-lived.

The clearance space alone of the engine is subjected to the maximum explosion pressure; the pressures to which the walls are subjected become progressively less from the clearance space to the part of the cylinder at the lowest point reached by the top of the piston, below which point they become zero. In addition to the gas pressures the cylinder walls have to tie the cylinder head to the crankcase and shaft bearings and consequently have to withstand the maximum gas pressure exerted on the cylinder head. The thickness of wall required for this is given by

$$t_1 = \frac{\pi}{4}d^2 \cdot p / \pi ds = \frac{pd}{4s} = \frac{t}{2}$$

The lowest part of the cylinder consequently does not have to be more than one-half the thickness of the upper end. In many designs the cylinders are turned of diminishing thickness from head to crank end; in other designs (Mercedes, Lorraine Renault, Liberty, Curtiss K) the upper part of the cylinder is reinforced by stiffening ribs while the lower part is without such stiffening.

The following table gives some cylinder thicknesses and calculated stresses; a maximum gas pressure of 500 lb. per square inch is assumed.

Engine	Diameter, inches	Material	Cylinder-head thickness, inches	Cylinder-barrel thickness, inches	Calculated maximum stress, pounds per square inch
Benz, 230	5.71	Cast iron	0.2600	0.216	6,600
Liberty 12	5.00	Steel	0.1875	0.156	8,000
Curtiss, K-12	4.50	Steel	......	0.078	14,400
Hall-Scott A5a	5.25	Semi-steel casting	......	0.125	10,500
Austro-Daimler, 200	5.31	Steel	0.1970	0.138	9,620
Renault, 400	4.92	Steel	......	0.110	11,200
Bassé-Selve	6.10	Steel	0.2700	0.118	12,900
Maybach, 300	6.50	Steel	0.3100	0.110	14,800

The thickness of the cylinder head is determined mainly by considerations of stiffness. It is essential that the valve seats, which are located in the cylinder head, should be free from deformation and this cannot be secured unless the heads are stiff. In cases where the integral cylinder head is backed by an aluminum casting (Hispano-Suiza, Napier) the thickness cannot be reduced because the heat transfer through the double thickness of metal is poor and the exhaust valve seat, which is the hottest part of the cylinder, must have sufficient thickness of metal around it to conduct the heat away and prevent warping from overheating and unequal expansion. The head thicknesses of a few cylinders are given in the table above. Steel cylinders are generally machined out of solid sheet forgings, but in the case of the Liberty engine the cylinders have been made from steel tubing ½ in. thick, whereby a great reduction is obtained in the amount of metal to be removed by machining.

The **valve seats** are integral with the heads in steel cylinders but have to be cast, or otherwise fastened, into aluminum heads, With integral or non-detachable valve seats it is impossible to take out the valves without taking off the cylinder or taking out the piston. Detachable valve cages have been used, but these will always result in imperfect cooling of the valve seat. A compromise adopted in the Beardmore and Austro-Daimler engines is to have a detachable valve cage (Fig. 85) for the inlet valve, which requires little cooling; the exhaust valve can then be dropped into the cylinder and withdrawn if necessary through the inlet valve seat opening.

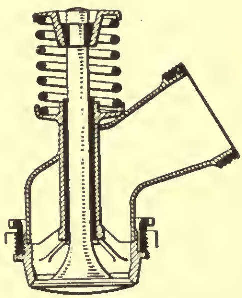


FIG. 85.— Detachable valve cage (Austro-Daimler engine).

The **valve ports** and **valve stem guides** in all steel constructions are integral for each valve and are welded to the cylinder head. The guides are provided with bushings of steel or bronze. The valves must work very freely in these bushings and at the same time there must be no leakage of air or exhaust gas between valve stem and guide; the guides are made quite long.

Many devices have been used for **attaching the cylinder to the crankcase.** The best method is to tie the cylinder, by through bolts, directly to the main bearing caps on the crankshaft on each side of the cylinder. With two long bolts on each side of the cylinder tying the cylinder flange to the main bearing caps, the gas pressure on the cylinder head is supported entirely by stresses set up in the bolts and is not transmitted to the crankcase, which can consequently be made lighter. As the bolts are parallel to the cylinder axis and are symmetrically disposed around it, this method of attachment avoids all distortion. One pair of bolts is commonly made to serve for two adjacent cylinders by the use of dogs through which the bolt goes and which are supported equally on the flanges of both cylinders (Fig. 80). In the Bugatti engine (Fig. 67) the through bolts are supplemented by studs in the crankcase.

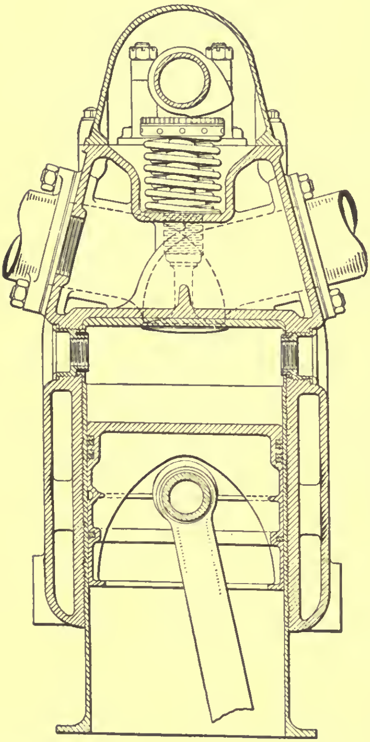


Fig. 86.—Cylinder of Hispano-Suiza engine.

It is only in engines with a single row of cylinders that the above method of attachment can be employed. In Vee and W engines other methods must be used; the most common is by studs in the crankcase which pass through the flange at the lower end of the cylinder (see Fig. 47). In the Curtiss K engines the steel cylinder is kept in place by the aluminum cylinder head, which is bolted to an aluminum jacket cast integral with the upper half of the crankcase (Figs. 87 and 56).

Typical Cylinder Designs.—Single cast-iron cylinders are seldom used; a typical example is in the Benz 230 engine of Fig. 77. The barrel and head are cast in one piece. Jackets are of die-pressed steel welded on and extending well down towards the base flange. They are provided with annular corrugations to take care of expansion. Plates welded in position in the jacket space above the crown of each cylinder deflect the water to the exhaust valve pockets. The cylinder barrels extend 0.39 in. below the base flanges into the crank chamber and are held down each by four bolts and four dogs. The cylinder walls taper from 6.5 mm. at the top to 5.5 mm. at the base.

An example of the block cast-iron construction is shown in Figs. 66 and 67 (Bugatti). The cylinders are in blocks of four.

Composite steel and aluminum constructions are made up in several ways. In the Hispano-Suiza engine (Figs. 86 and 51) the steel liner with integral head is threaded throughout the entire length of contact with the aluminum. The aluminum jacket is a block construction for four cylinders and completely surrounds the steel liners so that valve ports go through both the aluminum and the steel. For proper cooling it is necessary to

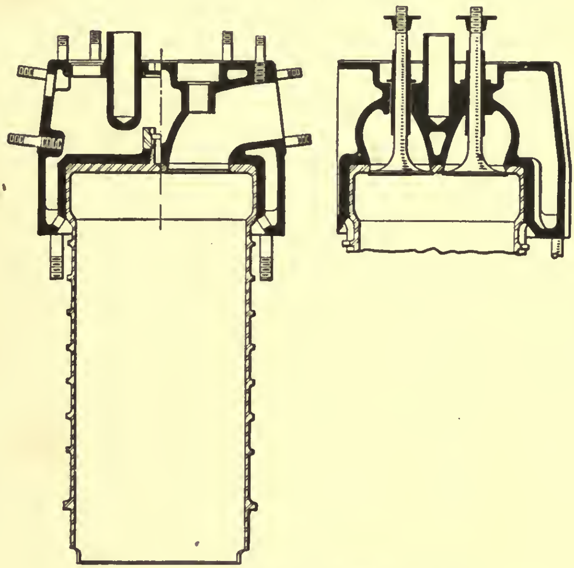


FIG. 87.—Cylinder of Curtiss K engine.

have perfect contact of the two metals at the cylinder head as well as the barrel, otherwise warping of the steel head will occur and the valve seats will distort. There is no actual contact anywhere of water with the steel cylinder. The steel liners are attached to the crankcase by bolts.

A different steel-aluminum block construction is employed in the Curtiss K engines. In this case the steel liners with integral heads are turned with stiffening flanges (Figs. 87 and 56) on the outside, with a packing retaining flange at the bottom and a central stud at the top. The upper end of the liner is of slightly enlarged diameter and is threaded on the outside. The aluminum cylinder head is a block casting into which the six cylinders are screwed. To ensure intimate contact the threaded stud at the center of the liner head, which passes through the head casting, is drawn up by a nut. The head casting matches up with and is bolted to a flange on the upper end of the aluminum cylinder block, which is integral with the upper half of the crankcase. Jacket water is in contact with the liner below the combustion space only. Water tightness is obtained by packing between the lower liner flange and the crankcase.

Still another construction is used in the Napier "Lion" engine (Fig. 73). This engine employs liners with integral heads, which are fastened to a four-cylinder aluminum-block head by four valve seats (inlet of bronze, exhaust of steel) in each cylinder. These seats are screwed into the cylinder head. The use of removable seats is in general objectionable as it means less perfect cooling of the seats than is possible with an integral construction. The barrel jackets in the latest design are separate and of pressed steel, made in two halves welded together and to flanges on the cylinders at the top and bottom of the water spaces. In the design shown in Fig. 73 a common steel water jacket is bolted to the head casting and makes a joint with each cylinder at its lower end by means of a rubber ring pressed against a flange on the liner by a large circular split nut.

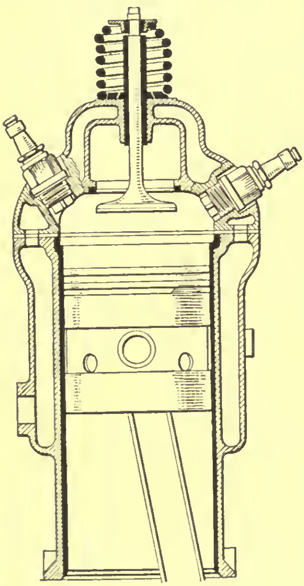


Fig. 88.—Cylinder of Sturtevant engine.

A steel-aluminum block construction in which the liner has no head is employed in the Siddeley "Puma" engine (Fig. 8). In this case the aluminum heads are cast in sets of three and the steel liners are screwed and shrunk into them. Aluminum water jackets are also cast in threes and are bolted to the heads. The lower joint between the aluminum jacket and the liner is made by a screwed gland, which squeezes a rubber ring against a shoulder on the outside of the liner. The valve seats are of phosphor bronze expanded into the aluminum head.

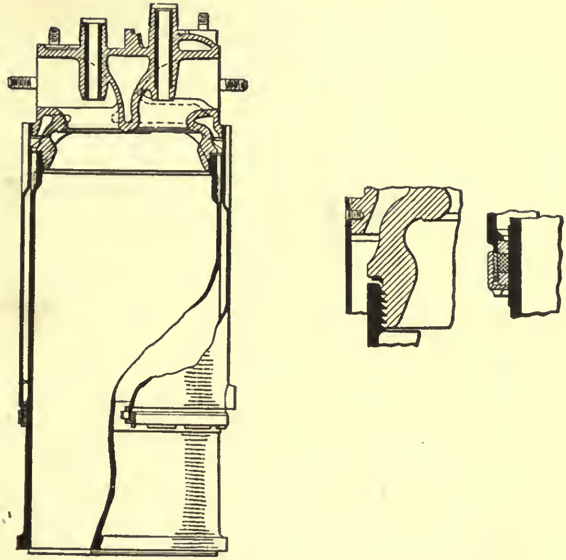


Fig. 89.—Cylinder of Maybach engine.

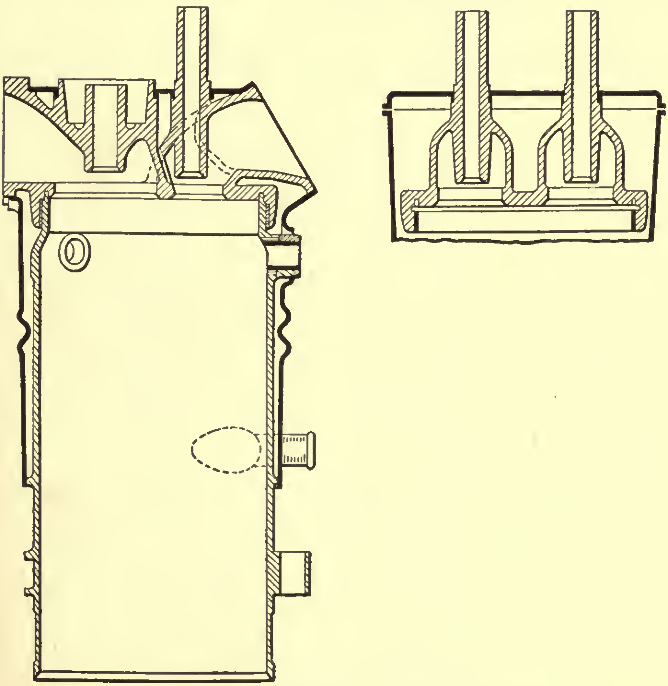


Fig. 90.—Cylinder of Benz 300 engine.

Another construction in which a headless steel liner is employed is the Sturtevant engine, Fig. 88. The aluminum cylinders are cast in pairs and are provided with closely fitting steel liners; the heads are also of aluminum with inserted valve seats and are in pairs. A peculiarity of this construction is that the aluminum cylinders are bolted direct to the crankcase; the steel liner does not transmit any longitudinal forces and is perfectly free to expand.

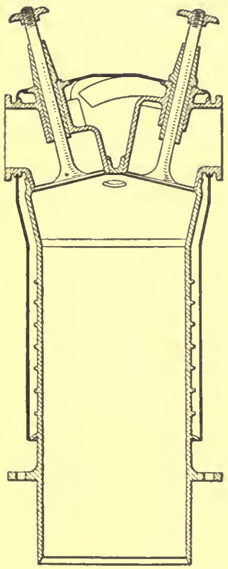


FIG. 91. — Cylinder of Liberty engine.

A composite steel and cast-iron construction is used in the Maybach engine (Figs. 89 and 81). The steel barrel is screwed with a buttress thread into a cast-iron head and comes up against a soft brass washer which prevents water leakage. The water jacket is a machined forging and screws on to the cylinder head (see details, Fig. 89) where it is sweated in position with soft solder; the lower joint is kept watertight by a gland and rubber ring. The valve-stem guides have cast-iron bushings.

In the Benz 300 engine (Fig. 90) the cylinder head is of cast iron and the rest of the cylinder is of steel. There are ports for two inlet valves and one exhaust valve. The steel liner screws into the cast-iron head and makes a watertight joint by bedding into cement in the small groove into which the top of the liner goes.

The all-steel construction is exemplified in the Liberty cylinder (Fig. 91). This design has a bumped head and obliquely set valves. The jackets and valve ports are welded to the sheet cylinder. Details of this welding are shown for the very similar Packard engine, Fig. 92.

In the Austro-Daimler engine (Fig. 93) the cylinders are all steel with pressed steel jackets and twin inlet and exhaust valves in the cylinder heads. The valve pockets are welded in position with the exception of one inlet valve (Fig. 85), which is detachable with its seat and guide. The cylinders taper from 4 mm. at the top to 3 mm. at the middle and increase again to 4 mm. at the bottom; the jackets are 1 mm. thick. The bottom

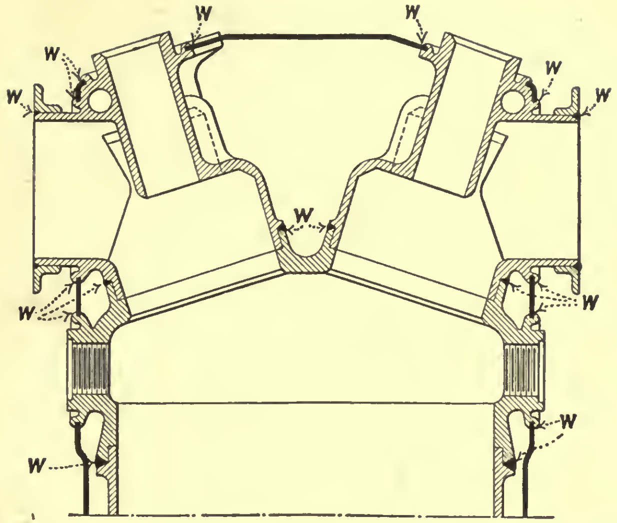


FIG. 92.—Cylinder head of Packard engine showing welding, W, of the water jacket.

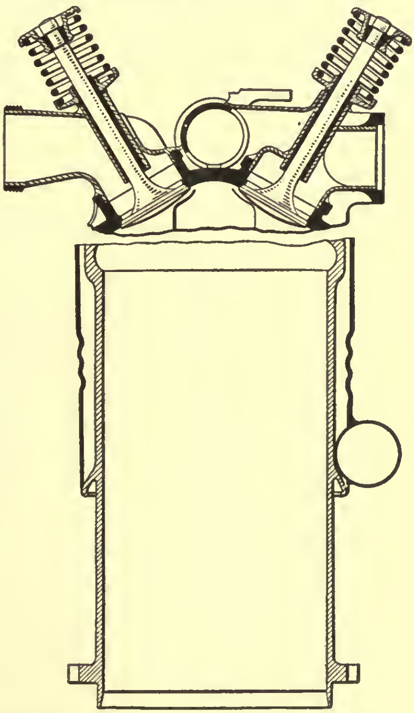


FIG. 93.—Cylinder of Austro-Daimler engine.

of each jacket is flanged over and welded to a bevelled flange machined on the barrel.

Pistons.—One of the most important steps in the improvement of the airplane engine has been the general substitution of an aluminum alloy for the cast iron that, until very recently, was universally employed for the piston material. The piston must be as light as possible in order to keep down inertia forces and at the same time must be thick enough in the crown to conduct the heat away rapidly from the center to the circumference, where it is taken up by the cylinder walls. With cast-iron pistons, reduction in weight has resulted in many troubles and especially in the burning and cracking of the piston head. Measurements by Hopkinson in a Siddeley engine show the cast-iron piston head has a temperature of over 900°F. This is borne out by similar measurements on the pistons of air-cooled cylinders by Gibson.

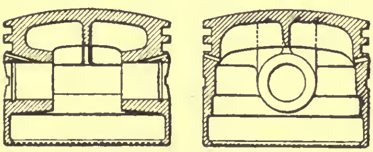


Fig. 94.—Aluminum piston.

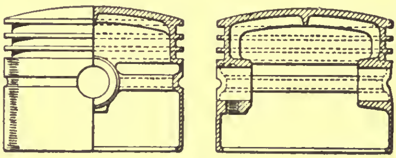


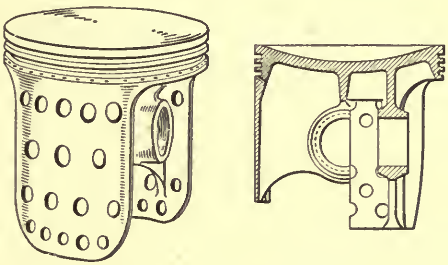
Fig. 95.—Cast-iron piston.

With aluminum pistons this temperature is reduced to about 400°F. and at the same time the piston is considerably lighter. The two pistons of Figs. 94 and 95 for a 100 by 140 mm. air-cooled engine have weights of 1.26 lb. and 1.77 lb., or a reduction in weight of 29 per cent by the substitution of aluminum alloy for cast iron; these pistons show the temperature difference noted above.

The lower temperature of the aluminum piston has other important results. It reduces the rise in temperature of the incoming charge during the suction stroke and thereby increases the volumetric efficiency of the engine, and it permits the use of a higher compression ratio without danger of preignition and thereby increases both the capacity and efficiency of the engine. The horse power of an engine can be increased at least 5 per cent by the use of aluminum pistons.

The piston should be designed with heat dissipation in mind as much as the other piston functions. The heat received at the center of the head must pass out radially to the circumference, and this should be provided for either by adequate thickness of metal from center to periphery or by the provision of radial ribs which serve the double function of heat carriers and stiffening members. Furthermore, the thickness of the cylindrical wall must not be cut down behind the first ring as the heat has to flow downward from the crown.

The friction between the piston and the cylinder is by far the largest item of mechanical loss (see p. 24) and should be kept as low as possible. Its high value results apparently from the partial carbonization of the lubricant clinging to the walls under the action of the high gas temperatures and of the slight but unavoidable leakage of burning gases past the piston rings. As a result, the viscosity of the lubricant is greatly increased. The



(a) (b)

FIG. 96.—Slipper piston.

extent of the friction depends upon: (1) The pressure of the piston against the cylinder walls, which governs the thickness of the film; the friction appears to be proportional to the average loading; (2) the area of the bearing surface; (3) the quantity of the lubricant on the walls; the friction increases with this quantity; (4) the temperature of the walls, which controls the viscosity of the lubricant. Of the means adopted to reduce this friction loss the most prominent is the cutting away of the piston skirt on the sides which do not support side thrust.

This practice has the further advantage of reducing the weight of the piston. An example of such a piston is shown in Fig. 96. It will be seen that the piston-pin bosses in this design are supported by vertical transverse ribs, which pass within a distance from the center of the head of a little more than half of the radius of the piston. This is an important feature in keeping the piston head cool; the heat absorbed by the central portion of the head can pass down these ribs to the gudgeon-pin bosses and to the skirt. This in turn permits the use of a thinner crown, especially as the stiffness of the crown is greatly increased by the support of the ribs. The thickness of the lubricant film is diminished by providing holes in the slippers through which excess oil is squeezed out. If both slippers are designed for the same intensity of pressure, then areas of the two slippers will be different. Such a design is shown in Fig. 97; the supporting ribs are no longer parallel.

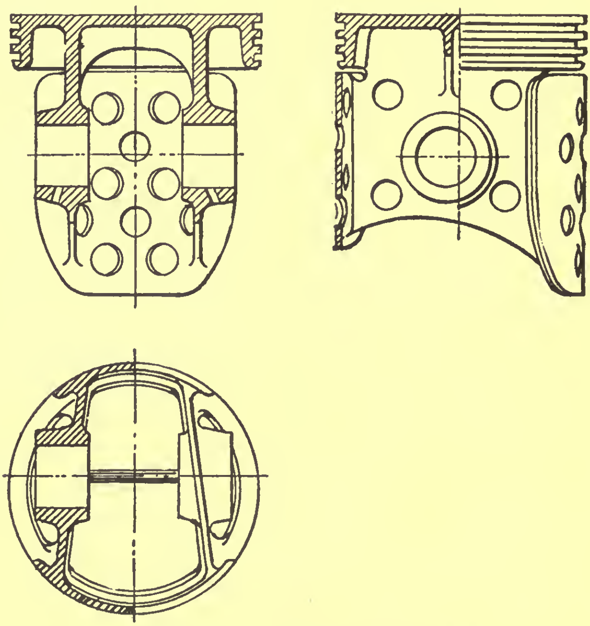


Fig. 97.—Piston with unequal slippers.

One of the important troubles with pistons is the slap which occurs when the side thrust is transferred from one side to the other. The amount of this slap depends on the clearance between the piston and cylinder. The cold clearance must be larger with aluminum than with cast iron as the expansion is much greater. The pistons of Figs. 94 and 95 have cold clearances of 0.026 and 0.020 in. respectively; the hot clearances for both are 0.008 in. The clearances should be greatest at the top and where the temperatures are highest and should diminish as the bottom of the skirt is approached. With aluminum pistons a cold clearance over the top lands of about 0.005 in. per inch diameter is necessary. Such a piston will be noisy when cold. For cast iron the cold clearance should be 0.003 in. per inch diameter at the top, and 0.00075 at the base. The clearances for special engines are given in Table 4.

In order to prevent piston slap, or the opposite danger of seizing when hot, the practice has arisen of insulating the piston skirt from the ring-carrying portion of the piston. This is most readily accomplished by the use of a piston with piston-pin bosses carried by ribs (Fig. 97) and with the skirt or slippers separate from the upper portion of the piston. A design of this character is shown in Fig. 98; in this case a complete skirt is used. Such constructions are satisfactory only for cylinder diameters up to about 5 in.; for larger sizes the piston cooling will be inadequate. The clearance necessary for the skirts of aluminum divided slipper pistons is about the same as that required for the normal cast-iron piston.

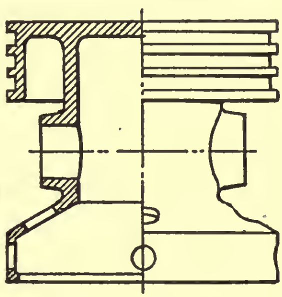


Fig. 98.—Divided-skirt piston.

Another method of reducing piston slap is by offsetting the wristpin by a small amount, usually not more than ¼ in. The object of this construction is to cause the piston to tilt slightly about the piston pin and therefore to pass progressively instead of abruptly from one cylinder wall to the other. Such offset is shown in Figs. 96 and 97.

The composition of the aluminum alloy used in German pistons is given below. These pistons are usually die castings and have a tensile strength of 28,000 to 31,000 lb. per square inch and extension of 4½ per cent as against about one-half those quantities for sand castings.

	Copper	Zinc	Iron	Silicon	Tin	Nickel	Manganese	Magnesium	Aluminum
Benz 230 h.p.	6.02	12.13	1.42	0.31	0	0	Tr.	Tr.	80.12
Austro-Daimler 200 h.p.	7.67	1.33	1.32	0.52	2.21	0	Tr.	0.29	86.66
Bassé-Selve 270 h.p.	1.90	15.62	1.06	0.45	0	0	0	0	80.97

Piston weights (including piston rings and gudgeon pin) vary from 0.19 lb. (Austro-Daimler) to 0.25 lb. (Liberty) per square inch of piston area in aluminum construction; for cast iron the weight may exceed 0.42 lb. (Maybach).

Typical Pistons.—Figure 99 is the Maybach cast-iron piston. Figure 100 is the Benz cast-iron piston with a thin crown and with

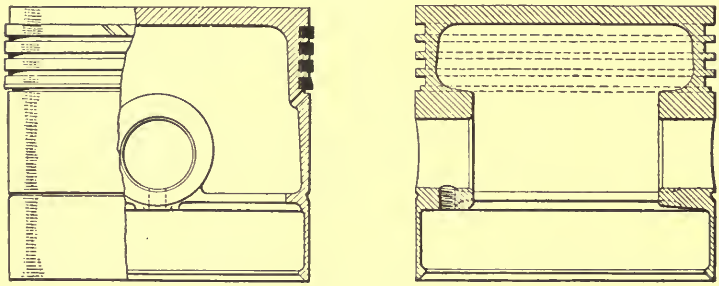


Fig. 99.—Maybach cast-iron piston.

a hollow conical steel pillar riveted to the piston head and resting directly on the middle of the piston pin. The small end of the connecting rod is cut away to avoid interference with this pillar. With this arrangement the gas pressure is transmitted in the line of the connecting rod and there is no bending moment on the wristpin. The ribbed aluminum construction is shown in Fig.

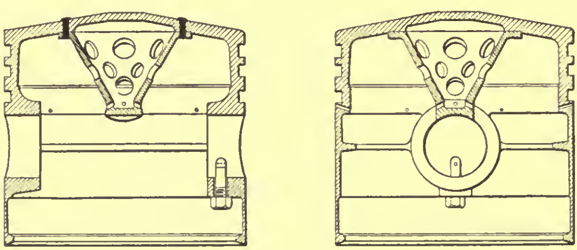


Fig. 100.—Benz 230 cast-iron piston.

101 for the same engine. The top clearances for the two pistons are 0.02 and 0.03 in. respectively; the bottom clearances are 0.004 and 0.014 in. respectively. The weights are 6.72 lb. and 4.90 lb. respectively, complete with rings and setscrews but without piston pins; the saving in weight is 27 per cent. The ribless construction is typified by the Liberty engine piston shown in Fig. 102; oil grooves are provided on the piston skirt.

Piston rings are of dense gray cast iron, fully machine-finished, peened on the inner curved surface and exactly ground to size upon the outer curved surface. With very narrow rings semi-steel is used. The rings may be either of the concentric or eccentric types. The ends are commonly chamfered at an angle of 30 to 40 deg. but stepped ends are also used; the gap when in the cylinder is about $\frac{1}{240}$ the diameter of the piston. Three

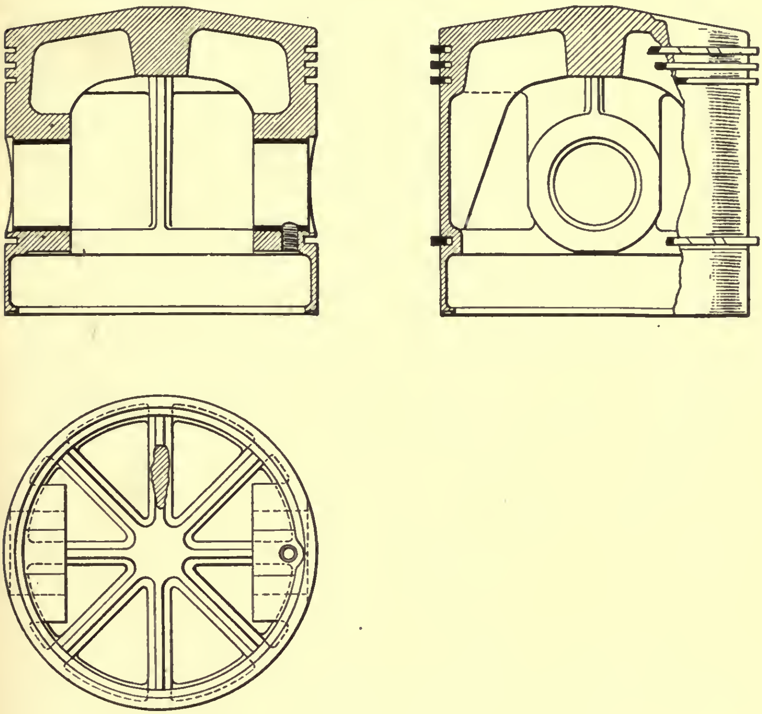


FIG. 101.—Benz 230 aluminum ribbed piston.

rings are commonly used, but four rings are found occasionally. Two narrow rings are sometimes used in one groove. A scraper ring near the bottom of the skirt is sometimes used to clear excess oil from the cylinder walls; the same result is obtained by the use of perforations through the skirt. In some pistons (Figs. 99 and 100) the lowest ring acts as a scraper and has a groove below it through which small holes are drilled to the interior of the piston to drain away any excess of oil.

Piston or Gudgeon Pin.—The piston pin is usually of steel, machined to size, case-hardened and ground. It is always hollow. It is most commonly fully floating, that is, it has bearing in the end of the connecting rod as well as in the piston bosses, with some end motion as well. The pin will then rotate and local wear will be avoided. The bushing in the connecting rod is also floating in many engines. On the cold motor the pin should be a mild driving fit in the bosses and a running fit in the connecting-rod bushing. When warm the aluminum bosses expand more than the bushing and the pin becomes free. With standard piston types, as in Figs. 99–102, the piston pin is comparatively long and is subjected to considerable bending stress. By the adoption of the slipper piston (Figs. 96 and 97) or the divided skirt (Fig. 98) the bosses are brought closer together and the pin is shortened. It can consequently be made of smaller diameter and lighter and if fully floating will show no wear.

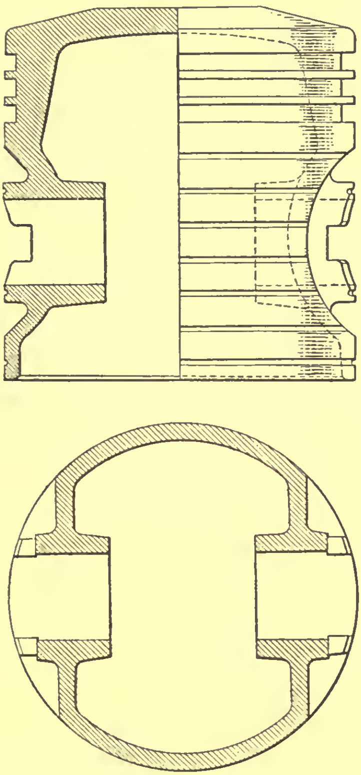


Fig. 102.—Liberty ribless aluminum piston.

Connecting Rods.—In vertical engines the connecting rods are of uniform (non-tapering) circular or I-section, with solid small ends and marine type big ends. An example of the circular section (Benz 230) is shown in Fig. 103; the I-section, assembled with the piston (Siddeley "Puma"), in Fig. 104. The small end is usually provided with a bronze bushing, although in the Maybach engine this is replaced by a perforated cast-iron floating shell 0.124 in. thick. The large end has a babbitted-bronze shell. The tubular rods used in several German engines are sometimes provided (Fig. 103) with a centered internal pipe for lubricating the small end. The Benz rod has a number of radial holes drilled in the big end to reduce weight and has the top of the small end cut away to permit direct application of the gas pressure load on the crankpin through a pillared piston as in Fig. 100.

In Vee engines several connecting rod arrangements are used. In the Curtiss OX and V, Sturtevant and Thomas-Morse engines

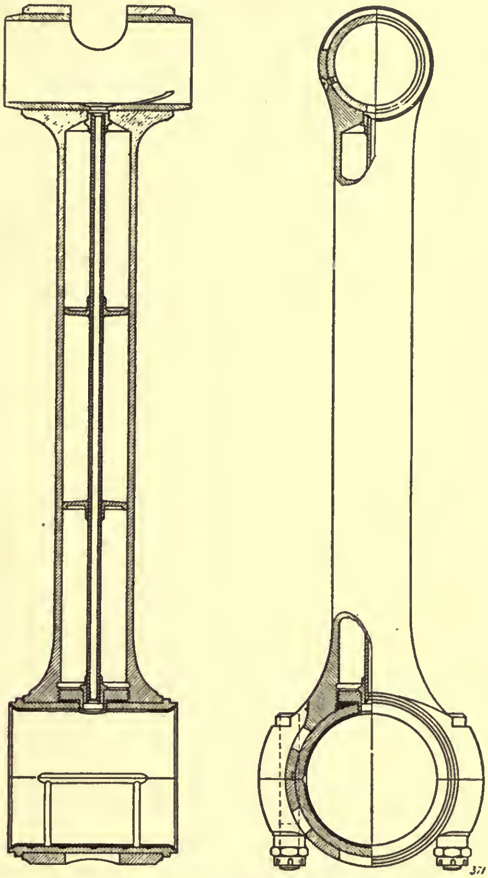


Fig. 103.—Benz tubular connecting rod.

the cylinders in the two rows of the Vee are staggered so that the connecting rods do not lie in the same plane. With this arrangement the big ends of each pair of cylinders lie side by side on the same crankpin. Such an arrangement results in an increase in the over-all length of the engine.

With the cylinders of each pair opposite one another, as in the general practice with Vee engines, the connecting rods are in the same plane and special arrangements must be made to connect them both to the crankpin. Two arrangements are in use, the forked rod and the articulated rod. The forked rod is used in the Liberty, Hispano-Suiza, Packard, and Fiat engines. Each pair of rods consists of a plain rod and a forked rod. The forked rod clamps the big end bronze bushing; the plain rod works on the

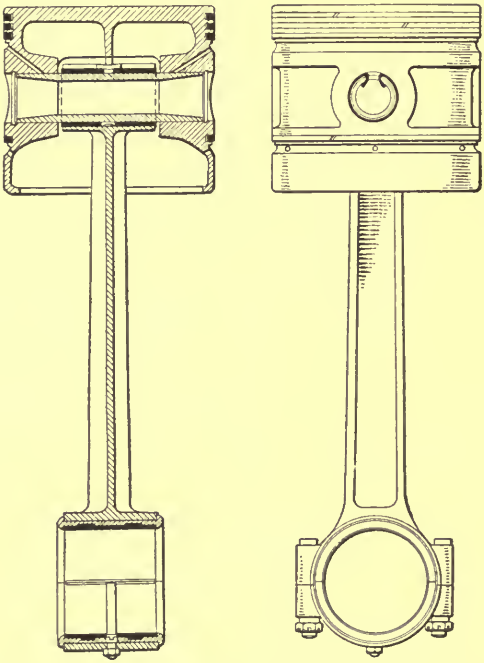


Fig. 104.—Siddeley "Puma" I-section connecting rod.

outside of this bronze bushing between the two forks of the forked rod. The rods for the Liberty engine are shown in Fig. 105. The bearing is prevented from rotating in the forked rods by dowel pins. In the Fiat engine the bottom ends of the fork are fastened together.

In the articulated rod assembly a master rod is used and a short rod is attached to a pin which is held in the upper half of the big end of the master rod. Ordinarily the master rods are all placed on one side of the Vee, but occasionally (Renault) the master rods alternate with short rods. A good example of a

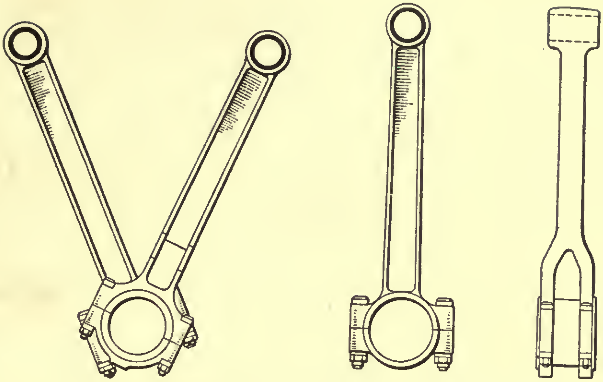


FIG. 105.—Forked connecting rods of Liberty engine.

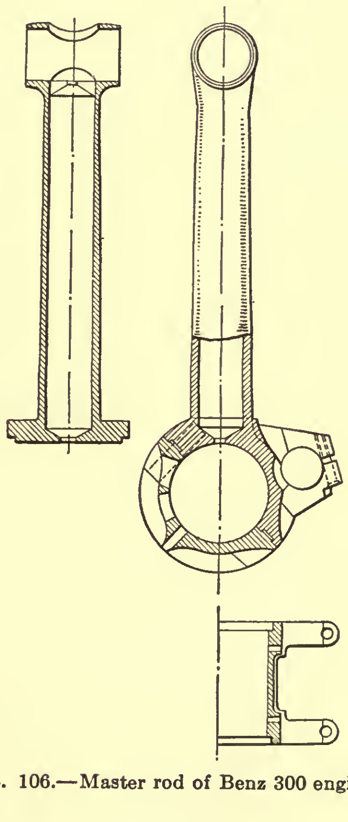


FIG. 106.—Master rod of Benz 300 engine.

master rod is shown in Fig. 106 for the Benz 300, 60-deg. Vee engine. The rod is tubular; the pin for the small rod is held by two clamp screws. In the Renault engine (Fig. 107) the pin for the small rod is of the same diameter as the piston pin so that both ends of the small rod are alike.

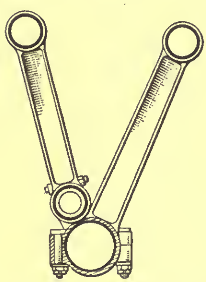


Fig. 107.—Articulated connecting rods of Renault engine.

In W engines the articulated rod is most common. The Napier "Lion" uses a central master rod, on each side of which is mounted an articulated rod (Fig. 108) carried on pins fixed in lugs integral with the big end of the master rod. The main rod is of I-section; the side rods are tubular and carry bronze bushings at both ends. Each side-rod pin is tapered at one end, fits into a tapered hole in the corresponding lug, and is drawn up tight by a bolt screwing into the pin at the taper end.

In the Lorraine W engines the two outer rods bear on the cylindrical outer surface of the big end of the master rod, the

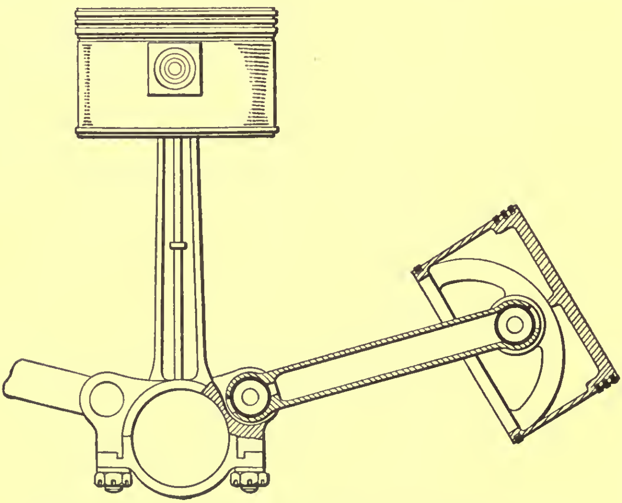


Fig. 108.—Articulated connecting rods of Napier "Lion."

bearing slippers covering less than half the circumference. Two circular steel rings hold the two halves of the big end of the master rod together and hold the slippers of the outer rods to the outer surface of the master rod.

The articulated arrangement of connecting rods suffers from some disadvantages as compared with an arrangement in which the connecting rods are always radial to the crankpin. The short rod is materially shorter than the master rod—usually at least 20 per cent shorter—and consequently causes greater angularity of that rod and increased side thrust in the cylinder. The explosion pressures transmitted along the short rod do not act directly on the crankpin but impose stresses on the master rod which under unfavorable conditions may be serious. For example, with a 90-deg. Vee and with explosion pressure reached 30 deg. before dead center in the short-rod cylinder, the force acting on the master rod *CD* (Fig. 109) would be directed along the line *AE*. The reaction of this force at *C* can be readily found and, treating the master rod as a cantilever loaded with this reaction force at *C* and held on the crankpin, the stress at any section of the rod can be ascertained.

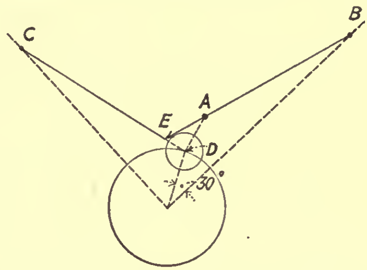


Fig. 109.—Diagram of articulated connecting rods.

Connecting rods up to the present time have always been made of steel. The use of forged aluminum rods would materially reduce the weight of the reciprocating parts and the bearing pressure at the big end.

Crankshafts.—Crankshafts are made of alloy steel (nickel or chrome-vanadium) and are usually forged in one piece. An exception to this is the Bugatti eight-throw shaft which is made in two lengths. In airplane practice there are usually bearings on both sides of each throw; this gives great stiffness to a light shaft. The arrangement common in automobile engines of two or more throws between main bearings has been employed by the Sturtevant, Thomas-Morse and Duesenberg engines and is still used in the Curtiss K engines (see Fig. 55). The long crankarms (between the first and second, and between the fifth and sixth cranks) of this engine have centers of gravity which do not coincide with the axis of the shaft (as in four-cylinder engines) and consequently produce an unbalanced moment about the crank axis. This can be balanced by a counterbalance weight between the two center crankpins which are in line. The addition of such a counterbalance weight sets up an undesirable bending moment on the long central crankpin and also puts more load on the two central main bearings, which have to resist the moments created by these unbalanced masses. To eliminate these objectionable conditions it is desirable to balance directly the masses of the two long crankarms. This is accomplished in the Curtiss K engines (Fig. 55) by applying balance weights directly to the long crankarms. To obtain the greatest possible balancing effect with the least weight, aluminum spacers are inserted between the steel balance weights and the crankshaft, the balance weights being held to the crankshaft by steel bolts.

The crankshaft and pins are always made hollow and holes are drilled through the crank webs for oil passages connecting the hollow crankpins and journals. The open ends of the shaft and crankpins are plugged by screw plugs (Hispano-Suiza) or by discs or caps which are expanded or brazed into place, or in some cases (Liberty, Siddeley "Puma") are held in position by bolts which tie together a pair of caps. In this last case the bolts can be used to obtain rotational balance, the method being to use special bolts thickened in the middle. The caps in different constructions are of duralumin, gun metal, steel and other metals and may be used with or without gaskets.

Main **bearings** and crankpin bearings are nearly always of babbitted bronze. Occasionally a ball bearing is used at one end; at the rear end in the Hispano-Suiza; at the front end in the Fiat. In the Napier "Lion" with three cylinders on each crankpin and with a heavy big end, the main-bearing pressures are very high and roller bearings are used throughout. Double-thrust ball bearings are usual and are placed just behind the propeller hub.

The pressure on main bearings is high and demands considerable oil circulation, not only for lubrication but also for cooling. The babbitt tends to flake off unless the bronze has been tinned before casting the babbitt, in which case a perfect bond can be obtained; mechanical holding of the babbitt by holes, dovetails or screw threads is generally found unsatisfactory.

The **bearing pressure** in a six-throw, seven-bearing crankshaft is greatest at the center main bearings because the crank throws on the two sides of it are in line so that the dynamic loads imposed on the two cranks are in phase. Consequently the center main bearing is often made longer than the intermediate bearings to diminish the intensity of pressure. In Rolls-Royce and Fiat engines the center bearings are 60 per cent longer than the intermediate bearings. In the Liberty engine the maximum load on the center main bearing is 7,700 lb. or 1,675 lb. per square inch of projected area; the mean unit bearing pressure is 1,265 lb. per square inch. As the rubbing velocity is 19.5 ft. per second, the friction work is $F = f \times 19.5 \times 1{,}265$ ft.-lb. per second, where f is the coefficient of friction. On the intermediate main bearings of this engine the maximum load figures out as 7,250 lb. or 1,580 lb. per square inch of projected area; the mean unit bearing pressure is 700 lb. per square inch. The end main bearings receive loading on one side only and show a maximum load of 4,025 lb. or a maximum unit pressure of 815 lb. per square inch and a mean unit pressure of 610 lb. per square inch.

The crankpin bearing pressure for the Liberty engine has a maximum value of 4,980 lb. or 932 lb. per square inch of projected area; the mean unit bearing pressure is 642 lb. per square inch. The crankpin pressures used in the German engines are somewhat lower, ranging from a mean unit bearing pressure of 402 lb. per square inch in the Austro-Daimler to 585 lb. per square inch in the Maybach. The crankpin pressure is mainly due to inertia and centrifugal forces—the gas pressures have comparatively little effect. This is evidenced by the fact that the wear on crankpins bearings is on the side remote from the piston, that is, on the side subjected only to inertia and centrifugal forces.

Crankshafts are subjected to stresses which vary rapidly in sign and magnitude and consequently are especially liable to fail from fatigue of material. The weakest point is generally at some place where there is a sudden change in cross-section and poor distribution of stress. It is particularly important that the fillets at the junctions of the crankpins and journals with the crank webs should be of adequate size. Tests to determine the desirable size of the fillet have been conducted recently in England; they show that the steel is materially weakened if the fillet is less than 3/8 in. radius.

In the discussion of torque on page 47 it has been shown that the maximum torque at the propeller end of the crankshaft of a six-cylinder engine is actually less than the maximum value at the rear crankpin. Consequently there is no need for any increase in diameter of the crankshaft from rear to front in that case. The free end of the crankshaft is subjected to much more severe conditions than the propeller end. The free end is, as it were, wound up when the maximum torque is applied to it and released when the torque diminishes. At certain speeds this alternate winding up and release may coincide with a natural period of vibration of the crankshaft, and in that case the shaft will vibrate excessively and the reciprocating masses attached to it will also vibrate and impart their vibration to the whole structure. Such **torsional vibration** could be reduced by the use of a flywheel on the free end. It has given much trouble in various six-cylinder engines and has been largely responsible for the failure of engines with eight crank throws.

With six-cylinder engines of 200 to 300 h.p. the freely vibrating shaft has a frequency which is usually about 6,000 vibrations per minute; for four-cylinder engines this frequency is higher, and in single-crank radial or rotary engines it may be as high as 20,000. The period of vibration can be determined by striking a series of light blows at regular intervals; the vibrations will increase markedly when the frequency of the blows coincides with the natural period of the shaft. In a six-cylinder engine the impulses are three per revolution so that the dangerous speed for a shaft with vibration frequency is 6,000 per minute of $6{,}000 \div 3 = 2{,}000$ r.p.m. The next most dangerous speed would be 1,000 r.p.m. With eight-cylinder engines the dangerous speeds would be $\frac{1}{4}$ and $\frac{1}{8}$ the vibration frequency.

The addition of **counterweights** to the crankshaft as a means of obtaining rotary balance is of value mainly in reducing bearing pressures.* The centrifugal force arising from unbalanced rotating weights acts radially from the center and produces considerable bearing pressures. Under airplane-engine conditions a lower total weight is obtained by omitting counterbalances and giving the bearing sufficient area and stiffness to support the centrifugal forces. With higher speeds of rotation the need for counterbalance weights increases.

Crankshafts are drop forgings and are usually made in dies when the quantity warrants it; in other cases they are cut from large billets. The dies may be made of cast iron when the number of forgings required is small; for quantity production they are of steel.

Strength of Shafts.—Shafts should be designed for their strength in shear. For a solid circular shaft of diameter d in.

* See G. D. Angle, *Aviation*, June, 1919.

subjected to a bending moment M in pound-inches and a torsion T, also in pound-inches, and with a maximum permissible intensity of shearing stress at the outer surface of the shaft of f lb. per square inch,

$$d^3 = 5.1\frac{\sqrt{M^2 + T^2}}{f}$$

For a hollow shaft of outside diameter d_2 and inside diameter d_1, the equation becomes

$$\frac{d_2^4 - d_1^4}{d_2} = 5.1\frac{\sqrt{M^2 + T^2}}{f}$$

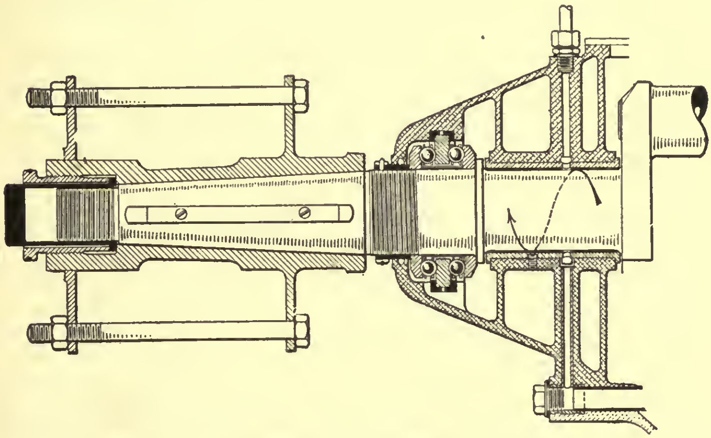


FIG. 110.—Propeller hub of Hispano-Suiza engine.

With $M = 24{,}500$, $T = 54{,}000$, and $f = 16{,}000$, these equations give $d = 2\frac{5}{8}$ in. and if d_2 is assumed to be 3 in. d_1 is 2.275 in. Under these conditions the hollow shaft will weigh 56 per cent as much as the solid shaft. A hollow shaft of still larger outside diameter would be lighter and stiffer but would require larger and heavier bearings and would result in increased rubbing velocities at the bearings and increased friction.

Propeller hubs in American practice are mounted on a tapered extension of the crankshaft. The Hispano-Suiza hub (Fig. 110) is a good example of standard practice. It is keyed to the engine shaft, which is given a taper of 1 in 10, and is threaded at the end to receive a long nut which is used for forcing the hub on the taper. The inner flange is integral with the tapered hub; the outer flange has splines which fit in grooves on the outer end of the hub and permit axial movement of about 1 in. for adjustment to the thickness of the propeller, which is held between the two flanges. Eight bolts hold the flange and the propeller together. Rotation of the hub on the engine shaft is prevented by a key. The long nut is held in position by a locknut or a locking pin.

The Benz engine (Fig. 111) employs a hub which is bolted to a flange at the end of the crankshaft and has an outer flange which fits on the splined end of the hub.

Crankcases.—The crankcase has to serve several functions: it has to tie various parts of the engine together; it has to withstand stresses due to gas pressures, and bending moments due to unbalanced forces; it contains the lubricating oil; it supports

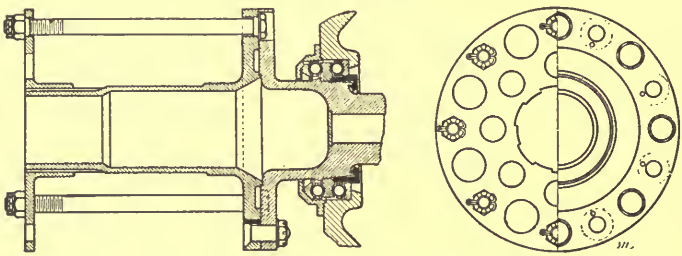


Fig. 111.—Propeller hub of Benz engine.

various auxiliaries; and it has to support the engine as a whole. The stresses in the crankcase are chiefly of the two kinds suggested above. The explosion pressure, acting on the cylinder head, puts the cylinder under tension and this should be supported as directly as possible by connecting members from the cylinder to the corresponding lower main crankshaft bearings. In vertical engines this can be accomplished very satisfactorily by the use of through bolts from the lower cylinder flange to the lower bearings (see Fig. 112), using transverse webs in the upper crankcase as distance pieces. In Vee and W engines this construction is not possible and it is necessary to fasten the cylinders and the lower bearings to the upper crankcase, and transmit the tension through the transverse webs to the lower bearings. The lower half of the crankcase is sometimes cast as a unit with the lower bearings but this practice has nothing to recommend it. The assembly of cylinders and upper crankcase should sustain all the stresses due to gas pressures.

The unbalanced centrifugal and inertia forces acting through the main bearings subject the crankcase to bending moments which change continuously in direction and magnitude. It is necessary that the crankcase should have sufficient stiffness to withstand these bending moments without objectionable deflections. For this purpose a webbed box structure has been found most satisfactory. It is easily possible to obtain the necessary stiffness by utilizing the upper crankcase only as a stressed member. The lower crankcase is preferably used only as an oil

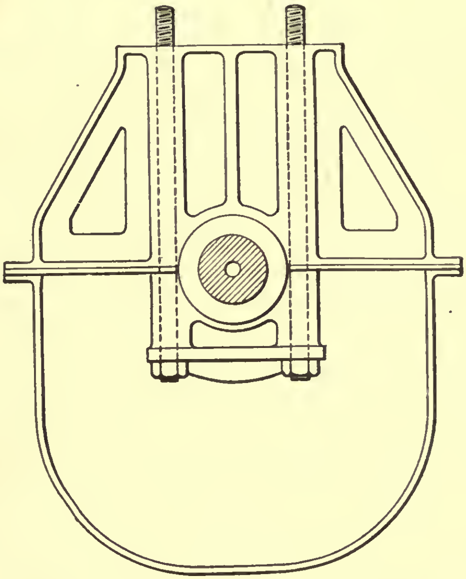


FIG. 112.—Transverse section of crankcase.

container and a support for oil pumps and other auxiliaries. This practice can be seen in the Hall-Scott L (Fig. 64), Bugatti (Fig. 67), Curtiss K and OX (Figs. 55 and 59), Napier "Lion" (Fig. 72), Maybach (Fig. 81), and Benz engines (Fig. 78). In other engines the lower crankcase carries also the lower halves of the end main bearings, as in the Hispano-Suiza (Fig. 51) and Siddeley "Puma" engines (Fig. 9). These arrangements permit of easy accessibility. In the Liberty engine (Fig. 47) the lower crankcase has transverse webs and the crank chamber is divided into six separate chambers; a similar construction with double transverse webs is employed in the Fiat engine (Fig. 76).

The transverse webs are often cut away in places for lightness. Aluminum alloy is universally used for crankcases.

The lower crankcase serves as an oil sump. The earlier practice of keeping a considerable body of oil (wet sump) in the bottom of the crankcase is now being superseded by the dry sump which is kept drained by a scavenger pump or pumps. Wet sumps are shown in the Benz engine (Fig. 78), Hispano-Suiza (Fig. 51), Hall-Scott L (Fig. 64), and Curtiss OX engines (Fig. 59). In the last case the sump is separated by drainage plates from the rest of the crankcase. The advantage of the dry sump is in avoiding drowning the cylinder with oil in case the engine operates momentarily upside down or in any posture approximating that position. The drainage point of the sump is usually in the middle, but in some cases the scavenger pumps take oil from both ends (Liberty, Napier "Lion"). Oil cooling is carried out in the lower crankcase in the Austro-Daimler engine by casting outside cooling ribs running longitudinally along the bottom of the crankcase and attaching a sheet of aluminum in such a way as to form an air duct along the whole underside of the engine. In the Bassé-Selve engine an oil cooler with air tubes is fastened to the bottom of the crankcase but has no direct communication with the inside. In the Curtiss K engine (Fig. 55) oil cooling is effected inside the crankcase by the jacket water on its way to the pump; this arrangement serves also to heat up the oil quickly after starting the engine and puts the lubrication system into normal operation earlier than would otherwise be possible.

German airplane engines usually have provision for air cooling of the crankcase. In the Benz engine (Fig. 77) the supporting webs for six of the main bearings form air passages transversely across the engine. Two of these serve as air intake passages to the two carburetors, which are thereby supplied with heated air. In addition the lower crankcase is traversed by 18 aluminum tubes, 30 mm. in diameter; air is scooped into the tubes through an aluminum louvered cowl on one side of the engine and discharged through a reversed cowl on the other side.

CHAPTER VII

VALVES AND VALVE GEARS

Location of Valves.—The diversity of valve locations which characterizes automobile practice is not found in modern airplane engines. L-head and T-head arrangements have been supplanted by overhead valves which permit the simplest form of cylinder head and water jacket, shortest and most direct passage of the gases, and, with overhead camshafts, a considerable simplification in the valve gear and a reduction in the number of rubbing or contact points. The valves may either be seated in a flat head in which case the stems are parallel to the cylinder axis or the head may be domed, in which case the valve stems are inclined to the cylinder axis.

Valve Lift.—Valves are always of the poppet type with bevelled seats. They are opened by cams which operate either directly or indirectly; they are closed by springs. The valve (Fig. 113) has a **face** which is usually about 25 per cent wider than the **seat** on which it closes and a **stem** which passes through a long guide (often provided with a bushing) and which connects with the valve by a rounded fillet. The bevel of the valve face is usually 45 deg. but 30 deg. is sometimes used. The width of the valve face must be small to ensure gas tightness and is usually about one-fourth the lift of the valve. The width of the valve seat is usually less than 0.1 in. The free area for the passage of the gas through the fully opened valve may be taken approximately as πdh where d is the smallest diameter of the bevelled valve face and h is the lift. This area should not be greater than the free opening through the valve seat. Neglecting the area occupied by the valve stem, $\pi dh = \frac{\pi}{4}d^2$, or $h = \frac{1}{4}d$, gives the lift which

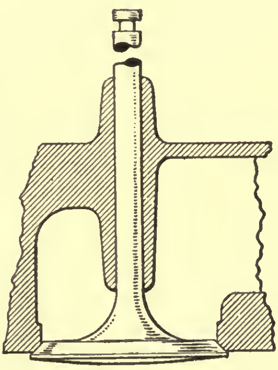


FIG. 113.—Typical airplane engine valve.

makes the gas passage area equal to the free opening through the valve; usually h varies from one-fifth to one-sixth of the outside valve diameter. Values will be found in Table 4.

With a valve lift of one-quarter of its diameter the gas flow for a given pressure drop is found by experiment to be about 67 per cent of the flow through an unobstructed port[1]; with a lift of one-half the diameter this is increased to from 80 to 90 per cent. These "coefficients of efflux" are found to be the same for all pressure drops, and for valves of different sizes at equal lifts expressed in per cent of their respective diameters. The experiments were carried out with continuous flow which presumably would give results differing from those actually occurring under the operating conditions of intermittent flow. The earlier investigations of Lucke[2] indicate coefficients of efflux lower than those given above; the variation with lift is probably of the right order of magnitude.

The volume of gas passing the inlet valve per unit of time is approximately equal to the piston displacement in that time; the volume of gas passing the exhaust valve is from two to three times as great. If the mean piston speed during the suction stroke is s in. per second and the piston diameter is D in. the mean gas velocity V in feet per second past the valve is given by

$$\frac{\pi}{4}D^2 \cdot s = 12V\pi dh$$

or

$$V = \frac{D^2 s}{48dh} \text{ ft. per second}$$

where d and h are in inches. This velocity is approximate, as the equation assumes the cylinder to fill with a mixture at atmospheric pressure and temperature. As the piston speed is varying throughout the stroke, the gas velocity will vary and will have a maximum value which is nearly twice the mean value. For the Liberty engine the mean gas velocity is 189 ft. per second; for most engines it varies from about 150 to 200 ft. per second.

The **pressure drop** past the valve to obtain this velocity can be obtained with sufficient accuracy from the equation $V^2 = 2gh$, where h is the pressure drop measured in feet of air. To convert this to a pressure drop, i, measured in inches of water, using the

[1] Lewis and Nutting, *4th Ann. Report, Nat. Adv. Comm. Aeronautics*, 1918.

[2] *Trans. A. S. M. E.*, 1905, Vol. 27.

conversion factor, 1 in. of water = 5.2 lb. per square foot, we have

$$h = 5.2 \times i \times v$$

where v is the volume of 1 lb. of the explosive mixture. The quantity v is obtained from the perfect gas equation

$$v = RT/p = 52\ T/p.$$

At ordinary atmospheric pressure and temperature, v has a value of about 13.

$$V^2 = 2g \times 5.2 \times 13 \times i$$

and

$$i = \frac{V^2}{4{,}350}$$

For gas velocities from 150 to 200 ft. per second the corresponding pressure drops are from 5.2 to 9.2 in. of water.

The pressure drop through the valve is important as affecting the pressure in the cylinder at the end of the suction period and, thereby, the volumetric efficiency. This pressure is controlled also by the frictional resistances to the flow through carburetor, manifold and gas ports, by the time and rate of valve opening and closing, and by other factors. At midstroke, when the piston velocity is a maximum, the gas velocity past the valve will be a maximum; if its value is twice the mean velocity the pressure drop will be four times the mean value. In the Liberty engine the pressure drop corresponding to a mean velocity of 189 ft. per second is 8.2 in. of water; the pressure drop past the valve at midstroke would then be about $4 \times 8.2 = 33$ in. of water. Frictional resistances will increase this quantity. During the latter half of the stroke the quantity of gas passing the valve will be greater than in the first half on account of the existence of this greater vacuum. That is, during the first half of the stroke a vacuum is being created in the cylinder; during the last half this vacuum is being filled up. Near the end of the stroke the valve is closing which cuts down the area for gas flow and limits the filling up process. In most engines (see p. 175) the final closure of the valve does not occur till after the dead center and the additional time so obtained for the admission of the charge has important results in increasing the weight of charge admitted. The continued flow of gas into the cylinder past the dead center position is due not only to the existence of a vacuum there but also to the inertia of the column of gas in the induction system. It is not desirable to delay the final closure of the valve long enough to establish atmospheric pressure in the cylinder; this procedure would not increase the volumetric efficiency because of the diminished volume of the charge resulting from the return of the piston. The actual closing of the valve is sometimes delayed till the piston has returned 10 per cent of its stroke so that the volume of gas at the beginning of compression is correspondingly reduced and the compression ratio is less than the cylinder dimensions would indicate.

The gas flow area past the valve has been seen to be proportional to dh, or since h is proportional to d, the area is proportional to d^2. The larger this area the lower will be the gas velocity, and the less the frictional resistances and the pressure drop.

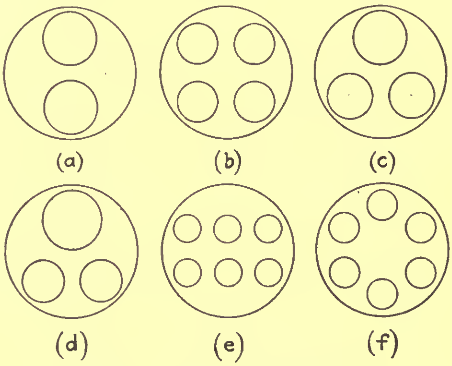


FIG. 114.—Valve arrangements.

With flat-headed cylinders and with no increase in diameter in the combustion space the valve diameter must be considerably less than half the cylinder diameter; in the Hispano-Suiza 180 it is 42 per cent, in the Hispano-Suiza 300 it is 40 per cent. With enlarged heads the ratio of valve diameter to cylinder diameter can easily be made 50 per cent as in Liberty engines, or even 55 per cent as in the Curtiss VX engine.

To obtain a larger valve area, **dual valves** may be employed. The Benz 200 has dual valves of 1.693 in. diameter as against one valve of 2.205 in. diameter in the Hispano-Suiza 300 engine of the same cylinder diameter; the possible gain in valve area is 17½ per cent. A comparison is shown in Fig. 114 of the maximum valve areas for the Hispano-Suiza 300 engine, *a* with single valves, *b* with dual intake and dual exhaust valves and *c* with dual intake and single exhaust. The single valves are of the same diameter as in the actual engine; the width of bridge between valves is kept constant and the clearance from the cylinder walls is the same as in the actual engine. The maximum dual valve area in *b* is 19 per cent and in *c* is 45 per cent greater than the single valve area in *a*. In Fig. 114 *d* the single inlet valve is shown of the same area as the dual exhaust valves; with this arrangement both inlet and exhaust valve areas are increased 23 per cent as compared with *a*. The increase in valve area obtainable by the use of dual valves becomes greater as the cylinder diameter increases since it is not necessary to change the width of bridge between valves or the cylinder wall clearance with change in cylinder diameter. For example, with a cylinder diameter of 7 in. and bridge width 0.69 in. and wall clearance 0.206 in. the use of dual valves increases the area 24 per cent as compared with 19 per cent gain in the Hispano-Suiza 300 with cylinder diameter 5.51 in. and the same bridge width and clearance.

A further development along the same lines is the use of **triple** inlet and exhaust **valves** as in the 800-h.p. Sunbeam Sikh engine. This arrangement leads to a decrease in valve area except for large cylinders. With a 7-in. cylinder and the same bridge width and clearance as above the decrease in valve area with triple valves in line (Fig. 114 *e*) is 35 per cent as compared with dual valves; with the valves arranged in a circle (Fig. 114 *f*) the decrease in valve area is 16 per cent. The latter arrangement would require a more complicated valve gear than is necessary with the three valves in line. With larger cylinder and smaller bridge width the comparative showing of the triple valves would improve.

The comparisons just presented between multiple and single valves are based on the assumption of the same lift expressed in per cent of diameter. If dual valves are used and the actual lift is kept the same as for the corresponding single valve, the gas flow through the multiple valves will be increased by about 20 per cent over the figures given; for example, the dual valves of Fig. 114 *b* will increase the gas flow about 40 per cent above that through the single valve for a given pressure drop. The diameters of the dual valves can be reduced, without decreasing the gas flow below that through a single valve, by maintaining the same lift as for a single valve. For example, a 2.5-in. valve with 25 per cent lift (0.625 in.) gives the same flow as two 1.75-in. valves with 25 per cent lift (0.44 in.) or two 1.5-in. valves with

0.625-in. lift (42 per cent). This last arrangement has certain advantages; the area of valve presented to gas pressure is only 72 per cent of the area of the single valve and the weight of the two valves will be only 56 per cent of the weight of the single valve, assuming the weights to vary as $d^{2.5}$. With valve spring tensions proportional to valve weights the power required to operate these dual valves will be less than half the power required to open the single valve. Other advantages are that a larger proportion of the cylinder head can be jacketed because it is not occupied by the valves; that the valve cooling will be better because the circumference of the two valves is 20 per cent greater than that of the single valve and the distance travelled by the heat is only 60 per cent as great; and the distortion of the valve will be less in consequence both of lower temperature and smaller diameter.

The area of the **exhaust valve** opening affects the volumetric efficiency since it determines the pressure of the residual gases in the combustion space at the time of opening of the inlet valve and determines the back pressure on the piston during the exhaust stroke. The main problem of the exhaust valve is heat dissipation. The exhaust valve is heated during the explosion stroke and by the exhaust gases as they pass out. The greater the velocity of the gases the greater is the heating of the valve. Heat abstraction from the valve is principally by conduction to the seat and thence to the jacket water, but is also by conduction to the stem and thence to the guide. The inlet valve gives no trouble from this source as it is cooled by the entering charge. Increase of exhaust valve area by increasing its diameter is objectionable because it results in increased temperature of the valve. The heat received by the valve is approximately proportional to the square of its diameter while the area of the heat-abstracting seat is proportional to the first power of the diameter. Furthermore, as valve diameter increases the mean distance the heat has to travel increases and this results in increased valve temperature and also in valve warping. The maximum practicable size of exhaust valve seems to be about 3 in. diameter. A limit to valve size is also set by the valve weight, which increases about as $d^{2.5}$ while the area increases as d^2. As the valves are closed by springs, the dimensions of the springs have to increase rapidly with increase of valve diameter and lift in order to give sufficiently quick closure. The desirable method of increasing exhaust valve area is therefore by the use of multiple valves and not by increase in diameter of a single valve. The arrangement of Fig. 114*b* with dual exhaust valves and single inlet valve, is often employed (Siddeley "Puma," Benz 300, A B C Dragon-fly), while the reverse arrangement of two inlets and one exhaust valve is seldom used (Bugatti). Ordinarily, if dual valves are used, they are used both for inlet and exhaust.

In addition to providing the maximum free area through the valves it is important that the form of the passage through which the gas is flowing should be such as to offer minimum resistance and as far as possible to permit smooth flow without the formation of eddies.

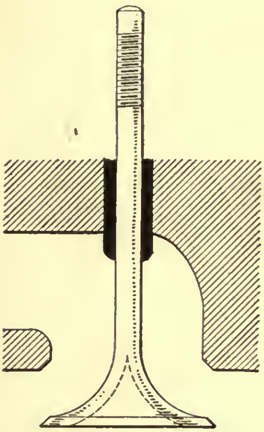


FIG. 115.—Trumpet-shaped valve.

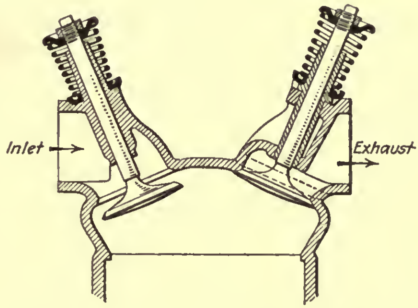


FIG. 116.—Valves of Bassé-Selve engine.

Especially is this true for the inlet valve as it will influence materially the volumetric efficiency of the engine. The limitations inherent in an engine as light and compact as an airplane engine will generally prevent the adoption of ideal forms of passage, but some improvements over common practice (as represented in Fig. 113) are possible. One of these is in the shape of the valve. If the valve is formed with a trumpet head which flows with a large radius into the stem as in the Siddeley "Puma" inlet valve (Fig. 115), the gas flow lines will be considerably improved; such valves should be hollowed out to reduce the weight. The approach of the gas to the valve is usually smoother when the valves are inclined than when they are vertical; compare, for example, the Bassé-Selve (Fig. 116) with the Curtiss K (Fig.

87), both of which are good examples of smooth passages without sudden enlargements. In the Liberty engine (Fig. 91) the form of the gas passages is such as to set up eddies. On the exhaust side the need for adequate water-jacketing of the valve seat and guide will generally lead to a less favorable form of gas passage than on the inlet side. This is especially noticeable in the Bassé-Selve engine (Fig. 116), in which the water jacket is carried all round the whole length of the valve stem guide; in most engines such complete jacketing is not attempted.

The cross-section **area of inlet pipes and ports** should be approximately the same as the valve area so as to avoid loss of head due to change of cross-section. With an engine using 0.6

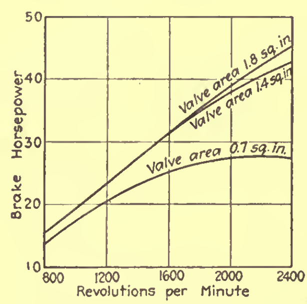


FIG. 117.—Variation of engine power with valve lift.

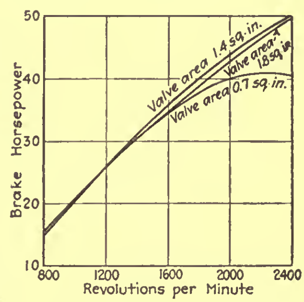


FIG. 118.—Variation of engine power with valve diameter.

lb. of fuel per horse-power hour and 15 lb. of air per pound of fuel, the volume of charge entering the cylinder will be approximately 120 cu. ft. and of the exhaust gases 250 to 300 cu. ft. per horse-power hour. With a velocity of 150 ft. per second past the inlet valve the inlet pipe cross-section is $4 \times \frac{120}{3{,}600} \times 144 \div 150 =$ 0.128 sq. in. per horse power. Good engines show usually from 0.14 to 0.16 sq. in. per horse power for the inlet pipes and ports. Exhaust pipes and ports are usually of the same size as inlet pipes and ports, but are occasionally larger. In the Austro-Daimler engine inlet pipes are 0.126 sq. in. per horse power and exhaust pipes 0.18 sq. in.

The **effect of valve lift on engine capacity** is shown in Fig. 117, which is plotted from Pomeroy's[1] test results. The engine was 90 × 120 mm. and had an adjustable inlet valve lift. The

[1] *The Automobile Engineer*, Feb., 1919, p. 44.

valve diameter was 1.75 in.; the valve areas (πdh) used in the test were 1.8, 1.4 and 0.7 sq. in. As indicated in the figure the b.h.p. was the same for the two larger areas up to 1,500 r.p.m. above which speed the larger area showed its superiority. The lowest lift showed marked inferiority.

Tests of another engine of the same size with constant lift and with the valve areas kept the same as in the previous engine, by fitting different diameters of valves ($1\frac{9}{16}$, $1\frac{3}{8}$, and 1 in.), gave the results shown in Fig. 118. It is noteworthy that the performances of these three valves were practically identical up to 1,600 r.p.m., above which speed the smallest valve showed inferior results and the intermediate size valve showed best results. It is seen by comparing Figs. 117 and 118 that valve area alone is not important; it is necessary to know the lift-diameter ratio also. The highest curve in Fig. 118 is obtained with a lift of 23.6 per cent of the diameter; in Fig. 117 the highest curve has a lift-diameter ratio of 18.7 per cent, but it is probable that still better results would have been obtained with a higher lift.

Valve Materials.—Inlet valves in airplane engines under normal operation may reach temperatures of over 1,100°F.; exhaust valves may go to 1,600°F. or higher. The heat received by the head of an exhaust valve is dissipated in three ways: (1) by conduction down the stem to the guide, (2) by direct radiation from the back surface of the head and (3) by direct conduction from the face to the valve seat. The last of these is by far the most important. To be effective it is essential that the valve should have good metallic contact with its seat throughout the whole of the explosion stroke. If, through valve warping or the lodging of scale on the seat, there should be any leakage of gas past the valve, there will be rapid heating at the place where the leakage occurs and the valve will burn away at that place. Another prolific cause of valve burning is persistent preignitions in the cylinders; it is found that valves which stand up satisfactorily under normal operation fail very rapidly when persistent preignitions occur; with such preignitions the temperature of the exhaust valve may rise to 2,100°F. If the exhaust ports are so designed that the exhaust gases play directly on the neck of the valve this may become highly heated and may actually supply heat to the valve head instead of taking it away; in such a case overheating of the valve is likely to occur. It is also important that the valve guides should be efficiently water-cooled and should not project into the exhaust pocket so as to be heated directly by the exhaust gases. A final cause of overheating the exhaust valve is the use of an overrich mixture which may be still burning during the exhaust stroke.

An interesting suggestion for valve cooling is the use of a hollow stem into which is put a small amount of mercury before plugging. The liquid mercury in contact with the hot center of the valve head is vaporized and is condensed again in the upper part of the stem. The mercury thus acts as a heat carrier abstracting from the valve head its latent heat each time it is vaporized. The vapor pressure of mercury at 820°F. is 50 lb. per square inch.

The principal types of **valve failure** are (1) elongation of the stem, (2) distortion of the head, (3) cracks in the valve face, (4) wear of the stem, (5) wear of the foot, (6) burning of the head, (7) scaling and (8) breaking due to self-hardening. Elongation of the stem results either from the use of a steel of insufficient strength at the working temperature or from overheating of the stem. Distortion of the head occurs usually when proper heat treatment has not been given to the valve forging before machining; in other cases unequal heating or softening under the action of high temperature may be the causes. Cracks come usually from cracks in the steel from which the valves are made; they are fairly common and are dangerous as they may result in the breaking away of a section of the valve. Wear on the valve stem occurs usually in rotary engines which produce a side pressure due to the inertia of the valve. Wear of the valve foot results from the hammering of the tappet or the wipe of the cam; it is diminished by hardening the foot or by the use of a cap. Burning is due to overheating.

A steel to be satisfactory for exhaust valves in airplane engines should have the following properties as stated by Aitchison.[1]

1. The greatest possible strength at high temperatures.
2. The highest possible notched bar value (resistance to impact).
3. The capacity of being forged easily.
4. The capacity of being manufactured free from cracks.
5. The capacity of being easily heat-treated.
6. The least possible tendency to scale.
7. The ability to retain its original physical properties after repeated heatings for prolonged periods.
8. Freedom from liability to harden on air cooling.

[1] *The Automobile Engineer*, Nov., 1920.

9. Freedom from distorting stresses after heat treating.
10. Hardness to resist stem wear.
11. Capacity of being hardened at the foot.
12. Reasonable ease of machining.

The best steels for exhaust valves are in five classes:

1. Tungsten steel with not less than 14 per cent tungsten and about 0.6 per cent carbon.
2. High chromium steels (stainless steel) with about 13 per cent chromium and about 0.35 per cent carbon.

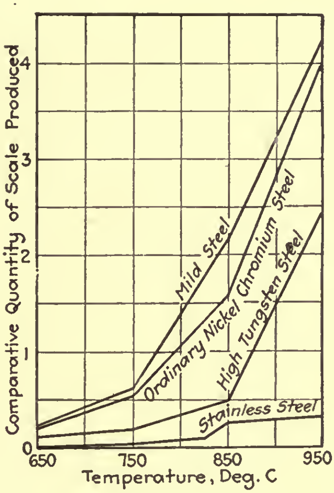


FIG. 119.—Resistance of valve steels to scaling.

3. Steel containing from 7 to 12 per cent chromium and about 0.6 per cent carbon.
4. Steels containing about 3 per cent nickel.
5. Ordinary nickel-chromium steels.

Of these steels the first four are superior to the last. The nickel-chromium steels are difficult to manufacture free from flaws, they tend to harden during the running of the engine, they scale rapidly and they show no superiority at high temperatures over the other steels. The relative resistances to scaling are shown in Fig. 119, from which it is apparent that stainless steel is superior to the others. The tensile strengths of these steels at higher temperatures are given in the following table.

ULTIMATE STRENGTH OF VALVE STEELS, POUNDS PER SQUARE INCH

Steel	Temperature, degrees Fahrenheit	
	1,300	1,650
High tungsten, high carbon	39,600	19,700
High tungsten, low carbon	34,700	14,100
High chromium, high carbon	33,800	16,800
High chromium, low carbon	27,100	10,700
Low chromium, high carbon	41,400	16,800
Low chromium, low carbon	38,000	15,800
3 per cent nickel, high carbon	25,800	10,100
3 per cent nickel, low carbon	21,000	8,700
Nickel chromium	23,500	10,100

The 3 per cent nickel steel is much cheaper than the others but is markedly inferior in tensile strength at high temperatures and consequently should be used only on inlet valves or for the exhaust valves of rotary engines. The high chromium (stainless) steel is highly resistant to scaling and, if of low carbon content, is readily machined but is not easy to forge and is liable to cracks. High tungsten steel retains its strength best of any steel at high temperatures and is fairly resistant to scaling. Exhaust valves which are liable to be subjected to unusually high temperatures should be of tungsten steel; for more moderate temperatures stainless steel will be more durable. Monel-metal valves have been used, and although they have stood up well under test on the Hispano-Suiza engine, they have failed rapidly on the Liberty engine.

Valve Operation.—The valves of modern airplane engines are mechanically operated; automatic action, which is found in some automobile engines and in a few of the earlier airplane engines, must always result in lowered volumetric efficiency and capacity. Actuation of the valves is by means of cams acting either directly or indirectly. The camshafts may be placed near the base of the cylinders and operate the valves through push rods and rocker arms, or overhead camshafts may be used acting on the valves directly or through rocker arms.

A good example of push-rod operation is shown in the Benz 230 engine (Fig. 78), which has separate camshafts for the inlet and exhaust valves, both located in the crankcase; a similar arrangement is used in the Maybach engine (Fig. 81). In Vee engines the usual practice, where push rods are employed, is to have a single camshaft, located in the angle of the Vee, inside the crankcase, carrying inlet and exhaust cams for both rows of cylinders; the Curtiss OX and V2 engines (Figs. 60 and 62) show this arrangement, which is also used on the Benz 300 (Fig. 131). In recent years the tendency has been to do away with push rods and to use overhead camshafts. This last arrangement reduces the weight and complexity of the valve gear, and, in consequence of the smaller number of joints involved, makes for better maintenance of the valve timing. There may be either (1) one camshaft over each row of cylinders acting directly on the valves as in the Hispano-Suiza (Fig. 51), or (2) one camshaft acting directly on one set of valves and indirectly through a rocker arm on the other set as in the Siddeley "Puma" (Fig. 8), or (3) one camshaft acting indirectly through rocker arms on both sets of valves as in the Liberty (Fig. 47), Bassé-Selve (Fig. 129), Bugatti (Fig. 67), Fiat (Fig. 76), Rolls-Royce (Fig. 70), Mercedes, Lorraine-Dietrich, Renault, Austro-Daimler, or (4) two camshafts acting directly on the two sets of valves as in the Curtiss K (Fig. 56) and Napier "Lion" (Fig. 73).

Cams.—The shape of the cam depends on the desired valve movement and on the form and location of the cam follower.

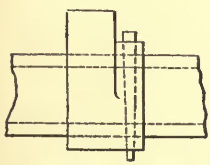


Fig. 120.—Individual cam.

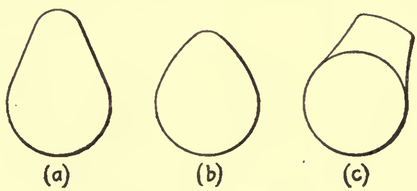


Fig. 121.—Cam forms.

The cams are usually integral with the camshaft, which gives maximum security and accuracy of location, but sometimes they are fastened by taper pins to the hollow camshaft, as in Fig. 120; this arrangement permits more satisfactory hardening of the cams and the replacement of a worn cam but is less secure and may become slack. The cams are sometimes made with convex flanks as in Fig. 121*b*, or with flat surfaces tangential to circular arcs as in Fig. 121*a*, or with flanks that change from concave to convex and a top which is concentric with the camshaft as in the constant acceleration cam of Fig. 121*c*. The cam follower may be flat as in Fig. 122*a*, rounded as in Fig. 122*b*, or a roller as in Fig. 122*c*, and it may be fixed on the end of the valve plunger or it may be mounted on a radius rod as in Fig. 122*d*. With a flat follower a convex flanked cam is used; tangential and constant acceleration cams are used with the other types of follower shown in Fig. 122.

The work which a cam has to do is in three parts. (1) It must overcome the difference of gas pressures on the two sides of the valve. This pressure difference is important only in the case of the exhaust valve, which just previous to opening has a pressure

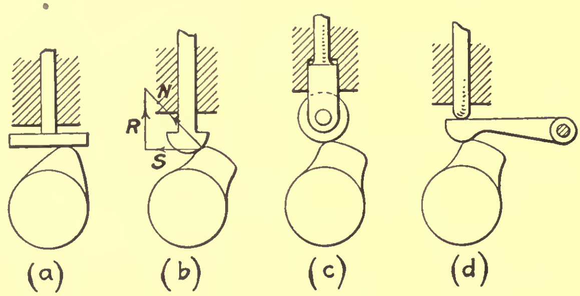


Fig. 122.—Cam followers.

of about 60 lb. per square inch (gage) on one side and atmospheric pressure on the other. With a valve 2½ in. diameter the pressure difference is nearly 300 lb. at the instant of opening and falls rapidly to a negligible quantity. (2) The valve spring is operating at all times to keep the valve closed; the compression on the spring is usually about 50 lb. when the valve is closed (see table 4). The cam must do work in compressing the spring. (3) The moving parts from the cam to the valve, including the valve and spring, must be accelerated and work must be done in giving them the necessary acceleration. The force required to accelerate these parts is determined by the design of cam and follower and by the masses that have to be accelerated.

For maximum volumetric efficiency of the engine the valves should open promptly, should remain wide open as long as possible and then should close promptly. If a valve is to be opened in a given time (number of degrees of crankshaft rotation) the force required to accelerate the moving parts will be kept a minimum by making the acceleration constant and thereby keeping the accelerating force constant. During the opening the moving parts must be first accelerated and then brought to rest; the deceleration is accomplished by the valve spring and, sometimes, if a push rod is used, by an additional spring acting on the push rod. Smooth action will be obtained when the deceleration is constant and has the same value as the acceleration. The cam does not necessarily do any work at all during the decelerating period.

The acceleration and the force required to produce it are readily calculable. Suppose the valve to move from the closed to the fully open position in 60 deg. of crankshaft rotation and that the moving parts are accelerating uniformly for half that period, or 30 deg., and are decelerating uniformly for the next 30 deg. Then at 1,500 r.p.m. the time, t, available for acquiring maximum velocity is $\frac{60}{1,500} \times \frac{30}{360} = \frac{1}{300}$ sec. If the lift is 0.5 in. the distance, d, moved in this time is 0.25 in. and the acceleration, a, is given by $d = \frac{1}{2}at^2$, or $a = 3{,}750$ ft. per second per second. If the weight, w, of the moving parts is 1 lb. and if all of the parts move with the same velocity as the valve, the force required to accelerate the parts will be $\frac{w}{g}a = 110$ lb. The force exerted by the cam will be greater than this by an amount equal to the valve spring compression and, at the instant of opening the exhaust valve, by the gas-pressure difference. With the numerical values given above the force exerted by the exhaust cam at the moment when the valve begins to open must be $110 + 300 + 50 = 460$ lb. As there is always some tappet clearance to permit expansion of the valve stem without forcing the valve to lift, this maximum force occurs a short time after the cam has come into action. The force will diminish rapidly as the gas pressure in the cylinder falls but will tend to increase later with increasing spring compression. During the decelerating period the spring pressure would have to be greater than 110 lb. to bring the valve to rest in 30 deg. of crank rotation. During the closing of the valve the gas pressure difference is absent, the acceleration is due to spring action and the deceleration is brought about by pressure on the cam.

The forces exerted by the cam, which have just been considered, are the radial forces, R, acting along the push rod, or, in the case of overhead camshafts, at right angles to the outer end of the rocker arm. The actual pressure, N, between the cam and its follower acts normal to the surface of contact and will be greater than the radial force throughout the accelerating period. The relation between these two forces is indicated by the triangle of forces in Fig. 122*b*. The side thrust, S, may be troublesome. The quicker the opening of the valve the greater will be the acceleration force, R, and the greater will be the ratio of both N and S to R. In other words, the normal pressure and the side thrust increase much more rapidly than the radial force. The side thrust is particularly objectionable with valve plunger guides as in Figs. 122*b* and *c*; with the arrangement of Fig. 122*d*, or with the cam operating directly on the rocker lever, a rapid opening of the valve can be obtained without trouble from side thrust.

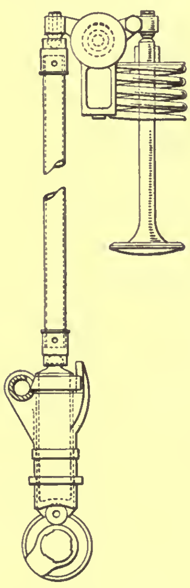


FIG. 123.—Maybach valve gear.

A good example of the constant acceleration type of cam with roller follower is shown in the Maybach engine, Fig. 123. The displacement, velocity, and acceleration curves both for inlet opening and for exhaust closure are given in Fig. 124; it will be seen that the valves open and close rapidly, and remain full open for considerable periods of time. The velocity of the exhaust valve when closing increases uniformly for 60 deg. of crank rotation, then decreases but not quite uniformly for the next 46 deg.; the inlet valve on opening has acceleration which increases for about 48 deg., when maximum velocity is attained, and then comes to rest after 60 deg. more of uniform deceleration. Valve displacements for the whole cycle are shown in Fig. 125; it will be seen that the valves are wide open for considerable fractions of the stroke.

The tangential cam is used in the Liberty engine (Fig. 130) operating on a roller at the end of the rocker arm. With this type of cam, the center of curvature of the highest part of the cam cannot coincide with the center of the camshaft (as in the constant acceleration cam), and consequently the valve cannot stay at its wide-open position. The actual valve lifts are shown in Fig. 126 plotted against crank position, and in Fig. 127 plotted against piston position. The valve opening is not so good as in the Maybach engine, but the forces required to acceler-

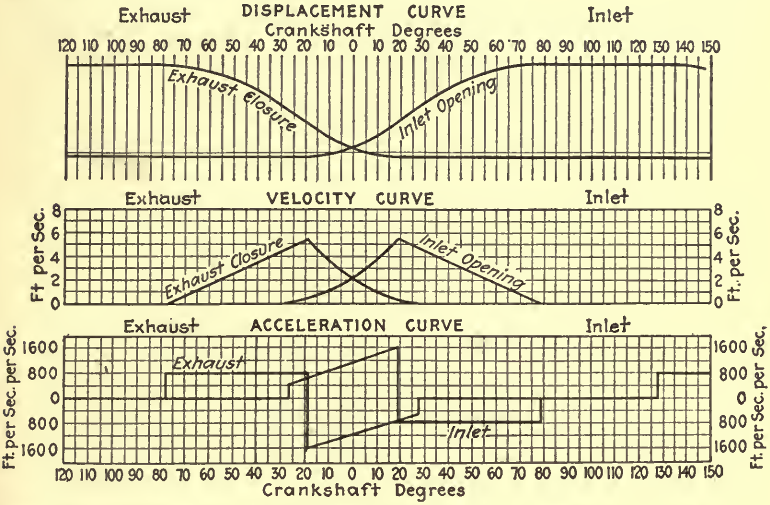


FIG. 124.—Displacement, velocity and acceleration curves for the valves of the Maybach engine.

ate the moving parts are less in consequence of the longer time available for opening or closing the valve; with symmetrical cams this time is one-half the total time the valve is open. In the Maybach engine the intake valve is open for 223 deg. of crank rotation and the valve, while opening, is accelerating for

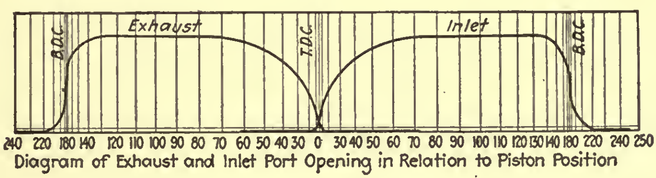


FIG. 125.—Valve openings of Maybach engine.

29 deg.; in the Liberty engine the inlet valve is open for 215 deg. and the valve, while opening, is accelerating for 54 deg. of crank rotation. As the acceleration is inversely as the square of the time taken to lift the valve through a given distance, the force

required to overcome the inertia of the moving valve parts would be 3.5 times as great for an engine using 29 deg. of crank rotation for the valve acceleration as for the same engine, with the same revolutions per minute, using 54 deg. of crank rotation.

Valve Springs.—The function of the valve spring is to decelerate the valve moving parts during the latter half of the valve opening and to accelerate them during the first half of the valve

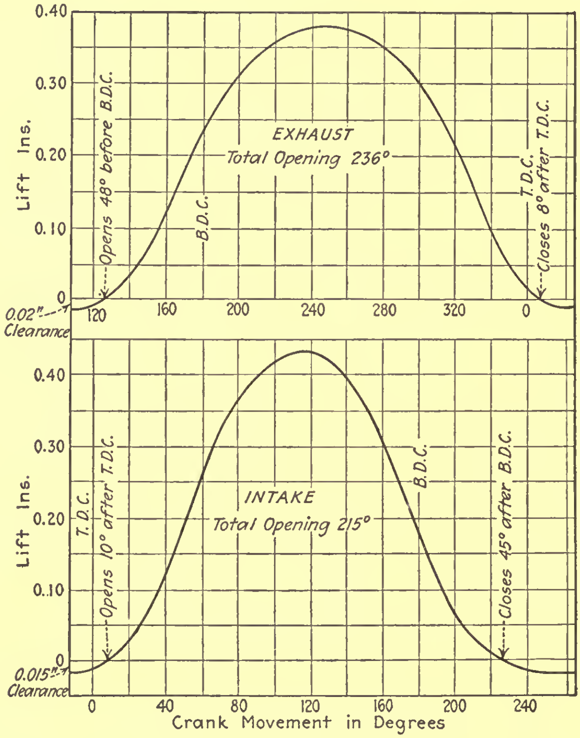


FIG. 126.—Valve lifts of Liberty engine plotted against crank positions.

closure. In addition, the exhaust valve spring must be strong enough to keep the valve closed during the suction stroke when the engine is idling. During idling the pressure in the cylinder may be 10 lb. per square inch below atmospheric pressure, which, with a $2\frac{1}{2}$-in. valve, gives a pressure difference of 50 lb. between the front and back of the valve. The spring pressure required for acceleration is normally in excess of that required to keep the exhaust closed during idling.

Cylindrical helical springs are employed almost universally, but occasionally conical helical springs may be used (Fig. 133), or, when minimum height is required, the rat-trap type of spring, as in the Curtiss engine (Fig. 128). In the larger engines two concentric cylindrical helical springs are used as in the Liberty engine

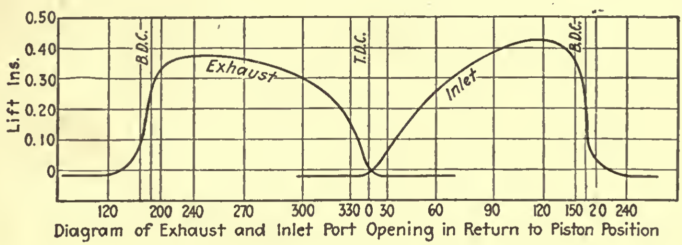


FIG. 127.—Valve lifts of Liberty engine plotted against piston positions.

(Fig. 130) and the Benz 300 inlet (Fig. 131), or even three springs as in the Bugatti exhaust valve (Fig. 67). An advantage of multiple springs is that, in case of breakage of one spring, the valve cannot fall into the cylinder.

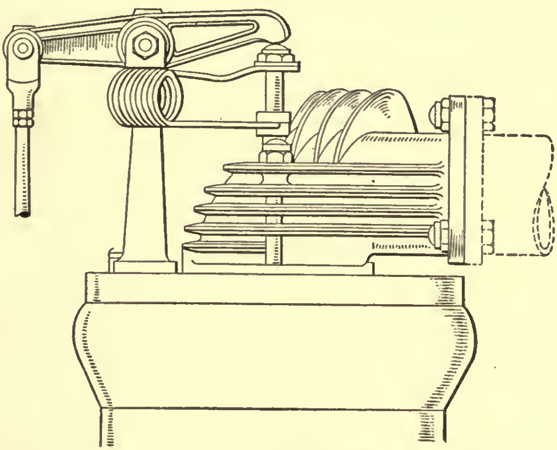


FIG. 128.—"Rat trap" valve spring (Curtiss engine).

The maximum safe working load, P, in pounds, on a cylindrical helical spring of outside diameter D in. and with steel of diameter d in. is given by

$$P = \frac{\pi}{8} \cdot \frac{d^3}{D} \cdot S$$

where S is the safe shearing stress of the material in pounds per square inch. The value of S varies from about 80,000 to 150,000 (increasing as the diameter of the wire diminishes) for springs that are used intermittently. For continuous use, as in an airplane engine, about half these values should be employed, or, for the usual sizes of spring steel, about 60,000 lb.

The deflection, f, in inches of one coil of a cylindrical helical spring is given by

$$f = \frac{8nPD^3}{d^4G}$$

where G is the modulus of elasticity in shear and may be taken as 12,000,000 lb. per square inch. The following table gives safe loads, P, and the corresponding deflection, f, of one coil for various springs. It is calculated for S = 60,000 lb. per square inch.

SAFE LOADS AND DEFLECTION OF CYLINDRICAL HELICAL SPRINGS

Wire gage	d in.		Outside diameter of spring, D in.			
			1.75	2.00	2.25	2.50
No. 8	0.162	P	64.	55.5	48.8	43.5
		f	0.23	0.33	0.42	0.57
No. 6	0.192	P	107.	92.5	81.	72.
		f	0.20	0.26	0.34	0.42
No. 5	0.205	P	131.	113.	99.	88.5
		f	0.18	0.25	0.32	0.36
No. 4	0.225	P	175.	150.	132.	118.
		f	0.16	0.22	0.29	0.36
No. 3	0.242	P	225.	195.	170.	152.
		f	0.11	0.16	0.20	0.30

For square steel of side d in. the tabular valves of P should be multiplied by 1.2, and the values of f by 0.59.

The **valve spring retainer** is usually a washer or cupped disc with a downwardly turned flange to center the spring. The retainer is fastened to the valve stem in various ways. It is sometimes held by a nut which screws on to the threaded upper end of the valve stem and is locked by a split pin as in the Bassé-Selve engine (Fig. 129); or it is held by a cotter through the valve stem, locked in position by wire clips as in the Maybach engine (Fig. 123); or the valve stem is turned to a smaller diameter for a short length near the top and held by a conical split collar as in the Liberty engine (Fig. 130), the Benz 300 (Fig. 131) and the Fiat engine (Fig. 134). In those engines in which the cam acts directly on a flat cam follower on top of the valve stem this follower may serve to hold the spring retainer. In the Hispano-Suiza engine (Fig. 132) the upper end of the valve stem is slotted to receive the disc-shaped retainer and is threaded internally. The retainer slips over the valve stem and is provided with a key which fits into the slot and prevents rotation; it has fine notches on its upper surface to mesh with corresponding notches on the lower face of the follower. The follower stem screws into the hollow valve stem and after being screwed into the desired position the retainer is permitted to come into contact

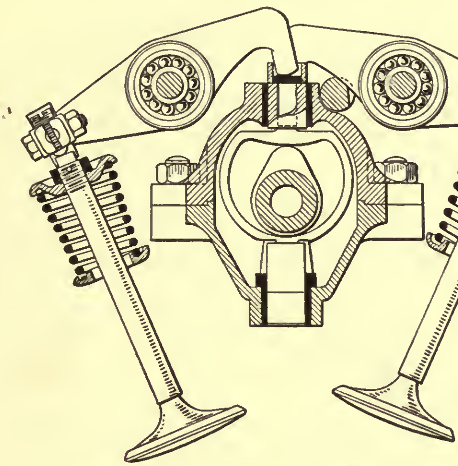


Fig. 129.—Valve gear of Bassé-Selve engine.

with it and locks it in position. A similar arrangement is shown for the Siddeley "Puma" engine (Fig. 133), but in this case a volute steel spring of rectangular section is used.

With dual valves one spring, or a pair of concentric springs, can be made to serve for the two valves as in the Fiat engine (Fig. 134). The springs are in a steel yoke or cage. The inner spring is mounted on a central guide tube while the outer spring is retained by the yoke. The lugs on the two sides of the yoke fit over split locking cones which are held by a spring ring against the grooves turned in the upper ends of the valve stems.

Rocker arms are usually pivoted in plain bearings except in German practice, where ball or roller bearings are commonly used, as in the Bassé-Selve engine (Fig. 129) and the Benz 300 (Fig. 131). In this last engine, which has one intake and two exhaust valves, the rocker for the intake valve is brought obliquely under the double rocker of the exhaust valves and the push rods are thereby kept in line and close to the cylinders without unduly shortening the length of the rocker. Double springs of very low height are required for the intake valve and are sunk in the dished cylinder head.

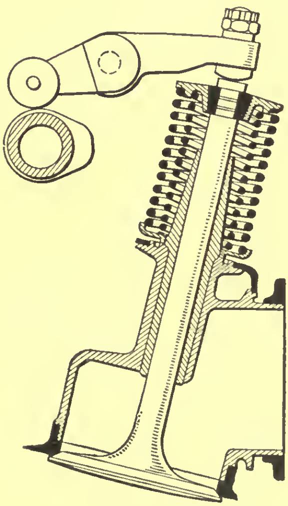


Fig. 130.—Valve of Liberty engine.

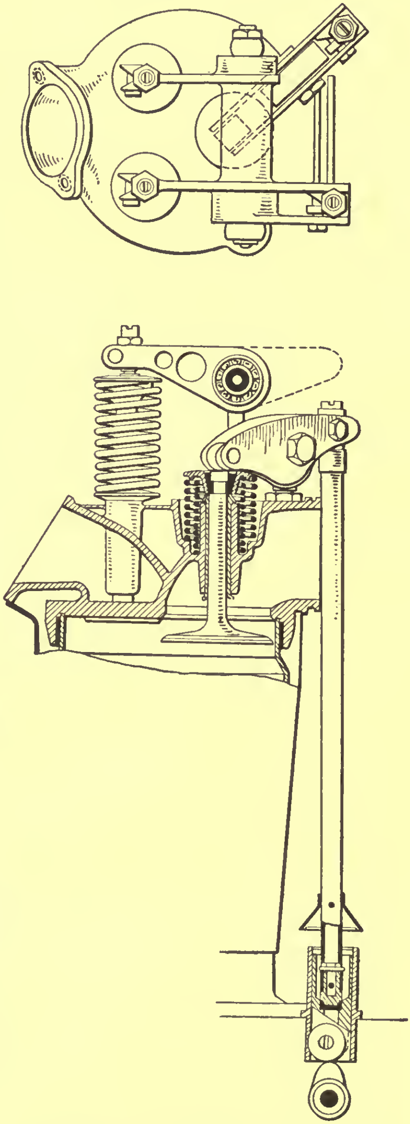


Fig. 131.—Valve gear of Benz 300.

Adjustment has to be provided for the **tappet clearance.** A common method is that shown for the Hall Scott A7A engine

(Fig. 135), in which the hardened steel set screw at the end of the rocker arm is clamped in position by a lock screw at the end of the split rocker arm; in this engine the clearance is 0.02 in. when the engine is cold. In the Liberty engine (Fig. 130) the adjusting

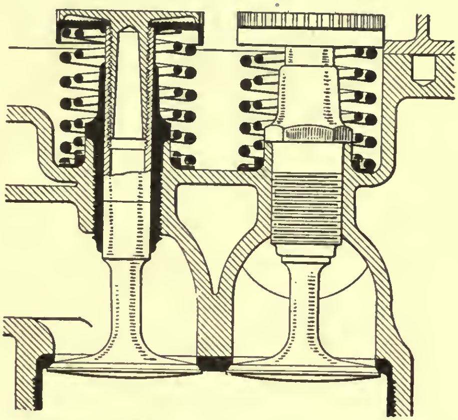


FIG. 132.—Valves of Hispano-Suiza engine.

screw is held in place by a lock nut. The amount of clearance should exceed the expansion of the valve stem and depends mainly on the length of the valve stem. As the inlet valve is colder than the exhaust valve it requires less tappet clearance; in the Liberty engine the inlet clearance is 0.015 in., the exhaust

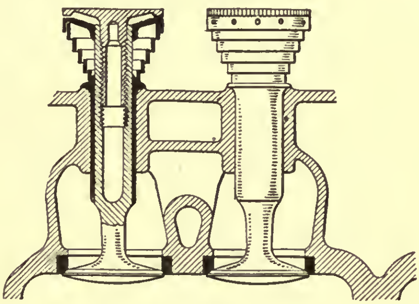


FIG. 133.—Valves of Siddeley "Puma" engine.

0.020 in. Too much clearance is to be avoided, as producing noise and possible breakage of parts.

Valve Timing.—If the inlet and exhaust valve openings could be made unobstructed and large enough, and if the openings and closings were sufficiently rapid, each valve could be timed to open at the beginning of a piston stroke and to close at the end of that stroke. In actual engines it is necessary for the valves to depart from this timing in the interests of high volumetric efficiency and capacity. Especially must the exhaust valve open early and the inlet valve close late. The time of opening of the

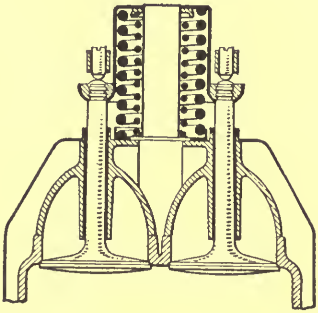


FIG. 134.—Valves of Fiat engine.

inlet valve is related to the time of closing of the exhaust valve: the exhaust valve usually closes completely shortly after the inlet valve starts to lift. The valve does not start to open until the tappet has moved a distance equal to the tappet clearance. The inlet valve opening is usually 10 to 15 deg. past top dead center (T.D.C.) but sometimes occurs before top dead center.

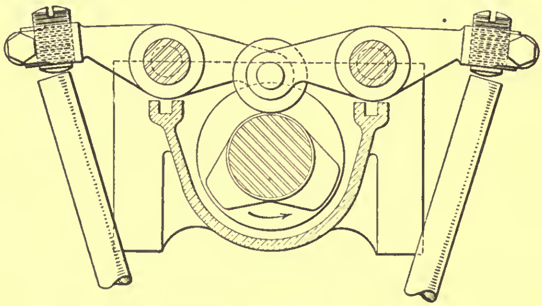


FIG. 135.—Tappet adjustment, Hall-Scott engine.

Inlet valve closure is usually about 40 deg. past bottom dead center (B.D.C.), which corresponds to about 10 per cent of the return stroke of the piston and causes a corresponding decrease in the compression ratio. The exhaust valve opens from 45 to 50 deg. before bottom dead center and closes about 10 deg. past top dead center. The actual timing for any engine should be determined by operation of the engine and observation of the timing giving maximum capacity. The inlet valve should not be opened until a partial vacuum is established in the cylinder; too late a closure will result in reduction of compression below that obtainable with proper timing. Valve timings for various engines are given in the following table:

EXAMPLES OF VALVE TIMING

Engine	Inlet		Exhaust		Duration of opening	
	Opens	Closes	Opens	Closes	Intake	Exhaust
Hall Scott—A7a	15°L	40°L	45°E	10°L	205°	235°
Hall Scott—A5a	15°L	45°L	50°E	10°L	210°	240°
Curtiss—90	12°L	40°L	45°E	TDC	208°	225°
Liberty	10°L	45°L	50°E	10°L	215°	240°
Curtiss—K12	TDC	37°L	47°E	TDC	217°	227°
King-Bugatti	TDC	45°L	47½°E	17½°L	225°	245°
Hispano-Suiza—180	10°L	52½°L	48°E	10°L	222½°	238°
Hispano-Suiza—300	10°E	62°L	62½°E	29½°L	243°	272°
Mercedes—180	TDC	40°L	40°E	10°L	220°	230°
Mercedes—240	2°L	51°L	52°E	16½°L	229°	248½°
Maybach—300	8°E	35°L	33°E	7°L	223°	220°
Benz—200	5°L	45°L	55°E	18°L	230°	253°
Average	10°–15°L	35°–50°L	45°E	10°L	200°–220°	235°

E = early. L = late. TDC = top dead center.

CHAPTER VIII

RADIAL AND ROTARY ENGINES

Radial and rotary engines are characterized by having the cylinders disposed at equal angular intervals around a complete circle. The number of cylinders may range from 3 to 20 or more. It is not possible to arrange more than 10 or 11 cylinders in a circle without increasing the size of the crankcase to dimensions which give an over-all diameter too large for use in an airplane. If more cylinders are desired they have to be arranged in two planes or banks, with an equal number of cylinders in each plane; with air cooling, the cylinders of the rear row are staggered with reference to those of the front row; with water cooling, they may lie exactly behind those of the front row.

Fixed-radial engines have stationary cylinders and a revolving crankshaft; there are usually as many cranks as there are rows of cylinders, although two cranks have been used with a single row of cylinders.

Rotary engines have rotating cylinders and a fixed crankshaft; in this case the propellor hub is attached to the rotating crankcase.

Double-rotary engines have both cylinders and crankshaft rotating but in opposite directions. With this type, two arrangements are possible for utilizing the power developed: (*a*) two propellers may be used mounted on the crankcase and the crankshaft respectively and therefore rotating in opposite directions but with right- and left-hand pitches respectively, so that both give thrust in the same direction; (*b*) the crankcase may be geared to the crankshaft (with reversal of motion) and the power absorbed by a single propeller on the crankshaft or crankcase.

Air Cooling.—The disposition of the cylinders of a radial or rotary engine, in a plane at right angles to the wind and the slip stream, gives these types a unique opportunity for direct air cooling of the cylinders. This is especially the case with rotary engines, which churn through the air as well as meeting the incoming wind.

With cylinders in line, as in multicylinder vertical or Vee engines, air cooling is practicable only with a blower for supplying the cooling air and with a system of ducts for distributing the air to the different cylinders. Renault Frères have built engines of this type up to 12-cylinder Vees but there are considerable difficulties in obtaining high mean effective pressures and low fuel consumption with such cooling as can be obtained in this manner. This type has not met with general favor.

Radial engines are often water-cooled although this practice sacrifices one of the most important potential advantages of the engine. Rotary engines are always air-cooled and could not be readily water-cooled, both as a result of the mechanical difficulties in getting water to and from the rotating cylinders, and also because of the excessive centrifugal stresses which would be set up in the connections between the cylinder and the crankcase as a result of the increased mass of the cylinder with its water and jacket.

Advantages of Radial and Rotary Engines.—The primary advantage of radial and rotary engines over other types is in small weight per horse power. This results from two principal causes.

1. The engine can be air-cooled, and the water pump, water, jackets, water pipes and radiator eliminated. The only important additional weight is that of the cooling fins on the cylinder.

2. The crankcase and crankshaft are much shortened and lightened; the big ends of the connecting rods are also lighter.

The net result of these reductions is a decrease of almost 40 per cent in weight per horse power of the power plant. The Liberty engine with water and radiator weighs about 2.6 lb. per horse power; large air-cooled fixed radial engines have a weight of about 1.6 lb. per horse power.

Other advantages of the radial and rotary engines are short over-all length, which permits better location of gasoline tanks, pilots, etc.; immediate accessibility of the cylinder heads, and easy accessibility of the rest of the engine on removal of the cowling; ease of mounting on and detaching from the fuselage, the attachment being to a vertical plate; lowered air resistance in small sizes which can be accommodated without special enlargement of the fuselage—in larger powers it is necessary to increase the size of the front of the fuselage and thereby increase its resistance until it is likely to be as great as that of a water-cooled engine and its radiator; engine balance as good as in the best arrangements of vertical and Vee engines.

Disadvantages.—The principal disadvantage of the air-cooled radial and rotary engines up to the present time has been a lower m.e.p. and higher fuel consumption than in water-cooled engines. These features are especially true of the rotary engine, which suffers the further disability that its revolutions per minute must be low in order to keep down the air-churning resistance and the centrifugal forces exerted by the cylinders on the crankcase. With improved constructions of cylinders (see p. 202) it is probable that the performance of fixed air-cooled radial engines will not be markedly different from that of water-cooled engines. Other disadvantages are larger oil consumption, which is particularly marked in rotary types but can be kept down in fixed radials; large over-all diameter necessary for large powers, requiring an enlarged fuselage and possibly limiting the power developable per unit; large crankpin pressures in fixed radials; large gyroscopic effect in rotaries, affecting the maneuvering qualities of the plane; large inertia effect in rotaries, retarding the speeding up of the engine on opening the throttle but giving more uniform speed at low speeds or with missing cylinders.

Number of Cylinders and Firing Order.—With a single crank, the number of cylinders of a radial or rotary engine must be odd; the firing order will follow the direction of rotation of the crank in fixed radials and will be in the opposite direction to rotation of the cylinders in rotaries. In both cases it will skip alternate cylinders and will have occurred in all the cylinders in two complete revolutions of the crank or cylinders. If the cylinders are numbered serially the firing order will be 1, 3, 5, 7 . . . 2, 4, 6, 8 . . . .

With a two-throw crank and equal angular spacing of the cylinders the number of cylinders acting on each crank should be odd; the total number of cylinders will be even. (The Smith 10-cylinder engine[1] with one row of cylinders has a two-throw crank so that it is really equivalent to two rows of five cylinders each.) The firing should occur alternately on the two cranks which are at 180 deg. The firing order for regular impulses for a 10-cylinder engine, with serial numbering, will be 1, 8, 5, 2, 9, 6, 3, 10, 7, 4. The 20-cylinder Anzani air-cooled rotary has four rows of five cylinders each. This arrangement keeps the over-all diameter small but impairs the cooling of the two rear rows.

[1] *Jour. S. A. E.*, Jan., 1919.

With a two-throw crank and equal spacing of cylinders in each row but with the cylinders of the second row immediately behind those of the first row[1] the number of cylinders in each row may be odd or even. Firing is alternately from front to back row cylinders except when there is an even number of cylinders in each row. In that case two successive impulses will come on one crank at the end of each revolution but the angle between impulses will be constant.

Rotary Engines.—In a rotary engine (Fig. 136) the cylinders rotate about the crankshaft as a center; the pistons rotate about

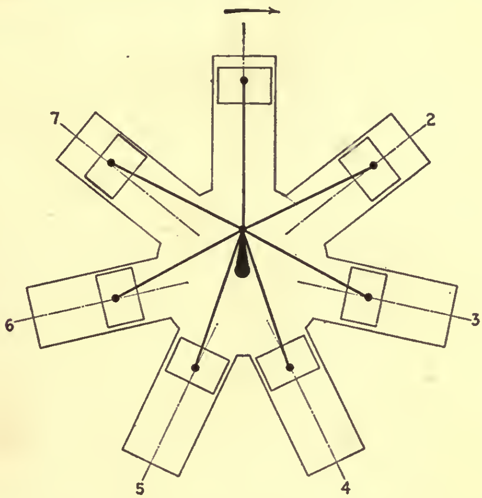


FIG. 136.—Diagram of rotary engine.

the crankpin as a center. The angular velocity of the pistons about the crankshaft as center is constant since it must be the same as that of the cylinders which are rotating at uniform velocity; the angular velocity of the pistons about the crankpin as center would be constant only when the connecting rods were infinitely long or the crank throw infinitesimally small. The result of the rotations about the two centers is to give the piston a reciprocating motion relative to the cylinders as shown in Fig. 136 (an analysis of this motion is given on p. 507). As the cylinders rotate the pistons will assume the positions shown relative to the cylinders. If the crank is fixed with its throw vertically upward the piston will always be at the in dead center

[1] See 20-cylinder water-cooled Anzani, *Aviation*, Feb. 15, 1920.

when the cylinder reaches the position 1; it will be at the out dead center for the position vertically below 1. One revolution of the cylinders completes one in and out stroke of each piston.

Certain special problems arise in the construction of the rotary engine. One of the most important of these is the accommodation of seven or nine connecting rods on one crankpin; this problem occurs also with fixed-radial engines. Another is the attachment of various members to the rotating cylinders. The exhaust manifold is always eliminated and the exhaust gases discharge directly past the valve to the air. The carburetor (or equivalent device) must be stationary and discharges into the crankcase; separate induction pipes to the individual cylinders may or may not be provided. There is no possibility of continuous circulation of lubricating oil; the oil is used up as it is fed. The cylinders must be machined all over to exact dimensions to avoid unbalanced centrifugal forces.

Gnome Engine.—A longitudinal section of the 100-h.p. Gnome Monosoupape (single-valve) engine is shown in Fig. 137. The engine has nine cylinders (each machined out of a 6-in. solid nickel-steel bar) arranged at equal angular distances around the crankcase. The bore is 110 mm., the stroke 150 mm., clearance volume 365 cu. cm., normal speed 1,200 r.p.m., weight 260 lb.

The cycle of operations is as follows: starting with any cylinder on top center and the exhaust valve open, the cylinder draws in air through the exhaust valve until its closure at 45 deg. before the bottom center; that is, air is drawn in for 135 deg. rotation of the crank. During the next 25 deg. a vacuum is created in the cylinder until, at 20 deg. from the bottom center, the admission ports are uncovered and a rich mixture enters from the crankcase, mixing with the air already in the cylinder and forming an explosive charge. The ports are again covered at 20 deg. past bottom center and, as the cylinder rotates to its top center, compression occurs. Ignition takes place at 20 deg. before top center and the cylinder rotates on its power stroke until it is 90 deg. past top center, when the exhaust valve opens and remains open for the following 405 deg. rotation. It will be seen that the exploded charge is expanded for only half the stroke and consequently there is no possibility of high capacity or efficiency with this engine. One important reason for the early exhaust of the exploded mixture is to prevent overheating of the cylinders. It is found that a late exhaust will cause overheating.

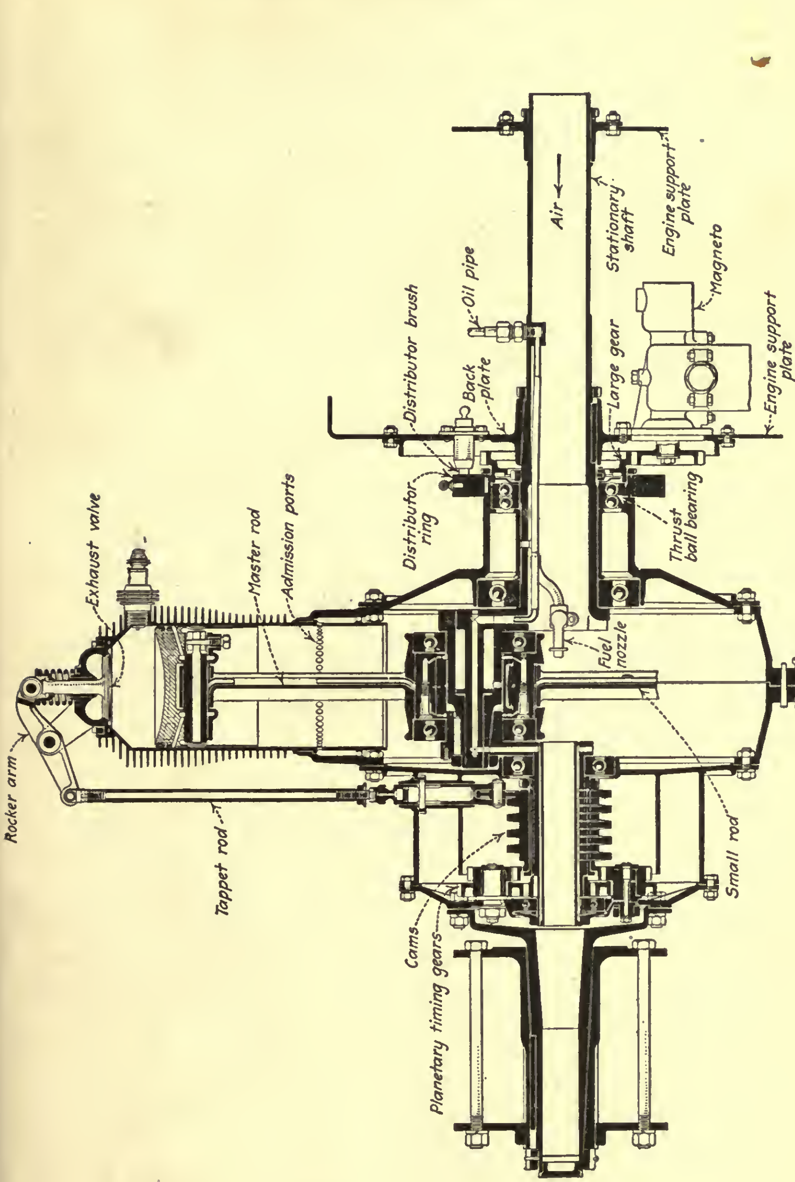


Fig. 137.—Longitudinal section of Gnome 100.

The mixture in the crankcase is at all times too rich to be explosive. Air is drawn in through the open rear end of the hollow stationary crankshaft. The fuel is supplied under air pressure and sprays from the fuel nozzle into the crankcase, where it is churned up with the air. There is no throttle valve, but the fuel supply can be controlled by a regulating valve to adjust the mixture strength. The power output of the engine can be varied through a small range only, by the use of the regulating valve or switching the ignition on and off. Variation of the power by cutting out the ignition on one or more cylinders gives unequal impulses, and causes fouling of the cylinders which are not firing.

The *magneto* is mounted on the face of the back plate remote from the engine. It is driven through a spur gear which meshes with the large gear keyed to the thrust box casing. The gear ratio is 4 :9, that is, the magneto armature makes nine revolutions to four of the engine, and as the magneto gives two sparks per revolution there will be nine sparks in two revolutions of the engine. The current from the magneto goes to the distributor brush, which makes contact in turn with the nine metal segments of the distributor ring. The distributor ring revolves with the engine and consequently 18 contacts are made in two revolutions of the engine, but no spark passes on the exhaust stroke as none is generated at the magneto.

The *air pump* (for the fuel pressure) and *oil pump* are mounted on the back plate and driven from the same large gear as the magneto. The oil pump delivers oil into two pipes of equal size. Of the oil going into one pipe about one-third flows through a branch into the thrust box, oiling the thrust ball race and main engine ball race. The surplus oil overflows into the crankcase through holes drilled for this purpose, and passes on to the cylinder walls through the ports in the base of the cylinder. The main supply of oil passes up the big crank web through a hollow plug in the center of the hollow crankpin, down the short end crank web into the hollow short end of the crankshaft, whence it is conveyed by a series of holes to the cam pack. The oil then passes through grooves between the cams and is thrown centrifugally over the interior of the cam box, lubricating the cams, cam rollers, tappets, planet gear wheels, and the cam box and nose-piece ball races. The oil then overflows back into the crankcase and passes on to the cylinder walls as in the case of the overflow from the thrust box. Some of the oil also passes along the hollow tappet rods to the rocker arm pins.

The oil from the other pipe flows up inside the long end crank web into an annular space around the brass plug in the long end crankpin and out of holes in its balls to two grooves or channels cut in the ends of the bore of the master connecting rod big end. Holes are drilled from these grooves to each wristpin and the wristpins are drilled to correspond with these holes, so that the oil may pass through to lubricate the wristpin bushings. From these the oil passes into steel tubes (which are fixed to the connecting rods) and along the tubes, oiling the gudgeon or wristpins and bushes. In later type engines the steel tubes are dispensed with, and the oil passes along the face of the connecting rods to the gudgeon pins and bushes. The overflow from the gudgeon pins passes through holes in the side of the pistons, and lubricates the rings and the cylinder walls. The surplus oil is blown out through the exhaust valve and lubricates the exhaust valve-guide and stem.

The *crankcase* is made of two steel stampings bolted together by steel bolts and centered by dowel pins. The nine cylinders are each gripped tightly by the two parts of the crankcase and prevented from turning by a small key. The crankcase is not directly supported on the crankshaft, but carries on its faces plates or covers, known respectively as the cam box and the thrust box. The thrust box contains the main ball race and a self-aligning double-thrust race. The cam box contains the planet gears and the cam pack which actuates the exhaust valves, and one radial ball race. The nose piece which carries the propeller is mounted on the cam box.

The *pistons* are of cast iron with concave heads. A portion of the trailing edge is cut away to allow the piston in the adjoining cylinder to clear. Each piston is fitted with an obturator ring about 0.6 mm. thick in a groove around its top. This obturator ring is of cupped form and is pressed out against the cylinder wall by the gas pressure, thus preventing leakage past the piston. A packing ring is fitted behind the obturator ring and in the same groove. A wipe ring which is made of cast iron is also fitted in a groove situated just below the obturator ring. The piston is fastened to its connecting rod by means of a hollow steel gudgeon pin fixed in lugs on the underside of the piston head by means of a tapered set screw.

Connecting Rods.—The connecting rod assembly consists of a master connecting rod, to which eight auxiliary connecting rods are attached by means of wristpins. All the rods are of H-section and the auxiliary rods are bushed at both ends with phosphor-bronze bushes. The master connecting rod big end runs on two ball bearings; its small end is bushed with phosphor bronze.

The single *valve* in the cylinder head performs the following functions: (*a*) It acts as an exhaust valve; while so doing its temperature is raised; (*b*) it admits to the cylinder a quantity of air sufficient for combustion of the charge entering later through the ports at the base of the cylinder. During this portion of the cycle it is cooled.

The valve is 60 mm. diameter and has a lift of 10.5 mm. It is mounted in a steel cage which also carries the rocker arm fulcrum pin, and is mechanically operated by means of a hollow steel tappet rod and steel rocker arm. The valve stem slides in a cast-iron bush at the center of the cage which is held in position by means of a locking ring screwed into the cylinder head. The valve is made heavier than is necessary for mechanical strength and is of such weight as to balance the centrifugal action of the tappet rod which would otherwise tend to keep the valve open. The valve spring is spiral and encircles the valve stem, taking its bearing against the valve cage and a detachable collar on the valve stem. The valves are operated by the *cam pack*, which consists of nine cams keyed to a bronze-bushed sleeve rotating on the small end of the crankshaft. The cams operate the tappet rods which work the overhead rocker arms. Each tappet rod is formed of a tappet and a rod jointed together. The tappet works in a guide in the cam box, and at its inner end is a roller which bears against the cam. The tappet rod extends from the joint to the rocker arm of the exhaust valve, and is adjustable. The cam pack is driven at half the engine speed by planet gears, which are fitted on the inner face of the cover of the cam box. The engine is running at twice the speed of the cam pack, so that the rollers at the bases of the tappet rods are overtaking the cam pack. The clearance between the rocker arm and the bottom of the slot in the valve stem, when the tappet roller is at the bottom of the cam, should be 0.5 mm. when the engine is cold. In later type engines the rocker arm engages the valve stem by means of a roller which bears against the end of the stem.

Le Rhone.—The nine-cylinder 110-h.p. Le Rhone engine is shown in Figs. 138 and 139. The bore is 112 mm., stroke 170 mm., the normal speed is 1,200 r.p.m., weight 323 lb. The 80-h.p. nine-cylinder engine is of 105 mm. bore, 140 mm. stroke. The cylinder has a cast-iron liner. This engine differs from the Gnome engine in several important respects. It has an inlet valve as well as an exhaust valve and consequently has to be

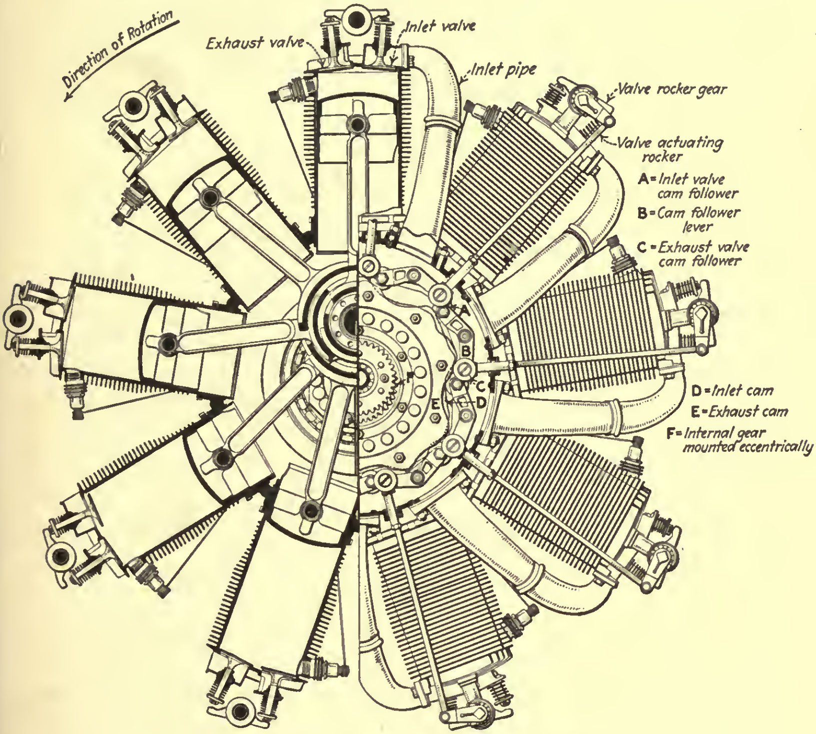


Fig. 138.—Transverse section of Le Rhone 110.

provided with inlet pipes from the crankcase to each inlet valve cage. The valve timing is as follows: the exhaust valve closes 5 deg. after top center on the suction stroke; the inlet valve opens 13 deg. later or 18 deg. after top center; the inlet valve closes 35 deg. after bottom center and compression goes on till 26 deg. before top center, when ignition occurs. The expansion occupies 125 deg. of the power stroke when the exhaust valve opens at 55 deg. before bottom center and remains open for 140 deg.

This timing differs notably from that of the Gnome engine and gives much more complete expansion of the gases.

A rudimentary carburetor (see p. 288) is located at the rear end of the hollow crankshaft, admitting an explosive mixture to the crankcase. Control of power output is through the throttle valve. Ignition is by a magneto and distributor similar in location and general construction to those of the Gnome

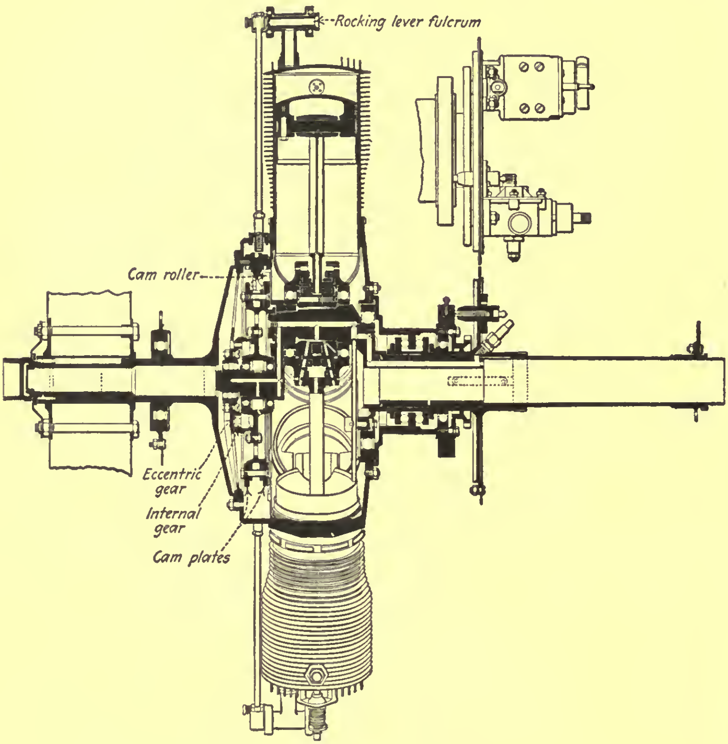


FIG. 139.—Longitudinal section of Le Rhone 110.

engine. The pistons are convex and of semi-steel. The connecting rods are of the slipper type (see p. 204).

The two valves on each cylinder are actuated by the motion of a rocking lever which is fulcrumed at its middle in ball bearings. This lever is operated by a valve-actuating rod which receives its motion from the trailing end of a cam-follower lever. The cam-follower lever is fulcrumed at its middle and carries an inlet valve cam follower at the forward end and an exhaust valve cam follower at the trailing end. The two cam-followers are in different planes and are actuated by the inlet and exhaust cams respectively. These cams are lobed plates mounted on a spider running in ball bearings on a shaft eccentric to the crankshaft. The spider carries an internal gear into which meshes an external gear mounted in ball bearings on the crankshaft and rotating with the crankcase. These arrangements are shown in the right half of Fig. 138 and also in Fig. 139. The external gear has 45 teeth; the internal gear 50 teeth; consequently one complete revolution of the engine will produce nine-tenths of a revolution of the cams. The engine is overrunning the cams and would

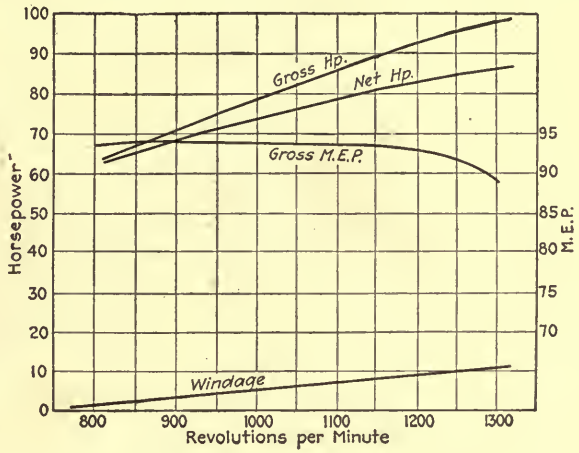


FIG. 140.—Performance curves of Le Rhone 80.

overrun them one complete revolution in 10 revolutions of the engine. Each cam has five lobes. Each inlet and exhaust valve should be opened once only in two revolutions of the engine, and this will be accomplished when the engine overruns the cam one-fifth of a revolution. As the engine overruns the cams one-tenth of a revolution each engine revolution it is evident that two revolutions of the engine are required to complete the opening and closing of all the valves.

The action of centrifugal force on the valve-actuating rod causes it to press continuously against the valve rocker lever. At low speeds of revolution this force may not be sufficient to open the exhaust valve at the desired time and consequently an exhaust cam is necessary to push the valve rod out. At high speeds the operation of both valves can be taken care of by the

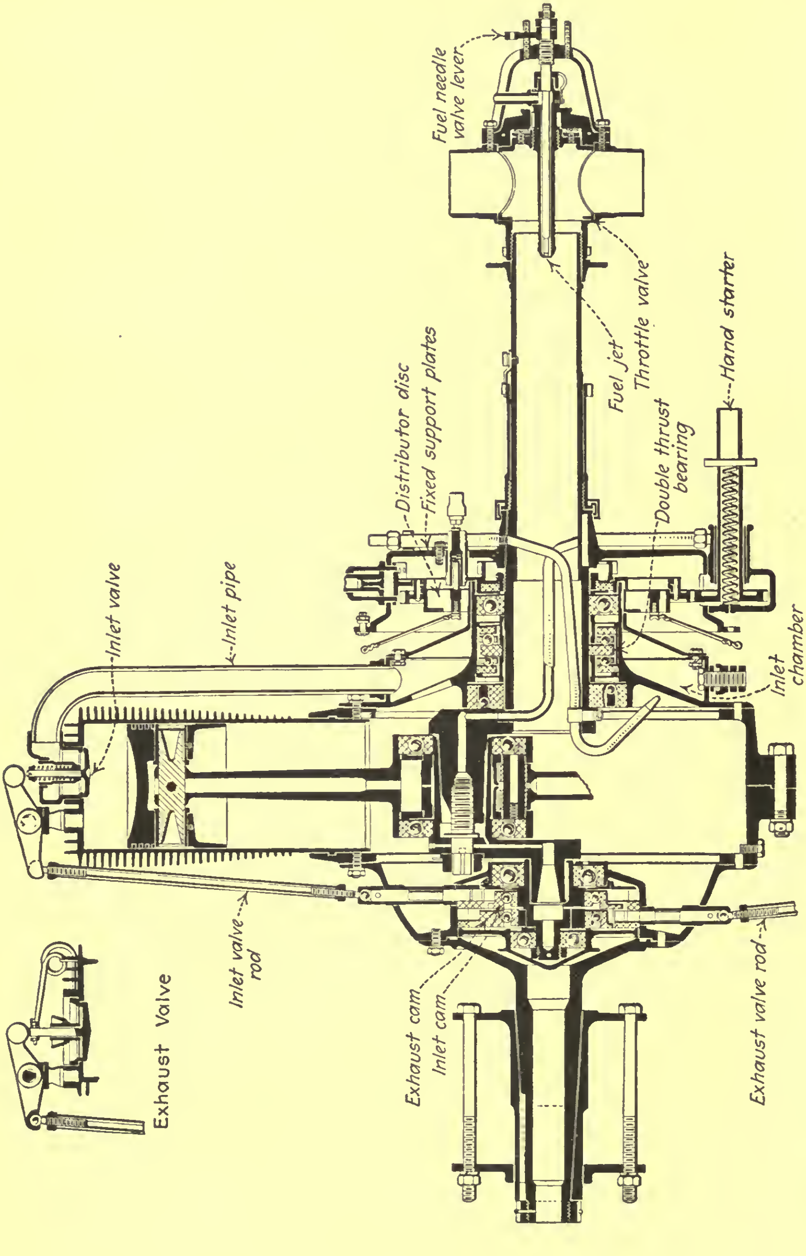


Fig. 141.—Longitudinal section of Clerget 130.

inlet cam if it is properly shaped for that purpose. The valves are brought back to their seats by spiral springs at low speeds; at high speeds centrifugal force closes the valves.

Performance curves for the 80-h.p. Le Rhone are given in Fig. 140.

Clerget.—The Clerget rotary engines are built with 7, 9 or 11 cylinders. The 130-h.p., nine-cylinder engine shown in longitudinal section in Fig. 141 is 120 mm. bore, 160 mm. stroke, makes

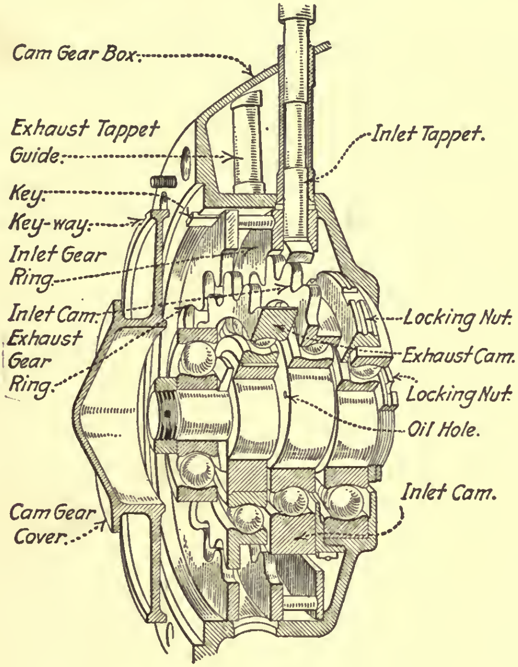


FIG. 142.—Cam gears of B.R. 2.

1,250 r.p.m., weighs 381 lb., develops 135 h.p. and has a compression ratio of 4. Its points of difference from the previous engines include the use of an aluminum piston, tubular connecting rods, inlet and exhaust valves operated by means of separate cams, tappets and rocker arms, and a double-thrust ball race which is a pure thrust bearing and distinct from the combined thrust and radial bearings of the other engines.

The inlet and exhaust cam plates are driven at nine-eighths the engine speed by separate internally-toothed gears mounted inside and keyed to the cam-gear case. These mesh with external gears mounted eccentrically on the crankshaft; the cams are attached to these external gears. This arrangement is the reverse of that used on the Le Rhone engine. The cam plates overtake the engine once in eight revolutions. Each cam plate has four lobes so that in eight revolutions each tappet will be lifted four times, or once in two revolutions. A sectioned perspective view of the similar cam-gear box of the B.R.2 rotary engine is shown in Fig. 142. The four cams on each gear are simply rearward extensions of every fourth tooth.

The valve timing differs in some respects from that of the Gnome and Le Rhone. At top center of the suction stroke

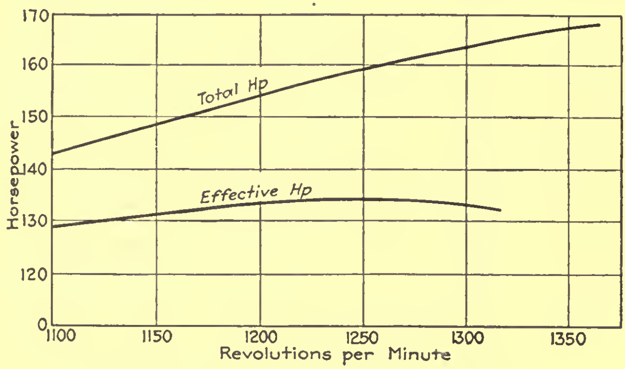


FIG. 143.—Performance curves of Clerget 130.

both exhaust and inlet valves are open. The inlet opens 5 deg. before top center; the exhaust closes 5 deg. past top center. The inlet remains open till 58 deg. past bottom center (or a total of 153 deg.) and compression begins. Ignition is at 25 deg. before top center and exhaust begins 68 deg. before bottom center.

The carburetor is located at the rear end of the hollow crankshaft. Fuel is injected under air pressure through a jet which is controlled by a needle valve. The air supply is controlled by a cylindrical throttle valve. Equal movement of both throttle lever and needle valve lever controls air supply only. Operation of the throttle lever alone controls both air and fuel. The charge entering the crank passes to the annular inlet chamber at the rear of the crankcase and then by the separate inlet pipes to the cylinders. The connecting rod assembly is similar to that of the Gnome engine (see p. 203).

The performance curve for this engine (Fig. 143) is typical of rotary engines. The effective horse power goes through a maximum at 1,250 r.p.m. but is very flat for a considerable range of speed. The rapidly increasing difference between the effective horse power and the indicated horse power is due to the rapid increase in the air-churning resistance.

B.R.2 Engine.—The British B.R.2 engine is one of the largest air-cooled rotary engines. In general construction it is similar to the Clerget. The cylinders are of aluminum with steel liners and steel head. The cylinder diameter is 140 mm., stroke 180 mm., compression ratio 5.01, brake horse power 230 at 1,300 r.p.m., weight dry 498 lb., weight per brake horse power dry

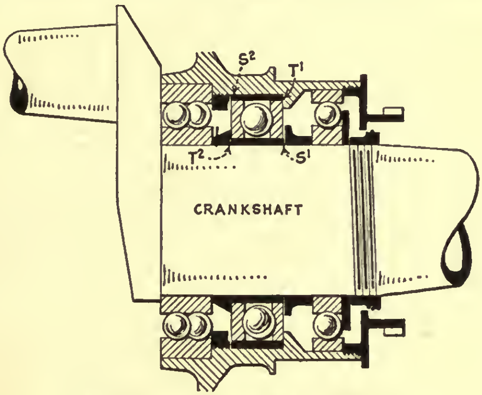


Fig. 144.—Thrust box of B.R.2.

2.16 lb. The cam gear for this engine is shown in Fig. 142 and follows exactly the same principle as the Clerget cam gear. The thrust box contains two ball bearings and a thrust bearing which differs from the Clerget in having one row of balls only. This single-thrust bearing is adapted both for pusher and tractor use as indicated in Fig. 144; a very small clearance is left for the travel of the crankcase along the crankshaft when changing from pusher to tractor.

Double Rotary.—The double-rotary engine has cylinders revolving in one direction while the crankshaft revolves in the other direction. The effective speed is the sum of the two speeds so that the power of an engine in which both cylinders and crankshaft revolve at 900 r.p.m. is the same as that of a radial or rotary engine of the same dimensions operating at 1,800 r.p.m. Such a speed is permissible in radial engines but would give excessive air-churning resistance in a rotary. There is no reason why even higher effective speeds—up to 2,400 r.p.m.—may not be practicable with this type, if the volumetric efficiency of the engine can be maintained and if the cylinders can be kept cool enough. In any case this arrangement leads to a combination of high engine speed and low propeller speed with consequent high propeller efficiency. It has important advantages over all other types in (*a*) the possibility of the elimination of unbalanced

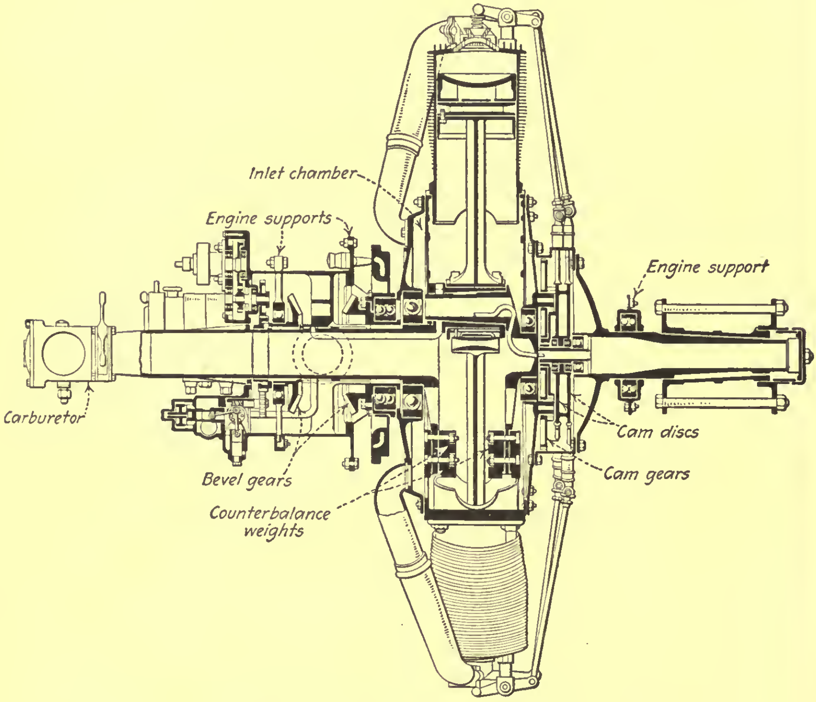


FIG. 145.—Longitudinal section of Siemens-Halske double rotary.

gyroscopic effects, which is an advantage for maneuvering, and (*b*) the elimination of the unbalanced turning moment exerted by the engine on the plane. This unbalanced turning moment is a constant, though small, power drag on the plane and its elimination is a distinct advantage

The only engine of this type which has been in production is the Siemens-Halske 11-cylinder engine (Fig. 145), which was brought out in 1918 and develops 200 h.p. at 900 r.p.m. of both cylinders and crankshaft, or a virtual speed of 1,800 r.p.m.

The single propeller is mounted on a nose attached to the revolving crankcase; the torque of the crankshaft is transmitted to the crankcase by securing a bevel wheel to the crankshaft and a similar gear, facing it, to the crankcase and mounting an intermediate pinion between the two on a stud which is fastened to the stationary cylindrical housing at the rear of the engine.

The carburetor is mounted on a stationary hollow extension of the crankshaft. The combustible charge is drawn in through the hollowcrank shaft to the crankcase and goes from the annular inlet chamber at the rear of the crankcase to the individual inlet pipes. The inlet and exhaust valves are operated through two cam plates which are loose on the crankshaft and are rotated through double reduction gears from an internal gear attached to the crankcasing. The engine is supported by steel rods both before and behind the cylinders.

The weight of the engine complete is 427 lb., which at 240 maximum h.p. gives a weight of 1.78 lb. per horse power. The fuel economy is as good as with stationary engines and is much better than with other rotaries.

Other designs of double rotaries with two propellers (right- and left-hand respectively) forward of the cylinders, attached to the crankshaft and crankcase respectively, have not passed the experimental stage. The efficiency of a pair of propellers close together but operating in opposite directions has been found to be but little inferior to that of a single propeller. There is consequently the possibility of the development of a satisfactory double-rotary engine on the lines indicated.

Radial Engines.—In a fixed radial engine the cylinders are stationary and the crankshaft revolves. Three to eleven cylinders can be accommodated in a single row or bank, but two rows with a two-throw crank must be adopted if a larger number of cylinders is desired or if it is necessary to cut down the over-all diameter. The two-throw crank eliminates the need for counterbalance weights but increases the length of the engine and introduces difficulties in the air cooling of the rear row cylinders.

Radial engines offer certain special construction problems. The most important are the balancing of the masses at the crankpin and the avoidance of excessive pressures on the crankpin. It is possible to operate radial engines at speeds as high as those used in vertical and Vee engines, but special care must be taken to prevent the overheating of air-cooled cylinders and the over-

13

loading of the crankpin. There is no fundamental reason why the mean effective pressure and economy of radial engines should not be as good as those of any other type.

A B C.—The development of *air-cooled* fixed radial engines has been carried on in England more than elsewhere. The

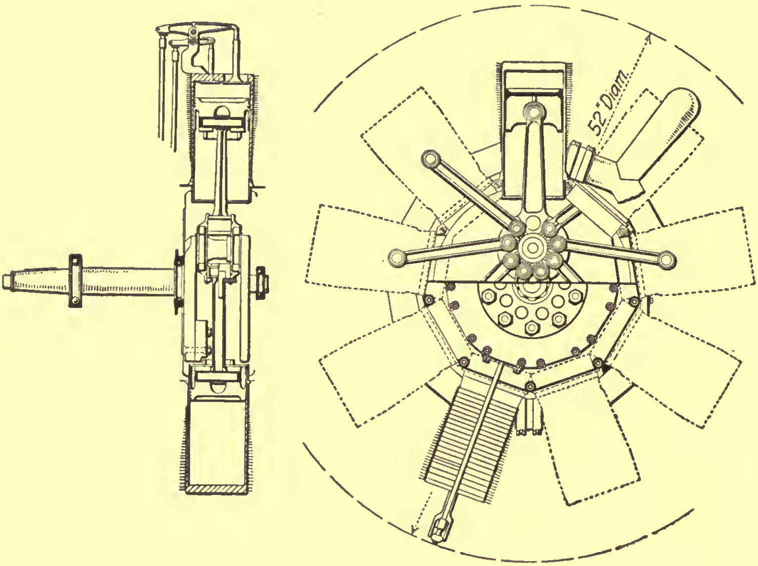


FIG. 146.—Sectional outlines of A B C "Dragonfly." Radial engine.

A B C engines, built by the Walton Motors Co., have the following general characteristics:

Type name	Gnat	Wasp	Dragonfly
Number of cylinders (copper-coated steel fins)	2	7	9
Bore, inches	4.75	4.75	5.5
Stroke, inches	5.5	6.25	6.5
Normal brake horse power	45	200	340
Revolutions per minute	1,800	1,800	1,650
Oil consumption, pints per hour	1.7	4	7
Gasoline per brake horse power hour, pints	0.56	0.56	0.56
Weight of engine, dry, pounds	115	320	600
Weight per brake horse power, pounds	2.3	1.6	1.765
Over-all diameter, inches	....	42.7	50.5

The Wasp and Dragonfly engines have each two exhaust valves and one inlet valve per cylinder. Their engines use the master-rod connecting-rod assembly with roller bearings (see p. 207) and have counterbalance weights. Sectional views of the Dragonfly engine are shown in Fig. 146.

Cosmos.—The Cosmos Engineering Co. has fixed radial engines with the following characteristics:

Type name	Lucifer	Jupiter, direct drive	Jupiter, geared	Mercury	Hercules, geared
Number of cylinders	3	9	9	14	18
Number of rows	1	1	1	2	2
Bore, inches	5.75	5.75	5.75	4.375	6.25
Stroke, inches	6.25	7.5	7.5	$5^{13}/_{16}$	7.5
Normal brake horse power	100	400	450	315	1,000
Brake mean effective pressure, pounds per square inch	....	113	113		
Revolutions per minute	1,600	1,650	1,850	1,800	1,750
Propeller speed, revolutins per minute	....	....	1,200		1,150
Weight of engine, dry, pounds	220	636	757	587	1,400
Weight per brake horse power, pounds	2.2	1.59	....	1.863	1.4
Weight per brake horse power at maximum power, pounds	....	1.413			
Over-all diameter, inch	....	52.5	52.5	41.625	

The Jupiter engine has two exhaust and two inlet valves; the Mercury engine has two exhaust and one inlet valve.

Performance curves for the Jupiter engine are shown in Fig. 147. It will be seen that the brake mean effective pressure reaches a maximum of 117 lb. per square inch at 1,700 r.p.m.

A special feature of the Jupiter engine is the method of conveying the explosive charge to the cylinders. There are three independent carburetors at the rear of the engine discharging into the cover of the annular inlet chamber which forms the rear of the crankcase. This chamber (Fig. 148) contains an aluminum spiral casting which fits closely into the chamber. The

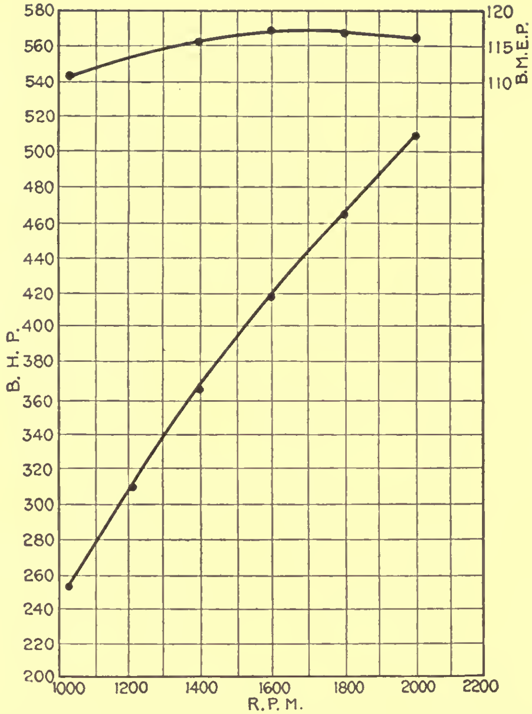


FIG. 147.—Performance curves of Cosmos "Jupiter" radial engine.

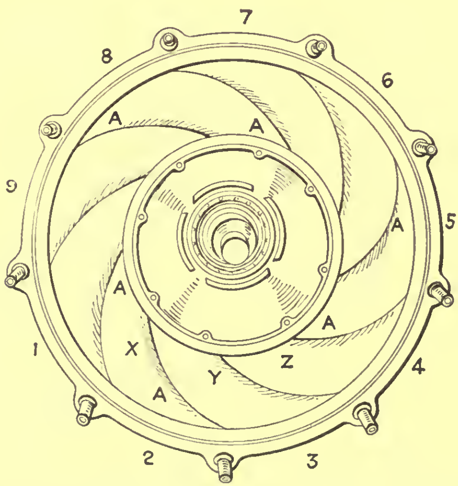


FIG. 148.—Induction chamber of Cosmos "Jupiter."

casting constitutes a three-part spiral. The carburetors discharge into the spaces marked X, Y and Z respectively. The

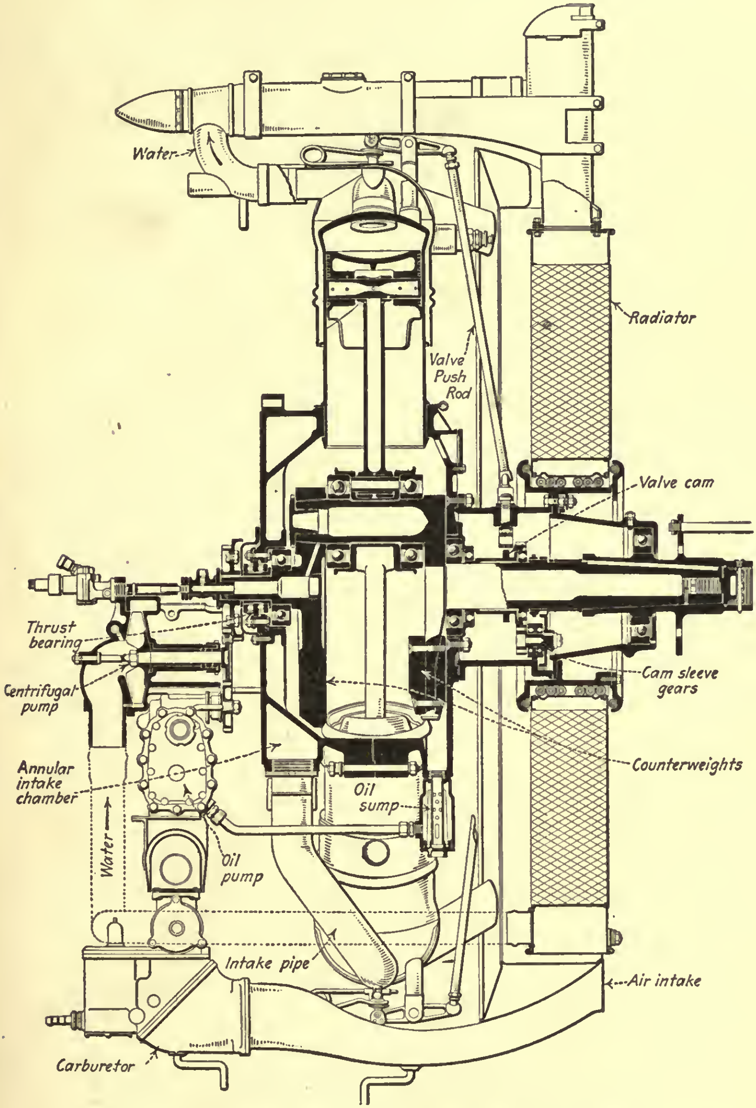


FIG. 149.—Longitudinal section of Salmson radial engine.

space X is part of the spiral marked AAA, so that the mixture drawn into X will flow along the spiral groove AAA. This groove is opposite the inlet pipes for cylinders 2, 8 and 5; similarly the middle carburetor will supply cylinders 3, 9 and 6. This arrangement gives the mixture a clean sweep from the carburetor to the cylinder and isolates the cylinders in three groups so that should one carburetor fail to act properly there would still be six cylinders in normal action.

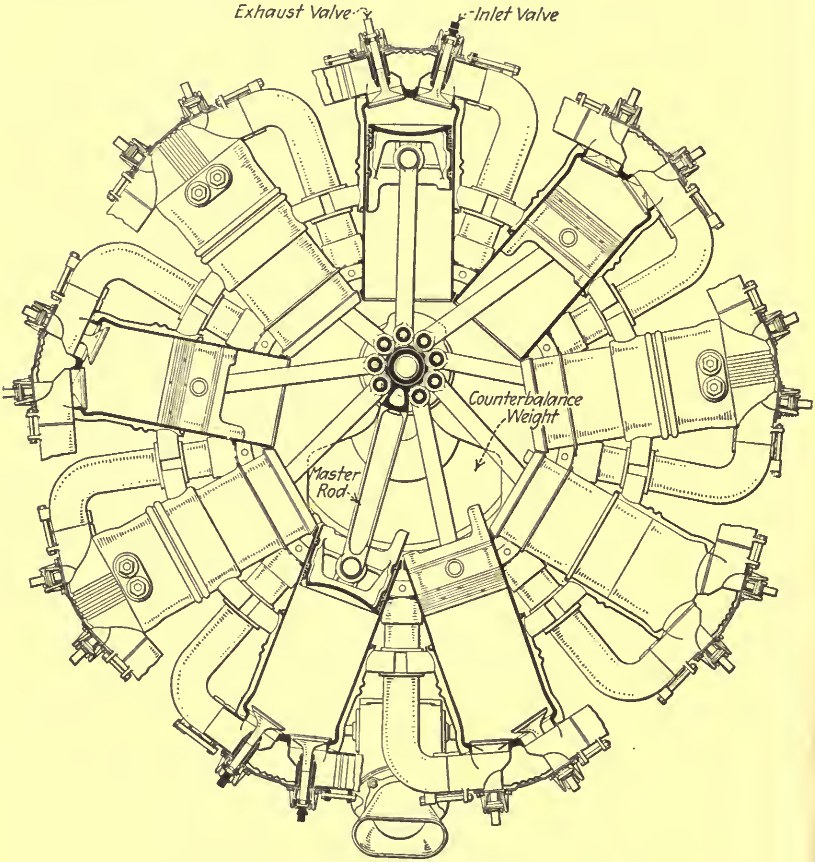


Fig. 150.—Transverse view of Salmson radial engine.

Salmson.—The Salmson (Canton-Unné) engine is a good example of the *water-cooled fixed-radial* engine. Figure 149 shows a longitudinal section of a nine-cylinder engine; Fig. 150 is a transverse view of the same engine. The general dimensions of the engine are: bore, 125 mm.; stroke, 170 mm.; ratio of compression, 5.3; weight of engine without water or radiator, 474 lb.; weight of water in jackets 20 lb.; power at 1,500 r.p.m., 250 h.p.; weight dry per horse power, 1.89 lb.; gasoline consumption per horse-power hour, 0.507 lb.; oil consumption per horse-power hour, 0.077 lb. The variation of brake horse power with engine speed is shown in Fig. 151.

The cylinders are steel forgings 3 mm. thick; the jackets are of sheet steel welded to the cylinders. The inlet and exhaust valves are symmetrically located and are both 62.5 mm. diameter; they are held to their seats by rat-trap springs. The connecting-rod assembly is of the master-rod type (see p. 203) with ball bearings on the crankpin. The crankshaft is of the

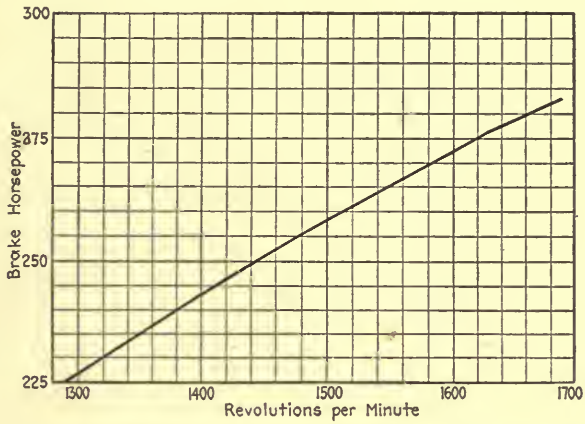


FIG. 151.—Performance curve of Salmson radial engine.

built-up type with counterweights. The valves are operated through push rods and rocker arms from a cam sleeve which is revolved on the crankshaft at one-fourth the engine speed by means of an epicyclic gear set. There are three pairs of cams on the cam sleeve, each pair at opposite diameters in its own plane. In each of the three planes are the cam followers of both valves for three cylinders. All the valves will be opened twice in one revolution of the cam sleeve. Consequently in two revolutions of the engine, or one-half revolution of the cam sleeve, each of the valves will have been operated once.

The inlet valve opens at top center and closes 55 deg. after the bottom center, or at about 16 per cent of the return stroke. Ignition occurs about 30 deg. before top center. The exhaust opens 65 deg. before bottom center and closes at top center.

The water circulation is shown in Fig. 152. A centrifugal pump taking water from the bottom of the radiator discharges it through two pipes into the heads of the two lowest cylinders. The top and bottom of each jacket is connected by pipes to the tops and bottoms respectively of the adjacent cylinders. The water is finally delivered from the top of the highest cylinder to the radiator.

The carburetor (Zenith) discharges through long vertical pipes into the annular inlet chamber at the rear of the crankcase and thence through separate inlet pipes to the individual cylinders.

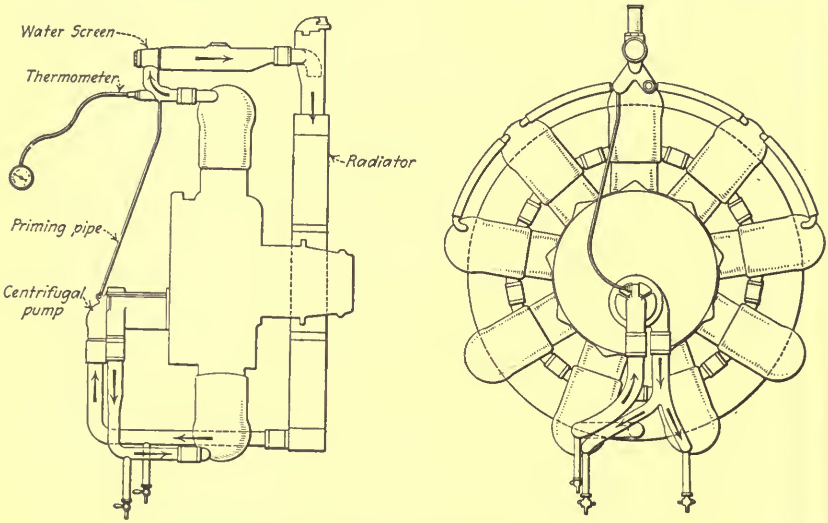


FIG. 152.—Water circulation in Salmson radial engine.

The exhaust passes from each cylinder into a sheet metal exhaust duct which encircles the engine, discharges at the sides of the fuselage, and is stream-lined to serve as a cowling for the engine.

Details of Radial and Rotary Engines.—Air-cooled cylinders are either made from solid steel, as in the Gnome, Le Rhone and Clerget rotary engines, or they are composite with steel barrel or liner and aluminum alloy head. All-aluminum cylinders have been tried with fair success but there is doubt of their durability; they are no lighter than the other types and their considerable longitudinal expansion increases tappet clearances and alters valve timing to a greater extent than with other constructions.

The satisfactory operation of an air-cooled cylinder depends on keeping down its temperature. When overhead valves are used, this temperature is highest in the middle of the head. With open exhaust as in rotary engines and in some radial engines there is not much difficulty in arranging for adequate cooling of all parts of the cylinder.

With overhead valves it is essential to make the cylinder head of the best available conductor (see p. 346), which in practice turns out to be an aluminum-copper alloy. The valve seats and the working surface of the cylinder barrel must be of some harder material. When an aluminum head is used the valve seats should consist of rings of steel or bronze, cast or expanded into position. Bronze seats, in consequence of their high coefficient of expansion, are less likely to come loose than steel seats.

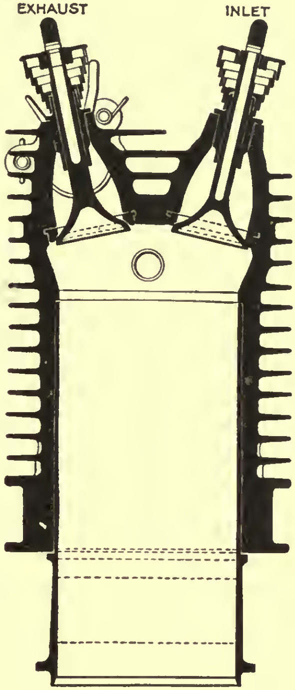


Fig. 153.—Aluminum air-cooled cylinder with steel liner.

One type of construction is shown in Fig. 153. An aluminum casting forms the head and surrounds the greater part of the steel liner, which is shrunk into the casting at about 300°C. Cylinders of this type have given excellent results, but the difference between the coefficients of expansion of the steel and aluminum tends to cause separation of the liner and casing at working temperatures and a film of oil may work in between them. With cylinders below 4 in. in diameter there is little trouble. A shrinkage allowance of about 1 in 600 should be made. The expansion trouble can be overcome by the use of bronze liners if a bronze sufficiently hard to resist wear is developed. The holding-down bolts go through lugs in the aluminum casting.

Screwed-in liners have not given good results owing to the impossibility of maintaining adequate contact between liner and casing. If the contact is good when cold, the difference of expansion when hot causes contact at points only.

The best method of composite construction is one with an aluminum cylinder head into which is cast or screwed a steel barrel with its own cooling fins (Fig. 154). This construction is mechanically sound and has been used successfully with cast-in barrels for sizes up to 6 in. in diameter and with screwed-in barrels up to 5½ in. in diameter. The length of the screwed portion should be about one-fourth of the cylinder diameter. The holding-down bolts grip a ring integral with the steel barrel and thereby avoid the breakages of holding-down lugs which have been rather frequent with the construction of Fig. 153. In another type of construction the barrel and head are formed of steel in one piece and an aluminim cap embodying the inlet and exhaust ports is bolted to the cylinder head.

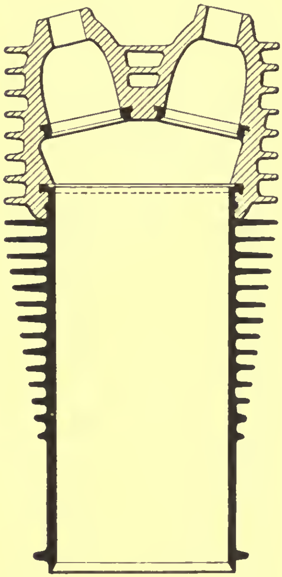


Fig. 154.—Steel air-cooled cylinder with aluminum head.

Tests of all-steel cylinders such as are used in Le Rhone and Clerget engines, with cylinder diameters ranging from 4 to 6 in., show that the all-steel cylinder gives very appreciably higher fuel consumption and lower brake mean effective power than does the aluminum-headed cylinder.[1] A 5½ by 6½-in. steel cylinder with one aluminum inlet and two cast-iron exhaust ports bolted to it was changed (1) by having an aluminum cap bolted to its head and (2) by having the original head cut off and an aluminum head cast on to the same barrel. Tests showed that under maximum load conditions at 1,450 r.p.m. and in a wind of 82 miles per hour the aluminum headed cylinder gave 15 per cent more power than either of the others. The fuel consumption was 26 per cent less than that of the steel cylinder and 20 per cent less than that of the capped cylinder.

A capped steel cylinder is usually not much better than the normal steel cylinder; however well fitted initially, "growth" and distortion of the aluminum impair the contact after a few hours' running.

[1] A. H. Gibson, *Inst. Aut. Eng.*, Feb., 1920.

The largest all-steel air-cooled cylinder tested by Gibson was 6 by 8 in. With a compression ratio of 4.48 and in a wind of 75 miles per hour this cylinder developed 115 lb. brake mean effective pressure on a fuel consumption of 0.68 lb. per brake-horse-power hour at 1,250 r.p.m., and 105 lb. brake mean effective pressure at 1,600 r.p.m. An aluminum-headed cylinder of the same dimensions developed under the same conditions 121 lb. brake mean effective pressure on a consumption of 0.56 lb. per brake horse power per hour.

Cylinder distortion may arise from the fact that the cooling air blast is directed against one side of the cylinder. Such distortion is negligible when the blast is directed on the exhaust side. This side is normally the hottest and needs most cooling. Tests on a 5½-in. aluminum cylinder with the blast on the exhaust side showed a maximum temperature difference between the front and back of the barrel of 58°C., and a mean difference of 19°C. With the blast on the inlet side the maximum temperature difference was 180°C. and the mean 120°C. In spite of this the cylinder, which was fitted with an aluminum piston of only 0.025-in. clearance, gave no sign of binding, showing that even in this extreme case the distortion was not serious.

With **longitudinal fins** and a comparatively uniform distribution of air flow, the distortion is not noticeably less. The exhaust side will be the hottest and the temperature will be less uniform than with circumferential fins and a free blast on the exhaust side. Furthermore, longitudinal fins do not stiffen the cylinder as strongly against distortion as do circumferential fins.

Connecting-rod Assembly.—The problem of connecting seven or nine big-ends to a single crankpin is usually solved either by the "articulated or master rod" assembly or by the "slipper" assembly.

The **master rod** assembly is used on the Gnome and Clerget rotaries and on most of the radials. Details of the assembly, as installed in the Salmson engine, are given in Figs. 155 and 156. The big end of the master rod encircles the crankpin, holds the wristpins for all the short rods, and carries the outer races of the ball bearings. It will be seen that this construction shortens the effective length of all rods except the master rod; that the axes of the short rods pass through the crankpin only twice in the revolution; and that the obliquity of the short rods is considerably greater than that of the master rod.

The **slipper** type of assembly is used in the Le Rhone and Anzani engines. The crankpin carries on ball bearings (Figs.

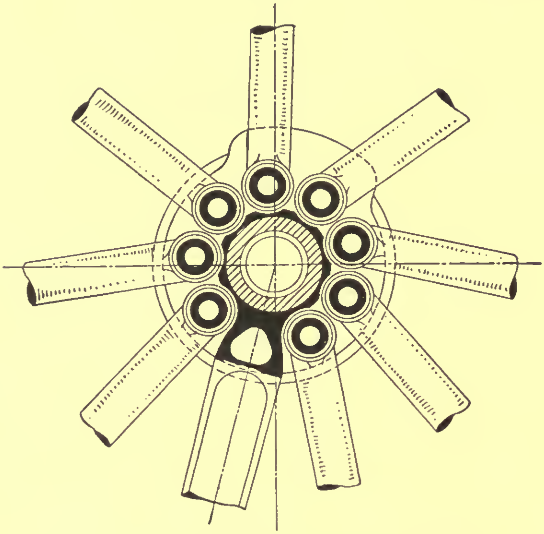


FIG. 155.—Articulated connecting-rod assembly.

157 and 158) two thrust blocks each of which has three annular grooves lined with bearing metal. The two discs are fastened

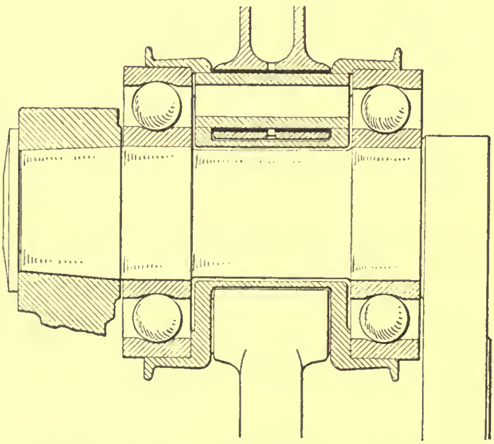


FIG. 156.—Section through articulated connecting-rod assembly.

together with the annular grooves opposite one another. The big ends of the nine connecting rods are provided with slippers

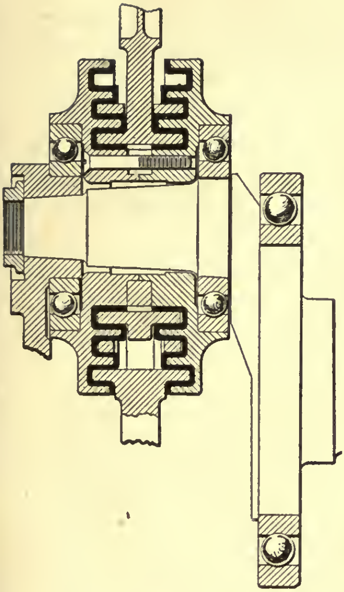


Fig. 157.—Section through slipper type connecting-rod assembly.

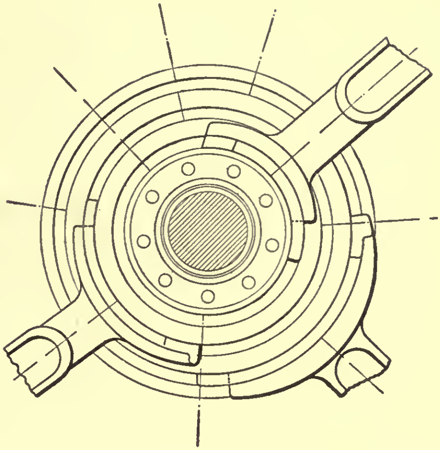


Fig. 158.—Assembly of slipper type connecting rods.

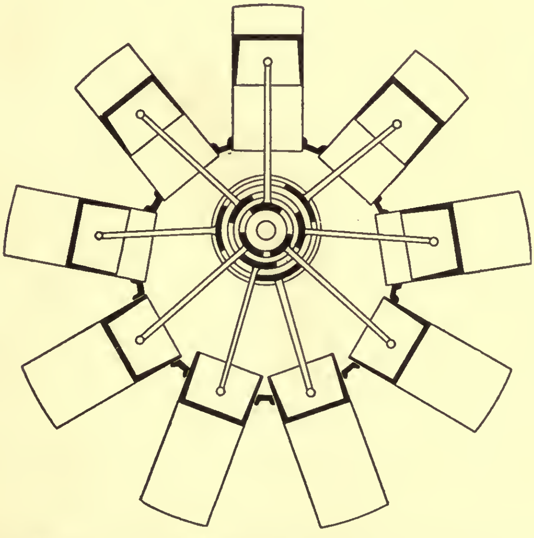


Fig. 159.—Diagram of rotary engine with slipper type connecting-rod assembly.

each of which is turned with the same radius of curvature as one of the annular grooves. Three connecting rods act on each groove and consequently there are three designs of slipper. The slippers for the middle and outermost grooves are slotted to avoid contact with the connecting rods for the innermost and middle grooves. The arrangement is shown in outline in Fig. 159. The plan of the slippers in Fig. 160 shows the slotting to prevent interference with adjacent connecting rods.

The slipper assembly is considerably heavier than the master-rod type and consequently is better adapted to rotaries than to radials. It has the advantage that the connecting rod is of maximum length and consequently of minimum angularity and also that the thrust (or tension) of the rod always passes through the center of the crankpin. Furthermore a large bearing surface is provided at the thrust block which is easily lubricated by the oil thrown off from the ball bearings.

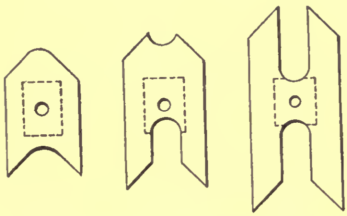


Fig. 160.—Projected views of slippers.

Dynamical Comparison of Radial and Rotary Engines.—The fixed-radial engine presents the special problem of a large mass rotating with the crankpin and consequently large centrifugal force. The inertia forces of the reciprocating parts are additive to this. The result is a considerable total pressure on the crankpin, which is relieved somewhat by the gas pressures during the explosion strokes. Roller or ball bearings are necessary at the crankpin if high speeds of rotation are to be maintained.

Balancing of the primary inertia forces of a single-crank fixed-radial engine is readily effected by a mass, approximately equal to half the mass of all the reciprocating parts, used as a counterbalance opposite the crankpin at crankpin radius. The counterbalance weight will add 7 to 10 per cent to the weight of the engine and can be avoided only by using two rows of cylinders and a double-throw crank. In the last case there is an unbalanced primary couple. Balancing the centrifugal and inertia pressures on the crankpin has been accomplished in an ingenious manner in the latest design of Cosmos "Jupiter" engine. Two bob-weights are suspended on the outer sides of the master rod;

their other ends are connected to the main crankshaft balance weights through hardened blocks working in slots machined in the bob-weights. The bob-weights serve not only to relieve the pressure on the crankpin but also as part of the weight necessary to balance the engine as a whole. The general arrangement of these bob-weights is shown in Fig. 161.

In rotary engines the pistons and connecting rods rotate about a stationary crankpin with an angular velocity which is variable. With the master-rod type of connecting-rod assembly there is

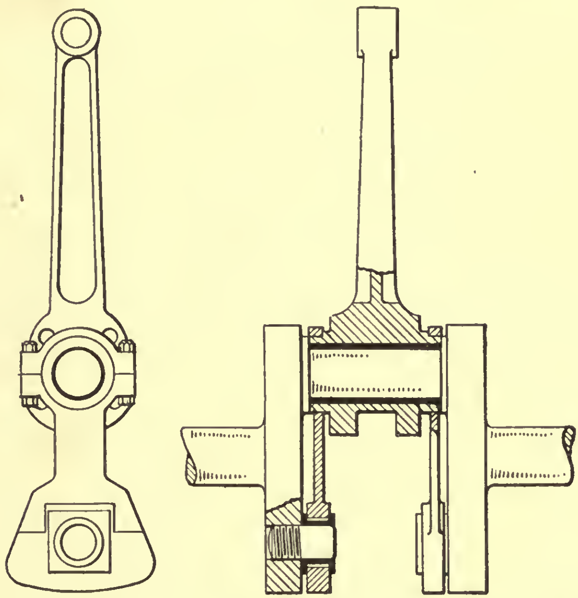


Fig. 161.—Balanced connecting rod of Cosmos "Jupiter" engine.

some lack of centrifugal balance at the crankpin, but it is usually negligible. The connecting rods are subjected to centrifugal tensional loading; the pressures on the pins at the ends of the rods increase as the square of the revolutions per minute. With the same connecting rod loading a fixed-radial engine may run at approximately twice the speed of a rotary engine with the same moving parts; before that speed is reached, however, the crankpin loading of the radial becomes excessive.

Ball and roller bearings for crankpins of radial engines offer special problems. The bearing rotates as a whole and presents conditions of loading quite unlike those of stationary bearings. Considerable investigation of this matter was made by the British Department of Aircraft Production[1] as a result of the failure of both caged and uncaged bearings.

An analysis of the situation showed that with cageless bearings the balls are crowded away from the center of rotation, by centrifugal force, and rub against one another. The balls rotate usually at about 2,500 r.p.m. about their own centers; the points that touch are always moving in opposite directions and the abrasion is considerable. When a cage is used the centrifugal force on the cage and the balls causes a displacement of the cage until the bearing load on the balls nearest the crank center due to the cage wedging between them is equal to the total centrifugal load on the cage. This causes a heavy abrasive action between the balls and the cage.

For successful operation it is necessary to have a cage which will carry independently the rubbing loads on each ball due to centrifugal force. To accomplish this (1) the cage must be strong enough to take the independent loads from the balls without distortion, (2) sufficient bearing surface must be provided at the surface of location of the cage to carry safely the total centrifugal load, (3) sufficient bearing surface must be provided between the balls and the cage to prevent wear on the cage, (4) the cage must be made of a metal of minimum abrasion, and (5) all surfaces must run with a continuous flow of oil.

To meet these requirements a cage as in Fig. 162 may be used. This type of cage must be definitely located and not displaceable by centrifugal force for more than a few thousandths of an inch. The design shown is made in two halves with eight hemispherical holes with 0.10 in. clearance for the balls. The outside circumference is turned in a flat V to avoid the actual ball path, and on either side of the V is a true cylindrical surface about $\frac{5}{16}$ in. wide. The cage is of phosphor bronze and fits the outer ball race with a clearance of 0.005 to 0.007 in. This bearing proved entirely satisfactory on the crankpin of a 10-cylinder, 115 by 150-mm. radial Anzani engine developing 150 h.p. at 1,300 r.p.m.

The details of a satisfactorily located cage for rollers for the crankpin for a 320-h.p. nine-cylinder radial engine making 1,700

[1] J. B. Swan, *The Automobile Engineer*, July, 1919.

r.p.m. are given in Fig. 163. The cage is located on the roller track, which in practice works out advantageously in polishing the track and keeping it free from foreign matter.

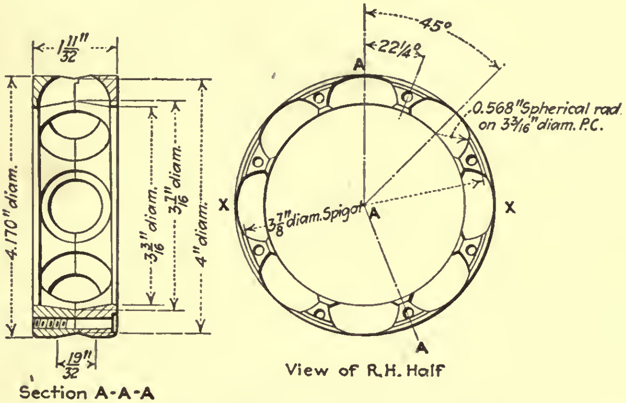


FIG. 162.—Located cage for ball bearings.

Details of successful and unsuccessful ball and roller bearings are given in the following table. At speeds above 1,600 r.p.m. and with a radius of rotation above 2½ in. cageless bear-

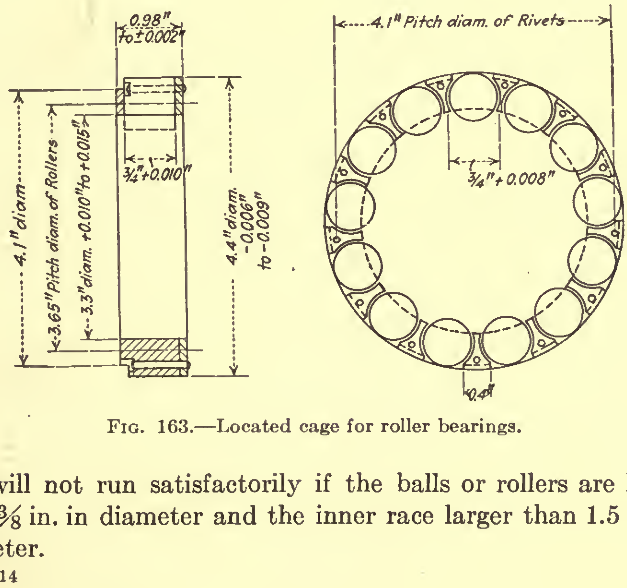


FIG. 163.—Located cage for roller bearings.

ings will not run satisfactorily if the balls or rollers are larger than ⅜ in. in diameter and the inner race larger than 1.5 in. in diameter.

ENGINES WITH UNCAGED ROLLER OR BALL BEARINGS

Type of engine	Horse power	Bore, inches	Stroke, inches	Revolutions per minute	Roller or ball diameter in inches	Number of rollers or balls	Diameter of crankpin, inches	Durability
2-cyl. opposed	5	2.75	2.52	3,000	5/16 rollers	12	0.93	Indefinite wear
2-cyl. opposed	6	2.75	2.52	4,000	5/16 rollers	12	0.93	Slight wear after 30 hr. running
2-cyl. opposed	40	4.33	4.73	2,000	3/8 rollers	14	1.5	Slight wear after 100 hr. running
6-cyl. radial	115	4.33	5.50	1,900	7/16 rollers	16	1.5	Slight wear after 30 hr. running
7-cyl. radial	170	4.53	5.90	1,800	9/16 rollers	14	2.01	Bad wear after 2 hr. running
10-cyl. radial	150	4.53	6.10	1,300	1 1/16 balls	9	2.06	Bad wear after 2 hr. running

ENGINES WITH LOCATED CAGES

Type of engine	Horse power	Bore, inches	Stroke, inches	Revolutions per minute	Roller or ball diameter in inches	Number of rollers or balls	Weight of cage, pounds	Bearings surface of cage, projected area, square inches	Durability
7-cyl. radial	170	4.53	5.90	1,800	9/16 rollers	12	0.288	1.212	Indefinite
9-cyl radial	320	5.50	6.50	1,700	3/4 rollers	13	0.722	1.801	Indefinite
10-cyl. radial	150	4.53	6.10	1,300	1 1/8 balls	8	1.090	1.95	Indefinite

The **crankshaft** in a radial engine will be solid or built-up according as the big end of the connecting rod has a plain bearing or a ball or roller bearing. The plain bearing is likely to give trouble in view of the heavy loading of the bearing except for low-speed engines, and can be used only with high-pressure forced lubrication; ball or roller bearings are very generally used.

The built-up crank necessary with ball or roller bearings may be either a two-web shaft with equal loading on front and rear bearings or an overhung crank with a drag crank for driving auxiliaries. The former type is the more desirable. With the overhung crank the main balance weight is on a single-crank web, which leads to an unbalanced couple, and also, the diameter of shaft and bearings has to be made greater.

Valve Operation.—Apart from the design of multilobed cams and their driving gears, the valve operation of radial engines does not present any special problems. In rotary engines the effects of centrifugal force on the push rods and tappets have to be met. Counterweights have sometimes been used on the valve side of the rocker arm, but their use increases the load and wear on the cam profile. The necessity for keeping down the over-all engine diameter is likely to result in the selection of an unfavorable type of valve spring and an undesirable reduction in the length of the valve stem guide; failures have been frequent when volute valve springs have been used.

Lubrication.—Rotary engines are always wasteful of oil, using almost $\frac{1}{10}$ lb. per brake-horse-power hour. There is no return of surplus oil to the pump, which consequently has to determine the amount of oil used. Plunger pumps are always used discharging directly to the main bearings, big end and cam gear, and relying largely on centrifugal force for the lubrication of wristpins and cylinders.

In radial engines either a plunger pump or a gear pump may be used with a dry sump. There is danger of over-oiling the lower cylinders. Most of the oil goes direct to the crankpin and is distributed thence by centrifugal force to the bearings, connecting-rod assembly, and cylinders. The oil consumption of radial engines runs from about 0.02 to 0.04 lb. per brake-horse-power hour.

CHAPTER IX

FUELS AND EXPLOSIVE MIXTURES

The properties desired in an airplane engine fuel are as follows:

1. It must have a high heat of combustion per pound. This determines the cruising radius for a given weight of fuel, since efficiency does not vary appreciably with the fuel.

2. It must have a high heat of combustion per cubic foot of explosive mixture if it is to develop high horse power per cubic foot of piston displacement. Alcohol and gasoline have about the same heats of combustion per cubic foot of explosive mixture but very different heats of combustion per pound of fuel. The heat of combustion is nearly constant for all the available fuels.

3. It must be able to withstand high compression without preignition or detonation.

4. It must vaporize readily (preferably with little or no preheating of the air) upon admixture with air and should be completely vaporized at the beginning of explosion. For good distribution it should be completely vaporized upon reaching the admission manifold, but this is not usually attained.

5. Combustion should be complete, leaving no solid residue in the cylinder.

6. The fuel and products of combustion must not be corrosive.

7. The explosion rate must be neither too rapid (as with hydrogen, acetylene and ether) nor too slow.

8. The bulk of fuel and the weight of the container must be low. This eliminates gaseous fuels.

The liquid fuels which meet the above conditions best are: (*a*) certain hydrocarbons, which form the constituents of gasoline and of certain coal tar products, and (*b*) the alcohols. The hydrocarbons under consideration may be divided into two main groups, saturated and unsaturated. The latter term is here applied to the behavior and not to the composition of the substance. The saturated hydrocarbons are again subdivided into the aliphatic or acyclic group, and into the aromatic or cyclic group. The hydrocarbons all form series in which the members differ from each other by the addition of CH_2. The members of the groups of most importance are listed in Table 9, together with

TABLE 9.—CLASSIFICATION AND PROPERTIES OF THE HYDROCARBONS OCCURRING IN VARIOUS ENGINE FUELS

		Group formula	Name	Chemical formula	Boiling point, degrees Centigrade	Density of the liquid	Latent heat, B.t.u. per pound
Saturated	Aliphatic and acyclic (paraffins)	C_nH_{2n+2}	Pentane	C_5H_{12}	36.0	0.645	
			Hexane	C_6H_{14}	69.0	0.676	156
			Heptane	C_7H_{16}	98.4	0.700	133
			Octane	C_8H_{18}	124.5	0.718	128
			Isopentane	C_6H_{12}	28.0	0.639	
			Isohexane	C_6H_{14}	61.0	0.673	
			Isoheptane	C_7H_{16}	90.0	0.697	
			Isooctane	C_8H_{16}	117.0	0.71	
	Aromatic and cyclic	C_nH_{2n-6} (Aromatics)	Benzene	C_6H_6	80.4	0.90	172
			Toluene	C_7H_8	110.3	0.88	151
			Xylene	C_8H_{10}	141.0	0.85	148
		C_nH_{2n} (Naphthenes)	Cyclopentane	C_6H_{10}	51.0	0.75	
			Cyclohexane	C_6H_{12}	81.0	0.80	155
Unsaturated	Acyclic	C_nH_{2n}	Ethylene	C_2H_4	−103.0		
			Propylene	C_3H_6			
			Butylene	C_4H_8			
			Amylene	C_6H_{10}			
			Hexylene	C_6H_{12}	69.0	0.754	
		C_nH_{2n-2}	Acetylene	C_2H_2	−82		
			Allylene	C_3H_4			

The members of the isopentane group are isomers of the members of the pentane group, that is, they have the same chemical compositions but different physical properties. The same is true of the naphthalene and ethylene groups.

the boiling points (temperatures of vaporization at atmospheric pressure), densities as compared with that of water and latent heats of evaporation.

Certain properties of these compounds are given in the following tables, in which there are also included, for convenience, the properties of air, and its constituents, and of the products of combustion of the fuel elements. In Table 10 data are given for the gaseous state only. Column 3 gives the density of each substance compared with air at the same pressure and temperature; column 4 gives the weight in pounds per cubic foot of the substance. Some of the quantities there given are fictitious in that the substance is not gaseous at 32°F. and 760 mm., but the quantity is of value in permitting the easy determination of specific weight at those temperatures and pressures at which it is gaseous. Column 5 is the reciprocal of column 4. The quantity R is the constant in the gas equation $pv = wRT$.

In Table 11 are given combustion data for the fuel constituents. Column 4 gives the volume of air necessary to burn one volume of the gaseous fuel, both being at the same pressure and temperature; the products of complete combustion are in all cases CO_2, H_2O and N_2 and their volumes are given in columns 5, 6 and 7. The mixture usually experiences a change in volume as a result of the chemical changes resulting from combustion, entirely independent of the change in pressure and temperature; this change in volume is given in column 8. Columns 9 to 12 give the weight of air required for combustion of 1 lb. of fuel and the weights of each of the resulting products.

Table 12 gives the heat of combustion of each of the substances listed. There is also given the heat of combustion per pound of explosive mixture, and the heat of combustion per cubic foot of explosive mixture measured at 60°F. and standard atmospheric pressure, the mixture being assumed to contain only that amount of air which is chemically necessary. It will be noted that two values of heat of combustion are given under each head, a higher and a lower heat value, and that these are different for all the fuels which contain hydrogen. The higher heat value is the total heat liberated by combustion, or the heat which would be given up by the mixture when burned in a closed vessel and cooled to its initial temperature. Whenever there is hydrogen in the fuel, water is formed by combustion and part of the heat of combustion is absorbed in vaporizing it. If,

TABLE 10.—PROPERTIES OF FUELS IN THE GASEOUS STATE

	Chemical formula	Relative density, air = 1 (Gaseous state)	Specific weight, pounds per cubic foot at 32°F. and 760 mm.	Specific volume, cubic feet per pound at 32°F. and 760 mm.	R (varies inversely as the density)	Specific heat at constant pressure, C_p.	Specific heat at constant volume, C_v	Ratio of specific heats, $\frac{C_p}{C_v}$
Oxygen	O_2	1.105	0.0892	11.21	48.1	0.217	0.156	1.40
Nitrogen	N_2	0.967	0.0783	12.78	55.0	0.244	0.173	1.41
Air	......	1.0000	0.0807	12.39	53.2	0.237	0.169	1.40
Hydrogen	H_2	0.0696	0.00562	177.9	764.0	3.41	2.42	1.41
Steam	H_2O	0.622*	0.0502*	19.91*	85.5*	0.480	0.373	1.29
Carbon monoxide	CO	0.967	0.0781	12.80	55.0	0.242	0.173	1.40
Carbon dioxide	CO_2	1.529	0.1227	8.15	34.8	0.202	0.156	1.30
Methane	CH_4	0.558	0.0447	22.37	95.4	0.58	0.453	1.28
Ethane	C_2H_6	1.075	0.0838	11.93	49.5			
Propane	C_3H_8	1.522*	0.1228*	8.14*	34.9*			
Butane	C_4H_{10}	2.015	0.1619	6.175	26.5	0.37	0.30	1.23
Pentane	C_5H_{12}	2.490*	0.2009*	4.98*	21.37*			
Hexane	C_6H_{14}	3.053	0.2464	4.06	17.42			
Heptane	C_7H_{16}	3.547	0.2862	3.50	15.00			
Octane	C_8H_{18}	3.992	0.3222	3.10	13.32			
Nonane	C_9H_{20}	4.600	0.371	2.694	11.56			
Benzene	C_6H_6	2.697*	0.2177*	4.60*	19.73*			
Toluene	C_7H_8	3.18*	0.2572*	3.89*	16.73*			
Xylene	C_8H_{10}	3.69*	0.298*	3.35*	14.42*			
Cyclo-hexane	C_6H_{12}	2.91*	0.235*	4.25*	18.30*			
Ethylene	C_2H_4	0.985	0.0782	12.79	54.0	0.37	0.30	1.23
Propylene	C_3H_6	1.498	0.1173	8.53	35.5			
Butylene	C_4H_8	1.937*	0.1563*	6.40*	27.5*			
Acetylene	C_2H_2	0.92	0.0725	13.79	57.8	0.35	0.27	1.28
Alylene	C_3H_4	1.383*	0.1116*	8.96*	38.4*			
Naphthalene	$C_{10}H_8$	4.42*	0.357*	2.80*	12.02*			
Methyl alcohol	CH_4O	1.12	0.0904	11.06	47.5	0.458		
Ethyl alcohol	C_2H_6O	1.617	0.1283	7.795	32.9	0.453	0.409	1.107

* From computed data.

TABLE 11.—PRODUCTS OF COMBUSTION

Fuel	Chemical formula	Molecular weight	Volume of air necessary for combustion of unit volume of fuel at same temperature and pressure	Products of combustion of 1 cu. ft. of fuel in theoretical amount of air			Volume after combustion in theoretical air in per cent of total mixture volume	Weight of air necessary for combustion of unit weight of fuel	Products of combustion of 1 lb. of fuel in theoretical amount of air		
				CO_2 cu. ft.	H_2O cu. ft.	N_2 cu. ft.			CO_2 lb.	H_2O lb.	N_2 lb.
Oxygen	O_2	32	....								
Nitrogen	N_2	28.08									
Air											
Hydrogen	H_2	2.016	2.387	0	1	1.887	85.2	34.2	0.0	8.94	26.28
Steam	H_2O	18.016									
Carbon monoxide	CO	28.00	2.387								
Carbon dioxide	CO_2	44.00									
Methane	CH_4	16.03	9.55	1	2	7.547	100.0	17.21	2.745	2.248	13.22
Ethane	C_2H_6	30.05	16.71	2	3	13.21	102.8	16.07	2.929	1.799	12.34
Propane	C_3H_8	44.06	23.87	3	4	18.87	104.0	15.65	2.996	1.635	12.02
Butane	C_4H_{10}	58.1	31.03	4	5	24.53	104.7	15.44	3.03	1.551	11.86
Pentane	C_5H_{12}	72.1	38.2	5	6	30.2	105.1	15.31	3.05	1.499	11.76
Hexane	C_6H_{14}	86.1	45.3	6	7	35.8	105.4	15.22	3.07	1.465	11.69
Heptane	C_7H_{16}	100.1	52.5	7	8	41.5	105.6	15.15	3.08	1.439	11.64
Octane	C_8H_{18}	114.1	59.7	8	9	47.2	105.8	15.11	3.08	1.421	11.60
Nonane	C_9H_{20}	128.2	66.8	9	10	52.8	105.9	15.07	3.09	1.406	11.57
Benzene	C_6H_6	78.0	35.8	6	3	28.3	101.4	13.26	3.38	0.693	10.18
Toluene	C_7H_8	92.1	42.9	7	4	34.0	102.3	13.50	3.35	0.783	10.36
Xylene	C_8H_{10}	106.2	50.1	8	5	39.6	102.9	13.57	3.31	0.845	10.42
Cyclo-hexane	C_6H_{12}	84.0	43.0	6	6	34.0	104.5	14.76	3.14	1.285	11.34
Ethylene	C_2H_4	28.03	14.32	2	2	11.32	100.0	14.76	3.14	1.285	11.34
Propylene	C_3H_6	42.0	21.48	3	3	16.98	102.2	14.76	3.14	1.285	11.34
Butylene	C_4H_8	64.1	28.64	4	4	22.64	103.2	14.76	3.14	1.285	11.34
Acetylene	C_2H_2	26.02	11.93	2	1	9.43	96.1	13.26	3.38	0.693	10.18
Allylene	C_3H_4	40.0	19.09	3	2	15.09	100.0	13.78	3.30	0.900	10.59
Naphthalene	$C_{10}H_8$	128.1	57.3	10	4	45.28	101.7	12.93	3.44	0.563	9.93
Methyl alcohol	CH_4O	32.0	7.16	1	2	5.66	106.1	6.46	1.374	1.125	4.96
Ethyl alcohol	C_2H_6O	46.0	14.32	2	3	11.32	106.5	8.99	1.911	1.174	6.90

TABLE 12.—HEATS OF COMBUSTION

Substance	Chemical formula	B.t.u. per pound		B.t.u. per cubic foot of theoretical mixture at 60°F. and 760 mm.		B.t.u. per pound of theoretical mixture	
		High	Low	High	Low	High	Low
Hydrogen	H_2	62,100	52,920	95.6	81.5	1,760	1,500
Carbon-monoxide	CO	4,380	4,380	94.6	94.6	1,265	1,265
Methane	CH_4	23,850	21,670	95.8	87.0	1,310	1,190
Ethane	C_2H_6	22,230	20,500	99.7	91.8	1,300	1,200
Propane	C_3H_8	21,600	20,055	101.2	94.0	1,297	1,205
Butane	C_4H_{10}	21,240	19,780	101.7	94.6	1,285	1,203
Pentane	C_5H_{12}	21,140	19,600	103.0	95.6	1,297	1,203
Hexane	C_6H_{14}	20,800	19,380	102.5	95.4	1,284	1,197
Heptane	C_7H_{16}	20,600	19,200	102.0	95.0	1,275	1,188
Octane	C_8H_{18}	20,400	19,020	101.4	94.5	1,266	1,176
Nonane	C_9H_{20}	20,380	19,015	101.7	95.0	1,266	1,184
Benzene	C_6H_6	18,070	17,400	101.0	97.2	1,184	1,140
Toluene	C_7H_8	18,250	17,490	100.8	96.5	1,261	1,208
Cyclo-hexane	C_6H_{12}	......	18,900		95.5		
Ethylene	C_2H_4	21,600	20,420	104.9	99.2	1,371	1,295
Propylene	C_3H_6	21,330	20,150	104.8	99.3	1,356	1,280
Butylene	C_4H_8	20,880	19,700	104.4	98.5	1,325	1,254
Acetylene	C_2H_2	21,600	21,020	115.0	112.0	1,515	1,475
Allylene	C_3H_4	21,200	20,325	112.1	107.0	1,434	1,375
Naphthalene	$C_{10}H_8$	17,410	16,860	101.1	98.0	1,250	1,210
Methyl alcohol	CH_4O	9,550	8,460	99.5	88.2	1,281	1,133
Ethyl alcohol	C_2H_6O	13,000	11,650	105.2	94.3	1,303	1,167

as is usual in gas engines, the gases escape at so high a temperature that no water vapor is condensed in the cylinder, the latent heat of vaporization of the water is not available for raising the temperature of the products of combustion, or for doing work. The useful heat of combustion is consequently the total heat less the heat absorbed in vaporizing the H_2O formed by the combustion. The heat of vaporization depends on a number of factors. A value of 950 B.t.u. per pound may be assumed and if this is multiplied by the number of pounds of water formed per pound of fuel burned it will give (with an approximation adequate for ordinary purposes) the unavailable heat. The lower heat value is the total heat minus the unavailable heat and is the value commonly used by engineers in dealing with gas engine problems.

Gasoline.—The gasolines at present on the market are of three different types:[1]

[1] DEAN, *Motor Gasoline*, Technical Paper 166, U. S. Bureau of Mines.

1. "Straight" refinery gasoline.
2. Blended casing-head gasoline.
3. Cracked and blended gasoline.

"Straight" refinery gasoline is produced by distillation. Crude petroleum is first distilled from a fire still, and the condensed product is collected until it reaches some predetermined density. This so-called *crude naphtha* or *benzine* is then acid-refined and steam-distilled. Several products of different ranges of volatility may be produced or the steam distillation may simply separate the product from the less volatile "bottoms." Straight refinery gasolines consist mainly of aliphatic hydrocarbons (see Table 9) and are generally characterized by a low content of unsaturated and aromatic hydrocarbons and by a distillation range free from marked irregularities.

Blended casing-head gasolines are of recent development. Casing-head gasoline is obtained, by compression or absorption, from natural gas and is too volatile for general use. Before marketing, it is generally blended with enough heavy naphtha to produce a mixture that can be handled safely. As a result of this blending, the volatility range is usually characterized by a considerable percentage of constituents of both low and high boiling points and a lack of intermediate constituents. Skilful blending may change this characteristic.

In chemical properties the blended casing-head gasoline seems to be identical with a straight refinery product of the same distillation range.

Cracked or **synthetic gasoline** is also a recent development. An oil consisting mainly of heavier hydrocarbons is subjected to high temperature and pressure and is thereby broken down or "cracked" into lighter constituents. This cracked gasoline is generally marketed in the form of blends with refinery and casing-head gasoline. Cracked gasolines differ chemically from straight-refinery and casing-head gasolines in having a larger amount of unsaturated and aromatic hydrocarbons. The heat of combustion of the aromatic compounds averages about 15 per cent less than that of the acyclic compounds, but, as shown on page 238, the presence in moderate amounts of certain aromatic compounds may improve the thermal efficiency of the engine enough to offset any disadvantage, for airplane use, of a lower heat of combustion.

Specifications.—In the past, gasolines have usually been described and bought on a **gravity** specification. So long as the gasolines in the market were straight refinery products such a specification was reasonably satisfactory, but, with the development of blended casing-head gasolines, it has become impossible to determine the volatility of a gasoline by density measurements. A given density may represent either a narrow "cut" consisting of a product which evaporates with a very narrow range of temperature, or a mixture of a volatile low-density product with a product of high density and low volatility. The former fuel might be admirable for airplanes while the latter might be quite unsuitable.

Specific gravity is best expressed as the ratio of the density of the fuel to that of water, both at 60°F. The trade practice has been to use the Baumé scale of density. This arbitrary scale has nothing to recommend it, and suffers the disadvantage

TABLE 13.—SPECIFIC GRAVITIES AT $\frac{60°}{60°}$ F. CORRESPONDING TO DEGREES BAUMÉ FOR LIQUIDS LIGHTER THAN WATER

Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity
25	0.9032	40	0.8235	55	0.7568	70	0.7000	85	0.6512
26	0.8974	41	0.8187	56	0.7527	71	0.6965	86	0.6482
27	0.8917	42	0.8140	57	0.7487	72	0.6931	87	0.6452
28	0.8861	43	0.8092	58	0.7447	73	0.6897	88	0.6422
29	0.8805	44	0.8046	59	0.7407	74	0.6863	89	0.6393
30	0.8750	45	0.8000	60	0.7368	75	0.6829	90	0.6364
31	0.8696	46	0.7955	61	0.7330	76	0.6796	91	0.6335
32	0.8642	47	0.7910	62	0.7292	77	0.6763	92	0.6306
33	0.8589	48	0.7865	63	0.7254	78	0.6731	93	0.6278
34	0.8537	49	0.7821	64	0.7216	79	0.6699	94	0.6250
35	0.8485	50	0.7778	65	0.7179	80	0.6667	95	0.6222
36	0.8434	51	0.7735	66	0.7143	81	0.6635	96	0.6195
37	0.8383	52	0.7692	67	0.7107	82	0.6604	97	0.6167
38	0.8333	53	0.7650	68	0.7071	83	0.6573	98	0.6140
39	0.8284	54	0.7609	69	0.7035	84	0.6542	99	0.6114
..	......	..	......	..	......	..	......	100	0.6087

that the greater the density the lower is the number of "degrees" on the Baumé scale. For liquids lighter than water, the relation between the specific gravity and Baumé scale, B, is given by the expression

$$\text{Specific gravity} = \frac{140}{130 + B}.$$

Numerical values are given in Table 13. Commercial gasolines range from about 55 to 75°Bé. (Sp. gr. 0.758 to 0.684). The Eastern gasolines are lightest (60 to 75°Bé.) and the California gasolines heaviest (57 to 63°Bé.).

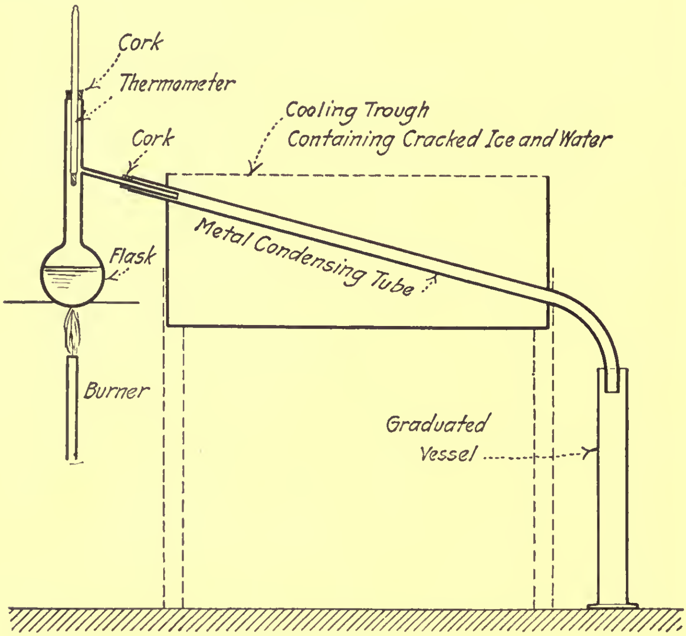


Fig. 164.—Distillation apparatus.

Volatility.—Volatility is the basic property that determines the grade and usefulness of a gasoline. The presence of low-boiling constituents is desirable to permit easy starting of a cold motor but may result in excessive evaporation losses in the commercial handling of the fuel.

The volatility is determined by distillation (Fig. 164). A 100-gram sample of the fuel is heated slowly while the vapor given off is condensed and collected. The first drop of gasoline should fall from the end of the condenser tube in 5 to 10 min. The rate of evaporation is kept about 4 c.c. per minute. A thermometer indicates the temperature of the vapor above the fuel. Readings of this temperature are taken when the first drop of distillate falls (initial point) and as each 10 per cent or other selected percentage has distilled, until at the end a dry point is reached. The observations are usually plotted as in Fig. 165 and give a record of the volatility of the fuel.

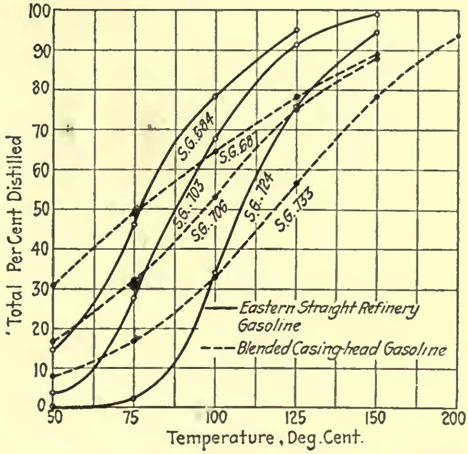


FIG. 165.—Distillation curves for straight refinery and casing-head gasoline.

Specifications for aviation gasoline have been prepared by the U. S. Committee on Standardization of Petroleum Specifications[1] and are as follows:

GASOLINE DISTILLATION TEST SPECIFICATIONS

Grade	Aviation gasoline, domestic grade	Aviation gasoline, fighting grade
Thermometer reading range when 5 per cent is recovered in receiver	122 to 167°F.	122 to 149°F.
Thermometer reading when 50 per cent is recovered in receiver, not more than	221°F.	203°F.
Thermometer reading when 90 per cent is recovered in receiver, not more than	311°F.	257°F.
Thermometer reading when 96 per cent is recovered in receiver, not more than	347°F.	302°F.
End-point shall not be higher than	374°F.	329°F.
Distillate recovered in the receiver from the distillation at least	96 per cent	96 per cent
When the residue is cooled and added to the distillate in the receiver the distillation loss shall not exceed	2 per cent	2 per cent

[1] Bureau of Mines, *Bulletin* No. 5, effective Dec. 29, 1920.

In addition the specifications require for both grades of aviation gasoline the following properties:

Color: Water white.

Doctor test: Negative.

Corrosion test: 100 c.c. of the gasoline shall cause no gray or black corrosion and no weighable amount of deposit when evaporated in a polished copper dish.

Unsaturated hydrocarbons: maximum proportion of the gasoline soluble in concentrated sulphuric acid, 2 per cent.

Acid heat test: the gasoline shall not increase in temperature more than 10°F.

Acidity: the residue after distillation shall not show an acid reaction.

The gasoline shall be free from undissolved water and suspended matter.

The Doctor Test is made by shaking two volumes of gasoline with one volume of "doctor" solution (sodium plumbite) in a test tube, shaking for 15 sec., adding a pinch of flowers of sulphur, shaking again and allowing to settle. If the liquid remains unchanged in color and the sulphur remains bright or only slightly discolored, the test is negative and the gasoline is "sweet."

The Acid Heat Test is made by adding 30 c.c. of 66° commercial sulphuric acid to 150 c.c. of gasoline, both being at room temperature. After mixing, shake for 2 min. and observe the rise in temperature.

Volatility curves for three straight refinery gasolines and for three blended casing-head gasolines, of approximately the same densities, are given in Fig. 165. The casing-head gasolines are seen to have larger percentages distilled below 50°C., but have longer distillation ranges. This results in a fairly uniform slope of the distillation curve. The large percentage unevaporated at 150°C. shows that the fuel is a blend or mixture of heavier and lighter fuels.

It should be noted further that the "cut" or fraction distilling off at any given temperature will be different from different gasolines. This is demonstrated in Fig. 166, which shows that the 100°C. cut may vary in specific gravity from 0.710 to 0.733 and the 150°C. cut from 0.748 to 0.780. In other words, volatility alone is not sufficient to characterize a gasoline.

From the volatility curves for straight refinery gasoline, Fig. 165, it will be seen that the average temperature of evaporation (boiling temperature) from a high-grade gasoline is 100°C. From the specific gravity curves it appears that the average density at

100°C. is almost 0.7. As the constituents are mainly aliphatic hydrocarbons it is safe to assume that a high-grade gasoline consists principally of hexane (C_6H_{14}) and heptane (C_7H_{16}), whose boiling points are 69 and 98.4°C. and densities 0.676 and 0.7, respectively (see Table 9).

Calorific Value.—The calorific value of commercial gasoline varies very slightly with type of fuel, field of origin, or density. Exhaustive tests by the U. S. Bureau of Mines show only 1.5 per cent difference between the highest and lowest values for a range of density from 0.687 to 0.745 (73.8 to 57.9°Bé.), the samples investigated including all the commercial types. The average high heat value is 20,200 B.t.u. per pound. It should be noted, however, that gasoline is sold by the gallon and that there is considerable difference on that basis; the calorific value ranges from 124,000 B.t.u. per gallon for sp. gr. 0.745, to 116,500 B.t.u. for sp. gr. 0.687, a difference of over 7 per cent in favor of the heavier fuel. This difference is not important in airplane use, since the weight of fuel that has to be carried is the important factor, and not its volume; in automobile use it may more than offset the disadvantages resulting from the use of a less volatile fuel.

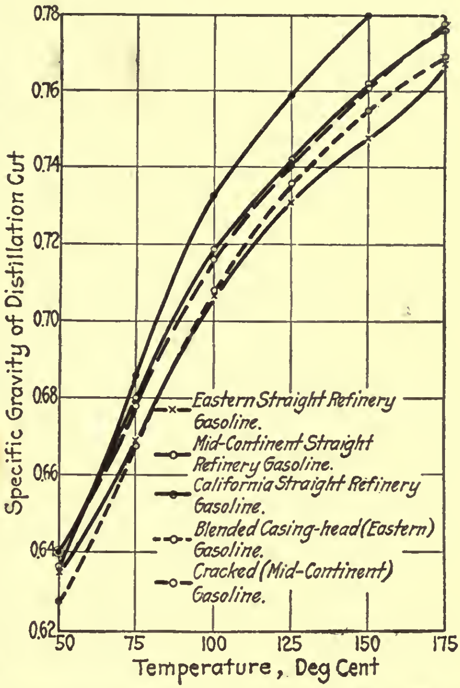


FIG. 166.—Density of "cuts" from various gasolines.

Benzene or **benzol** (C_6H_6) is a fuel which has been used considerably in airplanes, though generally mixed with gasoline. It is obtained chiefly from by-product coke-ovens.

Commercial 90 per cent benzol has a specific gravity of about 0.88. Its distillation curve should show an initial point not lower than 74°C., 90 per cent at or below 86°C., 95 per cent at or below 95°C., and end point not above 150°C. With a calorific value of 18,000 B.t.u. per pound the heating value per gallon is 132,000 B.t.u., or considerably higher than that of gasoline.

When mixed with gasoline there is no change in total volume. The distillation curve for such a mixture, containing 20 per cent of benzol and 80 per cent high-grade gasoline, is shown in Fig. 167, together with the distillation curves of the benzol and the gasoline.

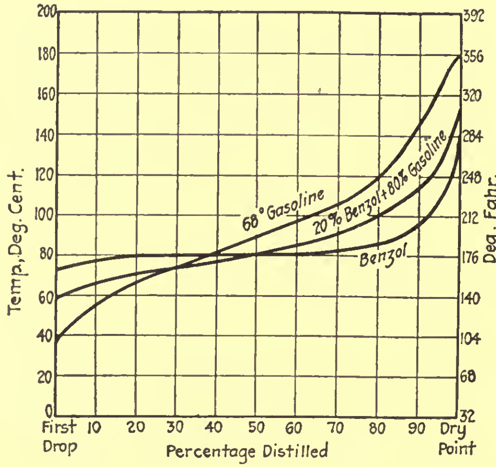


FIG. 167.—Distillation curve of benzol-gasoline mixture.

It will be observed that the effect of the addition of benzol is to increase the volatility of the mixture; with 30 to 50 per cent distilled the volatility is greater than that of either of the constituents, after which it becomes intermediate to the volatility of the constituents. This improvement in volatility has been found to be distinctly advantageous in increasing engine power at high altitudes. The lower heat of combustion of the mixture results in the consumption of a greater weight of fuel per brake horse-power hour than with straight gasoline.

Alcohol has been used mixed with gasoline or benzol or both, as an airplane fuel. Methyl alcohol, CH_4O (wood alcohol) has a boiling point of 65°C. and sp. gr. 0.81, heats of combustion, high 9,550, low 8,460, B.t.u. Ethyl alcohol, C_2H_6O (grain alcohol) has a boiling point of 78°C., density 0.79, heats of combustion, high 13,000, low 11,650 B.t.u. Commercial alcohol, either pure or denatured, contains water (*e.g.*, 10 per cent by volume in 90 per cent alcohol) and has a higher boiling point than pure alcohol. The effect of the addition of alcohol to gasoline is to improve the volatility. The calorific value of alcohol is so low compared with gasoline that its use inevitably increases the weight of fuel burned per unit of power developed. It does not, however, diminish the power developed, because the heat of combustion per unit volume of explosive mixture (see Table 12) is about the same as for gasoline; it may even increase the power output slightly. Its use also permits an increase in the permissible compression ratio for the engine and thereby improves the thermal efficiency.

Hydrogen has the highest calorific value of any of the fuels, per pound, but not per cubic foot of explosive mixture (Table 12). Apart from its high cost, it is objectionable on account of the great violence of the explosion when mixed with the proper amount of air. It cannot be carried in airplanes unless compressed to very high pressure in steel tanks, which makes the fuel system too heavy, or in the liquid form, which increases greatly the cost of the fuel. Liquid hydrogen has a temperature below −400°F. and cannot be kept from evaporating rapidly without the very best of heat insulation; no sufficiently robust container with adequate heat-insulating qualities has been devised as yet. Hydrogen gas has been used for starting cold engines.

It is often necessary to waste some of the hydrogen contained in a dirigible balloon. Attempts to burn the hydrogen alone in the engine have shown that only about one-third of the maximum horse power of the engine could be developed without serious detonations. By mixing hydrogen with the gasoline it is possible to develop the maximum power without trouble and thereby to save gasoline; at the higher powers only a small quantity of hydrogen can be burned.

Acetylene, (C_2H_2), like Hydrogen, gives explosions of great violence in the cylinder. Its heat of combustion per cubic foot of explosive mixture is highest of all the fuels given in Table 12. It may be stored either in the gaseous or liquid forms, but with the same objections (though to a less degree) as hydrogen. It can be generated by adding water to calcium carbide, leaving a residue of slaked lime. As the residue is considerably heavier than the acetylene, the total weight of the fuel system becomes excessive, if it is attempted to generate the acetylene in an airplane.

Ether has, as its principal advantage, the fact that its boiling point (35°C.) is lower than that of any of the other available fuels which are liquid at ordinary temperatures. This gives it a special value in starting a cold engine. Its use has been restricted to that purpose. The heat of combustion is rather low.

EXPLOSIVE MIXTURES

Properties of Vapors.—Every liquid gives off vapor continuously until the pressure exerted by that vapor at the surface of the liquid reaches a limiting value, which depends, for any given liquid, on its temperature only. The vapor liberated is always at the temperature of the liquid and it is said to be "saturated" when it is at the limiting pressure. The relation between the pressure and temperature of a saturated vapor is determinable only by experiment.

The pressure exerted by a vapor will, in time, reach the saturation pressure if the liquid is contained in a vessel of moderate dimensions; the presence, above the liquid, of gases or other vapors which are inert to the vapor under consideration and are at the same temperature will not affect the saturation pressure. The total pressure in the vessel (assuming no change of temperature) will be the sum (1) of the pressures of the gases and vapors already there and (2) of the saturation pressure of the liquid.

If the containing vessel is very large or if the time available is too short, or if the weight of liquid put into the vessel is less than the weight of saturated vapor necessary to fill the vessel, the vapor will have a pressure less than the saturated pressure and will be in the condition known as "superheated." Suppose the temperature of the superheated vapor to be T. If this vapor is cooled at constant pressure, with consequent diminution in volume, a temperature, T_o, will eventually be reached at which the vapor is saturated. The cooling process is similar to that used for determining the dew-point of air; the dew-point is the saturation temperature. The vapor is said to be superheated T-T_o degrees. All unsaturated vapors are superheated. When superheated they may be considered to behave like perfect gases.

A saturated vapor cannot exist, as such, at a pressure greater than the saturation pressure. If a saturated vapor is cooled at constant volume, thereby lowering the saturation pressure, some of the vapor will condense. If a saturated vapor is compressed, keeping the temperature constant, condensation will take place; if, on the other hand, it is expanded at constant temperature it will become unsaturated (superheated) unless liquid is present to supply more vapor by evaporation. The presence of other inert vapors will not affect these phenomena.

If air is passed through or over a liquid (as in certain obsolete types of carburetor), and if the contact is sufficiently intimate and prolonged, the air will leave carrying with it the saturated vapor of the liquid. If a liquid is injected into a current of air (as in modern carburetors) and if the contact is sufficiently intimate and prolonged and if, furthermore, the weight of liquid present is sufficient for that purpose, the air will carry its own volume of the saturated vapor of the liquid. If more liquid is injected than is necessary for this purpose the excess liquid will remain in the liquid state. In any case, the total pressure of the *carbureted air* is the sum of the *partial* pressures of the air and of the vapor. If the pressure of the carbureted mixture is p, and the pressure of the vapor is p_v, and of the air in the carbureted mixture is p_a, then

$$p = p_a + p_v$$

The relation between the saturation pressures and temperatures of the liquid fuels which are of importance in airplane engines is given in Fig. 168. Table 14 gives their values for certain selected temperatures.

The *specific volumes* (volumes of 1 lb.) of the saturated vapors are calculated on the assumption that they are perfect gases. This assumption is fairly satisfactory for the low vapor pressures which alone are of interest in engine mixtures. Taking, for example, heptane (C_7H_{16}) at 60°F., the molecular weight is $7 \times 12 + 16 = 100$. The gas constant R is inversely as the molecular weight of the gas; taking R for oxygen as 48.25, R for heptane is $\frac{32}{100} \times 48.25 = 15.45$, and the specific volume at 60°F. and at the saturation pressure of 0.54 lb. per square inch is

$$v = \frac{RT}{p} = \frac{15.45 \times 520}{0.54 \times 144} = 103 \text{ cu. ft.}$$

The *weight of air required for combustion* is obtained from the chemical equation,

$$C_7H_{16} + 11O_2 = 7CO_2 + 8H_2O$$

The relative weights of heptane and oxygen are 100 and 11 × 32. As the oxygen content of air is 23.4 per cent by weight, the air required for the combustion of 1 lb. of heptane is $\frac{11 \times 32}{100} \times \frac{100}{23.4} = 15.1$ lb. The *volume of this air* at 60°F. and 14.7 lb.

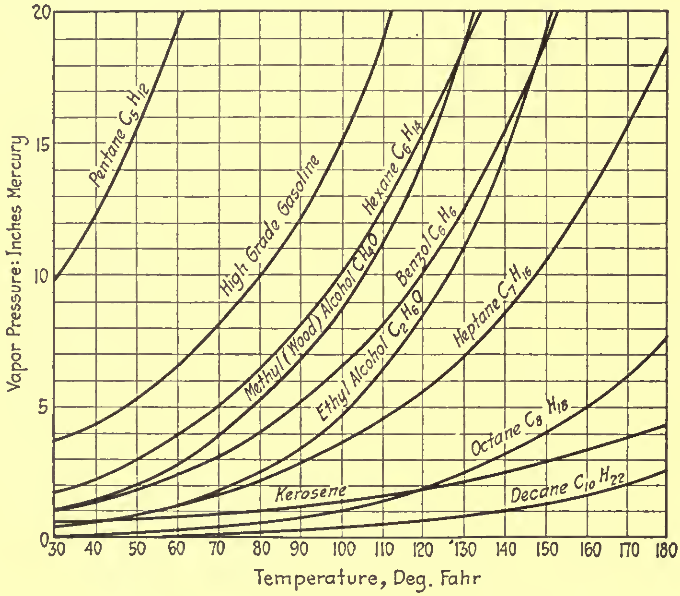


FIG. 168.—Vapor pressures of various liquid fuels.

per square inch pressure is given by the equation $v = \frac{wRT}{p} = \frac{15.1 \times 53.4 \times 520}{14.7 \times 144} = 197$ cu. ft. approx.

It is desirable that the fuel entering the inlet manifold should be entirely in the vapor form, either superheated or just saturated. This condition is necessary to obtain a homogeneous mixture and an equal distribution of the fuel to all the cylinders. The possibility of obtaining this condition may be determined by continuing the preceding calculation. If the air is saturated with heptane vapor at 60°F. its partial pressure, p_a, will be 14.7 − 0.54 = 14.16 lb. per square inch and the volume of the air at this

TABLE 14.—PRESSURES AND SPECIFIC VOLUMES OF SATURATED VAPORS OF VARIOUS FUEL ELEMENTS

Fuel	Symbol	Boiling point (at 14.7 pound pressure) degrees Fahrenheit	Specific gravity of the liquid (water = 1)	Vapor pressures, pounds per square inch, at				Approximate specific volume of saturated vapor, cubic feet per pound at				Air required for combustion, cubic feet per pound of fuel				Mean specific heat of liquid between 32° and 100°F.
				40°F.	60°F.	80°F.	100°F	40°F.	60°F.	80°F.	100°F	40°F.	60°F.	80°F.	100°F	
Pentane	C_6H_{12}	97	0.645	5.92	9.37	13.8	.....	12.6	8.27	5.83	....	196	204	212	219	
Hexane	C_6H_{14}	156	0.676	1.11	1.88	3.11	4.91	59.9	36.7	23.1	15.2	194	202	210	218	0.516
Heptane	C_7H_{16}	209	0.700	0.290	0.54	0.89	1.16	185	103	59	37	194	202	209	217	0.495
Octane	C_8H_{18}	258	0.718	0.058	0.14	0.27	0.52	813	352	188	101	193	201	208	216	0.508
Nonane	C_9H_{20}	304	0.741	0.039	0.077	0.15	0.29	1,089	565	313	162	193	200	207	216	0.503
Decane	$C_{10}H_{22}$	343	0.755	0.023	0.044	0.068	0.148	1,630	880	572	292	192	200	207	215	0.544
Benzol	C_6H_6	177	0.90	0.65	1.14	1.97	3.19	106	63	38	24	169	176	183	190	0.41
Methyl alcohol	CH_4O	148	0.812	0.66	1.33	2.46	4.25	241	131	74	44	82.7	86	89	93	0.62
Ethyl alcohol	C_2H_6O	173.5	0.806	0.29	0.65	1.26	2.28	403	187	100	57	115	120	124	129	0.59

reduced pressure will be $\frac{14.7}{14.16} \times 197 = 202$ cu. ft. This quantity, and similar quantities for other air temperatures and other fuels, are given in Table 14. The vapor coexists in the same space with the air (202 cu. ft.) and as this volume is greater than the volume which the saturated vapor of heptane occupies (103 cu. ft.) the vapor cannot be saturated in a chemically correct mixture; the vapor will be superheated. At the temperature of 40°F. the air volume is seen to be 194 cu. ft. and that of the saturated vapor of heptane 185 cu. ft.; as these are approximately equal the vapor will be practically saturated. At temperatures lower than 40°F. the volume of the air will be less than the volume of the saturated vapor and in that case part of the fuel will necessarily be in the liquid form. An excess of air above that chemically necessary will lower the temperature at which liquid must begin to appear; an excess of fuel will raise that temperature.

With a less volatile fuel such as octane it will be seen by inspection of the table that a higher temperature (a little under 80°F.) will be necessary if the fuel is to be in the vapor form. With benzol the temperature is well below 40°; with methyl and ethyl alcohol between 70 and 80°F.

It should be noted that the temperatures of the table are the temperatures after the vapor is formed. In the carburetor, the latent heat of vaporization of the fuel is taken from the air and the liquid fuel, with the result that the temperature of the mixture falls below the temperature of the entering air and fuel, unless heat, equal to the latent heat, is supplied from the jacket water or exhaust gases. If the latent heat of the fuel is 135 B.t.u., the specific heat of the liquid 0.45 and the specific heat of air at constant pressure 0.241, the fall in temperature ΔT for a mixture of 1 lb. of fuel with 15.1 lb. of air is given by the equation

$$135 = \Delta T(0.45 + 15.1 \times 0.241)$$

or

$$\Delta T = 33°\text{F.}$$

If the fuel is just saturated at 40°F., the entering temperature of the air and fuel would have to be at least $40 + 33 = 73$°F. to permit all the fuel to be vaporized, if no heat is supplied to the mixture from outside.

The following table[1] gives data of a similar nature for various fuels. Column 3 gives the temperature of the fuel-air mixture at

[1] From KUTZBACH, *Technical Note No.* 62. *National Advisory Committee for Aeronautics*, 1921.

which the vapor of the fuel is just saturated; the mixture is supposed to be chemically correct and the pressure of the mixture is atmospheric pressure. In column 4 is given the fall in temperature of the air and liquid fuel required to supply the latent heat for complete vaporization.[1] The initial temperature of the air must be at least equal to the sum of the quantities given in columns 3 and 4; this sum is given in the last column.

SATURATION TEMPERATURES OF AIR-FUEL MIXTURES

Fuel	Boiling point, deg. F.	Saturation temperature of the fuel mixture, deg. F.	Drop in temperature due to evaporation, deg. F.	Minumum temperature of the air for complete vaporization, deg. F
Hexane	158	0	54	54
Benzene	176	23	54	77
Ethyl alcohol	172	72	198	270
Decane	320	108	63	171
Naphthalene	428	198	72	270

It is evident that the temperature of the air-fuel mixture with decane or naphthalene as fuel is so high as to reduce considerably the volumetric efficiency and the power of the engine if all the fuel enters in the vapor form.

Gaseous Explosions.—In an airplane engine making 1,800 revolutions per minute, the duration of the explosion should not be greater than the time of one-sixth of a revolution or $\frac{1}{180}$ second. The possibility of employing a gasoline engine depends on the possibility of carrying out the explosion process with a high degree of completeness in this extremely short time.

Explosion is a chemical reaction attended by the liberation of a considerable amount of heat. It is a combustion process. Combustion results from the chemical union of a fuel with oxygen and this union may take place either (1) at the place where the two are brought into contact as with the ordinary gas burner, or (2) in an intimate mixture of the two, as in a bunsen burner or in a gas engine cylinder. Explosive reaction can take place only with an intimate mixture.

The reaction in an intimate mixture is not necessarily explosive; for example, no explosion occurs in the bunsen burner. An

[1] Some of these values are calculated by Kutzbach from values of latent heat which are apparently too high.

explosion is always self-propagating: that is, if part of the mixture is ignited the combustion will spread throughout the mass of the mixture. The term "explosion" is commonly reserved for the case where the velocity of such propagation is high; but there is no definite line of demarcation between explosion and slow burning.

The velocity of propagation of combustion in an explosive mixture depends on the kind of fuel, the amount of oxygen present, the amount of inert gases present, the temperature, pressure, and a number of other factors. The strength of the explosive mixture is the most important factor. No explosion is possible if the ratio of air to fuel exceeds certain limits. Bunte[1] has found the explosive limits for various air-fuel mixtures at atmospheric pressure and temperature as given in the following table:

EXPLOSIVE LIMITS OF AIR-FUEL MIXTURES

Fuel	Ratio of air to gas by volume		Theoretical ratio of air to gas by volume
	Lower limit, air in excess	Upper limit, gas in excess	
Carbon monoxide	5.06	0.33	2.4
Hydrogen	9.58	0.50	2.4
Water gas	7.06	0.49	2.4
Acetylene	28.8	0.91	11.98
Coal gas	11.6	4.23	5.7
Ethylene	23.4	5.84	14.4
Alcohol	24.3	6.32	14.4
Marsh gas	15.4	6.81	9.63
Ether	35.7	12.0	28.41
Benzene	36.7	14.4	36.0
Pentane	40.7	19.4	37.5

Burrell and Gauger[2] give explosive limits of air-gasoline mixtures as 66 and 16 (ratio of air to gasoline vapor by volume).

The above results were obtained with mixtures at ordinary atmospheric pressures and temperatures. They show that a self-propagating combustion is possible with most fuels where

[1] *The Engineer*, March 28, 1902.

[2] *Technical Paper* 150, U. S. Bureau of Mines.

there is a considerable excess present either of air or of fuel. These limits are considerably extended as temperature and pressure increase. For example, at 600°C. it is possible to explode a mixture of CO with 12 times its volume of air, as compared with 5.06 times at atmospheric temperature. The presence of carbon dioxide in place of some of the excess air diminishes the explosive limits.

The temperature to which part, or all, of the mixture must be brought to initiate an explosion is called the **ignition temperature.** This varies with the fuel, strength of mixture, the volume or mass of the mixture heated, the temperature and dimensions of the containing vessel, and the method of ignition. A weak spark, although it has a temperature much higher than the ignition temperature, may fail to cause an explosion. It may start combustion at the place where it passes, but the heat loss by convection, conduction and radiation may be in excess of the heat of combustion and the flame will fail to propagate. A sufficient duration of spark is also necessary. A flame may ignite a mixture that cannot be exploded by a spark, because it gives, initially, so large a volume of flame that the radiation loss to the containing vessel does not cool it below the ignition temperature. If the whole mass is raised in temperature simultaneously (as by adiabatic compression) the ignition temperature will be less than when part of the mixture only is heated. This ignition temperature, with adiabatic heating of fuel-air mixtures, is from about 1,200°F. for hydrogen to about 1,700°F. for carbon monoxide. With the usual gas engine fuels it falls between those limits, the value depending on the hydrogen and the neutrals present.

The ignition temperature has great importance as it determines the permissible ratio of compression, and thereby, the limit of efficiency in the engine. Compression must stop just short of that temperature at which ignition will occur. Any means for increasing the cooling of the mixture during compression (such as improved water jacketing) will permit a greater ratio of compression. Local heating of the mixture, as by carbon deposit, may result in preignition.

Combustion once started in an explosive mixture may either die out or be propagated. If it once starts to propagate itself, it is likely to continue and there will result an explosion. The velocity with which the combustion is propagated increases progressively in all true explosions. In the case of a bunsen burner the velocity remains constant and the combustion is not explosive. The flame in that case is stationary, but as the gas is moving the flame is really moving relative to the gas, in the opposite direction and with the same velocity. If the velocity of the gas is diminished too much by partly closing the gas supply, the flame will shoot back, *i.e.*, the flame will travel more rapidly than the gas. The flame remains at the mouth of the burner under considerable variations of gas velocity in the burner because the velocity of the mixture decreases rapidly as it issues from the burner, so that there will be some place, close to the burner, at which the gas velocity equals the velocity of flame propagation. The flame will remain stationary at that place. The cooling effect exerted by the metal burner also reduces the flame propagation velocity.

If the velocity of the gas which will just keep the flame away from the burner is measured, it will give a rough indication of the velocity of flame propagation in the mixture. The results will not be very accurate because of cooling and diluting influences of the atmosphere. Experiments of that general nature show that, at atmospheric temperature and pressure, for H and O, the velocity of propagation is about 115 ft. per second, and for CO and O about 4½ ft. per second. This is for the combining proportions, which give approximately maximum velocities. With H and air the velocity drops to about 10 ft. per second at 212°F. For gasoline-air mixtures, at atmospheric temperatures, velocities of about 3.5 ft. per second and for alcohol about 3 ft. per second are realized. These results apply only to linear propagation at atmospheric pressure.

In a **closed vessel**, such as a gas engine cylinder, the conditions are quite different. The propagation, starting from a point, is spherical; the increase of temperature results in increase of pressure and as the flame spreads the unburned portion will be compressed adiabatically and will increase continually in pressure and in temperature. As the temperature increases the rate of propagation will increase. The velocity of propagation will then be continually accelerated. The flame, moreover, is carried forward bodily by the expansion of the burned portion.

Experiments on explosions in closed vessels have determined the time required to reach maximum pressure with various mixtures exploded in vessels of various shapes. If the maximum distance from the ignition point to the boundary of vessel is divided by this time, the quotient gives a measure of the average rate of flame propagation. With illuminating gas at atmospheric temperature, in a tube ½ in. in diameter and with 7½ in. travel of flame, this varies from 5 to 24 ft. per second, according to the strength of the mixture. It increases rapidly with increased initial temperature; in some cases as the tenth power of the absolute temperature (= 1,000-fold for doubled temperature).

With the largest existing gas engines (using blast-furnace gas) the available time for a good explosion is about ⅛ sec. and the maximum distance the flame must travel is about 1⅓ ft.; this gives a mean velocity of 11 ft. per second. The addition of a third igniter has sometimes increased the capacity 20 per cent and shows that the speed limit has been reached. Blast-furnace gas consists mainly of CO, which, at low temperature, has a velocity of propagation not greater than one-third that of gasoline.

With an airplane engine at 1,800 r.p.m., the time for explosion is about $\frac{1}{80}$ second; if the flame travels 2 in. the mean velocity will be 33 ft. per second. By increasing the ignition lead, still more time might be provided; the speed of the airplane engine is not yet limited by the velocity of propagation of the explosion. Alcohol is slower so that alcohol engines could not be run as fast as gasoline engines if the rate of propagation of the explosion should ultimately determine the limit of speed, instead of valve areas and inertia effects as at present.

The observed velocities of propagation in actual engines are higher than those which experiments with closed vessel indicate. This results from another factor, **turbulence.** The velocity with which the gases enter gas-engine cylinders is very much higher than the velocity of propagation of flame. With 1-lb. drop of pressure into the cylinder and no frictional resistance the velocity of the entering air would be about 350 ft. per second; with ¼ lb., 175 ft.; with $\frac{1}{10}$ lb. about 120 ft. per second. This gas velocity causes turbulent conditions which cannot be quieted down by the time explosion starts. The propagation is not spherical but is by currents and eddies of burning gas which carry flame to all parts of the vessel more rapidly than is possible with spherical propagation. Recent experimental work bears this out. Dugald Clerk found, in a common gas-engine cylinder, that after quieting down turbulence, the explosion takes nearly three times as long as when the usual conditions exist. Experiments in closed vessel without stirring gave the time of explosion as 0.13 sec.; with vigorous stirring the time required was only one-sixth as long.

Detonation.—During explosion in a closed vessel the advancing flame sphere sends off compression waves which travel through the unburned mixture with the velocity of sound in that medium. If the vessel is of sufficient dimensions the increasing velocity of the flame and the continuously increasing pressure and temperature of the unburned mixture will result in the formation of a wave in which the pressure will be such as to bring the mixture (adiabatically) to the ignition temperature. In that case the wave will cause combustion as it moves on. The velocity of this wave will be greater than that of sound because the process is not merely one of wave transmission but of chemical reaction also. Investigations of the explosive wave show velocities of the order of magnitude of 3,000 to 6,000 ft. per second and pressures of 1,000 to 2,000 lb. per square inch. These pressures are destructive to engines and should be avoided.

The "detonations" or "pinking" which are both felt and heard in engine cylinders under certain conditions of operation probably indicate either the generation of an explosive wave or breaking down of the fuel with the liberation of free hydrogen, which explodes with extreme rapidity. In such cases the combustion is notably incomplete, the exhaust containing much free carbon, and the power and efficiency of the engine fall off. Fuels consisting of paraffins have a low-ignition temperature, and are readily detonated. Fuels belonging to the aromatic group have higher ignition temperatures and can be used with higher compression pressures without detonation.

The **maximum pressures** to which fuels can be compressed **without serious detonation** have been determined by Ricardo,[1] who used for that purpose a variable compression engine with compact combustion space, central igniter, and other features making for maximum capacity and efficiency. His results, including the corresponding indicated mean effective pressures and thermal efficiencies, are given in Table 15. The data for toluene, xylene, and acetone are for a compression ratio of seven, which gives a compression pressure well below their detonation pressures; it was not considered desirable to go above that compression ratio for hydrocarbon fuels on account of the excessive

[1] *The Automobile Engineer*, Jan. and Feb., 1921.

EFFECTIVE PRESSURE AND THERMAL EFFICIENCY

Fuel	Sp. Gr. at 15°C.	Approximate composition: Paraffins, per cent by weight	Approximate composition: Aromatics, per cent by weight	Approximate composition: Naphthenes, per cent by weight	Maximum compression ratio	Maximum compression pressure, pounds per sq. in. (gage)	Indicated m.e.p., lb. per sq. in. (gage)	Relative indicated m.e.p. (Toluene = 100)	Indicated thermal efficiency, per cent	Relative thermal efficiency (Toluene = 100)
Aromatic free gasoline	0.718	63.0	1.7	35.0	4.85	105.5	130.0	88.4	31.4	83.7
"A" gasoline	0.782	26.0	39.0	35.0	6.0	148.5	140.1	95.3	34.9	93.0
"B" gasoline	0.723	62.0	14.9	23.0	5.7	133.5	137.5	93.5	34.1	91.0
"C" gasoline	0.727	61.0	8.5	30.5	5.25	118.0	133.9	91.0	32.5	86.7
"D" gasoline	0.760	38.0	14.6	47.0	5.35	121.5	134.9	91.7	33.1	88.3
"E" gasoline	0.719	68.0	11.3	20.0	4.7	100.5	128.6	87.5	30.7	81.8
"F" gasoline	0.704	80.0	4.3	15.2	5.05	111.5	132.7	90.2	32.1	85.6
"G" gasoline	0.750	.....	7.5	....	4.55	96.0	127.4	86.6	30.2	80.5
"H" gasoline	0.767	10.0	4.8	85.0	5.9	140.5	139.5	94.8	34.6	92.2
Heavy aromatics	0.885	.....	71.5	....	6.5	163.5	142.5	96.9		
Kerosene	0.813	.....	.....	....	4.2	86.0	123.0	83.6		
Paraffin Series										
Hexane (80%)	0.685	77.0	2.7	20.0	5.1	113.5	133.1	90.5	32.4	86.4
Heptane (pure)	0.691	100.0		....	3.75	72.0	119.0	81.0	26.7	71.2
Aromatic Series										
Benzene (98%)	0.884	negligible	98.0	negligible	6.9*	179.0	146.5	99.6	37.2	99.2
Toluene (99%)	0.870	negligible	99.0	negligible	>7.0	>183.0	147.0	100.0	37.5	100.0
Xylene (91%)	0.862	5.0	91.0	4.0	>7.0	>183.0	146.8	99.9	37.3	99.5
Naphthene Series										
Cyclohexane (93%)	0.786	.....	4.6	93.0	5.9*	140.5	139.2	94.6	34.9	93.0
Hexahydrotoluene (78%)	0.780	.....	10.0	78.0	5.8	136.5	137.9	93.7	34.3	91.5
Hexahydroxylene (60%)	0.744	.....	.....	60.0	4.9	107.0	130.0	88.4	31.5	84.0
Olefine Series	.....									
Cracked spirit	0.757	Olefines 53.0	10.0	Paraf. & Naphth. 37.0	5.55	128.0	136.0	92.5	33.9	90.4
Alcohol group										
Ethyl alcohol	0.798	Water 1.5	.....	....	>7.5	>204.0	156.5	106.4	40.4	107.9
Methyl alcohol (purified wood naphtha)	0.829	(Not fully determined)			5.2*	116.5	146.6	99.6	35.0 approx.	93.3
Methylated spirits	0.821	6 approx.	.....	....	6.5*	163.5	155.5	105.8	38.5 approx.	102.7
Acetone	0.798	0.5	.....	....	>7.0	>183.0				
Ether	.....	.....	.....		....	....	(116.5)	79.2	(19.5)	52.0
Miscellaneous										
Ether 50% with gasoline	0.727	2.5 approx.	.....	Aromatic free by vol. 50.0	3.9	77.0				
Ether	0.735	5.0 approx.	.....	....	(2.95)	(47.5)				
Carbon disulphide, 50% and gasoline	0.994	.....	.....	50.0	5.15*	115.0				
Carbon disulphide	1.270	.....	.....	....	(5.4)	(123.0)				

* Preignition occurred before audible detonation.
(Values obtained indirectly.)

explosion pressures reached. Ricardo concludes from his investigations that detonation is less the lower the rate of burning of the fuel, and that no fuel has a rate of burning too low to permit of maximum efficiency being obtained at the highest practicable engine speeds. All the hydrocarbon fuels give the same power output and efficiency, within 2 per cent, for the same ratio of compression so long as that compression is not high enough to produce detonation. Figure 169 gives the indicated mean effective pressure and indicated thermal efficiency practically attainable in a high-grade engine with a non-detonating hydrocarbon fuel. With alcohol the high latent heat permits the use of a greater weight of charge and consequently of increased output as compared with the hydrocarbons.[1]

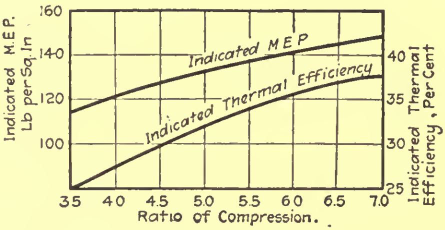


FIG. 169.—Maximum attainable m.e.p. and thermal efficiency in Otto cycle engine using hydrocarbon fuel.

The performance of a mixture of hydrocarbons is found to be the mean performance of the components. The highest permissible compression pressure is determined by the relative proportions of aromatics, naphthenes and paraffins; the smaller the proportion of the last the higher may be the compression pressure. The heavier compounds in the paraffin series detonate at lower compression pressures than the lighter compounds.

The tendency of a fuel to detonate is expressed by Ricardo in terms of its "**toluene value.**" The scale of toluene values is based on compression pressures at detonation; it is 0 per cent for a selected standardized gasoline detonating at a compression ratio of 4.85, is 100 per cent for toluene, and varies in direct proportion to the change in compression pressure. Its value is negative for fuels which detonate at lower compression pressures than the standard gasoline.

[1] The vaporization takes place mainly in the cylinder, during admission and compression, and keeps down the compression temperature.

The influence on the detonating temperature, and consequently on the permissible compression pressure, of the addition of toluene to a paraffin fuel is shown by the following tests.[1]

Toluene, per cent....	0.0	10.0	20.0	30.0	40.0	50.0	60.0
Compression ratio...	4.85	5.20	5.57	5.94	6.32	6.67	7.05
Indicated m.e.p.....	132.5	135.4	138.7	142.0	144.9	147.5	150.0

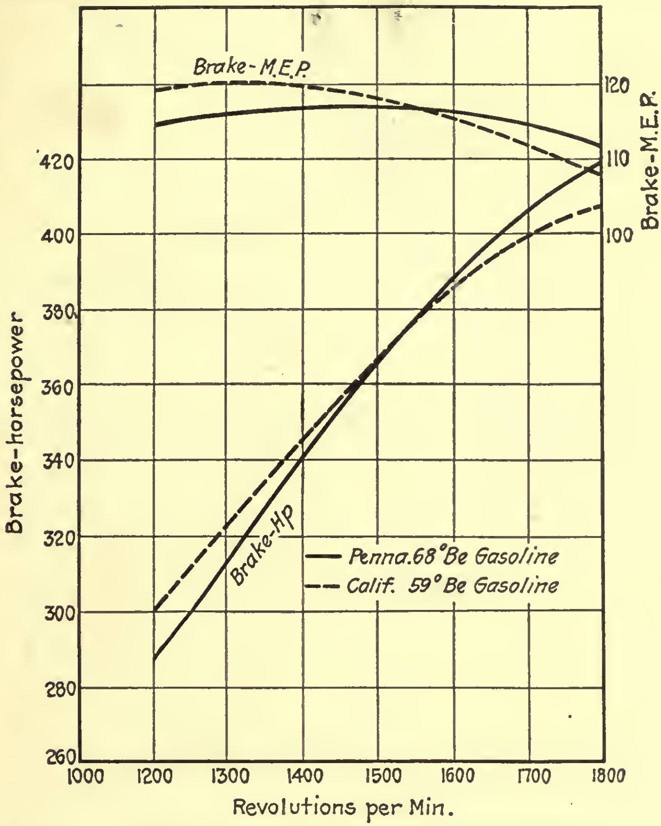


Fig. 170.—Variation of power of 12-cylinder Liberty engine with fuel.

The effect of the addition of ethyl alcohol is even more marked than that of toluene; only three-fifths as much alcohol need be added to obtain the same compression ratios. The effect of these additions on engine capacity are indicated by the mean effective pressure values of the table; the increase in engine efficiency can be seen from Fig. 169.

The preponderating importance of the detonating character-

[1] Ricardo: *Proc. Royal Aeronautical Society,* 1920.

istics of a fuel has received more recognition in England than in this country. Fuels are blended with toluene, benzol or other aromatics or naphthenes so as to give a standard toluene value. This amounts to giving a standard detonating compression pressure when the fuel is used in a standard engine. The actual detonating pressure and therefore the permissible compression ratio is largely determined by the characteristics of the engine in which the fuel is used. With a poorly shaped combustion space, non-central ignition, and other unfavorable features, a fuel will detonate at a much lower compression ratio than when the conditions are favorable. In such an engine, detonation may be prevented by the use of overrich mixtures and late ignition, with a resulting sacrifice of both power and economy.

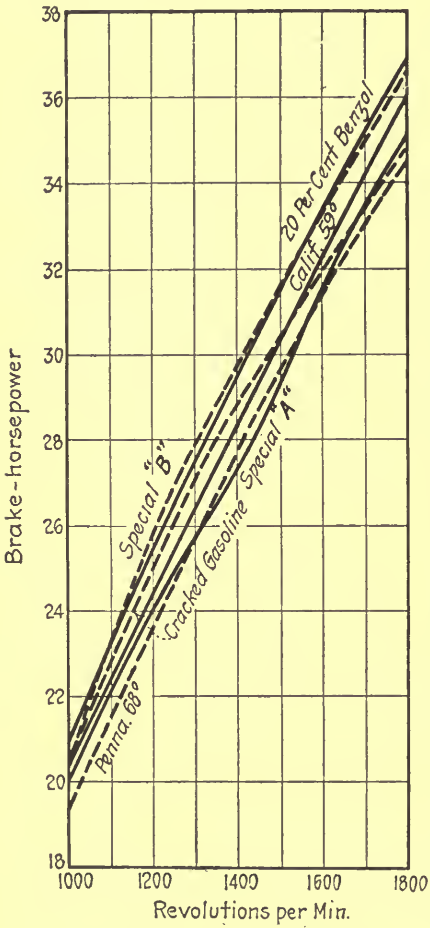


Fig. 171.—Variation of power of single-cylinder Liberty engine with fuel.

Influence of Fuel on Capacity.—A comparison of the power output of a Liberty 12 with two grades of gasoline is shown in Fig. 170; the low-grade (59°Bé.) gasoline falls off rapidly in brake mean effective pressure above 1,500 r.p.m., while the 68°Bé. gasoline maintains its mean effective pressure well to 1,800 r.p.m.

A comparison of six fuels is shown in Fig. 171. These fuels were used in a single-cylinder Liberty engine. Their distillation curves are given in Fig. 172; these represent about the full range of commercial airplane fuels. It will be seen from Fig. 171 that the total range of power is 2.8 h.p. at 1,800 r.p.m. with a maximum value of 37 h.p. at that speed; this power range is only 7.6 per cent.

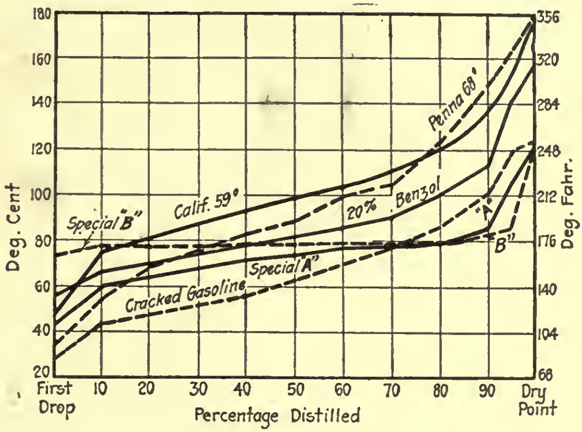


FIG. 172.—Distillation curves of the fuels of Fig. 171.

The tests of the Bureau of Mines[1] show the comparatively small range in efficiencies resulting from the use of different fuels. The fuels include straight refinery, blended casing-head,

CALORIFIC VALUE, POWER DEVELOPED IN ENGINE TESTS, AND SPECIFIC GRAVITY OF VARIOUS TYPICAL GASOLINES FROM MID-CONTINENT AND EASTERN FIELDS

Field from which sample was obtained	Process of manufacture	Gravity		High calorific value of gasoline		Power developed, horse-power hours per pound of gasoline
		Specific gravity	Bé.	Calories per gram	B.t.u. per pound	
Mid-Continent.....	Cracking plant	0.745	57.9	11,165	20,097	1.345
Mid-Continent.....	"Straight" refinery	0.742	58.7	11,174	20,113	1.403
Mid-Continent.....	"Straight" refinery	0.733	61.0	11,180	20,124	1.350
Eastern...........	"Straight" refinery	0.718	65.0	11,187	20,137	1.405
Mid-Continent.....	"Straight" refinery	0.724	63.4	11,215	20,187	1.395
Mid-Continent.....	"Straight" refinery	0.727	62.6	11,221	20,198	1.396
Eastern...........	Blended casing-head	0.733	61.0	11,230	20,214	1.376
Eastern...........	"Straight" refinery	0.724	63.4	11,236	20,225	1.420
Mid-Continent.....	"Straight" refinery	0.715	65.8	11,250	20,250	1.365
Eastern...........	"Straight" refinery	0.687	73.8	11,315	20,367	1.487

and cracked gasoline with densities varying from 57.9 to 73.8°Bé. The accompanying table shows that the work done per pound of

[1] *Technical Paper* 163, U. S. Bureau of Mines.

gasoline varied from 1.345 to 1.487 h.p.h., the higher value being generally obtained with the lighter gasolines. The heats of combustion also increased slightly as the gasoline became lighter, but the total range of higher heat values is only slightly greater than 1 per cent. Neglecting this, and assuming a low heat value of 18,500 B.t.u. for all the fuels, the range of efficiencies is seen to be from $\frac{1.345 \times 33{,}000 \times 60}{18{,}500 \times 778} = 18.45$ per cent to 20.48 per cent. The engine on which the tests were made is of low compression and low efficiency; the indications are that the change in efficiency with the fuel is even less in engines of higher efficiency such as are used in airplane practice.

A **mixture of alcohol and gasoline** has been used in high-compression aviation engines. This mixture eliminates detonation and has good starting characteristics. An "alcogas" tested at the Bureau of Standards[1] contained 40 per cent alcohol, 35 per cent gasoline, 17 per cent benzol and 8 per cent of other ingredients. With a compression ratio of 5.6, the maximum power at ground level was the same as for a high-grade aviation gasoline, but as the level increased the power output became 4 to 9 per cent greater than for gasoline. The thermal efficiency was perisuor by about 15 per cent; the fuel consumption was increased con acount of the lower heat value per pound of fuel. At a compression ratio of 7.2 the power output was increased by about 16 per cent as compared with gasoline at 5.6 compression ratio and the thermal efficiency increased about 22 per cent, which just offsets the lower heat of combustion of the fuel and gives the same weight of fuel per brake horse-power hour for both fuels. The distillation curve of the alcogas is very flat, 80 per cent distilling off between 140 and 175°F.; the end point is high, 360°F.

EXPERIMENTAL DETERMINATION OF STRENGTH OF MIXTURE

The accurate determination of the ratio of air to fuel used by an engine requires the separate measurements of the weights of fuel and air. The weight of fuel is readily ascertained, but the weight of air offers difficulties. The simplest method is to connect the air intake of the carburetor, in an airtight manner, with a large box to which air is admitted through a standard or calibrated sharp-edged orifice. If there is more than one air intake it is better, if practicable, to enclose the whole carburetor

[1] *National Advisory Committee for Aeronautics*, Report No. 89.

in an airtight chamber, connected with the orifice box. The air should enter the orifice quietly, passing through a honeycomb screen to eliminate the effect of wind or air currents.

The weight flowing through such an orifice is given by the equation

$$M = 1.1 F \sqrt{\frac{p}{T}(p - p_o)} \times C \quad .$$

where, M is the weight of air flowing per second in pounds.

F is the area of the orifice in square inches.

p is the external atmospheric pressure, pounds per square inch absolute.

T is the atmospheric temperature, degrees absolute, Fahrenheit.

p_o is the pressure inside the orifice box, pounds per square inch absolute.

C is a constant

The value of C has been determined by Durley[1] with great accuracy for circular sharp-edged orifices in steel plates, 0.057 in. thick. The following table gives his values for the coefficient C:

Coefficients of Discharge for Sharp-edged Orifice

Diameter of orifice, inches	Pressure difference on two sides of orifice, inches of water				
	1	2	3	4	5
5/16	0.603	0.606	0.610	0.613	0.616
1/2	0.602	0.605	0.608	0.610	0.613
1.0	0.607	0.603	0.605	0.606	0.607
2.0	0.600	0.600	0.600	0.600	0.600
3.0	0.599	0.598	0.597	0.596	0.596
4.0	0.598	0.597	0.595	0.594	0.593
4.5	0.598	0.596	0.594	0.593	0.592

It will be observed that the coefficient is constant for a 2-in. orifice.

In most cases it will not be found practicable to measure the air in this manner. A good approximation can be obtained from a measurement of the pressure drop from the mouth to the

[1] *Trans. Am. Soc. Mech. Eng.*, 1906.

throat of the choke or venturi tube of the carburetor. This drop can be obtained by connecting a water manometer with a very small hole ($\frac{1}{32}$ in.) drilled into the smallest section of the choke. Tests carried out on a considerable number of carburetors show that the coefficient of discharge, C, varies only slightly with the form and dimensions of the venturi tube. The weight of air flowing through a carburetor of F sq. in. free area at the throat is given by the equation

$$M = \frac{122.58\,p\,F}{\sqrt{T}}\sqrt{\left(\frac{p_o}{p}\right)^{1.422} - \left(\frac{p_o}{p}\right)^{1.711}} \times C$$

where M, p and T have the same meanings as for an orifice and p_o is the pressure at the throat. The coefficient C varies from 0.82 to 0.85 and may be assumed to have the mean value 0.84.

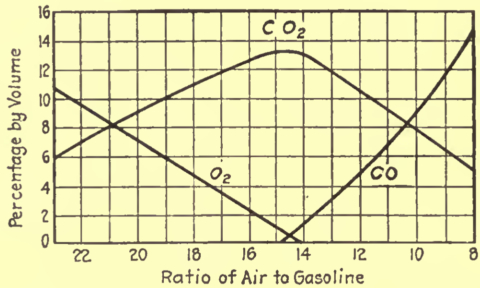


Fig. 173.—Composition of the exhaust gases from a gasoline engine.

Still another method is available if actual air and fuel measurements are impracticable. The investigations of Watson, on automobile engines, have shown that the composition of the exhaust gases varies in a regular manner with the strength of the mixture of air and gasoline admitted to the cylinder. His results are shown graphically in Fig. 173. With a chemically perfect mixture of about 14.5 parts of air to one of gasoline the exhaust gases contain about 13 per cent of CO_2 by volume and about 0.5 per cent each of O_2 and CO. If the air is present in excess (weaker mixture) there is more free O_2 and less CO_2 in the exhaust; if the mixture is richer, the free O_2 disappears and the amount of CO increases while the CO_2 decreases. All that is necessary for the test is an Orsat or other volumetric gas-analysis apparatus and the determination of the CO_2 and O_2 content, or in case no O_2 is present, the CO_2 and CO content.

CHAPTER X

THE CARBURETOR

An ideal explosive mixture arriving at the intake manifold of an engine should have the following characteristics: (1) it should be homogeneous throughout, (2) it should be of the composition or strength to develop maximum economy under each condition of engine operation, and (3) it should permit of the development of the maximum possible power.

In a stationary constant-speed engine, in which engine torque alone is variable, these results might be approximated by the use of an injection valve, under the control of the governor, spraying finely atomized fuel into the current of air going to the cylinders. In an automobile engine, with both engine torque and speed variable, this simple injection method cannot give satisfactory results. In the airplane engine with the three main variables of torque, speed and air density, the problem is even more complicated. For such engines the explosive mixture is formed by the use of a carburetor.

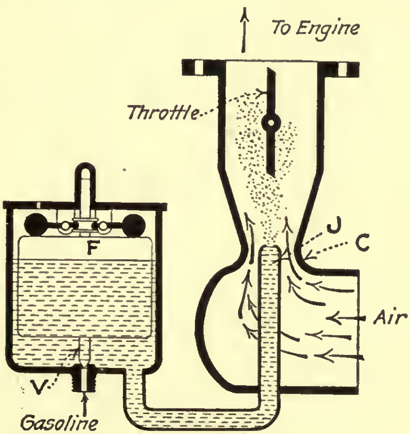


FIG. 174.—Diagram of simple carburetor

A **carburetor** is a device in which part or all of the air going to the engine passes through a restricted passage, thereby acquiring velocity with consequent fall of pressure; the fuel is sucked into the current of air in an amount which varies with the pressure drop. In the simplified standard form of carburetor shown in Fig. 174, air flows through the restricted "choke," C, and creates a partial vacuum. Gasoline is maintained at a constant level in the float chamber by the action of the float, F, which controls the position of the needle valve, V, past which the gasoline enters. As the float chamber is open to the atmosphere the level of gasoline in the nozzle or jet, J, will be the same as that in the float chamber so long as the engine is not operating. The discharge orifice of the nozzle is placed higher than the gasoline level in the float chamber to prevent overflow of the gasoline into the air passage when the engine is standing in such position as to incline the carburetor at a moderate angle to the position shown in the figure. When air is drawn through the carburetor, increasing reduction of pressure at C, resulting from increasing velocity of the air, will give an increasing head on the gasoline and will cause an increasing weight flow of the fuel. The mixture of air and fuel will be of constant strength if the weight of gasoline discharged by the jet is directly proportional to the weight of air flowing through the choke. The actual strength of the mixture whether constant or not is controlled by the size of the gasoline jet.

A carburetor built as in Fig. 174 would not discharge a mixture of constant strength for all rates of air flow, nor is such constancy desirable. It is common experience that the mixture delivered to the engine should be richer at very light loads (idling) than for heavier loads, and also that it should be richer for maximum power than for maximum economy. A satisfactory carburetor should vary the strength of the mixture so as to maintain the desired strength under all conditions of operation of the engine.

A study of the action of a carburetor requires a knowledge of the laws of flow of gases and liquids through such passages as are found in carburetors. The more important results of experiment on such flow are given in the following pages.

Theoretical Flow of Air through a Constricted Tube.—When air is flowing steadily through a tube whose cross-section varies, the weight and the total energy passing each section of the tube per second are constant.

Let W = Weight of air passing in pounds per second.
p = The air pressure in pounds per square foot absolute.
P = The air pressure in pounds per square inch absolute.
v = The specific volume of the air in cubic feet per pound.
V = The velocity of the air in feet per second.
T = The absolute temperature of the air in degrees Fahrenheit.
I = The internal energy of the air per pound in foot-pounds.
A = The cross-section of the tube in square feet.
a = The cross-section of the tube in square inches.
g = Gravitational acceleration = 32.16 feet per second per second.

The total energy passing any cross-section with unit mass of air is the sum of the internal energy I, the displacement work pv, and the kinetic energy $V^2/2g$ at that section. Assuming no heat transfer through the tube the total energy at 1 (Fig. 175) can be written equal to that at 2.

$$I_1 + p_1 v_1 + \frac{V_1^2}{2g} = I_2 + p_2 v_2 + \frac{V_2^2}{2g} \tag{1}$$

FIG. 175.—Venturi tube of optimum proportions.

Air is practically a perfect gas. If the expansion is without eddies or friction and without transfer of heat (adiabatic)

$$I_1 - I_2 = \frac{p_1 v_1 - p_2 v_2}{n - 1} \tag{2}$$

$$\text{Where } n = \frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}} = 1.403$$

Furthermore, with adiabatic expansion

$$p_1 v_1^n = p_2 v_2^n \tag{3}$$

Substituting from equations (2) and (3) in equation (1) there may be obtained the equation

$$\frac{V_2^2}{2g} - \frac{V_1^2}{2g} = \frac{n}{n-1}\, p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right] \tag{4}$$

The weight flow past any section is equal to the volume passing that section per second divided by the specific volume, or,

$$W = \frac{VA}{v}$$

Since the weight flow is constant at all sections

$$W = \frac{V_1 A_1}{v_1} = \frac{V_2 A_2}{v_2} \tag{5}$$

and substituting from equation (3)

$$\frac{V_1 A_1}{v_1} = V_2 A_2 \frac{1}{v_1} \cdot \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$V_1 = V_2 \frac{A_2}{A_1} \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \tag{6}$$

Substituting this value of V_1 in equations (4) and (5)

$$V_2 = \sqrt{2g\frac{n}{n-1}\cdot p_1v_1\left[\frac{1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}}}\right]} \tag{7}$$

and

$$W = A_2\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\sqrt{2g\frac{n}{n-1}\,\frac{p_1}{v_1}\left[\frac{1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}}}\right]} \tag{8}$$

If the section A_1 is taken just outside the tube where the cross-section may be regarded as infinite and the air velocity zero, these last equations become

$$V_2 = \sqrt{2g\frac{n}{n-1}\cdot p_1v_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]} \tag{9}$$

and

$$W = A_2\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\sqrt{2g\frac{n}{n-1}\,\frac{p_1}{v_1}\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= A_2\sqrt{2g\,\frac{n}{n-1}\cdot\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]} \tag{10}$$

In the use of equation (10) the specific volume, v_1, has to be determined from the known pressure, p_1, and absolute temperature, T_1, by the gas equation, $p_1v_1 = RT_1 = 53.34T_1$. Furthermore, it is practically most convenient to deal with pressures in pounds per square inch, P, and with areas in square inches, a. Substituting the numerical values of g and n, substituting P and a for p and A, and substituting $53.34T_1/p_1$ for v_1, equation (10) becomes

$$W = \frac{2.043P_1a_2}{\sqrt{T_1}}\sqrt{\left(\frac{P_2}{P_1}\right)^{1.422}-\left(\frac{P_2}{P_1}\right)^{1.711}} \tag{11}$$

This equation is difficult to use when the desired weight flow, W, is known and the pressure drop is required. The curves of Fig. 176 are based on this equation. The ordinates are weight flows per square inch of area per *minute*, and the abscissae are pressure drops measured in *inches of water*. One pound per square inch equals 27.70 inches of water. An initial temperature of 60°F.

(520° absolute) is assumed. The separate curves are for the initial pressures, P_1, marked on them.

Equations (8) and (10) have a limit to their range of application. If air, initially of pressure p_1 and specific volume v_1, flows through a tube the *smallest section* of which is A_2, the weight flow varies with the pressure, p_2, at that section. The weight

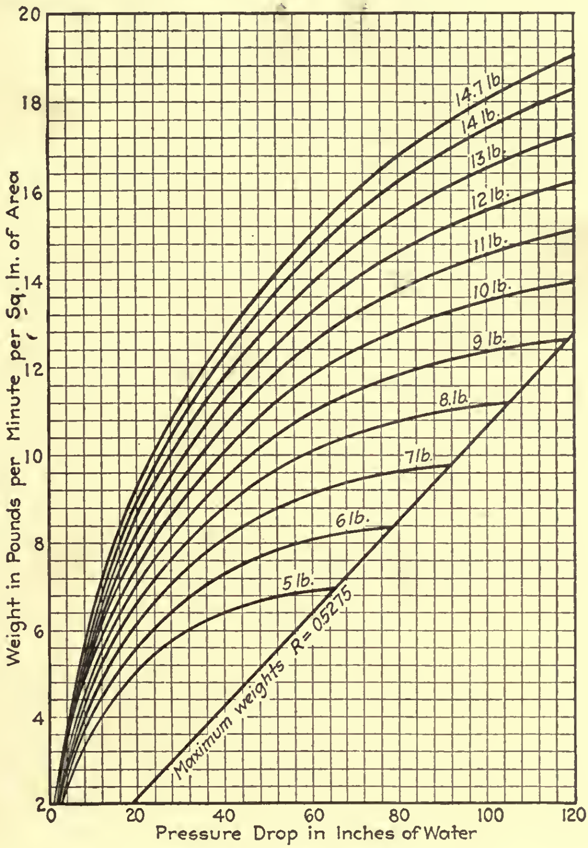


FIG. 176.—Weight flow of air.

flow will be found from the equation to reach a maximum value as p_2 diminishes to a certain critical value, and then will apparently diminish as p_2 is still further reduced. This **critical pressure** occurs when

$$\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = 0.53 \text{ (for air)}$$

That is, the maximum weight flow will occur when the pressure at the smallest cross-section of the tube is 53 per cent of the initial or maximum pressure. It is found by experiment that the pressure at the smallest cross-section is never less than this amount, and that it remains at that exact value so long as the pressure on the downstream side is equal to or less than that pressure. The weight flow through a frictionless tube is determined by the area of the smallest cross-section and cannot be increased by decreasing the pressure on the downstream side of that section below the critical pressure. In equations (8) and (10) p_2 can never have a value lower than $0.53p_1$.

For *very small pressure ranges*, for example when $(p_1 - p_2)$ is equal to or less than 1 per cent of p_1, the expansion of the air resulting from the pressure drop is so small as to be negligible and the flow may be assumed to follow the simpler laws of flow of incompressible fluids. In this case $I_1 = I_2$, and $v_1 = v_2$, and equation (1) becomes

$$\frac{V_2^2}{2g} - \frac{V_1^2}{2g} = p_1v_1 - p_2v_1$$

If the initial velocity is zero

$$\frac{V_2^2}{2g} = (p_1 - p_2)v_1 \tag{12}$$

or V is proportional to $\sqrt{p_1 - p_2}$, and since the weight flow is proportional to V (with constant cross-section and constant air density),

$$W = K\sqrt{p_1 - p_2} \tag{13}$$

With air at atmospheric pressure, the error resulting from the use of this equation would be about -2.3 per cent for 1 lb. per square inch pressure drop, and is roughly proportional to the pressure drop for small pressure drops. Equation (12) shows that the velocity and therefore *the weight of air flowing is proportional to the square root of the pressure drop* so long as the pressure drop is small.

The **actual flow of air through a constricted tube** is found to be less than the amount indicated by equation (10). Actual flow is always accompanied by frictional resistance and the formation of eddies. The ratio of the actual flow to the theoretical flow of equation (10) is called the **coefficient of discharge** of the tube and its value has to be determined by experiment.

The **choke** of a carburetor is usually of the general form shown in Fig. 175, that is, it consists of three parts: (1) a converging entrance; (2) a throat; and (3) a diverging discharge; such a tube is generally described as a **venturi tube.** With equal areas at the entrance and exit, the pressure drop from the entrance to the throat would be entirely regained at the exit, if the air flow were frictionless and eddyless. In actual carburetors, the pressures at discharge will be less than that at entrance, and the difference will depend on the velocity of the air, the "streamlining" of the passage, the degree of obstruction offered by the gasoline jet, and the weight of gasoline carried by the air. The total pressure drop in the venturi is important in determining the volumetric efficiency and capacity of the engine; to develop maximum power the charge should enter the cylinder with the maximum possible density. Loss of pressure in the carburetor is a direct source of loss of power in the engine.

The results of published tests on comparatively large venturi tubes, with straight axes, and without obstruction at throat or entrance, show discharge coefficients varying from 0.94 to 0.99 for cases where $A_1 = A_3$, and A_2 is equal to or less than 0.5 A_1. In these tubes it is found that, for minimum friction and eddy loss, the included angle for the converging entrance should not exceed 30 deg., and the diverging discharge tube should have an included angle between 5 deg. and 7.5 deg.; these should be joined to a short cylindrical throat by well rounded junctions. Figure 175 shows a venturi tube of these optimum proportions.

Such optimum proportions are generally not practicable for airplanes. Considerations of space available make it necessary to modify the entrance by curving its axis, and force the adoption of larger included angles. Furthermore, the air passage is obstructed by the gasoline jet and its supporting bosses, and, in many cases, by the throttle valve. All these factors will cause a diminution in the discharge coefficient and an increase in the pressure loss. An investigation at the Bureau of Standards[1] gives data on certain carburetors which were designed for the Liberty engine. The air passages of these carburetors are shown in Fig. 177. The tests were made with various air densities (corresponding to different altitudes), and both with and without fuel admission. Figure 178 shows the coefficient of discharge; Fig. 179 the ratio of the exit to the entrance pressure,

[1] P. S. Tice: *National Advisory Committee for Aeronautics, 4th annual report*, pp. 608–615.

for both carburetors, with air of 750 mm. pressure and with various weights of air flowing. Figure 180 shows the pressure recovery ratio for the Zenith carburetor, with various air densities, and both with and without fuel admission to the air. The conclusions derived from these tests are as follows:

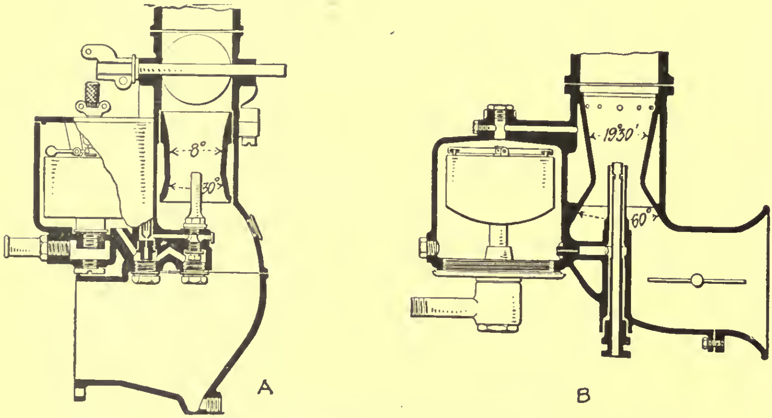


Fig. 177.—Zenith (*A*) and Stewart-Warner (*B*) carburetors.

1. The coefficient of discharge for the carburetor passages tested has an almost constant and maximum value for effective throat velocities greater than about 150 ft. per second.

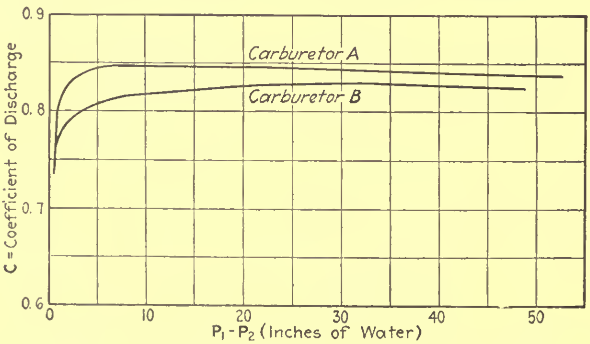


Fig. 178.—Venturi discharge coefficients for Zenith (*A*) and Stewart-Warner (*B*) carburetors.

2. The value of the coefficient of discharge for the carburetor passages tested lies between 0.82 and 0.85, under service conditions. These values are probably typical of reasonably well formed passages of similar type.

3. The coefficient of discharge for carburetor passages of this type is apparently only slightly modified as a result of consider-

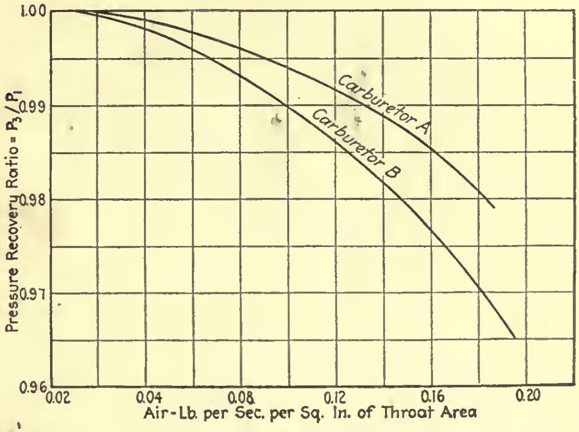


FIG. 179.—Pressure drops at partial loads in Zenith (*A*) and Stewart-Warner (*B*) carburetors.

able changes in passage form, with respect to angles of entrance and exit.

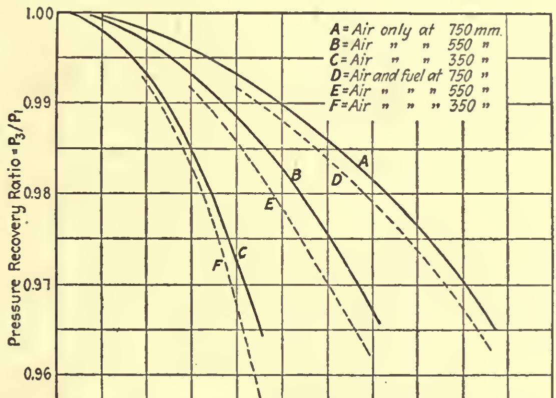


FIG. 180.—Pressure drop through a Zenith carburetor as affected by air density and the injection of fuel.

4. The coefficient of discharge for a carburetor passage is practically unaffected by wide variations in atmospheric density

(less than 1 per cent maximum variation between the density limits of 0.075 and 0.035 lb. per cubic foot).

5. The coefficient of discharge for a carburetor passage is practically unaffected by the introduction of fuel to the air stream (fuel discharge introduces irregularities not to exceed plus or minus 1 per cent).

6. The pressure loss in the carburetor outlet changes with the turbulence or internal motion of the air stream.

7. The pressure loss in the carburetor outlet changes with the quantity of fuel admitted to the air stream, and with the method of dividing the fuel by spraying.

Pulsating Flow.—The previous discussion relates to steady flow of air through the choke. In actual operation the flow is pulsating; each carburetor usually supplies three or four cylinders. With a maximum of four cylinders the carburetor will be supplying one cylinder only at any instant. The flow of the air through the carburetor is determined by the velocity of the piston in the cylinder to which the air is going. As this velocity is zero at the ends of the stroke and a maximum at midstroke, the variation in velocity of flow through the carburetor would be considerable were it not for the steadying effect of the intake manifold. The volume interposed between the carburetor and cylinder acts as an equalizing device and cuts down the pressure pulsations at the exit of the carburetor. Tests made in England and at the Bureau of Standards[1] show that for a given weight of air flowing under pulsating discharge the coefficient of discharge of the carburetor (as determined from pressure measurements at the throat), the pressure recovery ratio, and the strength of the mixture are practically the same as for steady flow.

The Flow of Fuel through a Nozzle or Jet.—The flow of a liquid through an orifice is given by the expression $V = C\sqrt{2gh}$, where V is the velocity of flow, C a coefficient, and h the head under which the flow is occurring. This expression becomes

$$W = 60.2\, C\, a\sqrt{sh} \tag{14}$$

where W = Weight of liquid discharged in pounds per minute.
a = Area of passage in square inches.
s = Specific gravity of the liquid (referred to water at 60°F.)
h = Head or pressure drop across the jet expressed in inches of water.

[1] *National Advisory Committee for Aeronautics, 4th annual report,* p. 616.

The coefficient C includes losses due to skin friction, fluid friction, contraction, and end effects. Its value varies with the head, h, with change in shape of the entrance to the jet, with change in ratio of length, L, to diameter, D, of the passage, and with the viscosity of the fuel.

Investigations by Tice[1] on the flow through jets show the influence of these different factors on the value of C. The effect of the alteration of the shape of the jet entrance from square to chamfered is shown in Fig. 181. The diameter and length are the same for both jets. The major effect of the chamfering is to reduce the contraction of the stream in the entrance, in this case, at heads above 2 in. in water. While the coefficient,

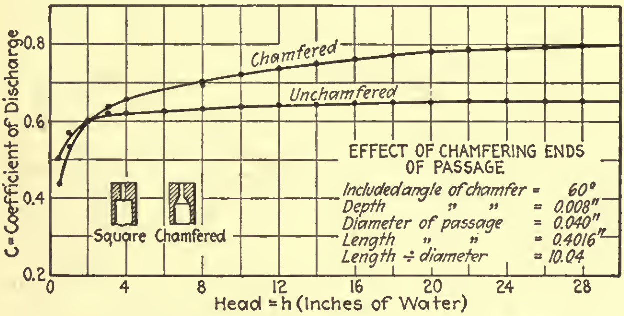


FIG. 181.—Discharge coefficients of square and chamfered jets.

C, has considerably higher values with increase of h with the entrance chamfered in this way, it will be noted also that its value varies through wider limits. Chamfering has the very practical advantage in carburetor manufacture, that the angle and depth of the chamfer, within comparatively wide limits, have an almost negligible effect on the discharge; while, on the other hand, small departures from truth in the making of sharp square edges result in wide variations in the discharge. This, together with the great difficulty of producing duplicate parts having square edges free from burr, practically rules out the square edge for carburetor metering passages.

Within the range of metering passage diameters used in general carburetor practice, it is found that the value of C increases with increase of D (Fig. 182).

The effect upon C of change in the ratio $L:D$ is brought out in Figs. 183 and 184. In the former, C is plotted against h for

[1] *Loc. cit.*, p. 603.

several values of $L:D$ with D a constant. In Fig. 184, C is plotted against $L:D$, each curve being representative of a constant value for h.

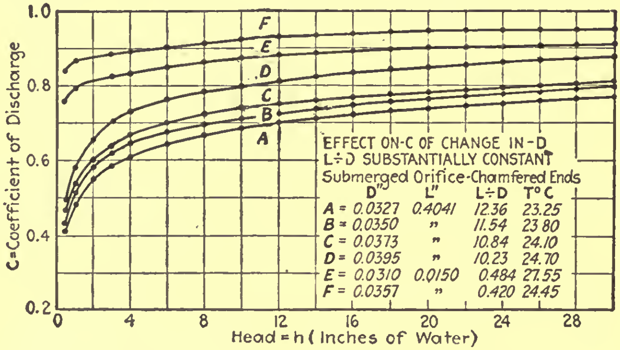


FIG. 182.—Influence of diameter on the discharge coefficients of jets.

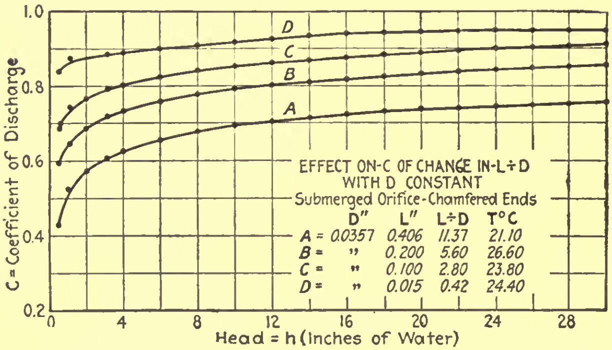


FIG. 183.—Influence of the ratio of length to diameter on the discharge coefficients of jets.

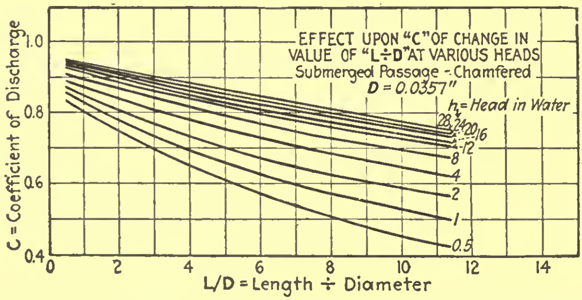


FIG. 184.—Influence of the liquid head on the discharge coefficients of jets.

A change in temperature, T, affects the discharge from a passage in two ways—through its influence on the density, s, and through the change in fluidity. For ordinary variations in T, the change in s is comparatively small and has very slight influence on the discharge. The curves A, B and C, in Fig. 185, for gasoline discharged from a jet at three temperatures, expresses the order

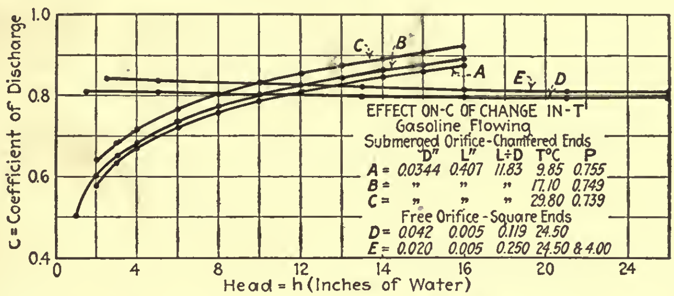


Fig. 185.—Influence of temperature on the discharge coefficients of jets.

of magnitude of the effect upon C of change in fluidity resulting from change in T. These results are for a comparatively long passage, in which this effect is much greater than with the smaller values for $L:D$ found in carburetor practice. The curves D and

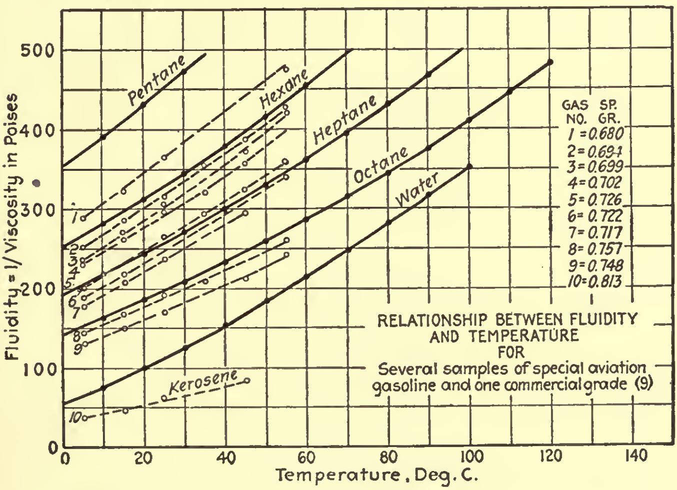


Fig. 186.—Variation of the fluidity of liquid fuels with temperature.

E are for sharp-edged orifices; they show great constancy of C with variation both of h and of T: a change in T from 24.5°C. to 4°C. shows no appreciable change in C at any value of h.

The fluidity of a liquid is the reciprocal of its viscosity. The variation of the fluidity of aviation engine fuels with temperature has been investigated by Herschel,[1] who finds the results shown in Fig. 186. The value of C for a jet will increase as the temperature and, therefore, the fluidity of the fuel increases. There is no fixed relation between the densities and fluidities of different fuels; a change of fuel will ordinarily result in a change in C.

Mixture Characteristics of a Carburetor with Constant Air Density.—It has been shown by equation (13) that, for moderate pressure drops in the choke, the theoretical air flow, W, is sensibly proportional to the square root of the pressure drop, and with a constant coefficient of discharge this means that the actual air flow follows the same law. It has been further shown that with discharge through a sharp-edged orifice the flow of the fuel follows the same law. Consequently, it would seem possible to construct a carburetor in which the air-fuel ratio would remain constant for moderate air flows. Actual carburetor constructions do not, however, employ sharp-edged orifices on account of the production difficulties already mentioned. Furthermore, the air flow does not increase as rapidly as the square root of the pressure drop—for example, in Fig. 176, with air initially at 14.7 lb. per sq. in. pressure, as the pressure drop increases from 10 in. to 40 in. of water, the weight flow instead of doubling increases only from 6.56 to 12.6 or 1.92 times. At the same time, using the standard form of chamfered jet, as shown in Fig. 181, the coefficient of discharge increases and thereby increases the flow of fluid more than two fold. The mixture will therefore increase in richness as the load increases. To offset this increase, the structure of Fig. 174 is modified in all commercial carburetors. These modifications are extremely diverse in character and can be such as to produce a constant mixture

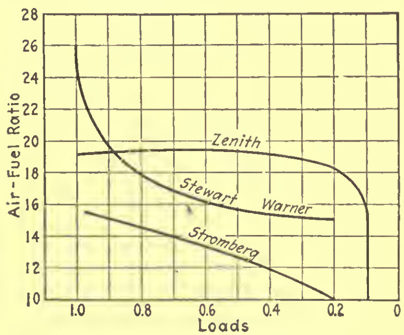


Fig. 187.—Variation of mixture strength with load in Zenith, Stewart-Warner and Stromberg carburetors.

[1] *Bureau of Standards Technologic Paper No.* 125.

or almost any desired variation of mixture with load. Some of these constructions will be considered later.

The results of tests on three special airplane carburetors at standard air density, shown in Fig. 187, are characteristic of the methods of variation of the air-fuel ratio with the load in actual carburetors. The Zenith carburetor shows a very constant mixture; the other two show enrichment of the mixture with diminishing load, a characteristic exactly opposite to that of the simple carburetor of Fig. 174. The actual value of the air-fuel ratio depends on the size of the fuel orifice and is not characteristic of the type of construction.

Mixture Characteristics of Carburetor with Variable Air Density.—When the air density changes, as a result of change of air pressure and temperature during the ascent of an airplane, a new disturbing element is introduced into the behavior of the carburetor. With level flight and wide-open throttle, the engine speed may be assumed, as a first approximation, to be constant at all altitudes; this is not the case since the engine speed may fall off as much as 10 or 12 per cent. The volume of air passing through the carburetor is equal (approximately) to the piston displacement of the engine per unit of time and may also be assumed to be constant. The weight of air, W, taken in will then be proportional to the air density, D. At any altitude x,

$$\frac{W_x}{W_o} = \frac{D_x}{D_o}$$

where o indicates ground condition. With a sharp-edged fuel nozzle of constant coefficient of discharge, the weight of fuel discharged, w, is proportional to the square root of the pressure drop at the carburetor throat (equation 14) and this is, approximately, proportional to the air density, D (equation 12). This may be written:

$$\frac{w_x}{w_o} = \sqrt{\frac{D_x}{D_o}}$$

If R is the air-fuel ratio $\frac{W}{w}$, then

$$\frac{R_o}{R_x} = \frac{W_o}{W_x} \cdot \frac{w_x}{w_o} = \sqrt{\frac{D_o}{D_x}}$$

that is, the strength of the mixture varies inversely as the square root of the air density. As the air density is proportional to its pressure and inversely as the absolute temperature, T, this becomes

$$\frac{R_o}{R_x} = \sqrt{\frac{P_o}{P_x} \cdot \frac{T_x}{T_o}}$$

On going from the ground to an altitude of 30,000 ft., where the air density is 40 per cent of ground density, the air-fuel ratio would fall from 20 to 14, or the strength of the mixture would be enriched $\left(\frac{R_o - R_x}{R_x}\right) \times 100 = 43$ per cent.

The strength of mixture desired can only be determined by engine tests. Such tests have been carried out on Hispano-Suiza and Liberty engines at the Bureau of Standards.[1] The best

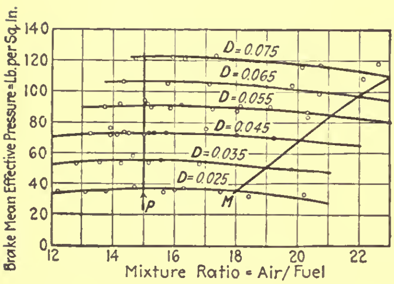


Fig. 188.—Influence of air-fuel ratio on brake m.e.p. at various air densities.

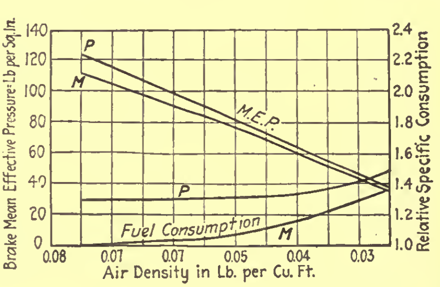


Fig. 189.—Influence of air density on maximum power and maximum thermal efficiency.

mixture to use depends on whether maximum power or maximum economy is wanted. The curves of Fig. 188 show how the brake m.e.p. varies with the mixture ratio at air densities, D, from 0.075 to 0.025. As these results are for constant engine speed, they also show the method of variation of the horse power developed. It will be seen that maximum power, P, is obtained with an air-fuel ratio of 15 at all air densities. Maximum economy (minimum fuel consumption per brake horse-power hour) is obtained at the points crossed by the curve M; it is seen that at ground level ($D = 0.075$) the most economical air-fuel ratio is 23, and that this value diminishes (richness increases) as the air density decreases. The maximum economy curve runs very near to the limit of explosibility, this limit requiring an increasingly rich mixture as the compression pressure

[1] P. S. Tice: *Nat. Adv. Comm. Aeronautics, 4th Annual Report*, p. 624.

diminishes. The air-fuel ratio for maximum economy is given approximately by

$$R = 106D + 15$$

where D is the air density. In Fig. 189 the same data are replotted to show the variation of brake mean effective pressure, and of fuel consumption per brake horse-power hour, with the air density, both at maximum power, P, and maximum economy, M.

It is not possible for a carburetor operating with wide-open throttle to give both maximum power and maximum economy without some kind of manual control since the demands for these two conditions differ only in the amount of fuel supplied. The condition of maximum economy alone is important, except for war purposes. For a flight of several hours' duration the combined weight of an engine and its fuel consumption will be less for a larger engine operating at maximum economy than for a smaller engine operating at maximum power and developing the same total power. The carburetor should be devised to give maximum economy at full throttle, with a manual control to increase the fuel supply so as to give maximum power if desired.

Economy of operation at low loads is unimportant in heavier than air machines since this condition of operation is not possible for other than very short periods. In lighter than air machines the economy at low loads may be of more importance. At full throttle the most economical air-fuel ratio varies from 23 at the ground to 19 at half-ground density; for operation at partial loads these figures must be reduced. It is not desirable to operate an engine with the mixture giving maximum economy because this mixture is so close to the limit of explosibility that slight changes in condition might result in exceeding that limit. Since the economy changes but slowly with change of mixture in the neighborhood of the optimum value, it is the practice to operate with smaller mixture ratios; a value of 20 at the ground is seldom exceeded.

The optimum mixture at partial loads may be presumed, as at full load, to be fairly near to the upper explosive limit. This limit changes with the load as a result of change in compression pressure and temperature and of change in the percentage of diluting residual gases present. The compression pressure exerts considerable influence on the explosive properties of a weak mixture, necessitating the use of a stronger mixture as the load diminishes. The temperature at the end of compression does not change much since the ratio of temperatures at the beginning and end of compression is a function of the ratio of compression which remains constant; it may be presumed that the temperature effect is negligible. The effect of charge dilution on explosibility has been investigated for mixtures of air with methane and with natural gas, the diluting agent being CO_2.[1] Some of the results of this investigation are plotted in Fig. 190. It is seen that with 20 per cent CO_2, a mixture of natural gas and air cannot be made to explode at atmospheric pressure and temperature;

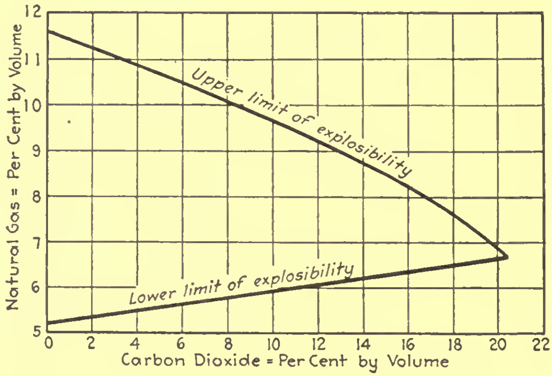


FIG. 190.—Influence of carbon dioxide dilution on the explosibility of a mixture of natural gas and air.

as the percentage of CO_2 diminishes the upper and lower limits recede until with no CO_2 present we have the lower limit with 5.2 per cent and the upper with 11.6 per cent of natural gas present. The figures for a gasoline-air mixture are probably not very different. With higher pressures and temperature the explosibility limits will be changed but the method of variation will be the same.

The amount of dilution of the charge by residual gases can be calculated approximately if the temperature of these gases is assumed. The amount of such dilution will vary with the load since the residual gases fill the clearance at exhaust pressure and are approximately constant in weight at all loads. The amount of such dilution, $d = W_r/W_c$ (where W_r = weight of residual gases and W_c = weight of fresh charge), is shown in

[1]CLEMENT: *Bureau of Mines Technical Paper No.* 43; *"The Influence of Inert Gases on Inflammable Gaseous Mixtures."*

Fig. 191,[1] which also shows the corresponding compression pressures. These curves are for a ratio of compression of 5.5 and must be regarded as approximations only. The pressure at the end of compression is well above atmospheric pressure and as the temperature is probably about 1,100°F. absolute the dilution can be carried further than indicated in Fig. 190 without exceeding the explosive limit. The pressure and dilution of the charge at partial loads are such as to demand a richer mixture

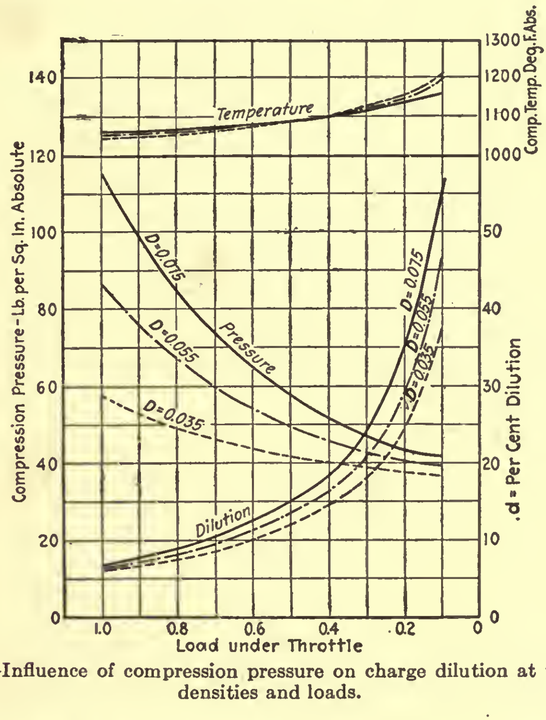


Fig. 191.—Influence of compression pressure on charge dilution at various air densities and loads.

than at full load if a satisfactory explosion is to be obtained. It seems probable that the air-fuel ratio for maximum economy does not fall below 15 for any operating condition that is likely to be met; that is, the maximum-economy mixture approximates to the maximum-power mixture as the air density and load decrease. With this in mind the performance curve for carburetors under partial loads can be examined. It would appear that constancy of mixture ratio under varying load is not desirable, and that

[1] P. S. Tice, *loc. cit.*, p. 634.

a carburetor should show enrichment of the mixture with diminishing load.

Performance of Representative Carburetors.—Several carburetors have been investigated at the Bureau of Standards[1] to

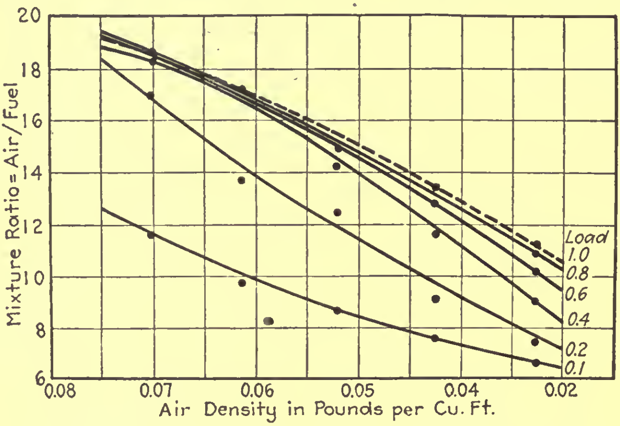


FIG. 192.—Variation of air-fuel ratio in Zenith carburetor.

ascertain the variation in air-fuel ratio with variation (1) of air density and (2) of load. Three of these carburetors are considered here. The Zenith carburetor, *A*, Fig. 177, which is

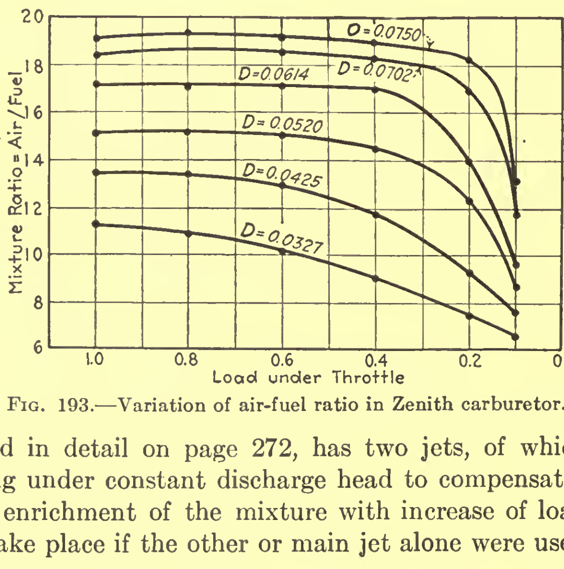


FIG. 193.—Variation of air-fuel ratio in Zenith carburetor.

described in detail on page 272, has two jets, of which one is operating under constant discharge head to compensate for the natural enrichment of the mixture with increase of load which would take place if the other or main jet alone were used. The

[1] P. S. TICE, *loc. cit.*, pp. 620–636.

Stewart-Warner carburetor, *B*, Fig. 177, has the throttle in the intake (anterior) and compensates for load changes by reducing

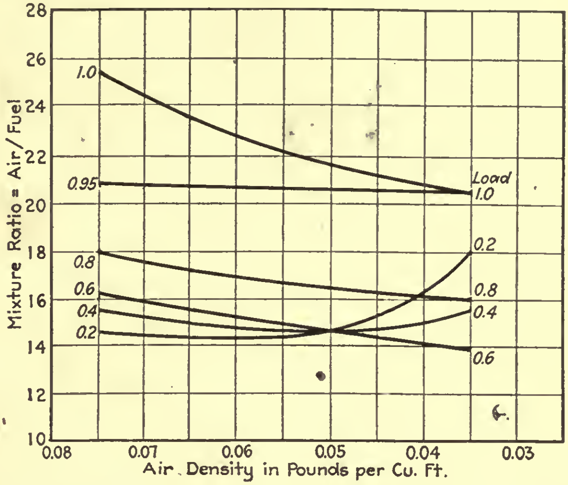


FIG. 194.—Variation of air-fuel ratio in Stewart-Warner carburetor.

the air pressure in the float chamber as the load increases by means of a passage connecting the choke discharge to the float chamber. The Stromberg carburetor, *C*, is described in detail

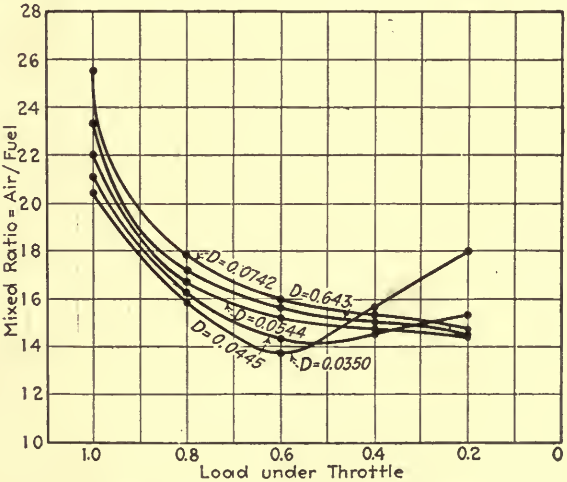


FIG. 195.—Variation of air-fuel ratio in Stewart-Warner carburetor.

on page 278. The results of the investigations are exhibited in Figs. 192 to 197. For each carburetor there is shown the variation of air-fuel ratio with constant throttle opening and variable air density, and with constant air density and variable throttle opening. The absolute values of the air-fuel ratio are unimpor-

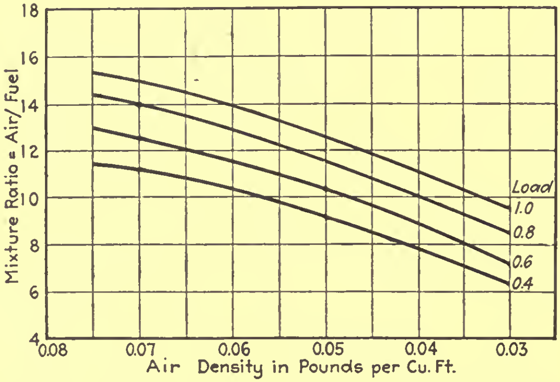


FIG. 196.—Variation of air-fuel ratio in Stromberg carburetor.

tant in this connection since they are controlled by the size of the fuel jet, which can be readily changed; the method of variation of that ratio may, however, be considered as characteristic of each type of carburetor. In the Zenith and Stromberg carbu-

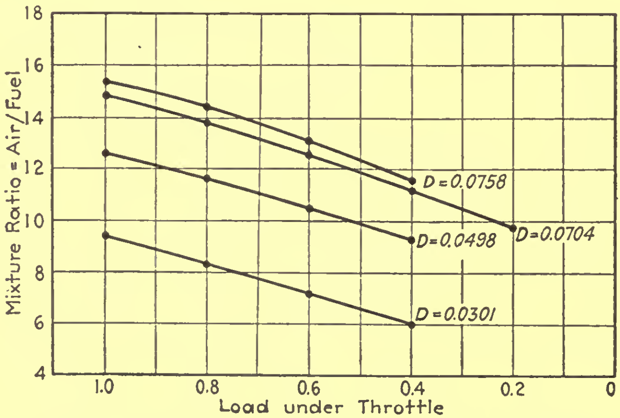


FIG. 197.—Variation of air-fuel ratio in Stromberg carburetor.

retors, the need for an additional altitude control device is obvious; the mixture ratio at full load varies from 19 to 10.5 in the Zenith (Fig. 192) and from 15.5 to 9.5 in the Stromberg

(Fig. 196) as the air density diminishes from 0.07 to 0.03. The enrichment is considerably in excess of that which has been shown (Fig. 188) to be necessary. With load variation at constant air density, the mixture is practically constant in the Zenith carburetor (Fig. 193), but enriches with diminution of load in the other two (Figs. 195 and 197); it has previously been shown (p. 263) that such enrichment is desirable.

Altimetric Compensation.—The importance of maintaining an economical mixture at high altitudes is attested by general experience in the air. British tests, to ascertain the advantages of a special altimetric control of the carburetor, have shown with water-cooled engines an increase in endurance from 4 to 4¾ hr., and in ceiling from 19,000 to 21,000 ft.; with air-cooled cylinders an increase in endurance from 5½ to 6¾ hr., of ceiling from 15,000 to 18,000 ft. and of speed from 84 to 92 miles per hour. In addition to this there is less fouling of the spark plugs, the cylinders keep cleaner, and there is less danger of stalling the engine.

Viscous Flow Carburetor.—It has been shown (p. 259) that with a standard simple carburetor with sharp-edged fuel orifice, the air-fuel ratio varies as the square root of the air density with full throttle and constant engine speed. There is a possibility of making this ratio constant, under varying air density, by substituting for the sharp-edged orifice a capillary passage in which the flow is entirely viscous. The laws of viscous flow are complicated,[1] but, with velocities below those of turbulent flow, it is approximately true that the velocity of flow is proportional to the pressure head. In that case, referring to page 259, we have

$$\frac{W_x}{W_o} = \frac{D_x}{D_o} = \frac{w_x}{w_o}$$

or $\frac{W_x}{w_x} = \frac{W_o}{w_o}$, that is, the air-fuel ratio remains constant with varying air density.

Carburetors have been built embodying the above principle, the viscous flow being obtained by the use of long capillary tubes, or by flow between flat discs or cones as in Fig. 198. At partial loads the fuel supply will fall off in proportion to the decrease in pressure head instead of in proportion to the square root of the pressure head and the mixture will consequently be too weak at low loads. Load control is obtained by raising or lowering the

[1] See HERSCHEL, *Bureau of Standards, Technologic Paper* 100.

disc (or cone) of Fig. 198 and thereby changing the width of the capillary passage; this can be done by interconnection with the throttle lever. The principal objection to this type of carburetor is that the fuel flow varies with the fluidity of the oil and this varies both with the grade of oil used and with its temperature (Fig. 186). A further difficulty is sluggishness in response to quick opening or closing of the throttle valve.

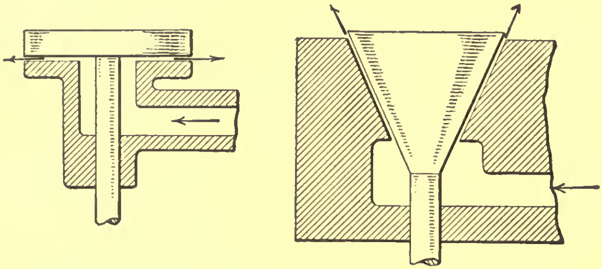


FIG. 198.—Diagrams of viscous flow carburetors.

Altimetric Control.—The only practical method at present available for adjusting the air-fuel ratio to the desired value at all air densities, as well as at all throttle positions, is by the use of an additional or altimetric control. A carburetor may be designed so as to give correct mixtures for varying load or for varying air density but it cannot satisfactorily meet both conditions, since, with the same weight of air flowing the weight of fuel will be different in the two cases. For example, the weight flow of air at half load at the ground will be the same as at full load at an altitude where the air has half ground density; the pressure drop and the fuel flow will, however, be different in the two cases and therefore the air-fuel ratio will be different. If the carburetor is designed to give correct mixture at all altitudes at full load there would have to be added to it a load control (preferably connected with the throttle valve) which would enrich the mixture at partial loads. The other method of procedure is, however, usual; the carburetor is designed to give correct mixtures at full and partial loads, and an altitude control is installed to permit a diminution in the fuel supply at higher altitudes. This control is nearly always manually operated but it can be made automatic without much complication.

A diminution of fuel supply can be brought about either

(1) by diminishing the size of the fuel orifice, or (2) by controlling the pressure head under which the fuel is flowing. The former is most readily accomplished by the use of a needle in the jet; the latter is the method generally employed because it is less sensitive in adjustment and turns out to be more robust as a structure.

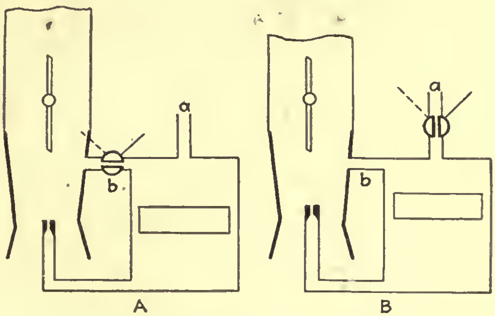


FIG. 199.—Altitude control by regulation of the float-chamber pressure.

Schematic diagrams of some of the more promising methods of altitude control are shown in Figs. 199 and 200.[1] For the control of the float-chamber pressure, Fig. 199, the top of the float chamber must be provided with a vent, *a*, to the atmosphere, and a connection, *b*, to some place where the pressure is less than atmospheric. The control valve may be in either of these passages.

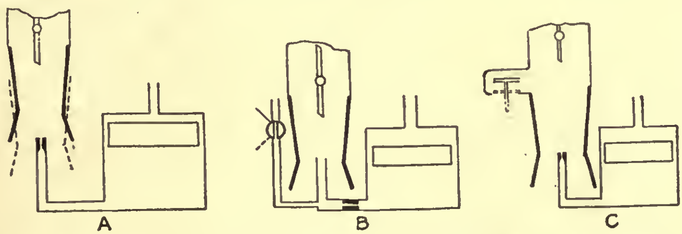


FIG. 200.—Altitude control by regulation of the jet discharge pressure.

The nozzle outlet pressure can be controlled in several ways. The position of the outlet relative to the air passage can be changed, either by shifting the choke, as in Fig. 200*A*, or by shifting the outlet. The amount of air passing the outlet can be reduced by the use of an auxiliary air valve located at a point

[1] *National Advisory Committee for Aeronautics, 4th annual report,* p. 637.

beyond the fuel outlet, as in *C*. A third method is to admit (or bleed) air to the fuel jet past the metering orifice, as in *B*, thereby reducing the pressure head on the orifice.

The structures involving a small plug valve controlling an air stream (Figs. 199 and 200*B*) are the simplest and most easily produced. Their regulation is comparatively direct and involves small forces and a minimum of parts; furthermore, they adapt themselves readily to automatic control. For such reasons, these methods are the ones usually encountered in service. The objection to them is that they do not permit of one setting for all loads at any given air density but require adjustment for each throttle position, if maximum economy is to be maintained.

The method of Fig. 200*A* is structurally clumsy and would complicate the carburetor considerably. The method of Fig. 200*C*, using a balanced auxiliary valve, would offer little resistance to operation and little complication. Moreover, the mixture should be satisfactory at partial loads without further manipulation. The auxiliary valve would have to be large to give complete compensation up to one-half ground density. A simple calculation shows that for this range the area of the auxiliary port must be approximately 1.5 times that of the carburetor throat.

Manual operation of the altitude control is extremely undesirable. The operation should be continuous as the plane changes its altitude or speed and can at best be only intermittent with manual operations. Moreover, the pilot has no definite means of knowing how far to move the control but must rely chiefly on the engine tachometer readings. He can find the maximum power position but not the more important maximum economy position. As he is already burdened with a large number of controls it is much better to make the altimetric compensation automatic.

The simplest automatic operating device is an aneroid bellows. A sealed flexible-walled chamber will expand under reduced pressure and under increased temperature, that is, it will respond to change in air density. If correction for pressure only is desired, the bellows can contain a spring under compression and can be exhausted before sealing (see Fig. 212). Such devices can only operate satisfactorily if the resistance which they have to overcome is small and if the method of control is such as not to disturb the compensation at partial loads.

Atomization.—The preceding discussion has concerned itself with the metering or mixture-making characteristics of carburetors. Other qualities which are of importance are (*a*) the degree of atomization of the fuel and the homogeneity of the mixture; (*b*) the pressure drop through the carburetor at wide-open throttle; (*c*) satisfactory idling performance; (*d*) acceleration. All carburetors, in order to be acceptable, must be satisfactory not only in mixture making but also in these other characteristics. Favorable conditions for fine **atomization** of the fuel are high velocities of the air and, to a minor degree, of the fuel. The air velocity is always much greater than that of the entering fuel and the atomization is largely due to the high relative velocity of the air. This is particularly marked if the fuel is not discharged in the axial direction. The use of an anterior throttle, as in Fig. 177*B*, by increasing the air velocity at the jet improves atomization at partial loads. The admission of air before the fuel outlet but past the orifice (see Fig. 200*B*) is a further favorable condition.

Good atomization may be impaired by the impinging of the mixture on obstacles such as a butterfly throttle valve, placed centrally above the jet (see Fig. 177*A*). Here again an anterior throttle has an advantage. The mixture will impinge on the inlet manifold and the valves before getting into the cylinder, but it is better to have such actions take place as far away from the mixing point as possible. Best results have been obtained with a long pipe leading from the carburetor to the manifold, giving more time for vaporization and the formation of a homogeneous mixture before the mixture is taken into one or other branch of the manifold.

Pressure drop through the carburetor has been touched on in page 251 in the discussion of the discharge characteristics of the air passage. Its importance is solely in affecting the maximum power output.

Idling.—An engine requires a richer mixture at lighter loads. When the engine is cold a still richer mixture is necessary. None of the carburetors in use on airplanes will give a satisfactory idling mixture without the use of some auxiliary device. This consists of a fuel discharge above the throttle which utilizes the high vacuum above the closed throttle to suck in the necessary amount of fuel.

Acceleration.—It is of importance that the mixture should respond rapidly to sudden changes in load. If the throttle valve is opened suddenly, the greater density and inertia of the fuel tend to make the mixture too weak, with the result that the engine will back-fire or misfire. To avoid this, it is common to have an auxiliary supply of gasoline which, at partial loads, collects near the fuel outlet and is drawn on first when the throttle is suddenly opened, keeping up the strength of mixture until the regular flow is established.

Certain special conditions have to be met with by an airplane carburetor as a result of manœuvres of the plane. The changing inclination of the plane will change the hydraulic head at the jet unless it is placed at the center of the float chamber. With the usual non-concentric arrangement of parts (see Fig. 177) it is desirable to have the float chamber placed in advance of the jet as this will give a greater hydraulic head and richer mixture on climbing and will cut down the fuel supply on descent or diving. It is necessary to see that the gasoline does not overflow from the jet when the plane is resting on the ground. The action of the float and float valves during a dive must be examined. The usual float, guided by a central spindle which is normally vertical, will go out of action during a dive, with the probable result of flooding the carburetor. Special float mechanisms are desirable and have been devised. In case of flooding during a dive, the air horn or intake pipe should be so arranged that gasoline cannot spill out into the fuselage. As the air horn is usually facing forward to get the advantage of the increased air pressure due to the relative wind velocity, such spilling will occur unless the air intake pipe is led upward before being turned forward.

The usual dual carburetor has one float chamber, and one air intake to the two chokes. A dual air intake pipe is to be recommended as reducing the risk from back-fire, by making each group of three or four cylinders a separate unit so far as carburization is concerned. With a common air pipe, back-fire may cause the engine to stop; with double intake, back-fire into one intake will not interfere with the operation of the cylinders fed from the other intake, the engine continues to run and the flame in the back-firing intake is drawn up into the engine, reducing the risk of fire. Furthermore, a dual intake increases engine power by diminishing the resistance to air flow.

CARBURETOR CONSTRUCTION

Zenith.—The carburetor which has been used most for airplane engines is made by the Zenith Carburetor Co. In this carburetor, an attempt is made to maintain constant mixture strength at varying throttle positions by the use of two jets or nozzles, one of which, the *main jet*, acts in the usual way, while the other, the *compensating jet*, delivers an amount of fuel which is entirely independent of engine speed and load. This arrangement was devised by Baverey in 1906. The main jet alone would give a mixture which is at all times too weak, but which becomes richer as the engine speed and load increase; the compensating jet alone would give a mixture which is at all times too weak but which becomes weaker still as the engine speed and load increase. The two jets working together tend to compensate one another, and, if properly proportioned, will give a mixture of fairly constant strength under varying speed and load. This is shown in Fig. 193. In this case, the jet sizes are No. 140 for the main jet and

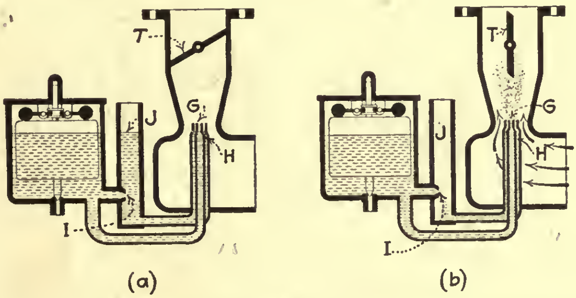


FIG. 201.—Diagram showing action of the Zenith carburetor.

No. 150 for the compensating jet, the number indicating the cubic centimeters of water discharged per minute under a 12-in. head. The discharge for the compensating jet is under a constant head of 2 or 3 in. of water; the main jet discharge is under the variable head due to the pressure drop at the throat of the venturi, which depends on the size of the throat and may amount to 40 in. of water in usual designs. The arrangement of these jets is shown diagrammatically in Fig. 201, in which *a* shows conditions at rest, and *b* at full throttle. The main jet, *G*, is located as usual; the compensating jet, *I*, discharges into the well, *J*, which empties into a nozzle, *H*, concentric with the main jet, *G*. When at rest, the levels in the float chamber, the wells, and the nozzles *G* and *H*, are the same. On opening the throttle, the capacity of the nozzle, *H*, is so much greater than that of the jet, *I*, that the well, *J*, is kept drained and both air and fuel are sucked up the nozzle, *H*. As the pressure in the well, *J*, is atmospheric, the discharge through *I* is due to the hydrostatic head of the liquid in the float chamber and is therefore constant. The well, *J*, serves also as an accelerating well, giving a body of fuel immediately available on opening the throttle from the idling position. At low speed, when the throttle valve, *T*, is nearly closed, the suction at the throat is not sufficient to draw in any gasoline and it enters only through the *idling device*. This device, shown diagrammatically in Fig. 202*a*, consists of the idling tube, *M*, within the secondary well, *P*, which is inserted in the main well, *J*, into which the discharge from the compensating jet, *I*, occurs. The well *P* is provided with a small metering orifice at the bottom through which gasoline can enter from *J*, and with small air

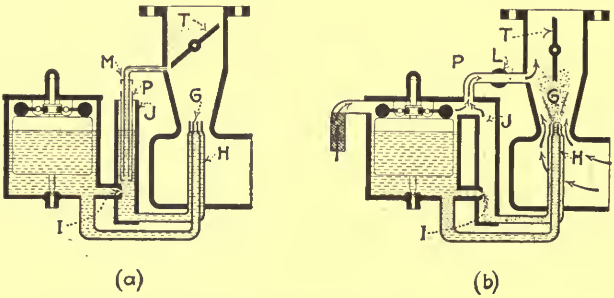


Fig. 202.—Diagram showing (*a*) idling device and (*b*) altitude control of the Zenith carburetor.

holes at the top. The idling tube, *M*, terminating opposite the throttle valve, is subjected to a very strong suction whenever the throttle is nearly closed and discharges gasoline from the well *P*. This gasoline meets the air passing with great velocity through the small opening around the throttle valve and forms the idling mixture. As the throttle is opened, the vacuum at the throttle diminishes while that in the choke increases, so that discharge through *M* ceases and that through *G* begins.

The *altitude control* of the Zenith carburetor is shown diagrammatically in Fig. 202*b*. It is of the type illustrated in Fig. 199*A*. The float chamber is open to the air through screened air inlets. The well *J* is in open communication at its top with the float chamber. A passage, *P*, from the float chamber to the choke discharge, is fitted with a stop cock, *L*, which is manually operated by the pilot. This cock is closed at the ground and is opened gradually as higher altitudes are reached; it should be opened as far as is possible without appreciably diminishing the revolutions of the engine.

The actual construction of a Zenith carburetor is shown in Fig. 203. Gasoline enters the float chamber through *D* and the needle valve seat, *S*. As soon as it reaches a predetermined height the metal float, *F*, acting through the levers, *B*, and the collar, *N*, closes the needle valve, *C*, on its seat *S*. From the float chamber the gasoline flows (1) through the compensating jet, *I*,

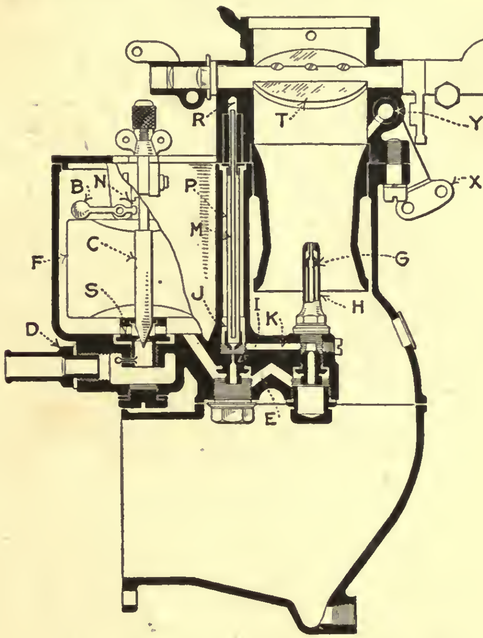


FIG. 203.—Section of Zenith carburetor.

into the bottom of the well, *J*, and then through the channel, *K*, to the cap jet, *H*, which surrounds the main jet, *G*, and (2) through the channel, *E*, to the main jet, *G*. The idling tube, *M*, is inside the secondary well, *P*, and discharges through the passage, *R*, to an opening (not shown) opposite the throttle valve. The altitude control valve, *Y*, is a tube which is shown communicating with the choke discharge; the other communication to the float chamber is not shown. It is operated by the lever *X*.

As in other carburetors, a single float chamber is used to supply two air chokes if the engine has six or eight cylinders. One air horn or intake commonly serves the two chokes of a duplex carburetor, but it has been found that greater engine power can be obtained if separate intakes are used. Tests of special Zenith carburetors for the Liberty engine showed maximum power developed with separate air intakes about 4 in. long.[1]

The special feature of the Zenith carburetor which has recommended it is the absence of all moving parts. It is general experience that auxiliary air valves, metering pins, and other moving devices will stick at times and cause irregularity of action. For maximum reliability and fool-proofness the compensating device should be fixed.

The **Claudel** carburetor, which has been used very extensively for airplane engines, especially in Europe, is now being made in this country. Like the Zenith, the compensation for load and speed is made without any moving parts. A general view is shown in Fig. 204. The fuel discharges into the choke from a *diffusor* which is shown assembled in Fig. 205*b*. The diffusor has four concentric tubes, the air tube *e*, guard tube *d*, diffusor tube *c*, and idling tube *a*. The main jet is in a small plug screwed into the bottom of the diffusor. Air at atmospheric pressure enters the bottom of the air tube, passes over the top of the guard tube (which prevents the fuel from overflowing when the engine is at rest), then goes through such holes in the diffusor as are above the fuel level, and out through the nozzle holes to the throat of the venturi. The fuel is at the level shown when the engine is idling or at rest. As the throttle is opened, the suction in the diffusor increases, thereby lowering the liquid level in the diffusor bore and uncovering progressively a series of air-bleed or compensating holes. Through these holes the air rushes into the ascending column of fuel and atomizes it as it leaves the nozzle holes at the top. At maximum load the diffusor is practically emptied and all the air-bleed holes are in action, cutting down the effective head on the fuel. The compensation is by controlling the jet outlet pressure along the lines indicated in Fig. 200*B*. Any desired kind of compensation can be obtained by appropriate design of the size and location of the air-bleed holes.

The diffusor acts also as an accelerating well. When idling the diffusor is out of action and all the fuel goes through the cen-

[1] *Bulletin*, Experimental Department, Airplane Engineering Division, U. S. A., Jan., 1919.

tral idling tube, mixed with some air entering compensating holes from the air tube.

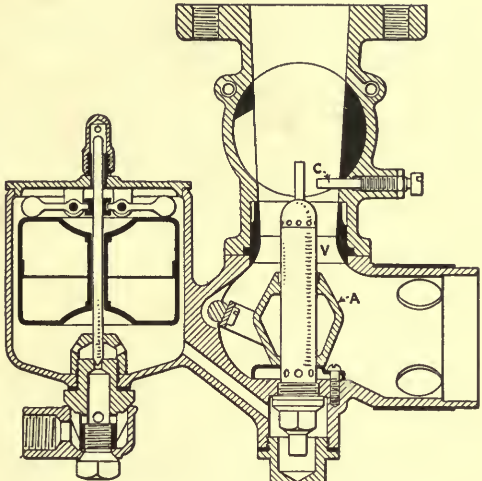


Fig. 204.—Section of Claudel carburetor.

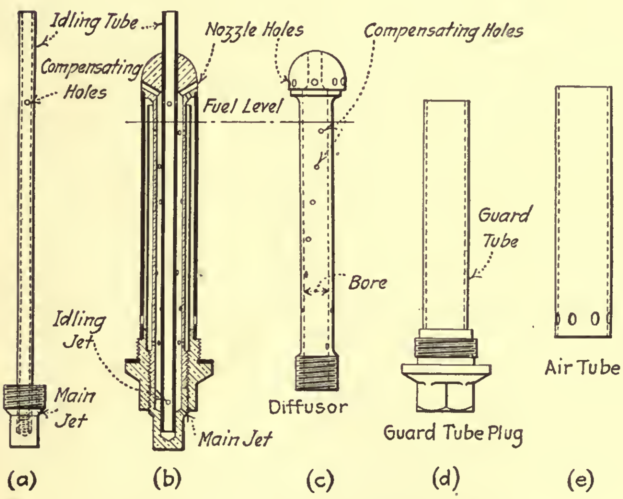


Fig. 205.—Details of Claudel diffusor.

The throttle is a cylindrical or barrel throttle, bored out so as to form a smooth continuation of the venturi when it is wide open. It offers no resistance at maximum load and consequently leads to maximum volumetric efficiency and power. As the idling tube projects into the throttle space, the throttle is slotted out wide enough to pass around it. To diminish the area through this slot when the engine is idling a screw, *c*, extends into the air space. Advancing the screw lessens the air area and enriches the idling mixture. Figure 206 shows the idling position.

Another feature of this carburetor is the sliding air cone, *A* (Fig. 204), which is controlled by an external lever. When the cone is raised to contact with the venturi, it shuts off all air supply and puts maximum suction on the diffusor. This greatly enriches the mixture and is advantageous for starting in cold

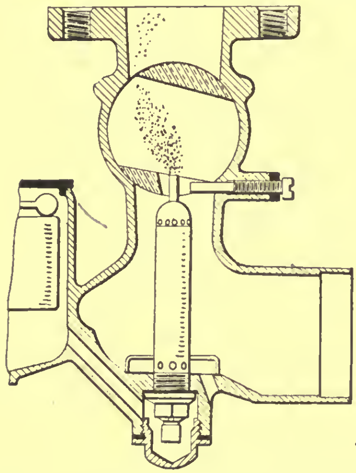


FIG. 206.—Idling device of the Claudel carburetor.

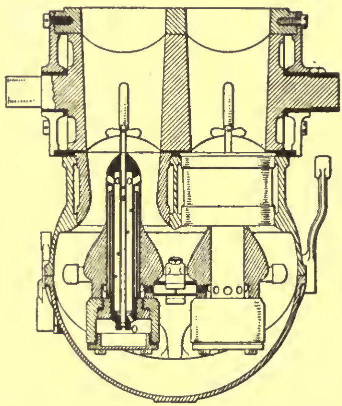


FIG. 207.—Section of dual Claudel carburetor.

weather. The same device is used for altitude control. The venturi used in airplanes is larger than is necessary at the ground. At low elevations the air cone is kept in a raised position in order to increase the suction in the diffusor to the amount necessary to give the desired mixture. As elevation is gained the air cone is gradually lowered, thus compensating for the natural increase in richness.

A cross-section through the diffusors and throttle valves of a duplex Claudel carburetor as used on the Hispano-Suiza engine is shown in Fig. 207.

The **Stromberg** carburetor, Fig. 208, although structurally very different, uses the same general method of compensation for speed and load as the Claudel. The special features of this carburetor are the float mechanism and the double venturi.

The float (Fig. 208) is spherical or cylindrical (with horizontal axis) and is hinged as shown with the pivot toward the tail of the

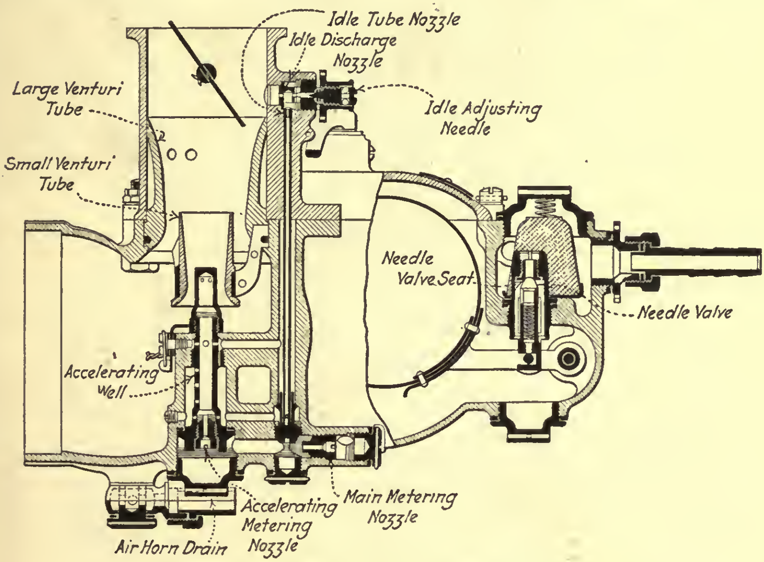


FIG. 208.—Section of Stromberg carburetor.

plane. With this mounting, the float is in action during all ordinary manœuvres of the plane (Fig. 209), that is, it keeps the needle valve closed with a moderate amount of gasoline in

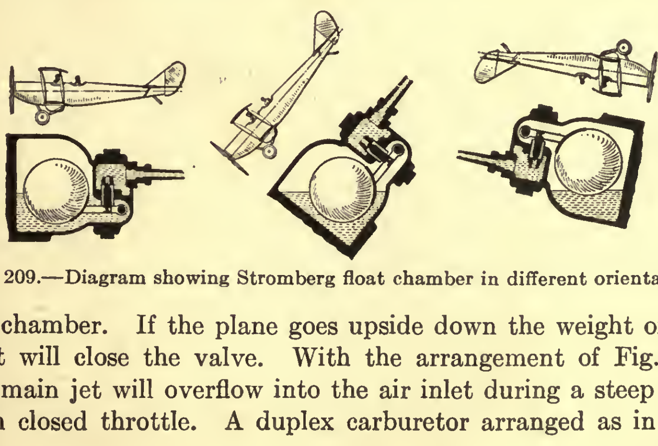


FIG. 209.—Diagram showing Stromberg float chamber in different orientations.

the chamber. If the plane goes upside down the weight of the float will close the valve. With the arrangement of Fig. 208 the main jet will overflow into the air inlet during a steep dive with closed throttle. A duplex carburetor arranged as in Fig.

210, with the float between the two discharge jets, leaves no possibility of such leakage of fuel.

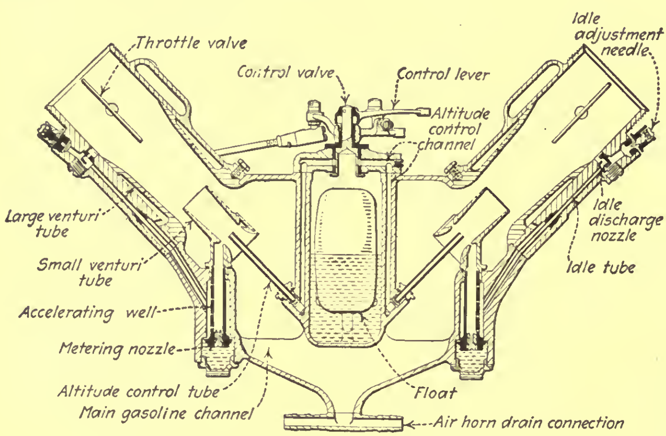


FIG. 210.—Section of dual Stromberg carburetor.

The diagrammatic sketch (Fig. 211) shows the metering jet, *E*, discharging into channel, *A*, with air-bleed holes, *D*, through which air at atmospheric pressure enters from the outer channel, *B*. The outer channel is also the accelerating well.

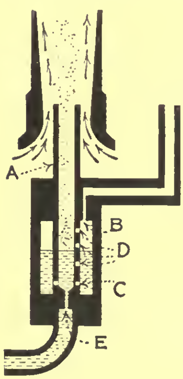


FIG. 211.—Diagram showing load control of Stromberg carburetor.

The fuel and the atomizing air are discharged radially into the choke through a ring of small holes, located at the throat of a small venturi tube. This small venturi is concentric with a larger venturi and discharges at its throat. The discharge pressure of the small venturi is considerably below atmospheric pressure and the depression is still greater at the throat of the small venturi. This results in very high velocity for that portion of the air supply which passes through the small venturi, giving good atomization of the fuel without having to make the whole air supply acquire a very high velocity. This arrangement gives a small total pressure drop in the carburetor, and consequently high volumetric efficiency of the engine.

The idling device is a miniature carburetor with discharge just above the closed throttle. The idling tube connects directly with the main jet passage and has a fuel nozzle discharging into a mixing chamber where it meets air entering through holes which are controlled by a needle valve. The discharge nozzle into the main choke is a slot of which more is exposed as the throttle moves from its closed position. The increased opening of the slot increases the suction in the mixing chamber, and sucks up more fuel as the throttle begins to open. With still further opening the suction at the main discharge nozzle increases while that at the idling nozzle decreases. There is a throttle position at which fuel discharges through both, but with still further opening the idling nozzle goes out of action.

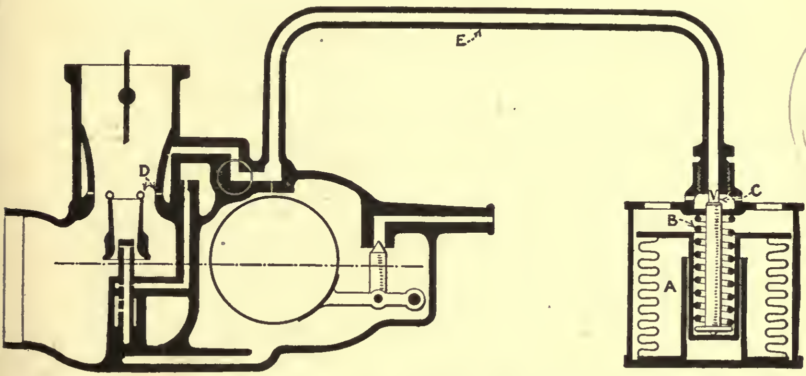


Fig. 212.—Diagram showing automatic altitude control attached to Stromberg carburetor.

Altitude compensation is effected by controlling the pressure in the float chamber. An arrangement for automatic control is shown diagrammatically in Fig. 212. The aneroid chamber, *A*, which has been exhausted before sealing, is compressed by the joint action of the air pressure and the spring *B*. As the air pressure diminishes the aneroid expands compressing the spring and raising the valve *C*. The valve point is slotted and offers a decreasing aperture for the admission of air as the valve rises. Air is sucked through this slot by the action of the venturi at *D*, and as the only air vent from the float chamber is into the pipe *E*, the pressure in the float chamber will vary with position of the valve *C*. An additional manual control is a necessary safety device.

The design in Fig. 210 is especially adapted to a 90-deg. Vee engine, and as previously pointed out, permits a position of the float chamber between the two carburetor outlets which largely eliminates the disturbing factor of changing inclinations of the plane. The carburetor barrels are water-jacketed for high altitude service. The main fuel nozzles are in an annular groove around the small venturi. The altitude-control suction is through the small axial tubes shown terminating at the throats of the small venturis and consequently give the maximum possible suction and range of action of the control. The altitude control has a partial connection with the throttle in such way that the

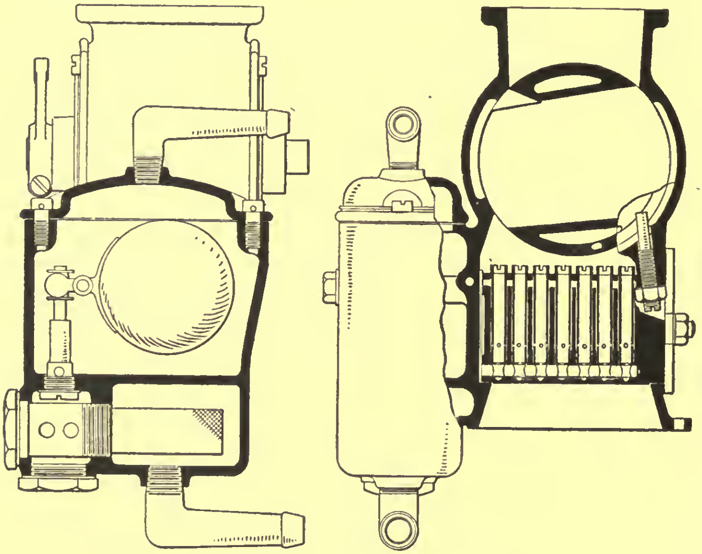


FIG. 213.—Sections of Miller carburetor.

mixture is enriched during the latter part of the closing of the throttle.

The **Miller** carburetor has been used on the U. S. Bugatti engine. It is of the multiple-jet type in which load compensation is effected by bringing more jets into action as the throttle is opened, the sizes of the jets being designed to give correct mixture at all loads. The jets are air-bled, giving compensation for varying speed. The jets are held in a narrow holder (Fig. 213) and discharge across a diameter at the throat of the venturi. The drill sizes for the Bugatti engine are No. 76, which is the idling jet, No. 76, No. 75, No. 71, No. 68, No. 57, No. 53. The corresponding diameters in inches are 0.020, 0.021, 0.026, 0.031, 0.043, 0.0595; the areas consequently increase very rapidly. These jets come into action progressively as the throttle is opened. Each jet has four small air holes just above the metering orifice; air enters at atmospheric pressure through a $\frac{3}{16}$-in. hole near the top of the jet holder and passes down around the outside of each jet to the air holes. The gasoline flows from the float chamber to the lower $\frac{3}{16}$-in. hole in the jet holder. The idling jet is the first in the holder.

The throttle valve is of the barrel type bored out to give a venturi form when wide open. The stop for the idling position is seen in the figure. Altitude compensation is obtained by varying the pressure in the float chamber, the air space of which is at all times in direct connection with the venturi. A manually-

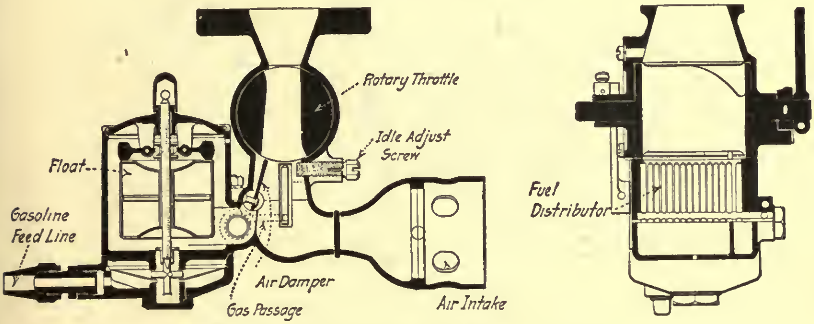


Fig. 214.—Sections of Master carburetor.

operated valve controls the size of the free air connection to the top of the float chamber.

The **Master** carburetor is also of the multiple-jet type, but differs from the Miller in that the jets are all of the same size and are not air-bled. The throttle is of barrel type (Fig. 214) with an opening that is curved so as to uncover the jets progressively as the throttle is opened. An air damper controlled by the pilot restricts the venturi opening and consequently enriches the mixture when desired for starting. The number of jets is usually from 14 to 21, which demands extremely small metering orifices.

The **Ball and Ball** carburetor (Penberthy Injector Co.) is of the single metering orifice, air-bled type. The float is spherical in a spherical chamber. The venturi throat, *A*, (Fig. 215), has the main nozzle tubes, *B*, connecting through the annulus, *C*, with the passage, *D*, and the mixing chamber, *E*. The metering jet, *F*, is at the bottom of the nozzle, *G*, and the fuel overflows through the four air holes, *H*, into the chamber, *E*, which connects to the outside air through the passage, *M*, and the air orifice, *N*. Gasoline arrives from the float chamber at *J*. The idling jet, *P*, connects through the passage, *O*, with the mixing chamber, *E*, and discharges just above the closed throttle. An auxiliary air valve, *S*, is sometimes used to reduce the strength of the mixture at heavy loads.

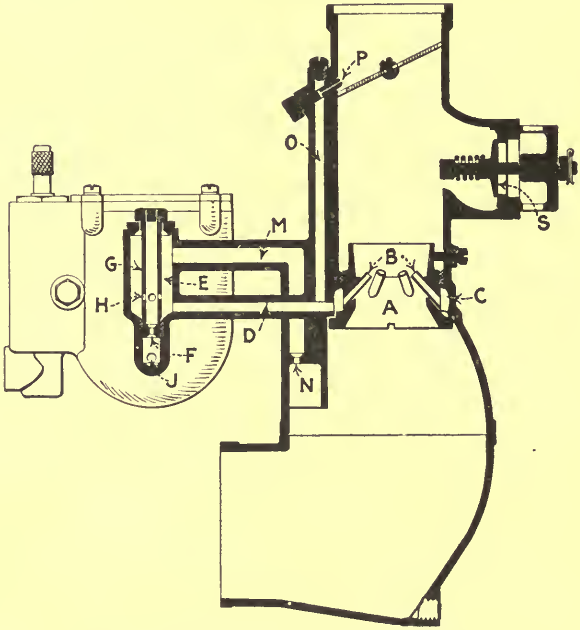


FIG. 215.—Section of Ball and Ball carburetor.

The altitude control is by variation of the pressure on the discharge side of the main jet. This is accomplished by substituting a larger valve-controlled opening for the air orifice, *N*; opening this valve increases the pressure on the discharge side of the main jet and weakens the mixture.

The carburetor used on the (German) **Bassé-Selve** engine is simpler and lighter than any of the types previously discussed. The float (Fig. 216) is annular, and concentric with the choke, thereby reducing the possibility of overflow of gasoline from the main jet when the carburetor is inclined. The main jet is formed by a hole drilled in a tube which is screwed diagonally into the water-jacketed body of the carburetor and lies across the choke tube. The jet tube is open at its lower end and projects into the bottom of the float chamber. The idling jet is formed by a second tube of small diameter inside the jet tube. This idling tube is also open at the bottom and is drilled radially with a small hole just below the main jet. It communicates with the mixing chamber just above the throttle by a passage drilled in the carburetor body. Altitude compensation is by varying the air pressure in the float chamber.

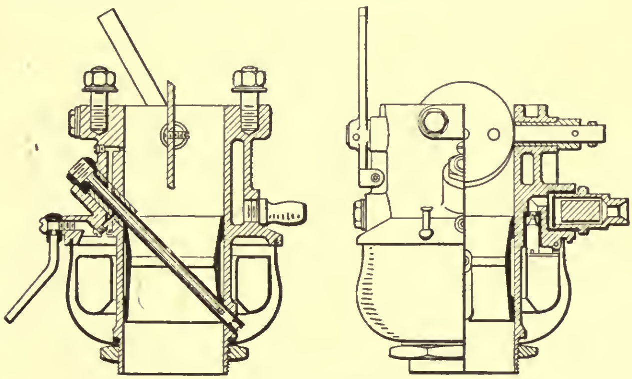


Fig. 216.—Sections of Bassé-Selve carburetor.

The float chamber is made of pressed sheet steel of very light gage. The needle valve (Fig. 216) is acted on directly by the float without the intervention of levers.

The carburetor of the **Bayerische Motoren Werke** engine has some noteworthy features. It consists of three carburetors with a common float chamber (Fig. 217). Each of these carburetors has a separate discharge pipe leading to a common induction manifold. The central carburetor has both idling and main jets; the outer two have main jets only. There are five throttle valves arranged in two systems with independent control. The main system has three throttles, one to each carburetor. The secondary system, which is an altitude control, has valves on the outer carburetor only.

The action is as follows: When the main throttle is opened slightly, the side throttles remaining closed, the idling jet (center carburetor) alone is in action; mixture from the center carburetor alone reaches the cylinders. As the throttle is opened further the main jet of the center carburetor comes in action and supplies the whole mixture until the throttle is half open. After this, the two side carburetors, which are controlled by slotted links, begin to open. The normal continuous ground level full-power operation is at the point where the side jets are just about to begin to discharge.

So long as the secondary throttles remain in their closed position with relatively small passages past them, a comparatively rich mixture is supplied by the side carburetors. As altitude is gained the secondary throttles are opened and give increased power while keeping the mixture of the desired strength.

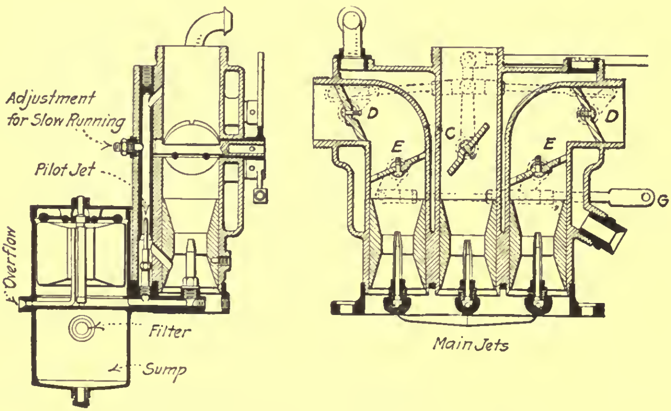


FIG. 217.—Sections of B.M.W. carburetor.

An entirely different type of carburetor is used on the **Maybach** engines on large German dirigibles. These have been designed to dispense with the use of a float chamber and to work in conjunction with a gasoline-pump system. The construction is shown diagrammatically in Fig. 218. The throttle valve, J, is of the rotary-barrel type and admits carbureted air from N and fresh air from L. The throttle lever is interconnected with the sliding shutter, K, controlling the air that flows past the jets, and with a rotatable cover, P, regulating the size of the jets. Fuel from the gasoline pump enters an upper vessel, A, by the pipe, B. The level in this vessel is kept constant by an overflow pipe, C, which conducts the excess fuel back to the supply tank. An air vent fitted with a baffle plate is provided at F. The fuel passes from *A* through a strainer, *M*, to the vessel, *D*, whence it is sucked through the orifice, *H*, into the induction pipe. Excess fuel in *D* overflows and joins the excess from *A* in the pipe *C*. At the top of vessel *D* two holes are drilled—the main and idling jets. These orifices are controlled by the eccentrically-mounted cap, *P*, which is rotated through interconnection with the throttle

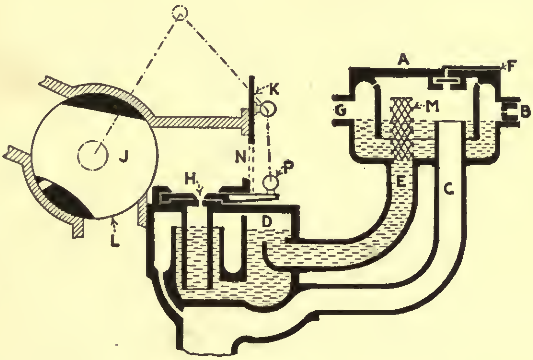


FIG. 218.—Diagram of Maybach carburetor.

lever. The fuel has a constant liquid head equal to the difference in levels between the liquid in *A* and the level of the orifices; in addition it is subjected to the suction in the passage above *H*.

In the idling position, *L* is open slightly (Fig. 219), *K* is closed, and the idling jet only is uncovered by *P*. The throttle-lever

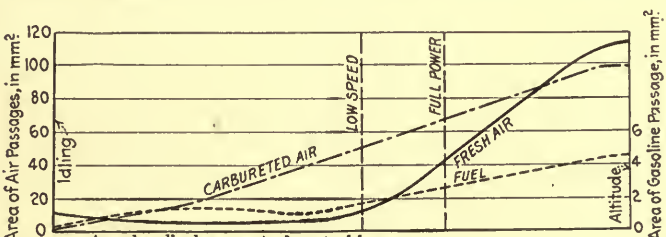


FIG. 219.—Action of the Maybach carburetor.

quadrant is marked with the positions "idling," "low speed," "full power," and "altitude." As the throttle is rotated from the idling position, which demands a rich mixture, the shutter *K* opens but the fuel opening does not increase much till the "low speed" position is reached; the fuel discharge increases in consequence both of increased fuel orifice and of the increased suction at *H*. The "full power" position is not maximum power but is the maximum at which it is desirable to operate the engine at ground level. The fuel orifice is nearly wide open at the full-power position. With further opening of the throttle the fresh-air inlet *L* opens more, thereby preventing the enrichment of the mixture which otherwise would occur at high altitudes and maximum power.

It is evidently possible to design the dimensions and the interconnections of the three orifices *G*, *N* and *H* in such way as to give any desired mixture to an engine operating at ground level and at maximum power at various altitudes. Partial loads at high levels are not provided for. This method of meeting the carburetor problem is undesirable because of the complexity of the design and the practical impossibility of making the varying fuel orifices of the desired dimensions. This particular carburetor is very heavy and offers a large air resistance, thereby reducing the volumetric efficiency and power of the engine which it supplies.

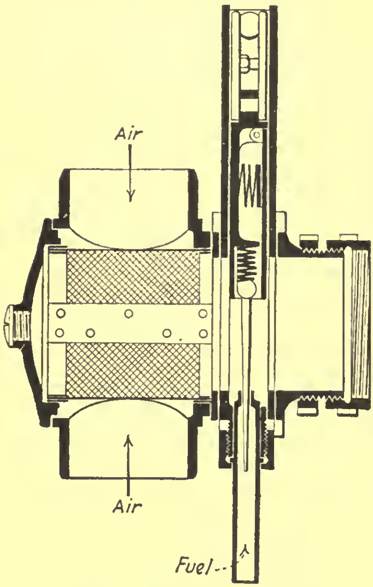


Fig. 220.—Section of LeRhone carburetor.

A very simple type of carburetor is used on the rotary **Le Rhone** engine. The air-fuel mixture enters the rotating crankcase through a stationary hollow crankshaft. The screened air supply is controlled by a throttle which is in the form of a shutter (Fig. 220) carrying at its lower end a long metering pin which controls the size of the fuel jet. The pressure at which the fuel arrives at the orifice is controlled by a by-pass valve; this serves to control the mixture when altitude or load is changed. The inherent mixture control is irregular and uneconomical with a device of this nature.

CHAPTER XI

FUEL SYSTEMS

The following statement of the requirements of the fuel system of an airplane engine is abstracted from the "Handbook of Instructions for Airplane Designers" prepared by the Engineering Division of the U. S. Air Service.

There should always be more than one means of supplying fuel to the engine.

Main-feed System.—Gravity feed should be used throughout if it is possible to maintain a sufficient head with the airplane at maximum angles of flight. It has been found that a head of 18 to 30 in. is required for satisfactory operation of current types of carburetor. Unless it is possible to maintain a sufficient head by gravity, pumps must be installed to supply gasoline from the main tanks to the engines. Pressure in supply tanks is not permitted on fighting planes.

The main fuel pumps should have a capacity at least 50 per cent greater than the maximum requirement of the engines. Two pumps, other than hand pumps, are desirable, either of which can supply sufficient fuel. They should have automatic pressure regulation to eliminate the use of relief valves or other means of adjusting the pressure at the carburetor. The gasoline pressure at the carburetor must always be at least 1 lb. and the system should be so adjusted that this pressure can never rise above 3 or 4 lb. as a result of change in position of the airplane. Pumps capable of a discharge pressure higher than 4 lb. should have relief valves connected between the suction and discharge, so adjusted as to limit the maximum discharge pressure to 4 lb. The fluctuation of pressure at the carburetor, due to pulsations of the pump, should not be over 25 per cent. Where air pressure is used, the power air-pump must be capable of keeping a pressure of 2 lb. on the tanks at the ceiling of the airplane and both spring- and manually-controlled relief valves should be furnished, the former set to relieve at 4 lb. per square inch.

Pumps should preferably be located below the lowest point in the supply system. If they are located higher than the bottom of the main tanks, means must be provided for admitting gasoline from the auxiliary supply to the suction side of the pumps. It should be possible for the pilot to make use of this connection during flight. A non-return valve must be installed to prevent this gasoline from going into the main tanks instead of the pumps.

Pumps which do not require glands are preferred, although satisfactory glands will be accepted; in case glands are used, they must be so located that any leakage from the glands will be drained to a point outside of the fuselage. Pumps should preferably be connected to and driven by the engine.

Auxiliary-feed System.—The auxiliary-feed system supplies gasoline to the engine in case of failure of the main supply; this auxiliary system should be such that fuel can be supplied to the engine in the shortest possible time never to exceed a period of 10 sec. from the time the pilot starts to make use of the auxiliary system. For emergency use, gravity tanks are best, but must not be used if a head of 12 in. in level flight is not obtainable; they should have sufficient capacity to operate the engines for 30 min. at an altitude of 10,000 ft. with wide-open throttle and should be so connected to the system that they can be shut off and used for reserve or emergency only. They must be so constructed or connected that they can be entirely emptied with the airplane inclined at maximum angles of flight. An overflow pipe from the gravity tank returns any excess gasoline to one or more of the main tanks; this overflow must be so constructed that there can be no gasoline trapped in it when the airplane is in normal flying position.

Unless there are three means of delivering fuel to the engine, such as two engine or wind driven pumps and a gravity tank, a hand gasoline pump must be provided which will permit the pilot, while controlling the airplane, to pump, without undue exertion, sufficient fuel from the main supply at proper pressure for the operation of all engines at full throttle. The capacity of this auxiliary system must be such that the pilot will not need to operate the pump during more than one-third of the time.

Tanks should be of tinned steel and of such thickness that the tank will stand 5 lb. per square inch pressure on the inside without undue distortion. Flat surfaces are to be avoided. Wherever the width or length (horizontally) of a tank is greater than 12 in., a splash plate for reinforcing purposes must be installed at least every 12 in.; wherever the height of a tank is greater than 18 in., a splash plate for reinforcing purposes must be installed at least every 18 in. All seams, including the connection between the splash plates and the walls of the tanks, should be riveted and soldered. Copper or soft iron rivets must be used throughout; the exposed parts of the rivets to be tinned in case iron rivets are used.

Drains leading to a point outside the fuselage must be installed in the bottom of each main tank. **Fillers** must be conveniently located on each tank, and in such a position that the entire tank can be filled while the airplane is on the ground. A removable screen must be installed at the point of filling of each main tank, and also, if practicable, in the gravity tank. **Vents** must be located at the highest point on all tanks, usually in the filler tube, except on wing gravity tanks where the overflow pipe shall act as a vent.

Line and Carburetor Strainers.—A line strainer, with removable screen and bowl, must be installed between the tanks and pumps, located as low as possible and in such a position as to be readily accessible for draining and cleaning. The strainer screen should be of brass, bronze or copper of about 100-mesh and 0.005-in. diameter wire, and should have at least 1 sq. in. for each 6 gal. which must pass through per hour.

Each carburetor should be provided with a strainer having a readily removable screen of brass, bronze or copper of about 50 mesh and approximately 0.009-in. diameter wire and having an area of at least 2 sq. in.

Service Pipes and Connections.—Service pipes should be 3/8 in. outside diameter where flow is 30 gal. per hour or less; 1/2 in. outside diameter where flow is between 30 and 60 gal. per hour; and 5/8 in. outside diameter where flow is between 60 and 100 gal. per hour. All vent and air tubes should be $\frac{1}{4}$ in. outside diameter. Wall thickness should be 0.028 in. for $\frac{1}{4}$-in., $\frac{1}{32}$ in. for $\frac{3}{8}$-in.. and $\frac{3}{64}$ in. for $\frac{1}{2}$-in. and $\frac{5}{8}$-in. outside diameter. All service pipes, or tubing, should be seamless and of annealed copper, soft enough to withstand vibration. At all points where the tubing is connected to solidly mounted objects, such as pumps or tanks, flexible connections must be provided. The tubing must be properly protected at points of possible chafing. Sharp bends are not permitted. Tube fittings are to be of brass or bronze.

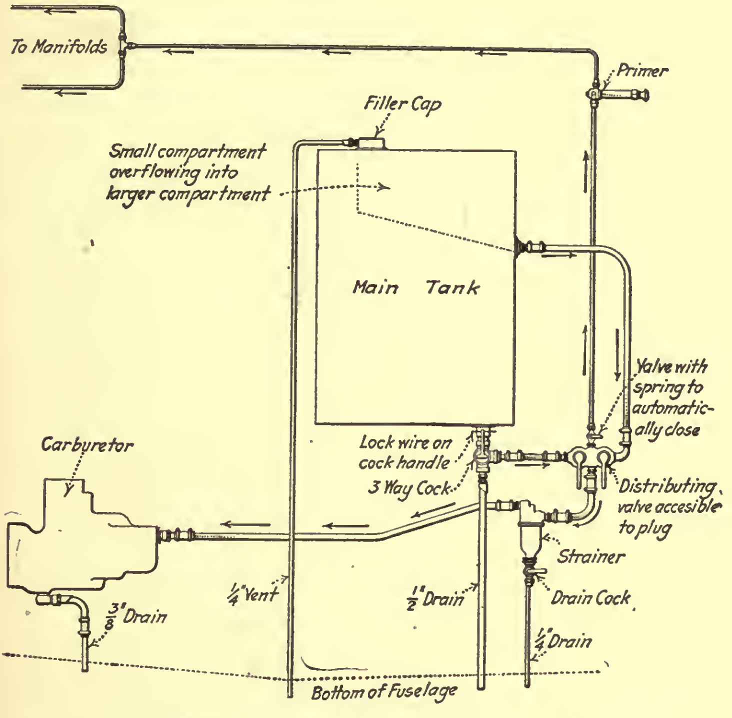


FIG. 221.—Fuel system for a single-engine airplane with gravity feed.

Multi-engine Installations.—When more than one engine is used, each should have its own gasoline system, consisting of pumps, main tanks. gravity or reserve tank, distributing valve and other apparatus required for a single engine system. A cross connection with shut-off valve should be provided so that any engine can take fuel from the tanks of the other engines, and unless two pumping units, not operated by hand, are provided for each engine, a cross connection should be provided so any engine may receive fuel from the pumps of the other engines.

Priming Devices.—A priming system should be installed on every engine, with the priming pump mounted in the cockpit in an accessible position.

Typical arrangements of the fuel system are shown in Figs. 221 and 222. Figure 221 shows a system in which the carburetor is near the bottom of the fuselage so that gravity feed can be employed. The auxiliary tank is most conveniently and simply made a portion of the main tank. A pump system with auxiliary tank incorporated in one of the wings is shown in Fig. 222.

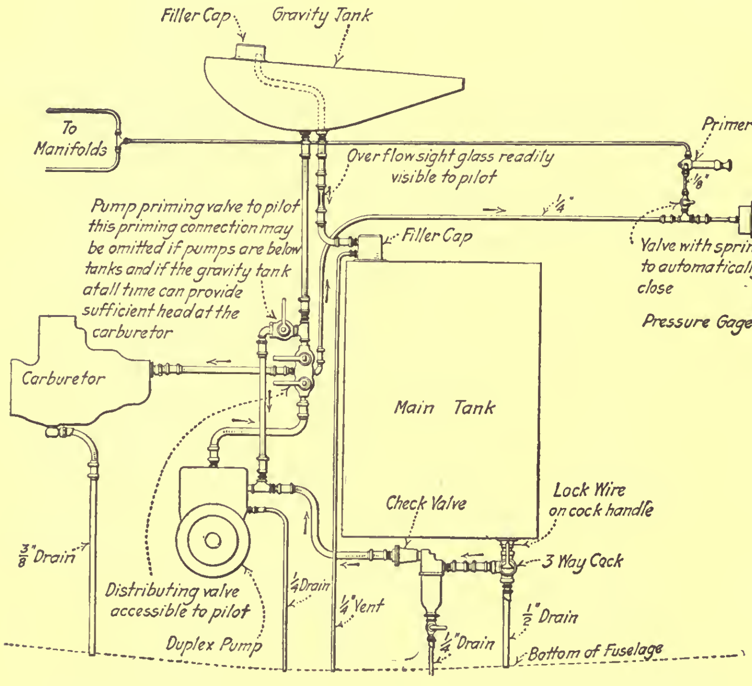


FIG. 222.—Fuel system for a single-engine airplane with pump feed.

Pumps.—A simple form of air pump, used in the Hispano-Suiza engine, is shown in Fig. 223. It is operated by a cam on the camshaft which gives the pump its compression stroke; the return stroke is by the action of the spring. A cup leather on the piston acts as a suction valve on the return stroke. The Mercedes pump (Fig. 224), which is driven from the end of the camshaft, takes in air through ports uncovered by the piston near the end of the suction stroke. A relief valve is incorporated in the pump.

Fuel pumps are made in many forms and are driven either from the engine or by small windmills. Sliding vane and gear

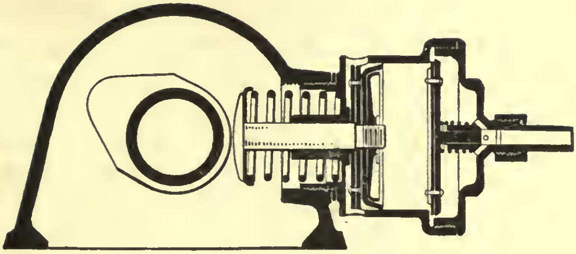


FIG. 223.—Hispano-Suiza air pump.

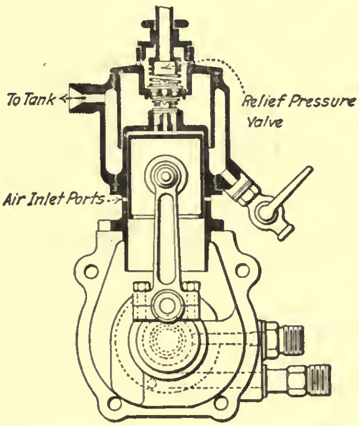


FIG. 224.—Mercedes air pump.

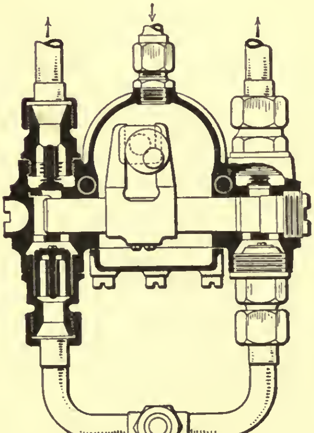


FIG. 225.—Maybach fuel pump.

pumps (see p. 338) are often used and differ from the oil pumps only in smaller capacity. The compact duplex reciprocating pump of the Maybach engine (Fig. 225) is driven from a crank on the end of the oil-pump shaft through a yoke with a sliding bushing. Any leakage of gasoline past the plungers is into the crank chamber, which is filled with lubricating oil under pressure. Another method of avoiding the use of glands past which fuel leakage might occur is the employment of castor oil as the displacing medium. In the Benz engine (Fig. 227) the fuel pump is driven by worm gearing from the end of the inlet camshaft.

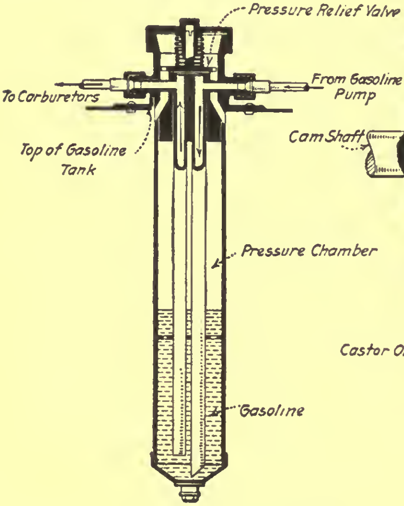


Fig. 226.—Benz pressure reservoir.

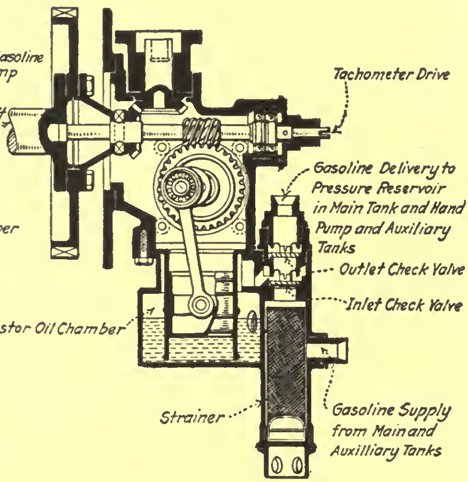


Fig. 227.—Benz fuel pump.

The lower portion of the cylinder is near the bottom of a chamber containing castor oil and the reciprocation of the piston produces a rise and fall of the castor oil in the annular space around the cylinder. The castor oil acts like an annular piston sucking in gasoline as its level falls and discharging it as the level rises. As the speed of the pump is slow (worm-gear reduction 10.75 to 1) it is necessary to keep the discharged gasoline under air pressure during the suction stroke and this is accomplished by the use of a pressure reservoir (Fig. 226) located in the main fuel tank; the pressure reservoir also serves to damp out pressure pulsations.

CHAPTER XII

IGNITION

Ignition is produced by the passage of an electric arc through the explosive mixture, at a time which varies somewhat with operating conditions, but in airplane practice is about 25 to 30 deg. before dead center on the compression stroke. At the operating speed used in airplane engines the moving electrode of the make-and-break system is impracticable. The spark passes between stationary electrodes and is incorporated in "spark plugs" which are screwed into the cylinder head. For the production of the electric arc the following pieces of apparatus are necessary.

1. A **source of electric energy**; this may be a primary or secondary (storage) battery, or more usually, a magneto.

2. As the potential required to cause arcing is very large the low potential current generated in a battery or low-tension magneto has to be transformed into a high-potential current by the use of an **induction coil**; this is usually incorporated in the magneto.

3. The current from a single source has to be sent in succession to each of several cylinders; this is accomplished by the use of a **distributor** which is located in the high-tension circuit.

4. The distributor connects up the circuit to that cylinder in which ignition is next to occur and maintains that connection throughout a short period. The actual timing of the ignition within that period is controlled by a **timer, breaker,** or **interrupter** located in the low-tension circuit.

Other minor but essential elements will be discussed later.

Electric ignition systems utilize electro-magnetic phenomena. An electric current is induced whenever a conductor is moved through a magnetic field or when the magnetic field around a conductor is varied. The intensity of the induced current is proportional to the rate at which the conductor cuts the lines of magnetic force and to the number of coils cutting the lines of force.

The simplest kind of electric ignition system is shown in Fig. 228; B is a source of current, N' a coil of wire surrounding an iron core (forming an electric magnet), S is a switch, timer, or other device for breaking the circuit at any desired moment. The magnetic flux is represented by the arrowed lines. If the switch, S, is opened the current falls to zero and N' is surrounded by a diminishing magnetic field; if S is closed N' is surrounded by a rising field. In both cases self-induction occurs in the coil N' and a current is generated in it, whose magnitude depends on the rate at which the magnetic field through N' changes and on the number of turns in the coil. The phenomena on closing and on opening S are quite different. On closing the circuit, current can flow only after the switch is actually closed and the flow is opposed by the resistance of the circuit (which is small) and the self-induction pressure. When, however, S is opened, an air gap of great resistance is introduced into the circuit with the result that the current diminishes very rapidly and therefore establishes a high electromotive force by self-induction in N'. This electromotive force is sufficient to overcome the resistance of the small air gap formed at the instant of breaking contact and an arc is established across the gap. The resistance of the arc is considerably less than that of the air gap so that the current may continue to flow for a short time across a considerable arc. The more rapid the opening of the gap, the longer will be the arc. The energy for the arc is almost entirely the magnetic flux through the coil N' and is of comparatively small magnitude.

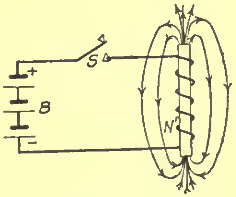


FIG. 228.—Inductance or spark coil ignition system.

If an additional or secondary coil N'' be wound concentric with the primary coil N', as in Fig. 229, the same magnetic changes will occur in both coils and an electromotive force will be generated in N'' which is proportional to the number of turns in the coil. When the number of turns is very large, a high tension, sufficient to jump an air gap such as ab, will be produced. A coil wound as in Fig. 228 is called an **inductance** or **spark coil.** A coil with a double winding as in Fig. 229 is called an **induction coil** or **jump-spark coil.**

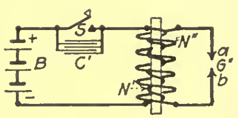


FIG. 229.—High-tension or jump-spark ignition system.

The formation of an arc results in the vaporization or burning of the metal of one of the points between which the arc springs and results in deterioration of that point. To reduce the arcing at S, a **condenser** is shunted around it. The condenser consists of two conductors separated by insulating material; it is usually made of a large number of sheets of very thin metal, such as tin foil, separated by thin paraffined paper sheets. Every other sheet of metal extends to one side and the balance to the other. All the sheets of one side are connected to one terminal and the remainder to another. By connecting the condenser across the switch, S, (Fig. 229) the energy which would otherwise go into the formation of an arc is absorbed in the system.

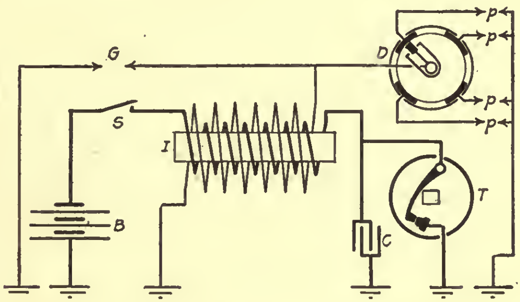


FIG. 230.—Circuit diagram of battery ignition system.

The schematic arrangement of a **battery ignition system** for a four-cylinder engine is shown in Fig. 230. The primary circuit includes the battery, B, switch, S, primary winding on the induction coil, I, the interrupter, breaker, or timer, T, which breaks the primary circuit whenever ignition is required, and the condenser, C, shunted around the timer to prevent arcing. The secondary circuit consists of the secondary winding of the induction coil, the distributor, D, and the spark plugs, p,p,p,p,; a safety spark gap, G, (see p. 310) is shunted on this circuit. The revolving arm of the distributor, D, establishes contacts successively with the four spark plugs in any desired order; the interrupter, T, breaks the primary circuit and the current thereby generated in the secondary circuit arcs across the spark plugs. The circuits are grounded as indicated.

The Magneto.—Most airplane engines at the present day have magnetos as sources of electric current. A magneto differs from a dynamo or electric generator in having permanent magnets in place of electro-magnets for the fields.

In Fig. 231 is shown the action of an armature type magneto, consisting of pole pieces, *N*, *S*, which are permanent magnets, and an armature, *AB*, consisting of core and end pieces, revolving between the shoes of the pole pieces. The clearance ("air gap") between armature end pieces and magnet shoes is only about 0.005 in. A coil is wound on the armature core, one end of the coil being grounded; the other end is carried away, insulated, through a collector ring and brush. As the armature revolves (being driven from the engine shaft) the lines of magnetic force take the successive directions indicated by the long arrows. The magnetic circuit is *NABS* for positions I and II. In the vertical position flux through the core ceases, and no current is generated

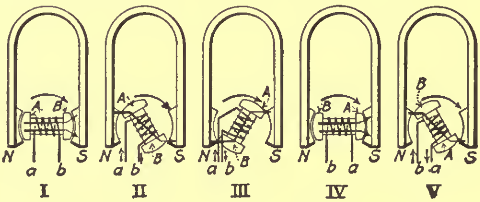


FIG. 231.—Armature type magneto.

n the coil. As the armature passes the vertical position, the circuit reverses to *NBAS*. This continues for 180 deg. more, when the original direction of flow is restored. The strength of the magnetic field influencing the armature coil is greatest at horizontal positions of the armature; but the rate of change of field strength is greater near the vertical positions, where the direction of magnetic flux is reversing itself. The air gap (in a construction like Fig. 231) is then large, so that the maximum effective rate of change occurs shortly after leaving positions II and V. Hence at these positions, twice in every revolution of the armature, the induced current reaches a maximum value, and is capable of producing a vigorous spark.

Figure 232 shows the method of variation of the induced current with magneto position. Starting at position II, Fig. 231, the magnetic flux begins to diminish and has completely reversed itself by the time position III is reached. The duration of this period depends on the width of the armature end pieces. From position III to V there is practically no induced current.

A high-tension magneto differs from that just described in that it has both primary and secondary coils wound on the same armature. Both coils link with the same magnetic circuit and therefore the armature becomes an induction coil and replaces the separate induction coil which would otherwise be necessary. Figure 233 shows a high-tension magneto. The two windings are shown with one terminal grounded to the machine. The primary coil is short-circuited by the contact at the interrupter, at M, until the proper moment, when it is opened suddenly and the induced high-tension current goes through the distributor to one of the spark plugs. The magneto and interrupter must be properly synchronized so that the break occurs when the primary e.m.f. is a maximum. The switch, when closed, short-circuits the primary circuit and thereby prevents the building up of a high-tension current in the secondary circuit, and so shuts off the ignition.

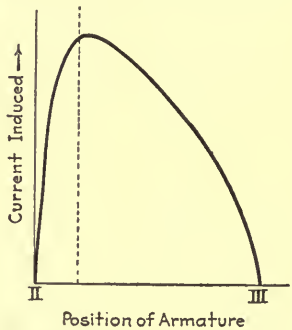


Fig. 232.—Induced current in armature type magneto.

Of the elements shown in Fig. 233 the condenser and interrupter are usually incorporated in the actual construction of the magneto. The distributor may also be incorporated when the magneto speed is one-half the engine speed.

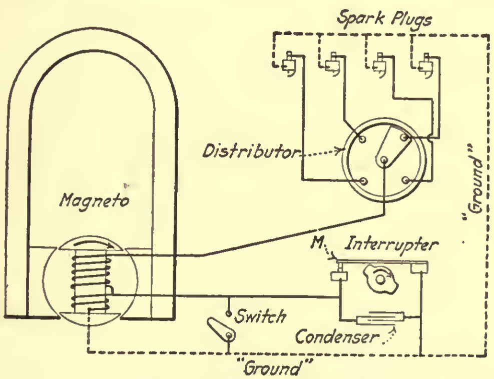


Fig. 233.—Circuit diagram of high-tension magneto ignition system.

The ordinary construction of a magneto with revolving armature gives sparks at 180-deg. intervals corresponding to the positions

(II and V, Fig. 231) of maximum induced current. With Vee type engines it may be necessary to have unequal time intervals between sparks; for example, with a two-cylinder 45-deg. Vee engine the sparks instead of occurring at 180-deg. rotation of the armature should occur alternately at 157½-deg. and 202½-deg. intervals. This unequal interval can be obtained in various ways. In one of the constructions of the Bosch Magneto Co. the armature end-piece is cut away on opposite

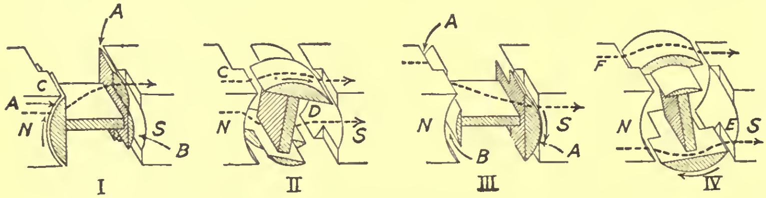


Fig. 234.—Magneto with unequal firing intervals (Bosch).

sides of each half of the core so as to increase the air gap and the tips of the pole shoes are also cut away on diagonally opposite halves of the two poles so as to make the positions of maximum induced current (II, Fig. 231) come earlier. The construction and operation are illustrated in Fig. 234. The large air gap, B, effectively cuts off the lines of force. Maximum induced current will occur shortly after the armature has left the trailing pole tips $C-D$ (position II) and also after the armature has left the trailing pole tips $E-F$ (position IV). These two positions

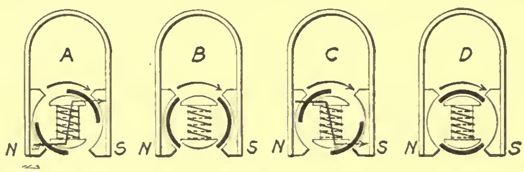


Fig. 235.—Inductor magneto.

are made less than 180 deg. apart as a result of cutting away tips of the pole pieces at E and F.

The magneto with revolving armature has to be provided with insulated moving wires, collector rings, brushes, and moving contacts to convey the induced current from the armature to the stationary conductors. To avoid this complication a **rotor** or **inductor** type of **magneto,** with stationary windings, is often used. Figure 235 shows a construction with a rotating element, or inductor, consisting of two cylindrical segments of soft iron; all the rest of the magneto is stationary. The magnetic condition of the armature core depends on the position of the inductor. In the positions A and C the segments form a magnetic bridge between the magnet poles and the heads of the armature core; in

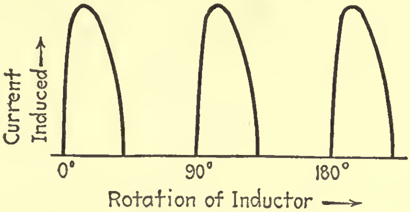


FIG. 236.—Induced current in inductor magneto.

these positions the magnetic flux is a maximum. In passing the positions B and D the magnetic lines are abruptly changed in direction and a vigorous induced current is set up. The reversal takes place four times per revolution of the inductor and succeeding reversals give current in opposite directions. This inductor magneto can give twice as many ignitions per revolution and consequently has to be rotated only half as fast as the rotating armature type of magneto. All the electrical connections are stationary. Typical current curves are shown in Fig. 236.

Another construction of inductor magneto is shown in Fig. 237. The rotor is a steel shaft carrying two laminated soft-iron arms with a space between them which is occupied by the stationary winding (not shown). The arms project on opposite sides of the shaft and are of such radius as to give the smallest practicable air gap between them and the pole shoes. The magnetic flux in the position shown is from N to R, then back along the shaft to the other arm and so to S. When the inductor has rotated 180 deg. from the position shown, the flux will be from N to the rear arm, then forward along the shaft to R and S. The flux through the shaft is reversed twice every revolution and induces a current in the winding around the middle length of the shaft.

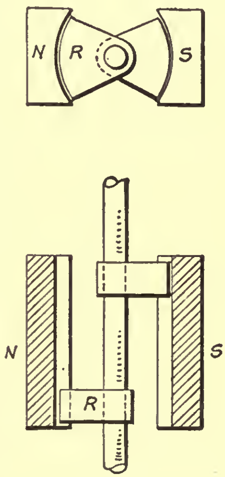


FIG. 237.—Inductor magneto with two arms.

The **Dixie magneto** uses a different type of inductor. The rotor, Fig. 238, consists of two revolving wings, *N* and *S*, separated by a bronze center-piece, *B*. Each wing is always in contact with one pole of the magneto (Fig. 239) and consequently keeps the polarity of that pole. The rotor is surrounded by the field structure, shown in Fig. 240, which carries laminated pole extensions on which the winding with its core is mounted. As the

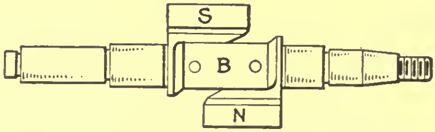


Fig. 238.—Rotating element of Dixie magneto.

rotor revolves the direction of magnetic flux through the core changes twice every revolution.

Construction of Magnetos.—The constructive features of a **Bosch high-tension magneto** of the rotating armature type are shown in Fig. 241. The armature rotates at engine speed and gives two electrical impulses per revolution. The distributor is geared to the contact breaker and rotates at half its speed.

The end of the primary winding is connected to the brass

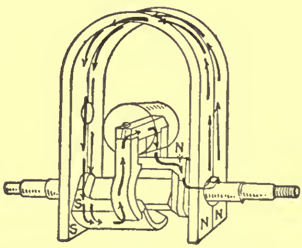


Fig. 239.—Diagrammatic outline of Dixie magneto.

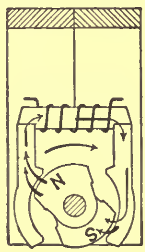


Fig. 240.—Field structure of Dixie magneto.

plate, 1. In the center of this plate is screwed the fastening screw, 2, which serves, in the first place, to hold the contact breaker in its position, and, in the second, to conduct the primary current to the platinum screw block, 3, of the contact breaker. Screw, 2, and screw block, 3, are insulated from the contact breaker disc, 4, which has metallic connection with the armature core. The platinum screw, 5, goes through the screw block, 3. Pressed against this platinum screw by means of a spring, 6, is the contact-breaker lever, 7, which is connected to the armature core and to the beginning of the primary winding. The primary winding is therefore short-circuited as long as lever, 7, is in contact with the platinum screw, 5. The circuit is interrupted

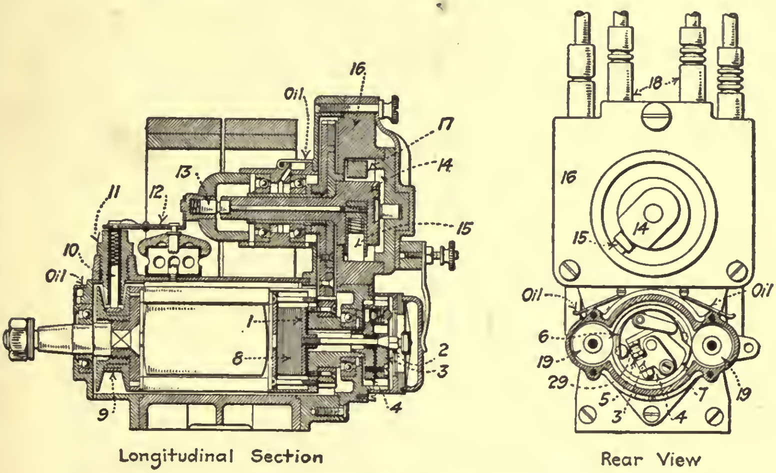


FIG. 241.—Constructive features of Bosch high-tension magneto.

when the lever is rocked. A condenser, 8, is connected in parallel with the gap thus formed.

The end of the secondary winding leads to the slip ring, 9, on which slides a carbon brush, 10, which is insulated from the

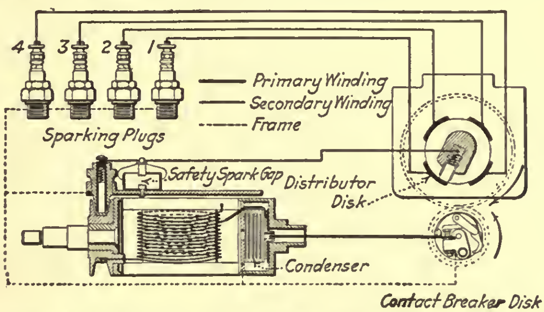


FIG. 242.—Wiring diagram for Bosch high-tension magneto ignition system.

magneto frame by means of the carbon holder, 11. From the brush, 10, the secondary current is conducted to the connecting bridge, 12, fitted with a contact-carbon brush, 13, and through the rotating distributor piece, 14, which carries a distributor carbon, 15, to the distributor disk, 16.

In the distributor disc, 16, are embedded four metal segments, 17. During the rotation of the distributor carbon, 15, the latter makes contact with the respective segments, and connects the secondary current with one of the contacts.

The contact breaker is fitted into the rear end of the armature spindle, which is bored out and provided with a keyway. The short-circuiting and interrupting of the primary circuit is effected by means of the contact-breaker lever, 7, and the fiber rollers, 19. As long as the lever, 7, is pressed against the contact screw, 5, the primary circuit is short-circuited, and the rocking of the levers by the fiber rollers, 19, breaks the primary circuit; at the same moment ignition takes place. The distance between the platinum points, when the lever is lifted on the fiber rollers, must not exceed 0.5 mm. (approximately 1/50 in.). This distance may be adjusted by means of the screw, 5.

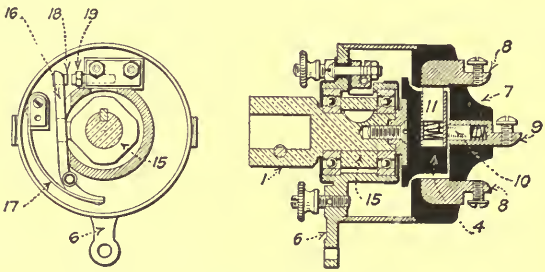


Fig. 243.—Bosch interrupter-distributor.

Another type of Bosch interrupter-distributor is shown in Fig. 243. This is connected at 1 to the engine cam shaft and carries a cam, 15, which has as many lobes as there are cylinders to the engine; four lobes are shown. The interrupter lever, 16, is held against this cam by the spring, 17, and when the rubbing plate of the lever is between two lobes the platinum points, 18 and 19, are in contact; the contact is interrupted at the passage of each lobe and the primary circuit is thereby broken. The distributor rotor is on the same shaft and carries a rectangular brass tube in which is located the carbon brush, 11. This brush sweeps the cylinder cavity in the distributor body, 7, and the contacts, 8, which are as numerous as the cylinders. The central carbon brush, 10, keeps contact with the rectangular brass tube. Adjustment of the interrupter is by rotation of the whole distributor through the timing arm, 6.

Permanent magnets are of steel alloyed with 5 per cent of tungsten or with chromium. All parts at which sparks may occur (breaker, distributor, safety gap) should be enclosed to reduce the fire risk.

The distributor speed is half the engine speed on all stationary four-cycle engines; the distributor may either be incorporated in the magneto or be driven direct by the cam shaft. The **Dixie distributor** for an 8-cylinder engine is shown in Fig. 244. The rotor carries two carbon brushes, which in an 8-cylinder engine are 180 deg. − 45 deg. = 135 deg. apart, and in a 12-cylinder engine are 180 deg. − 30 deg. = 150 deg. apart. These brushes are not in the same plane of revolution. In the plane of the outer brush

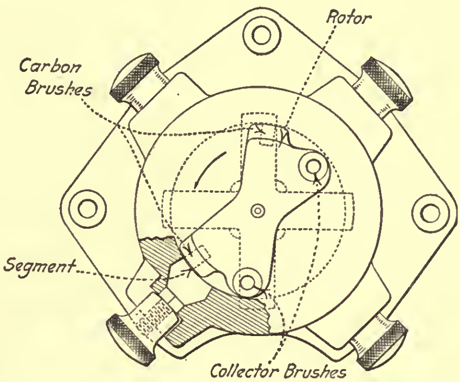


Fig. 244.—Dixie distributor for 8-cylinder engine.

are located four metal segments embedded in the insulating distributor block; the other four segments are located immediately behind the first four in the plane of the second carbon brush. Contacts are made each 45 deg. of rotation of the distributor rotor. The collector brushes are in continuous contact with the secondary circuit and with the carbon brushes.

When rubbing contact is employed in a distributor a deposit of carbon from the brush will be left on the distributor block which must be cleared off periodically. To avoid this a gap distributor is sometimes used with a nickel point and a small air gap (from 0.01 to 0.02 in.) across which the current arcs. The use of a gap distributor has the additional advantage of increasing the secondary voltage when the spark plug has its resistance lowered either by carbon deposit or high temperature, and thereby giving a spark under conditions in which it would otherwise fail (see p. 310).

The **spark advance** in airplane engines is generally fixed at about 30 deg. A slightly greater advance is desirable at high altitudes, but the advantage from its use is so slight and the complication of an additional control so undesirable that spark control is not used in airplane practice. Spark adjustment is obtained by adjusting the breaker. A corresponding adjustment of the magneto is sometimes provided. Figure 245 shows the adjustable driving gear arrangement of the magneto of the King-Bugatti engine. The bevel gear on the magneto shaft is fitted

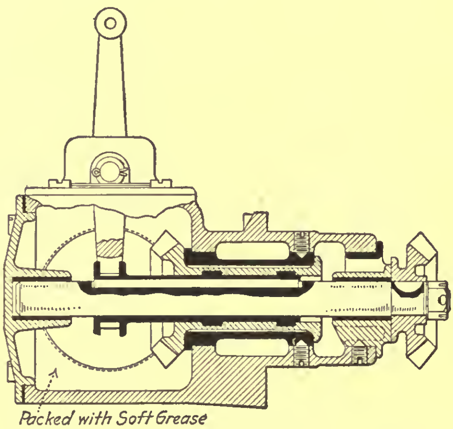


FIG. 245.—Adjustable driving gear for Bugatti engine magneto.

on a taper with a key. The gear has eight keyways, so spaced that the magneto timing may be set within $1\frac{1}{2}$ deg. of any desired position. The gear which adjusts the spark advance has four internal spiral grooves sliding over splines on the sleeve, which is keyed to the driving shaft, but may be moved along the shaft by a lever. Movement of this sleeve revolves the magneto driving gear with relation to the shaft-driving gear.

Adaptation to Engine.—In four-cycle engines having n cylinders, there are $\frac{n}{2}$ sparks necessary per engine revolution. If these are supplied by a magneto giving m sparks per magneto revolution, the ratio of magneto speed to engine speed is $\frac{n}{2m}$. Since m varies from one to four (the interrupter may work only once per revolution even though conditions are right twice), the speed ratio varies from $\frac{n}{2}$ to $\frac{n}{8}$. The great majority of magnetos give two sparks per revolution, and run at $\frac{n}{4}$ times engine speed. A multiple-spark magneto runs at relatively low speed.

In an engine with equal ignition intervals, one magneto may serve any number of cylinders. Thus a two-spark magneto for 15 cylinders would run at 3¾ times engine speed: the same magneto for 9 cylinders runs at 2¼ times engine speed. With unequal ignition intervals, a special magneto (see page 300) or a plurality of magnetos must be employed.

The **cycle of operations of a jump-spark ignition system**[1] can be considered as consisting of five periods. For quantitative values a typical magneto may be considered to have the constants given in the following table.

Constants of Typical Magneto

Primary turns (N_1)	160
Secondary turns (N_2)	8,000
Ratio of turns (n)	50:1
Primary resistance (R_1)	0.5 ohm
Secondary resistance (R_2)	2,500 ohms
Primary inductance (L_1)	0.015 henry
Mutual inductance (M)	0.74 henry
Secondary inductance (L_2)	36 henrys
Primary condenser (C_1)	0.2 microfarad
Secondary (distributed) capacity (C_2)	50 micro-microfarads
Normal speed of operation	2,000 r.p.m.
Primary current at break (I_b)	4 amperes
Maximum current in spark	0.075 amperes
Breakdown voltage of gap	5,000 volts
Sustaining voltage of gap	600 volts

Period I includes the building up of current in the primary winding as a result of either the impressed voltage from a battery or the voltage generated by the rotation of a magneto armature. During this period the breaker or interrupter is closed and the armature rotates from the position of maximum flux to the position where the interrupter opens. For the typical magneto this corresponds to about 100-deg. rotation and lasts 0.008 sec. at 2,000 r.p.m.; the current builds up to 4 amperes. Typical

[1] See Report 58, *5th Annual Report, Nat. Adv. Comm. Aeronautics.*

curves for armature flux are shown in Fig. 246, in which A shows the flux with open primary circuit and is due to the permanent magnetos only. B and C give the total flux under normal operating conditions at 500 and 2,000 r.p.m. respectively.

Period II is the very short period (about 0.00002 sec.) extending from the opening of the interrupter to the breakdown of the spark gap in the engine. During this period, the magnetic energy of the coil is in part transferred into electrostatic energy and charges the condenser and the capacity of the secondary circuit. The primary current flowing into the condenser against

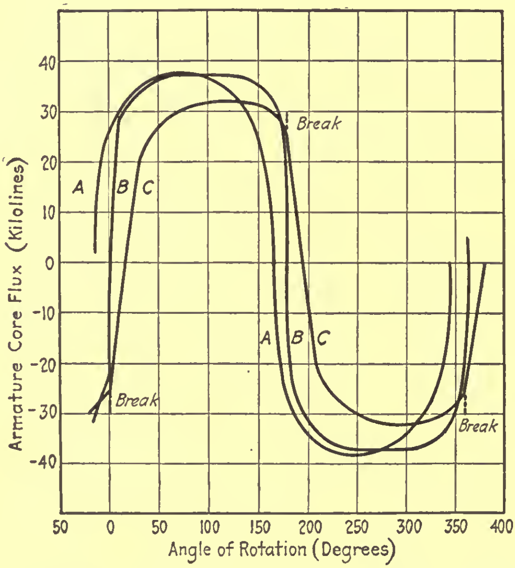


FIG. 246.—Typical curves for armature flux of magneto.

a constantly increasing e.m.f. will decrease at a constantly increasing rate. The decrease in magnetic flux resulting from this decrease of primary current generates an e.m.f. in the secondary windings which in turn sends a charging current into the distributed capacity of the secondary circuit. If the spark gap were not present this process would continue until the magnetic energy had been entirely converted into electrostatic energy; the maximum voltage which would be reached in the typical magneto would be about 70,000 volts. As a result of loss of energy due to resistances, eddy currents in the iron core, etc., this maximum voltage is greatly reduced. The curves of Fig. 247 show the rate of rise of secondary voltage as calculated; (*A*) with no energy loss except that in the resistance of the windings, (*B*) with the usual eddy currents. It is assumed that the spark gap does not break down and there is no arcing at the interrupter points.

Period III is the very short period (about 0.00005 sec.) beginning at the instant at which the spark gap breaks down and lasting until a steady arc is established. When this gap has broken down it affords a conducting path into which the charged secondary capacity discharges. As the secondary is now short-circuited by the arc the current increases rapidly. The energy

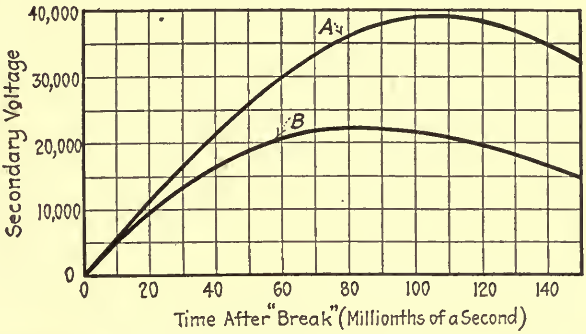


FIG. 247.—Typical curves for secondary voltage of magneto.

discharged into the gap during this time is about 0.002 joule, which is just about sufficient to ignite the explosive mixture (see p. 312).

Period IV extends from the establishment of the gap to the extinction of the spark.. During this time there is a steady discharge which lasts for a considerable time (0.003 sec. for a 5-mm. spark gap in air). The cessation of the arc is due usually to the exhaustion of the energy supply, although occasionally it may be extinguished by the closing of the interrupter if the r.p.m. is very high or the spark gap very short.

Period V covers the remainder of the cycle during which time both circuits are practically free from current.

Of these periods, II is the most important, as it determines whether or not a spark passes at all; the distributed capacity of the secondary circuit is of great moment in determining the maximum voltage in this period. While 5,000 to 6,000 volts is usually required to jump the gap, it may be much increased by oil films on the points and a cold cylinder. If the spark plug is fouled with a conducting film of carbon some of the energy will be drained by leakage during Period II.

A typical oscillograph showing the variations of the primary and secondary circuits is given in Fig. 248. The maximum current delivered to the secondary circuit is usually from 0.05 to 0.10 amperes.

A safety spark gap is sometimes shunted on the secondary circuit (Figs. 230, 241, and 242) to prevent the formation of excessive voltages, and the consequent possible breaking down of the insulation, in case the secondary circuit is open. This would occur when a spark plug is being tested out of the cylinder and is not grounded. The width of the safety gap is from 5⁄16 to 3⁄8 in., the higher value being used for high compression in the engine.

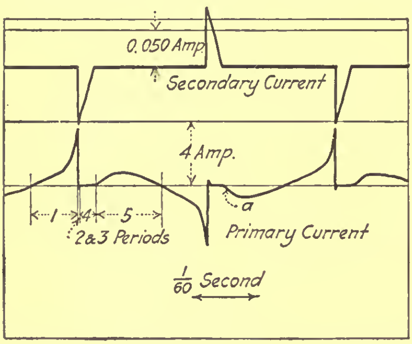


Fig. 248.—Typical oscillograph of primary and secondary currents in magneto.

The air surrounding the safety spark gap becomes ionized and ozone is liberated. If the air is confined the ozone will rust adjacent steel parts and slowly decompose organic insulating materials. A rotary safety gap with one electrode on the gear driving the distributor and the other integral with the distributing metal electrode is used sometimes; the air is churned up and expelled through a suitable gauze window.

A **series** or **subsidiary spark gap** is frequently used in order to maintain sparking even when the spark plugs are fouled. The series gap is placed in the connection between the plug and the magneto.[1] Investigations at the Bureau of Standards show that it is possible, by the use of a series gap, on an average ignition system, to spark a plug having a resistance lowered to only 4,000 ohms by fouling. At least 100,000 ohms insulation resistance is ordinarily necessary at the plug if a series gap is not used. For example, with secondary current limited to 0.08

[1] For elementary theory and results of tests see Report 57, *5th Annual Report, Nat. Adv. Comm. Aeronautics.*

amperes and insulation resistance of 50,000 ohms the maximum voltage across the air gap is $0.08 \times 50{,}000 = 4{,}000$ volts. This is not sufficient; 6,000 volts is usually required.

The efficacy of the series spark gap is well shown in Fig. 249, giving the results of some tests by Young and Warren.[1] The four resistances indicated were put in parallel with the spark plug to simulate different degrees of fouling. With each resistance the length of the main gap was varied while the series gap was kept constant. The curves show the maximum length of the main gap at which sparking occurred. With a parallel resistance of 58,000 ohms the main gap could be increased from 0.9 mm. to 3.4 mm. as the series gap was increased from zero to 0.02 in. The voltage across the main gap was also measured and was found to increase from 2,200 to 3,600 by the introduction of a 0.02-in. series gap, when the shunted resistance was 112,000 ohms.

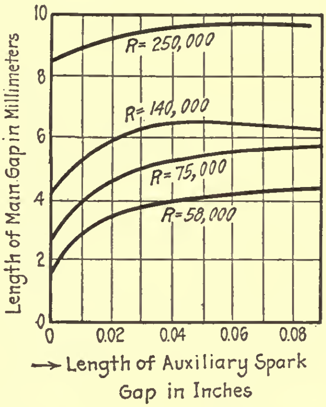


Fig. 249.—Effect of auxiliary spark gap on length of main gap.

The series gap is sometimes integral with the plug; sometimes at the plug but not integral with it; sometimes in the distributor. The amount of the gap should be variable to suit the degree of fouling of the plug. For maximum effectiveness the gap should be at, or integral with, the spark plug. These desiderata are conflicting. It is not practicable to adjust subsidiary spark gaps at each spark plug, while the engine is operating; it is easily possible to have a single adjustable spark gap at the distributor, but it cannot be adjusted to suit the different degrees of fouling in the different cylinders which it serves.

The effect of temperature and pressure on sparking voltage has been investigated at the Bureau of Standards.[2] Sparking voltage is a linear function of the density of the gas and depends on pressure and temperature only as they affect the density. For a typical spark plug with 0.5 mm. gap the sparking voltage

[1] "The Process of Ignition," *The Automobile Engineer*, March, 1920.

[2] See Report 54, *5th Annual Report, National Advisory Committee on Aeronautics.*

in air varies from 2,800 volts at atmospheric density to 9,400 volts at a density five times as great. Other measurements indicate that the sparking voltage in an explosive mixture of gasoline in air is about 10 per cent less than in pure air and that the change in voltage is proportional to the percentage of gasoline present. Figure 250 shows the observed sparking voltage for plugs with different gaps; No. 1, 1.8 mm. (0.071 in.); No. 2, 1.2 mm. (0.047 in.); No. 3, 2.2 mm. (0.086 in.); No. 4, 0.5 mm. (0.020 in.). The voltage required for a spark plug set at 0.5 mm., in an aviation engine of moderate compression, is about 6,000 volts.

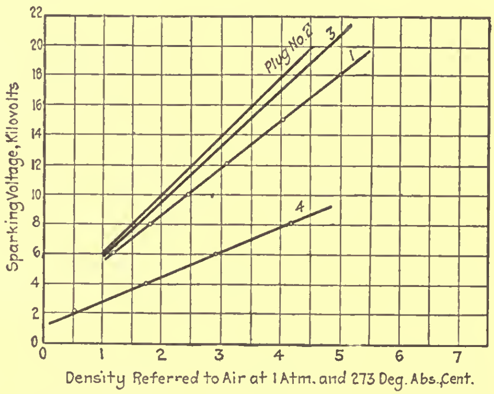


FIG. 250.—Sparking voltages for plugs with different gaps at various air densities.

The sparking voltage is not affected appreciably by the material of the electrodes but is diminished by the use of finer points. The dimensions of the points are, however, determined by considerations of mechanical strength and durability.

The "fatness" of a spark has no influence at all on the power developed in an engine. If the current is sufficient to charge the plug and its connections to the sparking potential, the maximum engine power will be developed. The energy represented by that condition is usually about 0.002 joule; the energy per spark varies from 0.03 joule in battery systems up to 0.16 joule in the more powerful magnetos. The excess of energy above that necessary for ignition has no discernible effect on the power developed.

Battery Ignition.—In a **storage cell,** electrical energy is stored as chemical energy but returns to electrical energy when the cell is connected to supply an external circuit. The desired voltage is obtained by connecting a sufficient number of cells in series, forming a storage battery.

The chemical reaction in a lead cell may be expressed by the following equation:

$$\underset{\text{Positive plate}}{PbO_2} + \underset{\text{Negative plate}}{Pb} + \overset{\overset{\text{Charge}}{\longleftarrow}}{\underset{\underset{\text{Discharge}}{\longrightarrow}}{\underset{\text{Electrolyte}}{2H_2SO_4}}} = \underset{\text{Positive and negative plates}}{2PbSO_4} + \underset{\text{Electrolyte}}{2H_2O}$$

Discharge results in the formation of lead sulphate; charging restores the plates to their original conditions of lead sponge (negative) and lead peroxide (positive).

The voltage of a fully charged idle cell is 2.05 to 2.10 volts. Discharge lowers the voltage in proportion to the current flowing. Complete discharge is reached at 1.7 volts, at the normal discharge rate fixed by the manufacturer. The capacity of a battery is expressed in ampere-hours at normal discharge rate; the capacity increases as the discharge rate decreases. The maximum discharge rate falls as the temperature decreases.

The acid or electrolyte is an aqueous solution of density 1.255 resulting from the addition of 1 part of sulphuric acid of sp. gr. 1.84 to 4⅓ parts (by volume) of distilled water. The strength and density of the solution fall as discharge progresses; when the density falls below 1.2 the cell needs recharging.

Self-sustaining battery systems require a generator for recharging. The system may then be regarded as a generator system on which the battery "floats." The generator furnishes low-tension direct current, and must have a commutator and brushes. In starting, or at low speed (up to 650 r.p.m.), ignition current comes from the battery. At some definite speed, say 650 r.p.m. of the engine, the generator begins to supply the ignition current. Its rate of delivery is then considerably in excess of that needed for ignition and the surplus goes to the battery. The recharging rate has a maximum value of 10 amperes when the battery is nearly discharged. The delivery voltage of the generator may be automatically controlled by a potential regulator.

The outlines of the **Liberty engine battery ignition** system are shown in Fig. 251. Figure 252 shows the circuit diagram. It includes a low-voltage generator in connection with a storage

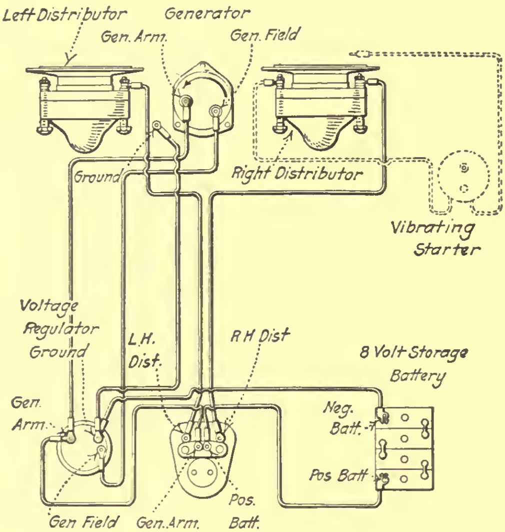


Fig. 251.—Liberty engine ignition system.

battery of light weight and small liquid content. Through a switch, the current is sent to two distributors which are mounted

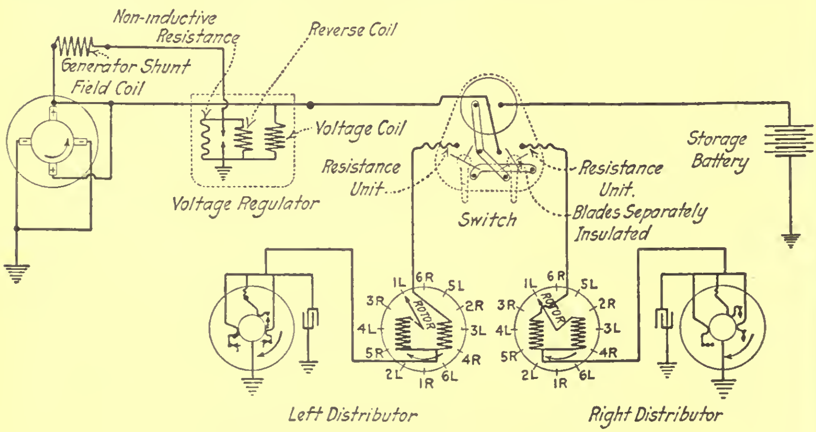


Fig. 252.—Liberty engine circuit diagram.

on the camshaft housings and are direct-driven by the camshafts. Right-hand distributors supply distributor-end plugs and left-hand distributors supply propeller-end plugs, there being two plugs to each cylinder. The entire system, exclusive of plugs and wiring, weighs 35 lb., and consists of generator, battery, switch, voltage regulator, and two distributors.

The **generator** is shown in Fig. 253. It is four-pole, shunt-wound, $4\frac{1}{8}$ in. diameter, $8\frac{1}{2}$ in. high, and weighs $11\frac{1}{4}$ lb. It is mounted with its shaft vertical on top of the crankcase between the two rows of cylinders at the rear end. The armature shaft extends downward into the crankcase, where it is driven by the auxiliary gearing which also drives the vertical shaft. Its speed is 1.5 crankshaft speed. The end housings of the generator field frame are of cast aluminum. Gearing from the armature shaft forms the tachometer drive. The upper housing contains the ball bearing for the shaft, which is the only bearing in the generator proper. This housing also supports the four brush holders: two positive brushes, insulated, and two negative, grounded to the frame.

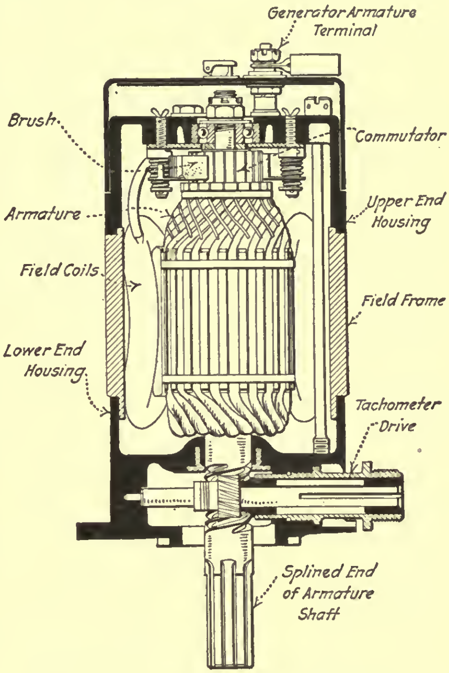


FIG. 253.—Liberty engine generator.

The ground side of the field is grounded through the voltage regulator, (Fig. 252), the generator voltage being determined by the amount of current flowing through the field, which in turn is controlled by the regulator. The armature has 21 slots and is wave-wound. Insulation between commutator segments is slotted down $\frac{1}{32}$ in. below the surface of the copper bars. The shaft is hollow, and ground through the bearing. The maximum generator voltage is 10 to $10\frac{1}{2}$ volts. A current of 5 to 6 amperes may be carried without overheating.

The **voltage regulator** keeps the voltage constant at all speeds above 650 r.p.m. of the engine. It weighs 1½ lb. and is mounted in a cast aluminum cup on the back of the dash behind the switch. It consists of a soft-iron core over which a pivoted iron armature is so mounted as to be normally held away from the core by an adjustable tension spring. When so held, the generator field current passes through a tungsten contact point on the armature to the ground.

The core carries three windings (Fig. 252). The **voltage winding** is of fine wire leading from the positive terminal of the generator armature to the ground. The generator voltage is impressed on this winding. Increase in this voltage increases the core magnetism and opens the contact gap of the regulator armature. This cuts off the direct flow of field current and decreases the armature voltage of the generator. The **reverse winding** is superimposed on the voltage winding and is also of fine wire, wound in a reverse direction. The **non-inductive winding** consists of resistance wire wound so as to produce no magnetic effect on the core and so as to be itself free from induction due to changing core-flux. These two windings are connected in parallel from the regulator armature contact point to the ground: they form a permanent high-resistance ground for the field current when the contact is open. The reverse winding rapidly destroys residual magnetism and enables the spring again to close the contacts, which in regular operation vibrate rapidly.

The **battery** is designed for light weight and no leakage. It is 7 by 4 by 5½ in. and weighs 10¼ lb. It can provide 3 amperes for 3 hr., which is sufficient energy for dual ignition on 12 cylinders. It floats on the line and normally supplies current only when the engine speed is under 650 r.p.m. The ammeter shows whether the battery is charging or discharging. Charging is automatic at speeds above 650 r.p.m. The hard-rubber battery jar has four compartments or cells, each cell (Fig. 254) containing 3+ and 4− plates, burned to connecting straps and separated by perforated rubber with wood. Plates are 3 by 3 in., and rest on ⅜-in. bottom ribs. Above the top of each plate is a flat sealing or baffle plate of hard rubber. The top of each cell is further sealed by a rubber cap through which the lead terminal posts extend. These also are sealed by gaskets or by burning.

In normal position, the electrolyte completely fills the plate compartments. When the battery is turned upside down, the electrolyte seeps through small holes to the compartment which is normally above the plates. This compartment is of such capacity as to hold all of the fluid in the cell, without danger of overflow at the vent plug.

The **switch**, Fig. 252, is located on the dash and weighs 1 lb. It is built on a Bakelite base. The circuits are controlled by

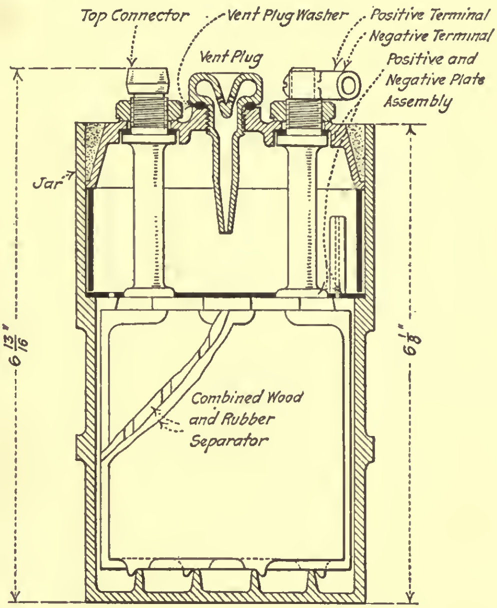


FIG. 254.—Section of storage cell.

two aluminum switch-levers operating spring-bronze contact fingers which connect with contacts molded in the base. The left lever supplies the left distributor, the right lever the right distributor. The engine is started on one distributor, and both levers are switched on only when the speed is above 650 r.p.m. (To start on both distributors would require that the battery supply two sets of plugs and would also waste battery current through the generator.) Resistance coils are mounted on the back of the switch in series with the distributor circuits. These prevent an excessive flow of current should the switch be left on with the engine idle. The switch has four external connections:

positive battery, generator armature, and two to distributors.

Two 12-cylinder **distributors** (Fig. 255) are used, each supplying one plug in all cylinders. Each weighs 5½ lb. and is 7¾ in. diameter by 5⅝ in. high. They are mounted one on each of the overhead camshafts. The transformer coils and breaker mechanism are incorporated. The Bakelite distributor head forms a cover for the breaker mechanism and a seal for the coil. The moving contact is through a soft carbon brush bearing on terminals molded in a hard-rubber track.

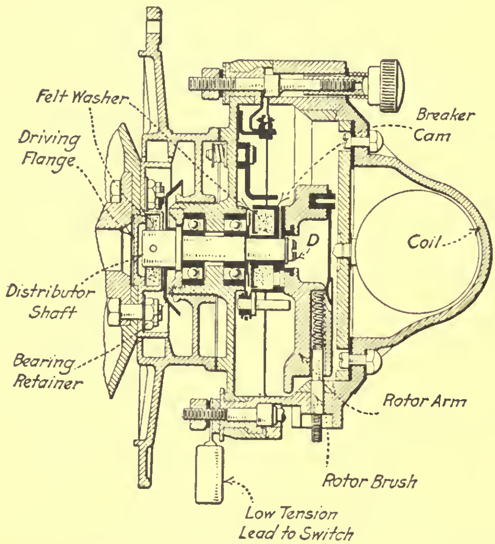


FIG. 255.—Liberty engine distributor.

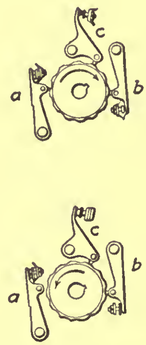


FIG. 256.—Liberty engine breaker mechanism.

The breaker mechanism is operated by a 12-lobed cam, having lobes spaced 22½ deg. and 37½ deg. (12-cylinder engine). Tungsten contact points are used. Two main-circuit breakers *a* and *b*, Fig. 256, connected in parallel, are provided and are timed to operate simultaneously; the duplication is a precautionary measure. The auxiliary-circuit breaker, *c*, is provided to prevent the production of a spark when the engine is "rocked" or turned backward. This auxiliary breaker is connected in parallel with the other two through a resistance unit (Fig. 252) which reduces the amount of current flowing through it and is so timed that it opens slightly *before* the other two when the engine is turned in a forward direction. The opening of the main breakers then results in the production of a spark. When the engine is turned in a *backward* direction the two main breakers open first and no spark is produced due to the fact that the current continues to flow through the coil through the auxiliary breaker but in diminished quantity due to the resistance unit. By the time the circuit is opened at the auxiliary breaker the intensity of the magnetic field of the coil has weakened to such an extent that no spark is produced. The whole breaker mechanism may be revolved to advance or retard the spark.

Spark Plugs.—The conditions to which the spark plug is subjected in aviation engines are difficult to meet. The requirements are:

1. The maintenance of a gap having a breakdown voltage of about 6,000 volts.
2. The maintenance of an insulation resistance of at least 100,000 ohms.
3. Practically complete gas tightness.

These conditions must be maintained under pressures of 500 to 600 lb. per square inch while immersed in a medium which alternates, 15 cycles per second, in temperature between 50° and 2,500° C. and in an atmosphere which tends to deposit soot, and possibly oil, on the surface of the insulator. The inner end of the insulator and central electrode may have an average temperature of 900°C. while the body of the insulator, well up in the shell, is in contact with a jacket containing water at 70°C. For successful operation the insulating surface must remain clean, the insulator must not fracture or disintegrate under the varying temperatures and no part of the plug must become hot enough to cause preignitions of the charge.

The shell is of steel with standard thread. The S. A. E. standard plug dimensions are: outside diameter, 18.2 mm.; pitch diameter, 17.22 mm. ±0.02 mm.; root diameter, 16.09 mm.; 16.9 threads per inch; 1.5 mm. pitch. In order to keep down its temperature it is often made with fins for radiating heat. Investigations at the Bureau of Standards[1] have shown that brass shells average from 90° to 270°F. hotter than steel shells.

The most common insulating materials are mica and porcelain. Other materials used are fused quartz, steatite and molded materials such as Bakelite and Condensite. Porcelain of the highest grade is an excellent material except for its brittleness and conse-

[1] Report 52, *5th Annual Report, Nat. Adv. Comm. for Aeronautics.*

quent liability to fracture either from temperature or mechanical effects. Mica, while free from this trouble, is more likely to foul in consequence of its rough surface. Fused quartz is free from both the above objections.

The insulation resistance of these materials diminishes with rise in temperature. At a temperature of 900°F. the order of merit of the insulating materials is mica, quartz, steatite, porcelain. Porcelain plugs which show a resistance of millions of ohms when cold may have a resistance of only 100,000 ohms at 900°F.

The most common source of failure in spark plugs is fouling.[1] It causes more than 50 per cent of spark plug troubles and is most serious at high altitudes. Fouling is due to the deposit of a layer of carbon and causes a short circuit. The **carbon deposit** results from either or both of two causes: (*a*) The chilling of the flame by a cool portion of the plug and the consequent incomplete combustion; this effect is particularly common when the mixture is overrich and is of frequent occurrence at high altitudes with an imperfectly compensated carburetor. (*b*) The decomposition of lubricating oil which is splashed on heated portions of the insulator. The oil itself is an insulator and when it wets a layer of soot in the plug it makes it an insulating layer. Such a deposit chars and becomes more and more conducting. The oil acts as a binding material and also increases the rate of deposition of soot since the carbon particles in the flame adhere to it readily. The conduction through the deposit seems to take place through a narrow path where the oil film between the particles has been broken down by electric stress. Such fouling causes misfiring.

A method of attempting to reduce the deposit of carbon is to keep the insulation at such high temperature that all carbon deposit is burned off. This can be accomplished by making the insulator with petticoats, ridges or other projections, but there results the danger of preignition, particularly in high compression engines or in those that are not well cooled. Another method is to shield the insulator with a metal baffle plate which protects it from oil spray, but if this is done the flame does not get good access to the insulator and any deposit which has formed will have very little opportunity of being burned away. The use of a series gap (p. 310) is useful in maintaining firing after the plug is fouled.

[1] Report 52, *5th Annual Report, Nat. Adv. Comm. for Aeronautics.*

Fouling with oil, either in the form of a surface film over the electrodes or as a drop between the points, will often prevent firing. The breakdown strength of oil is several times that of air, and the voltage required may easily exceed that which the ignition system is capable of delivering. The trouble is intensified if the insulation of the plug is at the same time reduced by a layer of soot, thereby diminishing the maximum voltage which the system can develop.

The oil trouble usually occurs on starting but may also be met when the plane is recovering from a long glide during which the engine is turning over slowly and pumping oil into the cylinders. It may sometimes be identified by the sparking of the safety gap (see p. 310). It seems to occur most often when the form of the electrodes is such as to drain the oil away by capillary forces. The only real remedy is to keep down the amount of lubricating oil going to the cylinders at starting and during glides.

Cracking the insulator is one of the most common causes of spark plug failure. The thickness of the insulator is usually so great that a clear crack may not interfere with ignition, but after a while the cracks become filled with carbon and form a conducting path.

Cracking may result from several causes. The high temperature gradient from the hotter inner end of the insulator to the relatively cold shell and the consequent unequal expansion is a frequent cause. Such cracks are most likely to occur at a shoulder or other place where there is a sudden change in diameter. Cracking may also occur if the metal parts of the plug are so arranged that their relatively greater expansion produces pressure on the insulator. The mechanical vibration of the engine as a whole may break the porcelain; such breakage often occurs in the outer portion of the porcelain. There is also considerable breakage from accidental mechanical injury such as striking the plug with a wrench.

Mica plugs are free from this trouble. If porcelain is used it should combine high mechanical strength, low modulus of elasticity, low coefficient of thermal expansion and high thermal conductivity. The porcelain may also be made in two or more pieces-permitting the innermost porcelain to heat and expand considerably, while the outer pieces are cooler. The passage of a spark through the joint between the pieces is prevented by a wrapping of mica around both the shell and the electrode. It is very difficult to make such plugs gas-tight.

In plugs in which the central electrode is cemented in the porcelain, the differential expansion of the two can be taken care of only if the electrode is kept of small diameter.

Breakage by mechanical vibration can be reduced if the insulation is cushioned by a considerable thickness of asbestos or other packing material between the shoulder of the insulator and the bushing. One-piece plugs in which the edge of the shell is crimped over the shoulder of the insulator are especially liable to cracking at the edge of the shell.

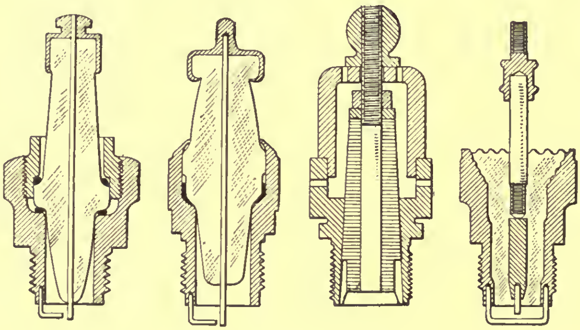


FIG. 257.—Types of spark plug construction.

A minor cause of failure is the change in the width of the spark gap either through warping of the wires or corrosion. Warping occurs only when the wires are relatively long. Corrosion is very slow with the alloy commonly used (Ni 97 per cent, Mn 3 per cent), although a chemical reaction between the cement and the metal of the electrode may sometimes cause the tip to drop off.

Gas leakage is an evil in that it causes a rapid heating of the plug if its amount is considerable; such leakage is usually a matter of workmanship rather than of design of the plug. There are two joints to keep tight, that between the central electrode and the insulator, and that between the insulator and the shell.

The general methods of **construction** are shown in Fig. 257. In the *screw bushing, a,* the insulator has a shoulder, one side of which is seated on a shoulder in the shell while a bushing is screwed down inside the shell on the opposite side. A gasket of brass, copper, asbestos, or some soft heat-resisting material is used and can be placed on either side of the shoulder of the insulator, or on both. With a mica insulator it is possible to dispense with the gasket. To relieve the insulator from the mechanical strain resulting from differential expansion of the shell and the insulator such constructions as those shown diagrammatically in Fig. 258 may be used. In *a* the shell *A* and sleeve *B* are made of different metals and *B* is of such length as to maintain constant pressure on the washer. The expansion of the central electrode is compensated in a similar manner. In *b* the steel clamping nut is very thin and flexible. Expansion of the central electrode is provided for by a strong spring washer under the nut *B*.

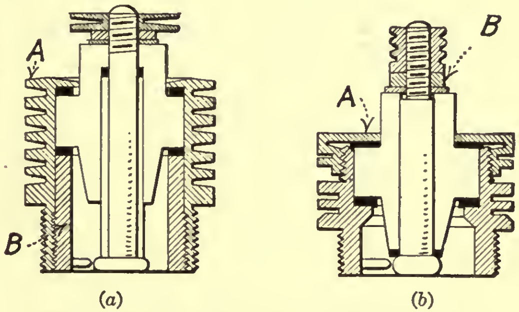


FIG. 258.—Special spark plug constructions.

The *crimped shell*, *b* (Fig. 257), is most common (Champion, Titan, etc.) and is formed by forcing the top edge of the shell over a gasket, which rests on the upper side of the shoulder of the insulator. These plugs cannot be disassembled.

The *taper fit*, *c* (Splitdorf), is used with a mica or steatite insulator. The mica does not stand well the pressure exerted on it during assembly. If a thin steel jacket is placed over the taper it protects the mica and is flexible enough to form a gas-tight fit with the shell.

A *molded-in* insulator, *d* (Anderson), consists of glass which has been forced between the central electrode and the shell while in the molten state. It adheres to both electrode and shell and is gas-tight.

The shape and arrangement of the electrodes seem to have little effect on the operation with the exceptions already noted of the greater liability to fouling with oil of plugs in which the side wall of the shell forms one of the electrodes. The variation in breakdown voltage with the shape of the tips is slight. With fine wires any oil film at starting is burned off rapidly but there is greater liability to preignition. With a central electrode consisting of a disc (Fig. 257*c*) the danger of short-circuiting with carbon is increased while the likelihood of complete fouling with oil is diminished and greater protection is afforded to the insulating material back of it.

The location and number of spark plugs used are important in their effect on engine performance. An effort should be made to reduce as much as possible the distance through which the flame has to be propagated. The greater that distance, the greater is the liability to detonation. If the distance is shortened, a higher compression may be employed without detonation. Where a single plug is used its location should be in the center of the head. Multiple spark plugs are desirable, not only to insure ignition in case of failure of one plug but also because they permit higher compression pressures. With a constant compression pressure the power is increased by increasing the number of plugs; for example, in a 5½ by 6½-in. four-valve single-cylinder test engine with a compression ratio of 5.4 and at 1,800 r.p.m., the brake horse power was increased 5.7 per cent by the use of two spark plugs and 11.1 per cent by the use of four plugs.

A typical wiring diagram is shown in Fig. 259, which shows the wiring of a 12-cylinder Vee engine equipped with two magnetos for regular operation, and a starting magneto. The regular magnetos also have radio connection.

The **relative advantages of battery and magneto ignition** have been much debated. The performance of the engine is not affected by the source of primary current. With battery ignition the engine is started by turning it with the current on but fully retarded. With magneto ignition the engine is pulled over a few turns with no current on so as to fill the cylinder with an explosive mixture, and a starting magneto is then operated by hand, giving a shower of sparks in one cylinder; the starting magneto is arranged to be fully retarded. The element of danger in starting the engine is eliminated by this latter method of operation. The battery requires more attention than the magneto, particularly if the engine is to stand idle for a while; if it runs down there remains no means for starting the engine.

The battery system is also more complicated than the magneto system but it lends itself better to irregular explosion intervals; the magneto must be of special type in this case (see p. 300). The generator in the battery system requires attention to keep commutator and brushes in good condition; there is no corresponding attention necessary with the magneto. The total weight of the magneto system, including dual magnetos and a starting magneto, is somewhat greater than that of a battery

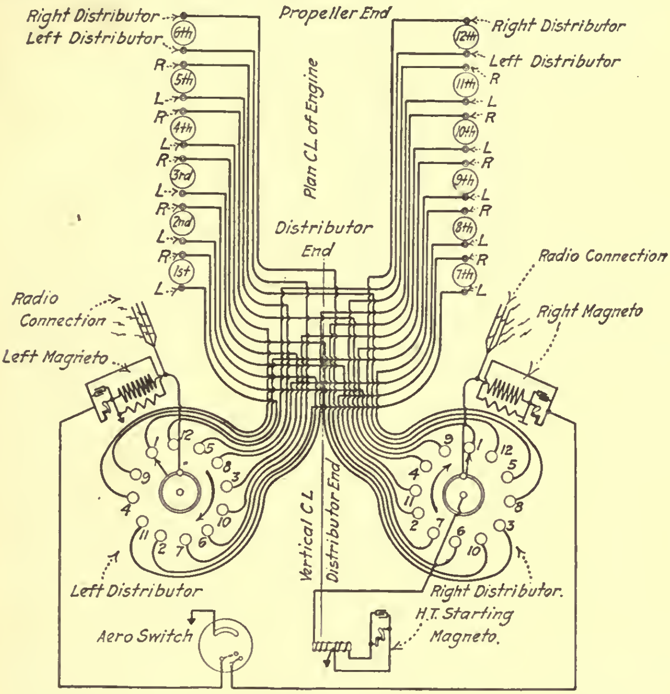


FIG. 259.—Typical wiring system for 12-cylinder Vee engine.

system. A magneto system is easier for the pilot to operate; there is only one switch handle to control and no ammeter to be watched. The battery system has distinct advantage when current is required for an electric starter, lights and other uses.

The **firing order** adopted in actual engines is given below.

8-cylinder 90-deg. Vee	1L, 1R, 2L, 2R, 4L, 4R, 3L, 3R (Curtis OX5, VX; Sunbeam Arab)
	1L, 4R, 2L, 3R, 4L, 1R, 3L, 2R (Hispano-Suiza) (counting from propeller)

12-cylinder 45-deg. Vee: 1L, 6R, 5L, 2R, 3L, 4R, 6L, 1R, 2L, 5R, 4L, 3R (Liberty) (counting toward propeller).

12-cylinder 60-deg. Vee:
- 1L, 2R, 5L, 4R, 3L, 1R, 6L, 5R, 2L, 3R, 4L, 6R (Rolls-Royce Falcon and Eagle)
- 1L, 1R, 5L, 5R, 3L, 3R, 6L, 6R, 2L, 2R, 4L, 4R (Sunbeam Maori, Cossack)

9-cylinder Rotary: 1, 3, 5, 7, 9, 2, 4, 6, 8 (Gnome, LeRhone, BR1, BR2, Clerget)

6-cylinder Vertical: 1, 5, 3, 6, 2, 4 (Beardmore, Galloway, Siddeley, Austro-Daimler, Mercedes, Fiat)

7-cylinder Radial: 1, 3, 5, 7, 2, 4, 6 (A. B. C. Wasp)

9-cylinder Radial: 1, 3, 5, 7, 9, 2, 4, 6, 8 (A. B. C. Dragonfly)

CHAPTER XIII

LUBRICATION

All rubbing surfaces in an engine should be lubricated. The most important of these surfaces are the cylinder walls, main bearings, crankpins, piston pins, and camshaft bearings, but there are also numerous other parts to be lubricated.

The coefficient of friction of a bearing with good lubrication, moderate pressures and high speeds is practically independent of the materials composing the rubbing surfaces, but is proportional to the viscosity of the oil, to the rubbing speed and to the area; it is independent of the pressure. For high pressures and low speeds these laws do not hold; for velocities from 100 to 500 ft. per minute the coefficient decreases about as the square root of the velocity, for velocities from 500 to 1,600 ft. per minute it decreases about as the fifth root of the velocities, while above 1,600 ft. per minute it is practically constant. With high pressures the coefficient of friction increases.

In an airplane engine the loads on the principal bearing surfaces are variable, going through a cycle of changes every two revolutions of the engine, and varying from a maximum to a low minimum. For example, in the Liberty engine the total force on the crankpin varies from 4,980 to 1,500 lb.; on the intermediate main bearing from 7,250 lb. to 800 lb.; on the end main bearing from 4,025 lb. to zero; on the center main bearing from 7,700 lb. to 2,500 lb.; the piston side thrust from 930 lb. to zero. Furthermore, the direction of the force changes in these principal bearing surfaces, so that the portion of the bearing which at one instant is supporting maximum pressure is later relieved of all pressure. This intermittent application of the load is favorable to good lubrication and permits the use of maximum pressures greatly in excess of what would be possible with continuous loading. The oil film which is squeezed out by the application of the maximum pressure is replaced during the reduction or reversal of the pressure.

The **maximum load** per square inch of projected area is greatest on the piston pin, which has a diameter considerably less than the crankpin; in the Liberty engine this is 2,580 lb. per square inch as against 932 lb. on the crankpin; 1,675 lb. on the center main bearings; 1,580 lb. on the intermediate main bearings; 815 lb. per square inch on the end main bearings.

The **rubbing speeds** of the main bearings of airplane engines range usually from 16 to 20 ft. per second; the rubbing speeds at the crankpins will be somewhat lower.

The **total friction work** at any bearing is proportional to the product of the mean total load by the rubbing speed. The limiting factors for a bearing for continuous operation are the mean pressure per square inch of projected area and the rubbing speed. The product of these two is a good index of the service of the bearing. In the Liberty engine this load index is 13,500 lb.-ft. sec. for the crankpin; 24,670 for the center main bearing; 13,650 for the intermediate main bearing; and 11,900 for the end main bearings.

The permissible pressure on the bearing depends on the viscosity and therefore on the temperature of the oil. The temperature tends to rise and must be kept down by oil cooling. Temperatures of 160°F. and higher are common.

The **lubrication of the cylinder** offers problems quite different from the lubrication of the rest of the engine. The side thrust pressures are moderate; the piston speed is high, reaching 2,000 ft. per minute, or 33 ft. per second, and the maximum speed (at mid-stroke) is about 52 ft. per second. The friction work under these conditions would not be a serious charge against the engine if the oil film could be maintained in good condition, but as pointed out on page 24 the viscosity of the oil film on the cylinder walls is greatly raised by carbonization. The oil film on the walls and around the piston rings has to serve another purpose besides acting as a lubricant; it acts as a seal to prevent the blowing of the gases past the piston. For this purpose high viscosity is useful.

In starting cold, in idling with overrich mixtures, and in cold weather, a certain amount of liquid fuel will meet the cylinder walls and, being perfectly miscible with the mineral lubricating oil, it will dilute the oil film and the thinned oil will then run down the cylinder walls and dilute the oil in the crankcase. This phenomenon, which is very common in automobile practice, is not so usual in airplane engines because of the higher volatility of the aviation fuels. The ordinary gasoline of commerce has a high end point, which means that it has kerosene constituents which will not vaporize during admission and will be deposited in the liquid form on the cylinder walls. With aviation fuel the dilution when it occurs is by more volatile elements which tend to vaporize out from the hot body of oil so that the dilution of the crankcase oil is not cumulative as with automobile engines.

The friction at the bearings of an engine results in heat which has to be taken away as fast as it is generated if the bearings are not to rise in temperature. Much of this heat is conducted through the metal to cooler parts of the engine but part is carried away by the oil itself. For this purpose a large flow of oil is desirable. The amount of oil circulated in the Liberty engine is about 12 gal. per minute and the temperature rise may average about 10°F. This corresponds to a heat abstraction of about 45 B.t.u. per minute. If the bearing friction work is taken as 1½ lb. per square inch of piston area (see p. 24), the corresponding heat generated will be about 200 B.t.u. per minute. The volume of oil circulated in this case is not sufficient for complete cooling of the bearings.

Viscosity.—The oil which gives minimum temperature rise of the bearing is the best to use, other factors being equal. An oil of lower viscosity will cause greater friction because it will squeeze out and allow a closer contact of metallic parts and increase metallic friction and wear. An oil of higher viscosity results in increased fluid friction of the oil. With a complete film of oil, the oil flows like a pack of playing cards sliding over each other, the outer layers adhering to the surfaces and not sliding with reference to them. The actual fluid friction F, in pounds, is given by the equation

$$F = \frac{P \times A \times V}{5{,}760 \times t}$$

where P is the absolute viscosity in poises; A is the rubbing area in square inches; V is the rubbing velocity in feet per second; and t is the oil film thickness in inches. This formula indicates that the friction diminishes as the thickness of the film increases.

The measurement of viscosity has been standardized in this country and is determined by the Saybolt Universal Viscosimeter in which the fluid to be analyzed flows through a tube 0.1765 cm. diameter, 1.225 cm. long, under an average head of 7.36 cm., from a vessel 2.975 cm. diameter. The time in seconds required for 60 c.c. of oil to flow through the tube is the viscosity in seconds

Saybolt. The absolute viscosity is obtained from the equation

$$100\,P = G\left(0.22S - \frac{180}{S}\right)$$

where G is the specific gravity of the oil and S is the Saybolt viscosity.

One of the most troublesome features of lubrication is the decrease in viscosity of oils with rise in temperature. It is found that if the logarithm of the Saybolt viscosity is plotted against the temperature (Fahrenheit), on cross-section paper, the points lie close to a straight line—this is a purely accidental relation.

The viscosities of the principal American lubricating oils at temperatures of 100°, 150° and 212°F. are given in Table 16. It will be noticed that the very heavy oils fall off most rapidly in viscosity as the temperature is raised so that whereas the range in Saybolt viscosities of the oils tabulated at 100°F. is about 11 to 1, at 212° it is only 3 to 1.

The **desired physical characteristics of a lubricating oil** are as follows:

(*a*) Body sufficient to prevent metallic contact under maximum pressure and maximum temperature.

(*b*) Lowest viscosity in keeping with the above conditions.

(*c*) Capacity of resisting high temperatures without decomposition.

(*d*) Fluidity at minimum temperatures.

(*e*) High fire test.

(*f*) Freedom from oxidation.

(*g*) Freedom from corrosive action on metals.

The standard specifications for lubricating oils for airplane engines adopted by the U. S. Army and Navy are given below. Grade 1 is the Navy specification, Grade 2 the Army. The specifications are also different for summer and winter use.

Flash Point.—The flash point of Grade 1 shall not be lower than 400°F.; for Grade 2 not lower than 500°F.

Viscosity at 210°F. shall be within the following limits:

Grade 1 (summer)	90–100 sec.
Grade 1 (winter)	78– 85 sec.
Grade 2	125–135 sec.

Pour Test.—Grade 1 not above 45°F. for summer, or 15°F. for winter.

Cold Test.—Grade 2 not above 35°F.

TABLE 16.—PROPERTIES OF REPRESENTATIVE AMERICAN LUBRICATING OILS FOR USE IN INTERNAL COMBUSTION ENGINES

Kind of oil	Physical properties							
	Baumé grav.	Flash, deg. F.		Deg.F. burn	Deg. F. chill	Viscosity, seconds		
		Open cup	Closed cup			100° F.	150° F.	212° F.
Havoline:								
Light	25.9	370	380	430	33	173	66	42
Medium	25.0	385	395	450	34	237	80	46
Heavy	25.6	395	410	455	46	361	111	54
Mobiloil:								
"E" Light	28.1	370	380	420	0	167	66	44
"A" Medium	21.8	360	360	420	24	330	97	49
"B" Heavy	26.3	500	470	580	41	1,640	397	122
Arctic Lt. Med	23.3	370	380	425	6	221	74	45
Arctic Medium	21.1	370	585	430	8	300	87	46
"BB" Med. Heavy	25.8	460	...	540	46	926	243	86
Monogram:								
Light	27.6	360	360	410	20	140	60	41
Medium	26.0	375	370	430	23	289	95	50
Heavy	28.9	430	445	505	34	340	108	55
Ex. Heavy	24.7	465	425	535	58	1,583	356	110
Perfection:								
"A" Light	29.1	400	410	470	26	181	71	45
"B" Medium	24.9	390	400	450	32	243	81	47
"C" Heavy	29.3	420	430	495	40	316	103	54
Socony:								
Zero	24.3	395	410	470	Sl at 0	219	77	46
Polarine Heavy	25.4	385	380	450	35	300	103	51
Texaco:								
Light	21.3	335	340	380	Sl at 0	205	69	42
Medium	20.9	350	350	400	Sl at 0	301	85	46
Heavy	19.3	356	360	420	10	495	119	51
Veedol:								
Aero No. 1	26.2	455	450	535	39	795	212	78
Aero No. 2	27.1	450	445	530	38	814	222	80
Aero No. 3	26.3	435	435	520	38	517	149	63
Aero No. 4	27.6	440	430	515	34	513	151	64
Aero No. 5	24.7	440	450	520	28	413	135	55
Aero No. 6	24.7	460	460	540	27	474	134	58
Zero Heavy	26.2	410	...	480	33	329	107	52
Zero Extra Heavy	26.1	465	...	550	44	.....	355	111
Wolf's Head:								
Heavy	28.6	415	...	475	32	334	108	52
No. 8	27.6	485	460	550	46	1,196	300	100
Castor oil	15.0	...	...	...	..	1,270	305	90

Acidity.—Not more than 0.10 mg. of potassium hydroxide shall be required to neutralize 1 gram of Grade 1 oil.

Emulsifying Properties.—The oil shall separate completely in 1 hr. from an emulsion with distilled water at a temperature of 180°F.

Carbon Residue in Grade 1 shall not be over 1.5 per cent; in Grade 2 not over 2.0 per cent.

Precipitation Test.—When 5 c.c. of the oil is mixed with 95 c.c. of petroleum ether and allowed to stand for 24 hr. it shall not show a precipitate or sediment of more than 0.25 c.c.

The oil shall not contain moisture, sulphonates, soap, resin or tarry constituents.

The *Flash Test* shows the temperature at which the vapor from a sample, heated in an open cup, will ignite. It has some relation to loss by evaporation. The open cup is 2½ in. diameter, 1⁵⁄₁₆ in. high and is filled to within ⅜ in. of the top. It is placed on a metal plate and heated so that its temperature rises not less than 9° nor more than 11°F. per minute. A test flame ⁵⁄₃₂ in. in diameter is passed across the top of the cup, taking 1 sec. for the passage, at every 5°F. rise of temperature of the oil. The flash point is the temperature at which a flash appears at any point on the surface of the oil. Drafts must be avoided.

The *Viscosity Test* has been discussed on page 329.

The *Pour Test* indicates the temperature at which a sample of the oil will just flow. The oil is placed in a glass jar 1¼ in. diameter and 4 to 5 in. long to a depth of about ¼ in. and the jar is corked. It is then placed in a freezing solution and at each 5°F. drop in temperature it is taken out and tilted. The pour test is 5° higher than the temperature at which the oil will not flow when the jar is placed in the horizontal position.

The *Cold Test* has a similar purpose to the Pour Test but in this case the oil is first frozen and is then stirred with a thermometer until it will run from one end of an ordinary 4-oz. sample bottle to the other. The temperature reading at that time is the Cold Test.

Carbon residue is obtained by heating 10 grams of oil in a porcelain crucible placed inside two iron crucibles with covers. It is heated so as to maintain a vapor flame of specified length and heated further after the vapors cease to come off. After cooling the weight of carbon residue in the porcelain crucible is determined.

Reclaiming Oil.—The oil used in a stationary airplane engine not only becomes diluted by the heavier constituents of the fuel but also becomes dirty by the accumulation of free carbon from the cylinder walls, of metal particles worn off the bearing surfaces, and of other solid impurities. The oil usually has to be changed between the fifth and twentieth hour of flying service; most oil is not used more than 5 hr. It can be reclaimed by allowing it to stand 30 hr. in a tall bucket, decanting off the upper two-thirds, filtering and warming to 150°F. A more thorough process is to put the oil with some water and soda ash in a steam-jacketed tank, raising the temperature to 212°F., forming an emulsion and obtaining a precipitation of carbon, iron, and dirt after a period of rest. The steam drives off the 2 or 3 per cent of diluent coming from the volatile aviation gasoline. A recovery of 85 per cent is possible in this way and the reclaimed oil is at least as good as new oil.

A centrifugal oil cleaner has been tried on the Liberty engine with considerable success. This consists of a spun copper bowl, 5 in. diameter, rotating at 1½ crankshaft speed; the centrifugal force at 2,550 r.p.m. is about 45 times that of gravity. The oil is led into the center of the bowl and is thrown out at the top. Examination of the contents of the bowl after a run show that it collects metal particles, sand, carbon, rubber and other solids. Its use should increase considerably the periods between changes of oil and should prolong the life of all bearing surfaces by preventing their abrasion by solid particles in the oil.

Castor oil is employed in rotary engines in which the gasoline is admitted to the crankcase on its way to the cylinders. Mineral oil (petroleum) cannot be used in this case as it is miscible with gasoline and its use would result in a thinning of the lubricant and a wastage of fuel.

Methods of Lubrication.—The splash system of lubrication often employed in automobile engines is not satisfactory for airplane engines on account of the high loading at which they are operated and also because of the extreme variations in engine orientation during flight. For the last reason also a wet sump is undesirable since it will deluge some of the cylinders during such airplane evolutions as a nose dive and may result in trouble from excess of oil in the cylinder. Consequently the modern airplane engine is provided with a pressure oiling system and a sump, which is usually kept dry by a scavenger pump.

The normal lubricating system is as follows. The scavenger pump or pumps take oil from the sump and deliver it to the external oil tank. The pressure pump takes oil from the oil tank and discharges it into a distributing main from which branches go to each of the main bearings. Oil enters some of the hollow main journals through small holes which register with corresponding holes or channels in the bearing and thereby with the branch oil pipes. Usually, alternate journals are filled with oil and the crank cheeks on the two sides of these journals are drilled to connect with the hollow crankpins and thereby permit lubrication of the crankpins through appropriate holes in them. The oil-containing journals and all the crankpins have closed ends (see p. 144). The lubrication of the piston pin is sometimes carried out by oil pipes running along the connecting rod and registering every revolution with the oil hole in the crankpin; in other cases the oil thrown out by centrifugal force from the crankpin is relied on to lubricate the piston pin as well as the cylinder wall.

The camshaft is lubricated from an oil pipe from the end of the distributing main, connecting with it usually by an annular groove in the front main bearing. The camshaft is hollow and acts as an oil carrier discharging oil through small holes at each bearing. The oil escaping from the bearings lubricates the cams and returns to the crankcase over the distributing gears, meeting there the oil escaping from the main bearings, crankpins and cylinders.

The **lubricating system of the Liberty engine** follows the lines indicated above. The cylinders, pistons and wristpins are lubricated by oil spray from the crankpin. A double scavenging pump at the rear of the engine (Fig. 261) keeps the two sumps drained and returns the oil to the outside oil tank. In the **Hispano-Suiza** engine (Fig. 51) a dry sump is also used. A single gear-type scavenging pump is driven directly from the rear of the crankshaft; an eccentric sliding-vane pressure pump is mounted on the vertical water-pump shaft and rotates at 1.2 times the crankshaft speed. The vane pump forces the oil through a filter to the main oil pipe in the lower crankcase. There are four oil holes in each crank pin through which oil goes to the bearing and is thrown off to lubricate the cylinder and wrist pin. A small hole is provided in the leading face of each cam to lubricate the cam and its follower.

In the **Curtiss K-12** (Fig. 55) the pumps are located in the lower part of the crankcase and are driven through a horizontal shaft. There is a triple-gear scavenging pump and two pressure pumps arranged in a unit with the spiral driving gears and surrounded by a filtering screen. The oil is supplied to the main bearings in the usual way and is then conveyed to the crankpins through small tubes built into the crankshaft. The oil is cooled by a temperature regulator through which the jacket water circulates on its way from the radiator to the pump.

The **Curtiss OX** engine uses a wet sump (Fig. 59) covered by an oil-pan partition. A single-gear pump driven from a beveled gear on the crankshaft at the propeller end sucks oil from the sump and discharges it into the rear end of the hollow camshaft whence it goes through tubes to the crankshaft bearings. In this engine there is a continuous closed oil passage from one end of the crankshaft to the other. The cylinder is lubricated by oil spray. The oil-pan partition has a half-inch hole at its center for the return of oil to the sump.

The **Hall-Scott L-6** engine (Fig. 63) uses a wet sump and has scavenger and pressure pumps mounted as a unit inside the lower crankcase and driven through an inclined shaft from a bevel gear which is at the rear of the crankshaft. The system is otherwise similar to the Liberty engine. An oil sight gage (Fig. 64) shows the level in the sump. Splash plates in the lower case prevent excessive splash from the dipper action of the connecting rods.

In the **Napier "Lion"** engine three oil pumps are combined as a single unit at the extreme rear of the engine; they are of the gear type and are driven at half engine speed. The suction pumps draw the oil away from the two ends of the lower crankcase by two separate steel pipes (Fig. 72) and discharge to the tank through a common pipe. The pressure pump delivers to both ends of the crankshaft and to the three camshaft casings. There is a continuous passage for oil through the crankshaft.

In the **Fiat-650** engine (Fig. 76)there are scavenger pumps at the two ends of the lower crankcase and a pressure pump directly under the rear scavenger pump. They are driven by bevel gears from the horizontal tubular shaft inside the crank casing and are mounted in ball bearings as well as the horizontal shaft and the spur gear which drives it. The oil drawn from the main tank is discharged into a copper main cast into the crankcase. The main bearings are fed from copper branch pipes. In other respects the system is normal.

In the **Benz-230** (Fig. 77) a wet sump is used and the triple oil pump is submerged in the sump. The main pressure pump *A* (Fig. 260) draws oil from the reservoir in the sump and discharges it to the main bearings through a distributing main and branch pipes. The supply to the piston pins is through small pipes inside the tubular connecting rods. Fresh oil is fed into the sump by the small suction pump, *B*, from the oil tank while the correct working oil level is maintained in the reservoir by the pump, *C*, whose curved suction pipe (see Fig. 77) terminates at the desired oil level. All return oil passes over the transverse air pipes in the lower crankcase and is cooled by them.

In the **Maybach** engine (Fig. 80) scavenger pumps are mounted at both ends of the crankcase and the pressure pump is placed behind the rear scavenger pump. All three are operated by the same horizontal shaft driven by spur gearing from the front end of the engine shaft. The oil is discharged into an

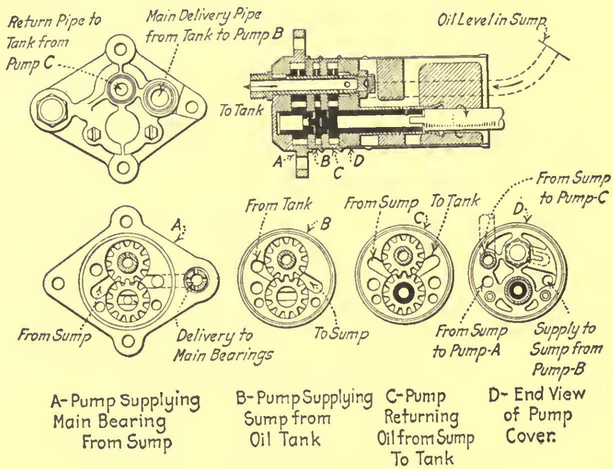


Fig. 260.—Triple-gear pump of Benz engine.

external oil main on the upper crankcase and past individual screens into branch pipes drilled through the transverse webs into the main bearings. The oil thrown off from these bearings is caught in aluminum scoops bolted to both ends of each crankpin, is carried by centrifugal force into the hollow crankpin and thence through radially bored holes to the bearing. The piston pins are oiled through the internal pipes in the connecting rods. Baffle plates bolted to the upper crankcase just below the cylinders prevent an excess of oil reaching the cylinders.

Relief Valves.—All pressure-feed systems are provided with a relief valve on the discharge side of the pump. This is a spring-loaded valve as in Fig. 261 and is set for the maximum allowable pressure. The oil pressure is high in starting especially in cold weather when the viscosity of the oil is very great. The normal operating oil pressures after fully warming up are about 25 to 30 lb. per square inch in Liberty engines, 50 to 60 lb. in Curtiss engines, 40 to 65 lb. in Hispano-Suiza engines. In cold weather it is best to drain off the oil after a flight and to fill up with hot oil before starting.

The **location of oil grooves** in the bearings is a matter of considerable importance on which there is much divergence of practice. The actual pressure on the oil film will vary from zero at the ends of the bearings and at the split of the bearing to possibly as much as 10,000 lb. per square inch in the center of the loaded area at the moment of maximum loading. As the oil pressure does not exceed 50 lb. per square inch it is obvious that oil cannot be forced in at the place of maximum loading unless the pressure at that place falls below 50 lb. per square inch during some part of the cycle. There should be no oil grooving lengthwise in the middle of the most loaded half of a bearing; such grooves are channels of escape for the oil and may result in such thinning of the film as to increase the friction and, possibly, to cause seizing. The most heavily loaded part of the main bearing is usually the middle of the lower cap and it is at this place that the oil usually enters. The grooves should then be two helical grooves intersecting at the oil hole at the center of the bottom half of the bearing and running to the split but not too near the ends of the bearing. Short helical grooves in the upper half of the bearing may start at the split opposite the lower grooves but should not go more than half-way up each side. Similar grooving should be provided at the crankpin bearing, which also is most heavily loaded at the lower half.

Wherever practicable it is desirable that the oil should enter at the place of minimum average bearing pressure. It is the cyclical variation in the loading that makes possible the proper lubrication of the heavily loaded bearing surfaces of aviation engines. The oil pressures employed are in themselves not nearly adequate to support the loads but are required to overcome the viscous and frictional resistance to the flow of the oil to the various bearings and also to ensure that the oil channels will clear themselves of small obstructions.

The **amount of oil circulated** per minute is determined not only by the lubrication needs but also, as pointed out on page 329, by the extent to which the oil is used as a cooling medium. For lubrication the amount should probably be some function of the total projected areas of the main bearings and crankpins; expressed in this way the use of oil varies from 0.1 to 0.5 lb. per square inch of projected bearing area per minute. In terms of the power delivered by the engine the oil circulated varies from 0.025 to 0.15 lb. per horse-power minute.

The oil consumption of an engine as usually measured is the amount of oil which has to be added to the system to make up for oil burned or otherwise used up during the engine operation. This quantity varies from 0.02 to 0.05 lb. per horse-power hour in stationary water-cooled engines but may go as high as 0.15 lb. in rotaries.

Oil Pumps.—The great majority of airplane engines use gear pumps both for scavenging and pressure pumps. A simple gear pump consists of a power-driven spur gear meshing closely into an exactly similar driven gear, the gears being enclosed in a casing with the minimum working clearance above and below the gears and also around them except where they mesh. The oil inlet is on the side where the gears separate; the discharge is on the side where the gears meet. If there were no leakage the oil carried from the inlet to the discharge side would be equal to the space between the teeth, but some of this is brought back to the inlet side since a tooth going into mesh does not fill the space between adjacent teeth and consequently does not displace all the oil content of that space. The capacity of each gear can be taken approximately as equal to half the annular space between the roots and tips of the gear, or, for the two gears, as equal to the whole of the annular space of one gear. As the width of this annular space increases with decrease of the number of teeth (decrease of pitch) it is evident that, for a given pitch diameter, capacity can be increased by decreasing the pitch.

Two scavenger gear pumps are sometimes combined into a triple-gear pump as in the Liberty engine. In this case the driving gear is central and the inlets to the driven gears are on opposite sides of the pump.

The driving gears for scavenger and pressure pumps are usually placed close to one another and driven by the same shaft. In the Benz engine (Fig. 260) three such gear pumps are mounted on the driving shaft and function as indicated in the figure. The Liberty oil pump is driven at one and one-half times crankshaft speed and has a capacity of 1.9 gal. per minute at normal speed.

The pump consists of a double scavenging pump with three gears, *A*, *B* and *C*, Fig. 261, drawing oil from the two sumps and

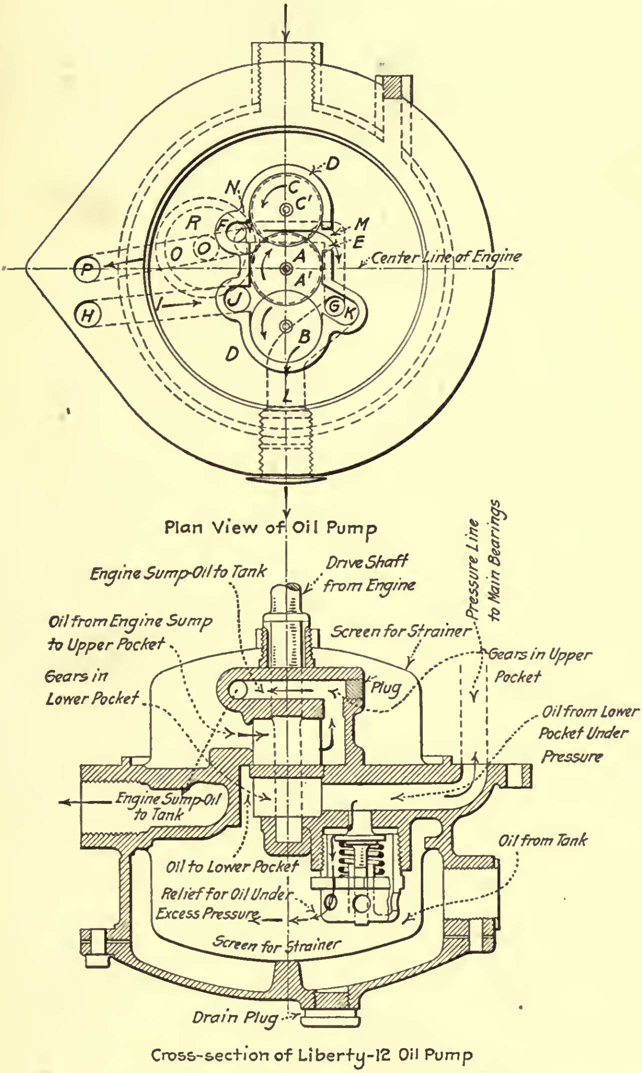


Fig. 261.—Gear pump of Liberty engine.

a pressure pump immediately below it giving an oil pressure of 35 to 50 lb. per square inch at engine speeds from 1,500 to 1,800 r.p.m. The pressure gears, *A'*, *C'*, (Fig. 261) are immediately below the gears A and C of the scavenging pump. The gear A' is driven from the pump shaft and the upper train is operated through a vertical-shaft connection from C' to C. The pressure pump draws oil from the tank through a copper pipe, the oil passing through the large lower strainer and entering the gear housing at M; it is discharged through the passage NOP to the distributing manifold. The pressure relief valve is shown in

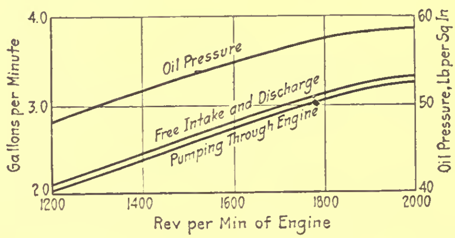


Fig. 262.—Performance curves of gear pump of Packard-180.

Fig. 261; the spring is usually set for a pressure of 50 lb. per square inch. The discharge from the relief valve goes to the suction side of the pressure pump.

The oil from the rear sump entering the scavenging pump after passing through the upper strainer is drawn through E and discharged to the outlets F and G, which are connected by a passage in the pump body. From G, the oil goes through the passages K

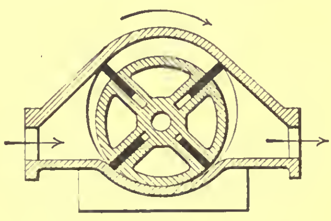


Fig. 263.—Sliding-vane pump.

and L to the oil tank. Oil from the front sump is taken through an internal pipe and the passage HIJ and is also discharged at F and G.

The volumetric efficiency of the gear pump can be made very high—probably up to 90 per cent; the over-all efficiency is 50 to 60 per cent. Tests of the pressure pump of the Packard 180-h.p. engine give the results shown in Fig. 262. It will be seen that the slip is very low since the discharge curves are almost the

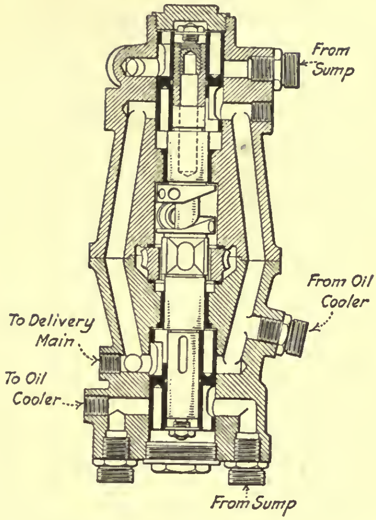


FIG. 264.—Plunger pump of Bassé-Selve engine.

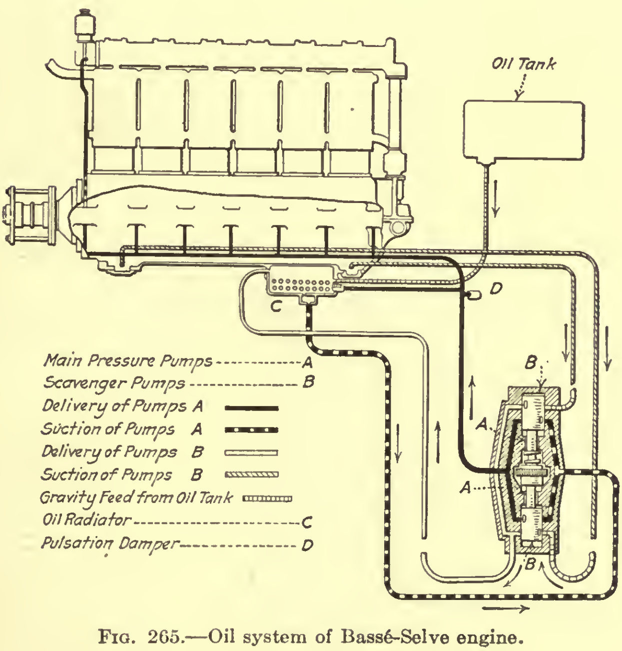


FIG. 265.—Oil system of Bassé-Selve engine.

same with free discharge and with discharge against pressures which vary from 48 lb. at 1,200 r.p.m. to 58 lb. at 2,000 r.p.m. It may be further noted that the slopes of the discharge curves up to 1,600 r.p.m. are such as to go through the zero of ordinates; that is, the pump discharge is directly proportional to its r.p.m.

The eccentric sliding-vane pump (Fig. 263) is used occasionally but is being displaced by the gear type. The sliding vanes are

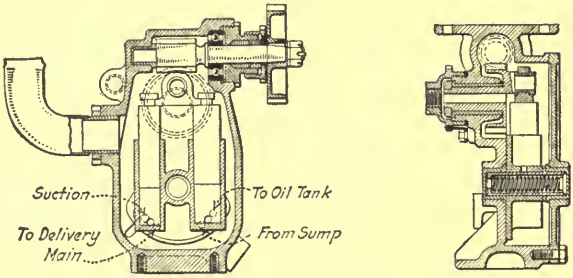


FIG. 266.—Plunger pumps of Salmson engine.

pressed out by springs in a slotted cylinder which is mounted eccentrically in a cylindrical casing and is power-driven.

Plunger pumps are used in some of the German engines, being operated by eccentrics or by scroll cams. In the Bassé-Selve engine, (Fig. 264) the oil pump is driven by a worm gear. The pump consists of two double-acting steel plungers which work vertically in aluminum barrels and make in effect four pumps.

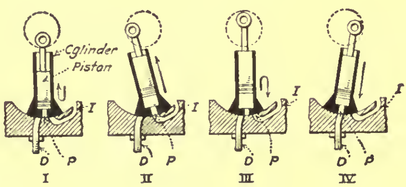


FIG. 267.—Action of LeRhone oil pump.

The plungers are rotated by the worm gear and are simultaneously reciprocated by the action of the scroll cam cut in the spindle and operated by a hardened steel roller working on a pin screwed into the pump body. At each stroke of the two plungers oil is drawn from the cooler tank to one of the inner pump chambers and is discharged from the other inner pump chamber into the delivery main. At the same time oil is sucked out of one of the engine sumps into one of the outer pump chambers and is pumped from the other outer pump chamber into the cooler tank. A diagrammatic view of the system is shown in Fig. 265.

An entirely different arrangement of plunger pumps is used in the Salmson engine (Fig. 266). In this case two plungers are used, the larger one being the scavenging pump. The plungers are pivoted on a crankpin rotated by worm gearing. The pump bodies are pivoted and oscillate through a small angle as the crankpin rotates and the plungers make their strokes. The connections to suction and discharge are made when openings in the oscillating pump bodies register with appropriate openings

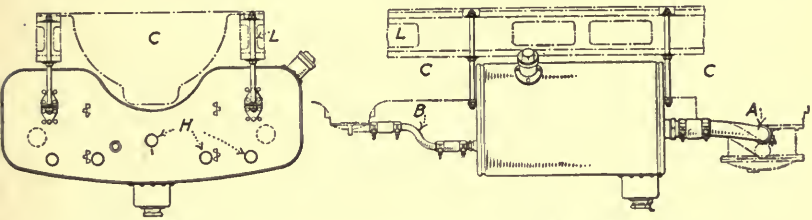


FIG. 268.—Mounting of oil tank.

in the pump casing. The method of action of a single pump of this type, used on the LeRhone engine, is shown diagrammatically in Fig. 267. The port, *P*, in the oscillating cylinder comes alternately opposite the intake port, *I*, and the delivery port, *D*.

The method of mounting the oil tank and of connecting it to the engine in the USD-9A airplane is shown in Fig. 268. The tank is slung from the engine sills, *L*, and is located just below the crank case, *C*. The tube, *A*, is the oil return pipe from the scavenger pump to the tank; the supply pipe from the tank to the pressure pump is shown in dotted lines. The front sump connects by a pipe, *B*, to a Y-shaped air-pressure-relief pipe inside the tank, the upper ends of which extend nearly to the top of the tank. This acts as a pressure-relief vent for the tank. The oil is cooled by longitudinal air pipes, *H*.

CHAPTER XIV

THE COOLING SYSTEM

All airplane engines are ultimately air-cooled. The only option is as to whether the air shall be applied directly or indirectly. In the latter case, the heat is removed by water which is then cooled by the air. Indirect (water) cooling offers two advantages: (1) the transmission of heat from the cylinder is more rapid to water than to air, and (2) the ultimate cooling surface (in the radiator) can be made much greater than is possible at the cylinder. Direct-air cooling has the advantage of reduced total weight and diminished vulnerability; a bullet hole through the radiator or jacket will put the whole engine out of action while a hole through one cylinder of an air-cooled engine may put that cylinder alone out of action.

Air cooling in airplanes is used almost exclusively in rotary and radial engines. In vertical or Vee engines with several cylinders in line, the cooling problem is much more difficult, although it has been met by the use of suitable cowling to direct the air on to the different cylinders. With the rotary and radial types the motion of the plane and the location of the engine in the slip stream of the propeller ensure an adequate flow of air for cooling; with the multicylinder vertical or Vee type it is sometimes necessary to add a fan to improve the air circulation.

The resistance or drag of air-cooled cylinders is considerable but has not been determined experimentally in a satisfactory manner. In the rotary engine there is, in addition to the drag, the resistance due to churning which reduces directly the b.h.p. of the engine. This resistance increases so rapidly with speed that it is not found desirable to operate rotary engines at speeds in excess of about 1,400 r.p.m.; the increase in indicated power which results from increased speed is largely used up in overcoming the increased air resistance. The total work done by an air-cooled engine in overcoming air resistance is probably greater under ordinary conditions of operation than the total work done by a water-cooled engine in overcoming the drag of its radiator. Until quite recently, air-cooled cylinders were at a great disadvantage both in fuel economy and in power developed per unit volume of piston displacement, but recent constructions have put the air- and water-cooled engines very nearly on a par in these respects. They have always had an advantage in weight per horse power.

The heat which has to be removed from the cylinder in order to keep the temperature of the cylinder within the limit which permits satisfactory operation of the engine is usually about equal to the heat equivalent of the work done in the cylinder, or 42 B.t.u. per brake horse power per minute. This quantity will increase or decrease with change in operating conditions; its limits are apparently between 30 and 60 B.t.u. per brake horse power per minute.

The **cooling surface** of the modern air-cooled cylinder almost invariably takes the form of a series of fins. Tests on the rate of heat dissipation from such surfaces, made by the British Advisory Committee for Aeronautics,[1] show that, for wind speeds between 20 and 60 miles per hour, the heat loss for a given material is independent of the roughness of the surface. Steel shows 5 to 10 per cent greater heat dissipation than aluminum or copper. Aluminum is improved about 10 per cent by a coating of stove enamel.

Throughout the usual range of cylinder diameters the heat dissipation for copper fins in a parallel air blast at the ground is given by

$$H = [0.0247 - 0.0054(l^{0.8}/p^{0.4})]V^{0.73}$$

where H is the heat dissipated in B.t.u. per square foot of fin surface per minute per degree Fahrenheit difference between the mean fin temperature and the incoming air temperature; l is the length of the fins in inches; p is the pitch in inches measured from surface to surface of adjacent fins; and V is the wind speed in miles per hour. With tapering fins p should be taken at the mean height of the fin. The heat dissipation, depending on the weight of air brought into contact with the cylinder, is proportional to $(dV)^{0.73}$, where d is the air density.

Shape and Size of Fins.—The fin which gives the maximum heat-loss per unit of weight is one having slightly concave surfaces and a sharp tip; a plain triangular fin is only very slightly less efficient. The best proportions for such a fin depend on the conductivity of the material and on the wind speed. For a speed of 40 miles per hour, the following table shows the best

[1] A. H. Gibson, *Institution of Automobile Engineers of Great Britain*, 1920.

proportions for fins of aluminum alloy (conductivity = 0.38 C.G.S. units) and steel (conductivity = 0.12 C.G.S. units) and also for rectangular copper fins (conductivity = 0.90 C.G.S. units).

Bottom breadth, centimeters	0.025	0.05	0.1	0.2	0.3	0.4	0.5
Length, centimeters:							
Aluminum	.....	...	2.0	2.9	3.5	4.1	4.5
Steel	.....	...	1.1	1.5	1.8	2.1	2.3
Copper	1.6	2.3	3.3	4.8			

If such a fin be truncated until the tip breadth is one-fifth of the bottom breadth, the lengths become 80 per cent of those given above. The heat dissipation is about 0.88 time and the weight 0.96 time as great as for the complete triangular fin.

Since the heat dissipated from a fin of given shape varies directly as the length of the fin, while the weight varies as the square of the length of the fin (other things being equal) cooling fins should be as short as possible, a large number of short thin fins being used in preference to a smaller number of longer and thicker fins. While in practice this is to be borne in mind, many other factors besides that of weight have an important bearing on the best size of fin to be adopted. Thus, in a thin steel cylinder, or in a cylinder of cast iron or cast-aluminum alloy, the circumferential ribs add greatly to the strength and resistance to distortion. Comparatively deep and heavy fins have a greater effect in this direction than a larger number of similar but smaller fins giving the same cooling. Again, as the number of fins is increased, the pitch is correspondingly diminished. This diminution in pitch reduces the air flow between the fins to an extent which may, with very small pitches, render the fins practically useless for cooling purposes.

In a cylinder of cast iron or of aluminum alloy, foundry difficulties put a definite limit to the minimum pitch of the fins. On the barrel itself a somewhat smaller pitch may be adopted than on the head, or the barrel fins may be turned out of the solid if desired. On account of the complicated form of the cylinder head and ports, however, it is difficult to machine their cooling fins, and the length of many of the cores necessitates the pitch being made fairly large. The minimum practical pitch of fin for such cylinders, having a diameter of from 4 in. to 6 in., is about 8 to 9 mm. or about 5/16 in. Foundry difficulties also prevent the casting of a fin having a tip less than about 0.5 mm. in thickness, or a root thickness less than about $l/10$, so that an aluminum fin 1 in. long would not have a root thickness less than 0.1 in.

For steel cylinders with fins turned out of the solid, the pitch may with advantage be cut down to about 1/4 in. on a cylinder of 3 in. or so in diameter, but there appears to be little to be gained by reducing the pitch beyond this point.

The **mean fin temperature** depends on the total amount of heat which has to be dissipated and its value depends on many factors. Determinations of the actual temperatures of the cylinder walls, pistons, and exhaust valves show no definite differences between well designed air-cooled and water-cooled engines; the air-cooled cylinder may be, and often is, the cooler of the two.

The **influence of engine speed on the wall temperature** is shown in the following table giving the temperature at a point on the side of the combustion space of an aluminum air-cooled cylinder operating at maximum load.

R.p.m.	800	1,000	2,200	1,400	1,600	1,800
B.h.p.	10.2	12.8	15.4	18	19.7	20.6
Temperature, degrees Centigrade	100	103	124	123	136	138

The **compression ratio** is more important than engine speed in determining the wall temperature. There is usually a definite compression ratio giving minimum wall temperature; variations on either side increase that temperature. The following table gives tests of a 100 by 140 mm. aluminum air-cooled cylinder with varying compression ratio. The brake mean effective pressure and fuel consumption of this engine are noteworthy.

Compression ratio		4.6	5.0	5.4	5.8	6.2	6.4
Brake mean effective pressure, lb. per sq. in.		116.1	119.3	122.0	125.0	129.0	123.0
Fuel, pounds per brake horse power per hour		0.530	0.507	0.490	0.475	0.480	0.520
Mean barrel temperature, degrees Centigrade	Top	180	170	157	154	183	212
	Bottom	105	95	89	85	110	135

The increased temperatures in the last two columns result from preignitions which were occasional with compression of 6.2 and frequent with 6.4.

Increase in **cylinder diameter** diminishes slightly the heat loss to the walls per brake horse power; the experimental evidence suggests a decrease of about 3.5 per cent for 10 per cent increase in cylinder diameter. As the ratio of cooling surface to b.h.p. varies inversely as the diameter in similar cylinders, the ratio of cooling area to heat given to the walls decreases as the diameter increases. The temperature difference between an air-cooled cylinder and the cooling air in a given wind may be taken as inversely proportional to $D^{0.6}$, where D is the cylinder diameter.

The **air-fuel ratio** has considerable influence on the heat transmitted to the cylinder walls. The cylinder is hottest with the weakest mixture capable of sustaining maximum load; or, approximately, with an air-fuel ratio of 13.5. Further weakening of the mixture makes a cooler cylinder on account of the reduction in brake horse power and in the heat loss per b.h.p. At the same time it gives a hotter exhaust valve. The last point is brought out in the following table giving test results for a 100 by 140 mm. air-cooled aluminum cylinder.

Air-fuel ratio	11.1	11.9	13.8	15.2	15.7
Brake m.e.p., lb. per sq. in.	122	122	119	116	114
Fuel, pounds per brake horse power per hour	0.622	0.589	0.515	0.480	0.470
Exhaust valve temperature, degrees Centigrade	706	717	747	752	747

An increase in mixture strength beyond that necessary for maximum power reduces the temperature of the cylinder surfaces, as shown below for an engine operating at full throttle and constant speed.

Air-fuel ratio	10.5	11.4	12.9	13.5	15.4
Cylinder head temperature, degrees Centigrade	200	215	237	229	215

Tests show that the **maximum temperature of the head** of an air-cooled cylinder must not exceed 270°C. for satisfactory working. If the temperature exceeds 280° there is usually trouble from preignition. Higher working temperatures are permissible with larger cylinders. If the temperature is kept at 200° to 220°C., the economy and capacity obtained are quite as good as for water-cooled cylinders of similar design and size.

The **temperature of the exhaust valve** at its hottest point should not exceed 720°C.; with valves not exceeding 1.5 in. in diameter it is possible to reduce this temperature to 650°C.

In a well designed aluminum cylinder of the overhead-valve type, operating in a 60-mile-per-hour wind, a provision of 0.28 to 0.35 sq. ft. of cooling surface per brake horse power is sufficient to give satisfactory operation, the larger area applying to cylinders of about 4-in. bore, and the smaller to cylinders of about 6-in. bore. For steel or cast-iron cylinders with overhead valves this area must be increased about 50 per cent and for L-head cast-iron cylinders by 100 per cent.

At reduced wind speeds the cylinder temperature increases; the mean temperature difference between the fins and the air varies inversely as $V^{0.4}$. Thus in a given series of tests a reduction of wind speed from 80 to 40 miles per hour increased the cylinder temperature from 229° to 296°C. There are practical difficulties in the way of providing sufficient cooling surface for operation at full throttle below certain limiting wind speeds. The wind speeds can be less with smaller cylinders. The minimum air velocity for good performance of air-cooled cylinders of good design and material under full throttle is given below. At lower air speeds partial throttle only should be used.

Diameter, inches	2	3	4	6	8
Minimum air velocity, miles per hour	30	40	50	70	90

Cylinder Materials.—With cylinders of normal design the middle portion of the head is the hottest point. Free air flow to this point is impeded by the valve ports and gears so that it is almost impossible to provide adequate cooling surface there. The heat has to travel outward and is dissipated mainly from the cooling surface surrounding the combustion head. It is therefore important to use a material of maximum thermal conductivity. The three practical materials for cylinder construction, steel, cast-iron and aluminum alloy, have conductivities (in C.G.S. units) of 0.12, 0.10 and 0.38 respectively. Aluminum is consequently the most desirable material. The alloys most suitable for cylinders are copper-aluminum alloys with about

90 per cent of aluminum. The high-zinc alloys are unsuitable because their tensile strength is low at 200°C. All the alloys show rapid decrease of strength as the temperature increases beyond 250°C. The following table gives data on this point.

Composition per cent						Tensile strength, lb. per sq. in.	
Cu	Sn	Mg	Al	Ni	Mn	At 250°C.	At 350°C.
7.0	1.0	...	92.0	..	..	12,300	6,700
12.0	...	...	88.0	..	..	23,500	13,400
14.0	...	...	85.0	..	1	21,300	14,500
4.0	...	1.5	92.5	2	..	24,600	11,200
9.0	...	...	89.0	2	..	19,000	10,100

Water Cooling.—By water-cooling the cylinder and the exhaust ports it is possible to run with higher speeds and compression ratios than are practicable with air-cooled cylinders. The possible increase in speed and ratio of compression are relatively unimportant when compared with the performance of the best recent constructions in air-cooled cylinders; they are considerable as compared with the average air-cooled cylinder.

With water cooling it is possible to maintain almost any desired cylinder temperature. If the temperature is low the volumetric efficiency and the capacity of the engine will be improved (see p. 37) but the engine friction increases and its efficiency falls off. The temperature of the jacket water after leaving the radiator must be below the boiling point of water at the pressure existing on the suction side of the pump, otherwise the pump will not function well but will suck in water vapor. As fuel economy is ordinarily more important than capacity, the jacket water is usually kept at as high a temperature as the boiling point will permit. The mean jacket temperature at the ground is usually 160 to 180°F.

Water is not the ideal cooling agent. A less volatile fluid would permit higher cylinder temperature; higher efficiencies might be obtained without running into such temperatures as would cause preignition. The same result might be obtained by operating a closed water-cooling system under pressures greater than atmospheric, but this would necessitate heavier material for the radiator core and consequent increase in weight and decrease in airplane efficiency. The heat which is removed by the jacket water is practically equal to the b.h.p. or is 42.4 B.t.u. per brake horse power per minute. In a closely cowled engine this same amount of heat would have to be removed from the radiator. With the usual cowling there is considerable removal of heat by the air stream from the engine and water-jacket surfaces, so that only about 31 B.t.u. per brake horse power per minute has to be removed from the radiator: with an uncowled engine this quality falls to 23 or 25 B.t.u.

The principal parts of a water-cooling system are the jackets, the pump and the radiator. The last of these will be considered first.

Radiators.—Airplane radiators have developed from automobile practice but certain types of automobile radiators are entirely unsuited to airplane practice. The successful commercial types have cores made of thin brass, or copper ribbons or tubes from 0.004 to 0.006 in. thick. Common types are shown in Fig. 269, which illustrates: *a* and *b*, rectangular air passages; *c*, rhombic passages; and *d*, circular passages with hexagonal ends. Other common types have hexagonal or elliptical air passages. The water passages are narrow, varying from 0.03 to 0.08 in. The air tubes are commonly not more than ¼ in. in maximum cross-section dimension and are from 3 to 5 in. long (depth of core). The metal sheets are stamped or rolled to the desired form with the front and rear ends of the pair of sheets forming each water passage in contact with one another. These ends are soldered by dipping them into a shallow pool of molten solder. Great care must be exercised to keep down the weight of solder as much as possible; it often amounts to 25 per cent of the total weight of the radiator core. The top and

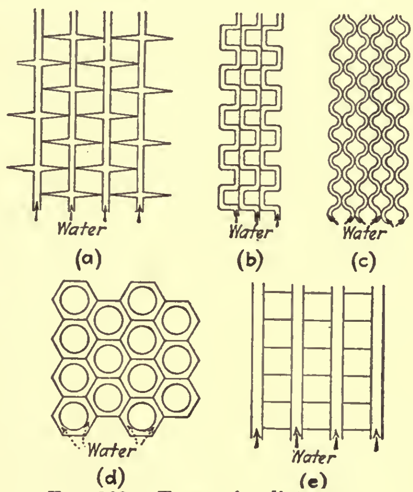


Fig. 269.—Types of radiator core.

bottom ends of the water passages are inserted through slots in the top and bottom headers respectively; the two sheets of each water passage are spread apart and soldered to the header. In the case of type *d*, Fig. 269, the ends of the circular tubes are expanded into hexagonal forms which are soldered together; the expansion is made enough to give the desired width of water passage between the tubes. Type *b* differs from type *a* not only in the method of assembly but may also be made of a corrugated surface which is intended to give greater strength and larger radiating surface. The types *a, b, c* and *d* in Fig. 269 have water in contact with all the radiating surface and are said to have only "direct" radiating surface. Many automobile radiators have extensions of this direct radiating surface in the form of fins on flat or circular tubes (*e*, Fig. 269), metal spirals, and so forth. Such "indirect" radiating surface is found to have too high a ratio of head resistance to heat-removing capacity to be satisfactory for airplane use.

The dimensions or external shape of a radiator can be adapted to suit its location and desired performance. The location may be such that air may pass through or around it without obstruction, in which case it is said to be in an "unobstructed" position. On the other hand, the radiator may be located in the nose of the fuselage, or in the plane of the wing, in which case the air flow is materially affected by other parts of the plane and the radiator is said to be "obstructed." The performance of such a radiator will depend not only on the size and type of the core but on its position or surroundings. Examples of typical unobstructed locations are shown in Fig. 270 (at the sides of fuselage) and in Fig. 271 (over the engine); common obstructed positions are in the nose of the fuselage, and in the wing.

A comprehensive study of the properties of various types and dimensions of radiator cores has been made at the Bureau of Standards and published in the *Fifth Annual Report* of the National Advisory Committee on Aeronautics. The following discussion is mainly from that source.

Two quantities are of importance in determining the **heat transfer** of a core. They are the **temperature difference** between the entering air and the mean water temperature; and the **mass flow of air.** The temperature difference should ordinarily be taken as the difference between the mean summer air temperature and the mean water temperature. The mass flow of air,

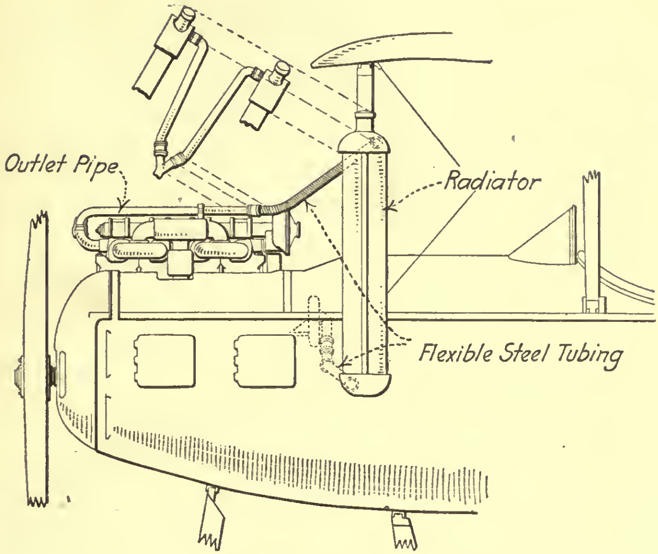


Fig. 270.—Side radiators.

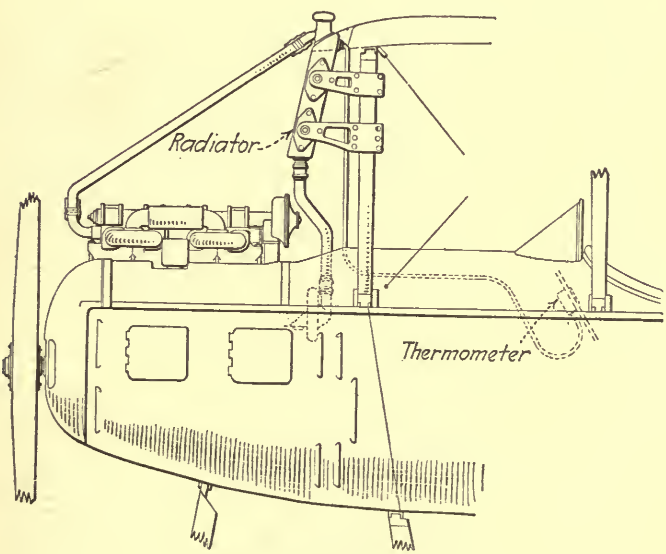


Fig. 271.—Overhead radiators.

M, is the weight of air flowing per second per square foot of frontal area of the core. Its amount (at constant air density) is found to be proportional to the free air speed or the velocity with which the core moves through the air when the core is unobstructed; the mass flow is always less for obstructed positions than for unobstructed.

The **energy dissipated** or **heat transfer** is expressed in horse power per square foot of frontal area, and, for purposes of comparison of the properties of various cores, a temperature difference of 100°F. is assumed between the air entering the radiator and the mean water temperature; the heat transfer is proportional to this temperature difference. One horse power is equivalent to 42.54 B.t.u. per minute.

The **head resistance** of the core is the force required to push it through the air and is expressed in pounds per square foot of frontal area. This head resistance is found to vary approximately as the square of the free air speed; in most cases the exponent is slightly less than 2. If R is the head resistance, and V the free air speed in miles per hour, then

$$R = cV^2$$

and c is called the **head resistance constant.**

The **horse power absorbed** by a radiator is the engine power required to overcome the head resistance and support the weight of the radiator. The work done in supporting the weight can be calculated if the lift-drag ratio of the plane as a whole is known. An average value of 5.4 may be assumed for this ratio. If W is the weight of the core and contained water in pounds per square foot of frontal area, the propeller thrust required to support the weight is $W/5.4$. The horse power absorbed is

$$\text{H.P.} = \left(R + \frac{W}{5.4}\right) \cdot \frac{V \times 5{,}280}{60 \times 33{,}000}$$
$$= \frac{1}{375}\left(R + \frac{W}{5.4}\right)V.$$

It should be noted that this method of calculation neglects the effect on the lift-drag ratio of the addition of the radiator. The lift-drag ratio varies between different planes and varies even more widely between climbing and level flight. The selection of a core for a given plane cannot be made satisfactorily without a knowledge of the relative importance of climbing speed and top speed. A lift-drag value of 5.4 is a good average and gives about equal value to climbing and level speed. If the rate of climb is of prime importance the value may be as low as 3, while if speed on the level is the most important, a value as high as 10 may be used.

A small additional power charge against the radiator is that required to overcome the resistance to water circulation in the radiator. It is usually so small as to be negligible.

The definition just given of horse power absorbed applies only to the case of an unobstructed radiator. If the addition of a radiator necessitates alterations in structure (such as the substitution of a flat nose for a stream-line fuselage or the enlargement of the fuselage to accommodate the radiator required) the consequent increase in resistance of the structure should be charged to the radiator.

A comparison of the performance of various cores can be obtained when the heat transfer per unit of power absorbed is known. This quantity is called the **Figure of Merit** and is a pure number. The comparison must be for the same temperature difference and free air speed. It applies only to unobstructed radiators.

The general conclusions derived from the tests at the Bureau of Standards are as follows:

Heat transfer is a function of mass flow of air and is independent of the air density.

Heat transfer is roughly proportional to mass flow for a core having only direct cooling surface. When there is a considerable amount of indirect cooling surface the heat transfer increases less rapidly than mass flow at high air speeds.

Heat transfer is proportional to the *temperature difference.*

Heat transfer is not greatly affected by the *rate of water flow* provided the rate is above 2 gal. per minute per inch of core depth per foot width of core. It should be noted, however, that this is true only when the mean water temperature is regarded as constant.

Heat transfer from *direct* cooling surface is not appreciably affected by the *composition of the metal.* When fins and other indirect cooling surface are used the thermal conductivity of the metal is important.

Heat transfer is somewhat increased, but at the expense of a large increase in head resistance, by spirals or other forms of passages which increase the turbulence of the air stream. Heat transfer is greater for smooth than for rough tube walls, for, if the surface is rough, it will be covered with a layer of more or less stagnant fluid.

Head resistance for any particular core varies approximately as the square of the *free air speed.*

The head resistance of a core appears to be closely related to its mass flow so that, in general, anything which tends to cut down the flow of air through the core will cause a considerable increase in head resistance.

Head resistance varies directly as the *air density* for a given free air speed, and inversely as the density for a given mass flow.

Head resistance is considerably increased by projections, indentations, or holes in the air tube walls.

Head resistance per square foot is not appreciably affected by the *size of the core* within the limits used, *viz.*, 8 by 8 in. to 16 by 16 in. and 12 by 24 in.

Special conclusions with reference to types of cores are as follows:

For a high figure of merit the core should have smooth, straight air passages, easy entrances and exits for the air and a large percentage of free area. Under these conditions the figure of merit increases as the depth increases up to at least 20 times the diameter of the air tubes, which is as far as experiment has gone. Even greater depths may be of advantage.

By far the **most satisfactory radiator for use in unobstructed positions** seems to be one of thin flat plates with water space not over 1/16 in. wide and spaced 1/2 in. on centers. The plate should be at least 12 in. deep. As the figure of merit changes but slightly with increase of depth beyond 12 in. the depth may be made 20 in. or more if it is desirable to reduce frontal area. The chief defect of the type is mechanical weakness. Of the commercial radiators tested, those have given highest figure of merit at high air speeds which have only direct cooling surface in the form of tubes about 1/4 in. square and about 5 in. deep. The figure of merit of this type at 120 miles per hour free air speed varies from about 8 to 8.4, whereas flat plates 9¾ in. deep and 1/2 in. on centers have a value of 10.7. The energy dissipated per square foot of frontal area is less in the above flat plate radiator than in the best square tube radiators so that a larger frontal area will be required with flat plates but the power absorbed will be less.

The British Air Ministry has adopted as standard a circular tube 10 mm. in diameter expanded at the ends to a hexagonal section 11 mm. across the flats (Fig. 269 *d*). The standard length of the tubes is 120 mm. The material is 70 − 30 brass with wall thickness of 0.005 in.

The actual power absorbed by the radiator in being lifted and pushed through the air (see Table 18) varies from about 3 h.p. to 6 h.p. per square foot of frontal area at 100 miles per hour. This amounts to from 5 to 20 per cent of the total engine power. A small gain in radiator performance may have an appreciable effect at high speed.

The selection of a **radiator core for an obstructed position** is more difficult. An obstructed position involves a large absorption of power. The resistance of a fuselage fitted with a nose radiator is two or three times the resistance of the same fuselage with a stream-line nose. The increase in resistance due to the substitution of a radiator for a stream-line nose is greater than the increase that would be caused by using a radiator of the same core construction and the same cooling capacity in an unobstructed position.

At any given free-air speed the total resistance of a fuselage with a flat nose radiator is increased by increasing the air flow through the radiator, either by opening exit vents for the air or by decreasing the resistance of the radiator to the passage of air. This indicates that a nose radiator should be of compact construction with high heat transfer, for low air flows through the core, requiring a core of high resistance. This fact is of special importance since the space available for a nose radiator is so limited that the highest possible mass flows are used in practice. A nose radiator with air exit vents equal in area to the free air passage through the radiator is found to cut down the heat transfer about 35 per cent as compared with the same radiator in an unobstructed position. Indirect cooling surface may be of advantage if it is made of copper, crimped from the water tube walls and well soldered to them at all possible places. Several types of core show good heat transfer at low speeds, but here again the square tube, with direct radiating surface only, gives best result of all commercial types, and flat plates spaced ¼ in. on centers show excellent performance. The fin and tube type with its small amount of direct surface has no use in airplanes except possibly in a wing position where a high head resistance is no disadvantage.

The **properties of cores** selected as typical of common construction are given in Tables 17 and 18. The constructions vary from the flat tubes (E-6 and E-8) with 100 per cent direct cooling surface and a minimum of head resistance to finned circular tubes (F-5) with only 12.3 per cent of direct surface and three times as much head resistance per square foot of frontal area as the best flat tubes. The dimensions of the core are given in Table 17; the performance at various wind speeds in unobstructed positions in Table 18. The "figure of merit" necessarily diminishes

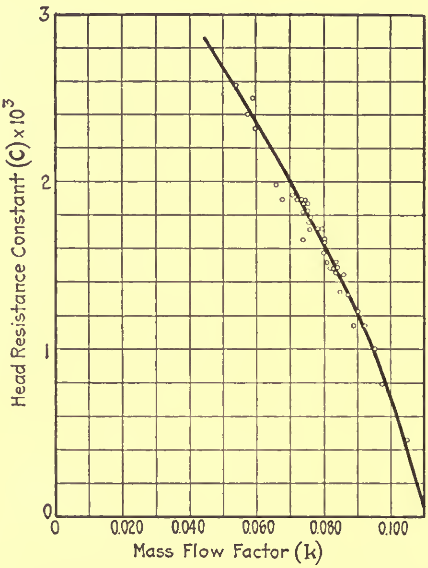


FIG. 272.—Head resistance constant and mass flow factor.

with increase of wind speed; the order of merit of the different cores is different at different speeds. At 120 miles per hour the flat tubes E-8 and E-6 are seen to be best and the finned circular tubes F-5 the poorest.

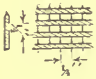
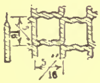
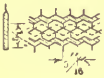
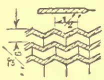
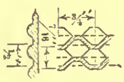
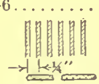
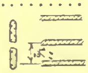
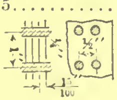
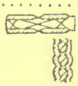
Table 19 gives the constants in the empirical equations for "Head Resistance," "Mass Flow" and "Energy Dissipated" for the cores listed in Table 17; these quantities are obtained from the data of Table 18. The Head Resistance Constant and the Mass Flow Constant appear to be connected by a simple relation; plotting these quantities for all the cores tested gives the curve of Fig. 272.

TABLE 17.—DIMENSIONS OF TYPICAL RADIATOR CORES

Number	A-7	A-23	B-8	C-4	D-1	E-6	E-8	F-5	G-3
Depth, inches	4⅞	5	5¼	2⅞	2⅝	9¾	9¾	3⅛	3⅜
Metal:									
Water tubes	Brass	Brass	Brass	Brass	Copper	Copper	Copper	Copper	Copper
Fins	....	....	....	....	Brass	....	....	Brass	
Thickness of metal, inches	0.005	0.005	0.005	0.005	0.010	0.004	0.004	0.010	0.006
Weight, pounds per square foot of frontal area:									
Empty	17.1	15.6	13.9	9.2	8.5	16.8	9.2	7.6	11.0
Water content	7.2	6.3	4.2	3.9	2.2	9.3	5.0	1.3	2.9
Filled	24.3	21.9	18.1	13.1	10.7	26.1	14.2	8.9	13.9
Per cent free area	64.2	68.5	73.3	62.6	75.8	75.0	87.5	....	72.0
Cooling surface, square feet per square foot front	62.6	54.5	56.1	32.0	23.7	78.5	39.2	34.5	44.7
Per cent direct cooling surface	52.2	91.8	30.3	57.2	46.8	100.0	100.0	12.3	42.9
Water tubes:									
Length, inches per foot of core	12.0	24.0	13.9	13.8	12.0	12.0	12.0	12.0	13.1
Width, inches	0.078	0.0402	0.0822	0.073	0.074	0.0525	0.0525	0.235	0.069
Depth, inches	4.45	4.64	4.84	2.50	2.20	9.74	9.74	0.235	1.30
Number per foot	48	39	21	43	29	48	24	72	68
Area normal to flow, square feet per foot	0.1156	0.0505	0.0597	0.0546	0.0348	0.170	0.085	0.0216	0.0212
Hydraulic radius, inches	0.038	0.0199	0.040	0.0355	0.038	0.0108	0.0108	0.059	0.033
Air tubes:									
Hydraulic radius, inches	0.050	0.064	0.069	0.062	....	0.0938	0.218		
Ratio, length ÷ hydraulic radius	97.2	78.1	76.2	47.1	....	104.0	44.7		

TABLE 18.—RADIATOR PERFORMANCE IN TERMS OF FREE AIR SPEED (FOR UNOBSTRUCTED POSITIONS ONLY)

Grade A represents very good performance; grade E, very poor

Radiator	Speed, miles per hour	Air flow, lb. per sq. ft. per sec.	Energy in h.p. dissipated per square foot of		Head resistance, lb. per sq. ft. frontal area	H.p. absorbed per sq. ft. of frontal area	Figure of merit
			Front	Surface			
A-7.......	30	2.20	27.2	0.43	1.72	.50	54.6
	60	4.40	45.9	0.73	6.70	1.79	25.7
	90	6.60	61.7	0.99	14.90	4.66	13.2
	120	8.80	76.5	1.22	26.3	9.87	7.8
Grade.........	...	.....	A	....	D	D	B
A-23............	30	2.29	20.3	0.37	1.40	0.44	46.6
	60	4.58	37.2	0.68	5.61	1.54	24.1
	90	6.87	52.9	0.97	12.10	3.88	13.6
	120	9.16	68.0	1.25	21.3	8.11	8.4
Grade.........	...	.....	B	....	C	C	B
-8............	30	2.12	20.2	0.36	1.72	0.41	49.8
	60	4.24	31.0	0.55	6.88	1.64	18.9
	90	6.36	39.8	0.71	15.48	4.52	8.8
	120	8.48	47.9	0.85	27.5	9.89	4.8
Grade.........	...	.....	D	....	D	D	D
C-4............	30	2.40	14.8	0.46	1.37	0.30	48.7
	60	4.80	26.4	0.83	5.47	1.26	20.9
	90	7.20	37.8	1.18	12.30	3.53	10.7
	120	9.60	48.5	1.52	21.9	7.79	6.2
Grade.........	...	.....	D	....	C	C	C
D-1............	30	2.41	13.8	0.58	1.25	0.26	53.5
	60	4.82	24.7	1.04	4.75	1.08	23.0
	90	7.22	32.3	1.32	10.48	2.99	10.8
	120	9.63	38.7	1.63	18.4	6.53	5.9
Grade...........	...	.....	D	....	B	B	C
E-6............	30	2.12	29.3	0.37	1.57	0.51	57.2
	60	4.24	51.1	0.65	6.27	1.78	28.8
	90	6.36	71.3	0.91	14.10	4.54	15.7
	120	8.48	90.5	1.15	25.1	9.59	9.5
Grade.........	...	.....	A	....	D	D	B
E-8............	30	2.74	17.2	0.44	0.78	0.27	63.3
	60	5.48	29.8	0.76	3.12	0.92	32.4
	90	8.23	41.0	1.05	7.03	2.31	17.8
	120	10.97	51.7	1.32	12.5	4.84	10.7
Grade.........	...	.....	C	....	A	A	A
F-5............	30	1.82	13.8	0.40	2.52	0.33	41.3
	60	3.64	21.7	0.63	9.65	1.81	12.0
	90	5.46	27.8	0.81	21.6	5.57	5.0
	120	7.28	33.0	0.96	38.3	12.8	2.6
Grade.........	...	.....	E	....	E	E	E
G-3............	30	1.88	23.1	0.52	2.40	0.40	58.0
	60	3.75	39.5	0.88	8.65	1.80	22.0
	90	5.62	53.6	1.20	19.1	5.21	10.3
	120	7.50	67.0	1.50	33.4	11.5	5.8
Grade.........	...	.....	B	....	E	E	C

A plotting of the data in Table 18 for core E-8 is given in Fig. 273.

Size of Radiator.—If a type of core has been selected and its properties are known and plotted as in Fig. 273, the necessary size of an unobstructed core is directly obtainable. The dimensions must be calculated for the most unfavorable condition, which, ordinarily, will be maximum climbing speed near the ground with summer temperatures. The mean water temperature will be fixed by the maximum temperature allowable in the jackets and by the quantity of water circulated. The

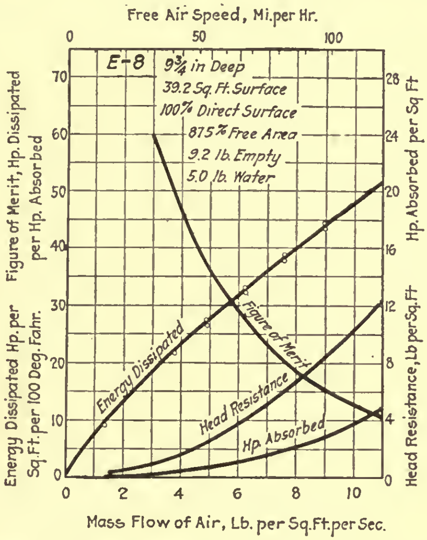


Fig. 273.—Properties of a flat tube radiator core.

maximum allowable water temperature is ordinarily 20 to 30°F. below the boiling point and varies with the altitude of the plane; its value will be determined by (1) its influence on the volumetric efficiency of the engine (see p. 37) and (2) the water resistance of the radiator (see p. 364). If the water system is closed, with no vent to the atmosphere, the last-mentioned factor disappears. The heat to be dissipated should be determined if possible but may be assumed equal to the brake work of the engine if more exact knowledge is not obtainable. The effect of propeller slip should be estimated and allowed for. Allowance should also be made for the cooling effect of the radiator headers and of the exposure of the engine to the wind.

Occasionally, instead of designing for maximum climb some other condition may impose maximum service on the radiator as, for example, in flying boats and seaplanes intended for training, where much taxi-ing is done at low plane speed and maximum engine power. If the radiator is in the nose of the fuselage some assumption must be made as to the relation of mass flow to the speed of the plane. The mass flow will usually vary from 0.04 to 0.07 time the speed of the plane (in miles per hour), depending on the type of radiator, the amount of cooling and the masking effect of the propeller. The power absorbed is seldom calculable because of the uncertain effect of the radiator on the resistance of the fuselage.

TABLE 19.—CONSTANTS IN THE EQUATIONS $R = cV^2$; $M = kV$; AND $Q = Gm^n$

R = Head resistance in pounds per square foot.
V = Free-air speed in miles per hour.
M = Mass flow of air in pounds per second per square foot.
Q = Energy dissipated in horsepower per square foot per 100°F. temperature difference.
m = "mass flow constant," which is the ratio of the mass of air passing through 1 square foot of radiator to the mass of air passing through 1 square foot of free area in front of the radiator.

Radiator	$c \times 10^3$	$k \times 10^2$	m	G	n
A-7	1.86	7.34	0.667	15.1	0.75
A-23	1.56	7.63	0.694	10.3	0.85
B-8	1.91	7.07	0.643	13.1	0.60
C-4	1.52	8.00	0.727	7.1	0.85
D-1	1.32	8.03	0.730	8.1	0.70
E-6	1.74	7.07	0.643	16.1	0.80
E-8	0.867	9.13	0.830	7.6	0.80
F-5	2.68	6.07	0.552	10 0	0.60
G-3	2.40	6.25	0 568	14.8	0.75

The mass flow for a wing radiator depends on the angle of incidence but is probably not over 0.01 time the plane speed even at the best climbing angle.

The relative efficiencies of radiators in various positions are given by Liptrot[1] as follows:

[1] *Aeronautics*, Apr. 29, 1920.

Position of radiator	Relative efficiency
Unobstructed	1.000
Underslung, side or overhead, but close to fuselage	0.973
Twin nose radiator	0.716
Nose radiator with core entirely above or below propeller shaft	0.656
Nose radiator with propeller shaft in center	0.585
Behind engine	0.423

The use of a small projecting lip or stream line entrance around the core may reduce the necessary core size slightly but at the cost of a considerable increase of head resistance.

Rate of Water Flow.—One gallon (231 cu. in.) of water at 200°F. weighs $\frac{60.12 \times 231}{1{,}728} = 8$ lb. approximately. With a temperature difference of 10°F., 1 gal. of water per minute will give up 80 B.t.u. or $\frac{80}{42.45} = 2$ h.p. approximately. With a temperature difference of 5°F. the flow of water in gallons per minute should equal the engine horse power.

The entering temperature of the water is fixed by the necessity of keeping at a certain point below boiling. With fixed entering temperature, if the amount of water circulated is increased the mean temperature of the water is raised and consequently the temperature difference between air and water is increased. The influence of water velocity on the heat transfer is found by experiment to be very small so long as the velocity is above 2 gal. per minute per foot width per inch depth of core, which is much below usual rates. With a circulation of ¼ gal. per minute per horse power the temperature fall of the water is 20°F.; increasing this to ½ gal. reduces the temperature fall of the water to 10°F. and increases the temperature difference between air and water by 5°F. With an infinite amount of water circulated this temperature difference could be increased only another 5°F. The increase in pump work with increased water flow makes it undesirable to circulate more than about ½ gal. per minute per horse power, and with radiators that are relatively long and narrow a flow of ¼ gal. per minute per horse power should be used.

The **pressures** required to **maintain water flow** through the cores of radiators vary greatly with the dimensions and type of construction. Those types having the widest and straightest water spaces offer least resistance whereas those with many right angle bends will offer much resistance. The range for 12 commercial radiators tested at the Bureau of Standards, all of them 8 in. square in frontal section and of depths varying from 2⅝ to 4 in. with a total water flow of 20 gal. per minute, was from 0.27 to 10.2 ft. of water pressure drop. These pressure drops may be assumed to vary directly as the height of the core, but the rate of change with change of water velocity follows an exponential law in all cases, though with a widely varying exponent in the different types. The resistance seems to depend largely on the care used in manufacture and on the form of the water tube entrances and exits. It would seem well to include a test for pressure necessary to produce water flow in acceptance specifications for complete radiators.

The water enters the top header of the radiator, at which place atmospheric pressure is usually maintained through the overflow pipe. The suction pressure at the pump cannot be less than the vapor pressure of the water leaving the radiator if the pump and radiator are at the same level. If the water leaves the radiator at 190°F. the corresponding vapor pressure is 9.2 lb. or about 5 lb. below atmospheric pressure. The maximum pressure available for overcoming the resistance of the radiator in this case will be 5 lb. per square inch or 11.5 ft. of water. With a reserve tank in the upper plane, as in Fig. 274, the head available in overcoming radiator friction is increased by the height of the tank above the suction. If the resistance of a proposed radiator is in excess of the available pressure, its height must be decreased and its width correspondingly increased in order to give the necessary radiating surface.

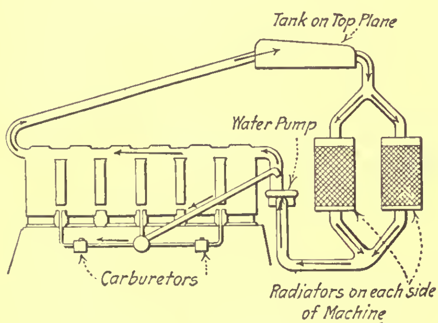


FIG. 274.—Cooling system of Benz engine.

Occasionally radiation or expansion tanks instead of being vented to the atmosphere are provided with safety valves opening at 2 or 3 lb. per square inch. This diminishes the loss of water from evaporation and may permit a higher water temperature.

Effect of Altitude on Radiator Performance.—The investigations at the Bureau of Standards have yielded the following general conclusions:

The effect of the lower air temperature is to increase the heat transfer in proportion to the increase in the mean temperature difference between the entering air and the water. The decrease in air density reduces the mass flow of air and decreases the heat transfer at any given plane speed in proportion to the air density.

Head resistance is proportional to air density and is therefore reduced with increased altitude. The combined effect of temperature and density changes is to decrease the heat transfer but not as rapidly as the engine power diminishes; consequently the cooling capacity of the radiator becomes excessive at high altitudes and may be more than double the required capacity. As the head resistance falls off more rapidly than the heat transfer the "figure of merit" of the radiator increases with altitude.

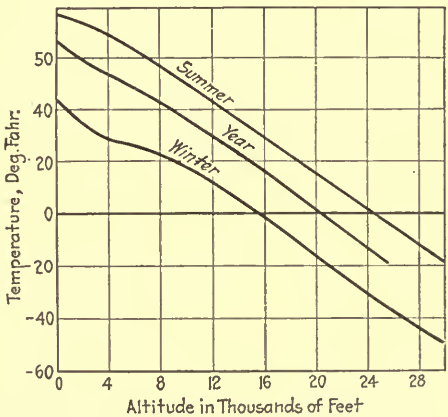


FIG. 275.—Variation of air temperatures with altitude.

From the above conclusions the performance of a radiator at any altitude can be calculated when its ground performance is known. For example, take the flat plate core (E-8) for which ground data are given in Tables 17 and 18. It is desired to calculate its performance in summer at 10,000 ft. altitude and 120 miles per hour. The ground data are:

Mass flow of air at 120 miles per hour = 10.97 lb. per square foot per second.

Head resistance at 120 miles per hour = 12.5 lb. per square foot.

Weight of core and contained water = 14.15 lb. per square foot.

The mean temperature of the water in the radiator may be assumed to be 30°F. below the boiling point. The pressure at 10,000 ft. is 10.2 lb. per square inch (see p. 389) and the corresponding boiling point is 194.2°F. The summer mean temperature at 10,000 ft. may be taken as 45°F. (Fig. 275). The mean temperature difference at 10,000 ft. will be 194.2 − 30 − 45 = 119.2°F. The air density at the same elevation is 0.0545 lb. per cubic foot (see Fig. 276). The mass flow at 10,000 ft. $= 10.97 \times \frac{0.0545}{0.0750} =$ 7.98 lb. per square foot per second. The energy dissipated at mass flow of 7.98 lb. is 40 h.p. per square foot per 100° F. temperature difference (see Table 18); with the increased temperature difference the energy dissipated becomes $40 \times \frac{119.2}{100} = 47.7$ h.p. per square foot. The head resistance (see Table 18) $= 12.5 \times \frac{0.0545}{0.0750} = 9.09$ lb. per square foot.

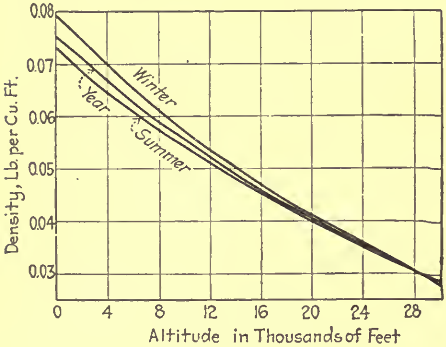


Fig. 276.—Variation of air densities with altitude.

The degree of masking required at altitudes may be readily calculated if the engine h.p. is assumed proportional to the air density. If the radiator is just adequate in level flight at a given speed at the ground, it will be capable of more cooling than is required of it in level flight at the same speed at higher altitudes. It is therefore possible to mask an increasing fraction (and cut down thereby the mass flow) as altitude increases. The curve of Fig. 277 shows how much masking is possible for the flat plate radiator E-8 at 120 miles per hour, but the curve is practically the same for other cores and speeds.

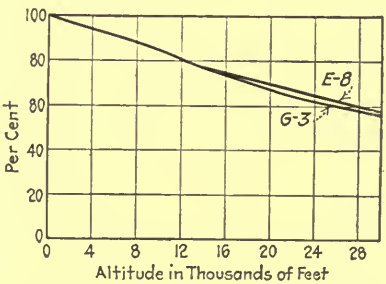


Fig. 277.—Radiator masking at altitudes.

It should be remembered that climbing should be considered as well as level flight in any discussion of radiators and of masking. The speed for maximum climb may be only one-half that of level flight at certain altitudes, and the cooling must be adequate for the climbing condition. This consideration alone would require a masking of 50 per cent for such planes in level flight. If the relation between maximum climbing speed and level speed is known, and also the change in engine revolutions and power, the mass flow of air can be determined under both conditions and the desirable degree of masking can be found.

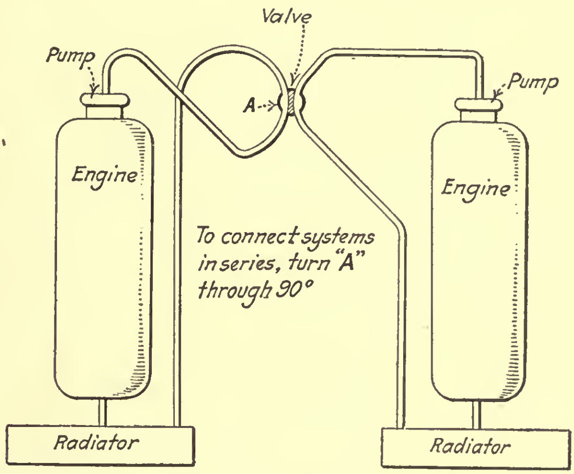


FIG. 278.—Radiator interconnections for dual engines on lighter-than-air machines.

The twin-engined dirigible offers a special case of importance. Such a craft may operate for long periods with one engine only, which therefore operates at low speed but full power. If the radiator is designed for maximum speed each radiator will be too small for its engine at this reduced speed. To obviate the use of a larger radiator the installation may be arranged as in Fig. 278 in case the water pumps are of such construction as to permit the water to pass through when they are idle. Turning the valve *A* through 90 deg. will circulate the water through both radiators and through the jackets of both engines, and will thereby prevent the idle engine from freezing up and will give more than adequate radiating surface. Some provision for masking the radiator is especially desirable in this case.

Masking can be partially accomplished by varying the water flow, as by by-passing some of the water from radiator inlet to outlet. The effect of reducing the quantity of water is to reduce the mean temperature of the water and thereby to reduce the mean temperature difference between air and water. The possible range of control by this means is small. Shutters across the radiator front answer the purpose more fully, although they add to the head resistance. They may be operated by the pilot, or as in some German planes, may be under the automatic control of an electrical resistance thermometer. Closed shutters on a nose radiator decrease the head resistance: on a free air radiator, they increase it. A retractable side or bottom radiator, which may be drawn within the body to decrease the cooling effect, is occasionally used. It may be arranged most conveniently as an auxiliary radiator in series with a fixed main radiator which has no masking device and is adequate for high-altitude level flight. The auxiliary radiator is retracted as altitude is gained. The increased water resistance from two radiators in series is objectionable. Yawing is another possibility.

Effects of Yawing Airplane Radiators.—The air stream does not always approach the radiator at right angles to its face. The most common causes of this are:

1. Radiator mounted in the propeller slip stream where the air strikes the radiator at angles other than normal to its face.
2. Radiator mounted in the wing (or other position) where the axes of its passages for the air are not parallel to the direction of motion of the plane.
3. Radiator pivoted about an axis perpendicular to the direction of motion of the airplane for the purpose of changing its inclination for the regulation of cooling capacity (masking).

The effects of yawing a radiator through angles from 0 to 45 deg. are (1) to decrease slightly the mass flow; (2) increase the head resistance by as much as 50 per cent in the case of cores of low head resistance but much less in the case of high-resistance cores; and (3) in some cases, for angles up to 20 or 25 deg. to increase slightly the heat transfer. These effects vary largely with different types.

The **complete radiator** consists not only of the core but of top and bottom headers. The top header may serve merely as a distributor or it may have sufficient capacity to serve as reserve and expansion tank also. The latter practice reduces complications and is therefore used on small machines intended for short flights. For large machines used for long flights, an adequate water capacity would entail a large frontal surface and excessive head resistance of the header. The desirable reserve capacity in British practice is given by the formula:

$$\text{Gallons} = \frac{\text{h.p.} \times (\text{endurance in hours})}{1{,}600}$$

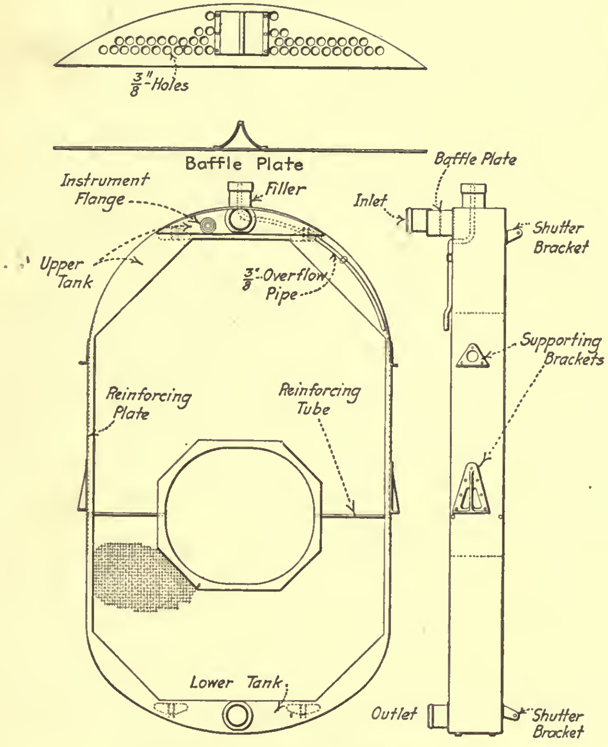


FIG. 279.—Details of typical nose radiator.

The reserve water tank is often located in the upper wing but there is danger of freezing unless, as in Fig. 274, the water circulation is through the tank; the objection to including it in the circulation is the increased length of pipe through which the water has to be forced.

The lower header is a collector only and should be as small as practicable. Both headers should be stream-lined. The headers and their contents will usually add 50 per cent to the weight of the core and its contents. Occasionally (as in the Maybach

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plant) the headers are divided into halves by vertical baffles on the fore and aft line. Water enters the left-hand side of the lower header, passes to the left-hand side of the upper header, then over the baffle to the right-hand side and down to its exit at the right-hand side of the lower baffle; this arrangement causes greatly increased water resistance if the same weight of water is circulated; if the weight of water is halved so as to maintain the same velocity in the radiator passage, the mean temperature difference between air and water will be diminished, necessitating the use of a large radiator.

A complete nose radiator is shown in Fig. 279. Among the details to be noted are the filler, inlet and outlet pipes; the perforated baffle plate between the inlet and the upper tank; the overflow pipe; the upper and lower supporting brackets; and the shutter brackets. The filler cap is of hard rubber with a safety chain; a better construction, avoiding loss from the snapping of the chain, is with a hinged cap held closed by a snap wire.

Pumps.—As previously pointed out (p. 363) the cooling water required is not more than ½ gal. per brake horse power per minute. Ordinarily it is ¼ gal. per minute or less. The resistance to the circulation of the water is chiefly in the radiator, but is considerable in other parts of the system; its magnitude is variable, but may be assumed to be from 4 to 8 lb. per square inch in good installations.

The **water horse power** of the pump of a 100-h.p. engine using ¼ gal. (2 lb.) of water per horse power per minute against 8 lb. per square inch pressure is $8 \times 144 \times \frac{2 \times 100}{60} \times \frac{1}{33,000} =$ 0.116 h.p. If the efficiency of the pump and its drive is 20 per cent, the horse power used to drive the pump will be $0.166 \div 0.2 = 0.58$ h.p., which is a very small fraction of 100 h.p. Consequently, the water pump efficiency is comparatively unimportant and the type selected should be one of maximum simplicity and minimum weight. The single impeller volute centrifugal pump meets these conditions best and is universally used.

In a volute pump, water enters axially, is caught by the impeller blades and is given a high velocity of rotation before it is discharged into the volute casing, from which it escapes through one or more outlets. The number of outlets is usually the same as the number of banks of cylinders. In order to keep down the size and weight of the pump the impeller rotates at a speed greater than that of the engine; one and one-half engine speed is common. If the impeller blades are radial (Fig. 280) the theoretical discharge pressure in feet of water is given by $V^2/2g$ where V is the tip speed of the impeller. Taking an impeller diameter of 4 in. and a speed of 2,400 r.p.m. this becomes $\left(\pi \times \frac{4}{12} \times \frac{2{,}400}{60}\right)^2 \div 2g = 32$ ft. = 13.8 lb. per square inch. The water velocity leaving the impeller cannot be converted completely into pressure head and there are various impeller and casing losses, so that the

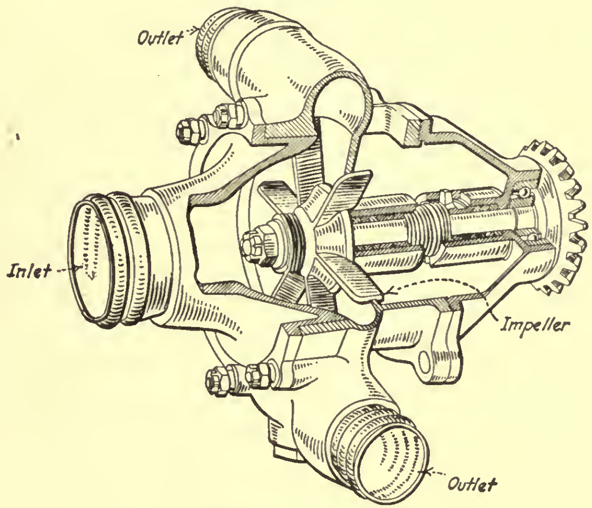


FIG. 280.—Water pump of Liberty engine.

actual discharge pressure will be much less than that calculated above; it would probably be less than one-half the theoretical value.

The resistance to be overcome by the pump is entirely frictional, and varies as the square of the amount of water circulated. The amount of water circulated is proportional to the speed of the pump. The work done by the pump is proportional to the volume of water circulated multiplied by the resistance, or is proportional to the cube of the pump speed.

The pump of the **Liberty engine,** Fig. 280, has a 2-in. inlet and two outlets. It runs at 1½ times engine speed, and has a capacity of 86 gal. per minute at 2,000 r.p.m. of the engine. The impellers are radial and are partly shrouded. The packing of the impeller shaft against water leakage is kept compressed by a coiled spring. The **King-Bugatti** (410 h.p.) pump (Fig. 281) has impellers completely shrouded on one side. As there is only one outlet the casing is of complete volute form. The impeller is 5⅞ in. diameter with eight vanes, the web being drilled to equalize the water pressure. The shaft is packed with graphited asbestos rope packing, held under compression by a coiled spring. The inlet is 2¼ in. in diameter; the outlet is 2³⁄₁₆ in. It is coupled direct to the engine shaft.

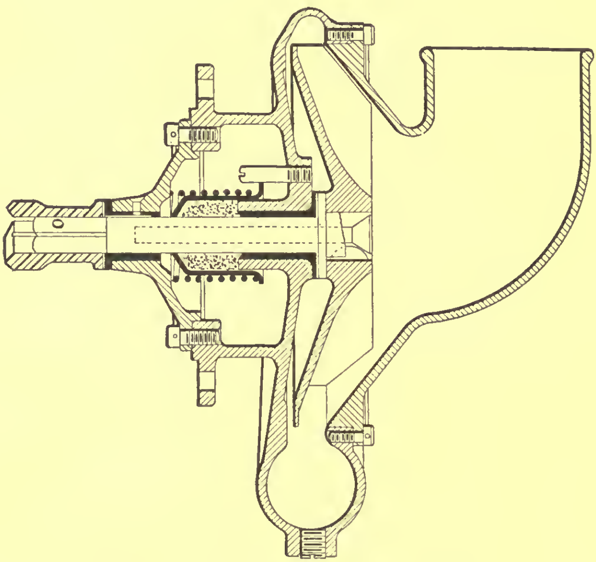


Fig. 281.—Water pump of Bugatti engine.

The **Austro-Daimler** (200 h.p.) pump weighs 7.6 lb., has an impeller 4.4. in. in diameter, inlet and outlet diameters 1.42 in., and a ratio of pump to engine speed of 1.89. It is driven from the rear end of the crankshaft by a bevel gear which is integral with a sleeve forming an extension shaft (Fig. 282). The pump bevel gear floats on the end of the pump spindle, and is fitted with a large-diameter thrust ball-race and retaining spring, which, being at the bottom end of the spindle, are as far away as possible from the impeller. Both the pump spindle bearings are lubricated through two holes drilled in the pump body and oil grooves cut in the spindle bearings. The impeller is formed with six vanes and is completely shrouded; it is keyed to the spindle and secured by a gun-metal nut and washer. A conically-faced shoulder is

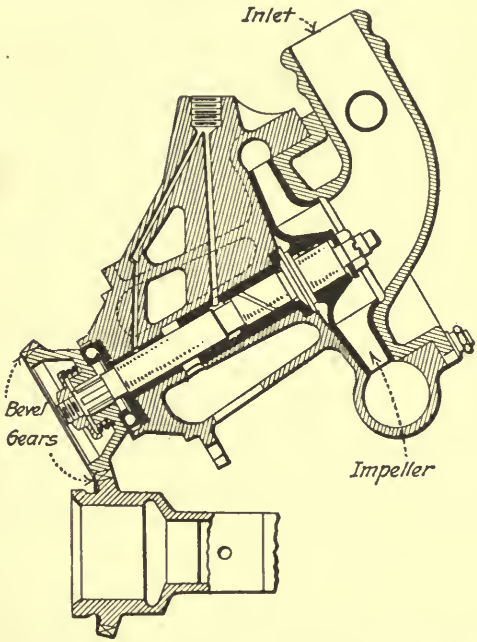


FIG. 282.—Water pump of Austro-Daimler engine.

machined on the pump directly beneath the impeller. This shoulder beds into the bevelled face of the bronze bearing, form-

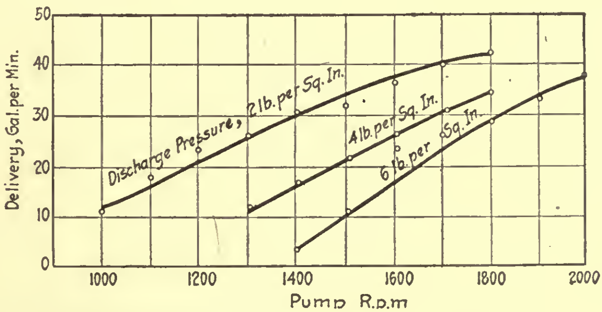


FIG. 283.—Performance curves of Austro-Daimler water pump.

ing a water-tight joint. The performance curves of this pump are given in Fig. 283.

The **Maybach** (300 h.p.) pump (Fig. 284) has an impeller 4.46 in. in diameter, inlet 2.13 in. in diameter, outlet 1.97 in. in diameter, and a ratio of pump to engine speed of 2. The pump spindle is driven through a dog clutch at its lower end by a short vertical spindle running in a bronze bushing; this spindle is driven by a bevel gear meshing with the main bevel fixed on the rear end of the crankshaft. The top portion of the pump spindle bearing is cupped to form the housing for a thrust ball-race, above which is fixed the impeller. The impeller is a gun-metal casting, having six helical vanes. The lower half of the pump body is an aluminum casting, to the inlet passage of which the diagonal water pipe from the radiator is coupled by a rubber connection. The top half of the water pump body, which is a gun-metal casting, is formed with six helical passages leading in

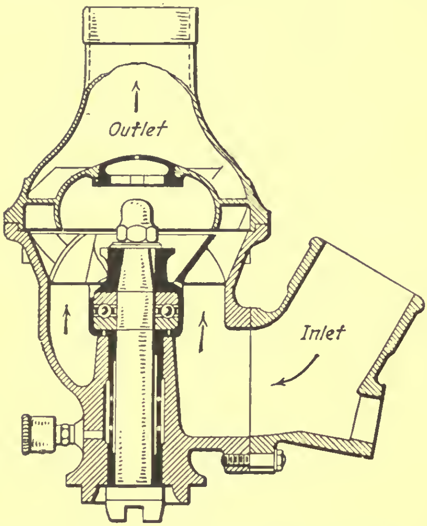


Fig. 284.—Water pump of Maybach engine.

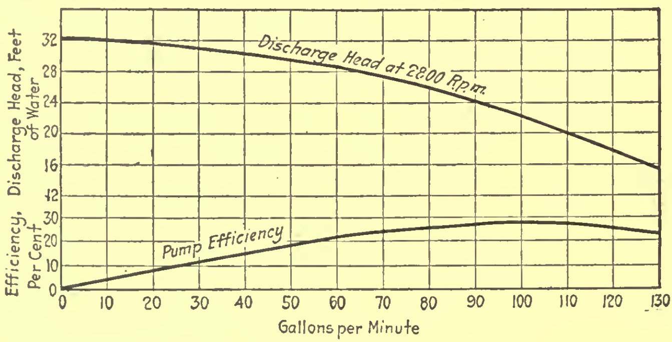


Fig. 285.—Performance curves of Maybach water pump.

a reverse helical direction to the impeller. These passages connect with the common vertical outlet passage in the top of the body casting. The center portion of the top body casting, inside the helical passages above the impeller, is domed and fitted with a screwed plug. This plug is drilled with a small hole, to prevent an air-lock. Two other holes are also drilled in the bottom of the impeller between the vanes for the same purpose. The steel ball thrust race is exposed to the flow of water, a disadvantageous feature. Performance and efficiency curves for this pump at 2800 r.p.m. are given in Fig. 285. It will be seen that the maximum pump efficiency of 28 per cent is obtained with a discharge head of about 22 ft. of water and a capacity of 100 gal. per minute.

Piping.—Water velocities in pipes vary from about 8 ft. per second in small engines to 16 ft. per second in large engines. Actual pipes sizes are from $1\frac{1}{4}$ in. diameter for 90 h.p. to 2 in. diameter for 400 h.p.

The frictional resistance to flow of water through straight pipes is given by $h = 4f\ (l/d)\ (V^2/2g)$ where h is the loss of head in feet, l and d are the length and diameter of the pipe respectively in feet, V the velocity in feet per second, and f is a coefficient whose value is likely to vary from 0.004 to 0.010, depending on the roughness of the pipe. Inlets, outlets and bends will each offer a resistance equivalent to a length of 10 to 20 diameters.

Large pipe sizes diminish the resistance and work of the pump but they weigh more and hold more water. The pump suction should be of ample diameter and as short and direct as possible. The connecting rubber hose should be firm and non-collapsible. Pipe lines should be of light tubing, bent to easy radii, with a minimum of bends and fittings. Hose connections at junction points should be very short, and should fit over corrugations. The fastenings should be by smoothly-bearing steel clamps which do not cut the rubber. Tape should be applied over hose and clamp and the whole shellacked. The pipes should be arranged to avoid air pockets if possible; if such occur, vent cocks must be applied. Particular care must be given to the vent cock on the pump casing.

Water.—The water used should be free from lime. Filling the system with boiling water makes starting easy in cold weather. Anti-freezing solutions are all more or less objectionable, and it is best to drain the system when the plane is not in use. Fig. 286 shows the properties of some anti-freezing mixtures. Alcohol lowers the boiling point and makes close control of temperature essential; the strength decreases and the freezing point is elevated

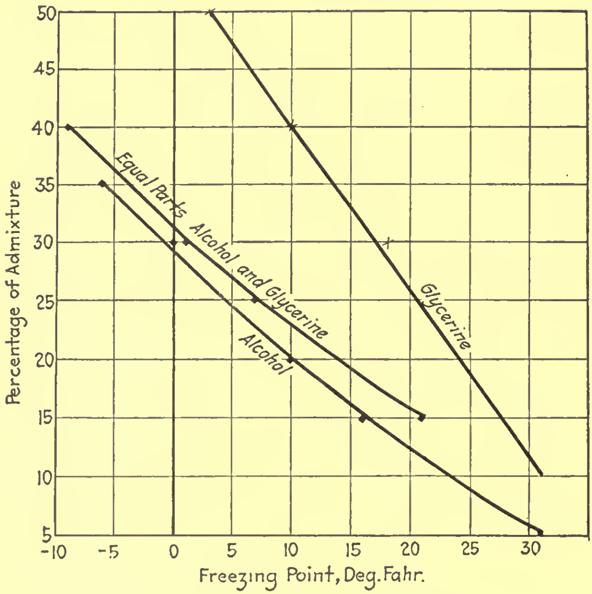


FIG. 286.—Properties of anti-freezing mixtures.

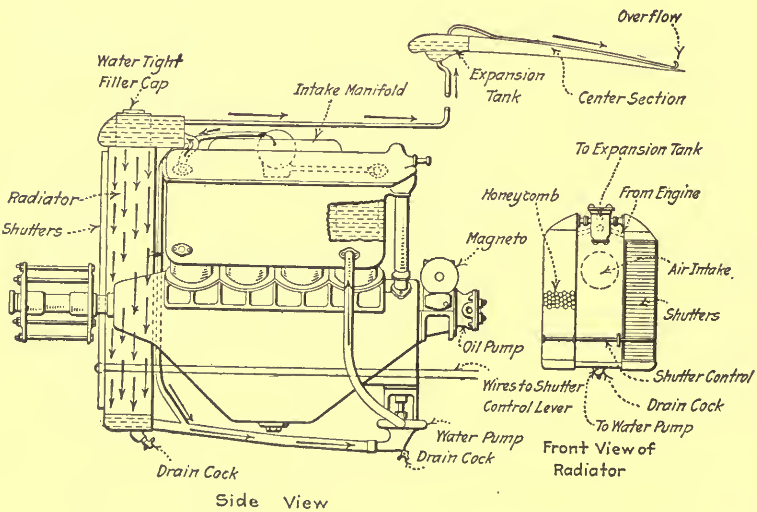


FIG. 287.—Cooling plant of S E-5 airplane.

as the alcohol evaporates. Periodical tests by hydrometer are advisable. Glycerine does not evaporate, but impairs circulation and is detrimental to rubber. The glycerine should be stirred slowly into the water.

Typical complete cooling systems are shown in Figs. 287 and 288. Figure 287 is for a 180 h.p. Hispano-Suiza engine in a SE-5 plane with two tubular nose radiators at the sides of the engine shaft, each 30 by 7½ by 3$^{15}/_{16}$ in., with a total frontal area of 450 sq. in. and a radiating surface of 12,700 sq. in. The water capacity is 83½ lb. and the flow rate 30 gal. per minute. The system is provided with an expansion tank which occupies the

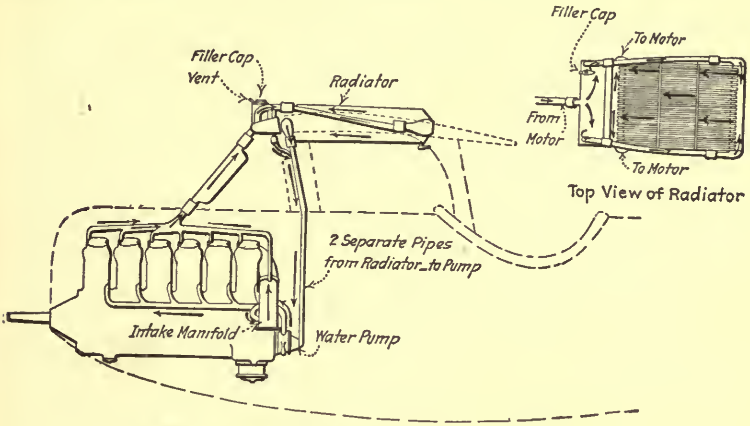


Fig. 288.—Cooling plant of Le Pere airplane.

leading section of the middle panel of the upper wing. A small portion of the water leaving the cylinders passes around the intake manifold and is then returned to the pump; the rest of it goes through the radiator. The radiator is masked by shutters.

Figure 288 shows a 360-h.p. Liberty engine in a Le Pere two-seater plane with a wing radiator in the center section of the middle panel of the upper wing. The radiator is 31 in. long, 27 in. wide and 7 in. deep; has a frontal area of 783 sq. in. and a radiating surface of 35,520 sq. in. The water capacity is 41.6 lb. and the flow 80 gal. per minute Free water area 61.6 sq. in., free air area 1,247 sq. in., weight 127 lb. The water pumped around the manifold goes to the radiator before returning to the pump.

CHAPTER XV

GEARED PROPELLER DRIVES

A well designed airplane engine develops its maximum power at a speed (r.p.m.) considerably in excess of the most efficient propeller speed. In order to combine maximum power development with most efficient utilization of that power it is necessary to resort to a geared drive.

Geared drives have been employed in a number of successful installations. A German analysis of these[1] is the basis for the discussion which follows. The simplest type is a **single-reduction** with spur gears as in Figs. 289 and 290. In the Renault engine (Fig. 289), the gear ratio is two to one and consequently can be used for driving both camshaft and propeller shaft; in the Hispano-Suiza engine (Fig. 290) the gear ratio is four to three. Gears of this type show heavy wear. A design for a single reduction with internal gear is shown in Fig. 291; the internal gear housing is attached to the crankcase by an eccentric centering flange which permits accurate adjustment of the gears. This type permits great simplicity but there is difficulty in arranging satisfactory bearings on both sides of the gear wheels.

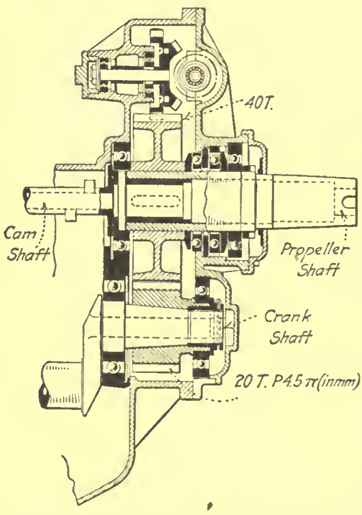


FIG. 289.—Renault single-reduction-gear.

With single-reduction gears the propeller shaft cannot be in the same axial line with the crankshaft; when this arrangement is desired **double-reduction gears** must be used. There are many possible arrangements; both pairs of wheels may be fitted with internal or external gears and in addition any one of the three shafts may be fixed while the other two drive and are driven respectively. Some of these arrangements are shown

[1] KUTZBACH: *Technische Berichte, Vol. III, Sec.* 3.

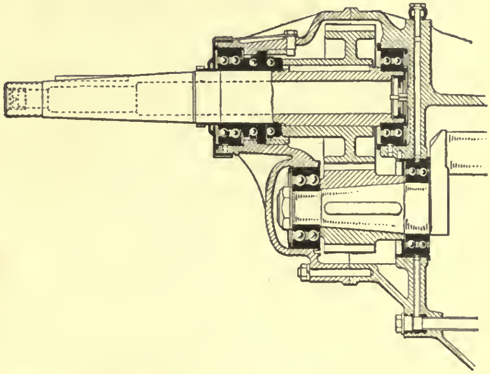


FIG. 290.—Hispano-Suiza single-reduction-gear.

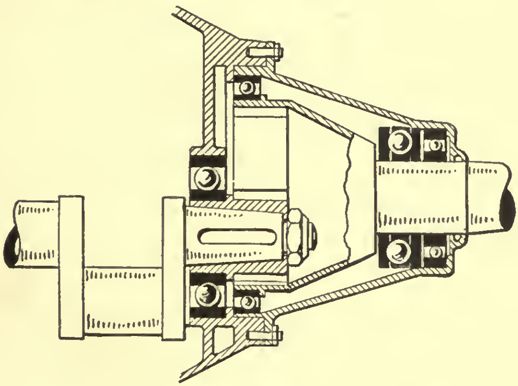


FIG. 291.—Single-reduction internal gear.

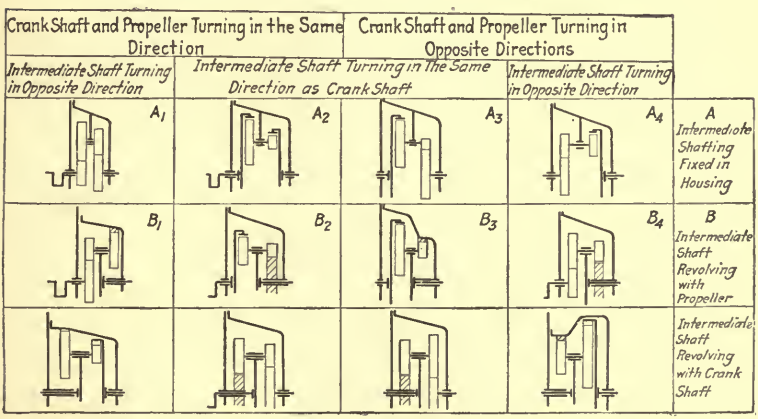


FIG. 292.—Possible arrangements of double-reduction gears.

schematically in Fig. 292. In the top row the intermediate shaft is fixed; in the second and bottom rows it revolves forming the so-called planetary gears. The shaded gears are fixed and do not revolve. Some of these arrangements offer considerable difficulties for actual construction, notably in the matter of providing suitable bearings on both sides of the gears; in others the space occupied may be great and the revolutions of the intermediate shaft very high.

A simpler arrangement is one in which both pairs of gears have one gear in common. Schematic outlines of such reductions are shown in Figs. 293, 294 and 295. Figure 293 is developed from A_1. Fig. 292; Fig. 294 from A_4; and Fig. 295 from B_2. The Rolls-Royce planetary gear, Fig. 296, is an actual construction of Fig. 295. The three revolving intermediate shafts are

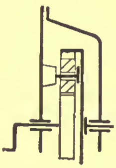


Fig. 293.

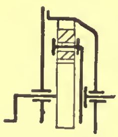


Fig. 294.

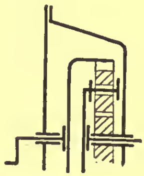


Fig. 295.

Double-reduction gears with a common gear.

carried in a spider, C. The internal gear, a, on the crankshaft drives the three gears, b, on the intermediate shafts, and the three gears, c, on the same shaft mesh with the gear, d, which is held against revolving in the housing. The spider, C, revolves and carries the propeller shaft.

The advantage of the double-reduction gear over the much simpler single-reduction gear lies in the perfectly axial transmission of the power, from which the best condition of loading of the housing (pure torsion) is obtained. When the power is transmitted through two, three or four intermediate gears at equal angles, springing of the gear shafts from unequal peripheral forces or inaccurate tooth forms is avoided. Certain arrangements also make it possible to use heavy revolving masses (for instance, those of the intermediate shafts or the larger internal gears), thereby improving the uniformity of transmission and avoiding reversals of tooth pressure. The principal advantage, however, consists in the fact that on account of the load being divided

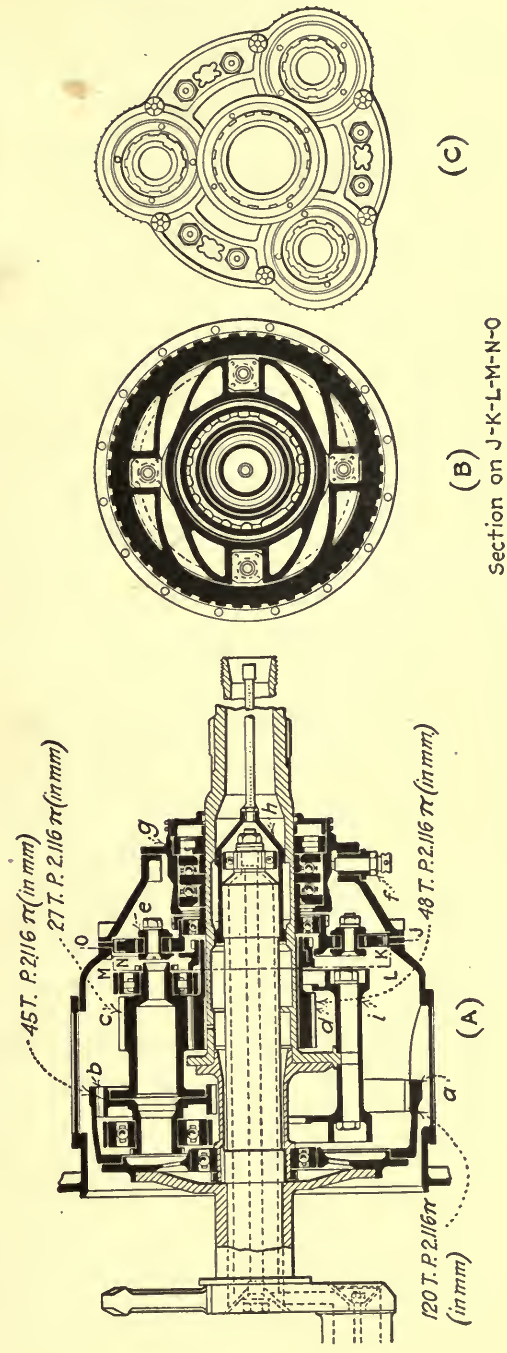


FIG. 296.—Rolls-Royce planetary gear.

between two to four intermediate gears the tooth pressures per unit of tooth face are low. Consequently small pitches and small gears can be used which in turn have smaller construction defects since the defects of manufacture resulting from the use of inaccurate dividing wheels increase with increasing radius. The disadvantage of the double reduction gear is its weight and cost and the need for exact adjustment of the intermediate shafts if all the gears are to work equally. Furthermore, a complicated construction is necessary to ensure a solid and secure assembly of the gear.

To obtain and keep proper **adjustment of the reduction gearing** as wear occurs, it is necessary to fit a joint between the crank case and the gear case, or, in the transmission, between crankshaft and

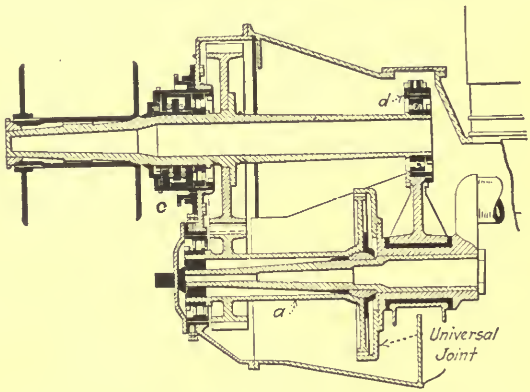


Fig. 297.—Rolls-Royce single-reduction gear.

gear, which will adjust itself automatically while running or can be adjusted in assembly. In the Rolls-Royce planetary gear, Fig. 296, a sliding cross linkage is used in a fixed housing. The link, (*B*) and *e*, lies between the outer engine housing and the intermediate gear wheel, *d*, which is held in the housing. Consequently the gear wheel, *d*, can adjust itself and always remains concentric with the crankshaft. The whole set of planetary gears also remains concentric with the crankshaft—which may shift in the casing—but not with the casing. The forward bearing, *g* and *h*, must be adjusted on each overhaul of the engine by the screws, *f*.

In the Rolls-Royce spur gear, Fig. 297, the upper gear can be adjusted by eccentrically-set ball-bearing cages, *c* and *d*, and the lower gear can be adjusted on the engine shaft by screws. A universal joint is used between the crankshaft and the gear, *a*.

Many difficulties have been encountered in the actual operation of reduction gears—principally, fracture, wear and heating of the gears.

The **bending stress in a gear tooth** of the common involute form may be taken as

$$f = 14 \times \frac{W}{bp} \text{ approximately,}$$

where W is the load in pounds on the gear tooth, b is its width and p is the circular pitch in inches. The mean value of the loading on the tooth can be determined from the known engine power, P, and the speed, V, of the pitch circle

$$W = \frac{550 \times P}{V}$$

The maximum loading on the teeth may be considerably greater than the mean loading either because of acceleration pressures, resulting from incorrect pitch or form of teeth, or because of irregular delivery of power from the engine, or on account of reinforced vibration near a resonance period of the shaft. The values of f calculated for a number of successful engines run from 30,000 to about 40,000 lb. per square inch; the material used is generally case-hardened chrome-nickel steel. These high stresses are calculated on the assumption that all the load is carried on one tooth. With accurate pitching the deformations of the loaded tooth will transfer load to the next tooth. With oblique teeth, such as herring-bone gears, the tooth pressure is distributed on an oblique line running from the root to the tip and the bending stress is thereby reduced. The stresses are worse if the teeth bear unevenly as a result of warping in hardening, untrue keying or poor forming.

The **surface pressure** of the opposing curved tooth faces must not be sufficient to squeeze out the oil film. The relative sliding speed of straight-toothed gears is zero at the pitch circle, and the oil is more easily squeezed out under this condition than when there is relative motion. The bearing pressure is given by the expression $\frac{W}{bd}$, where d is the diameter of the relative curvature of the teeth at the rolling circle. With involute teeth having radii of curvature of e_1 and e_2 at the rolling circle,

$$\frac{2}{d} = \frac{1}{e_1} \pm \frac{1}{e_2},$$

the + sign applying to external gears, the − sign to internal gears. Calculations from successful engines indicate that with hardened gears the bearing pressure may go up to 1,400 lb. per square inch; if the gears are not hardened it should not exceed 450 lb. per square inch. With internal gears the bearing pressures become low and hardening is, as a rule, unnecessary. Experience with roller bearings, where hardened rolls run between hardened rings, indicates a permissible bearing pressure of 2,800 lb. per square inch or more at low peripheral speeds; if the rolls bear directly on the unhardened shaft the value falls to from 150 to 300 lb. per square inch.

With oblique toothed gears the contact shifts with great speed from side to side, as a result of which there is less tendency to squeeze out the lubricating film.

Heating of the gears results from the sliding contact at the teeth and may be of such magnitude as to lead to trouble. The heat is best carried off by thermal conduction from the gears to the outer casing, but if this is not sufficient it must be assisted by oil cooling. The lubrication should not be so heavy that the oil heats up through churning; this may occur through the use of wide gears which catch the oil and force it out sideways with great force, or through locating the gears very close to the housing.

For smooth running it is necessary that there should be no reversals of pressure in the gears. Four-cylinder engines give such reversals of pressure and so do six-cylinder engines at a low torque, or, with very heavy reciprocating parts, at high speeds. With a larger number of cylinders with crank angles equally spaced reversals will not occur.

Central power plants have been used on several planes. The principal advantage which they offer is the possibility of concentrating power plants in a central engine room (where they can be under constant supervision) and the resulting reduction of drag of the complete machine. There is also the possibility of reducing the number of mechanics required in a multi-engined plane. The disadvantages are the loss of power (possibly 5 per cent) in the transmission shaft and gears, and the increase in weight.

Chain-driven propellers were used successfully by the Wright Brothers in 1903 and by others later. In recent years the chain drive has not been used but shaft and bevel gears have been employed with some degree of success. Siemens-Shuckert multi-engine planes of several sizes have used bevel gear drives. The largest of these with six engines and four propellers, is arranged with the four rear engines driving the two rear propellers at half engine speed and the two front engines driving the two front propellers with a reduction ratio of 14 to 9. The couplings between the engines on the main transmission gear are a combination of friction and independent couplings. The latter enable the engine to be disengaged and stopped if damaged. The articulated transmission shafts are connected at both ends through laminated spring couplings.

The main difficulty in the operation of shaft drives has been in the setting up of "torsional resonance," which has caused breakage of shafts and universal joints. This has been overcome by the use of a flywheel on the engine and a special clutch which combines a dog clutch and a friction clutch. The shaft should rotate at engine speed and the gear reduction should be near the propeller. Some trouble has resulted from "whirling" of long shafts but probably because bearings have been placed at nodal points; this can be avoided. With these difficulties overcome it is doubtful whether the expense, weight and complication of the flywheels, clutches, shafts and gears will not more than counterbalance the advantages of a central engine room.

CHAPTER XVI

SUPERCHARGING

Change of Engine Power with Altitude.—The indicated work in the cylinder of a gasoline engine is the product of the heat of combustion of the fuel by the thermodynamic efficiency of the engine. The thermodynamic efficiency is unaffected by the air density and depends only on the ratio of compression. The heat of combustion is determined by the weight of fuel which can be burned and this depends on the weight of air admitted and consequently on the density of the air. All other conditions remaining constant, the indicated power of an engine would vary directly as the density of the air.

The brake horse power of the engine is the difference between its indicated power and the power required to overcome engine friction. At constant engine speed the friction will not change greatly with the air density; it increases with lowered temperature of the lubricant and decreases with lowered pressures at rubbing surfaces. If the frictional resistance remained constant the brake horse power would fall off much more rapidly than the indicated power at high altitudes. For example, an engine developing 100 i.h.p. at the ground will give 85 b.h.p. with 15 friction h.p. Operating in air at one-half ground density the theoretical indicated power is 50 h.p. and with 15-h.p. friction there would be 35 b.h.p. The brake power would be diminished in the ratio 35/85 = 0.412, while the indicated power is halved.

The actual diminution in brake power is not as great as the preceding calculation would indicate; the conditions are complex and not susceptible of exact calculation.

The friction horse power is partly rubbing friction and water and oil pump work and partly work done in overcoming throttling losses at the intake and exhaust of the gases. The former losses may be assumed constant with varying air density; the latter may be assumed to vary directly as the air density. If the total fricton loss is 15 per cent of the full indicated power at the ground, and the throttling losses are assumed to be one-third of the total friction loss, and if the indicated horse power is proportional to the relative air density, d, then the brake horse power, B, at any air density is given by

$$B = \frac{d - 0.10 - 0.05d}{0.85} B_o = \frac{0.95d - 0.10}{0.85} B_o$$

where B_o is the brake horse power at the ground. The following table gives horse powers calculated for different altitudes. It will be seen that the brake horse power is nearly proportional to

Altitude, feet	Relative air density, d	Relative air pressure	Relative b.h.p., B/B_o
0	1.0	1.0	1.0
6,000	0.829	0.801	0.808
12,000	0.694	0.645	0.660
18,000	0.581	0.518	0.532
24,000	0.485	0.414	0.424
30,000	0.411	0.333	0.342

the air pressure. Actual tests support these calculated quantities for altitudes up to 10,000 ft.; for high altitudes the brake horse power does not decrease as rapidly as the air pressure nor so slowly as the air density but according to some intermediate law.

Tests made at the Bureau of Standards[1] show a variation of brake horse power with barometric pressure as in column 7 of Table 20. The ratio of brake power to air density is given in the eighth column. It is seen that the brake power falls off more rapidly than the air density and that at one-half ground density the brake power is about 0.43 time the ground power.

The variations with air density of the mechanical, volumetric and thermal efficiencies of the Liberty 12 engine and the Hispano-Suiza 300 at a speed of 1,600 r.p.m. are given in Fig. 298.

The relative horse powers of Table 20 are based on constant engine speed. They may more properly be regarded as relative engine torques. Engine speed falls off with increasing altitude so that the actual horse power developed falls off more rapidly than is indicated in Table 20. With constant revolutions per minute the resisting torque at the propeller diminishes in direct proportion to the air density and consequently falls off less

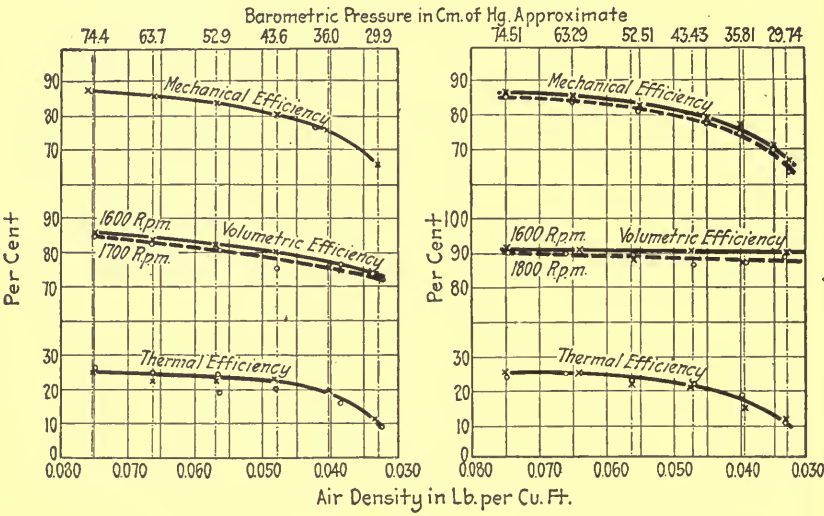
[1] *4th Annual Report, National Advisory Committee for Aeronautics,* 1918, p. 502, Fig. 6.

rapidly than the engine torque. Since the engine torque is practically independent of the revolutions per minute the engine speed will diminish as altitude is gained until that speed is reached at which propeller torque equals engine torque.

The actual engine power at any altitude is given by

$$P = P_G \times K \times \frac{N}{N_G}$$

where P_G is power developed at the ground,
N_G is revolutions per minute at the ground,
N is revolutions per minute at altitude,
K is the quantity in the seventh column of Table 20.



Liberty 12. Hispano-Suiza 300.

FIG. 298.—Variation of engine efficiencies with air density.

Table 20 shows that the engine power at constant speed is almost exactly proportional to the barometric pressure. On this basis the engine power at an elevation where the barometer is B cm. is

$$P = P_G \times \frac{B}{76} \times \frac{N}{N_G}.$$

Supercharging.—The diminution in power of a gasoline engine with increasing altitude results in a moderate reduction of speed in horizontal flight. If greater power were available the ground speed could be maintained at all elevations or exceeded, if desired. Much effort has been expended in attempts to prevent or reduce

TABLE 20.—VARIATION OF BRAKE HORSE POWER WITH ALTITUDE

Altitude		Barometric pressure		Air temperature, degrees Fahrenheit	Air density	Brake horse power developed	Brake h.p.	Brake h.p.
Feet	Meters	Inches	Centimeters				Density	Relative pressure*
0	0	29.92 14.7	76.0	66	1.0	1.00	1.0	1.0
2,000	610	27.8 13.6	70.6	59	0.94	0.940	1.0	1.01
4,000	1,220	25.75	65.4	53	0.88	0.880	1.0	1.02
6,000	1,830	23.92 11.7	60.9	48	0.829	0.823	0.99	1.025
8,000	2,440	22.3	56.6	42	0.785	0.766	0.975	1.025
10,000	3,050	20.75 10.2	52.7	36	0.733	0.712	0.97	1.025
12,000	3,660	19.3	49.0	29	0.694	0.660	0.95	1.025
14,000	4,270	18.0	45.7	22	0.655	0.614	0.938	1.02
16,000	4,880	16.7	42.4	15	0.617	0.572	0.927	1.02
18,000	5,490	15.5	39.4	8	0.581	0.529	0.91	1.02
20,000	6,100	14.35 7.05	36.4	1	0.545	0.488	0.895	1.02
22,000	6,710	13.3	33.8	−6	0.515	0.450	0.875	1.01
24,000	7,315	12.35	31.4	−13	0.485	0.413	0.853	1.00
26,000	7,920	11.5	29.2	−20	0.458	0.380	0.812	0.99
28,000	8,530	10.7	27.2	−28	0.435	0.349	0.806	0.98
30,000	9,140	9.95 4.9	25.3	−35	0.411	0.325	0.79	0.97

* Relative pressure means pressure expressed as a fraction of ground pressure.

this falling off in engine power. Such falling off would be entirely avoided if steam power could be substituted for gasoline power, since the boiler and condenser pressures would be independent of the barometer pressure. Attempts to design a light-weight steam plant have not been successful; there is no difficulty with engine or condenser (which takes the place of radiator), but it has not been found practicable to design a boiler to withstand high steam pressures and of sufficiently extended heating surface without arriving at weights which are prohibitive for airplane use. Furthermore, the lower fuel economy of a steam plant would necessitate the carrying of a greater weight of fuel.

Two general methods present themselves for increasing gas engine power at high altitude:

1. To select an engine so large that it will give the desired power when running with wide-open throttle at the high altitude at which the airplane is intended to fly, and to operate it at partial throttle at all lower altitudes.
2. To select an engine which gives the desired power at the ground and add some device for supplying the cylinder with air at a pressure greater than the barometric pressure when desired. This process is known as pre-compression or supercharging.

As an illustration, suppose it is desired to fly at 20,000 ft. developing 400 h.p. This can be accomplished either by installing an engine which would develop 800 h.p. with wide-open throttle at the ground; or by installing a 400-h.p. engine provided with a supercharging device which is able to maintain that horse power at all altitudes up to 20,000 ft. If the large engine is used the engine weight will be increased. An estimate made of the increase of weight which would result from doubling the power of a Liberty motor by doubling the piston area per cylinder, while keeping the stroke constant, indicates this increase would be about 40 per cent. If the power were doubled by doubling the number of cylinders the weight would be nearly doubled. It should be noted that if the engine is not permitted to develop more than 400 h.p. at any elevation, the radiator, water pump and general cooling system will not be larger than for a 400-h.p. engine. If the smaller engine is used the engine weight will also be increased by the addition of the supercharging apparatus and the engine becomes more complicated.

Oversized Engine.—In this system, the greater weight of the engine is offset not only by greater simplicity (as compared with a supercharging engine) but also by greater economy. Such engines should be provided with an automatically controlled throttle valve, actuated by some device (generally similar to an aneroid barometer) which responds to changes in atmospheric pressure. An example of such a device is given in Fig. 299 in which an airtight flexible chamber filled with air at low pressure actuates a balanced double-seated throttle valve. If the throttle is placed before the carburetor, the top of the float chamber must be kept in communication with the low-pressure side of the throttle. The control may be so adjusted as to give constant horse power at all altitudes up to that at which the throttle is wide open; the power cannot be maintained beyond that point. With an engine so operated it is possible to use a higher ratio of compression, without danger of preignition, than with an engine which has wide-open throttle at the ground.

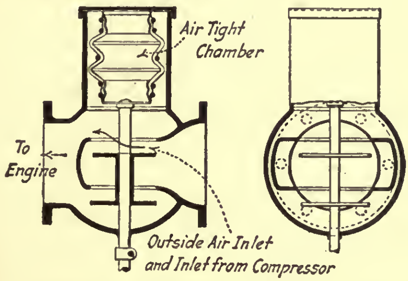


Fig. 299.—Automatic throttle control for oversized engine.

With constant power output the weight of the charge admitted per cycle will be approximately constant and the pressure in the cylinder at the beginning of compression is also approximately constant. The latter quantity is actually a little more at the ground than at higher altitudes because the engine is exhausting against a higher barometric pressure and consequently there is a greater weight and pressure of burned gases remaining in the cylinder to be mixed with the new incoming charge. Furthermore, the efficiency at the ground would be lower than at high levels on account of the higher back pressure. With constant power output the pressure in the cylinder at the beginning of compression would be less than 7 lb. per square inch at 20,000 ft. elevation, and probably less than 8 lb. per square inch at the ground. That is, the maximum pressure to be expected at the beginning of compression is 8 lb. per square inch as compared with 14 lb. in the usual engine. This results in lower compression and explosion pressures. Furthermore, the cylinder temperatures are lower throughout the cycle mainly in consequence of the smaller amount of heat developed by the explosion and the greater cooling effect of the water jacket. Under these conditions it is possible to employ higher compresssion without danger of

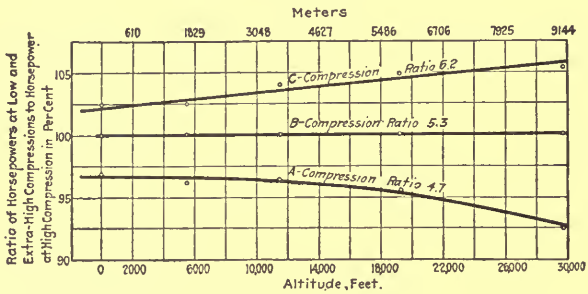


FIG. 300.—Effect of altitude on variation of engine power with compression ratio.

preignition. Engines have been operated in this manner with a ratio of compression as high as 7.

The employment of a **high ratio of compression** will increase the available power, particularly at high altitude. This is shown clearly in Fig. 300, which gives the results obtained at the Bureau of Standards with an engine supplied with three different sets of pistons to give different ratios of compression. The curve *B* is for a compression ratio of 5.3, which is here regarded as standard. The curves *A* and *C* are for compression ratios for 4.7 and 6.2 respectively. It will be seen that, calling the horse power with

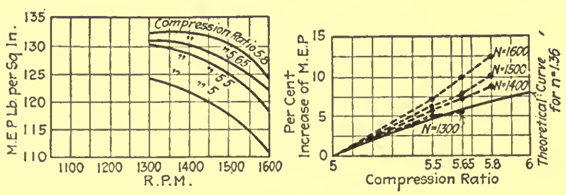


FIG. 301.—Variation of engine capacity with compression ratio and engine speed.

standard compression 100, at all altitudes, it is increased to 104½ at 20,000 ft. with the high compression and reduced to 95¼ with the low compression. At the ground the corresponding horse powers are 102¼ and 96¾. German tests on a Benz 200-h.p. engine, Fig. 301, show similar increase of power with ratio of compression and show also that the actual increase may be greater than the theoretical (air cycle) increase, particularly at high engine speed.

The increase in engine size necessary to maintain ground power is inversely as the density of the air at the altitude up to which full power is desired. The percentage increase is shown in Fig. 302.

The change in horse power actually developed with varying compression will be greater than these constant-speed tests indicate. With a given propeller the engine speed will increase

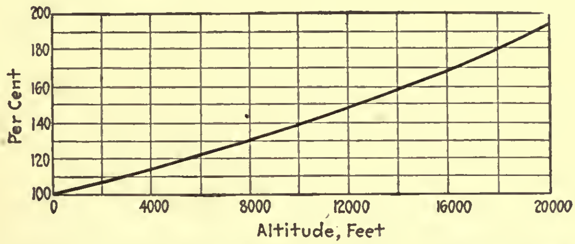


FIG. 302.—Required displacement volume of oversized engine.

with the engine torque and will cause a further increase in horse power. French tests show the following results:

	Hispano-Suiza		LeRhone			Clerget		
Ratio of compression	5.3	5.8	5.18	5.65	6.58	4.6	5.2	5.6
Revolutions per minute	2,040	2,070	1,230	1,260	1,290	1,290	1,350	1,360
Brake horse power	186	195	119	126	134	123	137	144

The **admission of inert gases** with the explosive mixture is now being developed as a means of maintaining high economy at low levels with an oversized engine. The inert gas is cooled exhaust gas. Its presence will permit operation with full throttle at much higher compression ratios than would otherwise be possible and consequently with higher thermal efficiency. As elevation is gained the percentage of inert gas in the charge may be reduced either by hand control or automatically; it should be possible to maintain the engine power at high altitudes and to operate continuously at very high efficiency by this device, (see p. 435).

Up to 12,000 ft. altitude the oversized engine (with about 46 per cent increase in volume) is preferable both in respect of total weight and of simplicity of construction to the supercharged engine. With altitudes in excess of 20,000 ft. the weight of the oversized engine becomes considerable and the preference may rightly fall on the supercharged engine. A combination of the two may possibly turn out to be best, with power maintained constant up to about 10,000 ft. by the gradual opening of the throttle valve, and then bringing into action a supercharging device to maintain constant power up to 20,000 ft.

Supercharging Engine.—In a supercharging engine, air is compressed by a blower or other device and is delivered to the

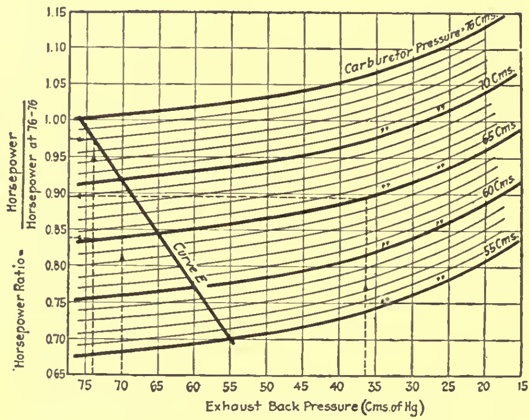


FIG. 303.—Chart for finding the power developed by a supercharged engine.

carburetor at a pressure in excess of the surrounding atmospheric pressure and consequently in excess of the exhaust pressure, except in the case (discussed later) where the blower is driven by an exhaust gas turbine. The **power that can be delivered** by an engine whose admission and exhaust pressures are different is readily calculable for an ideal engine, but it is necessary to have recourse to actual tests in order to ascertain its magnitude for an actual engine. Such tests have been conducted at the Bureau of Standards; the curves given in Fig. 303 show the results obtained. These curves give the horse power that will be developed by an engine with any exhaust pressure from 76 to 20 cm. of mercury and with air supplied to the carburetor at any pressure from 76 cm. down to 55 cm. of mercury. The horse power is given as a ratio to the horse power delivered at the ground with admission and exhaust both at a pressure of 76 cm. of mercury. The tests were conducted with the carburetor adjusted to give maximum power and the curves are all corrected to the same air temperature at the carburetor. The curves show that if the pressure at the carburetor is maintained at 76 cm. during flight the horse power developed in the engine will increase as a result of diminishing back pressure and at 20,000 ft. (36.5 cm. pressure) will be about 6 per cent greater than the horse power at the ground; this is to be compared with the diminution of 51 per cent in horse power (see Table 20) at the same altitude without supercharging.

The gain in engine horse power with supercharging is not of course net gain. Some of the additional work is used up in precompressing the air to the admission pressure. Furthermore, the precompression heats up the air so that it enters the carburetor at a temperature greater than that of the surrounding atmosphere. The actual horse power developed in the cylinder is given by the equation

$$P_C = P_G \times r \times F$$

where P_C is the horse power developed with the supercharging apparatus at the given altitudes; P_G is the observed horse power on the ground at the observed carburetor air temperature, t_1; r is the horse power ratio at the given condition of exhaust and carburetor pressures produced by the supercharging device at the given altitude, (obtained from curves, Fig. 303) and F is the temperature correction factor to correct from observed temperature at the ground, t_1, to temperature at the carburetor, t_2.

The **temperature of the precompressed air** can be calculated from the equation

$$T_2 = T_3\left(\frac{p_2}{p_3}\right)^{\frac{n-1}{n}}$$

where T_2 is the absolute temperature of the air entering the carburetor, T_3 is the absolute temperature of the air entering the supercharging device, p_2 and p_3 are the air pressures at the same places. The quantity n may be assumed for ordinary conditions to have the value 1.3. The quantity $\frac{T_2}{T_3}$ may be obtained from Fig. 304, which gives values also for $n = 1.2$ and $n = 1.4$.

The effect of this increase of temperature is to diminish the power of the engine. The temperature correction factor is obtained from the equation (compare p. 33).

$$F = \frac{920 + t_1}{920 + t_2}$$

where t_1 and t_2 are in degrees Fahrenheit (not absolute).

As an example of the use of the above, suppose an engine develops 200 h.p. at the ground with barometer 76 cm. mercury and temperature 40°F. and that it is taken to an altitude where the barometer is 35 cm. and temperature −4°F. and that it is supercharged to 65 cm. pressure. From Fig. 303 the horse power

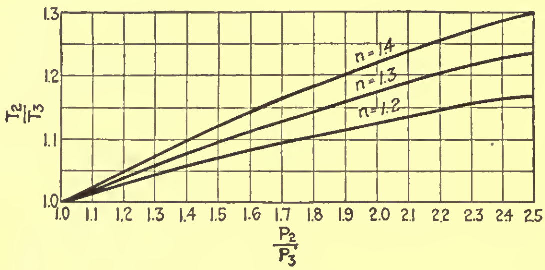


FIG. 304.—Temperature rise of air during compression.

ratio, r, corresponding to these conditions is 0.9. The pressure ratio $\frac{p_2}{p_3} = \frac{65}{36.5} = 1.78$. Assuming that $n = 1.3$, Fig. 304 gives $\frac{T_2}{T_3} = 1.142$, or $T_2 = 1.142 \times (460 - 4) = 520$. Consequently $t_2 = T_2 - 460 = 60°$. The temperature correction factor $F = \frac{920 + 40}{920 + 60} = 0.978$. The engine horse power will then be

$$P_C = 200 \times 0.9 \times 0.978 = 176$$

If the barometric pressure at which the ground horse power is observed is not 76 cm., a further correction may be introduced. For example, if the barometer reads 74 cm., the horse power ratio as compared with 76 cm. is found from curve E, Fig. 303, to be 0.972. The horse power actually developed, P_C, will then be $\frac{176}{0.972} = 181$.

As previously pointed out, this horse power is not the net horse power available for driving the propeller. There must be subtracted from it the work required to precompress the air.

The ideal method of compression is isothermal but this cannot be realized. If there were no addition or abstraction of heat during the compression and no frictional resistance or eddy losses, the compression would be adiabatic and this is what the centrifugal compressor, without cooling, might be expected to accomplish. The work of adiabatic compression is given by

$$\begin{aligned} W &= \frac{\gamma}{\gamma - 1}(p_2v_2 - p_3v_3) \\ &= w\,R\,\frac{\gamma}{\gamma - 1}(T_2 - T_3) \\ &= wJ\,C_p\,(T_2 - T_3) \end{aligned}$$

where p_2, p_3, are the pressures at beginning and end of compression respectively; v_2, v_3, the corresponding volumes; T_2, T_3, the

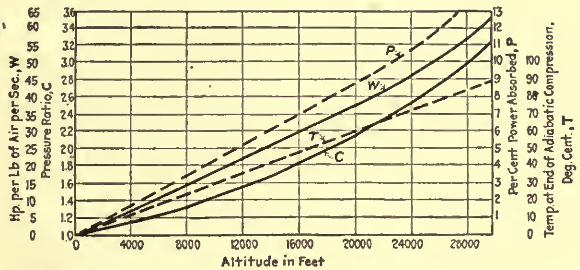


Fig. 305—Power absorbed in the compression of air.

corresponding absolute temperatures; R is the air constant, 53.4; J the mechanical equivalent, 778; C_p the specific heat of air at constant pressure, 0.241; γ the ratio of specific heats, 1.4; and w the weight of air compressed. The final temperature, T_3, is given by the equation

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$$

The work done per pound of air compressed (adiabatically) per second is given in Fig. 305, curve W, for compression from the mean pressure and temperature existing at any altitude to standard atmospheric pressure. This quantity is expressed in curve P as a percentage of the work which can be done by that air in an efficient engine. The ratio of compression (pressure ratio) is shown by curve C and the temperature rise of the air by T. The mean temperatures used as a basis for calculating the above curves are given by Fig. 275.

With *isothermal* compression the work required to compress 1 lb. of air is given by $W = RT_3 \log_e \frac{p_2}{p_3}$ ft.-lb. The actual work done on a centrifugal compressor in compressing air is found to be about twice the work of isothermal compression. For the example worked out above, the work of isothermal compression for 1 lb. of air is $W = 534 \times 456 \times \log_e 1.78 = 14{,}040$ ft.-lb. The actual work of air compression per pound of air will be $2 \times 14{,}040 = 28{,}080$ ft.-lb.

The amount of work done by 1 lb. of air in the cylinder is determinable from the assumption that the explosive mixture is 15 parts air to 1 part gasoline, by weight, and that the fuel consumption is ½ lb. gasoline per horse power hour. Every pound of air does work $\frac{2 \times 33{,}000 \times 60}{15} = 264{,}000$ ft.-lb. Of this work the fraction $\frac{28{,}080}{264{,}000} = 0.1065$ is used for precompression. Consequently in the case discussed the net available horse power will be $181 \times (1 - 0.1065) = 161$ h.p., that is, 20 h.p. will be used in driving the blower or other supercharging device.

There is one type of supercharging device in which the power required for precompressing the air is obtained from an exhaust gas turbine. This imposes a back pressure during the exhaust stroke in excess of the atmospheric pressure. For example, suppose that under the same conditions as those worked out in the example with 65 cm. carburetor pressure, an exhaust gas turbine is used and that the exhaust back pressure is 60 cm. mercury. The horse power ratio is now (Fig. 303) 0.85 and the net horse power is $\frac{0.85}{0.9} \times 181 = 171$ h.p.

In the **exhaust gas turbo-superchargers** that have been built up to the present, the selected operating conditions have generally been the maintenance of ground pressure in both admission and exhaust manifolds up to some limiting altitude. Assume a ground pressure of 76 cm., temperature 66°F. and the engine operating at an altitude where the barometer is 38 cm. (19,000 ft.), and air temperature 5°F. If the exponent n during the compression has the value 1.3, it is seen from Fig. 304 that $\frac{T_2}{T_3} = 1.174$ for $\frac{p_2}{p_3} = 2$. As $T_3 = 460 + 5$, $T_2 = 545$, or the temperature of the air entering the carburetor is 545 − 460 = 85°F. The engine horse power is thereby diminished in the ratio, $F = \frac{920 + 66}{920 + 85} = 0.984$.

The work W available from the exhaust gas turbine may be determined from the equation for adiabatic compression given above. The velocity, V, with which the gas discharges on the blades of the turbine (assuming a frictionless nozzle) is given by the equation $W = \frac{V^2}{2g}$, or it may be obtained from the equation

$$V = \sqrt{2g\frac{\gamma}{\gamma - 1}RT_2\left[1 - \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}}\right]}\ \text{ft. per second.}$$

where T_2 is the absolute temperature and p_2 the absolute pressure of the exhaust gases entering the turbine nozzle. Values of V from this equation are given in following table calculated for $T_2 = 1{,}800$. The average temperature of the gas leaving the engine is about 1,500°F., but loss from the exhaust manifold reduces it to about 1,300°F., or 1,800° absolute.

$\frac{p_2}{p_3}$	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0
V	754	1,026	1,230	1,384	1,514	1,622	1,716	1,798	1,871	1,940

With an exhaust gas turbine of 100 per cent efficiency, the work that can be done by the gas is equal to the kinetic energy of the gas. If an exhaust gas turbine, maintaining ground pressure at the carburetor and in the exhaust manifold, is fitted to a 200-h.p. engine, using 0.5 lb. gasoline per horse power hour and 15 lb. of air per pound of gasoline, the weight of exhaust gases will be $\frac{200 \times 0.5 \times 16}{60} = 26.7$ lb. per minute. At 19,000 ft. altitude (38 cm. barometer) the pressure drop ratio $\frac{p_2}{p_3}$ in the expansion nozzle is 2, and the corresponding gas velocity is 1,940 ft. per second. The kinetic energy of the gas is $\frac{26.7 \times 1{,}940^2}{2g \times 60} = 26{,}000$ ft.-lb. per second or $\frac{26{,}000}{550} = 47.3$ h.p. Assuming an efficiency of 50 per cent for the gas turbine the power available for driving the compressor is 23.7 h.p. The compressor work with isothermal compression would be $W = 534 \times 465 \times \log_e 2 = 17{,}400$

ft.-lb. per pound of air compressed. The weight of air compressed is $\frac{200 \times 0.5 \times 15}{60 \times 60} = 0.417$ lb. per second. The power required for isothermal compression is $\frac{0.417 \times 17{,}400}{550} = 13.2$ h.p. If the compressor efficiency is 50 per cent as compared with isothermal compression, the power required to drive the compressor will be $2 \times 13.2 = 26.4$ h.p.

For the operation of the turbo compressor just discussed, it is necessary that the **over-all efficiency** of the combination should be not less than $\frac{13.2}{47.3} = 0.279$. This over-all efficiency, E, is the product of the turbine efficiency E_T and the compressor (isothermal) efficiency E_C, or $E = E_T \times E_C$. Tests on the Rateau exhaust gas turbo-compressor indicate the possibility of values of E_T of about 0.53, and a value of about 0.5 for E_C. These correspond to $E = 0.53 \times 0.5 = 0.265$. The theoretical work of isothermal compression which can be done by this combination for the special case under discussion is $47.3 \times 0.265 = 12.55$ h.p. or is less than the 13.2 h.p. calculated as necessary to maintain ground pressure at the carburetor.

The actual pressure which could be maintained at the carburetor is readily calculable. The work of isothermal compression per pound of air is $W = RT_3 \log_e \frac{p_2}{p_3}$, and as the weight of air is 0.417 lb. per second, the horse power for isothermal compression is $(0.417 \times RT_3 \log_e \frac{p_2}{p_3})/550 = 12.55$. The value of T_3 has been given as 456. Solving the equation gives $\frac{p_2}{p_3} = 1.95$, and $p_2 = 74$ cm. That is, the power developed by the exhaust gas turbine is sufficient to compress the air to 74 cm. pressure. If a higher pressure is desired at the carburetor either the back pressure on the engine must be increased, thereby increasing the turbine power, or the efficiencies of turbine and compressor must be increased.

The inefficiency of the compressor has a further effect on the engine performance besides that just discussed. Practically all the work done on the compressor goes finally into **heating of the air** and as the (isothermal) efficiency of the compressor is about 0.5 the amount of such heating can be readily determined.

For the case under discussion with air at 38 cm. pressure, and 465° absolute temperature Fahrenheit and with compression to 74 cm. pressure, the work of isothermal compression per pound of air is $53.4 \times 465 \times \log_e 1.95 = 16{,}600$ ft.-lb. = 21.35 B.t.u. The total work done is $2 \times 21.35 = 42.7$ B.t.u. and the consequent heating of the air is $\frac{42.7}{c_p} = \frac{4.2}{0.241} = 177°$F. The final temperature of the air will be $177 + 465 = 642°$ absolute = 182°F.

This high temperature of the air entering the carburetor will cause a decrease in volumetric efficiency of the engine. To avoid the consequent loss of engine power the air should be partly cooled on its way from the compressor to the engine. Some heating of the air is highly advantageous in aiding the vaporization of the fuel in the carburetor and intake manifold.

Centrifugal compressors (single stage) do not appear to be suitable for ratios of compression greater than 2 to 1 on account of the excessive speeds which become necessary. This means that they can be used only up to altitudes of about 20,000 ft. if they are to maintain ground pressure at the carburetor. Multi-stage compressors permit higher ratios of compression, or the same ratio with lower peripheral speeds.

Supercharging Devices.—Two methods have been employed for supplying the engine with air at a pressure higher than that of the surrounding atmosphere.

1. The cylinder takes in an overrich charge in the usual manner and this charge is raised in pressure and diluted to the proper strength by the admission of compressed air at the end of the admission stroke.

2. The whole of the air going to the engine is compressed and is sent under pressure through the carburetor.

In the first method the demand for compressed air is intermittent and the compression is most suitably carried out in a reciprocating (piston) compressor. The second method requires a centrifugal compressor. It suffers the disadvantage that the pressure in the carburetor is greater than the external pressure so that the carburetor must either be made strong and tight enough to withstand this condition or must be entirely enclosed in a chamber under the compressor pressure, which makes it comparatively inaccessible.

An example of the first method is the **Ricardo system,** which has been experimented with considerably in England. The cylinder (Fig. 306) has a ring of ports uncovered near the lower end of the stroke. The piston is of two diameters, as shown, leaving an annular space, A, which diminishes in volume as the piston descends. This annular space communicates freely with the intercooler, B, which connects with the ring of ports. The closing of a hand-operated valve at F puts the supercharger out of action when desired. Air is admitted to the annular chamber, A, through the automatic valve, E.

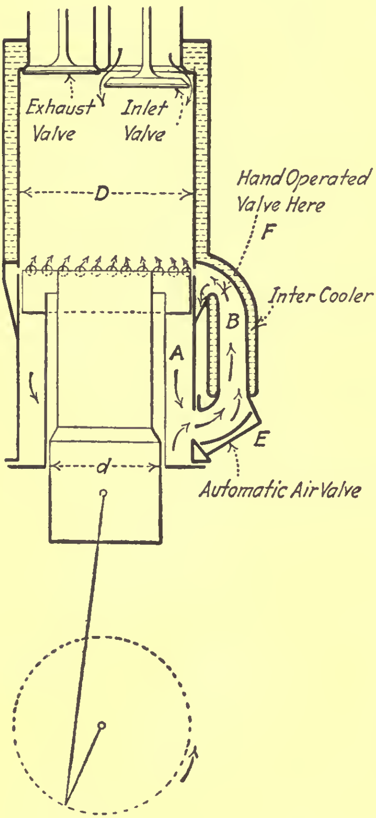


FIG. 306.—Ricardo supercharging engine.

In Fig. 306 the piston is shown near the end of the suction stroke. Compressed air from B is just beginning to flow into the cylinder through the ring of ports; the inlet valve is nearly closed and the pressure in the cylinder is raised. During the succeeding compression stroke air is admitted through E into A and is compressed in A and B during the expansion stroke. Near the end of the expansion stroke the ring of ports opens and some of the burned gases pass into B; the exhaust valve opens immediately afterwards and these burned gases together with the compressed air sweep back through the ports and scavenge the cylinder. A new charge of air is taken in through E during the exhaust stroke and is compressed during the following suction stroke.

It is obvious that this system can be used only for moderate degrees of supercharging since the additional air supplied to the cylinder per admission cannot be greater than

$$L \times \frac{\pi}{4}(D^2 - d^2)$$

where L is the piston stroke, and D and d are the two piston diameters. As the normal cylinder charge has a volume $L \times \frac{\pi}{4}D^2$ this represents an increase of $\frac{D^2 - d^2}{D^2} = 1 - \left(\frac{D}{d}\right)^2$. If $d = \frac{1}{2}D$, the increase in charge would be 75 per cent. The results of tests with this device are given in Fig. 307; they show an increase of power of approximately 50 per cent.

The more promising method of supercharging appears to be that in which all the air going to the engine is precompressed in a compressor. **Reciprocating compressors** operating at engine speed have been tried in England but show an over-all efficiency which is very low—only about 21 per cent. The **Roots** type of **positive** blower (Fig. 308) with aluminum rotors operating at

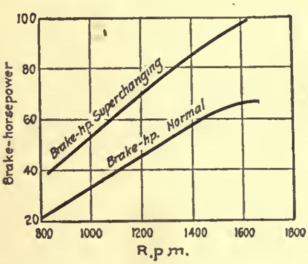


Fig. 307.—Performance curves of Ricardo supercharging engine.

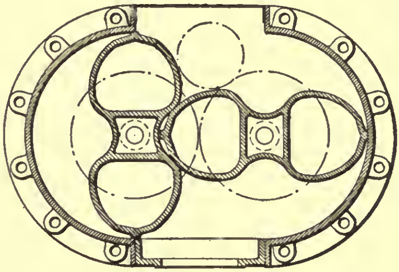


Fig. 308.—Roots blower.

twice engine speed gives an over-all efficiency of about 52 per cent but is exceedingly noisy. The most commonly used type is the **centrifugal compressor.** Such a compressor may either be directly coupled to the engine or driven by a separate engine or by an exhaust gas turbine. In any case, its peripheral speed must be high in order to keep down the number of stages and the weight and bulk of the compressor. If directly coupled to the engine shaft a train of gears must be employed to increase the speed up to 10,000 r.p.m. or more, depending on the number of stages employed and the amount of supercharging desired; if operated through a gas turbine no gears are necessary as the turbine speed will be from 20,000 to 30,000 r.p.m. and one compressor stage will be sufficient at these speeds.

Geared direct-coupled compressors have given much trouble from stripping of gear teeth. At the high speeds of rotation required, the kinetic energy of the rotor wheels is very great and considerable forces have to be employed for rapid acceleration. When the engine is started or the throttle valve is opened suddenly the pressure at the gear teeth is so high and so suddenly applied that breakage is likely to occur. To prevent this a friction clutch, spring coupling, centrifugal clutch or other equivalent device must be employed between the engine shaft and the rotor wheels.

One solution of this problem is shown in the **Sturtevant supercharger,** Fig. 309. The single-stage blower runs at 10 times the engine speed through a 2 to 1 belt drive in series with a 5 to 1 helical gear drive contained within the blower casing. The belt drive is vertical and is brought into action when desired by an

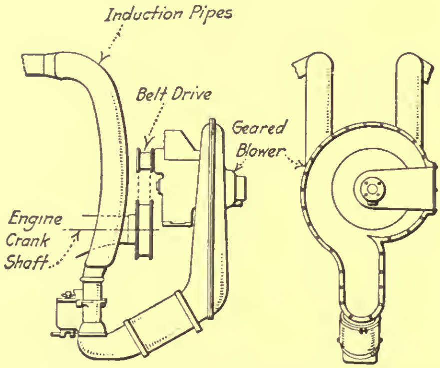


Fig. 309.—Sturtevant supercharger.

idler pulley. This arrangement gives ample slip when the engine speed charges suddenly. The weight of the supercharging device in this case is stated by the manufacturers to be 50 lb. for a 210-h.p. engine. The engine speed with constant-pitch propeller increases from about 2,100 r.p.m. at the ground to about 2,500 r.p.m. at 20,000 ft. altitude; the blower consequently increases from 21,000 to 25,000 r.p.m.

An English design, shown in Fig. 310, has a double reduction of 11 to 1 between the engine shaft and the blower disc with three intermediate wheels distributing the torque to the driven pinion; the over-all efficiency is 53 per cent.

German constructions show **multi-stage compressors** with the relatively low peripheral speeds of 400 to 500 ft. per second.[1] At these speeds the design works out to three stages to maintain

[1] Hildesheim, *Automotive Industries*, Oct. 21, 1920.

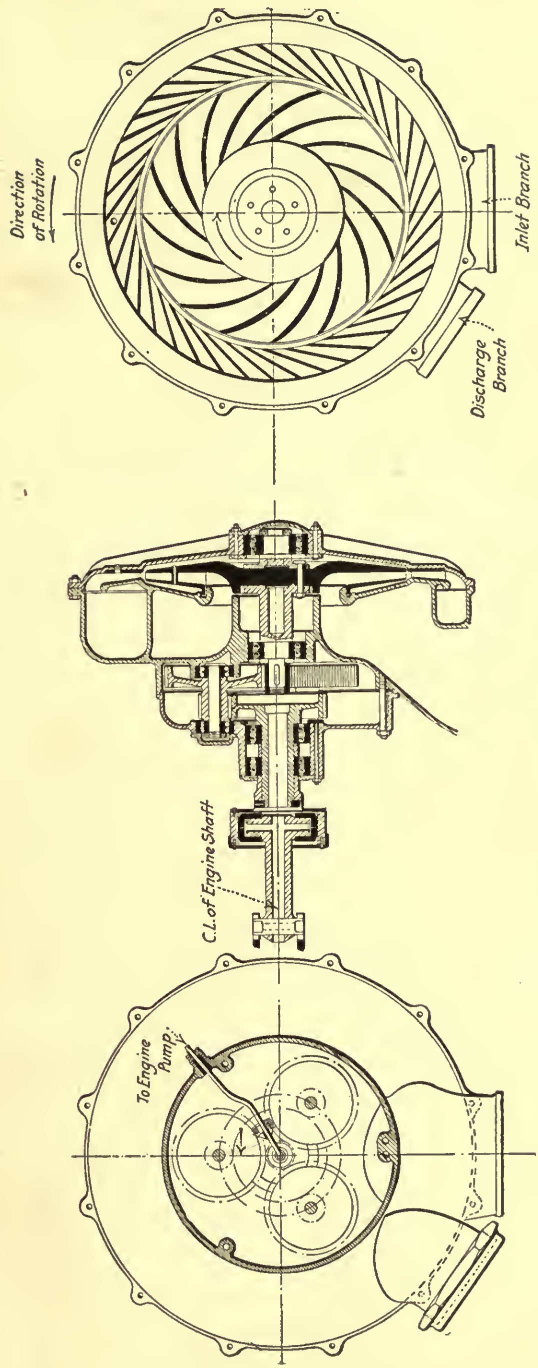


FIG. 310.—Centrifugal supercharger (British).

full power up to 11,500 ft. and four stages to 16,000 ft. The compressor makes 10,000 to 11,000 r.p.m. Such compressors have been used both as individual superchargers direct-coupled to a single engine, and as central superchargers driven by a separate engine and delivering compressed air to all the engines of a multi-engine plane. With a separate engine, no clutch is necessary between engine and compressor but the engine must be provided with a flywheel to prevent too great acceleration and consequent stripping of the gear wheels. Individual superchargers are generally driven from the rear end of the crankshaft,

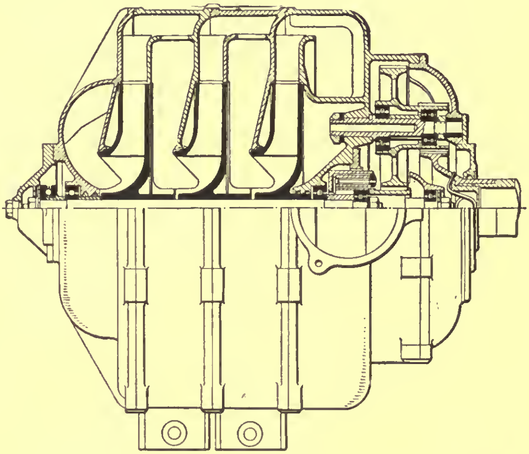


FIG. 311.—Schwade multi-stage centrifugal supercharger.

but in some cases the gears connect to the propeller end of the shaft to avoid the torsional oscillations which have sometimes given much trouble at the rear end.

The Schwade three-stage supercharger shown in Fig. 311 delivers 2,200 lb. of air per hour at a pressure ratio of 1 to 1.52 (11,500 ft. altitude). The shaft speed is 1,400, intermediate gears 3,500, blower 10,500 r.p.m. The rotor diameter is 10 in., peripheral speed 460 ft. per second. The pinion on the blower shaft is built in one with a friction clutch consisting of four bronze sectors which are pressed against the inside of the clutch housing by centrifugal force and which come into action when the engine speed reaches 600 r.p.m. The casing and its supports, partition walls, and diffusors are of aluminum. The supercharger complete weighs 105 lb. and was applied to a 260-h.p.

Mercedes engine weighing 925 lb.; it has also been used with rotary engines. A four-stage compressor for supercharging up to 16,000 ft. weighs 132 lb.

Brown-Boveri have built a central four-stage supercharger, 18.5 in. diameter and making 6,000 r.p.m., which gives a peripheral speed of 490 ft. per second. The gear ratio is 4.15 to 1. This machine supplies 9,200 lb. of air per hour at 0.52 atmosphere initial and 1 atmospheric final pressure; it is driven by a 125-h.p. engine and sends compressed air to engines aggregating 1,200 h.p. The gear teeth are of the Maag form, 0.5 in. circular pitch,

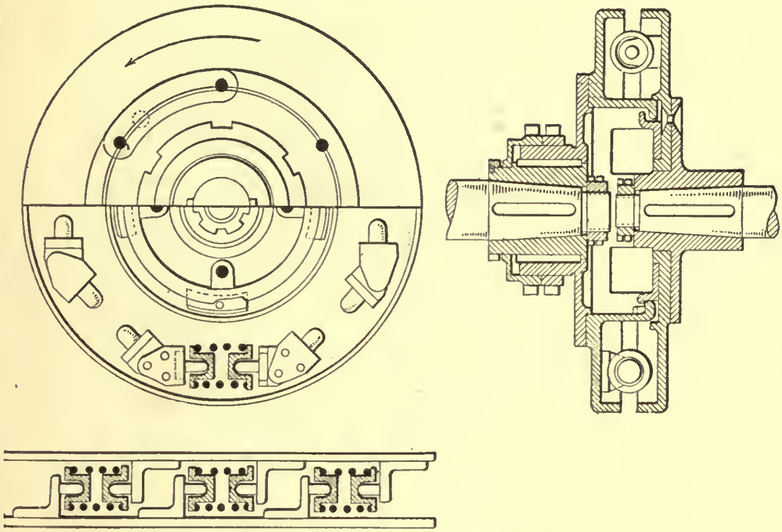


FIG. 312.—Spring coupling for supercharger drive.

2 in. face width, are hardened and ground and are loaded to 1,600 lb. per square inch at full load. Oil is injected directly between the teeth. The blower is connected to the engine shaft through a leather block joint. The coupling has a disc flywheel mounted on it (weight of both 44 lb.) to give smooth operation and protect the gears against shock. Tests with spring couplings have shown that the actual forces at the gear teeth are likely to be four times the normal driving force. The details of a successful spring coupling are shown in Fig. 312. The efficiencies of these multi-stage compressors (isothermal basis) average about 65 per cent; they may reach 68 per cent in some cases.

With a supercharger direct-coupled to the engine and operating all the time, a throttle valve should be placed on the suction side of the supercharger—either hand or automatically operated—to keep down the pressure and power at low altitudes. With this location of the throttle valve the power absorbed by the compressor will be less than with the throttle on the discharge side. With central superchargers, the air pressure is controlled by throttling the supercharger engine; the air pipe to each engine is fitted with a relief valve, a throttle

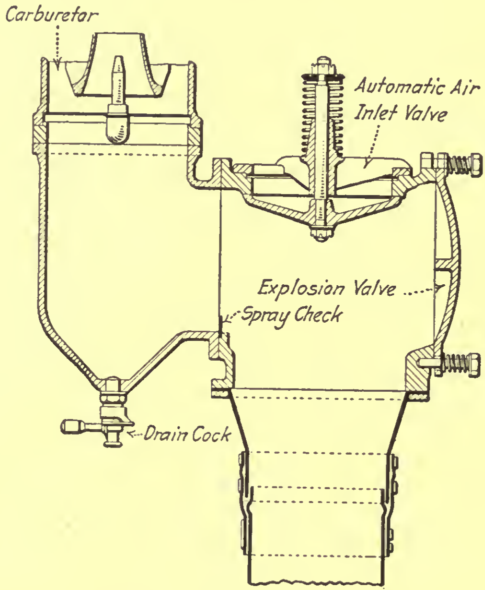


FIG. 313.—Explosion relief valve.

valve for the compressed air and an automatic air admission valve which closes when the supercharger comes into action. Relief or explosion valves are important; without such a valve a back fire would be likely to destroy the partition wall between the last two stages of the blower. Figure 313 shows an explosion valve held on its seat by springs and also an automatic inlet valve which closes whenever the compressor is in use.

The difficulties of geared drives can be eliminated and greater total power obtained by the use of an **exhaust gas turbine** for driving the compressor. In this case there is no fixed relation between the engine and compressor speeds. The exhaust manifold leads to the nozzle chamber of the turbine and the compressor is mounted on the turbine shaft. The scheme is shown in Fig. 314.

Much development work has been done on turbo-superchargers

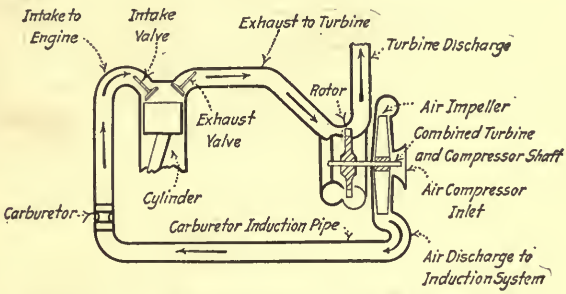


FIG. 314.—Diagram of exhaust-turbine supercharger.

but they must still be considered as in the experimental stage. The principal difficulties encountered have been with the exhaust valves, which are subjected to a higher temperature and which are not cooled by exposure to the outside temperatures; with the manifold, which is kept continuously at a high temperature and

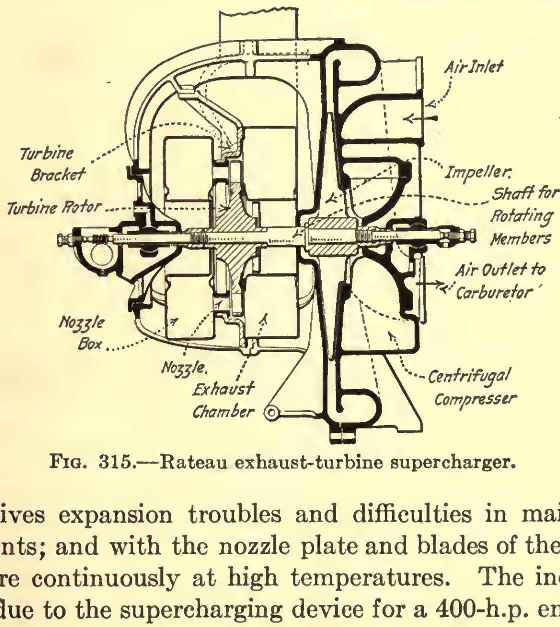


FIG. 315.—Rateau exhaust-turbine supercharger.

which gives expansion troubles and difficulties in maintaining tight joints; and with the nozzle plate and blades of the turbine, which are continuously at high temperatures. The increase in weight due to the supercharging device for a 400-h.p. engine can be made from 15 to 20 per cent of the engine weight when a peripheral speed of 900 ft. per second is used for the compressor.

The pioneer work on turbo-superchargers has been done by Rateau in France. Figure 315 shows a cross-section of his arrangement. The results of tests of the **Rateau supercharger** at

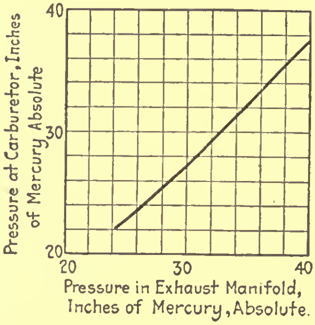


Fig. 316.—Pressures in Rateau exhaust-turbine supercharger.

an altitude of 9,000 ft. are given in Figs. 316–318. Figure 316 shows the relation between the back pressure on the engine and the pressure at the carburetor; the exhaust pressure stays at about 2 in. of mercury above the carburetor pressure. Figure 317 shows the variation of pressure and temperature ratios in the

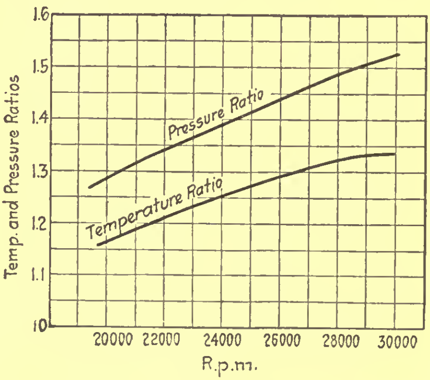


Fig. 317.—Performance of Rateau exhaust-turbine supercharger.

compressor with varying r.p.m. Figure 318 gives the variation of over-all efficiency of the turbo compressor with variation of r.p.m.; the turbine efficiency naturally increases as the bucket speed approaches the designed speed. Tests of a Lorraine-Dietrich 8-cylinder, 160-h.p. engine show an increase of power from 111 to 164 h.p. at 9,000 ft. altitude by the use of this supercharger; the engine speed increased from 1,370 to 1,550 r.p.m. Tests of a Breguet plane with a 300-h.p. Renault engine showed the time of climb to 16,400 ft. decreased from 47½ to 27 min. and the horizontal speed at that altitude increased from 91 to 120 miles per hour by the use of this supercharger. The ceiling was increased 13,000 ft. and the speed at the new ceiling was 25 per cent greater than that at the old ceiling.

British tests of a Rateau supercharger fitted to an air-cooled engine indicate the possibility of developing within 12 per cent of ground power up to a height of 17,000 to 20,000 ft. This is

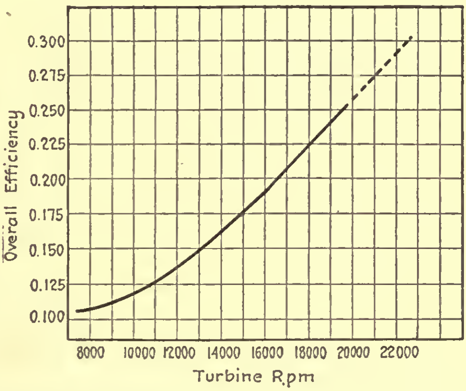


FIG. 318.—Over-all efficiency of Rateau exhaust-turbine supercharger.

obtained by maintaining ground pressure at the carburetor, which was found to entail a back pressure of about 19 lb. abs. at the exhaust.

The **Moss turbo-supercharger** (General Electric Co.) has been designed for, and used successfully on, the Liberty engine. The exhaust manifolds, of rectangular form, increase in cross-section as they come forward to the front of the engine and join at the nozzle box (Fig. 319) which is situated inside the Vee at the level of the tops of the cylinders. The nozzles cover about one-half the circumference of the wheel. The turbine wheel is 9.1 in. in diameter, the compressor 10.5 in.; the peripheral speed is about 1,000 ft. per second. The turbine and compressor spindle is supported at the rear in a water-cooled bearing mounted on the intake pipe; at the front, the bearing is in the air intake to the compressor. The compressor is provided with guide vanes

(Fig. 319) and discharges at the bottom into the intake pipe which extends horizontally backward into the Vee and supports the two carburetors. Performance data on this supercharger are not available but preliminary tests at Pike's Peak (barometer 18 in.) showed an increase of engine power from 251 h.p. to 367 h.p. when running at 1,800 r.p.m.

The use of a supercharger leads to some complication in the **fuel supply system.** The carburetor float chamber is kept at the compressed-air pressure, which is variable, and may be 7 or 8 lb.

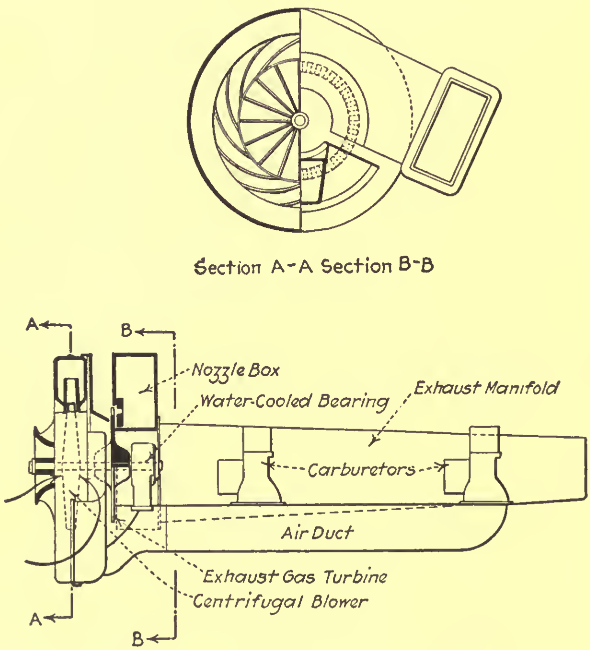


FIG. 319.—Moss exhaust-turbine supercharger.

per square inch in excess of the atmospheric pressure. The fuel has to be fed to the carburetor against this pressure. An air pressure system is undesirable both because of danger of leakage of fuel and also because the tanks would have to be made heavier to withstand such high pressures. The pressure in the fuel line must not exceed the pressure in the float chamber by an amount sufficient to lift the float valves and thereby flood the carburetor; the excess of pressure should be less than 5 lb. per square inch. A practicable system is to supply the fuel by a direct-acting engine-driven fuel pump delivering into a line equipped with a spring-loaded relief valve which is subjected to the compressed-air pressure on one side and the pump discharge pressure on the other. The spring is adjusted to lift at any desired excess of fuel over air pressure and by-passes some of the fuel to the suction side of the pump. Such a valve is shown in Fig. 320. The air is admitted to the inside of a corrugated copper "sylphon;" the fuel pressure must be sufficient to overcome both the air pressure and the spring compression.

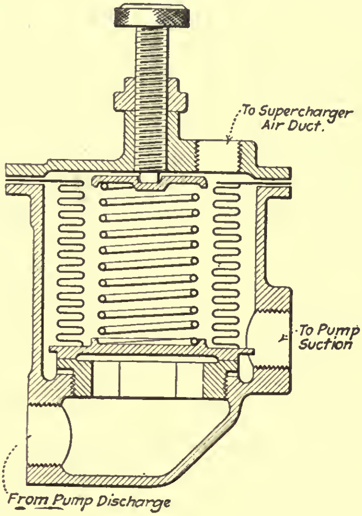


Fig. 320.—Relief valve for fuel line of supercharged engine.

Another method is to maintain the supercharger pressure in the gravity tank (Fig. 321) and to return to the main tanks the excess of gasoline pumped to the gravity tank through a float-operated valve.

Small fuel tanks with air pressure obtained from hand pumps are used for starting or for emergencies.

Another method of increasing the power of an airplane engine at high altitudes is by **supplying the engine with oxygen.** Approximately 4 lb. of oxygen is necessary to burn 1 lb. of gasoline. This weight is so considerable that oxygen can be carried only for emergency uses, as, for example, in combat, where it is desired to increase the speed of the plane for a short time. As the oxygen-gasoline mixture would give excessive temperatures and pressures, the oxygen can be used only for enriching the air and permitting a moderate increase in the heat developed per cycle. The oxygen must be carried in liquid form at atmospheric pressure, otherwise the weight of the container becomes excessive. As the boiling temperature at ground pressure of liquid oxygen is $-297°$F. it is necessary to have extraordinarily good heat insulation to keep the rate of evaporation down to a permissible figure. A Dewar flask is the only practicable insulator in this case, but since it is too fragile and brittle to withstand the slapping of the liquid in flight, it is necessary to put the liquid oxygen in a metal container which with proper cushioning is inserted in a Dewar flask. The liquid oxygen can be evaporated at any desired rate (1) by electrical heating through a coil immersed in the liquid and (2) by immersing a metal rod in the liquid to an adjustable depth and utilizing the thermal conduction along the rod, or (3) the liquid can be siphoned over into the charge going to the cylinder. It is evident that special adjustment of the carburetor is necessary to meet the condition of oxygen supply.

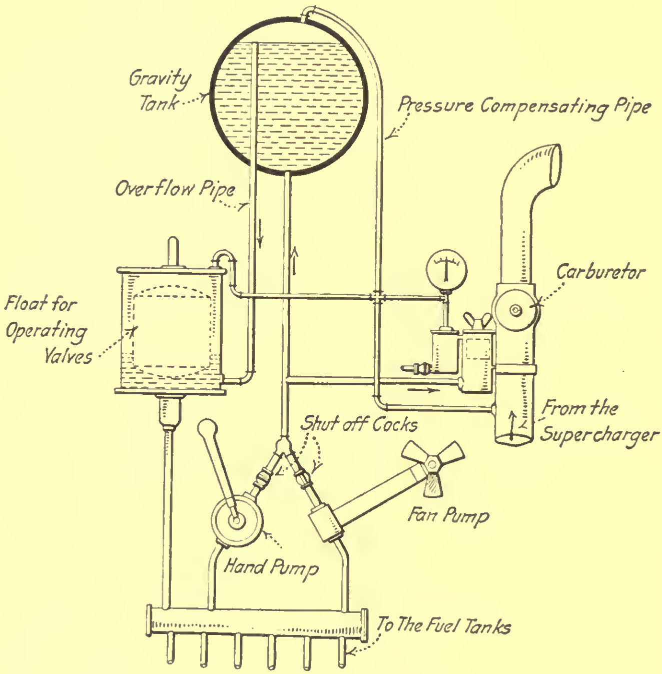


FIG. 321.—Fuel system for supercharged engine.

The power developed in a supercharged engine can be absorbed satisfactorily only by the use of a **variable pitch propeller.** A four-bladed propeller would be best in order to avoid excessive peripheral speeds at high altitudes. The supercharger may be used either (*a*) to maintain constant power, in which case the revolutions per minute will vary inversly as the cube root of the air density, and the torque will vary as the cube of the density or (*b*) to maintain constant torque, in which case the revolutions per minute and power will vary inversely as the density.

A calculation for a machine of standard type with a ground speed of 110 miles per hour gives the following results:

Type of engine	Altitude, 10,000 ft.		Altitude, 20,000 ft.		Ceiling, feet
	Maximum speed, miles per hour	Rate of climb, feet per minute	Maximum speed, miles per hour	Rate of climb, feet per minute	
Normal engine	105	450	...	.....	19,000
Constant power	126	1,050	141	930	61,000
Constant torque	132	1,420	159	1,770	No theoretical limit

CHAPTER XVII

MANIFOLDS AND MUFFLERS

Air Intakes.—The location, dimensions and orientation of the air intake to the carburetor have considerable influence on the capacity of the engine. Attempts are often made to increase the density of the air going to the carburetor by having the air intake face full forward, so as to establish in the intake a pressure which is the sum of the surrounding atmospheric pressure and the velocity head equivalent to the air velocity of the plane or of the slip stream. With a plane making 120 miles per hour at the ground this dynamic head would be about 0.24 lb. per square inch. The necessity for keeping the air intake pipe of moderate length does not usually permit the intake to be located in free air so as to take advantage of this velocity head. Inside the fuselage there is very small air velocity and the practical means of getting access to high velocity air is by extending the air intake pipe above the cowling. This arrangement has the great advantage that spillage of gasoline into the fuselage is thereby prevented and a frequent source of fires is eliminated. The disadvantage of this arrangement when applied to the usual form of carburetor is that the air intake pipe will have two or more right-angle turns. With an inverted type of carburetor, with jets discharging downward this difficulty is overcome and a short direct air intake pipe can be employed.

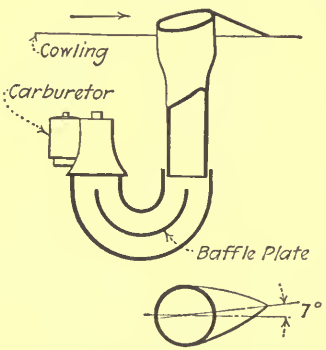


FIG. 322.—British design for air intake pipe.

Investigations in England on the best shape of air intake have shown that maximum power is obtained when the intake pipe is cut off at an angle of 25 deg. to the horizontal so as to form a scoop and that instead of mounting the scoops dead ahead they are best turned about 7 deg. to compensate for the propeller slip stream (see Fig. 322). The double right-angle turn of the air on its way to the carburetor results in a disturbed air flow which can be largely corrected by the insertion of the baffle plate indicated. In the Liberty engine a single standpipe has been used to serve two carburetors, by the use of the duplex air intake shown in Fig. 323.

The length of intake pipe will affect the engine capacity because the inertia of the air column tends to make the flow to the carburetor continuous in spite of the intermittent action of the cylinder suctions. In the Liberty engine the optimum length with inverted type of carburetor is found to be 6 in. at normal revolutions per minute.

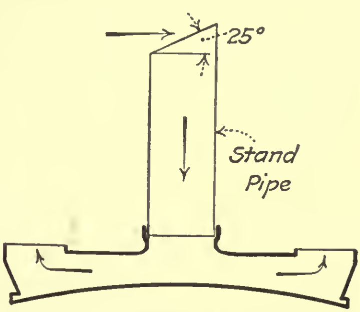


FIG. 323.—Liberty engine duplex intake pipe.

Intake Manifolds.—The main function of the intake manifold is the distribution of the mixture formed in a carburetor to several cylinders. For good efficiency and capacity the strength and density of the mixture reaching all cylinders should be the same. To ensure these results the amount of fuel entering, and the pressure drop from carburetor to inlet valve, should be the same in all branches. With a fuel which is not completely vaporized before the branching of the manifold occurs it is very difficult, if not impossible, to ensure proper distribution of the fuel. This difficulty increases as the volatility of the fuel decreases and is therefore greater with ordinary commercial gasoline than with airplane fuels.

For the vaporization of a volatile fuel the important factors are (1) air supply of sufficiently high temperature, (2) fine atomization at the jet, (3) avoidance of obstacles in the path of the mixture so that the atomized liquid drops may have no opportunity for coalescence before being vaporized, and (4) sufficient time for the vaporization. The temperature of the mixture after vaporization is much below that of the entering air because the latent heat of vaporization of the fuel is taken from the air. Assuming a latent heat of 135 B.t.u. per pound, complete vaporization will produce a temperature drop of 47°F. with an air fuel ratio of 10 to 1, and a drop of 25°F. with an air fuel ratio of 20 to 1. If all the fuel is not vaporized the temperature drop will be less. The fall of temperature in the

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manifold may bring about a deposit of ice both inside and outside the manifold as a result of the freezing of the moisture in the air. It has been suggested[1] that accumulation of ice inside the manifold may be the cause of numerous unexplained engine failures and resulting crashes. The relation between the manifold temperature, measured at the intake valve, and the air supply temperature in a Liberty engine is shown in Fig. 324. The manifold was water-jacketed. Below 20°F. there was so little evaporation that the manifold temperature was higher than the air supply temperature. As the air supply temperature increased up to 120°F. the temperature fall increased up to about 35°F. The air fuel ratio was about 16 to 1.

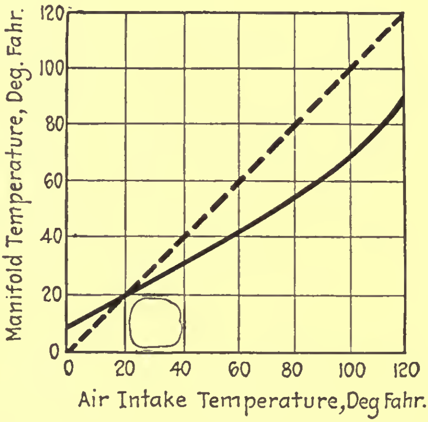

FIG. 324.—Temperature change in intake manifold.

With air initially cold it will not be possible to vaporize the fuel completely by heat absorbed from the air because at low temperatures the air will become saturated with the fuel vapor before all the fuel is vaporized. Table 14, page 229, shows, for example, that with a theoretically correct mixture, the air cannot hold all the pentane in the vapor form at a temperature below about 38°F. In such case the only chance for complete vaporization is by supplying heat from outside. This is accomplished by utilizing some of the heat either of the jacket water or of the exhaust gases. The possibilities are (1) to preheat the air before it enters the carburetor, (2) to heat the mixture in the manifold, and (3) to heat the manifold locally at some place on which the

[1] SPARROW, *Technical Note, No.* 55, *Nat. Adv. Comm. Aeronautics.*

liquid drops impinge so as to supply heat to the liquid only (hot-spot method).

All preheating is objectionable in that it diminishes the density of the charge and thereby decreases the capacity of the engine. That method of preheating is best which causes vaporization with the minimum resulting temperature of the mixture. All three methods of preheating are employed in airplane practice. In some of the German engines preheating the air is accomplished by taking the air through pipes in the crankcase (Fig. 78), which has the advantage of cooling the lubricating oil. The more common procedure is heating the manifold by jacket water.

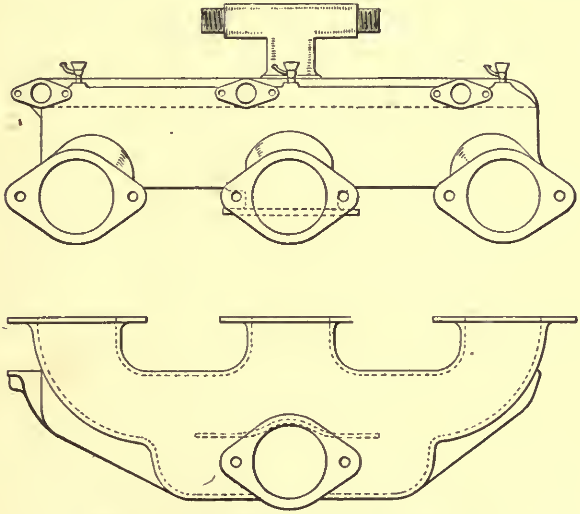

Fig. 325.—Liberty engine intake manifold.

There is no general consensus of opinion as to the best form of manifold. It is desirable that sharp turns should be avoided as far as possible, that the various branches should have approximately the same length, that sudden enlargements should be avoided and that the velocities should be high enough to prevent deposition of liquid drops but not so high as to cause a large frictional resistance. Mean velocities of 120 to 200 ft. per second are common but values up to 250 ft. per second have been used successfully.

Manifolds usually divide themselves into two classes, the short-branch type and the long-branch type. The standard Liberty manifold (Fig. 325) is a good example of the short-branch type; it is water-jacketed and has a baffle plate opposite to the inlet to equalize the lengths of the three branches. The Benz manifold (Fig. 326) is an example of an unjacketed long-branch manifold with all three branches of the same length and with long-turn elbows. Another design for accomplishing the same purpose is shown in the Hall-Scott engine (Fig. 63).

Manifolds for vertical engines can usually be arranged in any way the designer likes. In the Maybach engine (Fig. 80) the carburetors are at the ends and the manifolds run along the side of the engine with no attempt to equalize the lengths of the branches. Ordinarily such arrangements as those of Figs. 325 and 326 are used.

In 90-deg. Vee engines there is plenty of room in the Vee to accommodate the carburetors and the intake is usually inside the

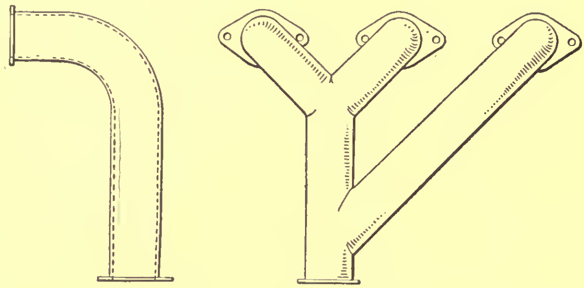

FIG. 326.—Benz intake manifold.

Vee. With this location a short-branch manifold must be used. The Hispano-Suiza engine (Fig. 50) shows a typical arrangement with the transverse pipe water-jacketed. If long-branch manifolds are to be used they must either be placed in the rear of the engine, as in the Curtiss engines, Figs. 59 and 62, or the intake valves must be on the outside of the Vee.

In 60-deg. and 45-deg. Vee engines the space inside the Vee is small and a favorable design of manifold is difficult if the carburetors are placed inside the Vee. If the inlet valves are placed outside the Vee, the exhaust pipes will be crowded inside the Vee and may give rise to troubles caused by their proximity to the valve springs, etc. An alternative arrangement is to provide a space between the two central cylinders of each block (as in the Bugatti (Fig. 67) and Fiat engines (Fig. 76)) and to lead the induction pipes from carburetors mounted outside the Vee through these spaces to manifolds inside the Vee.

In radial and rotary engines the distribution problem is comparatively simple, especially where an induction chamber is provided in the crankcase. The special distributing chamber of the Bristol "Jupiter" engine (Fig. 148) is noteworthy.

Exhaust Manifolds.—The function of the exhaust manifold is, primarily, to conduct the exhaust gases away from the airplane with the minimum back pressure at the engine and without fire risk to the airplane or annoyance from the discharged gases to the pilot. An additional function may be to muffle the sound of the exhaust, although this has usually been considered unimportant in military machines. The manifold is usually required to have a clearance of 2½ in. from wooden parts and of 3½ in. from fabric parts of the airplane.

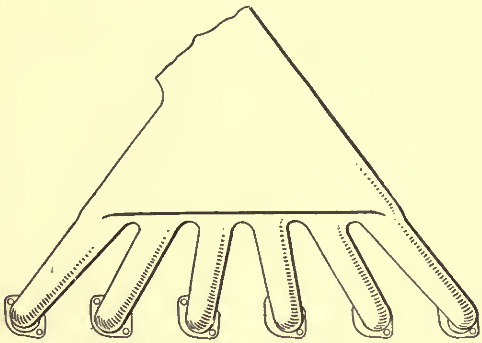

Fig. 327.—Hall-Scott exhaust manifold.

The simplest exhaust piping is either its complete absence as in rotary engines and certain stationary engines or the use of short tubes discharging outwards or upwards. In some engines these stub tubes are cut off on the outer end at an angle of 45 deg. so as to discharge backwards as well as outwards. The absence of exhaust pipes or the use of short straight pipes is advantageous not only in reducing back pressure but also in permitting better cooling of the exhaust valves by radiation and avoiding heating of the exhaust valve springs. This arrangement does not conduct the gases away from the crew of the airplane.

A method of discharging the gases overhead and to the rear with small back pressure is shown in the Hall-Scott manifold of Fig. 327. This arrangement obstructs the view of the pilot in a tractor machine but diminishes the noise heard from below. Long radius curves are very essential for all the branches if back pressure is to be kept down; the radius should be about 2½ times the diameter of the pipe. The arrangement of Fig. 328 with discharge to the rear and with an exhaust main of increasing diameter not only carries the gases away from the crew but slows down the gases before exit and thereby tends to diminish noise.

As the exhaust period lasts more than two-thirds of a revolution there is overlapping of exhausts in a manifold connecting three or more cylinders. The velocity of the gases immediately after the opening of the exhaust valve is extremely high and its effect on the exhaust from any other cylinder whose exhaust valve is open at that time should be carefully considered. By the use of two concentric exhaust mains to which the cylinders are connected so

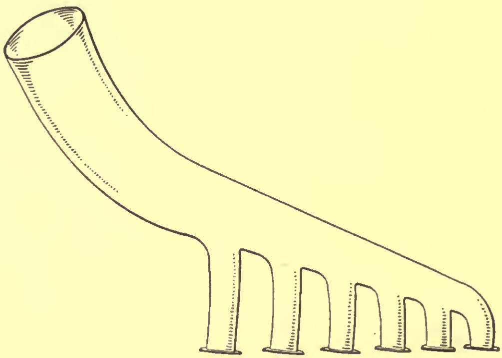

FIG. 328.—Hall-Scott exhaust manifold.

that no two consecutive exhausts go into the same main, an ejector effect can be obtained from the action of a newly opened exhaust on the exhaust which is closing, which may cause substantial scavenging in the closing cylinder.

The cross-section area of manifold branches is governed by the size of the ports in the cylinders; an average value is about 0.15 sq. in. per brake horse power of the cylinder.

Mufflers, to be efficient, must slow down the exhaust gases to velocities below that of sound (1,100 ft. per second); actual gas velocities probably exceed 2,000 ft. per second. For airplane use the muffler must be of light weight and yet durable. The constructions employing reversal of direction of the gases and discharge through small holes are likely to give excessive back pressure. Tests by Diederichs and Upton show that the volume of the muffler should be about three times that of a single cylinder of the engine, and that the inlet to the muffler should be tangential so as to give the gas a whirling motion. For durability the muffler should be attached to the end of a tail pipe 6 or 8 ft. long which will cool the gases sufficiently to prevent excessive oxidation of the muffler. One of the simplest and most successful mufflers is shown in Fig. 329. The tangential inlet pipe starts of circular cross-section and flattens out in a fan shape so as to give admission for almost the whole length of the muffler. An inner

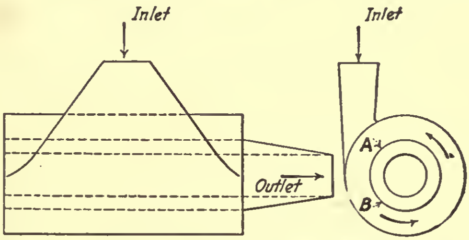

Fig. 329.—Diederichs and Upton exhaust muffler.

shell, *AB*, is provided with a large number of holes except in the small arc between *A* and *B*; these holes are smallest near *B* and increase in diameter from *B* to *A* (contra-clockwise). The gas passes through these holes and then through more holes in the innermost shell and finally escapes at the open end of the innermost shell. With this muffler the exhaust noise can be reduced 80 per cent with about two-thirds of 1 per cent reduction in engine power. Mufflers of this type are found to give less back pressure than those with axial admission.

CHAPTER XVIII

STARTING

The starting of an airplane engine depends on three things, (1) obtaining an explosive mixture in the cylinder, (2) a device for igniting it, and (3) a device for turning the engine over.

Hand Starting.—The idling device on the carburetor will furnish a rich mixture to the cylinders if the throttle valve is closed and the engine is turned at a sufficient speed. If the revolutions per minute of the engine is too low (below 20 to 30 r.p.m.) the velocity of the air in the intake manifold will be insufficient to carry the fuel into the cylinders; this will be the usual condition

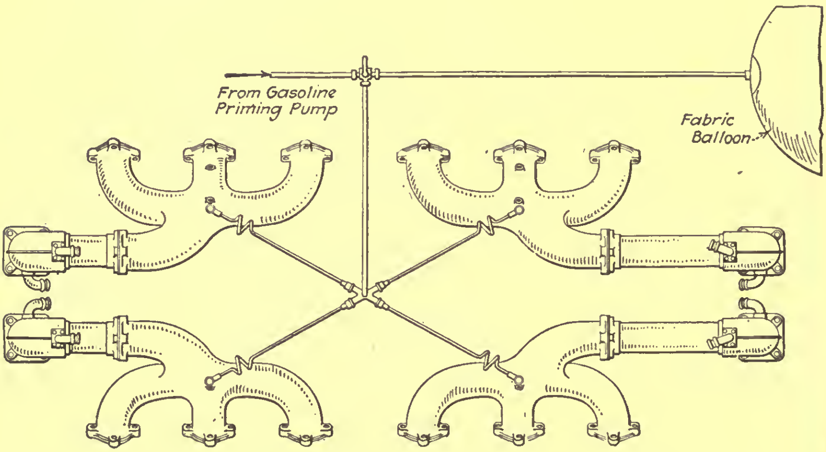

FIG. 330.—Priming system for a 12-cylinder Vee engine.

with hand starting. To overcome this difficulty the cylinders may be primed through individual priming cocks or through such a priming system as is shown in Fig. 330. With the engine cold only a small part of the gasoline will be vaporized and the rest will go as liquid into the cylinder and will dilute the lubricant. In very cold weather there will be difficulty in vaporizing enough of the gasoline to form an explosive mixture unless the jacket is filled with hot water or some other device has been used to heat the engine. In such a case a number of attempts may have to be made before the engine starts, and as an excess of gasoline is put in the cylinder before each attempt the walls may be washed clear of lubricant and scoring of the cylinder may occur when the engine finally starts.

To overcome these difficulties a more volatile fuel such as ether may be used for priming. A more satisfactory practice, for cold weather, is to use hydrogen as the starting fuel. This has the great advantage that it does not have to be vaporized and that it forms an explosive mixture throughout a great range of strengths. A hydrogen-air mixture will explode if the hydrogen forms from 10 to 66 per cent of the mixture by volume; with gasoline vapor the range is only from 1.5 to 4.8 per cent. The hydrogen may be admitted through the priming system as shown in Fig. 330; it is best to take the gas from a fabric balloon in which it is at atmospheric pressure and not from a high-pressure bottle which would be likely to give an excess of gas. The hydrogen must be shut off shortly after the starting as it gives more violent explosions than gasoline.

The regular magneto will not turn fast enough with hand starting to give a spark. If a battery system of ignition is used there is no difficulty from this source. If a magneto system is used it is customary to supply a hand-operated starting magneto which is geared to run at high speed and is turned independently of the engine. With hand starting the engine may be pulled over several times to fill the cylinders with explosive mixture, after which a shower of sparks is sent from the starting magneto through the distributor to the fully retarded cylinder which is ready to fire, or the ignition may be left on, fully retarded, while the propeller is pulled over either by hand or by rope.

The directions for starting the Liberty engine by hand are as follows: Inject ½ oz. of lubricating oil through each priming cock. Turn switch "off." Turn the engine over five times. Open throttle slightly. Retard spark fully. Prime each cylinder twice. Turn engine over twice. Turn on one switch. Pull down and forward on propeller blade. After the engine starts: Advance spark half way. Turn on both switches. Leave throttle undisturbed for 5 min. The lubricating oil should be warm by the time the jacket outlet water has reached 150°. If, in cold weather, it is not, stop the engine for 5 min.; then start all over again. Accelerate and slow down the engine a few times. Note that the oil gage registers pressure, 5 lb. at 600 r.p.m. At this speed, the ammeter should show "discharge." At 1,000 r.p.m. it should indicate "charge." After 5 min. more, open the throttle wide. The speed should rise to about 1,600 r.p.m.

The necessity of turning the engine over preliminary to starting can be avoided by the use of a mechanism which will lift the valves and permit the pumping of an explosive charge into the cylinders. In the Maybach engine this is accomplished (Fig. 331) by lifting all the tappets (inlet and exhaust) off their cams by depressing the hand lever, *A*, which rotates the two lay shafts, *BB*, and lifts the tappets through slots in the lay shafts engaging with small lugs projecting from the tappets. At the same time the shutter, *C*, in the exhaust main is closed. The hand pump, *E*, is then operated and air is sucked through the carburetor and the intake manifold and through the cylinder to the exhaust manifold and to the pump. When the cylinders are thus charged the lever, *A*, is brought vertical, which restores the engine to its operating position. The starting magneto is then used to start the engine. The whole operation can be carried out from the pilot's seat.

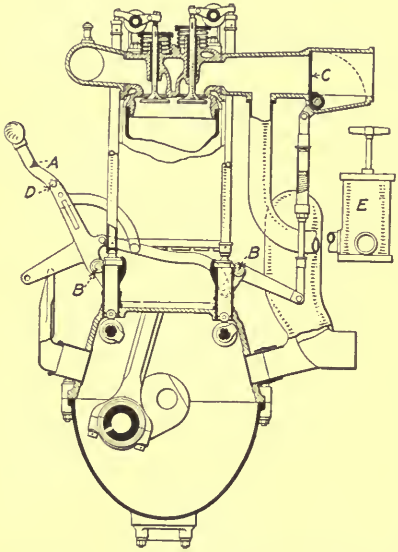

Fig. 331.—Starting mechanism of Maybach engine.

The **starting torque** in high-power airplane engines is very considerable. The load consists of two parts, (1) the compression load, and (2) the friction load. The compression load increases with the cylinder diameter and the compression ratio. The maximum torque does not change with the number of cylinders because high-compression pressure will exist at any moment in one cylinder only. As the compressed gas re-expands the mean torque for two revolutions does not increase with increase in number of cylinders. Friction is the principal resistance in starting the engine, especially in cold weather when the lubricating oil may be near the solidifying point. Tests at McCook Field[1] to determine the average starting torque for various engines have yielded the following results. The average air temperature was

AVERAGE STARTING TORQUE, POUNDS-FEET

Engine	Number cylinders	Rated horse power	Normal speed	Bore, inches	Stroke, inches	Number of engines tested	Number of trials	Starting torque, pounds-feet			
								Throttle open		Throttle closed	
								Maximum	Average	Maximum	Average
Liberty	12	400	1,700	5.00	7.00	2	6	130	124	156	143
Hispano-Suiza	8	300	1,800	5.51	5.91	2	6	106	102	101	96
Liberty	6	210	1,700	5.00	7.00	2	6	133	105	135	110
Packard	8	180	1,600	4.75	5.25	1	3	110	82	87	77
B. M. W.	6	185	1,400	5.90	7.08	1	3	150	139	153	145

75°F. With freezing temperatures the results would have been much higher—probably doubled. It should be noted that the starting torque does not increase nearly as rapidly as the engine horse power.

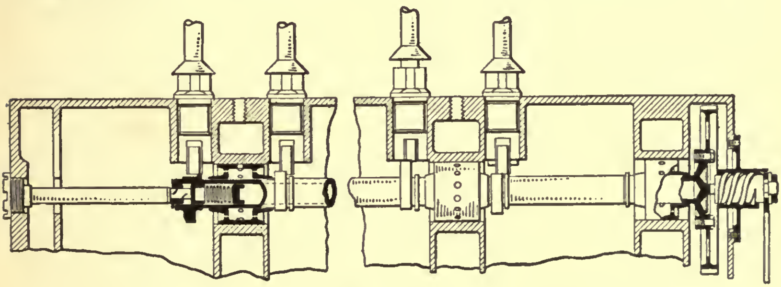

FIG. 332.—Compression release of Benz engine.

From the preceding table it may be presumed that with low temperature the mean starting torque for a 400-h.p. engine will be from 300 to 400 lb.-ft. It is difficult for a mechanic to exert this torque on the propeller even in a land plane without danger to himself from overbalancing; in a sea plane it is even more difficult to accomplish. The starting torque can be diminished by reducing or eliminating the compression. For this purpose **compression release** cams may be provided on the camshaft to keep the exhaust valves open during the compression stroke. In the Benz engine (Fig. 332) the compression release cams are brought

[1] *Air Service Information Circular, No.* 126.

into action by the axial movement of the camshaft effected by a square-thread screw operated by a small lever at the rear of the crankcase. The camshaft is returned to its normal running position by a spring inside the front end of the shaft. The relief cams open the exhaust valves 35 deg. early and close them at 22 deg. late.

In the Bassé-Selve engine (Fig. 333) the compression release is operated by means of a rod which lies horizontally along the outside of the camshaft casing directly underneath the exhaust-valve rocker arms. The rod is slotted in such a way (see separate detail, Fig. 333) as to form cams which lift the exhaust valve rockers when the rod is partially rotated by means of the hand lever at the end of the rod. With this device the exhaust valves remain open so long as the rod is in the rotated position.

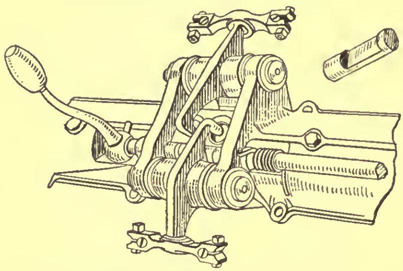

Fig. 333.—Compression release of Bassé-Selve engine.

In addition to the difficulty which is experienced in starting large engines by the propeller there is a considerable element of danger, which has proved fatal in many cases. To obviate this, recourse may be had to a portable engine cranker, which is usually available only at airdromes, or to a starting mechanism integral with the engine. Whatever the nature of the starting mechanism it should be thrown automatically out of action as soon as the engine fires and it should also go out of action if the engine fires before the dead center and starts to turn backward. The connection from the starter to the engine is through a dog or clutch, as in Fig. 334, which is pushed out of mesh when the engine starts forward but will remain in mesh if the engine starts backward.

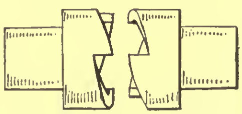

Fig. 334.—Dog clutch.

The acceleration of the engine shaft as a result of firing one of the cylinders is very great and if the engine starts backward this acceleration will be transmitted to the starting mechanism. As the starting mechanism is always geared, with a ratio which may be 100:1 or more, the motivating element will be given an enormous acceleration which will produce high teeth pressures in the gears and will probably strip the teeth of the gears or the dog projections of the clutch. If the gears are hand-operated through a crank the gear ratio may be 10 or 20 to 1, and, although the teeth may hold on a back fire, the mechanic will be endangered by the high speed of the handle. To safeguard the starting gear some kind of friction clutch or other safety drive should be placed close to the dog. Multiple-disc clutches are suitable on account of their compactness and low weight, but all such friction devices are uncertain in their action.

A different type of safety device with an oscillating member is shown in Fig. 335.[1] The oscillating member, *A*, takes no part in the transmission of the load. The engine drive is to the right of the figure. In normal rotation, *A* is driven by the gear wheel, *C*, and is free to oscillate as determined by the tapered sides of the stationary projections on *D*. If a reverse rotation occurs *D* prevents the rotation of *A*, which causes the sleeve on which *C* is mounted to move to the right, and thereby throws *C* out of mesh. As the action cannot take place until some backward movement has occurred this gear should be placed on a shaft geared to the engine shaft, so that release may occur quickly. In a hand-operated system it might be on the handle shaft.

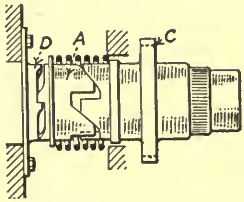

Fig. 335.—Safety clutch.

Integral cranking mechanisms are either operated by hand, by compressed air, or by electric motor. It is usually necessary to have the engine rotating at a speed of from 10 to 20 r.p.m. before regular operation will take place. Hand mechanism must have the operating handle at the side or rear of the engine and consequently must employ worm, bevel, or helical gears, usually in conjunction with spur gears. The efficiency of the gear train is poor and the attainable cranking speed very low. The hand mechanism for the Hispano-Suiza engine is shown in Fig. 336. It includes double-reduction spur gears, a dog clutch, a releasing spring and a gear-driven starting magneto.

Compressed air may be utilized (1) in a motor which cranks the engine, (2) it may be carbureted and sent through a distributor and exploded in the cylinders, or (3) it may go direct to the cylinders at a high pressure through a distributor and

[1] Sherman, *The Automobile Engineer*, December, 1919.

special non-return valves. If a compressed air motor is used, it can be run from the engine as an air compressor to store up the

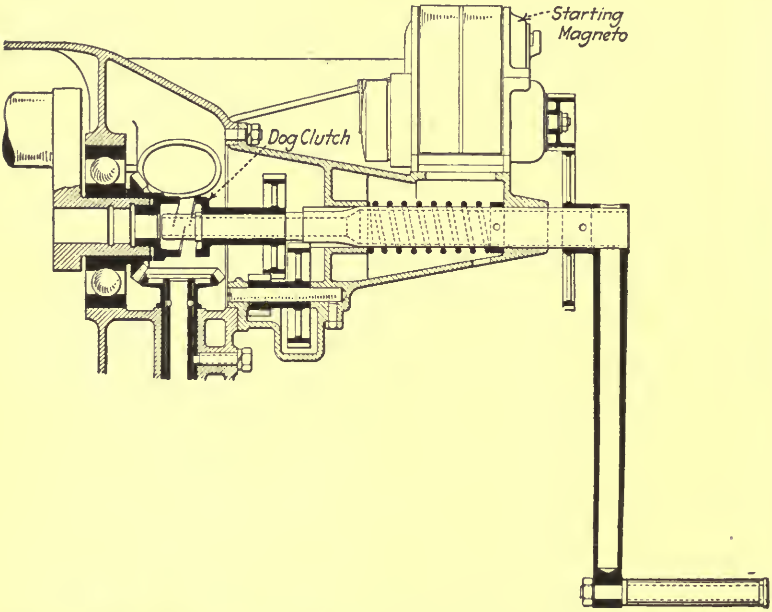

FIG. 336.—Hand-starting mechanism for Hispano-Suiza engine.

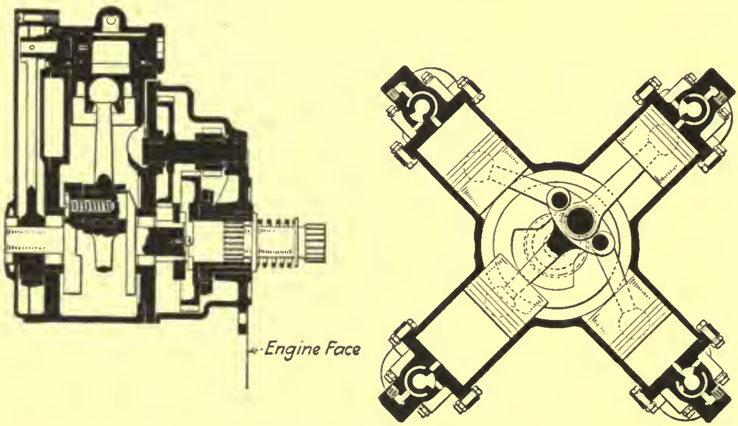

FIG. 337.—Radial air compressor.

air necessary for starting. In the Motor-Compressor Company's starter (Fig. 337) a multi-cylinder radial compressor is lined up with the crankshaft at the rear of the engine. When operated as a compressor it is directly connected through a positive clutch to the engine and runs at engine speed, compressing air up to 230 lb. pressure in a wire-wound tank. When used as a starter it drives the engine through a train of spur gears and rotates at seven times engine speed. Automatic arrangements are provided for throwing out the compressor when the air pressure reaches 230 lb. and for throwing out the cranker when the engine starts. The apparatus weighs about 50 lb. for a 200-h.p. engine.

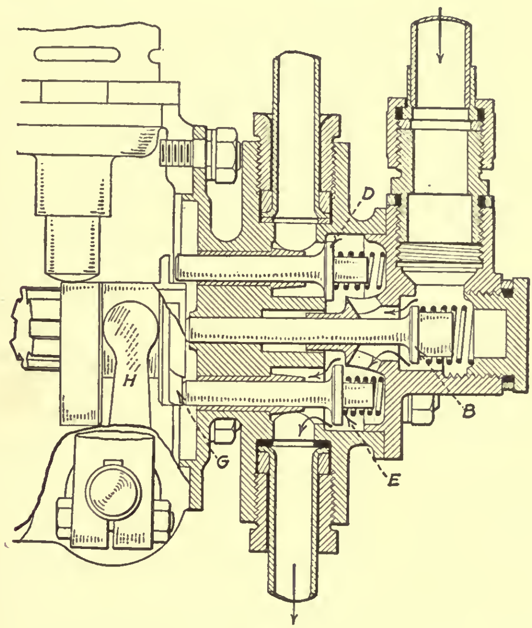

Fig. 338.—Compressed air distributor.

In the Christensen system compressed air is sent through a special carburetor, a distributor and non-return valves to the cylinders. The air pressure is sufficient to start the engine (say 100 lb.) and a retarded spark gives a late explosion and initiates regular operation. This system uses a minimum of compressed air for the starting. An air compressor driven by a hand-operated clutch from the crankshaft and an air tank are necessary parts of the system. The whole weighs about 40 lb. for a six-cylinder engine.

Compressed air can be used inside the cylinders without carbureting. If no air compressor is provided a steel air tank must be carried of sufficient capacity for several starts. This arrangement uses a distributor and non-return valves; it will weigh less than the Christensen system, but is likely to leave the aviator stranded on occasion.

The details of an air distributor are shown in Fig. 338. When the starting lever, *H*, is thrown in, the central main air valve, *B*, is opened by the boss at the end of the sliding sleeve and the individual valves, *D*, *E*, admitting air to the individual cylinders are opened in turn by the rotation of the face cam, *G*.

Another method which has been used but is now abandoned is to insert a black powder cartridge in a special fitting in the cylinder head. On detonation this will give a considerable pressure and should start the engine. It causes a deposit of carbon in the cylinder and is not adapted to remote control.

Electric cranking has been used considerably. It requires a 12-volt battery which will normally have to supply 100 amperes but may be called upon to give to 200 amperes for a half minute or more; an electric motor which will operate at 3,000 to 4,000 r.p.m., carrying a small spur gear on the armature shaft; and a double reduction gear with a speed reduction ratio of 100 to 150. The last gear is preferably fastened to the rear propeller hub flange, the motor reduction gears being mounted on the crankcase between the front cylinders and the propeller hub. The gears can be brought into mesh by the use of a solenoid wired in series with the motor. The Bijur starter used on Liberty-12 engines has a six-cell battery weighing 35 lb. with a capacity of starting the engine 150 times under normal conditions. The starting motor and gear may weigh 24 lb. and give an engine speed of 40 to 50 r.p.m. The maximum torque available at the engine crankshaft is 1,300 lb.-ft. An arrangement in which the gears are kept inside the nose of the engine is shown in Fig. 339;[1] this arrangement sup-

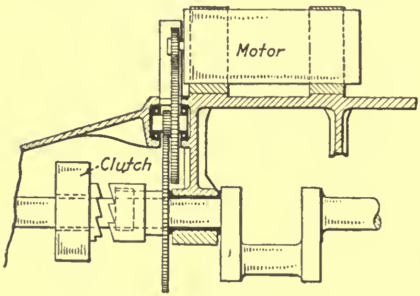

FIG. 339.—Electric starter.

[1] SHERMAN, *loc. cit.*

poses the starter incorporated in the engine design and not adapted, as usual, as an afterthought. If this starter is mounted at the rear of the engine it can be made a combined hand and electric starter by putting a dog clutch and cranking handle on the intermediate shaft.

Various **portable crankers** have been devised for airdrome use. The U. S. Air Service has used an electric cranker driven by an automobile-starting motor and storage battery through double-reduction gearing, and mounted on a motor truck. As the weight of the gearing does not have to be considered in this case it has been found unnecessary to provide an automatic release in case of engine back fire. The cranker is mounted in a spherical bowl permitting universal adjustment; it is brought up to the end of the propeller shaft and adjusted so that its shaft is in line with the engine crankshaft. An engagement lever then pushes the perforated face plate at the end of the cranker shaft against the front propeller hub flange when some of the nuts enter the perforations. The engagement lever is withdrawn as soon as the engine starts.

The Odier portable starter, which has had considerable use, employs a long single-acting steel cylinder and piston operated by carbon dioxide from a steel bottle of liquefied carbon dioxide. The piston carries a pulley at its free end, over which a cable is passed with one end fastened to the cylinder and the other wrapped around a drum and then fastened to an elastic cord. The drum has four bolts placed symmetrically around the periphery at one end—parallel to the drum axis—projecting sufficiently to engage a kind of dog clutch on the front propeller hub flange. The cylinder is carried on an inclined wooden arm and a vertical leg of adjustable height, so that the bolts can be brought up to the propeller hub level. The high pressure of saturated vapor of carbon dioxide (308 lb. per square inch at 0°F.) provides a large starting force in a cylinder of small diameter. The piston stroke is such as to give two revolutions of the engine with high speed. The cranker is thrown forward and out of mesh when the engine starts by the action of the dog teeth on the propeller flange. In case of a back fire the piston is pulled back, the gas is recompressed, and the elastic cord becomes slack and permits the drum to revolve freely. The weight of the whole apparatus is 44 lb. so that it can be carried in the plane if desired.

CHAPTER XIX

POTENTIAL DEVELOPMENTS

Increased Compression.—The best airplane engines give a notably better performance, both as regards fuel consumption and horse power per unit of piston displacement, than other gasoline engines. The possiblity of this improved performance results from the use of a higher compression ratio, which in turn is only possible through the use of the volatile aviation gasoline. Ordinary commercial gasoline containing larger fractions of the heavier paraffines (nonane and decane) would detonate at the compression pressures reached in airplane engines. The dependence of fuel consumption on the compression ratio is shown in the following table,[1] which gives the theoretical consumption of gasoline (lower heat value 18,600 B.t.u. per pound) per brake horse-power hour with an assumed mechanical efficiency of 90 per cent and with variable specific heats.

Ratio of compression	4.0	4.5	5.0	5.5	6.0
Gasoline per brake horse-power hour	0.416	0.398	0.375	0.361	0.350

The best modern engines use 0.45 to 0.50 lb. per brake horse-power hour with a compression ratio around 5.0; that is, the attained efficiency is 0.375 ÷ 0.45 = 83 per cent of the theoretical efficiency. With higher compressions this relative efficiency tends to increase. At maximum load, which is obtained only with richer mixtures (see p. 33), the relative efficiency averages 75 per cent for water-cooled and 76.5 per cent for radial air-cooled engines with aluminum cylinders.

The use of still higher compression pressures, and consequently higher efficiencies, is possible by the use of gasoline mixed with toluol, alcohol or other substance which will prevent detonation (see p. 236). This field of improvement is now under active investigation and promises considerable improvement.

[1] Gibson, *Trans. Royal Aeronautical Society*, 1920.

Use of Inert Gases.—A high compression pressure can be used without danger of detonations, and consequent preignitions, by taking in cooled exhaust gases with the charge. The influence of such admixture is shown in Fig. 340, which is taken from Ricardo's tests.[1] With a high-grade fuel which, when operating at full throttle and with an economical setting, detonates at a compression ratio of 4.85:1, the full power can be maintained by admitting inert gases in sufficient quantity to prevent detonation up to a ratio of compression of 6:1. That is, the decrease in weight of fresh charge taken in is fully compensated by the increase in engine efficiency up to that ratio of compression. If still higher compression is used (for an oversized engine for high-

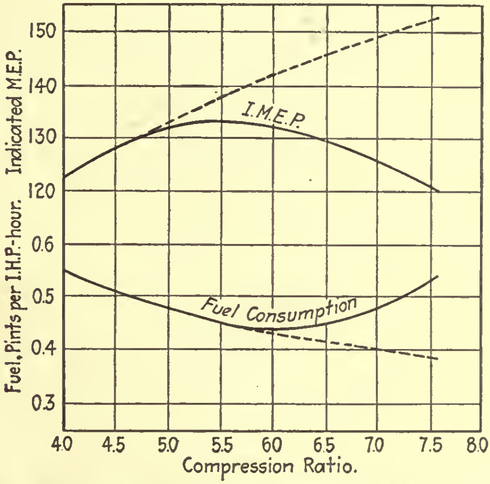

FIG. 340.—Effect of addition of inert gases on engine performance.

altitude flight, see p. 390) the power will fall off. The increase in economy by the use of the inert gases, with increase of compression ratio from 4.85 to 6, is seen to be about 6 per cent. The dotted lines show the performance obtained using a fuel doped to prevent detonation and without the admission of inert gases. With a compression ratio of 7:1 the exhaust-controlled engine develops 84 per cent of the power which would be developed by a pure non-detonating mixture.

Any increase in efficiency from increase of compression ratio will also increase the power output in practically the same ratio and is therefore doubly valuable.

[1] *Trans. Royal Aeronautical Society,* 1920.

The maximum brake m.e.p. possible for an engine with inlet valve closure of 50 deg. late, with volumetric efficiency of 88 per cent and mechanical efficiency of 90 per cent, is as follows:[1]

Nominal compression ratio	4.5	5.0	5.5	6.0
Maximum brake m.e.p	130.5	138.0	143.5	148.0

The best recorded results for both air- and water-cooled engines with compression ratios of 4.5 to 5.0 are very close to these figures.

Tests of single engines have shown consistently better results than those of multi-cylinder engines. The best air-cooled single-cylinder engines have shown a relative thermal efficiency of 91 per cent. The difference between the best performance of a single-cylinder water-cooled engine and the performance of a 12-cylinder Vee engine is from 8 to 10 per cent. The difference depends on the efficiency of the induction system and represents the possible saving by **better distribution** in the induction system.

As the efficiencies of the best single-cylinder engines using weak mixtures are within 10 to 15 per cent of the theoretical maximum, it is evident that little further progress is possible in improving the thermal efficiency of engines using the present cycle of operations. Increase in capacity (b.h.p. per unit of piston displacement) can be obtained by supercharging or by improving the volumetric efficiency. This latter method offers some chance for improvement as the measured values vary from 70 to 85 per cent with exceptional values up to 90 per cent.

The fire risk in airplanes could be practically eliminated by the use of kerosene as fuel. Kerosene may be used either (1) by vaporizing it outside the engine, or (2) injecting it into the cylinder as a liquid either during the suction stroke or at the end of compression. To vaporize a reasonable proportion the initial temperature must be not less than 140°F., which results in a reduction in the weight of the charge of about 20 per cent as compared with gasoline and a corresponding decrease in engine power. Furthermore, it is chemically much less stable than gasoline and detonates at a lower temperature so that a lower compression ratio must be used, which further diminishes the power and decreases the efficiency. The heavier fractions condense on the cylinder wall and, passing into the crankcase, thin the

[1] Gibson, *loc. cit.*

lubricating oil. Injection of the fuel during the suction stroke intensifies this last trouble but reduces the loss due to preheating. Injection at the end of compression presents many difficulties common to Diesel engines, which are not yet satisfactorily solved.

Modifications of the Otto cycle would seem to offer considerable possibilities for increased efficiency. The pressure at the opening of the exhaust valve is usually from 60 to 70 lb. per square inch. If more complete expansion could be obtained a considerable increase in efficiency might be effected. Attempts have been made to realize this potential increase in work along two lines, (1) by more complete expansion in the cylinder and (2) by expanding the gases after leaving the cylinder.

1. A lower terminal pressure in the cylinder can be obtained either (*a*) by throttling or cutting off the admission of the charge or (*b*) by making the expansion stroke longer than the compression stroke. The former method (*a*) is that of the oversized engine (p. 390) and is employed primarily for maintaining power at high altitudes. Its use requires a larger and therefore heavier engine for a given capacity, which is a serious detriment. The method (*b*) can be carried out by the use of a **variable-stroke engine** and has the incidental advantage of permitting a more complete scavenging of burned gases.

The **Zeitlin engine** which is now under development is an example of a variable stroke engine, but this feature is utilized in this case only to give better scavenging and thereby to permit the admission of a greater weight of charge. It is a single-valve, nine-cylinder air-cooled rotary which follows the Gnome engine in taking in its air supply through the open exhaust valve and mixing with it an overrich mixture through ports uncovered by the piston near the end of the suction stroke. The variable stroke is obtained by mounting, on the crankpin, eccentrics, which are driven by gearing around the crankpin in the same direction as the engine but at one-half engine speed. The connecting rods are mounted on these eccentrics and consequently the piston travel will vary throughout two revolutions of the engine. In the engine with 107.75 mm. crank throw, the working stroke of the engine is 181 mm.; the exhaust or scavenging stroke is 203.5 mm.; the suction stroke is 226 mm. and the compression stroke 203.5 mm. As the admission ports are not covered until the piston has made part of the compression stroke the effective compression stroke is practically the same as the working stroke. The admission ports in the cylinder are uncovered near the end of the suction stroke. The length of the scavenging stroke is such as to clear the cylinder almost completely of burned gases, so that the new charge is undiluted by them.

This engine is arranged to give **variable compression** by opening the exhaust valve during the early portion of the compression stroke and permitting some air to escape before it has time to mix with the overrich mixture. It is designed for a maximum compression ratio of 7 and has controls for decreasing this to 4.5. The maximum compression is for altitudes of 10,000 ft. or more; the minimum compression is for ground operation. The strength of the mixture will obviously change with the ratio of compression; the mixture will be rich at the ground and will be leaned to maximum economy strength at normal flying level.

2. The further expansion of the gases after leaving the cylinder can be carried out either in a gas turbine or in a reciprocating engine. The use of the **exhaust gas turbine** for driving a supercharging blower has already been discussed (p. 408); in view of the high speed of the turbine shaft, which is necessary if the turbine is to have fair efficiency, it is doubtful whether such a turbine could be geared down so as to help drive the propeller without excessive loss of power. For driving high-speed auxiliaries such as the supercharging blower it has an important field.

Compounding.—Expansion of the gases in a reciprocating engine can be accomplished by following the methods used in compound steam engines. The problem is simplified in some respects in the gasoline engine because one double-acting low-pressure cylinder can take the exhaust from four high-pressure cylinders and as the temperature of the gases is greatly reduced by the time they reach the low-pressure cylinder it might be possible to operate without trouble from excessive piston temperature. As the friction work in the high-pressure cylinders is equal to about 15 lb. per square inch of piston area there should be a pressure drop of at least that amount at exhaust, or with 70 lb. terminal pressure in the high-pressure cylinder the receiver pressure should be about 50 lb. Furthermore, in order to get a full charge in the high-pressure cylinder it would be necessary to have an atmospheric exhaust from the high-pressure cylinder at the end of the exhaust stroke and immediately after closure of the exhaust to the receiver. The piston displacement of the low-pressure cylinder would probably have to be about three times that of the high-pressure cylinder, or with the same stroke its diameter would be 1.7 times that of the high-pressure cylinder. Attempts to construct compound gas engines have been made in stationary types but without commercial success. The extra power obtainable has not justified the additional first cost and maintenance. It is possible, however, that more success may be met in the airplane engine where first cost is not of prime importance. The weight of the engine per horse power should not be increased by compounding and the weight of fuel used should be appreciably diminished.

Two-cycle.—A modification of the Otto cycle which offers possibilities of considerable reduction in weight per horse power is two-cycle operation. The normal Otto cycle requires four strokes for the completion of the cycle of which two are used solely for pushing out the burned charge and taking in the new charge. The essential parts of the cycle are unchanged if the exhaust and admission processes are speeded up and made, in part, simultaneous by using a slightly precompressed charge to sweep out the exhaust gases after the exhaust pressure has fallen substantially to atmospheric pressure. With this arrangement the cycle of operation may be completed in two strokes, the number of cycles per minute doubled and the horse power almost doubled. Considerable experience with two-cycle engines is available from stationary practice and marine practice and indicates the possibility of increase of the power output from a cylinder of given size by from 60 to 80 per cent but with a falling off in efficiency of 20 per cent or more.

There are two general methods of obtaining a precompressed charge, (1) by crankcase compression and (2) by the use of a separate compressor. With **crankcase compression** in a multi-cylinder engine, the crankcase must be divided to form a gas-tight compartment for each cylinder so that each piston on its down stroke may compress a charge which has been taken in during the up stroke. This arrangement is only possible in an engine with a single row of cylinders; it cannot be carried out in a Vee, W, or radial engine. It is simpler than the separate compressor but it limits the amount of charge which can be taken in to the volume sucked in during the up stroke of the piston and it will carry lubricating oil from the crankcase into the cylinder. A **separate compressor** should have a displacement volume greater than that of the engine cylinder in order to send some scavenging air into the engine cylinder for the more complete clearing out of the exhaust gases and also to give a pressure at the beginning of compression which is fully up to or slightly above atmospheric pressure. The compression pressure required is about 5 lb. per square inch. One double-acting air compressor would be required for two engine cylinders with discharge from the compressor direct to the cylinders. By the use of a receiver a larger compressor can be made to serve a larger number of cylinders. A centrifugal compressor would eliminate the need for a receiver.

The two-cycle engine is just beginning to be used in airplane service. The principal difficulties to overcome are low efficiency and heat trouble. If the additional weight of fuel that has to be

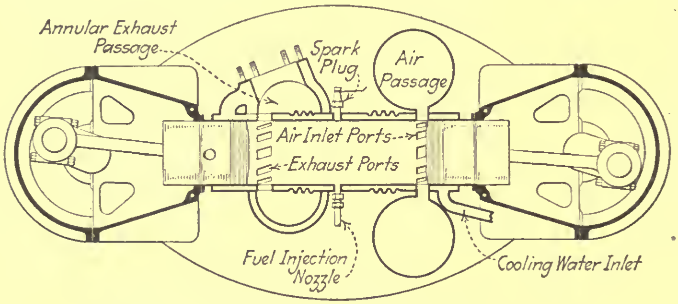

FIG. 341.—Junkers two-cycle solid-injection engine.

carried for a long flight is equal to the saving in engine weight there is little advantage in the lighter weight engine except for short flights. The doubled number of explosions in the engine per unit of time increases the cylinder temperature and leads to serious heat difficulties with the piston. Exhaust valve troubles are eliminated by the use of exhaust ports uncovered by the piston in place of exhaust valves. There has been considerable experimental work in this field but with no practical results so far except in the case of the Junkers engine.

The **Junkers engine** (Fig. 341) obtains high efficiency and eliminates heat trouble by departing entirely from the usual construction. There are six horizontal cylinders with their axes at right angles to the center line of the fuselage. The engine has two opposed pistons per cylinder and two crankshafts. All the pistons on each side of the engine are connected to a common crankshaft. The two crankshafts are geared together so as to make the two pistons of any one cylinder move in or out simultaneously. The combustion chamber is the space enclosed between the two pistons when they are on their inner dead center. Near the outer dead centers the pistons uncover cylinder ports, the exhaust ports (on the left) being uncovered first and the air admission ports (on the right) shortly afterwards. The propeller shaft carries the central gear with which the gears on the two crankshafts mesh; a blower is operated from the propeller shaft. There are no valves and no carburetors. When the pistons are on their outer dead centers, air from the blower passes through the cylinder from right to left and clears out the exhaust gases. This air is compressed, while the pistons make their inward strokes, to a pressure of 210 lb. per square inch or more. Fuel is then injected into the combustion space through the nozzle at the bottom of the cylinder, is ignited by a spark plug immediately above it, and expands, driving the two pistons outward. The pistons are equipped with a special cooling device. They are made with a cavity which is partly filled with a heavy oil and then sealed. The oil is violently dashed backwards and forwards by the motion of the piston, absorbs heat from the piston head and carries it to the cooled piston sides. The efficiency of this method of cooling is shown by tests with thermo-elements which indicated a maximum temperature of the piston head of 350°F. at maximum speeds and loads.

The advantages of this method of construction appear to be manifold. The high compression gives high efficiency, which is helped by the small heat loss during explosion resulting from the smallness of the cooling surface of the combustion chamber. The excellent scavenging permits higher volumetric efficiency, which in conjunction with higher efficiency gives a higher m.e.p. than is obtainable in other types. The i.h.p. is consequently more than twice that obtained for the same piston displacement in four-cycle engines and the weight is reduced to 1.5 lb. per horse power. The balancing of the reciprocating parts is practically perfect because the two pistons of each pair are at all times moving with equal accelerations in opposite directions; this condition is favorable to high engine speeds. The large size of the gas inlet and exhaust ports combined with positive admission of the air and fuel permits also high volumetric efficiency at high speeds; consequently this engine can be operated at higher speeds than the usual type. As the propeller is geared it can be run at its most favorable speed. A further feature of the engine is the great reduction of fire risk resulting from the direct discharge of the fuel into the engine cylinder; there is no explosive mixture outside the engine cylinder and less liability to fuel leaks. Moreover, a less volatile fuel can be burned.

The principal apparent objection to the Junkers engine is the difficulty of accommodating an engine of its width in the fuselage. In the design shown in Fig. 341, the over-all width is 8½ times the stroke, or with a 6-in. stroke the width is 4 ft. 3 in. This dimension is exceeded in large radial engines (see p. 195). The mechanical efficiency is low on account of the blower work and of the gearing losses; other Junkers engines, not adapted to airplane use, have given mechanical efficiencies of about 73 per cent.[1]

Among the possible developments for airplane engines are Diesel engines, gas turbines and steam plants either turbine or reciprocating. The **Diesel engine** would be most advantageous in view of its higher efficiency and the safety and low cost of the fuels which it could employ. The difficulties in the way of its employment in airplanes in its present stage of development are its excessive weight and the large size of individual cylinder below which it has not been found practicable to go. In the modified form of the Hvid and similar engines, smaller size cylinders become practicable but the weight is still excessive. It is by no means certain that it will be found practicable to operate this cycle successfully at the high speeds necessary for airplane use.

Gas turbines have been under active development for over fifteen years but the difficulties inherent in them have not as yet been overcome without sacrificing their potential efficiencies. The principal troubles are those resulting from the high temperatures to which the combustion chamber, nozzles, and buckets are subjected. When these temperatures are reduced, by injecting water or excess air, the efficiencies fall off. Moreover, the efficiencies are low unless the air is precompressed and as centrifugal compressors seldom have efficiencies above about 60 per cent, a large part of the power developed in the turbine is utilized in driving the compressor. Over-all thermal efficiencies are usually about 5 per cent, although an unsubstantiated value of

[1] SCOTT, *Jour. Soc. Aut. Eng.*, 1917.

20 per cent has been claimed for a 1,000-h.p. unit. The gas turbine, if applied to airplane propulsion, would have to be geared down, probably with double reduction gear. Its simplicity and light weight have attracted many inventors, but there are no indications that it is ever likely to become practically available.

Steam-power plants have the great advantage over internal combustion engines of maintaining their power at all altitudes. Both turbines and reciprocating engines of light weight are fully developed and can be regarded as immediately available for airplane use. The difficulties arise in connection with the boiler and oil burner. For efficiency the steam must be generated at high pressure and with high superheat, but no construction of boiler is known which does not entail weights which would be excessive for aircraft. A satisfactory kerosene (or other fuel) burner for operation with high rates of combustion in small space would also have to be developed; the fire risk from such apparatus would probably be considerable. And finally, the efficiency of the best steam plants is not nearly so high as that of existing airplane engines. It seems very unlikely that steam plants will ever be employed in aircraft.

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