MACHINE DESIGN

PART I.

American School of Correspondence

(Chartered by the Commonwealth of Massachusetts.)

BOSTON, MASS.,
U. S. A.

MACHINE DESIGN

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INSTRUCTION PAPER

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MACHINE DESIGN.

Machine design treats of the design and construction of machines and their various parts.

A **machine** is a combination of movable parts, arranged on a supporting frame, and placed between the source of power and the work.

The *object* of a machine is to transform the energy supplied at the point where the machine receives its motion, into work at the point where the resistance is overcome.

The various parts may be arranged to *change the direction or velocity* of the power or to *overcome great resistance with small force.* For instance, a steam engine changes rectilinear to circular motion while a planer changes circular to rectilinear. A lathe is a familiar example of the change in velocity. Hydraulic presses and testing machines illustrate the overcoming of a great resistance by a small force.

A machine cannot move of itself nor create power. According to the law of the conservation of energy, no increase of power can be obtained from any machine. If a machine were frictionless, the product of the force exerted at the driving point and the velocity of the driving point would equal the product of the resistance and the distance through which the resistance is overcome in the same time.

The operation of machines depends upon two conditions; the *transmission* of certain forces and the *production* of definite resultant motions.

In designing machinery both of these conditions must be considered. The machine must be constructed so that each part will bear the strains placed on it and also have the proper relative motions.

The nature of these **relative movements** is independent both of the power transmitted and of the dimensions of the parts. We see that this is true; for in the model of a machine, the dimensions of the parts may vary considerably from those requisite for strength. At the same time, the relative motions of the parts of a model which may be worked by hand, are the same as those of the larger machine which perhaps transmits 1000 H. P.

Pure Mechanism treats of the motion and form of parts of machines and the manner of supporting them.

Constructive Mechanism treats of the calculation of the forces acting on these parts. It involves the selection of materials, and the calculation of the dimensions for requisite strength and stiffness.

A **Mechanism** is a portion of a machine where two or more parts are so arranged and connected that the motion of one compels a definite motion of others.

Machines are made up of trains of mechanisms. The sewing machine, watch and printing press are examples.

Motion and **rest** are relative terms. If we consider some point or body as fixed, motion may be either relative or absolute. For this work the earth is assumed to be fixed and motion referred to it absolute.

A point moving in space follows a line, either curved or straight which is called its **path**.

Direction, like motion, is relative to continuous motion. If a point continues to move indefinitely in the same direction, it is said to have **continuous motion.** In this case, the path must be a closed curve. A shaft turning on its bearings or the crank-pin of an engine, is an example of this motion.

Intermittent Motion. When a part of a machine has motion in alternate directions and definite periods of rest, it has intermittent motion.

Reciprocating Motion. If a point travels in the same path, alternately in opposite directions, its motion is said to be reciprocating. If the motion is reciprocating and also circular, it is called **vibration.**

Velocity. The ratio of space to the time is called *linear* velocity, if the path of the moving body is a straight line. If the path of the body is a curve the ratio of space to time is called *angular* velocity.

Velocity may be uniform or variable according as the spaces traversed in equal times are equal or unequal.

$$\text{Velocity} = \frac{\text{Space}}{\text{Time}}$$

$$\text{Space} = \text{Time} \times \text{Velocity}.$$

The unit of space is usually one foot; the unit of time, one second. Hence velocity is expressed in feet per second.

Angular velocity is measured by the number of units of angular space passed over in a unit of time. Angular space is measured by circular measure of the ratio of the arc to the radius. The unit angle is the angle subtended by an arc equal in length to the radius. Angular velocity may be expressed in number of revolutions in a unit of time; one revolution is represented by 2π in circular measure.

$$\text{Angular velocity} = \frac{\text{Linear velocity}}{\text{Radius}}$$

$$\text{Linear velocity} = \text{Angular velocity} \times \text{Radius}.$$

Suppose we wish to find the linear velocity of some point on the periphery of a fly-wheel. Evidently the point will travel, during one revolution, a distance equal to the circumference of a circle of the given radius. The circumference is, from geometry, πd or $2\pi r$. Let n be the number of revolutions per minute, then the linear velocity is $2\pi r n$.

A fly-wheel is 20 feet in diameter and revolves at the rate of 50 revolutions per minute. Find the linear velocity of a point on the periphery.

$$\text{Linear velocity} = 2\pi r n = 2 \times 3.1416 \times 10 \times 50$$
$$= 3141.6 \text{ feet per minute.}$$

How far from the centre of this wheel must a point be to have a velocity of 20 feet per second?

$$V = 2\pi r n, \text{ or } r = \frac{V}{2\pi n}.$$

$$\text{Then, } r = \frac{1200}{2 \times 3.1416 \times 50}$$
$$= 3.82 \text{ feet. Ans.}$$

EXAMPLES FOR PRACTICE.

1. A wheel 3 feet in diameter makes 25 revolutions per minute. What is the linear velocity of a point on the circumference? Ans. 235.62 feet per minute

2. A wheel makes 300 revolutions per minute. How far from the center must a point be to have a linear velocity of 2827.44 feet per minute? Ans. $1\frac{1}{2}$ feet.

3. Find the linear velocity of a fly-wheel when the angular velocity is 188.5 revolutions per minute, the wheel being 10 feet in diameter. Ans. 942.5 feet per minute.

Revolution. A point revolves about an axis when it describes a circle the center of which lies within, and its plane of rotation is perpendicular to, the axis. If the axis passes through the body, as in the case of a wheel, the motion is called both rotation and revolution.

A body, like the earth for instance, may rotate about its own axis and also revolve in an orbit about another axis.

Cycle of Motions. In case the parts of a mechanism go through a series of motions which are repeated, each time the order of the motion of the several parts being the same, the series is called a cycle of motions. In the steam engine every revolution is a cycle because each series of motions is repeated for every revolution. Two revolutions of the crank are necessary for one cycle in some types of the gas engine.

The part or piece of mechanism which causes motion, or to which the power is applied is called the **driver** and the part whose motion is caused by the movement of the driver is the **driven** or **follower.**

Frame. The structure which supports or holds in position the moving parts and regulates the path of motion is called the frame. The motions are often referred to the frame, as it is usually fixed, that is, without motion. An exception to this is the locomotive frame.

Transmission. One object cannot move another unless the two are in contact or are connected by some body that is capable of communicating motion from one to the other. The above statement does not take into account the action of natural forces, such as gravity, magnetism, etc.

Motion transmitted by contact is seen in friction gearing, gear wheels, etc. Belts, rope gearing, levers, links, etc., are examples of motion by intermediate connection. Connectors are either rigid or flexible.

Motion. Mechanism may be used to change the motion of the follower from that of the driver. It may differ in direction, kind, or velocity.

Work is the overcoming of resistance through distance. It is the product of the resistance and the space through which it is overcome. If a body is lifted from the earth, against the attraction of gravity, the resistance is the weight of the body and the distance is the height to which the body is raised. The work done is equal to the weight of the body multiplied by the distance.

The **Unit of Work** is the foot-pound, that is, the amount of work done in lifting one pound through one foot. If F equals the force or weight and S equals the space or distance through which F is moved, then work $= F \times S$. $S =$ velocity multiplied by time $= V \times T$. Then if we raise 5 pounds to a height of 20 feet we do $5 \times 20 = 100$ foot-pounds of work.

Energy is the capacity for doing work. There are two kinds of energy, potential and kinetic.

Potential Energy is energy of position; water stored in a reservoir for example. The water is capable of doing work by means of a water wheel. Potential energy is measured in foot-pounds, that is, it is the weight of the body multiplied by the distance through which it is capable of acting.

Potential energy may also exist as stored heat or chemical energy, as in fuel, gunpowder or electric energy.

Kinetic or Actual Energy is the energy of a moving body. The energy in a moving body is the work which it is capable of performing against a resistance before it is brought to rest. It is equal to the work required to bring it from rest to its actual velocity. Kinetic energy is measured as the product of the weight of the body and the height through which it must fall to acquire the actual velocity. From the laws of falling bodies this height equals the square of the velocity divided by twice the value of the earth's attraction. Then

$$h = \frac{v^2}{2g}$$

and energy,

$$E = wh = \frac{wv^2}{2g}.$$

The **weight** of a body divided by g is called the **mass,** *i. e.*,

$$m = \frac{w}{g},$$

then substituting m for $\frac{w}{g}$ in the equation

$$E = \frac{wv^2}{2g}, \text{ it becomes } E = \frac{mv^2}{2}.$$

Energy is the capacity of doing work. The units of work and energy are the same; then

$$F\,S = wh = \frac{wv^2}{2g} = \frac{mv^2}{2}.$$

Power is the *rate* of performing work. It is equal to the work done divided by the time and is expressed as foot-pounds per minute or per second. Thus **horse-power** is a measure of power, being equal to 33,000 foot-pounds per minute or 550 foot-pounds per second.

$$\text{Power} = \frac{FS}{T}.$$

MATERIALS.

The principal materials used in the construction of machinery are cast and wrought iron, copper, wood, brass and other alloys. The properties and processes of manufacture for iron and steel have been described in Metallurgy.

CAST IRON.

Cast iron is used to a considerable extent in the construction of machines. For the heavy massive parts, the frames of lathes, steam-engines, planers, etc., for example, it is the best material. It is *not* suitable for parts requiring strength, elasticity or those subjected to shocks. For this reason piston-rods, connecting-rods, shafts, etc., are usually made of steel or wrought iron.

Many complicated shapes that cannot be forged are readily cast. The ease with which parts may be given the desired shape makes cast iron valuable.

Cast iron contains 3 to $4\frac{1}{2}$ per cent. of carbon with a little silicon. The hard and white varieties are used in the manufacture of wrought iron. The gray irons are used in the foundry.

Cast iron is made into the desired forms by melting it in a cupola and pouring into moulds. The moulds are made in sand or loam from patterns of pine wood. Patterns are made a little larger than the required casting because iron in solidifying contracts about $\frac{1}{8}$ inch per foot in each direction. This contraction is called shrinkage. In making a pattern a shrinkage rule is used which is about $\frac{1}{8}$ inch longer per foot than the standard.

Castings are likely to be put into a state of internal stress because of contraction when cooling. If some parts of the casting contract more than others, the casting may become twisted. Thin parts of the castings solidify first. The contraction of the fluid parts strains the portions already set and their resistance to deformation causes stresses to be set up in the parts which are solidifying.

For example, the form shown in Fig. 1 has a rigid flange

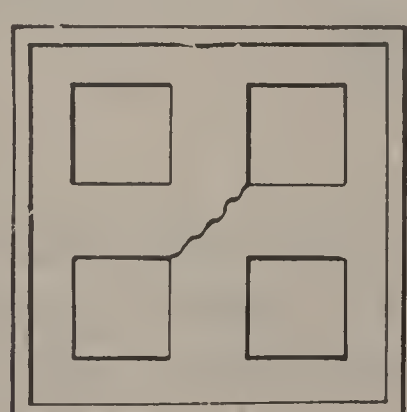


Fig. 1.

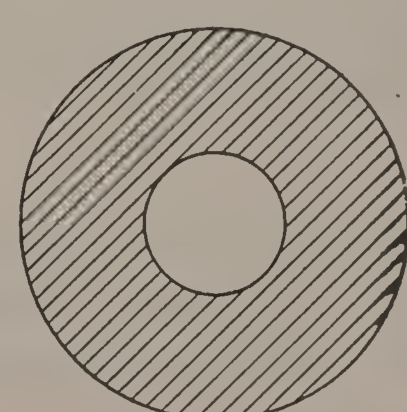


Fig. 2.

surrounding the inner part. If the contraction of the cross piece takes place more slowly than the rim, it is likely to fracture. In a thick cylinder, as shown in section in Fig. 2, the outer portions solidify and begin the contraction. The contraction of the inner induces pressure in the outer portion, which being rigid causes a resistance to contraction of the inner layers and puts them in tension. A cylinder so constructed is not strong to resist bursting pressure. If the cylinder is cast while water circulates through the core, the reverse distribution of initial strains is set up. This insures a stronger cylinder because the inner layers are in a state of compression and the outer portions are in tension.

The arms of pulleys may be broken by tension if the rim is thin and rigid. If the arms set first the rim may break near them. To have successful castings, the designer must carefully consider the dimensions of the various parts.

On account of these initial strains, that cannot be calculated, cast iron is unreliable. Cast iron structures usually have excessive dimensions to insure safety.

In cooling, the crystals of cast iron arrange themselves perpendicularly to the surfaces from which heat radiates. For this reason all corners should be well rounded as shown in Fig. 3, so that the arrangement of the crystals will make the castings strong.

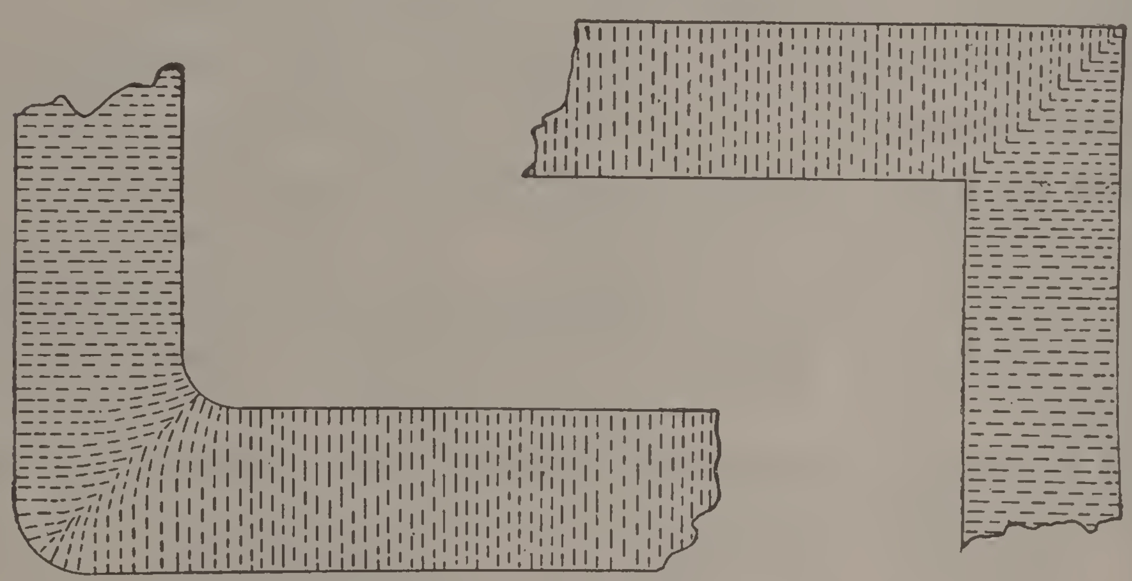


Fig. 3.

Chilled Castings. If castings are cooled rapidly during solidification, the graphite is prevented from separating from the iron. This causes the iron to become harder. In order to chill the cast iron, the mould is made of or lined with this same material. The mould which is lined with loam for protection, is a good conductor of heat and the molten cast iron is cooled or chilled during solidification.

The chilling usually extends to a depth of $\frac{1}{8}$ to $\frac{3}{8}$ of an inch from the surface; the interior remaining soft.

Malleable Cast Iron. Malleable cast iron is made by surrounding castings with oxide of iron, powdered red hematite or peroxide of manganese; keeping them at a high temperature for a considerable time according to the size of the casting. The elimination of carbon converts the cast iron into a crude form of wrought iron. Malleable castings will stand blows better than ordinary castings.

Cast iron is stronger than wrought iron when under pressure; but it is much weaker under tension and impact.

WROUGHT IRON.

Wrought iron is made from cast iron by eliminating part of the carbon. It is strong and tough and can easily be welded. For these reasons it is used for parts of machines and structures requiring strength and of simple form. Wrought iron parts are shaped by forging and finished in the machine shop; steam hammers being used on the heavy portions.

Wrought iron is rolled into plates, round and square bars, angle, tee, channel, I beam sections, etc. Large wrought iron structures are built up of bars or plates riveted or bolted together.

Wrought iron that has been rolled when cold has a greater tensile strength than before rolling; but its ductility and toughness is reduced. Annealing, or heating the iron to a red heat and allowing it to cool slowly, restores it to the original condition.

Compression of iron when cold increases its strength but reduces its ductility and toughness; annealing reduces strength and increases toughness and ductility. If the iron is rolled or hammered when hot, compression and annealing are carried on at the same time.

Wrought iron is used for piston-rods, shafts, connecting-rods, bolts, nuts, chains, etc.

STEEL.

Steel is far the most useful material used in machine design. Its properties depend upon the percentage of combined carbon, silicon, manganese, phosphorus, etc. Steel contains from .15 to 1.5 per cent. of carbon.

Formerly it was difficult to get sound castings, but by the use of silicon, aluminum and other elements and prolonged annealing, the internal stresses are destroyed.

Steel can be welded, but greater care is necessary than in the welding of wrought iron.

Tempering greatly increases the usefulness of steel, since it becomes hard if heated and cooled suddenly. With good steel almost any desired hardness may be obtained. The steel is heated to the temperature indicated by the color of the oxide which forms at its surface and is then quenched in oil or water. Hardness makes it suitable for cutting tools. When tempered it is hard, strong, has high elastic limit and little ductility.

COPPER.

Copper is a reddish metal of great ductility and malleability. It is usually rolled or hammered into shape because it doesn't cast well. Copper can be welded, but as it requires considerable care to make a good joint, pieces are more often joined by brazing. It can be drawn into wire. The tensile strength of cast copper is about 20,000 pounds per square inch; of forged copper about 30,000 pounds per square inch.

Hammering, rolling and wire-drawing increases the tensile strength, but makes it hard and brittle. It can be made soft and tough by annealing. It is expensive and is used for wire, fittings and tubing. Its strength is less than that of wrought iron and decreases rapidly with rise of temperature.

ALUMINUM.

Aluminum is a soft, ductile, malleable metal of bluish white color. It is very light; next to magnesium the lightest of the useful metals. Its strength is about one-third that of wrought iron. Aluminum casts well, the shrinkage being about the same as brass. The readiness with which aluminum unites with other metals makes it valuable for alloys. It can be electrically welded but doesn't solder well.

BRONZE.

Bronze, or gun-metal, is an alloy of copper and tin ; about 90 parts copper and 10 parts tin. It makes good castings. Bronze is harder and less malleable than copper. Copper-tin alloys are used for bearings because it is softer and wears faster than wrought iron or steel shafts.

The hardness of bronze depends upon the proportion of tin ; to increase hardness increase the amount of tin. An alloy of 92 parts copper and 3 parts tin is a soft bronze used for gear wheels.

Phosphor-bronze is made by mixing 2 or 3 per cent. of phosphorus with ordinary bronze. Soft phosphor-bronze has a tensile strength of about 45,000 pounds per square inch ; harder varieties have about 65,000 pounds and hard unannealed wire has about 150,000 pounds. It is used for pump-rods, propellor-blades, etc.

Manganese bronze, called white bronze, is an alloy of ordinary bronze and ferro-manganese. Like phosphor-bronze it is used in marine work, because it resists the corroding action of sea-water. Manganese bronze is equal in tensile strength and toughness to mild steel and can be easily forged.

BRASS.

The alloy of copper and zinc is called brass ; sometimes tin and a little lead are added. For bearings it has about 60 per cent. copper, 10 per cent. zinc and 30 per cent. tin and lead. Naval brass has 62 per cent. copper, 1 per cent. tin and 37 per cent. zinc. Red brass consists of about 37 per cent. copper and for the rest about equal parts of tin, zinc and lead. Brass is used for bearings, wire, fittings and ornamental work. Its tensile strength is about 23,000 pounds per square inch.

FUSIBLE ALLOYS.

Fusible alloys are made of tin, lead and bismuth. The melting point varies with the percentages of the various constituents. If made of 2 parts lead and 1 part tin, it melts at 475° F. ; if 1 part lead, 1 part tin and 4 parts bismuth, the melting point is about 200° F. An alloy of 1 part cadmium, 4 parts bismuth, 1 part tin and 2 parts lead melts at 165° F.

BEARING ALLOYS.

The principal constituents of bearing alloys are copper, tin, lead, zinc, antimony and aluminum. The bronzes contain a large per cent. of copper. A good bearing alloy is made of copper, 77 parts by weight, tin 3 parts and lead 15 parts.

Babbit metals have various proportions; hard babbit having about 89 per cent. tin, 4 per cent. copper and 7 per cent. antimony.

There are many other alloys containing the metals in varying proportions according to the intended use.

WOOD.

Wood is but little used in machine construction. Soft woods like pine are used for patterns; hard varieties, oak and lignum-vitæ for examples, are used for bearings. Sometimes levers are made of wood and the pulleys of some lathes are constructed of the same material. The cogs of mortise wheels are often made of beech or horn-beam.

SHOP PROCESSES.

In designing machinery it is necessary that the parts may be easily made. A finely finished pattern is of no value if it cannot be taken from the mould. Complicated castings and forgings should be used only when absolutely necessary. Simple designs are usually the best.

For **casting,** patterns or models are made from wood in the pattern shop. The pattern maker has to consider and make allowance for shrinkage in casting, for turning, boring and finishing. He arranges the patterns in such manner that the moulding, casting and finishing may be most cheaply done. Some parts can be moulded only by the use of cores. Parts to be finished by cutting tools must be so placed that they will not be unsound by reason of blow holes. The founder follows the specifications of the drawings; mixing the pig iron in different proportions so as to get the required strength and softness.

Forging is the operation of shaping wrought iron or steel without melting. These materials become plastic without fusing. The pieces are hammered or rolled into shape. If the work is light the smithing is done by hand; but when large forgings are made, steam hammers are used. When the pieces are very large, great skill is required to arrange the operation so that the result shall be a homogeneous sound piece.

Fitting, finishing, boring and **turning** are the operations of cutting the rough products of the foundry and forge to accurate dimensions. Fitting, boring and turning are done by steel cutting tools which shape the metal when cold. Cutting operations include chipping and filling, drilling, turning, planing, shaping and milling.

Conical surfaces, screws and nuts can be made in the lathe.

STRAINS IN MACHINES.

There are forces acting on the several parts of a machine which will cause them to give way if they are not sufficiently strong. Among these forces are the following:

1. The **useful load** caused by the power transmitted from the point of receiving the energy to the point where the useful work is accomplished.

2. **Resistance** due to friction in the machine.

3. Forces due to **inertia** caused by change of velocity of the moving parts.

4. **Weight** of parts of machines.

5. **Centrifugal forces** caused by changes in direction of motion.

The total action caused by the above forces is called the straining action on whatever part is considered. This straining action varies with the changes of working load, with the variation of position of the parts, with the change in speed, etc. In designing the various parts it is necessary to consider under what conditions the straining actions are greatest and calculate the dimensions of those parts to safely stand that action.

It is obvious that the maximum working load must be *less* than the breaking load. In most cases it should be very much less. Generally it is much easier to determine (by means of testing machines) the breaking strength than it is the working stress. In order to be sure of sufficient strength, it is customary to divide the breaking strength by the **factor of safety**, to find the allowable working load. Results from actual cases provide us with average factors of safety for various conditions. In case the straining actions are well known and the stresses are steady, the factors are small. A large factor is necessary when the straining actions are likely to be greatly in excess of the calculations, when the material is not reliable and when the parts are liable to shock. Some designers never use the term factor of safety, but know from experience that the various materials will safely carry a certain load under given conditions.

In most cases a permanent set would be injurious; it might prevent the movements of some parts of a machine. Under these conditions it is evident that the working stress must be less than the elastic limit.

MACHINE DRAWINGS.

Machines are designed from principles obtained by successful practice and from mathematical calculations. In order that both the designer and the mechanic may have a clear idea of the work, the designer makes a **drawing** of the machine. The drawing indicates the size and shape of the various parts and how they are to be put together. By means of the drawing, the designer calculates the relative motions of the parts and arranges them so that they will not interfere with each other. He calculates the sizes for strength and considers the modifications which will produce the greatest efficiency, or least cost of manufacture. The drawing indicates how the work is to be performed and distributed in the different shops. All dimensions, names of materials and finish marks should be clearly shown so that the workman, by carrying out accurately the ideas of the designer, may produce the desired machine.

Usually several views of the part to be made are shown. Sometimes it is necessary to show **sections** in order that the internal construction or sectional shape may be easily understood. These sections are usually drawn through the axis, or center, but it is sometimes advisable to show sections of other portions. Where the drawing shows a section, the portions of metal or wood supposed to be cut are covered with parallel lines at equal distances and usually oblique. These sections are called **hatched, cross hatched,** or simply **sectioned.** The character of the lines.

full lines, dotted, broken, light or heavy, indicate the material supposed to be cut. One kind indicates cast iron, another steel, another brass, etc. There is no standard for cross hatching, different draughtsmen using lines of various character. There is likely to be a confusion unless the parts have the name of the material printed on or near it, or a key is provided.

Fig. 4 shows the lines as generally used; those representing

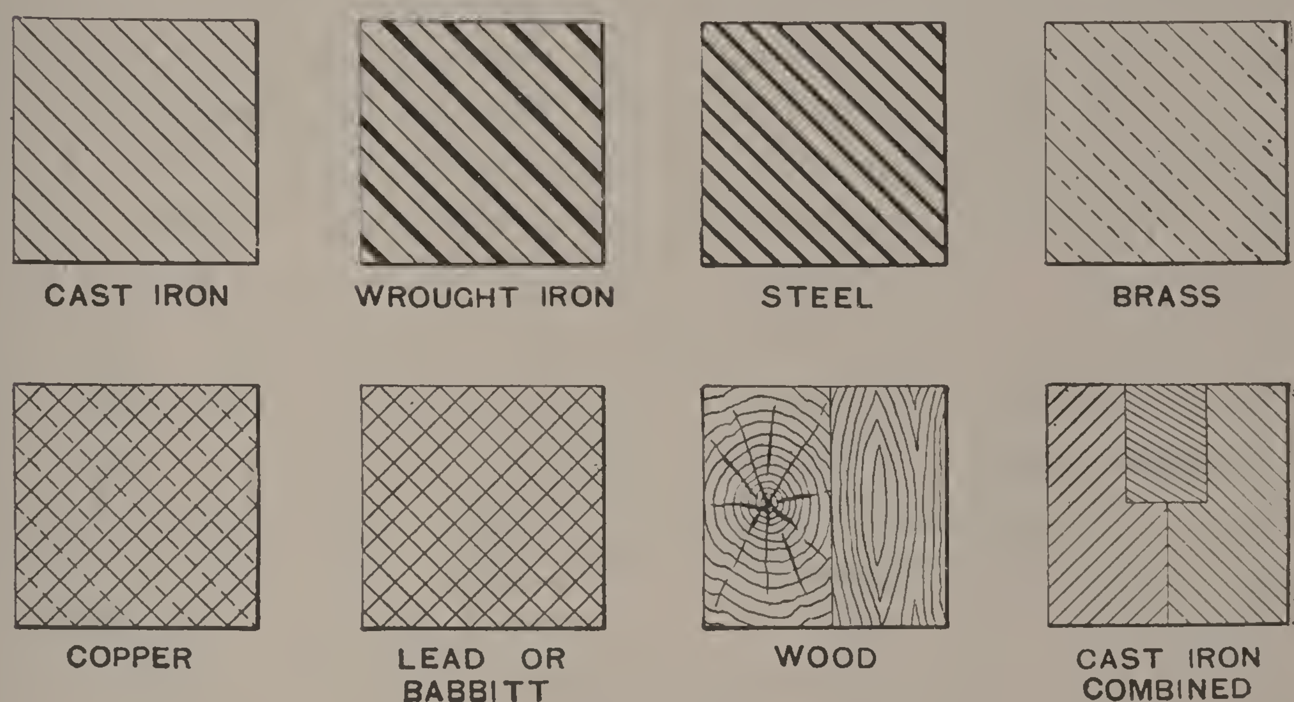


Fig. 4.

cast iron, brass, wood and lead being almost universal, the others are subject to more change. The lines may run from left to right, or right to left; in case two or more parts of the same metal are brought together it is necessary to avoid confusion by varying the direction and angles of the lines. If the hatching were to be the same, the parting line would be confused and one might think it all one piece.

When drawing designs of the details, it is well to make them as large as is convenient. The **scales** in general use are full size; half size, 3 inches or $1\frac{1}{2}$ inches = 1 foot. A drawing is never made $\frac{1}{3}$ size or by such scales as 2 inches or 1 inch = 1 foot.

A **working drawing** is one that shows all the dimensions of an object in such manner that the object may be made by reference to the drawing. It is a practical application of the study of projections. Usually three views are sufficient, **elevation, plan** or horizontal projection and **end view.** Besides these views, **sections** to show the interior construction are added.

It is not sufficient to draw the various views and sections the correct size; the **dimensions** also should be placed on the drawing. The workman can tell immediately the size of any given part without scaling it from the drawing. Although desirable, it is not necessary that a drawing be made accurately, provided all the dimensions are put in correctly. The chances of error are greatly reduced by removing the necessity to scale off dimensions.

Dimensions should be used systematically and wherever necessary. In placing dimensions on drawings, a line should be drawn from one point to the other. The number representing the dimension is placed in the space left for it at the centre of the line. These lines should be either fine full lines or dashes about $\frac{1}{2}$ inch long. Arrow heads are placed on the ends of the line, the heads or vertex of the arrow just touching the points or lines. In case the dimensions are very small, the arrow heads may be outside instead of between the lines, or pointing toward each other. The dimensions should be written in feet, inches, halves, quarters, eighths, sixteenths, etc. of inches. Fractions should be reduced to lowest terms. Write $\frac{3}{8}$, not $\frac{6}{16}$, nor $\frac{12}{32}$. The dividing line of the fraction should be parallel to the direction of the dimension line: never an oblique line because the oblique line may be mistaken for some part of a number. Feet are represented by the symbol $'$, inches by $''$. The inch marks should be placed after the fraction not between the whole number and the fraction; thus, eight feet, seven and three quarters inches should be written $8' — 7\frac{3}{4}''$, not $8'\ 7''\frac{3}{4}$. When the length is even feet it is usual to write it $8' — 0''$ in order that the workman may know that the inches were not left off by mistake.

It is necessary to get in all the important dimensions, especially the "over-all" dimensions, so that the workman will not be compelled to add up several small dimensions in order to select his stock.

Dimensions are often placed between two views and usually outside the several views. When placed outside, extension lines are used, that is, a fine or dash line is drawn as a continuation of lines or edges.

In placing the dimensions of a circle, give the diameter, not the radius. When an arc is used, give the radius. Holes may be located by giving the dimensions from the outside, or from the center of figure, to the center of the hole. The distance from center to center shows their distance apart. In case the holes are arranged in a circle, as in a cylinder head for instance, give the diameter of the circle whose circumference passes through the center of the holes.

In making sectional views the plane of the section passes through the center line of a shaft, bolt, screw, or cylinder, and the cylinder part is not represented in section but in full.

Sometimes, in addition to the above views an isometric or oblique projection is made. In the **isometric** projection only one view is used. The object is placed in such a position that its lines or edges are parallel to three rectangular axes. The dimensions are measured accurately on these lines, or lines parallel to them, and the lengths are true, not foreshortened, as in perspective drawing. Lines which represent length and breadth make angles of 30° with the horizontal and those representing thickness are vertical lines.

Oblique projections are similar to isometric projections except that the lines which make angles of 30° with the horizontal in the isometric projection make angles of 45° in the oblique.

It is usual when making mechanical or working drawings to do the work in pencil first, and then ink in the necessary lines, or a piece of tracing cloth is placed over the pencil drawing and the lines which show through are then inked. The latter method is used in case a number of blue prints are desired for the shops or office.

In the pencil work, **accuracy** is necessary. Some beginners think they can correct inaccuracies in pencil by care in inking. The lines should be located exactly of the required length. A hard pencil; 4 H or 6 H is generally used. A hard pencil, when sharp, makes a depression in the paper which cannot be erased. For this reason press lightly on the pencil.

In inking, it is better to make circles, arcs of circles and curved lines *first*. It is much easier to make straight lines meet arcs, or to make them tangent to circles or arcs, than the reverse. To indicate an edge or intersection of two planes a full line ——— is used; edges or intersections which are concealed are represented by dotted lines . . . . Dot and dash lines, — . — . — . — . or — . . — . . — . . — . . indicate center lines or axes. A fine line or a series of long dashes —— —— —— —— is used for dimension lines. Titles, various views, sections, names of materials, etc., are **lettered** on the drawings. The styles and the care taken in this work varies with the draughtsman or with the amount of time at his disposal. In every case all lettering should be neatly done and of some clear cut simple form. Marks indicating in which shop the work is to be done and for classification are also placed on drawings. Fig. 5 shows a working drawing of a three inch pillow block which illustrates the above principles. The three views side, plan and end are half in section.

DESIGN.

Parts of machines are designed from rules derived from "Strength of Materials," other rules are based on the wear of the parts, while others depend on the size or thickness necessary for stiffness or a sound casting. In case theory doesn't accord with the practice of successful designers, it is safe to follow the latter.

Some parts of machines, bolts, nuts, screws, pipes, etc., can be obtained in standard sizes from various factories. The designer should know these standard sizes and make his details of such shapes that they will conform. The designer must also keep in mind the processes and tools to be used so that the construction will not be too difficult or expensive.

Usually dimensions are expressed in feet and inches and such fractions of inches as $\frac{1}{2}$, $\frac{1}{4}$, $\frac{1}{8}$, $\frac{1}{16}$, $\frac{1}{32}$. $\frac{1}{3}$, $\frac{1}{5}$, $\frac{1}{6}$ or $\frac{1}{10}$ are never used since the scales in shops are not divided in these fractions. Decimals are used only for great accuracy or in the design of gear teeth.

All machines are made up of different combinations of *simple* principles. The designer must know these principles and the relations they bear to one another. He must also have a thorough knowledge of the machine; the work it is to do; the character of power to be applied and in some cases the location and surroundings. A study of machines that have been designed to do similar work will be of great assistance. A wide knowledge of machines

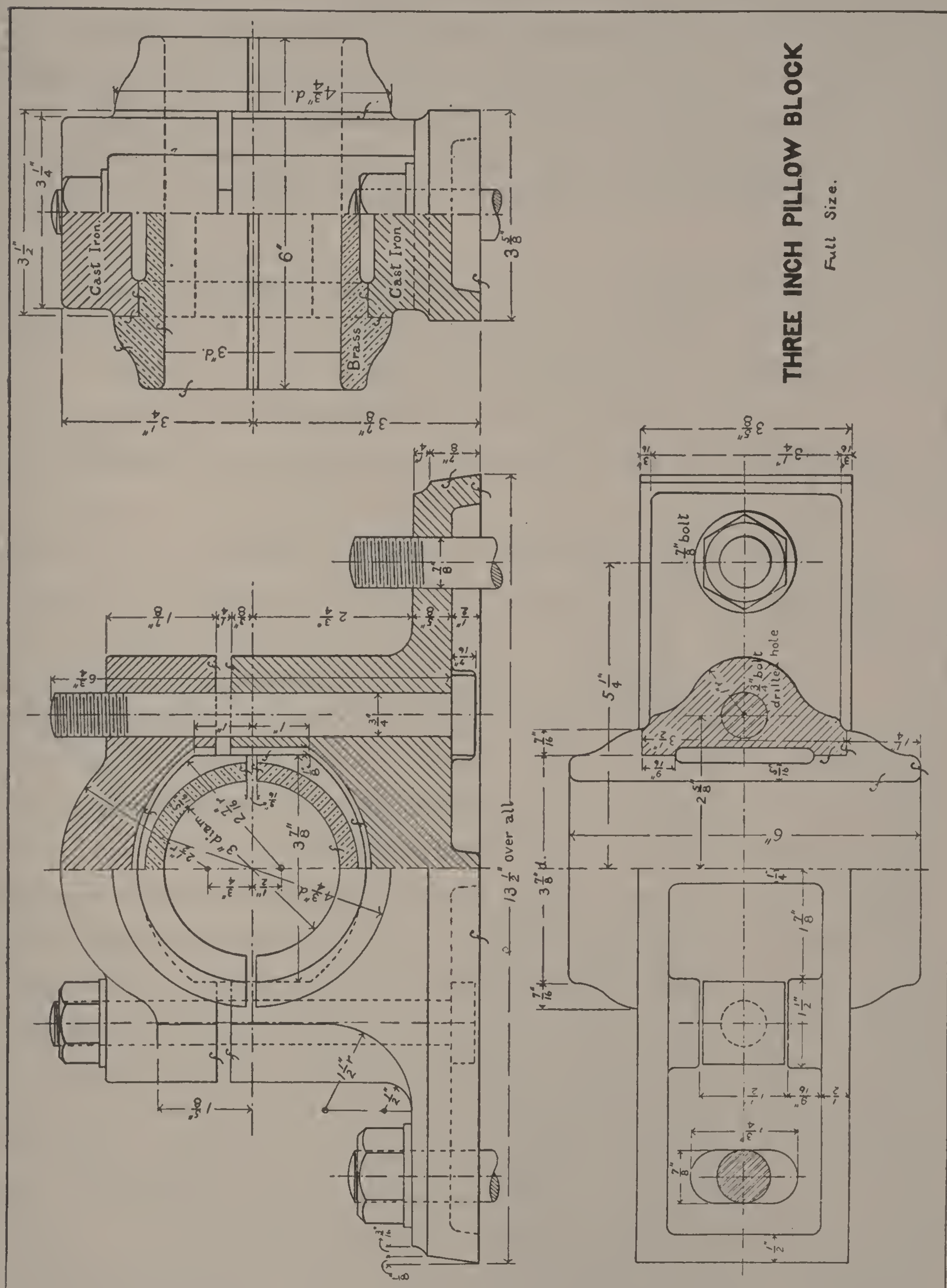


Fig. 5.

permits the designer to have many ideas of details for the new machine.

Often several complete drawings are necessary before the final result is attained. An idea that seems desirable may not prove so when the details are worked out. Sometimes the relative motions are all right, but when the parts are designed for strength, they are found to interfere with each other. The expense of construction or the difficulties of manufacture may render a good design impracticable.

All notes, calculations and sketches of details and combinations should be carefully preserved for reference. Ideas worthless for some particular machine may be found very valuable for another. It is well to keep sketches, calculations and memoranda in books rather than on loose sheets of paper that may become lost or misplaced.

The estimates for *weights* and *cost* of machinery are made from drawings. The volume is found by mensuration and when multiplied by the weight of a cubic unit of the material gives the weight. Considerable skill and experience is sometimes necessary to estimate the volumes of irregular shaped parts. If the weights are known the cost is estimated from market values. The *time* necessary for completion is also judged from the amount of work and the ease with which it can be accomplished.

At the beginning an inexperienced designer is usually taught by making drawings of details which have been designed by others. Often he is employed for some time simp'y tracing drawings. After a little he is given the principal dimensions of the simpler parts and instructed to make working drawings for the pattern, forge or machine shops. During this time he becomes familiar with methods adopted in shops, standard dimensions, allowable stresses for the various materials, methods of adjusting wear and lubrication, necessary dimensions for sound castings, etc.

The following are a few practical rules or suggestions that are unconsciously kept in mind by the successful engineer, but are often forgotten by the inexperienced.

Means for **adjusting** all parts subject to wear, should be provided.

Make the motion of all parts *positive* if possible ; avoid the use of springs and weights for producing motion.

Provide means for **lubrication** wherever necessary.

Construct the parts that may wear or break so that they will be **accessible** for adjustment or repairs.

Cranks, belts, levers and gear wheels are preferable to cams, screws and worm-wheels.

Avoid the use of **tap bolts** and **studs;** use through bolts or T head bolts if possible.

If convenient make the pressure per square inch on slides small.

FASTENINGS.

Bolts, Nuts, Keys, Cotters, etc.

A **screw** is a cylindrical bar, upon the surface of which a helical projection called the thread has been formed. A cylindrical helix is the curve generated by the revolution of a point about the surface of a cylinder, while moving along the axis at a constant rate.

A **nut** is a short hollow prism, upon the inside of which are formed grooves which correspond accurately to the threads of the screw.

The screw and nut when used for fastening is called a bolt. Screws are also used to transmit motion and to adjust the relative positions of two pieces. Screw threads are usually triangular or square in section. The Whitworth and U. S. standard threads are triangular. Square threads are used chiefly to transmit motion. There is less friction and less wear than with triangular threads, but they are more expensive.

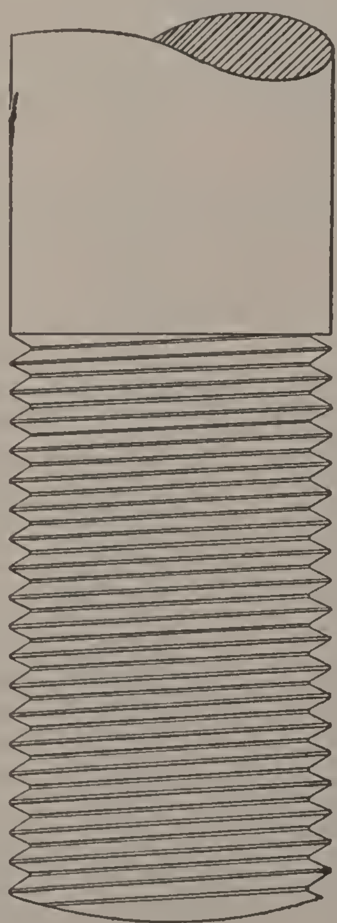


Fig. 6.

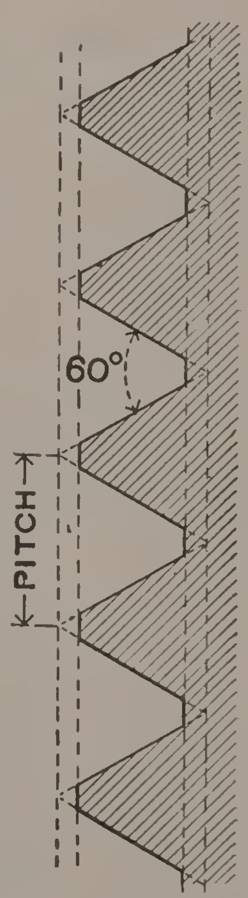


Fig 7.

The Standard Thread, Fig. 6, shows the American or Sellars triangular thread. The construction is shown enlarged by Fig. 7.

The sides of the thread are inclined at an angle of 60°. The section of the thread is an equilateral triangle having its top cut off so that the flat portion is $\frac{1}{8}$ of the pitch in width. The angles at the bottom are filled in similarly. The real depth of the thread is a little less than the altitude p of the triangle ; it is .65 p.

Sharp V threads are those cut without the flat top and bottom ; the section being an equilateral triangle.

The **pitch** of a screw is the distance the screw advances during one revolution ; it is the distance from one thread to the next. Another method of indicating the pitch is to give the **number of threads per inch.** For instance we say a screw has 11 threads per inch ; that is, it will advance one inch during 11 revolutions or the pitch is $\frac{1}{11}$ of an inch, or .091 inch.

The pitch of threads depends upon the diameter of the bolt. The following equations give approximately the pitch, and the diameter at the bottom of thread in the U. S. standard.

$$p = .24\sqrt{d + .625} - .175 \text{ inch},$$
$$d_1 = d - 1.3p = d - 2p_1$$

In these formulas p = pitch, d = diameter of bolt, d_1 = diameter at the bottom of the thread, and p_1 = real depth of the thread.

Let n = the number of threads per inch ; then

$$n = \frac{1}{p} \text{ and } d_1 = d - \frac{1.3}{n}$$

The external diameter of a bolt is $1\frac{7}{8}$ inches, to find the pitch, the number of threads per inch, the diameter at root of threads and the depth of thread we proceed as follows :

$$p = .24\sqrt{1.875 + .625} - .175 = .204 \text{ inch.}$$
$$n = \frac{1}{.204} = 5 \text{ (about) which makes the pitch .20.}$$
$$d_1 = d - 2p_1$$
$$2p_1 = d - d_1$$
$$p_1 = \frac{d - d_1}{2} = \frac{.26}{2} = .13$$

Then the pitch is $\frac{1}{5}$ or .2 inch : there are 5 threads per inch, diameter at root of threads is 1.615 inches and the depth of thread equals .13 inch.

The following table gives the principal dimensions of **U. S.** standard or Sellars threads.

Diameter of Bolt (Inches)	Number of Threads (Per Inch.)	Diameter at Bottom of Thread. (Inches.)	Area at Bottom of Thread (Square Inches.)
1/4	20	.185	.0269
5/16	18	.240	.0452
3/8	16	.294	.0679
7/16	14	.345	.0935
1/2	13	.400	.1257
9/16	12	.454	.1619
5/8	11	.507	.2019
3/4	10	.620	.3019
7/8	9	.731	.4197
1	8	.838	.5515
1 1/8	7	.939	.6925
1 1/4	7	1.064	.8892
1 3/8	6	1.158	1.0532
1 1/2	6	1.283	1.2928
1 5/8	5 1/2	1.389	1.5153
1 3/4	5	1.490	1.7437
1 7/8	5	1.615	2.0485
2	4 1/2	1.711	2.2993
2 1/4	4 1/2	1.961	3.0203
2 1/2	4	2.175	3.7154
2 3/4	4	2.425	4.6186
3	3 1/2	2.629	5.4284
3 1/4	3 1/2	2.879	6.5099
3 1/2	3 1/4	3.100	7.5477
3 3/4	3	3.317	8.6414
4	3	3.567	9.9930
4 1/4	2 7/8	3.798	11.3292
4 1/2	2 3/4	4.027	12.7366
4 3/4	2 5/8	4.255	14.2197
5	2 1/2	4.480	15.7633
5 1/4	2 1/2	4.730	17.5717
5 1/2	2 3/8	4.953	19.2676
5 3/4	2 3/8	5.203	21.2617
6	2 1/4	5.423	23.0978

The **Whitworth** triangular thread shown at D, Fig. **10**, is used in England. The angle between the surfaces is 55°, and $\frac{1}{6}$ of the depth or altitude of the triangle is rounded off instead of being flat as in the Sellars thread. The pitch and diameter at the bottom of the thread is found as follows:

$$p = .08d + .04,$$

$$d_1 = d - \frac{1.28}{n} = .9d - .05.$$

A screw with a **square** thread is shown in Fig. 8 and an enlarged section of the thread in Fig. 9. The pitch of the square thread is usually double that of the triangular or V thread for a bolt or screw of the same diameter. The pitch is about $\frac{1}{5}$ the

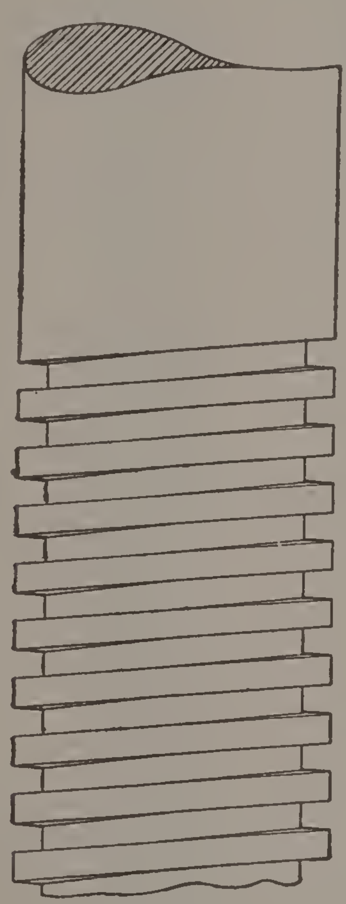


Fig. 8.

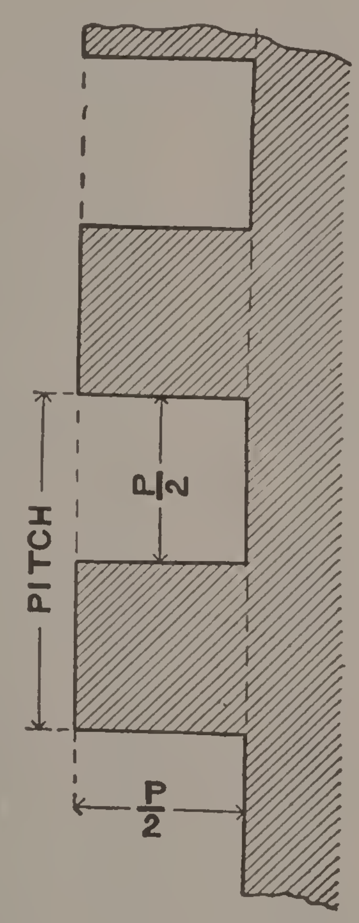


Fig. 9.

diameter of the screw; the diameter at the bottom of the thread is $\frac{4}{5}$ the diameter of the screw and the depth of thread $\frac{1}{2}$ the pitch or $\frac{1}{10}$ the diameter of the bolt. The edges of the square thread are slightly rounded to prevent flattening and to prevent binding of the nut.

If the thread is to be subjected to very rough usage the rounding is carried further so that the section of the thread is like that shown at A, Fig. 10. The thread shown at B is like a square thread, but instead of having a square section the thread **tapers** from the root to the point. This taper is given to the thread because it is easier to cut than a square thread. This form is often used as lead screws for lathes. The taper allows the nut, which is in two parts, to engage and disengage easily. The

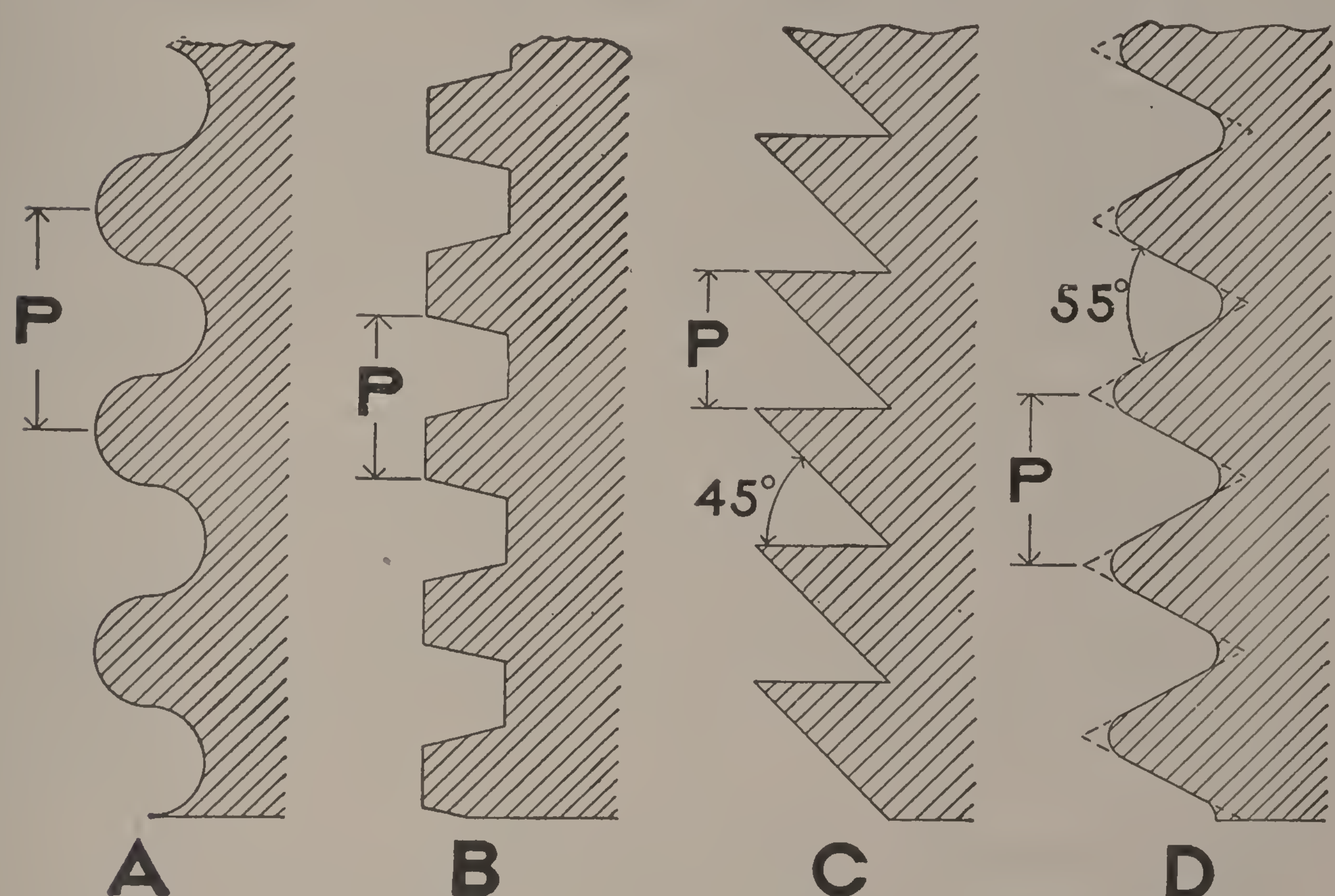


Fig. 10.

trapezoidal or buttress thread is shown at C. It is used for transmitting motion or when the force acts always in the same direction. One face is normal to the axis of the screw and the other is inclined at an angle of 45 degrees. The top is usually cut off and the bottom filled in as in the U. S. standard thread; the amount being about the same, that is, about $\frac{1}{8}$ the depth. The pitch is equal to the theoretical depth since the angle of the face is 45 degrees. The real depth is about $\frac{3}{4}$ the altitude of the triangle.

The relative advantages of the various forms of threads: and the uses to which they should be put may be understood by considering the forces which act on the faces.

It has been proved that the greater the angle of the screw thread the greater the friction between the bolt and nut, and the greater the force tending to burst the latter.

The friction of the square thread is less than that of the triangular thread because the angle between the sides is zero; there is moreover, no force tending to burst the nut. The triangular thread is, however, nearly twice as strong as the square thread. For the above reasons the square thread is better for transmitting motion and the triangular for fastening.

The trapezoidal thread should be used only when the pressure comes on the side perpendicular to the axis. In this case the thread has the same friction as the square thread and the same strength as the V thread. If the pressure is put on the inclined side the friction and bursting force are greater than is the case with the Sellars thread having angles of 60° between the sides.

Threads are formed for both *right handed* and *left handed* motion. Usually they are right handed. To determine the motion, hold the bolt or screw horizontal and turn it in the direction in which the hands of a watch revolve. If it advances into the nut or wood it is right handed. If when vertical the slope of the thread is from right to left it is right handed. For nuts reverse the above rule.

Multiple Threads. In tracing the thread about the screw, the next thread is reached in one revolution, if the screw is single threaded. In other words, the nut will advance a distance equal to the pitch for every revolution of the screw. If in tracing the thread through a turn one thread is missed, it is a double threaded screw; if two are missed it is a triple threaded screw, and so on.

Multiple threaded screws are used for transmitting motion, when it is desirable to have the nut advance a considerable distance for each revolution. This could also be accomplished by making the pitch large; but the multiple thread is better. The diameter at the root of the multiple thread is greater than that of a single thread and therefore stronger. The pitch is the distance the screw advances during one revolution, or it is the distance between two consecutive threads, if double threaded; or the distance between three threads, if triple.

Gas Pipe Threads. The Sellars thread is not suitable for the threads on gas pipe, for the calculated depth of thread would be greater than the thickness of the pipe. For this work a special system has been adopted having smaller pitch and cutting less deeply into the metal.

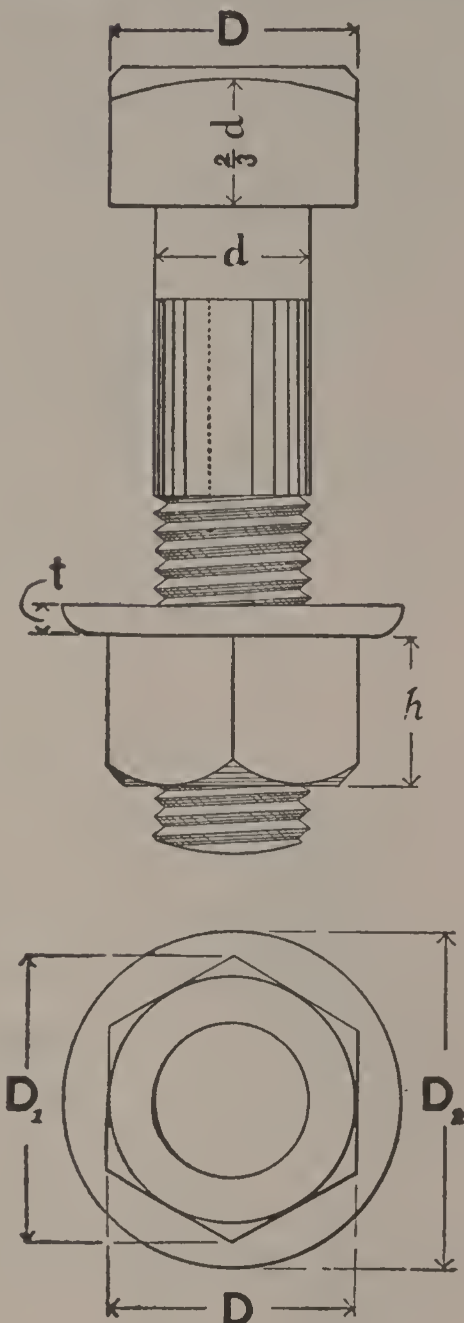


Fig. 11.

Proportions of Bolts and Nuts. The diameter of the bolt determines the dimensions of the nut. These dimensions may vary to suit circumstances. Sometimes in cramped places the nut must be made thin, or there must be little metal around the screw threads, or it must be made of peculiar shape. In altering the shape or size of a nut, the designer considers the strain put on it. The standard form is shown in Fig. 11. The head of the bolt is square. Sometimes the neck (the portion next the head) is made square also, to prevent rotation of the bolt when the nut is being screwed up.

The nut is hexagonal and the washer circular. The washer is used with rough castings to give a smooth surface on which to turn the nut. The following are the formulas for dimensions corresponding to the figure, d being the diameter of the bolt.

For Rough Work.

$D = 1\frac{1}{2}\,d + \frac{1}{8}''$

$D_1 = 1.73d + .14''$ for hexagonal

$D_1 = 2.12\,d + .18''$ for square

$D_2 = 1\frac{1}{8}\,D_1$

$h = d$

$t = .15\,d$

For Finished Work.

$D = 1\frac{1}{2}\,d + \frac{1}{16}''$

$D_1 = 1.73\,d + .07''$ for hexagonal

$D_1 = 2.12\,d + .09''$ for sq.

$D_2 = 1\frac{1}{8}\,D_1$

$h = d - \frac{1}{16}''$

$t = .15\,d$

STRENGTH OF SCREW BOLTS.

Bolts are generally used when the straining force is in the direction of the axis of the bolt; that is, bolts are used for *tension stresses.* It is evident that the effective area is not the area of the cross-section of the bolt, but the area at the root of the thread.

Let P = the total load on the bolt.

d_1 = diameter at root of thread.

a = area of cross-section at root of thread.

S_w = safe working stress in pounds per square inch.

Then for tension

$$P = a S^w = \frac{\pi d_1^2 S_w}{4}, \text{ from which } a = \frac{P}{S_w}.$$

$$\text{Then } d_1 = 2\sqrt{\frac{P}{\pi S_w}}.$$

The values of a are found directly from the preceding table. The value of P we can usually calculate from the machine. S_w varies with the material and the conditions of stress; if it is constant a good wrought iron bolt will stand 7,000 or 8,000 pounds per square inch. For variable stresses, S_w may be taken as about 5,000 or 6,000 pounds. Usually S_w is taken as 4,000 or 5,000 pounds. For bolts used in cylinder heads, S_w varies from 3,000 pounds per square inch for small to 6,000 pounds for large cylinders.

Suppose we wish to find the diameter of a bolt to sustain a steady stress of 15,000 pounds, allowing 8,000 pounds as the working stress.

$$a = \frac{P}{S_w} = \frac{15,000}{8,000} = 1.875 \text{ square inches.}$$

The number 1.875 lies between 1.7437 and 2.0485 of the table. The larger value should be chosen, the diameter of the bolt being $1\frac{7}{8}$ inches.

EXAMPLES FOR PRACTICE.

1. What is the safe working stress on a bolt 1 inch in diameter, if the value of P is 3,850 pounds?

Ans. 7,000 pounds (about).

2. Find the size of bolt used for varying stress. P = 14,000 pounds and S_w = 4,000 pounds. Ans. $2\frac{1}{2}$ inch bolt.

3. An engine cylinder head is bolted to the cylinder by 12 bolts. If the total steam pressure is 48,000 pounds, what is the diameter of the bolts? S_w being 4,250 pounds. Ans. $1\frac{3}{8}$ inches.

It would take considerable time to make the threads of all the screws and bolts of working drawings accurately. To save time a **conventional form**, shown in Fig. 12, has been adopted. The threads are represented by alternately light and heavy lines. The distance between these lines need not be equal to the pitch of the threads, because, the diameter being given, the number of threads per inch is found from the table. If the threads are not standard the number of threads per inch is noted on the drawing.

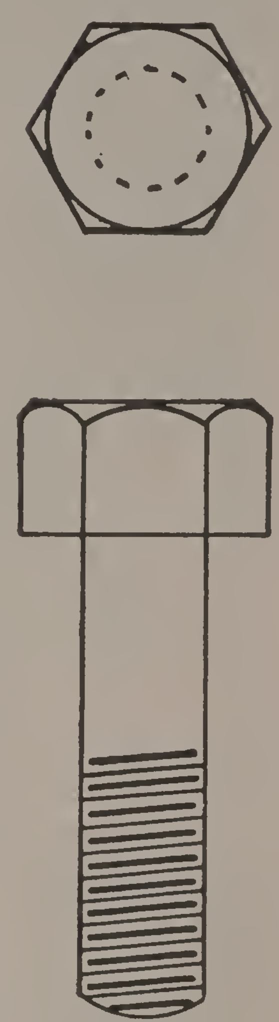


Fig. 12.

WRENCHES.

Forms of solid wrenches or spanners are shown in Fig. 13. The dimensions are given in decimals of the diameter of the bolt. They

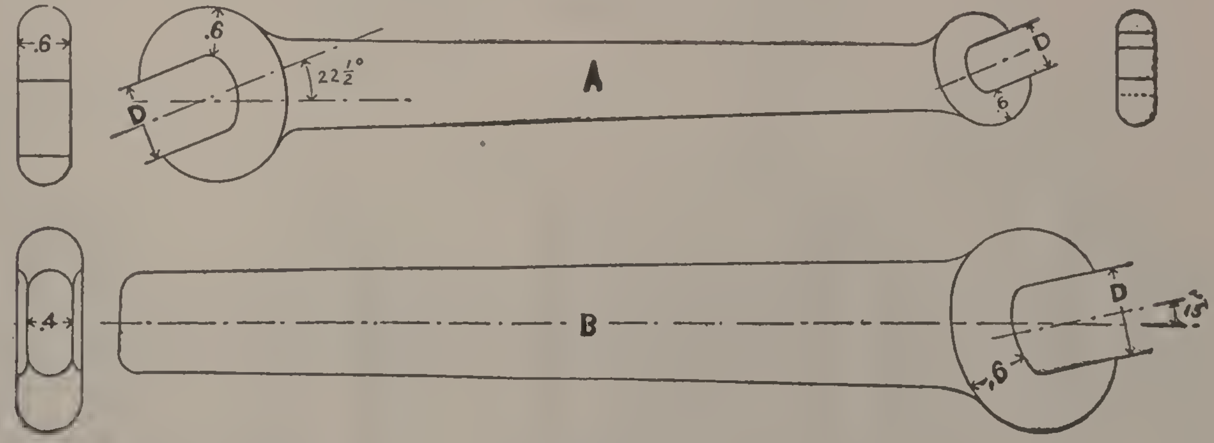


Fig. 13.

are made in many sizes, with ends shaped for hexagonal and square nuts. The unit for the proportions is D.

FORMS OF NUTS.

The most common form of nut is the hexagonal shown at A, Fig. 14; B shows the square nut. Usually both square and hexagonal nuts are chamfered off at an angle of 30 to 45.° Sometimes they are finished with a spherical bevel, having a radius of about twice the diameter of the bolt. C shows a round nut having holes in the sides into which a bar is inserted for tightening

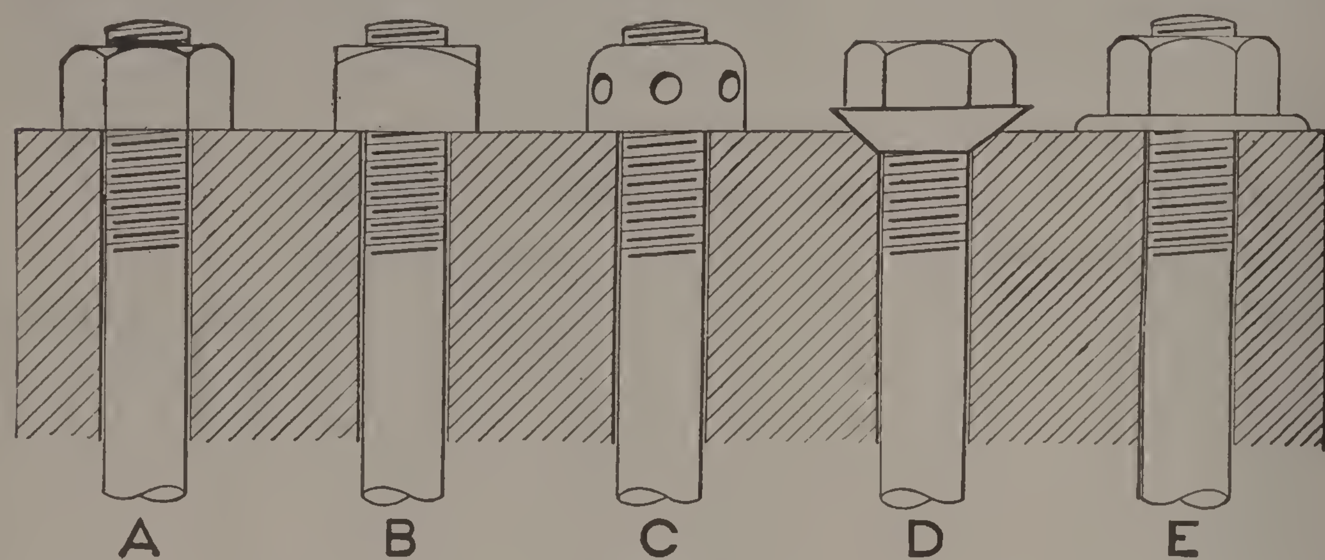


Fig. 14.

the nut. The nut shown at D is called a cap nut: it is used to prevent leakage past the screw thread. A thin copper washer is sometimes used with this form of nut. E represents a flange nut which is used when the hole in which the bolt is placed is considerably larger than the bolt. The flange covers the hole and gives greater bearing surface.

FORMS OF BOLT HEADS.

Fig. 15 shows several forms of bolt heads. The hemispherical or cup-shaped head, a common form, is shown at A. At B is

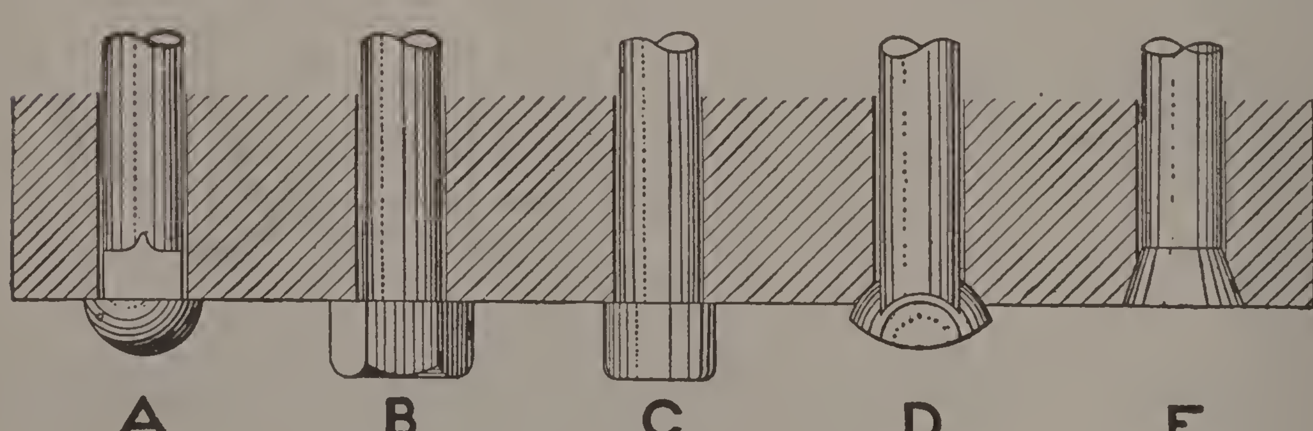


Fig. 15.

shown the hexagonal form. It is similar to the hexagonal nut and has about the same dimensions except the height which is usually less; it is from $\frac{2}{3}$ d to d. The cylindrical bolt head is shown at C; rotation of the bolt is prevented by the square neck. D shows the spherical head; the bearing surface rests on a seat of the same shape. It is used when the bolt tends to lean toward one side. The head remains in contact with the seat for every position of the bolt. E shows a bolt with countersunk head; rotation is often prevented by a set-screw.

Fig. 16 is the **hook** bolt. It is used when a piece would be

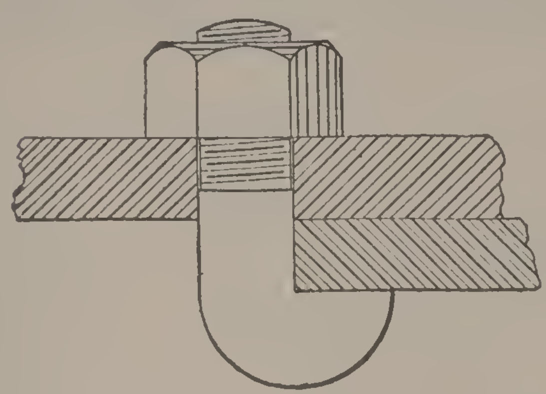


Fig. 16.

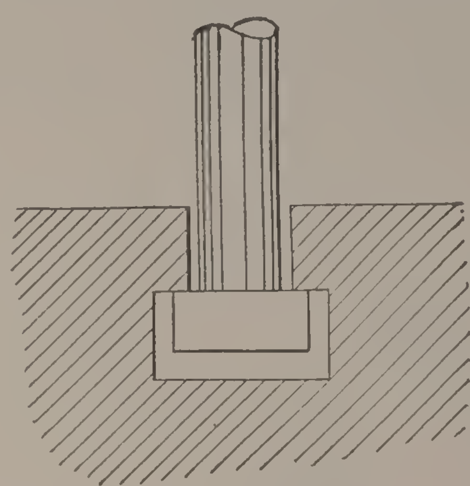


Fig. 17.

weakened or is too small to have a drilled hole. Fig. 17 is a **T-headed** bolt.

Set-screws are screws or bolts used to prevent by friction relative rotation between pieces. Set-screws are often used to prevent the hub of a pulley from turning on the shaft. They

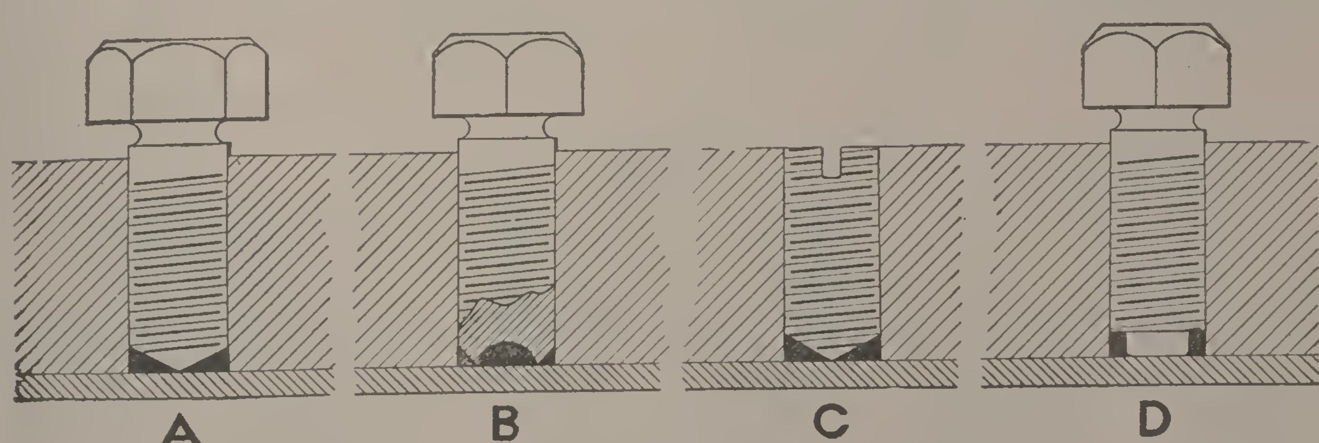


Fig. 18.

are screwed through the hub and prevent rotation by pressing against the shaft. At A, Fig. 18, is shown the cone point set-screw. The one shown at B is called the cupped set-screw; C, the headless cone point set-screw; and D, the round point set-screw.

A **stud bolt** is one that has threads cut on both ends. One end is screwed into one of the pieces to be connected and remains in position when the nut is on or off. Fig. 19 shows a stud bolt. They are sometimes used for cylinder-heads and valve-chest covers. A stud with a **collar** is shown in Fig. 20. The collar may be square or round; if square it is a convenient place for a wrench. The collar forms a shoulder against which the stud may be screwed.

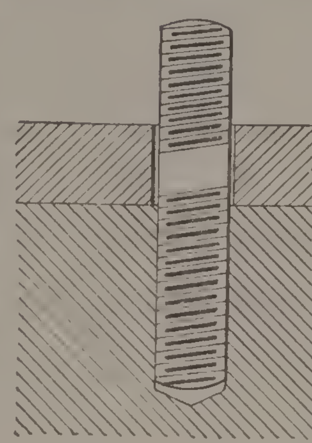


Fig. 19.

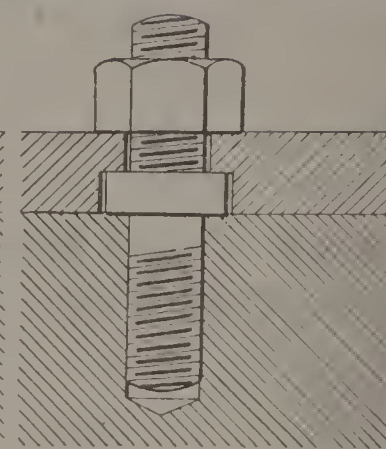


Fig. 20.

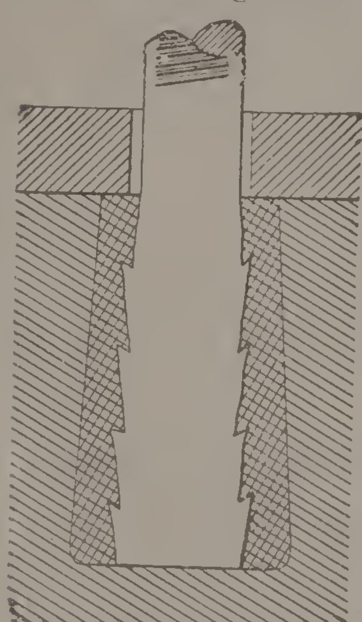


Fig. 21.

Fig. 21 shows an ordinary method of fastening a bolt to **stonework.** The head, which is long, is made jagged with a cold chisel. The hole is made larger at the bottom than at the top and after the head is placed, the space around it is filled with melted lead or sulphur.

Foundation bolts, which are used to fasten an engine-bed to its foundation, are often fixed as shown in Fig. 22. The head is formed by a cast iron washer and a cotter. This cotter passes through a slot and has gib ends to prevent slipping. The washer provides a large bearing surface. The bolt, washer, and cotter are placed in a recess in the wall and are accessible. The size of the washer is easily determined. The area of the washer multiplied by the compressive or crushing strength of the stone or brick should be equal to the pull on the bolt. The tensile strength of wrought iron is about 55,000 pounds per square inch and the compressive strength of

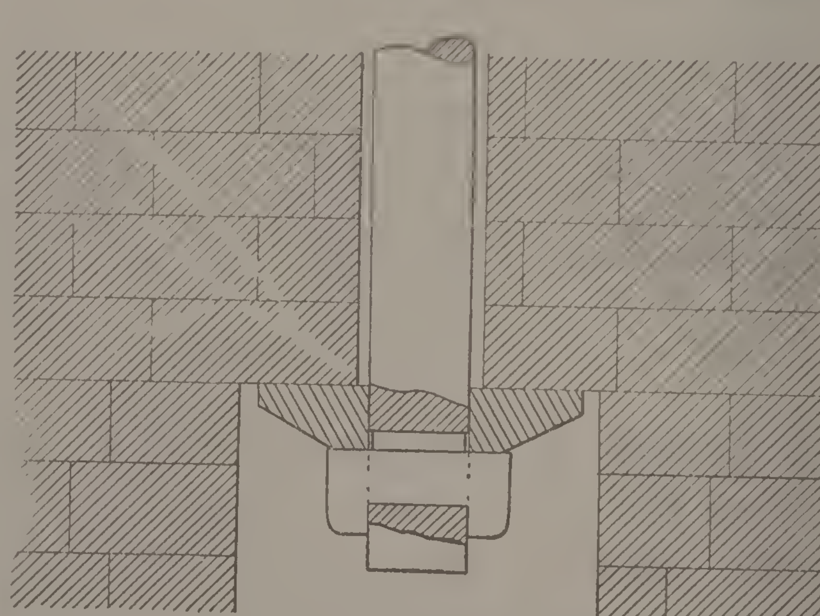


Fig. 22.

brick is about 2,500 pounds per square inch, or about $\frac{1}{22}$ of that of the bolt. Therefore, the bearing surface of the washer should be about 22 times the area of the cross-section of the bolt.

Screws. The three most common forms of machine screws

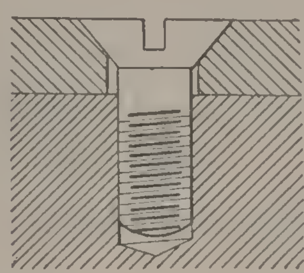


Fig. 23.

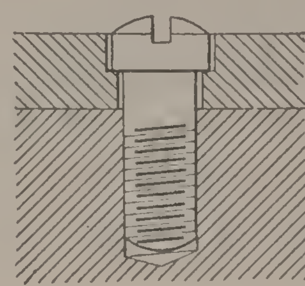


Fig. 24.

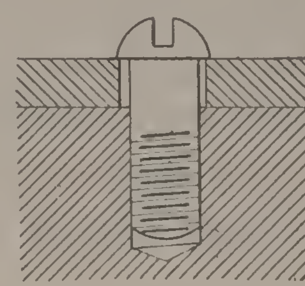


Fig. 25.

are shown in Figs. 23, 24, and 25. Fig. 23 is the **countersunk** head screw; Fig. 24, the **fillister head** and Fig. 25 the **button head.** The countersunk head is used when the head is not to project above the plate.

LOCKING ARRANGEMENTS FOR NUTS.

Nuts never fit the bolt accurately; some clearance is necessary to permit them to turn freely. If a nut is subject to frequent changes of load and vibration it gradually unscrews or slacks back. To prevent nuts from becoming loose various locking devices are used.

One of the most common is the double nut called a **locknut**

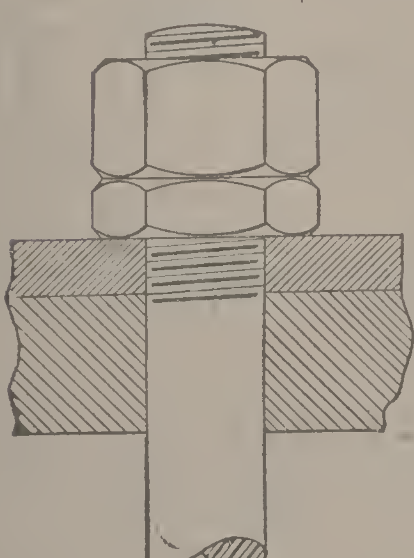


Fig. 26.

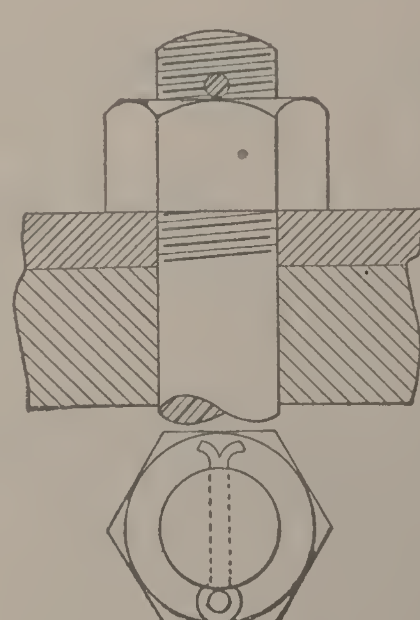


Fig. 27.

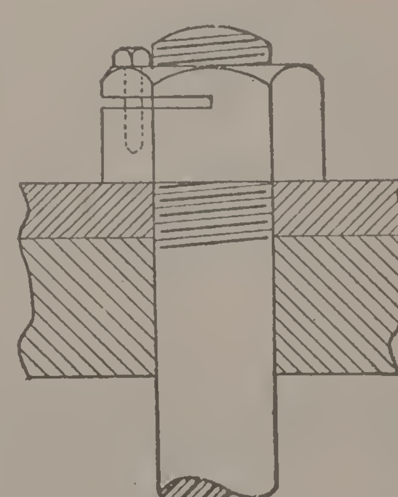


Fig. 28.

or jam nut shown in Fig. 26. There are two nuts, one about twice as thick as the other. The outer nut should be the thicker because the load is thrown on it. However, the thin one is often placed on the outside because the wrench is often too thick to act on it when it is inside. When the nuts are screwed home they are locked together by being turned in opposite directions.

In Fig. 27 the nut is kept in place by a **split pin** or cotter. A hole is drilled through the bolt and the pin driven through and the ends turned over to prevent it from backing out. This is not a very good method because the nut must always be close to the pin.

Another method is shown in Fig. 28. The nut is sawed about half way through and the parts closed slightly by a set-screw, after the nut is screwed home. The nut then grips the thread tightly. For small nuts the set-screw is not used but the parts are closed. just before it is set, by a slight blow of a hammer.

At A Fig. 29 is shown a **Grover's spring washer**. The upper

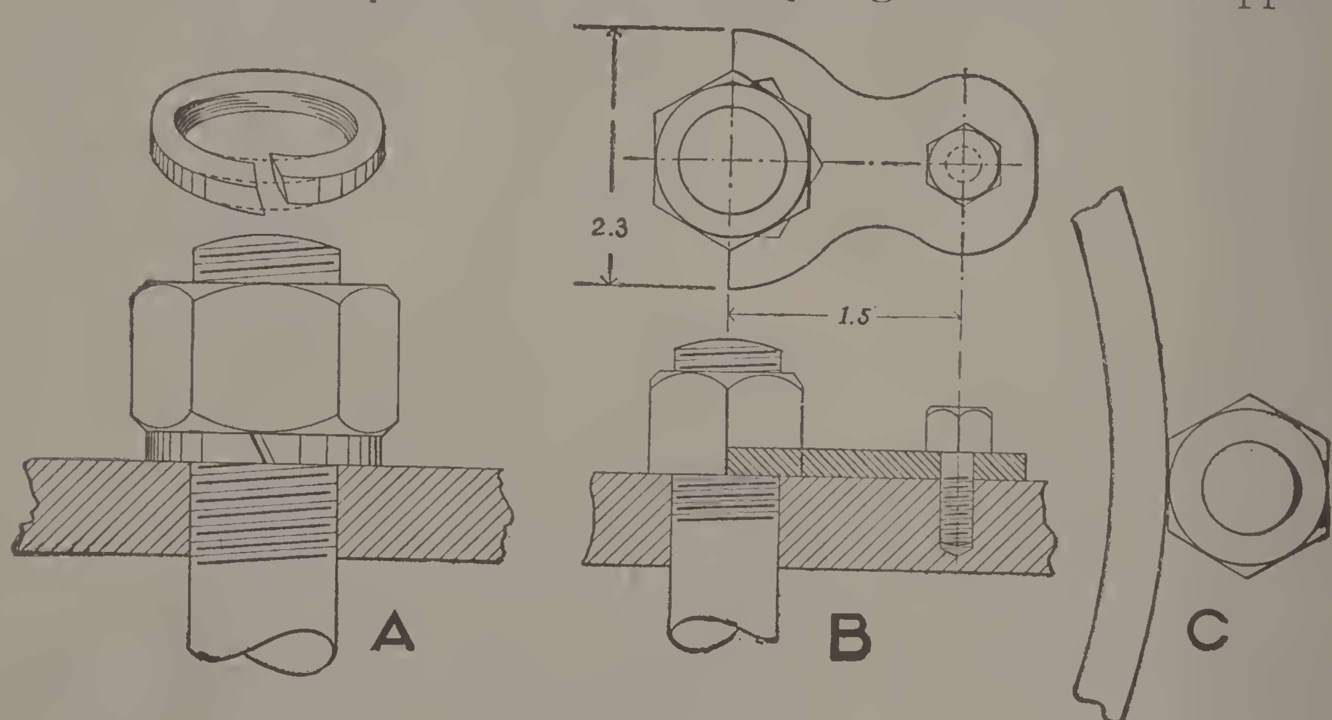


Fig. 29.

portion of the figure shows the washer when not held down by the nut. When the nut is screwed down tightly, the washer becomes nearly flat and its elasticity increases the friction between the threads of the bolt and nut.

In the device shown at B, Fig, 29, a **stop plate** is used. It is fixed by a set-screw to one of the pieces through which the bolt passes. It is shaped so that the nut may be locked at intervals of

$\frac{1}{12}$ of a revolution. The set screw may have a diameter equal to

$$\frac{d}{4} + \frac{1}{8}'',$$

d being the diameter of the bolt. The other dimensions are also in terms of the diameter. For bolts set in a circle, the bolts of a cylinder head for example, a **circular stop plate**, shown at C, Fig. 29, is used. It is placed inside the nuts and bears against one of the parallel sides.

There are numerous methods of locking by set-screws, their forms depending on the position of the pieces.

Usually bolts are used in tension, that is, when the straining force is parallel to the axis of the bolt. The joint pin is a

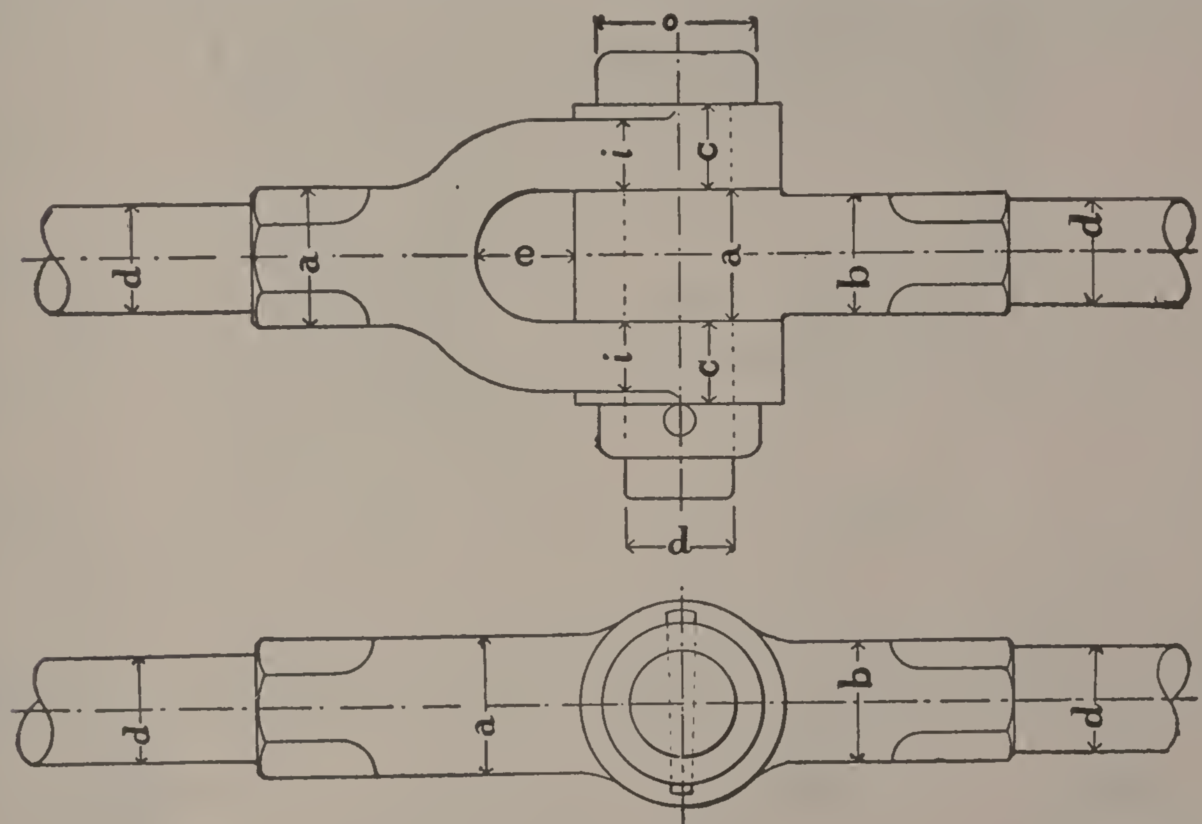


Fig. 30.

bolt placed so as to be in **shear.** Fig. 30 shows a **knuckle joint.** The joint pin is made the same size as the rods because of wear. If the pin were subjected to simple shear at the two sections it need be only about .7 the diameter of the rod, but it soon wears and is subjected to bending stresses also.

The other proportions are in terms of the diameter of the rods.

d = diameter of the rods.
$a = 1.2\ d$
$b = 1.1\ d$
$c = .75\ d$
$e = .8\ d$
$i = .6\ d$
$o = 1.5\ d$

Keys are small wedges, usually made of iron or steel, used to fix wheels, cranks and pulleys to shafts. It is the duty of keys to prevent the wheel or crank from rotating otherwise than with the shaft on which it is keyed. For example, a crank is keyed to the shaft in order that the shaft and crank will rotate together. Usually the friction of the key in the keyway will also prevent the wheel from sliding along the shaft.

Keys are usually plain rectangular pieces with their lateral sides parallel and a tapering thickness. In case the small end is

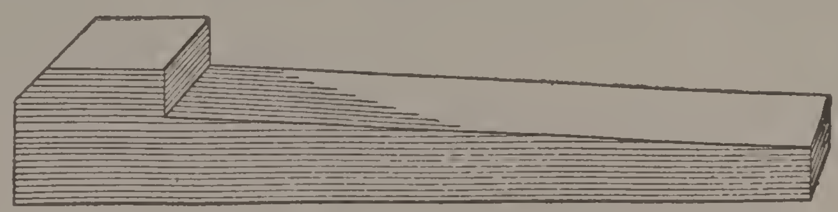


Fig. 31.

inaccessible, that is, the arrangement is such that it cannot be driven out, the key is made in the form shown in Fig. 31. A gib head is made at the large end which forms a shoulder to drive against.

Fig. 32 shows several forms of keys. The concave, or **saddle** key, is shown at A. The slot or keyway is cut in the wheel and the key hollowed to fit the shaft which is not cut at all. As the key holds by friction, it is suitable only for light work. The **flat**

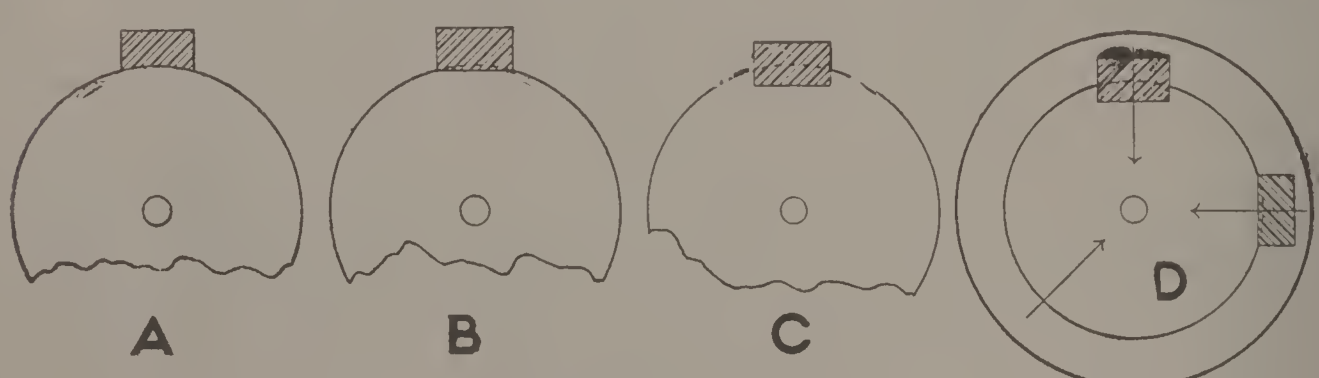


Fig. 32.

key is shown at B. A flat surface is planed on the shaft having a breadth equal to that of the key. This form is more secure than the saddle-key. The **sunk** key shown at C is more effective than either of the above forms because slipping is prevented unless the key shears. There are two forms of sunk keys, the rectangular and the square. The rectangular form is used for fastening cranks, gear wheels, pulleys, etc. They are driven in tightly and fit both at the top and bottom as well as the sides. As they fit on all sides they should not be used in accurate work because they are likely to spring the work out of true. Square keys are used for accurate work; they fit accurately only on the sides. In case a pulley is accidentally bored a little too large for the shaft it is fastened securely by using both a flat key and a sunk key as shown at D. The two keys are placed at about right angles. The pulley and the shaft have a bearing at three points on the circumference of the shaft.

Sometimes large wheels are keyed to the shaft by two, three or four keys. When the shaft is square eight keys may be used.

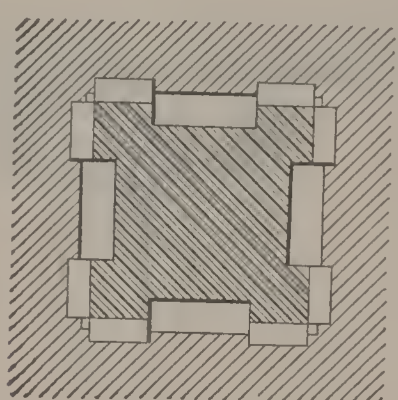


Fig. 33.

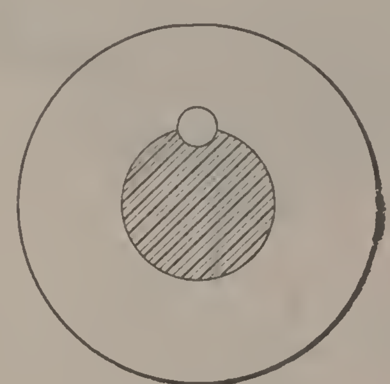


Fig. 34.

If they are arranged as shown in Fig. 33 no keyways need be cut on the shaft, and only the key seats are planed.

Pin-keys. A round taper pin is used in place of a key for small shafts; handles on valve stems for instance. The pin is sunk half in the shaft and half in the piece, as shown in Fig. 34.

Sometimes a secure connection, where the parts are not to be separated, is desired, as is the case with small cranks. The crank is bored slightly smaller than the crank shaft, then expanded by heat and shrunk on. It is further secured by a key or pin.

In order that keys may be easily driven in and removed they are tapered slightly. The taper is about 1 in 64 to 1 in 150. The more accurate the work the less the taper. This method of expressing taper means that the decrease in thickness is $\frac{1}{64}$ to $\frac{1}{150}$ of the length. Sometimes taper is expressed in inches to the foot as $\frac{1}{8}$ inch per foot.

Sliding Keys. Sliding or feather keys are used if the piece is to be prevented from rotating but at the same time is to be allowed to slide along the shaft. The key may be fastened to the piece and free to slide in the keyway of the shaft, or it may be fast to the shaft and the wheel free to slide. Figs. 35 and 36 show the various methods of fixing the key. In Fig. 36 the key

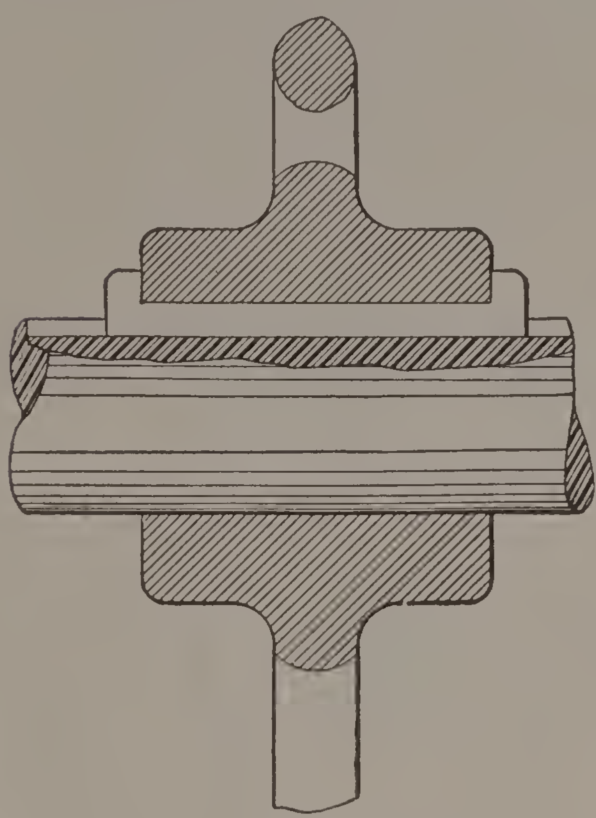


Fig. 35.

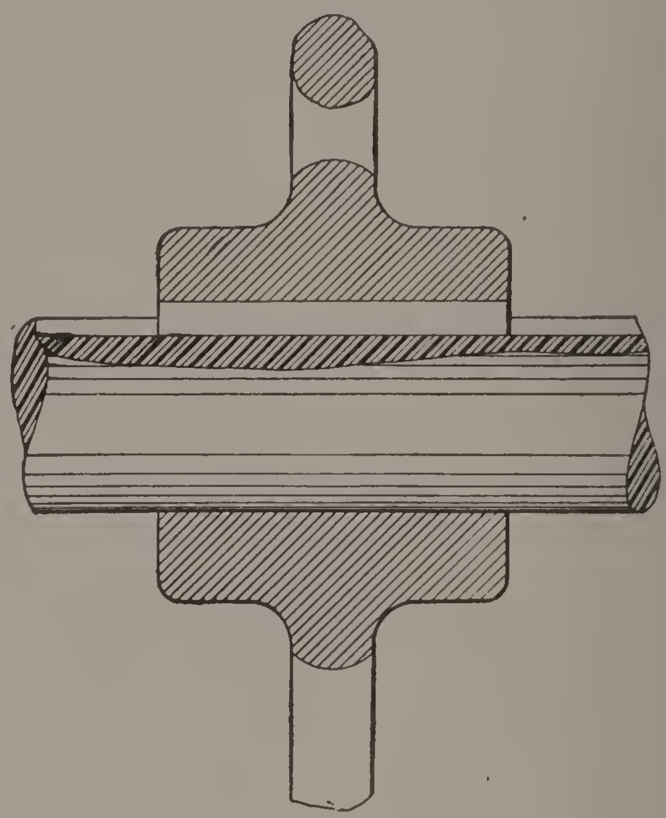


Fig. 36.

is dove-tailed in section and is fastened to the hub. The one shown in Fig. 36 is used when the hub comes against a collar or bearing, since it does not project from the hub. The feather key of Fig. 35 has gib heads.

Strength of Keys. Saddle keys are used most where the stress between the pieces is usually small. No exact rules can be given as they depend on friction to prevent rotation.

Sunk keys must resist both shear and compression. The twisting of the shaft tends to shear the key and crush it.

Let b = width of key in inches.
l = length of key in inches.
t = thickness of key in inches.
S_s = allowable shearing stress in pounds per square inch.
S_c = allowable crushing stress in pounds per square inch.
d = diameter of the shaft in inches.
r = radius of wheel or pulley in inches.
F = force in pounds acting at the rim.

The resistance to shearing is the shearing area multiplied by the shearing stress, blS_s. To find the diameter of the key, take moments about the center of the shaft and solve for bl

$$\tfrac{1}{2}\, d \times blS_s = F\, r, \text{ from which } bl = \frac{2\, F\, r}{d\, S_s}.$$

The sunk key usually has $\frac{1}{2}$ the thickness in the shaft and $\frac{1}{2}$ in the hub. In case the key is designed to be equally strong to resist shearing and crushing the shearing strength must be equal to the crushing strength.

The resistance to shearing is blS_s; and the resistance to crushing is equal to the product of the bearing surface and the stress, or $\frac{1}{2}\, tlS_c$.

If the crushing stress is assumed to be twice the shearing stress, $S_c = 2S_s$, then $b = t$ and the key is square in section. Usually the key is made wider than the thickness so that the shearing strength shall be greater than the resistance to compression, as there is little danger of crushing.

Let H. P. = the horse-power transmitted.
N = the number of revolutions per minute.
r = the radius of wheel in inches.

Then as the circumference of a circle is $2\,\pi\, r$, a point on the circumference will move $2\,\pi\, r\, N$ inches or $\frac{2\,\pi\, r\, N}{12}$ feet per minute.

The power transmitted (assumed to be constant) will be the force acting, F. multiplied by the distance $= \frac{2\,\pi\, rN}{12}$ or,

$$F \times \frac{2\,\pi\, r\, N}{12} = \text{foot-pounds.}$$

One H. P. = 33,000 foot-pounds per minute.

$$\text{Then H. P.} = \frac{2\,\pi\, rNF}{12} \div 33{,}000 = \frac{\pi\, rN\, F}{198{,}000}$$

$$\pi\, r\, NF = 198{,}000 \times \text{H. P.}$$

$$\text{or,} \quad Fr = \frac{198{,}000 \text{ H. P.}}{\pi\, N}$$

$$Fr = 63{,}025 \;\frac{\text{H. P.}}{N}$$

For shearing of keys the usual factor of safety is about 10 which gives an allowable stress of about 5,000 pounds per square inch for wrought iron and about 7,000 pounds per square inch for steel. Inserting these values in the equation $bl = \frac{2Fr}{dS_s}$,

$$bl = \frac{Fr}{d \times 2{,}500}, \text{ for wrought iron}$$

$$= \frac{Fr}{d \times 3{,}500}, \text{ for steel.}$$

$$\text{Then } bl = \frac{25.2 \text{ H. P.}}{d \times N}, \text{ for wrought iron}$$

$$= \frac{18 \text{ H. P.}}{d \times N}, \text{ for steel.}$$

Suppose the pressure on a crank-pin is 15,000 pounds and the crank is 12 inches long. If the shaft is 6 inches in diameter and the length of key 6 inches, what should be the dimensions of the wrought iron key?

$$bl = \frac{Fr}{2{,}500d}$$

$$b = \frac{Fr}{2{,}500 \times l \times d} = \frac{15{,}000 \times 12}{2{,}500 \times 6 \times 6} = 2 \text{ inches}$$

Then the key should be 2 inches broad. Its thickness is usually some fraction of the breadth. We will make it $\frac{3}{4}$, or $1\frac{1}{2}$ inches. Keys vary in thickness from $\frac{1}{2}$ to $\frac{5}{6}$ of the breadth.

In designing machinery, the dimensions are usually determined by empirical formulas. For the ordinary sunk key $b = \frac{1}{4}d + \frac{1}{8}''$, and the mean thickness, $t = \frac{1}{2}b$, to $\frac{5}{6}b$.

If we use these formulas for finding the dimensions of the key in the above example, $b = \frac{1}{4}d + \frac{1}{8} = \frac{1}{4} \times 6 + \frac{1}{8} = 1\frac{5}{8}$ inches. $t = \frac{3}{4}b = \frac{39}{32} = 1\frac{7}{32}$ inches.

When pulleys are keyed to large shafts which transmit only a small amount of power, the dimensions obtained from the above formulas are larger than necessary. In such cases the following formulas are used.

$$d = \sqrt[3]{\frac{100 \text{ H. P.}}{N}} \quad \text{or,} \quad d = \sqrt[3]{\frac{Fr}{360}}.$$

These values of d are inserted in the above formulas.

A pulley transmits 3 horse-power. It is keyed to a shaft 6 inches in diameter, which makes 130 revolutions per minute; find the dimensions of the key.

$$d = \sqrt[3]{\frac{100\text{ H. P.}}{\text{N}}} = \sqrt[3]{\frac{300}{130}} = 1.32 \text{ inches.}$$

$$b = \tfrac{1}{4}d + \tfrac{1}{8}'' = \frac{1.32}{4} + \tfrac{1}{8}'' = .455 \text{ or } \tfrac{1}{2} \text{ inch (about).}$$

$$t = \tfrac{1}{2}\,b = \tfrac{1}{4} \text{ inch.}$$

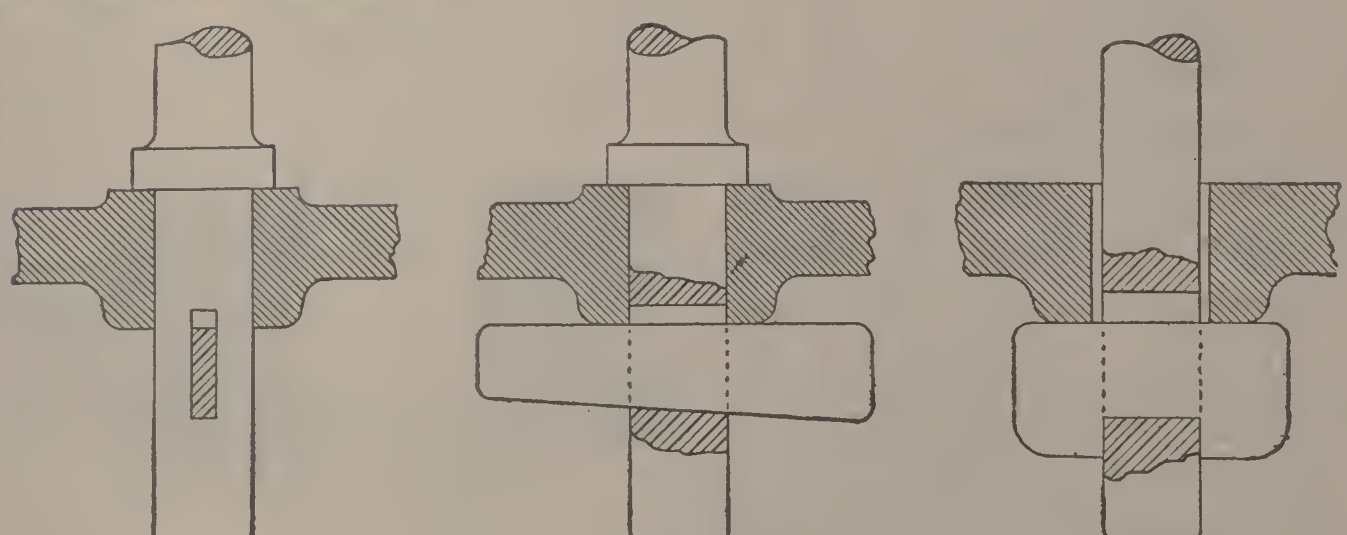
EXAMPLES FOR PRACTICE.

1. A shaft is 4 inches in diameter; what is the breadth of the sunk key? Ans. $1\frac{1}{8}$ inches.

2. Find the breadth of a steel key, when the length of the key is 4 inches, the horse-power transmitted is 100, the shaft is $4\frac{1}{2}$ inches in diameter and makes 100 revolutions. Ans. 1 inch.

COTTERS.

A **cotter** is an iron or steel bar driven through one or both of two pieces to be connected. It prevents their separation by its resistance to shearing at two transverse cross-sections. Cotters sometimes adjust the length of the pieces connected. They should be so designed as to decrease as little as possible the strength of the connected pieces.



A B C

Fig. 37.

A and B of Fig. 37 show two views of a simple cotter. In this form the cotter passes through the rod. The cotter resists tension. The collar or enlargement prevents movement in the other direction, and therefore resists thrust. An arrangement designed to resist tension only is shown at C, Fig. 37. The cotter has **gib ends** to prevent its moving out of place.

A construction to resist tension alone is shown at D and E of Fig. 38. This cotter is divided; one part with hooked ends is called

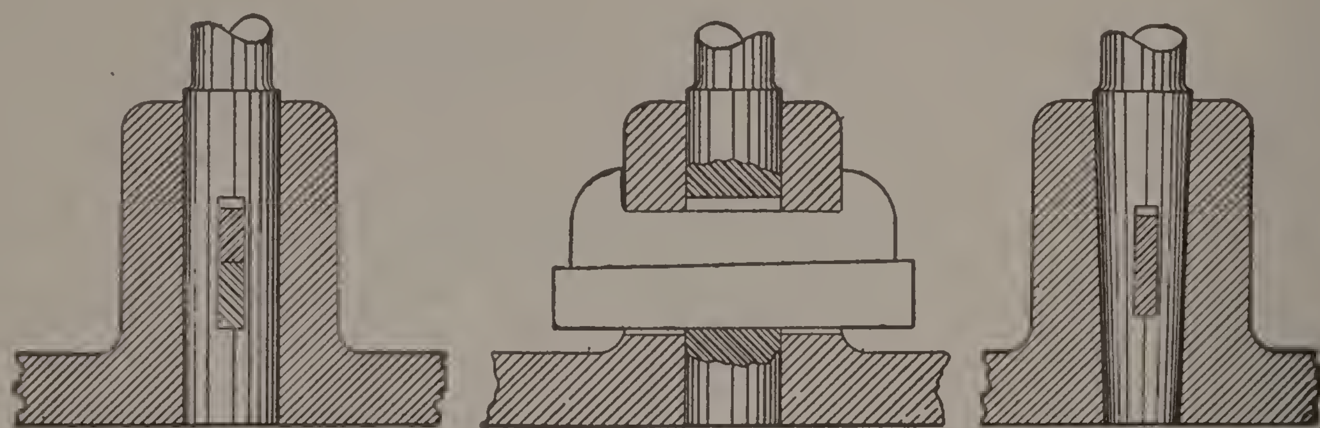


Fig. 38.

the gib and the other a plain cotter. Such a construction is often called the gib and cotter. At F of the same figure, the rod is tapered to provide for thrust.

Cotters like those shown at D and E in Fig. 38 are long and tapered and therefore may be used to adjust the length of the connected pieces. By driving the cotter in, the total length of the pieces is made less.

A cotter is often used to connect two straps, *a* and *b*, to the rod *l*, as shown in Fig. 39. If a plain cotter is used the excessive friction between the cotter and the straps, when the former is driven down, causes the lower strap to open as shown by the dotted lines. To prevent this a gib and cotter are used, or two gibs and a cotter. The gib prevents the strap from spreading. In Fig. 40 the side, *a b*, of the gib and the side, *c d*, of the cotter are parallel to each other and are at right angles to the straps. The parts of the gib and cotter that are in contact are tapered.

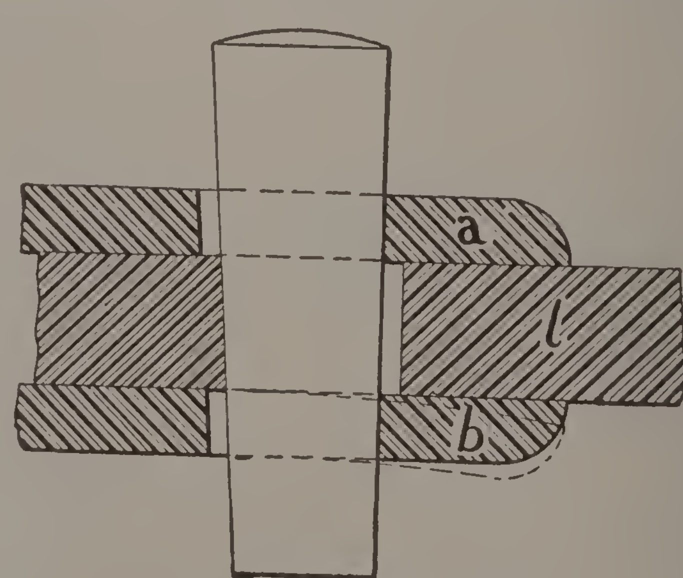


Fig. 39.

Taper of Cotters. A cotter that has a taper of more than 1 in 7 is likely to slack back. In general, the taper is from 1 in 24 to 1 in 48. If the cotter has a fastening device it may have a much greater taper, *i. e.*, 1 in 6 or 1 in 8. The taper of the cotter is found by dividing the increase in width by the length. Thus if a cotter is $2\frac{1}{4}$ inches in width at one end and $2\frac{3}{4}$ at the other the increase is $\frac{1}{2}$ inch and if the cotter is 10 inches long the taper is $\frac{1}{2}$ divided by $10 = \frac{1}{20}$ or 1 in 20.

Strength and Proportions of Cotters. In designing cotters a few facts must be remembered.

In the following demonstration the letters refer to Fig. 42.

The **cross-section,** $b\ t$, must be sufficient to stand the shearing stress.

The **thickness,** t, must be large enough to prevent crushing.

The **diameters** should be so designed that the rod will not be weakened by the cutting of the slot for the cotter.

Let F = the force in pounds on the rod,

S_t = allowable tensile stress of the rod in pounds per square inch,

S_c = allowable compressive stress of the cotter or rod in pounds per square inch,

S_s = allowable shearing stress of the cotter in pounds per square inch,

The net area of the rod is the area of a cross-section minus the area of the slot or,

$$\text{net area} = \frac{\pi\, d^2}{4} - d\,t\,.$$

The shearing area of the cotter is $2\ b\ t$. The area subject to crushing is $d\ t$. The area of the socket subject to tension is $\frac{\pi}{4}\left(D^2 - d^2\right) - (D - d)\ t$. The area of the rod itself is $\frac{\pi\, d_1{}^2}{4}$.

Then, as each of these values when multiplied by its respective allowable stress must be equal to the force, **F**,

$$F = \left\{ \frac{\pi d^2}{4} - d\, t \right\} S_t, \qquad \text{(a)}$$

$$F = 2\, b\, t\, S_s, \qquad \text{(b)}$$

$$F = d\, t\, S_c, \qquad \text{(c)}$$

$$F = \left\{ \frac{\pi}{4} (D^2 - d^2) - (D - d)\, t \right\} S_t, \qquad \text{(d)}$$

$$F = \frac{\pi}{4} d_1{}^2\, S_t, \qquad \text{(e)}$$

If the cotter is subjected to a force in one direction only, the allowable stresses may be,

Wrought Iron	$S_t = 10{,}000$	$S_s = 8{,}000$	$S_c = 20{,}000$
Cast Iron	$S_t = 2{,}800$		$S_c = 5{,}600$
Steel	$S_t = 13{,}200$	$S_s = 10{,}600$	$S_c = 26{,}400$

In case the forces act alternately in opposite directions the stresses are found by dividing the above values by 2.

The above values show that,

$$\frac{S_s}{S_t} = \tfrac{4}{5} \text{ and } \frac{S_c}{S_t} = 2, \qquad \text{then} \qquad \frac{S_c}{S_s} = \tfrac{5}{2} = 2\tfrac{1}{2}.$$

Then combining equations (b) and (c) and letting $t = \frac{1}{4} d$ we have,

$$2\, b\, t\, S_s = d\, t\, S_c$$

$$\text{or} \quad b = \frac{d\, S_c}{2\, S_s} = \tfrac{5}{4}\, d = 1.25\, d.$$

Combining (a) and (c)

$$\frac{\pi d^2}{4} - d\, t = d\, t\, \frac{S_c}{S_t}$$

$$t = \frac{\pi d}{12} = \tfrac{1}{4}\, d \text{ (about)}.$$

Combining (a) and (d) and taking $t = \frac{\pi d}{12}$,

$$\left\{ \frac{\pi d^2}{4} - \frac{\pi d^2}{12} \right\} S_t = \left\{ \frac{\pi}{4} (D^2 - d^2) - (D - d) \frac{\pi d}{12} \right\} S_t$$

$$D = \tfrac{4}{3}\, d.$$

To have the same tensile strength D should equal $\frac{4}{3}$ d but to prevent crushing, the bearing surface of the socket must equal that of the rod, or

$$(D - d)\ t = d\ t,$$

and $$D = 2\ d$$

Combining (a) and (e) and, as before, taking $t = \frac{\pi\ d}{12}$,

$$\left\{ \frac{\pi\ d^2}{4} - \frac{\pi\ d^2}{12} \right\} S_t = \frac{\pi\ d_1{}^2}{4}\ S_t,$$

$$d_1 = .816\ d.$$

Since the bearing surface of the collar should be equal to that of the cotter in the rod,

$$\frac{\pi}{4}\ (d_2{}^2 - d^2) = d\ t, \text{ and since}$$

$$t = \frac{\pi\ d}{12},$$

$$d_2 = 1.15\ d.$$

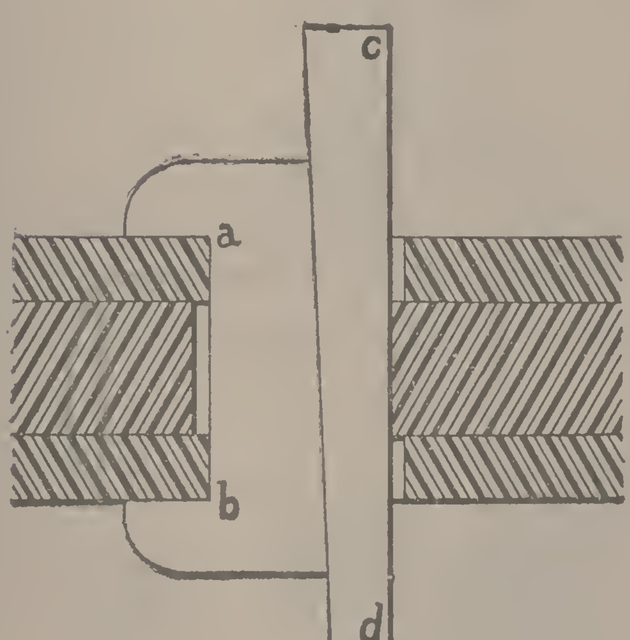


Fig. 40.

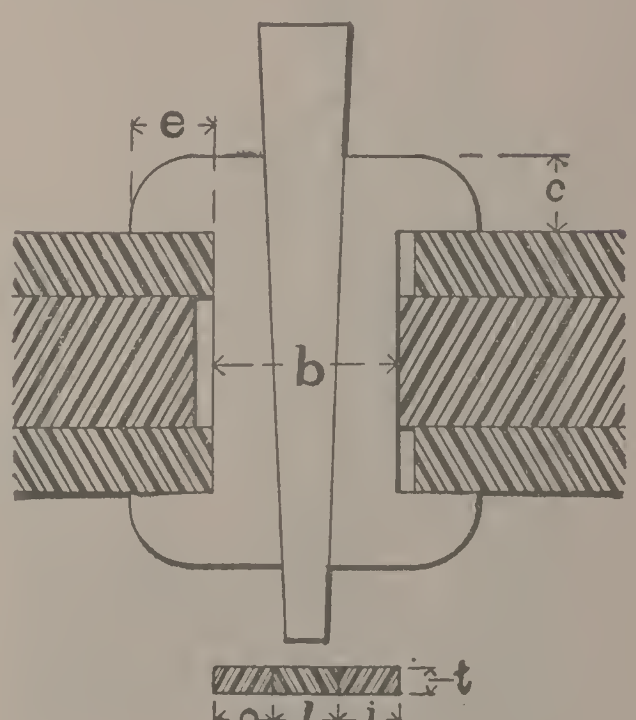


Fig. 41.

Tabulating the above results we find,

$$b = 1\tfrac{1}{4}\ d$$
$$t = \tfrac{1}{4}\ d$$
$$D = 2\ d$$
$$d_1 = .816\ d$$
$$d_2 = 1.15\ d$$

a is made equal to c and is from $\frac{3}{4}$ to $1\frac{1}{4}$ d.

Fig. 41 shows a cotter with **double gib**. In this case $b\ t$ is usually made $\frac{5}{4}$ the sectional area of the strap and $t = \frac{1}{4}\ b$. b is made the same for all cases, whether a single cotter, gib and cotter or two gibs are used. The other dimensions are.

$$c = \frac{b}{3}$$

$$e = \frac{b}{4}$$

$$i = \tfrac{5}{16}\ b$$

$$o = \tfrac{5}{16}\ b$$

$$l = \tfrac{3}{8}\ b$$

If a steel cotter is used in a wrought iron rod, b may equal d.

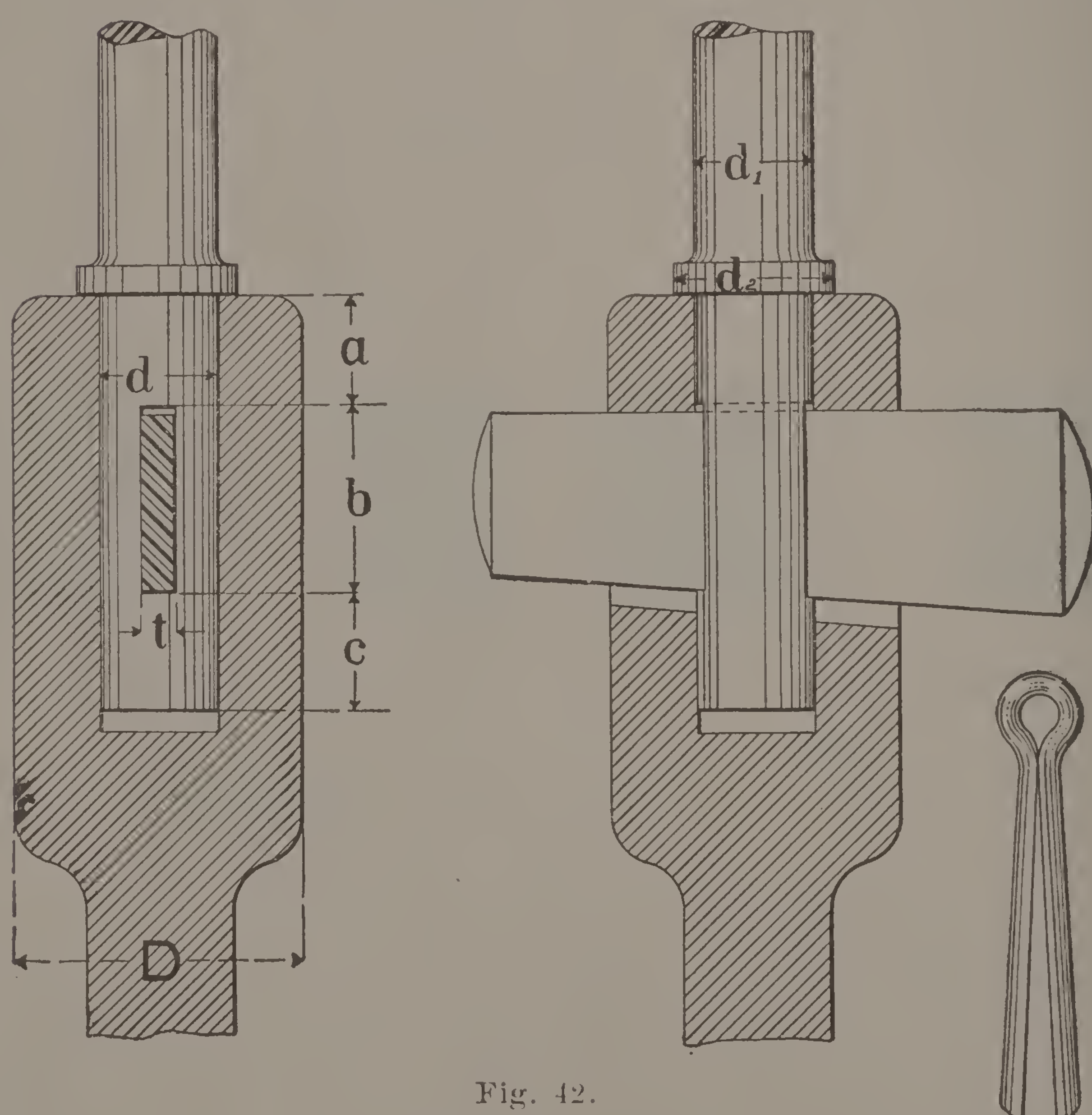


Fig. 42.

Fig. 43.

Fig. 43 shows a small **split pin**. Split pins are forms of cotters which are used to prevent two pieces from separating but do not connect them firmly. Large pins are made solid with a slight taper.

Locking Arrangements for Cotters. Fig. 44 shows a method

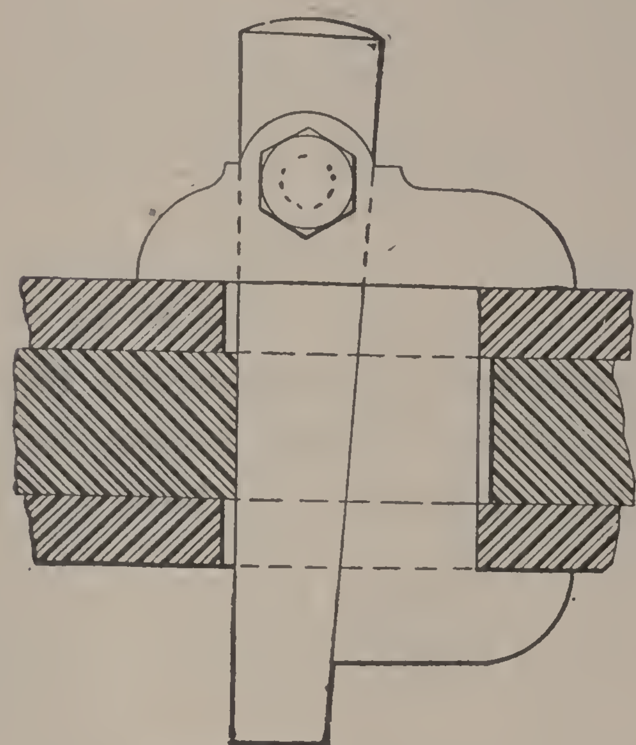


Fig. 44.

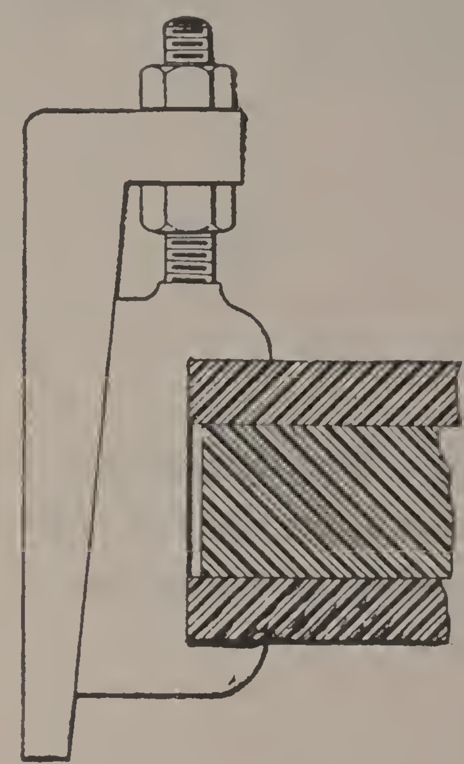


Fig. 45.

of securing the cotter by a **set screw.** The cotter passes through the head of the gib and is held by the screw. A simple way to secure the cotter is by prolonging the gib and having a screw

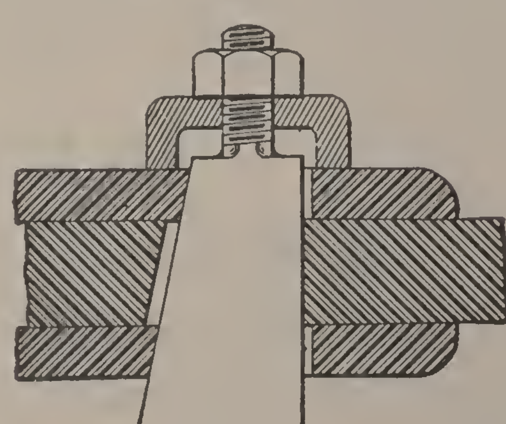


Fig. 46.

thread cut on it as shown in Fig. 45. Nuts on each side of the bent cotter keep it in place and adjust the length. In Fig. 46, the end of the cotter is a screw which passes through a recessed washer or extra seat. A nut above the washer holds it in place. In case a cotter has an excessive taper this method is sometimes used to prevent slacking back.

EXAMPLES FOR PRACTICE.

1. Find the dimensions of a rod and cotter of the form shown in Fig. 42. Assume $S_t = 6{,}500$ pounds. The load is 7,900 pounds.

$$\text{Ans.}\begin{cases} d = 1\frac{1}{2} \text{ inches} \\ d_1 = 1\frac{1}{4} \text{ inches} \\ d_2 = 1\frac{3}{4} \text{ inches} \\ D = 3 \text{ inches} \\ t = \frac{3}{8} \text{ inch} \\ b = 1\frac{7}{8} \text{ inches} \\ a = 1\frac{1}{8} \text{ to } 1\frac{7}{8} \text{ inches} \end{cases}$$

2. Two straps are connected to a rod as shown in Fig. 41. If the pull on the rod is 10,000 and $S_s = 5{,}500$ pounds, find the dimensions of the cotter and gibs. Assume $t = \frac{1}{2}$ inch.

$$\text{Ans.}\begin{cases} t = \frac{1}{2} \text{ inch.} \\ \text{width of cotter} = 1\frac{3}{8} \text{ inches.} \\ \text{breadth of gibs} = 1\frac{1}{8} \text{ inches.} \\ b = 3\frac{5}{8} \text{ inches.} \end{cases}$$

3. What are the dimensions of a wrought iron cotter used to fasten a wrought iron rod 2 inches in diameter?

$$\text{Ans.}\begin{cases} t = \frac{1}{2} \text{ inch} \\ b = 2\frac{1}{2} \text{ inches} \end{cases}$$

4. A cotter is $1\frac{1}{2}$ inches wide at the middle; it tapers on each side. If it is 18 inches long and tapers $\frac{1}{2}$ inch to the foot what is its width at each end?

Ans. $1\frac{7}{8}$ inches and $1\frac{1}{8}$ inches.

JOURNALS.

Journals are the portions of shafts and axles which turn in bearings and are supported by the frame of the machine. They are usually cylindrical but may be conical or spherical in form.

If the journal is at or near the end of the shaft it is called an **end** journal; if situated between two end journals it is called a **neck** journal.

The most common form for the bearing portion of a journal is a true cylinder as shown in Fig. 47. Collars or shoulders at the ends bear against the ends of the brasses, in which the journal revolves, and limit the play lengthwise. In case a light end play is desired, as in the car journal for instance, the bearing is made slightly shorter than the journal, thus permitting a little longitudinal

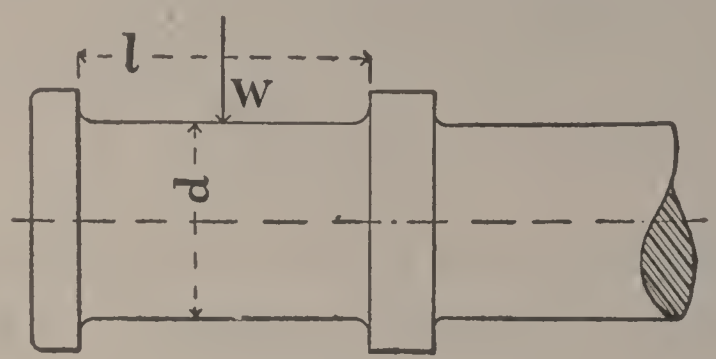


Fig. 47.

motion and causing uniform wear of the brasses. When longitudinal motion would interfere with the movements of other portions of the machine, it is made as small as possible.

The methods for calculating the requisite size of a journal depend upon the velocity of the shaft and the constancy with which it is run. In case it runs occasionally or at slow speed it is designed for strength; but if it runs constantly at high speed, durability and freedom from heating are as important elements as strength.

Journals may be subjected to straining forces in the plane of the axis (causing bending and shearing stresses) and also to combined torsional and bending stresses.

When designing for strength the journal is considered as a cantilever beam uniformly loaded.

If l = the length of the journal in inches,
d = the diameter of journal in inches,
S = the safe working stress,
I = the moment of inertia,
c = the distance of the fibre most remote from the neutral axis,
W = the total load in pounds,
w = load per square inch in pounds;

then from "Mechanics,"

$$W = 2 \frac{S}{l} \times \frac{I}{c},$$

and since for a circular section,

$$I = \frac{\pi d^4}{64} \text{ and } c = \frac{d}{2}$$

$$\text{then,}\quad W = 2\,\frac{S}{l} \times \frac{2\,\pi\,d^4}{64\,d} = \frac{\pi\,d^3\,S}{16\,l}$$

$$\text{and,}\quad 16\,W\,l = \pi\,d^3\,S$$

$$\text{from which } d = \sqrt{\frac{16\,W}{\pi\,S} \times \frac{l}{d}}.$$

The equation is used in the above form because the ratio of length to diameter $\frac{l}{d}$ is usually assumed, *i. e.*, fixed before the calculation is made. If this ratio were not assumed there would be two unknown quantities in the equation.

For journals which work intermittently the ratio l to d is usually 1, or $\frac{l}{d} = 1$. Where the speed is less than 150 revolutions per minute $\frac{l}{d}$ varies from 1.5 to 1.75. The greater the speed the greater the proportion of length to diameter; this causes a reduction of pressure per unit of bearing surface as the speed increases.

For example. Find the length and diameter of a steel journal, having a load of 2,000 pounds. Assume $\frac{l}{d}$ to be 1.5 and the safe working stress, S, as 9,000 pounds.

$$d = \sqrt{\frac{16\,W}{\pi\,S} \times \frac{l}{d}} = \sqrt{\frac{16 \times 2{,}000}{3.1416 \times 9{,}000} \times \frac{3}{2}} = 1.3 \text{ inches.}$$

In this case the journal would be $1\frac{5}{16}$ inches in diameter and $1\frac{5}{16} \times 1.5 = 2$ inches long.

The safe working stress varies with the conditions and the material. Average values of S are as follows:

Material.	Constant Load.	Variable Load.
Steel	12,000 to 13,000	9,000 to 12,000
Wrought Iron	7,000 to 9,000	6,000 to 7,000
Cast Iron	3,500 to 4,500	3,000 to 4,000

When designing to provide against heating, experience determines the allowable pressure per square inch on the area of projection of the journal. This is called the projected area and is equal to the length of the journal multiplied by the diameter, or,

$$\text{area} = l \times d.$$

The load per square inch evidently is the total load divided by this area, or,

$$w = \frac{W}{l \times d}.$$

If the pressure per square inch w, is too large the lubricant is squeezed out and the journal is likely to heat and increase in size. The better the lubricant, the larger w may be. For high speeds the pressure may be greater than for low speeds. In the case of journals, as for example, crank pins, where the pressure alternates in direction the limit of pressure may be about twice as great as where the load is in a fixed direction. This is because of the better lubrication in case of alternating direction.

Since $W = w\ l\ d$, and $W = \frac{\pi\ d^3\ S}{16\ l}$;

then $w\ l\ d = \frac{\pi\ d^3\ S}{16\ l}$, or $\pi\ d^2\ S = 16\ w\ l^2$,

and $\frac{l^2}{d^2} = \frac{\pi\ S}{16\ w}$, or $\frac{l}{d} = \sqrt{\frac{\pi\ S}{16\ w}}$.

Substituting this value of $\frac{l}{d}$ in the equation $d = \sqrt{\frac{16 W}{\pi\ S} \times \frac{l}{d}}$,

the result is, $d = 2\sqrt{\frac{W}{\sqrt{\pi\ S\ w}}}$,

and since $w\ l\ d = W$,

$$l = \frac{W}{w\ d}.$$

Therefore in calculating the diameter when the pressure w is assumed we use the formula $d = 2\sqrt{\frac{W}{\sqrt{\pi\ S\ w}}}$, and $l = \frac{W}{w\ d}$ for the length.

Find the length and diameter of a steel journal when the load is 11,000 pounds. The safe working stress is 10,000 pounds and the allowable pressure is 900 pounds.

From the formula we obtain;

$$d = 2\sqrt{\frac{W}{\sqrt{\pi}\,S\,w}} = 2\sqrt{\frac{11,000}{\sqrt{3.1416 \times 10,000 \times 900}}}.$$

The square root of $\pi = 1.77245$, then $d = 2.87 +$ inches, and the shaft would be 3 inches in diameter. We also find

$$l = \frac{W}{w\,d} = \frac{11,000}{900 \times 3} = 4.07 = 4\tfrac{1}{16} \text{ inches long.}$$

The following table of limits of pressure per square inch of projected area for different conditions is taken from Unwin's Machine Design.

PRESSURE ON BEARINGS AND SLIDES.

Pressure Calculated in lbs. per sq. in. of Bearing Surface.	Intensity of pressure, lbs. per sq. in.
Bearings on which the load is intermittent and the speed slow, such as crank pins of shearing machines	3000
Cross-head neck journals	1200
Crank pins of large slow engines	800 to 900
Crank pins of marine engines, usually	400 to 500
Main crank shaft bearings Marine engines (slow)	600
Main crank shaft bearings Marine engines (fast)	400
Railway journals	300
Fly wheel shaft journals	150 to 250
Small engine crank pins	150 to 200
Slipper slide blocks, Marine engines	100
Stationary engine slide blocks	25 to 125
Stationary engine slide blocks, usually	30 to 60
Propeller thrust bearings	50 to 70
Shafts in cast-iron steps (Sellers)	15

NECK JOURNALS.

A neck journal is considered as a simple beam supported at both ends and uniformly loaded. The cross head pin, or wrist pin is an example of this kind of journal.

From "Mechanics," the equation for the above beam is,

$$W = 8 \frac{S}{l} \times \frac{I}{c}$$

$$\text{Since } I = \frac{\pi d^4}{64}, \text{ and } c = \frac{d}{2},$$

$$W = 8 \frac{S}{l} \times \frac{\pi d^3}{32},$$

$$\text{from which } d = 2\sqrt{\frac{W}{\pi S} \times \frac{l}{d}}.$$

This formula is used to calculate the value of d when the ratio $\frac{l}{d}$ is assumed.

Example. Find the diameter and length of a wrought iron neck journal when the load is 4,400 pounds, S assumed to be 8,000 pounds and $\frac{l}{d} = 2$.

$$\text{Solution,} \quad d = 2\sqrt{\frac{W}{\pi S} \times \frac{l}{d}} = 2\sqrt{\frac{4{,}400}{3.1416 \times 8{,}000} \times 2}.$$

$$= 2 \times .59 = 1.18 \text{ inches.}$$

A $1\frac{3}{16}$ inch shaft would probably be chosen.

Then, also, $l = 2\,d$, $2 \times 1\frac{3}{16} = 2\frac{3}{8}$ inches.

As in end journals $W = w\,d\,l$, and since $W = \frac{8\,S\,\pi\,d^3}{32\,l}$ equating the two values for W, we get,

$$w\,l\,d = \frac{8\,S\,\pi\,d^3}{32\,l},$$

$$\text{or,} \quad 4\,w\,l^2 = \pi\,d^2\,S.$$

$$\text{Then } \frac{l^2}{d^2} = \frac{\pi S}{4\,w}, \qquad \text{and } \frac{l}{d} = \sqrt{\frac{\pi S}{4\,w}}$$

Substituting this value of $\frac{l}{d}$ in the equation $d = 2\sqrt{\frac{W}{\pi S} \times \frac{l}{d}}$

we get, $d = 2\sqrt{\frac{W}{\pi S} \times \sqrt{\frac{\pi S}{4w}}}$

and reducing, $d = 2\sqrt{\frac{W}{\sqrt{4 \pi S w}}}$,

as before, $l = \frac{W}{w d}$.

The same values for S and w, given for end journals may be used for neck journals.

Suppose we wish to find the length and diameter of a neck journal having a load of 8,000 pounds. Let us assume S to be 8,500 pounds and the bearing pressure 500 pounds ; then,

$$d = 2\sqrt{\frac{W}{\sqrt{4 \pi S w}}} = 2\sqrt{\frac{8,000}{\sqrt{4 \times 3.1416 \times 8,500 \times 500}}}$$

$$= 2\sqrt{1.09} = 2.08 \text{ inches.}$$

A $2\frac{1}{8}$ inch shaft would be used, and we also find

$$l = \frac{W}{w d} = \frac{8,000}{500 \times 2\frac{1}{8}} = 7\tfrac{1}{2} \text{ inches (about.)}$$

DIMENSIONS OF JOURNALS.

We have already seen how the diameter and length of journals are determined. Knowing the diameter, the height of the collar may be found from the formula $h = .1\, d + \frac{1}{8}''$ and the breadth equals $1\frac{1}{2}$ times the height.

Usually the journal is turned with a fillet in the corner, since the shaft is less likely to crack than if it has a square corner. The radius of the circular fillet is about one half the height.

FRICTION OF JOURNALS.

In every case, friction generates heat which causes the temperature of the journal to rise. This rise of temperature increases the size of the journal and causes unnecessary work.

In order to diminish friction, the journal is lubricated, that is, it is supplied with some lubricant (oil or fatty matter) which forms a thin film between the journal and the bearing. The lubricant diminishes friction and greatly reduces wear.

The surface velocity in feet per minute of a journal is evidently the circumference of the circle whose diameter is d, multiplied by the number of revolutions n, or expressed algebraically,

$$\text{Surface velocity} = \frac{\pi\ d\ n}{.12}\text{feet per minute.}$$

If f = the coefficient of friction, and W the load, the work expended in friction is expressed by the formula :

$$\text{Work} = f\ \text{W} \times \frac{\pi\ d n}{12}\ \text{foot ponnds per minute.}$$

As we have seen from the table on page 54, the pressure per square inch on the bearing surface cannot exceed certain limits. These limits varying with the kind of work done and the speed. For the load W a certain bearing surface $d l$ is necessary.

The value of the coefficient f varies greatly ; it is dependent upon speed, pressure, kind of metals, and kind of lubricant. Under some conditions it may be as low as .001 and under others .2.

From the above equation it is evident that the work used in friction is directly proportional to the diameter of the shaft. This fact shows that under a constant load it is well to obtain a large projected area, $d\ l$, by increasing l and making d as small as is consistent with strength.

Suppose we have two journals, $1\frac{1}{2}$ inches in diameter and $4\frac{1}{2}$ inches long ; the other 3 inches in diameter and 2.25 inches long. They have the same projected area, *i.e.*, 6.75 square inches ; but the work required to overcome friction of the first is only one-half as much as for the second.

Although the first, *i. e.*, the $1\frac{1}{2}$ inch journal is preferable for a steady load, it is not as strong as the three inch journal, and the latter should be used for high speeds and when the journal is subject to shocks.

PIVOT AND COLLAR JOURNALS.

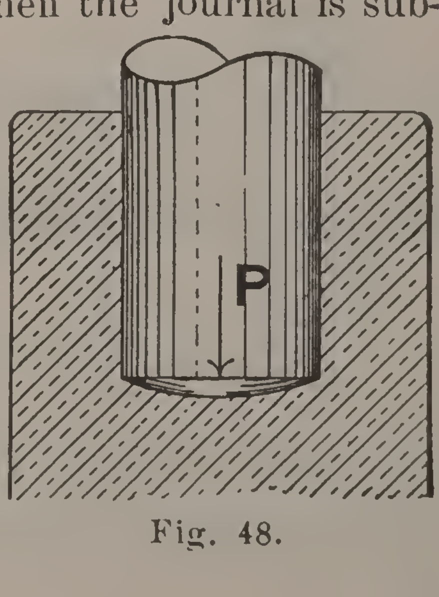


Fig. 48.

In journals already considered the direction of pressure is perpendicular to the axis of the shaft; in the pivot journal shown in Fig. 48 the direction of pressure is parallel to the axis. With end journals the bearing surface is the cylindrical surface ; but in pivot journals it is circular, and is the area of the end of the pivot. To find the diameter of such a journal knowing the load, assume a value for the pressure per square inch of bearing surface and solve for the area. This may be expressed by the formula,

$$W = \frac{\pi d^2}{4} \times w$$

Suppose the total load is 32,000 pounds and the allowable pressure on the bearing surface is 650 pounds per square inch. What is the diameter?

$$W = \frac{\pi d^2}{4} \times w, \text{ and } 32{,}000 = \frac{\pi d^2}{4} \times 650.$$

Then $\frac{\pi d^2}{4}$ = the area = 49.23 square inches.

and $d = 7.9$ inches.

An 8 inch shaft would be used.

In cases where the speed is not very high we may use the following as maximum values of w:

Wrought iron on gun metal	700 pounds
Cast iron on gun metal	470 pounds
Wrought iron or steel on lignum vitae	1,400 pounds

For high speeds with iron or steel on lignum vitae bearing, when moistened with water, the following formula may be used,

$$d = .035 \sqrt{W}.$$

The direction of load for a collar bearing is parallel to the axis but the bearing area is formed by a collar or collars on the shaft. Fig. 49 shows a journal with one collar and Fig. 50 one

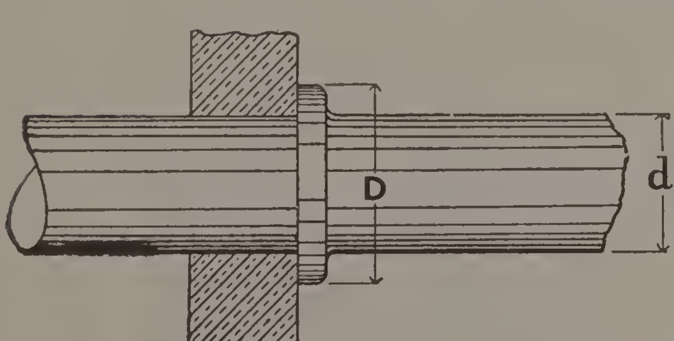


Fig. 49.

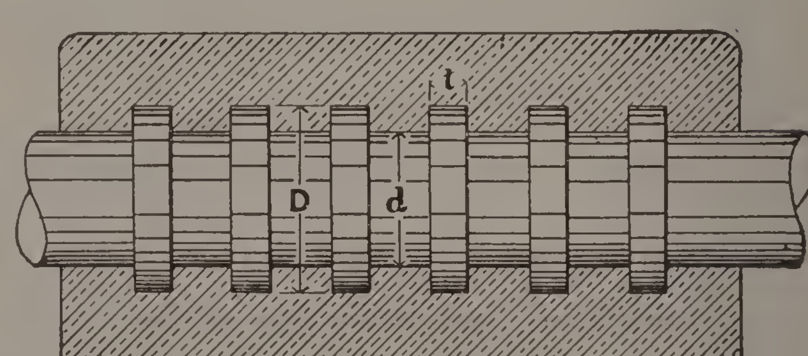


Fig. 50.

with several collars. To find the diameter of collars for propeller shafts we proceed as follows :

Let d = diameter of shaft,
D = diameter of collar,
N = number of collars,
w = pressure per square inch of projected area,
W = total load.

The total load is expressed by the equation,

$$W = \frac{1}{4} \pi (D^2 - d^2) N w.$$

Usually w is taken as 50 or 60 pounds for propeller shafts. Using $w = 60$ the formula becomes,

$$W = \pi (D^2 - d^2) 15 N,$$

or $$(D^2 - d^2) = \frac{W}{15 \pi N}.$$

Hence $$D = \sqrt{d^2 + \frac{W}{15 \pi N}}.$$

The number of collars is determined by the designer. By increasing the number, the diameter, wear and friction are decreased; but if too many collars are used all the thrust may be brought on a few of them. The distance between collars is sometimes equal to the thickness. But if the encircling rings are lined with white metal or are made hollow the distance may be made about twice the thickness.

EXAMPLES FOR PRACTICE.

1. What are the proportions of a wrought iron end journal, when $\frac{l}{d} = 1.2$, the load 1,800 pounds, and safe stress 8,500 pounds? Ans. $1\frac{3}{16}'' \times 1\frac{7}{16}''$.

2. Find the length and diameter of an end journal when S = 9,000 pounds, W = 8,000 pounds and w = 800 pounds. Ans. $3\frac{13}{16}'' \times 2\frac{5}{8}''$

3. Find the proportions of a neck journal, if the load is 7,000 pounds, the bearing pressure 400 pounds and safe stress 9,000 pounds. Ans. $2\frac{1}{16}'' \times 8\frac{1}{2}''$.

4. Find the proportions of a neck journal if the load is 6,000 pounds, the safe stress 8,500 pounds and $\frac{l}{d} = 1.75$.

Ans. $1\frac{1}{4}'' \times 2\frac{3}{16}''$.

5. Find the power used in friction when the load is 6,000 pounds, the diameter of the shaft 2 inches, and making 150 revolutions per minute. The coefficient being .002.

Ans. 942.48 foot-pounds per minute.

6. Find the diameter of a wrought iron pivot running at 90 revolutions per minute, with a load of 850 pounds, having gun metal bearings. Ans. $1\frac{1}{4}$ inch shaft.

7. Find the diameter of the collars on an 8 inch shaft, the end thrusts being 18,000 pounds. There are four collars.

Ans. 12.6 inches. Use $12\frac{5}{8}$ inches.

SHAFTS.

Shafts are parts of machines which support rotating pieces. They are usually circular in section.

Shafts may be divided into three classes. The classification being determined by the kind of stresses to which they are subjected.

1. Shafts subjected chiefly to torsion or twisting. For example, shafting used to transmit power, called line shafting.

2. Shafts subjected chiefly to bending; shafts of gearing ' example.

3. Shafts subjected to both torsion and bending, as engine shafts.

Shafts of the first class, those subjected to torsion, must be designed for both strength and stiffness. If a shaft is of large diameter or if it is short it may be designed for strength only; but for a long shaft of small diameter, sufficient strength may not insure sufficient stiffness. Line shafting is the name given to the long continuous lines of shafting used in mills, factories, etc. Numerous pulleys are keyed to the shafts from which power is taken by belts and gears. These shafts are strained by twisting stresses and also by a slight bending action due to the weight and the downward pull of the gearing.

Calculations for size are made by investigating the twisting moment which is equal to the force or pull multiplied by the radius.

From the study of "Mechanics" we know that the diameter of a round shaft may be found from the formula:

$$d = 68.5 \sqrt[3]{\frac{H}{n\,S_s}},$$

in which d = the diameter in inches,
H = the horse-power transmitted,
n = the number of revolutions per minute,
and S_s = the constant of torsion and is:
2,000 pounds per square inch, for timber.
25,000 pounds per square inch, for cast iron.
30,000 pounds per square inch, for wrought iron.
75,000 pounds per square inch, for steel.

Another formula for ordinary wrought iron mill shafting, is:

$$d = \sqrt[3]{\frac{H}{n \times .01153}}$$

The following table has been computed from this formula; by multiplying the second column by the number of revolutions per minute, the result is the power the shaft will transmit.

Diameter of Shaft in inches.	$\frac{H.\ P.}{n}$	Diameter of Shaft in inches.	$\frac{H.\ P.}{n}$
$1\frac{3}{4}$	0.0623	5	1.4536
2	0.0930	$5\frac{1}{2}$	1.9344
$2\frac{1}{4}$	0.1325	6	2.5112
$2\frac{1}{2}$	0.1817	$6\frac{1}{2}$	3.1944
$2\frac{3}{4}$	0.2418	7	3.9888
3	0.3139	$7\frac{1}{2}$	4.9056
$3\frac{1}{4}$	0.3993	8	5.9536
$3\frac{1}{2}$	0.4986	$8\frac{1}{2}$	7.1440
$3\frac{3}{4}$	0.6132	9	8.4800
4	0.7442	10	11.6288
$4\frac{1}{4}$	0.8930	11	15.4752
$4\frac{1}{2}$	1.0600	12	20.0896
$4\frac{3}{4}$	1.2470		

Hollow Shafts. The inner fibres of a shaft, that is, those near the centre are not as useful to resist torsion as the outer. Therefore, in making a shaft hollow considerable weight is removed and the strength is but little impaired. In other words, if a shaft is made hollow its weight is decreased in a greater measure than its strength. A hollow shaft is much stronger than a solid one of the same weight. For marine work hollow shafts are especially valuable on account of their light weight.

Let D = the outside diameter of the hollow shaft.

d = the diameter of a solid shaft having the same strength as the hollow shaft.

D_1 = the inside diameter of the hollow shaft.

From a mathematical consideration of moments and stresses the following formula is deduced:

$$D = d\sqrt[3]{\frac{1}{1 - \left\{\frac{D_1}{D}\right\}^4}}$$

Suppose we wish to find the diameter of a hollow shaft which shall have the same strength as a solid shaft of nine inches diameter, the internal diameter to be $\frac{1}{2}$ the external.

$$D = d\sqrt{\frac{1}{1 - \left\{\frac{D_1}{D}\right\}^4}}$$

$$= 9\sqrt[3]{\frac{1}{1 - \left\{\frac{1}{2}\right\}^4}} = 9 \times 1.022 = 9.198 \text{ inches.}$$

The shaft may be made $9\frac{3}{16}$ inches in outside diameter and $4\frac{19}{32}$ in inside diameter.

The **distance between bearings** depends upon the number of pulleys on the shaft. If the shaft has several pulleys keyed to it the bearings must be nearer than if there are only a few pulleys. Large pulleys and those giving out a large amount of power should be placed near the bearings. The distance should be such that the deflection can not be more than $\frac{1}{100}$ of an inch per foot.

The following table gives maximum distance between **bearings** for shafts of various sizes.

Diameter of Shaft in inches.	Distance between Bearings in Feet.	
	Wrought Iron Shafts.	Steel Shafts.
2	15.46	15.89
3	17.70	18.19
4	19.48	20.02
5	20.99	21.57
6	22.30	22.92
7	23.48	24.13
8	24.55	25.23
9	25.53	26.24

SHAFT COUPLINGS.

Since it would be inconvenient to manufacture shafts long enough for a large factory, some means must be provided to join short lengths together.

Shafting is usually made in lengths of 20 to 30 feet, and joined by shaft couplings. These couplings should be placed near bearings and on the side farthest from the power. If this is done the running part is supported even if a length is disconnected.

Couplings are made in various shapes according to the positions of the shafts to be joined. They may be designed for shafts having a common axis of rotation, that is, in line; for parallel shafts; or for shafts whose axes intersect.

Shaft couplings are usually divided into three classes.

1. **Mixed or permanent** couplings, which can be disconnected only by slacking keys or by removing nuts.

2. **Loose or disengaging** couplings, which are provided with arrangements for throwing a part of the shafting out of gear with slight effort.

3. **Friction Couplings** which are loose and so arranged that they put the shafting into gear gradually and slip if the resistance is great.

The simplest form of shaft coupling is the box or muff shown in Fig. 51. A short iron cylinder is fitted over the ends of the shafts. Relative rotation is prevented by a wrought iron key which is usually half in the shaft and half in the coupling. The shafts may be enlarged at the ends so that there will be no decrease in strength because of the key way. Couplings are

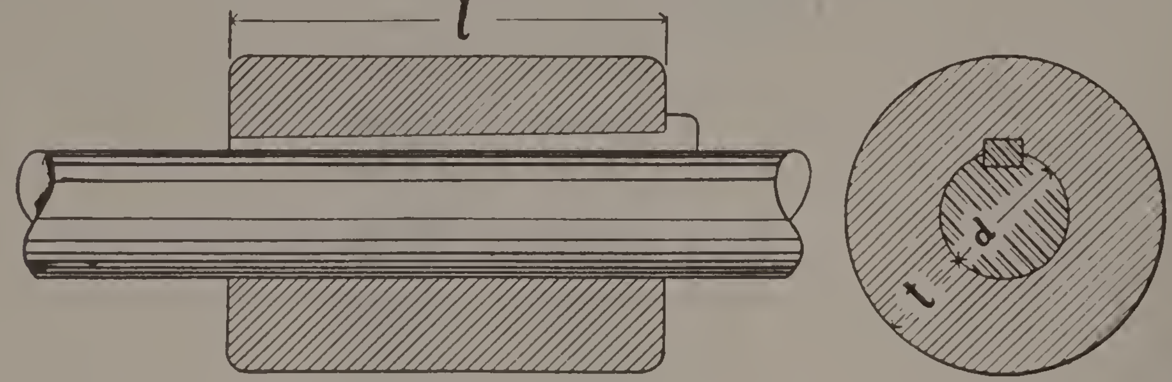


Fig. 51.

designed from empirical formulas and not from calculations for strength. In Fig. 51 d = the diameter of the shaft, and is taken as the unit.

The dimensions may be calculated from the formulas,

$$l = 2\tfrac{1}{2}\,d + 2''$$
$$\text{and } t = .45\,d + \tfrac{1}{2}''.$$

For the key,

$$b = \tfrac{1}{4}\,d$$
$$t = \tfrac{1}{6}\,d.$$

For example. Find the dimensions for a box coupling for a shaft 3 inches in diameter.

$$l = (2\tfrac{1}{2} \times 3) + 2'' = 9\tfrac{1}{2} \text{ inches}$$
$$t = .45\,d + \tfrac{1}{2}'' = 1.85 \text{ or } 1\tfrac{7}{8} \text{ inches.}$$

The key would be,

$$b = \tfrac{3}{4} \text{ inch}$$
$$t = \tfrac{1}{2} \text{ inch.}$$

The coupling shown in Fig. 52 is called the **clamp coupling.** It is made of cast iron, is easily removed, has no projecting parts

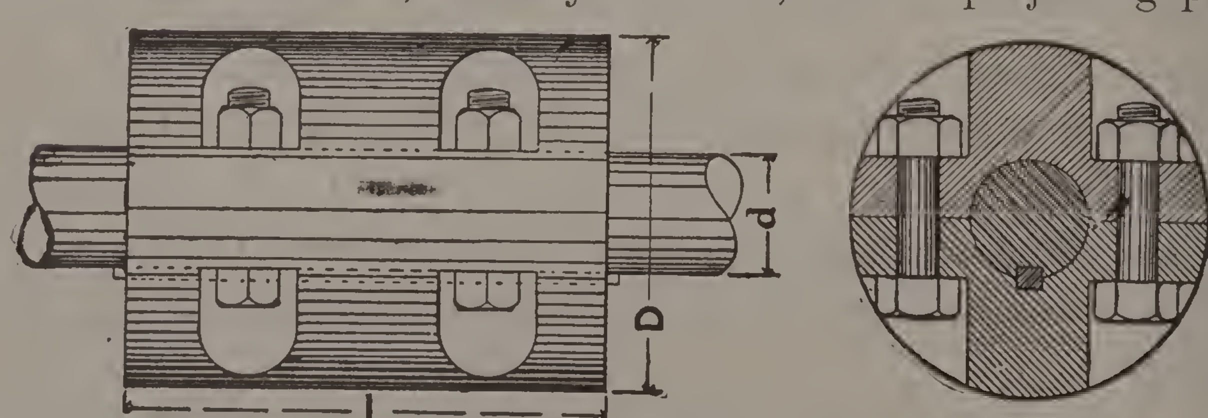


Fig. 52.

and on account of its cylindrical shape can be used as a pulley. The faces of the joint are first planed then the holes are drilled.

It is bored out after the two halves are bolted together with paper between them. When the paper is removed a slight space is left between the halves so that the coupling grips the shaft when the parts are bolted together. The key is straight and fits only at the sides so that it will not exert bursting pressure on the coupling. The following formulas may be used in finding the proportions; d being the unit.

$$D = 2\tfrac{1}{2}\, d + \tfrac{1}{2}''$$
$$l = 3 \text{ to } 4\, d.$$

The bolts may be $\frac{5}{8}$ inch in diameter for shafts under $2\frac{1}{2}$ inches in diameter and $\frac{3}{4}$ or $\frac{7}{8}$ inch for larger shafts. Usually four bolts are used for small and six bolts for large shafts.

Find the dimensions for a clamp coupling for a $3\frac{1}{2}$ inch shaft.

$$D = 2\tfrac{1}{2}\, d + \tfrac{1}{2}'' = 9\tfrac{1}{4} \text{ inches.}$$
$$l = 3\, d = 10\tfrac{1}{2} \text{ inches.}$$

We can use 6 bolts of $\frac{7}{8}$ inch.

The **flange coupling** is shown in Fig. 53. The cast iron

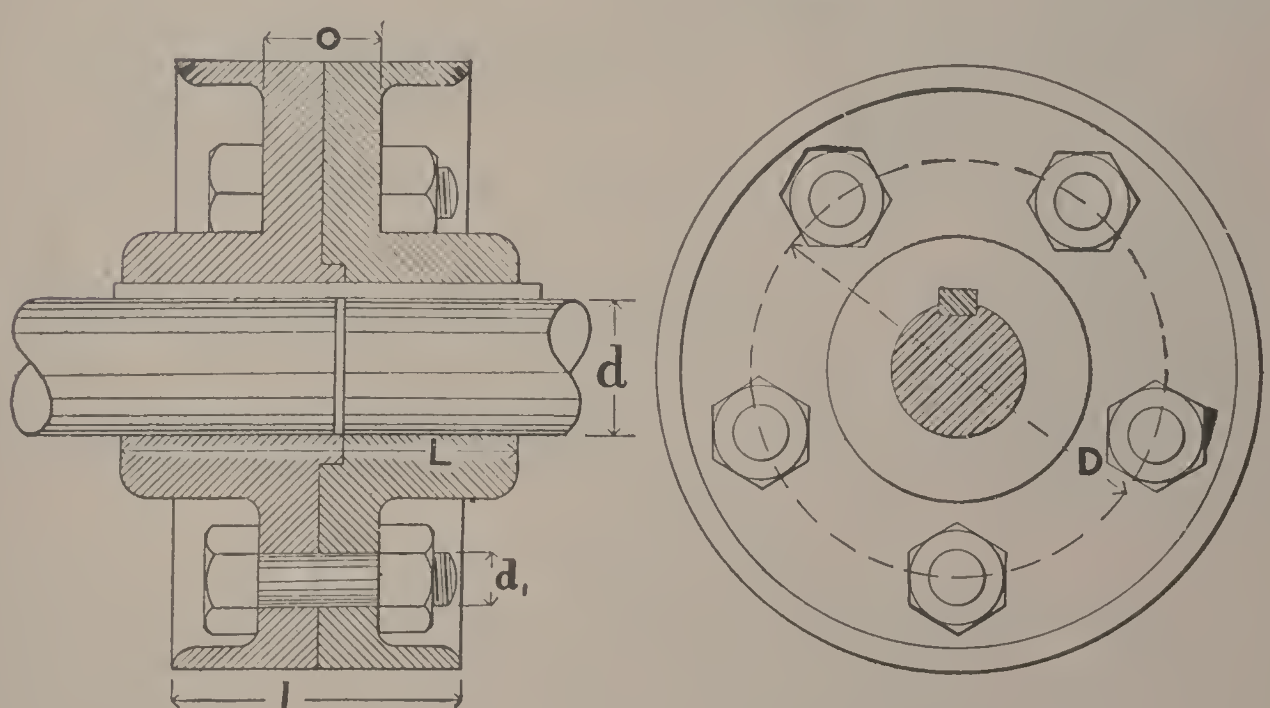


Fig. 53.

flanges are keyed to the ends of the shafts to be connected. The flanges are then brought face to face and bolted together. Sometimes the flanges are faced in a lathe to insure a good joint. One flange is often made to enter the other, as shown in the figure, to prevent the shafts from getting out of line. As in the other shaft couplings, the ends of the shafts may be enlarged for the key way. For designing, d = the diameter of the shaft, which is taken as the unit.

$$D = 2\tfrac{1}{2}\, d + 2''$$
$$l = 2d.$$

Number of bolts, n, $= 3 + \frac{d}{2}$ (the nearest whole number.)

$$d_1 = \frac{d}{n} + \tfrac{1}{4}''$$
$$o = \tfrac{1}{2}\, d + \tfrac{5}{8}''.$$

Propeller shaft coupling. The coupling shown in Fig. 54 is

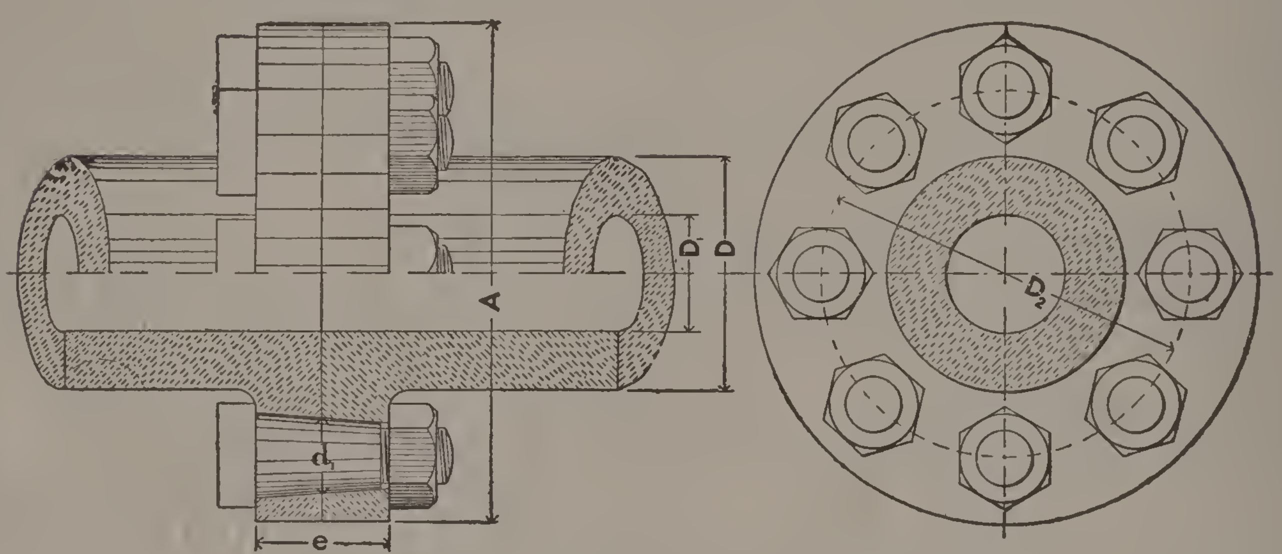


Fig. 54.

used for propeller shafts. In this case the hollow steel shaft is flanged at the ends and joined by bolts. Let d = the diameter of an equivalent solid shaft. Then,

$$d = \sqrt[3]{\frac{D^4 - D_1^4}{D}}$$
$$D_2 = 1\tfrac{3}{4}\, d$$
$$A = 2\, d$$
$$e = \frac{d}{2}$$

The bolts are made tapered. The number of bolts is usually assumed, and the diameter d_1, can be computed by a consideration of the twisting moment of the shaft and the shearing strength of the bolts.

Let n = the number of bolts, and a a constant: then $d_1 = a\,d$. The values of a for different values of n are as follows:

n =	3	4	5	6	7	8	9	10
a =	.318	.283	.258	.239	.224	.212	.201	.192

Find the dimensions of a shaft coupling for a hollow propeller shaft 4 inches in external diameter and 2 inches in internal diameter using 8 bolts.

$$d = \sqrt[3]{\frac{4^4 - 2^4}{4}} = 3.915 \text{ inches.}$$

$$D_2 = 1\tfrac{3}{4} \times 3.915 = 6\tfrac{7}{8} \text{ inches (about).}$$

$$A = 8 \text{ inches (about).}$$

$$e = 2 \text{ inches (about).}$$

$$d_1 = .212 \times 3.915 = \tfrac{13}{16}.$$

Bolts $\frac{3}{4}$ inch in diameter would be used.

Sellars Cone Coupling. A convenient coupling for shafts of equal or unequal size is shown in Fig. 55. It consists of an outer

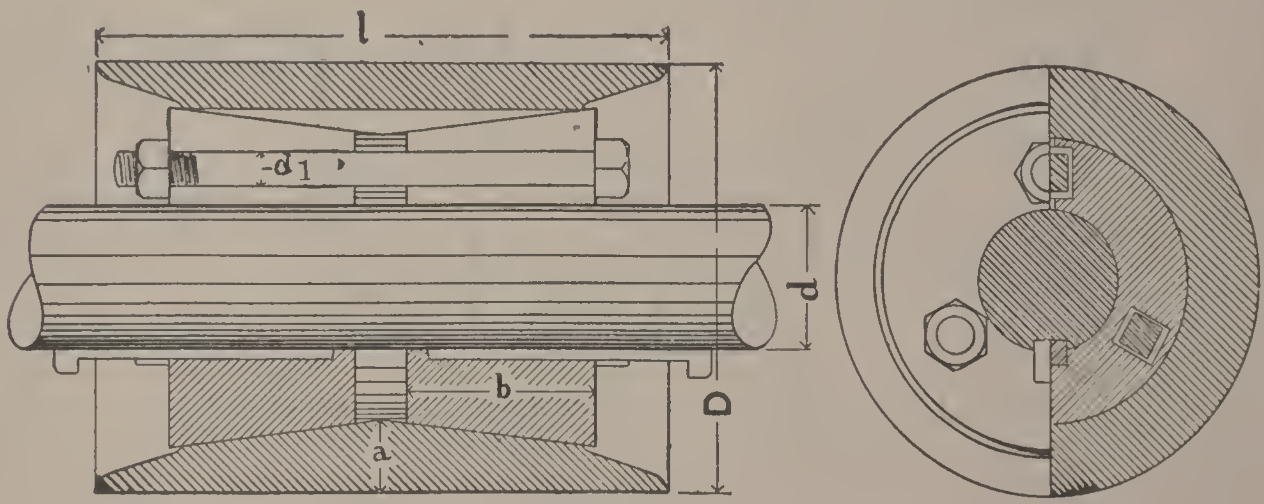


Fig. 55.

cylindrical box or muff which is turned of double conical form on the inside. Two sleeves, the exterior of which are conical and fit the conical surfaces of the box, are placed between the box and the shaft. The inside surfaces of the sleeves fit the shaft. Three square bolts parallel to the shaft and resting in slots cut in the sleeves, press them together. The sleeves are cut through on one side at the bottom of one of the bolt slots. This gives sufficient elasticity so that the sleeves may be drawn inward and grasp the shaft tightly. Each sleeve exerts the same force on the shaft and with the aid of a key prevents slipping. The keys fit at the sides only. These couplings are easily disconnected if the parts are well oiled before they are put together. The dimensions may be found from the following formulas:

$$D = 3\,d$$
$$l = 4\,d$$
$$b = 1\tfrac{1}{2}\,d$$
$$a = \tfrac{1}{2}\,d$$
$$\text{size of bolts } d_1 = \tfrac{1}{3}d$$

In the above formulas d is the unit, and is taken as the diameter of the shaft. The conical sleeves have a taper of about 4 inches per foot of length.

The Oldham Coupling. Fig. 56 shows a form of coupling used when two shafts are parallel. A disc is keyed on the end of each shaft. Between these discs lies a third which has a

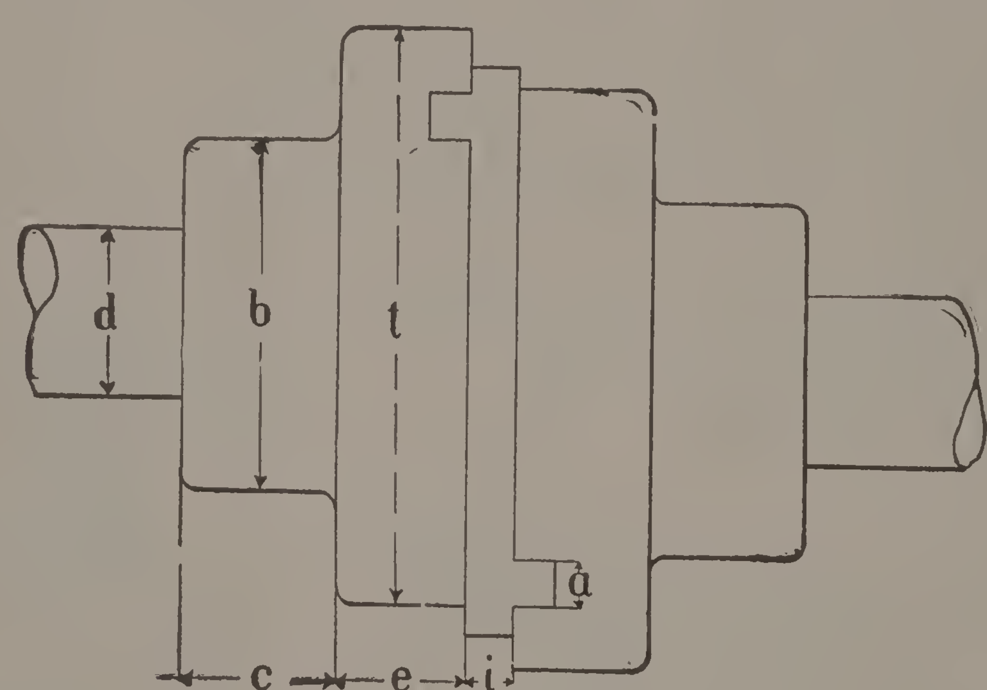


Fig. 56.

feather on each side fitting in a slot in the corresponding disc. The middle disc revolves around an axis parallel to the shafts and midway between them. The shafts and disc have equal velocities.

The proportions for this coupling may be as follows:

$d =$ diameter of shaft,
$a = .4\,d,$
$b = 1.75\,d,$
$c = .8\,d,$
$e = .7\,d,$
$i = .25\,d.$
$t = 3\,d.$

The Universal Coupling. In case two shafts are not in line they may be connected by a universal coupling shown in Fig. 57.

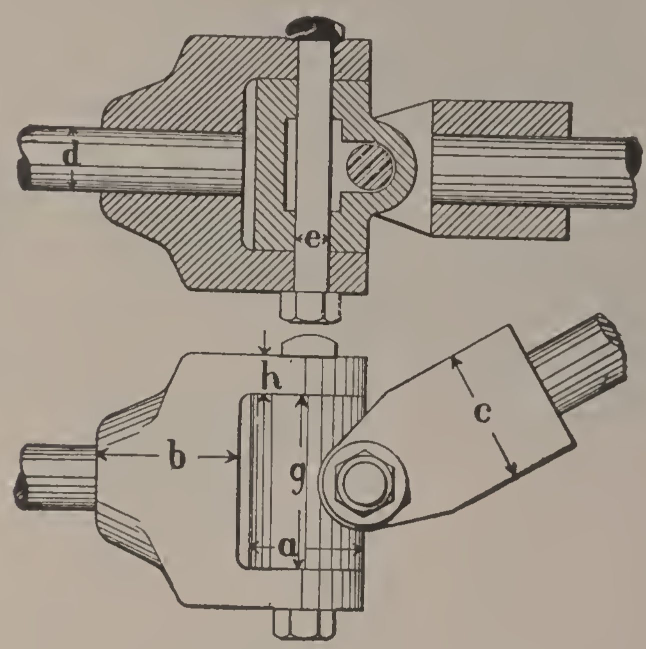


Fig. 57

The velocity ratio varies but little if the angle is small. They are constructed of wrought iron and may have the following proportions; d being the unit.

d = diameter of the shaft,

$a = d$ to $2\ d$,

$b = 1\frac{5}{8}\ d$,

$c = 1\frac{7}{8}$ d.

$e = \frac{1}{2}\ d$.

$g = 2d$,

$h = \frac{5}{8}\ d$.

Loose Couplings are used if shafts are to be connected and disconnected. A type called a claw coupling which somewhat resembles the flange coupling is shown in Fig. 58. It is used for large slow turning shafts, which always revolve in the same direction. This form is easily put in gear. In place of the flanges there is a set of projections or lugs which fit into recesses. One part is firmly keyed to the shaft by a sunk key; the other is fastened by a feather key. The part having the feather key (on the left hand) is prolonged and a groove cut for a lever with which to slide it back and forth.

A coupling which is easily put in gear but can drive only in one direction has its claws shaped as shown in Fig. 59.

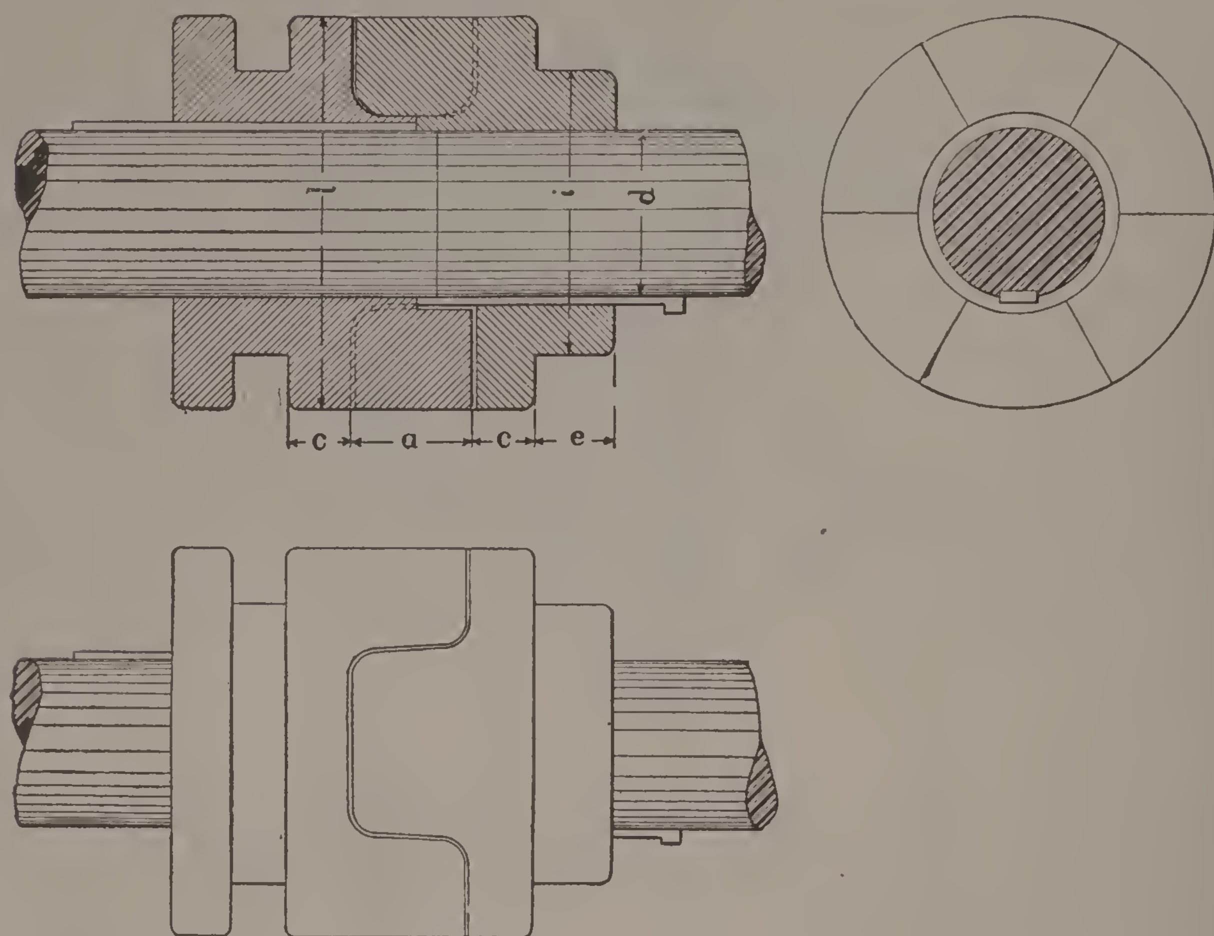


Fig. 58.

The dimensions may be as follows:

d = diameter of shaft,
$a = \frac{5}{8}\ d$,
$c = \frac{3}{8}\ d$.
$e = \frac{1}{8}\ d$.
$i = 1\frac{7}{8}\ d$.
$l = 1\frac{1}{2}\ d$ to $6\ d$.

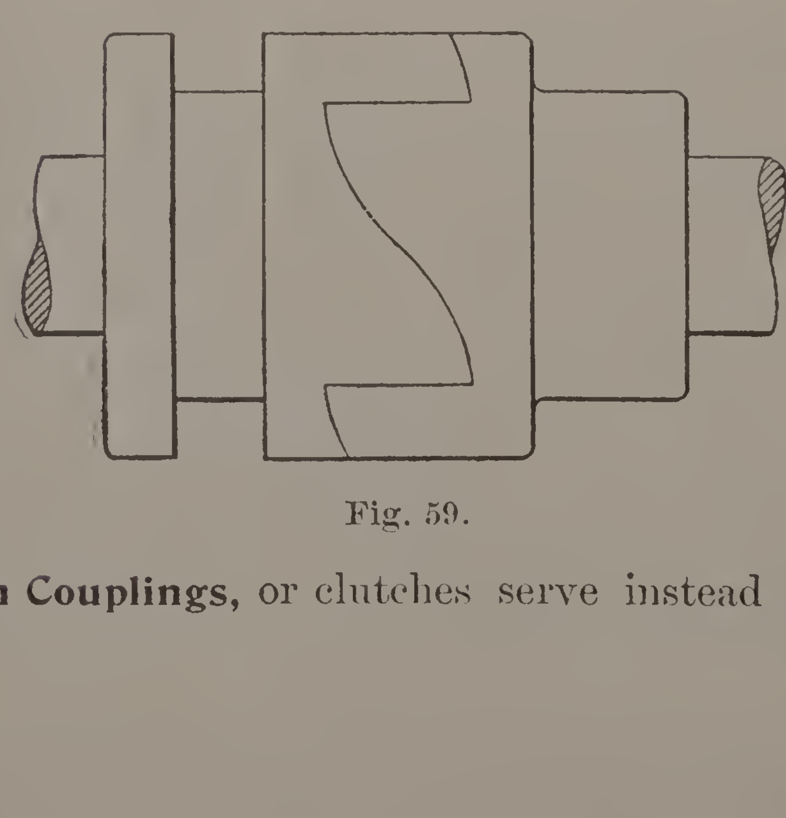


Fig. 59.

Friction Couplings, or clutches serve instead of loose couplings on shafts running at high speeds. Fig. 60 shows a good form of friction clutch. The ring, *e*, is keyed to the shaft *t*. This ring is split and fits inside the cylinder *c*, which is keyed to the shaft *i*. The split ends are connected by a screw having

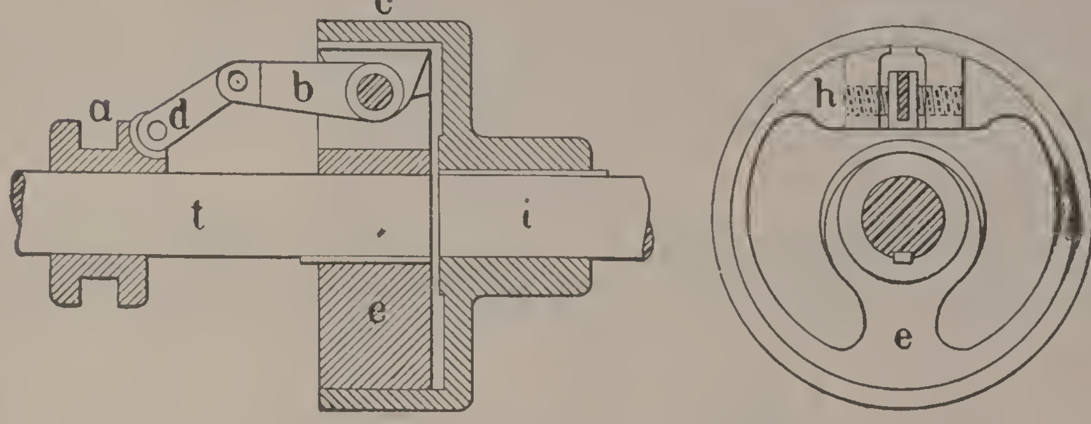


Fig. 60.

right and left hand threads. The link *d*, connects the lever *b*, to the sleeve or collar *a*. The lever *b*, turns the screw. The clutch is readily operated. When the sleeve is pushed toward the cylinder *c*, the rotation of the screw throws the ends *h* of the ring apart, and causes the ring *e* to press firmly against the cylinder *c*.

The proportions for the various parts are about the same as those for Fig. 58. The clutch shown in Fig. 62 is not as good as that of Fig. 60 because it causes an end thrust on the shaft and it is harder to put in gear. It is, however, simple in construction.

Weston Friction Coupling. The friction coupling shown in Fig. 61 is used both as a shaft coupling and for coupling a spur

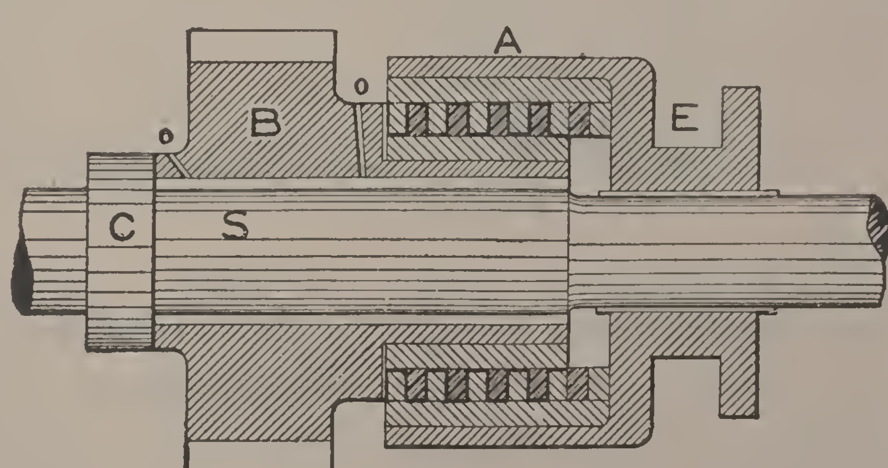


Fig. 61.

wheel to a shaft. The wheel B has a long hub and wrought iron rings or plates which slide on feathers. The clutch box A is fitted on the shaft, slides on feathers, and is moved by a lever working in the groove E. Inside the box A are six feathers upon which are strung alternately wooden and wrought iron rings. If the coupling box A is pressed to the left there is friction at each face between the rings. The wheel B is prevented from moving endwise by the collar C. One of the great advantages of this coupling is that if a sudden load comes on the spur wheel B, the plates merely slip over each other; if rigidly connected, some part would break.

Fig. 62 shows a simple form of friction coupling or clutch. It is used to couple wheels or pulleys to shafts and for loose

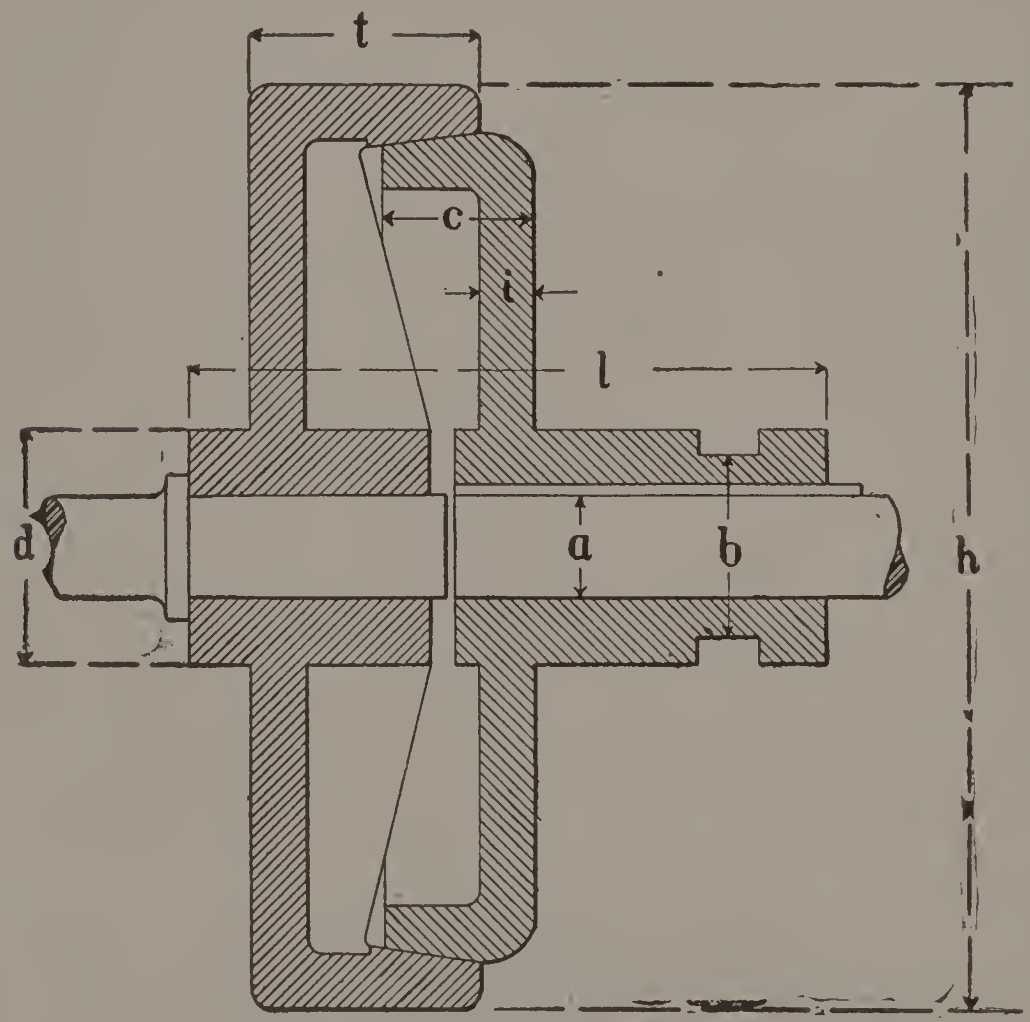


Fig. 62.

couplings for shafts running at high speeds. It consists of a cone keyed rigidly to one shaft and a movable cone sliding on a feather on a second shaft. The movable portion should be placed on the driven shaft so that it will be at rest when out of gear. If the resistance of the driven shaft is considerable the mean cone radius may be three or four times the diameter of the shaft. One great objection to this form is that the horizontal component of the pressure between the conical surfaces causes end thrust on the shaft. The angle of the cone may be from 4 to 10 degrees. The other proportions are as follows:

a = diameter of shaft,
$b = 1\frac{3}{4}\,a$,
$c = 1\frac{1}{2}\,a$,
$d = 2\,a$,
$h = 4a$ to $8a$,
$i = \frac{3}{8}\,a$,
$t = 2a$.

Shifting Gear for Clutches. Forked levers, having prongs which fit into the groove of the clutch, are used to put clutches in

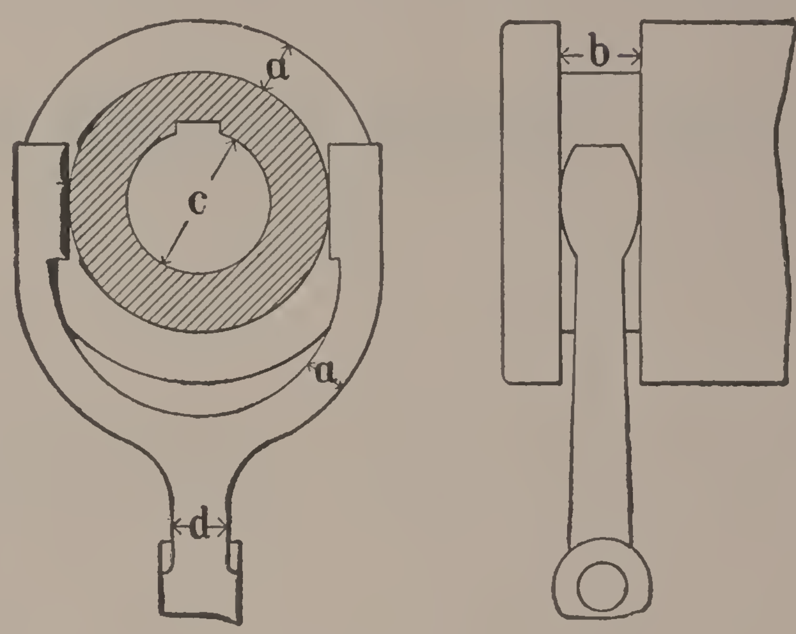


Fig. 63.

and out of gear. The lever is ordinarily worked by hand. Sometimes a brass strap is made to encircle the groove; this increases the wearing surface. The following dimensions refer to Fig. 63; the unit being the diameter of the shaft.

c = the diameter of the shaft,
$a = \frac{5}{16}\,c$,
$b = \frac{1}{2}\,c$,
$d = \frac{3}{8}\,c$.

EXAMINATION PAPER.

MACHINE DESIGN PART I.

MACHINE DESIGN.

Instructions to the Student. Place your name and full address at the head of the paper. Work out in full the examples and problems, showing each step in the work. Mark your answers plainly "Ans." Avoid crowding your work as it leads to errors and shows bad taste. Any cheap, light paper like the sample previously sent you may be used. After completing the work add and sign the following statement.

I hereby certify that the above work is entirely my own.

(Signed)

1. A wheel 42 inches in diameter makes 35 revolutions per minute. What is the linear velocity of a point on the circumference? Ans. 6.4 feet per second.

2. If the angular velocity of a wheel 6 feet in diameter is 30 per second, what is the linear velocity? Ans. 90 feet per second.

3. Define reciprocating motion, continuous motion, and intermittent motion.

4. What is a machine?

5. Why are the relative movements of the parts of a machine independent both of the power transmitted and the size of the parts?

6. Define power and its unit.

7. Why are not foot-pounds an indication of Horse-Power?

8. Name some forms in which potential energy may exist.

9. What is the energy of a body weighing 20 pounds and moving with a velocity of 4 feet per second? Ans. 4.97 foot-pounds.

10. In designing an engine why would you make the piston rod of wrought iron or steel rather than of cast iron?

11. Explain with sketch why all corners of castings should be well rounded.

12. Why are castings weak if some parts are much heavier than others?

13. What elements and processes are employed to get sound steel castings?

14. Name the principal strains in machines and the methods used in designing to allow for these strains.

15. What metals are used for bearing alloys?

16. If you were making a working drawing of an engine crank, how many views would you make?

17. When are sectional views used?

18. Which is the best method of designing, by theory, or practice, or by a consideration of both?

19. Name some part of a machine that is designed from calculation for strength. Some part designed to provide for wear.

20. Describe with sketch the standard U. S. screw or bolt thread.

21. What is the safe working stress on a bolt if the diameter is $1\frac{1}{4}$ inches and the load 3,800 pounds? Ans. 4,273 pounds.

22. Why are multiple threads used when motion is to be transmitted?

23. Describe the standard bolt and nut.

24. Why is the Sellars thread not suited for gas pipe?

25. An engine cylinder-head is bolted to the cylinder with 8 bolts. If the maximum total steam pressure on the piston is 28,000 pounds, what is the diameter of the bolts? Assume safe working stress as 5,000 pounds. Ans. $1\frac{1}{8}$ inches in diameter.

26. What is the pitch of a screw?

27. If a bolt is $1\frac{5}{8}$ inches in external diameter, what is the pitch? Find by formula. Ans. .185 inches.

28. Describe the Whitworth thread.

29. When are taper threads used? What is the advantage of the buttress thread?

30. Describe with sketch some method of fastening foundation bolts.

31. What is a key?

32. Describe the knuckle joint.

33. Is the Grover's spring a good locking device? Why?

34. Describe with sketch what you consider a good locking arrangement for nuts.

35. When are pin keys used?

36. Describe the method of fastening small engine cranks on shafts.

37. Why are keys tapered? About how much is the taper?

38. What is the most effective form of key? Why?

39. Find the length of a steel key which is an inch wide when a 3 inch shaft transmits 50 horse-power at 100 revolutions per minute.

Ans. 3 inches.

40. Find the dimensions of a cotter and rod of the form shown in Fig. 42. The load being 5,500 pounds and $S_t = 7,000$ pounds.

$$\text{Ans.} \begin{cases} d = 1\frac{1}{4} \text{ inches} \\ d_1 = 1 \text{ inch} \\ d_2 = 1\frac{7}{16} \text{ inches} \\ D = 2\frac{1}{2} \text{ inches} \\ t = \frac{5}{16} \text{ inch} \\ b = 1\frac{9}{16} \text{ inches} \\ a = \frac{15}{16} \text{ to } 1\frac{9}{16} \text{ inches.} \end{cases}$$

41. Make a sketch of a locking device for a cotter that has considerable taper.

42. A cotter is 2 inches wide at one end and $2\frac{7}{8}$ at the other. If it is 14 inches long what is the taper per foot?

Ans. $\frac{3}{4}$ inch per foot.

43 Find the length and diameter of a steel end journal when the load is 2,300 pounds. Assume $\frac{l}{d} = 1.75$ and the safe working stress S, as 8,500 pounds. Ans. $1\frac{9}{16}$ in. diam. and $2\frac{3}{4}$ in. long.

44. Find the proportions of a steel end journal when the load is 10,000 pounds, the safe working stress 9,000 pounds, and the allowable pressure 850 pounds. Ans. $2\frac{7}{8} \times 4\frac{1}{8}$ inches.

45. Find the diameter and length of a wrought iron neck journal when $W = 5,000$, $S = 7,500$ and $\frac{l}{d} = 2$.

Ans. $1\frac{5}{16}$ in. diam. and $2\frac{5}{8}$ in. long.

46. What is the height and breadth of collar for the above journal? Ans. $\frac{1}{4}$ in. in height and $\frac{3}{8}$ in. in breadth.

47. What is the diameter of a pivot journal when the load is 25,000 pounds and the allowable pressure is 600 pounds per square inch? Ans. $7\frac{3}{8}$ inches.

48. The end thrust on a 9 inch shaft is 12,000 pounds. Find the diameter of the collars assuming 5 are used.

Ans. $D = 11\frac{1}{2}$ inches.

49. What is the diameter of a mill shaft which transmits 55 horse-power at 90 revolutions per minute? Ans. $3\frac{13}{16}$ inches.

50. Why is a hollow shaft stronger than a solid one of equal weight?

51. Find the outside and inside diameters of a hollow shaft that equals in strength a solid shaft 8 inches in diameter. The inside diameter to be $\frac{1}{2}$ the outside. Ans. $8\frac{3}{16}$ and $4\frac{3}{32}$ inches.

52. Why is the distance between bearings small?

53. Describe with sketch a good simple shaft coupling.

54. Draw a sketch and calculate the dimensions of a clamp coupling like that shown in Fig. 52. The shaft being 4 inches in diameter.

OPTIONAL

For students taking Mechanical Drawing.

Assume convenient scale.

1. Design and draw a knuckle joint having $d = 2$ inches.

2. Design and make two sectional views of a cotter like the one shown in Fig. 42. Assume $d = 1\frac{3}{8}$ inches.

3. Make a drawing of some form of shaft coupling for a 3 inch shaft.

4. Design and make the drawings of a friction coupling for a $3\frac{1}{2}$ inch shaft.

5. Design and draw a Sellar's Cone coupling for a $2\frac{1}{4}$ inch shaft. Two views.