obtained by winding plaits of hemp, soaked in tallow, round the flange of the piston. These plaits were secured in place and packed tightly together by a ring, called the junk ring, which was bolted down to one face of the piston. Such a system would be quite inapplicable to modern high-pressure engines. One of the simplest and best modes of packing pistons of moderate dimensions was devised by Mr. Ramsbottom; it is illustrated in fig. 53. The flange of the piston *b* (fig. 52) is broad enough to allow of three rectangular channels being turned in it. Into each of these channels or grooves is placed a steel ring of rectangular section, as shown in the figure. These rings are not bent into an exact circle of the same diameter as the piston, but are so formed that they have a tendency to spring outwards, and consequently, when the piston is confined in the cylinder, they press against the working surface of the latter, and thus effect a steam-tight joint. Fig. 53 shows one of the rings, before it has been sprung into its place. The inner complete circle represents the diameter of the cylinder. The three rings are so arranged that the joints are not in a straight line. It has been found that if the spring on the rings is sufficient to cause them to press against the cylinder with a pressure of 3 lbs. to the square inch, the joint will be steam-tight.

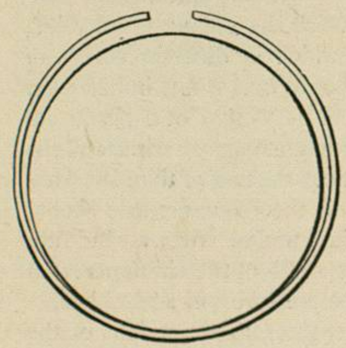

Fig. 53.

It is not necessary to make the rings of steel. In many cases they are made of cast iron, the ring when cast being slotted across. The outer face of the ring is turned, and the inner face left as it comes from the mould. This insures a good amount of spring.

There are many varieties of metallic packing rings in use. In some cases they are made without any initial outward spring of their own, but are pressed outwards by the action of the steam, which is admitted behind the ring by small holes in the body of the piston. In other cases the metallic packing, instead of being a simple ring, is made in the form of a spiral which makes rather more than two complete turns round the circumference of the piston, the groove in the latter being of course formed to correspond. In this manner the butt joint of the simple ring is avoided, and the leakage of steam through this joint is prevented.

In the case of marine engines, the packing ring is usually pressed against the barrel of the cylinder by means of a

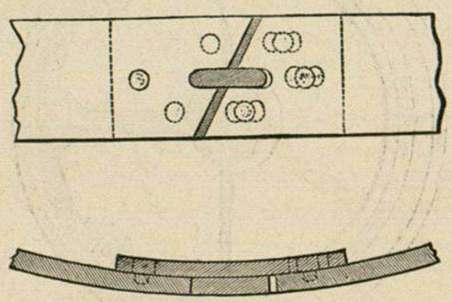

Fig. 54.

series of independent adjustable springs contained within the body of the piston. Figs. 54 and 55 [1] show the details of such an arrangement. The spring ring, which is of considerable depth, is held up against the sides of the cylinder by a series of steel springs, *aaa*, fig. 55. The joint in the ring is formed as in fig. 54 to prevent leakage. An oblique slot is taken out of the ring. A plate, fitted with a tongue piece, is fastened behind the slot, as shown in the section, fig. 54, and the tongue piece, which slides in a groove, allows the ring to expand and contract, and at the same

[1] Taken by permission from Mr. R. Sennett's work, *The Marine Steam Engine.*

time makes a steam-tight joint. The ring, with its springs, is covered by a flat circular piece of iron called the junk ring, which is shown in plan on one half of fig. 55. This enables the springs to be got at easily for examination and repair. The junk ring is attached to the body of the piston by bolts which work into brass nuts embedded in the metal of the piston. When the threads work loose, the nuts can be easily replaced. In the case of horizontal marine engines the part of the spring ring which is in contact with the lower

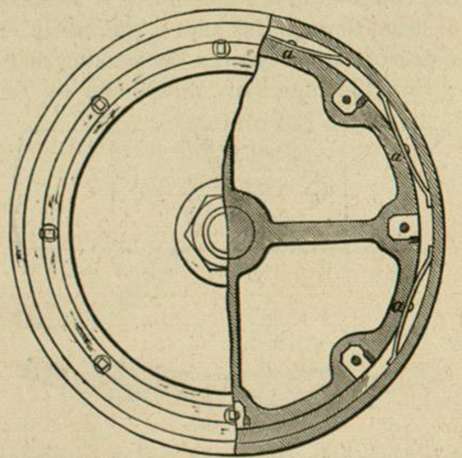

Fig. 55.

side of the cylinder is not backed by springs, but by solid blocks. These are necessary in order to support the weight of the piston, as otherwise the springs at the bottom would be abnormally compressed and those at the upper part of the circumference correspondingly slack. Within the last two or three years it has been possible, owing to the improvements introduced into the methods of casting steel, to manufacture pistons of that metal instead of cast iron, which is the material which has hitherto been almost invariably used. In marine engine pistons, it has been found possible

by the use of cast steel to effect a saving in weight of between thirty and forty per cent. as compared with cast iron.

Piston rods.—The piston rod is the member which transmits the motion imparted to the piston to the mechanism outside the cylinder. It consists of a truly cylindrical bar of wrought iron or steel, one end of which is fastened securely into the piston. The rod passes through the cylinder cover by means of a steam-tight stuffing-box, as shown in fig. 48, and the outer end terminates in the cross-head which will be described presently. There are various methods in vogue of fastening the rod into the body of the piston. Sometimes the end of the rod is turned cylindrical and a hole bored in the piston slightly less in diameter than the rod. The piston is then heated, which causes it to expand, when the rod can be inserted. After cooling, the piston contracts, and holds the rod firmly in its place. In the majority of cases the end of the rod is turned conical as in fig. 52, with a screw thread on the extreme end, by means of which, together with a nut, the rod is firmly embedded in a conical recess bored in the piston. Sometimes instead of a nut the rod is fastened by means of a cotter, and occasionally it is screwed and pinned into the boss of the piston.

The strength of piston rods has to be fixed with special reference to the fact that they are subject to alternating strains. When the piston is making the stroke towards the crank shaft the rod is in compression, and when making the return stroke the rod is in tension. The maximum stress per square inch of cross section at any part of the stroke is equal to the total pressure of the steam on the piston divided by the area of the rod. It is usual in designing pieces of machinery which have to bear alternating strains of tension and compression to make them much stronger than would be necessary were the strain always of one sort.

Cross-heads and slide-bars.—The outer end of the piston rod is attached to the cross-head, or motion block, which serves the double purpose of forming the means of connection between the piston rod and the connecting rod, and of guiding the piston rod so as to keep it straight and in the line of the axis of the cylinder, in spite of the bending moment due to the angular position of the connecting rod.

The cross-head generally consists of three principal parts, viz. (1) the body which often contains a conical hole into which the coned end of the piston rod is fastened by means of a cotter; (2) the part by which the joint with the connecting rod is effected; and (3) the guides or motion blocks which travel between fixed bars parallel to the axis of the cylinder, called slide-bars, and which prevent the end of the piston rod from being deflected as the connecting rod assumes an angular position. Fig. 56 shows a piston rod, cross-head, slide-bars, and connecting rod in position.

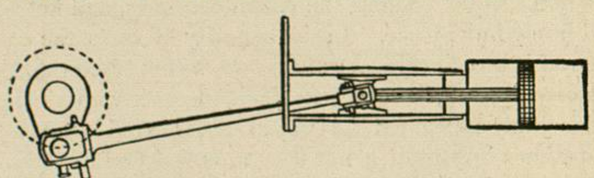

Fig. 56.

Pressure upon slide-bars.—Before giving examples of cross-heads, it is necessary to explain the nature of the strains which they transmit to the slide-bars. As the crank-pin revolves, the connecting rod assumes an angular position, the amount of the angularity increasing till the crank is at right angles to the axis of the cylinder and then again diminishing to nothing.

In the accompanying diagram (fig. 57) suppose the piston to be making a stroke towards the crank axle, the crank-pin revolving in the direction of the arrow. In this case the piston and connecting rods are both in a state of compression, and consequently there is a downward resulting force acting at the cross-head, which is opposed by the resistance of the slide-bar. Next take the return stroke, the connecting rod being in the position shown by the dotted line. In this case

both piston and connecting rods are in a state of tension, and consequently we have again a downward resultant force acting at the cross-head. Therefore so long as the engine runs in the direction shown by the circular arrow the pressure is always downwards. Similarly if the engine were reversed so as to run in the opposite direction, there would always be an upward resultant pressure acting at the cross-head and we should require a top slide-bar in order to prevent this pressure from deflecting the end of the piston rod. If the steam ceased to act on the piston during a portion of the stroke, and if at the same time the valves were so set that there should be considerable compression or back pressure on the other face of the piston, the piston rod might cease to be in a state of

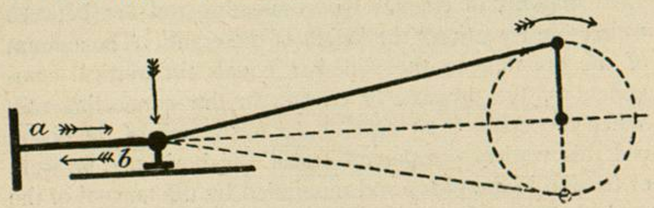

Fig. 57.

compression (or tension as the case might be) at a certain point in the stroke, and the above reasoning would not apply. In such a case we might have a downward pressure on the slide-bars during one portion of the stroke and an upward pressure during the remainder (or *vice versâ*), and thus two slide-bars are often necessary even in factory engines which are usually not reversed. Of course in the case of locomotives, marine and winding engines, which are constantly reversing, two slide-bars or some corresponding arrangement are an absolute necessity; but in factory engines if the valves are so set that there is no fear of a reversal of the direction of stress on the piston, one slide-bar is often sufficient. Of course in such cases the engine is arranged to run so that the pressure may act downwards, as it is much easier to keep the working face of the bottom bar lubricated than the corresponding top face.

The amount of the pressure on the slide-bar depends upon the angle (a) made by the axis of the connecting rod with the axis of the piston rod. The greater the angle (other things being equal) the greater the pressure. The size of the angle depends on two things: first, the position of the crank, the angle constantly increasing till the crank is at right angles to the axis of the piston rod, and then again diminishing; and second, the relative lengths of the crank arm and the connecting rod; the shorter the connecting rod, as compared with the crank arm, the greater will be the value of a for a given position of the crank. In the great majority of engines the connecting rod lies between 6 times and 3·5 times the length of the crank. The amount of the pressure on the slide-bar equals the vertical component of the pressure or tension in the connecting rod. When the crank is at right angles to the axis of the piston rod this vertical component equals the horizontal pressure or tension in the piston rod multiplied by the tangent of the angle between the piston and connecting rods. With the proportions of length of connecting rod to length of crank arm usually adopted, this angle is so small that there is no material difference between its sine and its tangent. Hence the usual rule is: Maximum pressure on slide-bar = pressure on piston × sine of angle between piston and connecting rods = pressure on piston × crank radius ÷ length of connecting rod.

We are only concerned to know the maximum pressure because the slide bar must be made wide enough to bear this pressure with safety. The safe amount of working pressure per square inch of slide-bar surface varies with the material of which the bar is constructed. In locomotives with steel or case-hardened wrought-iron bars the pressure is sometimes as high as 120 lbs. per square inch. This is, however, too high a pressure, and occasions very rapid wear

of both bars and motion blocks. In the best modern practice the width of bars and areas of motion blocks are so chosen that the pressure shall not exceed 40 lbs. per square inch when the bearing surfaces are of cast iron, and half as much again when they are of steel. The actual pressure acting on the piston rod when the crank arm is at right angles to the axis of the cylinder must be calculated from diagrams constructed as explained in Chapter V. page 194; it can only be taken from the indicator diagrams when the piston speed is very moderate.

The cross-head should be so designed that the point at which the vertical pressure is transmitted by the connecting rod shall be exactly over the centre of the guide or motion block, otherwise an undue proportion of pressure will be transmitted to the edge of the block nearest to the point

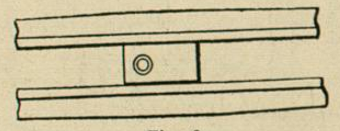

Fig. 58.

where the pressure is applied. Fig. 58 is an illustration of a cross-head in which this consideration has not been attended to.

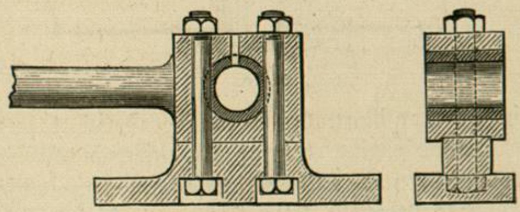

Fig. 59.

Fig. 59 shows a longitudinal and transverse section of a type of cross-head commonly in use in factory engines which require only one slide-bar.

Fig. 60 is an example of a cross-head provided with but one slide-bar. The motion block is so arranged as to envelope the bar, and the pressure comes on the upper or lower surface of this latter, according to the direction in which the engine is running.

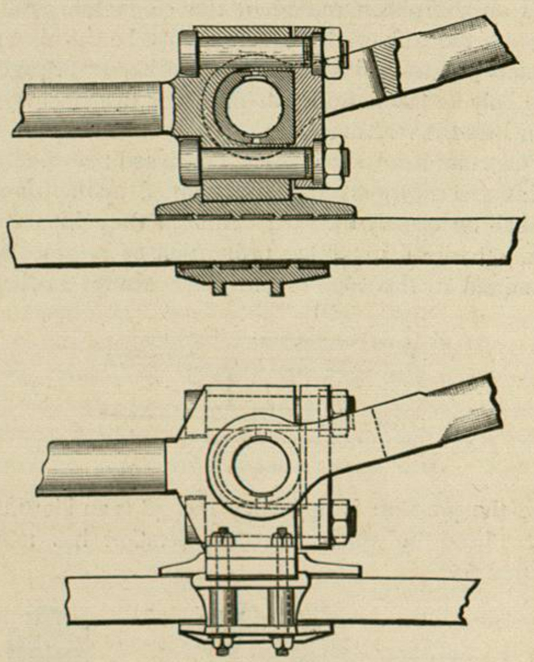

Fig. 60.

Fig. 61 is an illustration of the cross-head of the largest type of marine engine showing also the ends of the piston and connecting rods, and sections of the large slides and massive slide bars, the latter being attached to the main frames of the engines. Specimens of cross-head and slide bars are also given in pages 458 to 461.

In the older types of engines slide-bars were not used. The end of the piston rod was kept straight by means of a

parallel motion, which is an arrangement of link-work so contrived that a pressure is always brought to bear on the end of the rod equal in amount, but opposite in direction to the pressure due to the angular position of the connecting rod. Parallel motions are now seldom used for this purpose, and need not therefore be described in this volume. A

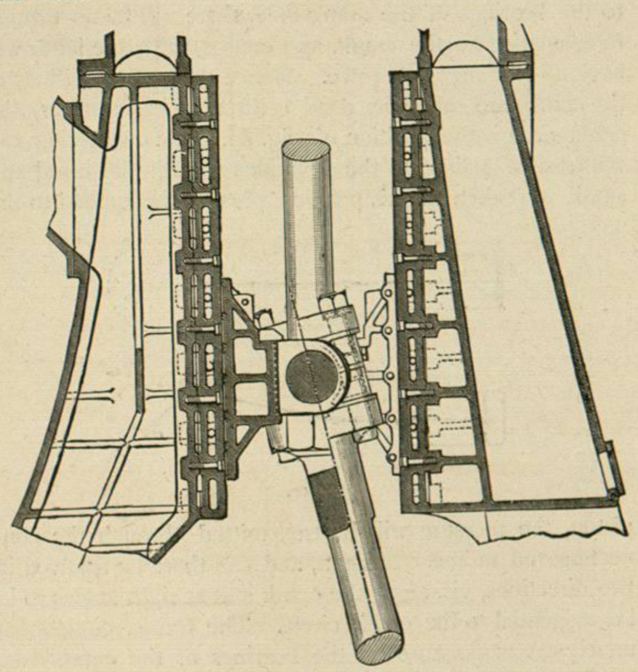

Fig. 61.

simple form of parallel motion, as used in the construction of indicators, is illustrated on page 319.

Connecting rods and cranks.—The connecting rod is the link which enables the reciprocating rectilinear motion of the piston to be converted into the circular motion of the crank pin. It is a link or rod of metal so formed at the two ends that it can be jointed to both the cross-head and