Radial paddle-wheels.—In paddle-wheels with radial floats only the float for the time at the bottom of the wheel is vertical, and giving direct sternward velocity to the water. All the others act obliquely and have a vertical as well as a horizontal reaction, the former, although absorbing a large proportion of the power of the engines, being wasted, so far as propulsive effect is concerned. The greater the immersion of the wheel, the greater will be the loss due to the vertical component of the pressure on the float. These wheels, therefore, should be so designed that at the maximum load draught of the ship, they should not be immersed more than one-quarter the diameter of the wheel; for beyond this limit the loss from the vertical reaction increases at a very rapid rate.

With these propellers it is impossible to determine, with accuracy, the area and speed of the race. Various assumptions may be made, but they are now of little practical value, as the use of the feathering paddle is almost universal for important vessels in which this means of propulsion is employed.

Objections to paddle-wheels.—The chief objection to the employment of paddle-wheels for ocean navigation arises from the practical difficulties attending the variation in immersion during long voyages, owing to the lightening of the ship by the consumption of coal, stores, &c. Even with feathering wheels, in which the floats are approximately vertical when in the water, the loss from forcing the floats in and out of the water, churning, &c., is much increased when the wheels are deeply immersed; and it is evident that if the wheels are to be sufficiently immersed at the end of a long voyage they must have been too deep in the water on starting. Even if water ballast be used to overcome this difficulty a larger expenditure of engine power would be necessary. For short voyages, in which the draught of water is comparatively unchanged, paddle-wheels may be advantageously employed, and they are almost essential for propulsion in many shallow rivers, where the depth of water is insufficient to admit the use of screws.

For ocean navigation, however, they have been superseded by the screw, for in addition to the loss of efficiency from alteration in draught, paddle-wheels are objectionable in consequence of the racing and straining of the machinery due to the rolling motion in a seaway, causing the paddles to often emerge from the water on one side and be correspondingly depressed on the other. For vessels of war the exposure of the paddles and engines to injury by shot and their interference with the deck arrangements render them doubly inadmissible. The paddle-wheel, as a propeller, is now only employed in the few special cases suitable to this form of propeller.

Screw-propeller.—The action of a screw-propeller is much more complex than that of the two types of propellers previously discussed, and is due mainly to the following causes:—

1. The action of the propeller on the water is oblique instead of direct.

2. The velocities of the several particles of water acted on by the screw are different from each other.

3. The screw acts on water that has previously been set in motion by the ship.

The difficulties attending an exact mathematical investigation of the action of screw-propellers are consequently so great that the problem is not yet solved. No formula yet in existence will give an accurate value for the thrust of an ordinary screw, even when working in undisturbed water, while for the case of propellers working behind actual ships the problem is still more difficult and complicated. The principles involved may, however, be easily understood. Oblique action, other things being equal, is always a cause of loss of efficiency in a propeller, and the fact that, in spite of this, the screw is a practically efficient propeller is explained by the circumstance that it operates upon a much greater quantity of water than could be acted on by a pair of paddle floats, or by any other propeller in a ship of the same size in the same time, and, as has been proved previously, the efficiency of any propeller depends to a large extent on the quantity of water acted on. The screw is the form of propeller best adapted to fulfil this condition.

It will now be desirable to define a few of the technical terms that will frequently be used.

Diameter.—The diameter of the screw is the diameter of the circle formed by the tips of the blades when revolving. The area of this circle is called the 'disc area' of the screw.

Pitch.—The pitch of the screw is the distance through which the screw would advance *in one revolution* provided it revolved in an unyielding medium such as a solid nut.

Speed of screw.—The speed of the screw is the distance it would advance *in a unit of time*, supposing the screw to be working in a solid nut. This is obviously equal to the pitch of the screw multiplied by the number of revolutions made per unit of time.

Slip.—In consequence of the screw-propeller working in a yielding medium, the speed of the ship is generally less than the speed of the screw. The difference between the speed of the screw and the speed of the ship is called the slip of the screw.

If v = speed of the screw.

V = " " ship.

$v - V$ = slip of the screw.

$\frac{v - V}{v}$ = slip of the screw expressed as a fraction of the speed of the screw . . . (2)

$\frac{v - V}{v} \times 100$ = percentage of slip.

This, however, is only the *apparent* slip of the screw. It assumes the screw to be acting on water previously at rest, which can never be the case with the water operated on by the screw-propeller of a ship. The friction of the ship on the water during its passage causes a wake to follow her, so that the screw-propeller acts on water already set in motion. The velocity of this stream must therefore be considered, in order to obtain the *real slip*, which represents the true value of the backward velocity impressed on the water by the propeller. The speed at which the water follows the ship depends on her form, and is difficult to ascertain, so that the slip generally referred to is the apparent slip only and not the real slip.

If we assume the water to be a stream of velocity μ and of sufficient breadth, the original velocity of the water acted on, relatively neglected, conditions which are far from being even nearly obtained in practice. It is assumed, for example, that the water enters the orifices at speed V, and that its velocity is *gradually* accelerated up to the speed v, so that all the energy in the supply-water to the pump is utilised. In practice much of this is lost, and in some designs practically all. The smaller is v for a given value of V, the greater is the efficiency.

From the formula (1) we see that the thrust mainly depends on the product A v, and that the smaller the value of v the greater must be the value of A for the same thrust. Since the efficiency becomes a maximum when $v = V$, which is the smallest value v can have, it follows that theoretically the larger A is made the more efficient would be the performance. Generally, A is made as large as practical considerations will admit, so as to keep v, the speed of the race, as small as possible; the sternward momentum of the race with respect to still water representing a loss, and the higher its velocity the greater will be the loss from shock, &c. With reference to screw-propellers, as we shall see in a later portion of this chapter, this principle requires modification.

Advantages and disadvantages of water-jet propulsion.—The advantages claimed for water-jet propulsion in warships consist in the freedom from damage in action by wreckage or grounding, greater control of motion of the ship from deck without altering the motion of the engines, and possession of large pumping power in case of a leak.

In practice they are at a great disadvantage, owing to the magnitude of the frictional resistances and the difficulty of operating on a sufficiently large body of water on this plan; and instead of being more efficient than other propellers, as they should be theoretically, they are in practice much less efficient. Their defective action therefore, due to the resistance of passages, &c., combined with the practical objections to the fitting of large orifices in the ship's side, either above or below the water, places the jet out of the region of practical propellers, except under very special circumstances, such as in lifeboats, &c.

Results of trials.—The 'Waterwitch' is the only example of a ship with a jet-propeller in the Royal Navy, and her trials demonstrated the inefficiency of the system. With 760 I.H.P. the 'Waterwitch' attained a speed of 9·3 knots, displacement 1,160 tons. The 'Viper,' twin-screw gun-vessel, of somewhat similar dimensions but inferior in form, attained a speed of 9·6 knots with 696 I.H.P., displacement 1,180 tons. The quantities of water acted on by the two kinds of propellers were very different. In the 'Waterwitch' the quantity of water passing astern per second was about 150 cubic feet, while the twin screws of the 'Viper' acted on over 2,000 cubic feet per second, or about fourteen times as much.

In 1883 one of the second-class torpedo boats by Messrs. Thornycroft & Co. was fitted with a turbine, and great skill and care were exercised to insure the best results. Efficiency of astern working was subordinated to that of ahead working, and the water inlet was placed at the bottom, and made into a scoop to utilise as much as possible of the energy of the entering water. All sudden changes of angles and velocity of water were avoided. On the comparative trials between the hydraulic boat and the other boats of equal size fitted with screw-propellers, it was found that the speed of the hydraulic boat was no greater than could be attained in the screw boats with about half the power. The actual results were: Hydraulic boat, I.H.P. 167, speed 12·6 knots. Screw boat, I.H.P. 170, speed 17·3 knots. The efficiencies were analysed as follows: Screw boat: Engine, ·77; screw propeller, ·65; total efficiency, ·5. Hydraulic boat: Engine, ·77; jet-propeller, ·71; circulating pump, ·46; total efficiency, ·251. The screw boat was therefore nearly double as efficient as the hydraulic, the principal loss in the latter being in the pump.

Feathering paddle-wheels.—With feathering paddle-wheels the floats are supposed to act in a direct sternward direction, and to enter and leave the water normally, the area of the race of both paddles being equal to that of a pair of floats. In ordinary radial paddle-wheels there is much local agitation and disturbance of the water, due to their oblique action on entering and leaving, which complicates the question, and the area of the race is not so clearly defined.

Before being acted on by the floats the water is assumed to be at rest, and therefore, relatively to the ship, it would have a sternward velocity V, equal to the speed of the ship.

Let v be the final velocity of the race *relatively* to the ship, and $A =$ the area of a pair of floats, one on each side of the ship, then the sternward momentum generated per second is equal to R, the propelling reaction, or $R = \frac{W}{g} A\, v\, (v - V)$, as in the jet-propeller.

The propelling reaction R acts on the paddle floats, and the velocity of the floats is assumed to be v, the same as that of the propeller race relatively to the ship; therefore the engine has to overcome a resistance R through a space v in one second. Hence, as regards the efficiency of the paddles the energy exerted in propelling is $= R\, v$, and the useful work done is clearly $= R\, V$.

Therefore the total work wasted is equal to

$$R\,(v - V),$$

and the efficiency is

$$\frac{R\,V}{R\,v} \text{ or } \frac{V}{v},$$

which is theoretically not so great as in the jet-propeller, other things being equal.

The work wasted in producing the race is equal to the half vis-viva generated

$$= \frac{W}{2g} A\, v\, (v - V)^2 = \tfrac{1}{2} R\,(v - V).$$

This, therefore, only amounts to one-half the total power wasted, the remainder being absorbed in producing the violent churning and agitation of the water which is always produced by paddle-wheels. In practice the loss from this cause would be even more than one-half of the total power wasted, for in the above investigation we have neglected the resistance to forcing the floats in and out of the water, which considerably increases the work of the engine.

The expression $\frac{V}{v}$ must therefore be regarded as the maximum possible efficiency with these propellers.

consumption is practically unaltered is considerable, and this being so it will generally be advantageous to select the higher limit of speed.

When the steaming distance of the ship for a given quantity of coal is under consideration, the amount of coal necessary to be expended for purposes other than the main engines must be taken into account. This is done by drawing a line O′ X′ at a distance below O X equal to the consumption per day for auxiliary purposes, such as electric lighting, culinary purposes, distilling drinking water, &c. The most economical speed of the ship is therefore obtained by drawing a tangent from O′ to the curve meeting it at B, which gives a greater speed than if the consumption for auxiliary purposes were not taken account of. The reason why the slowest speed is not the most economical will be understood from what has gone before; it lies in the increased proportionate waste by constant friction as the power is diminished, and the proportionately increased loss by radiation from boilers, steam pipes, &c., and the greater proportionate consumption of coal for the constant auxiliary services of the ship.

Amount of consumption for auxiliary purposes.—The consumption of coal on board warships for auxiliary purposes—i.e. for purposes other than propelling the vessel—is very considerable. This service includes the consumption for culinary purposes, warming ship, distilling, electric lighting, working guns, &c., and often amounts during the whole of the commission of a warship to much more than that expended in propelling the vessel, owing to the intermittent nature of the steaming generally required. The following amounts are the consumptions of coal for auxiliary purposes in modern war vessels: Large 1st class battle ships, 10 to 15 tons; 1st class cruisers, 8 to 15 tons; 2nd class cruisers, 5 to 8 tons; and 3rd class cruisers, $3\frac{1}{2}$ to 6 tons. The actual amount depends on the number and extent of the auxiliary machinery fitted, and tends to increase in the newer ships owing to the larger quantity of fresh water used, extensive electric light installations, fitting of refrigerators, &c.

The coal required to run one 300 ampère dynamo for 24 hours at full power has been found to average $2\frac{1}{2}$ tons in battle ships of British Navy, but even at this rate the cost is much less than if candles or oil were used.

Closed exhaust system.—In the newer ships of British Navy arrangements are made so that the heat of the considerable quantity of steam used for the auxiliary engines is not wasted in the auxiliary condenser, but utilised in the evaporators to make fresh water for the boilers and ship's company. A connection is made between the coils of the evaporators and the auxiliary exhaust pipe for this purpose, the valve on the exhaust pipe at the auxiliary condenser being closed. By this means a pressure of 10 to 15 lbs. can be maintained in the auxiliary exhaust system, the steam being condensed in the evaporator coils, producing fresh water. Piston relief valves are also fitted so that any excess of exhaust steam beyond that required for evaporation passes into the auxiliary condenser when in harbour, or into the L.P. receiver when the main engines are under way. By this means a saving is effected; the loss due to the back pressure on the auxiliary engines is more than counterbalanced by the fresh water obtained.

CHAPTER XXIV.

PADDLE-WHEELS.

Radial paddle-wheel.—The simplest form of paddle-wheel is generally known as the common or radial paddle-wheel. In this wheel the floats are bolted direct to the arms of the wheel, and consequently the pressure they produce on the water is always perpendicular to the radius, and the only float that produces a direct sternward reaction is the one at the bottom of the wheel, all the others having a vertical component tending to raise or depress the vessel, which is wasted so far as propulsion is concerned.

Width of floats.—The extreme width of the floats should not exceed one-half the width of the vessel, so that the combined width of the two paddle-wheels should not be greater than the width of the ship. In sea-going steamers the width of float generally does not exceed one-third the width of vessel. In still water the greater the width of float the more effective the wheel, as the required area of race can be obtained with less immersion, and the loss from oblique action is thereby reduced. This condition, however, is limited by the practical difficulties involved in supporting the overhanging end of the paddle-shaft. In rough weather extreme width would be objectionable from many causes.

Immersion of wheels.—The depth of immersion of paddle-wheels is practically limited by the draught of water of the vessel, as it is evidently undesirable to allow the lower edge of the propeller to be below the keel. The immersion of the wheels must also depend on their diameter, for if the floats act too obliquely on entering and leaving the water, a large proportion of the power would be wasted in producing vertical reactions. As an extreme case, we may point out that a radial paddle-wheel immersed to its centre would be of no value as a propeller.

In general the greatest immersion of a paddle-wheel should not exceed one-half the radius, or one-fourth the diameter of the wheel. When sea-going steamers were used for long voyages, the immersion at starting was about one-half the radius, and the mean draught for the voyage about one-third the radius of the wheel.

For effective working, the tops of the floats, when in their lowest position, should always be some distance below the surface of the water. In large sea-going paddle steamers the top of the lowest float was usually about 18 to 20 inches below the surface, at mean draught; in smaller vessels from 12 to 15 inches. In river steamers the immersion is generally much less, say from 3 to 6 inches; but these boats always work in smooth water, and their draught is practically constant. In sea-going steamers the immersion of the floats at their lightest draught should not be less than 6 inches.

Number and pitch of floats.—In radial paddle-wheels the number of floats is generally made equal to the number of feet in the diameter of the wheel, which practically sets them at about 3 feet apart from each other. In some fast ships, to reduce vibration, they have been set closer than this, or from 2 to $2\frac{1}{2}$ feet apart. If the floats be set too closely together the water will not escape with sufficient freedom from between them, whilst if too far apart the vibratory action will be excessive. The number and pitch of floats should be so arranged that there will always be at least three floats immersed at the same time.

Reefing paddle-wheels.—The floats are secured to the radial arms of the paddle-wheels by hook-bolts, in such a manner that if the draught of the vessel be increased, the floats may be readily unshipped and secured in other positions nearer the centre of the wheels. This operation is usually called *reefing the paddle-wheels,* and is equivalent to reducing the effective diameter of the wheel and the immersion of the floats, and thereby diminishing the loss from oblique action. Reefing is desirable when by increased draught it is found that the wheels cannot be driven fast enough to utilise all the steam generated in the boilers. This operation, by decreasing the resistance, enables all the steam generated to be used, and the piston speed increased, with a consequent gain in the power and speed of the ship.

The only points of advantage of the radial over the feathering paddle-wheel are its lightness, simplicity, and cheapness of construction. There are no working parts in it, and defects can be readily made good at little cost. Its propelling efficiency, however, is much less than that of the wheel with feathering floats, and the improvements in design and workmanship have made the latter so practically trustworthy, for the comparatively few services for which paddle-wheels are now required, that the radial paddle-wheel may be regarded as altogether a propeller of the past.

Feathering paddle-wheel.—In order to obviate the disadvantages resulting from the oblique action of the floats of radial paddle-wheels, especially in cases where the draught of the vessel varied considerably, feathering paddle-wheels have been introduced. The general form and arrangement of these propellers are shown in Figs. 284 and 285. The wheel consists of a wrought-iron framework, secured to a strong cast-iron centre or boss, keyed on the end of the paddle-shaft. The floats, instead of being fixed to the arms of the wheel, are carried on joint-pins, and their motion is controlled by the action of an eccentric, through rods and levers, in such a manner as to keep the floats approximately normal to the effective surface during their passage through the water, so that the whole of the thrust will be in a sternward direction. Its efficiency is at least 10 per cent. greater than that of the radial paddle-wheel when both work under suitable conditions, and the economy and efficiency resulting from its use far more than compensate for its increased first cost and expense of maintenance.

It is however more complicated, and requires more care and attention, than the radial wheel. It is very important that the working parts should be sufficiently strong to withstand the shocks to which

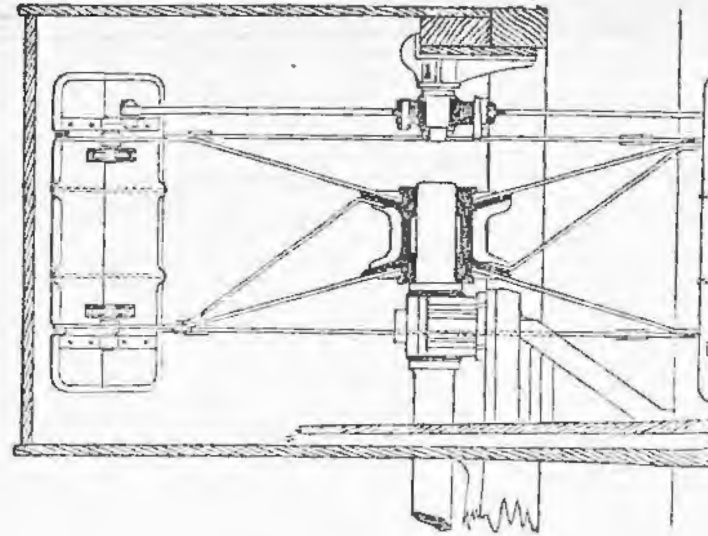

Fig. 285.

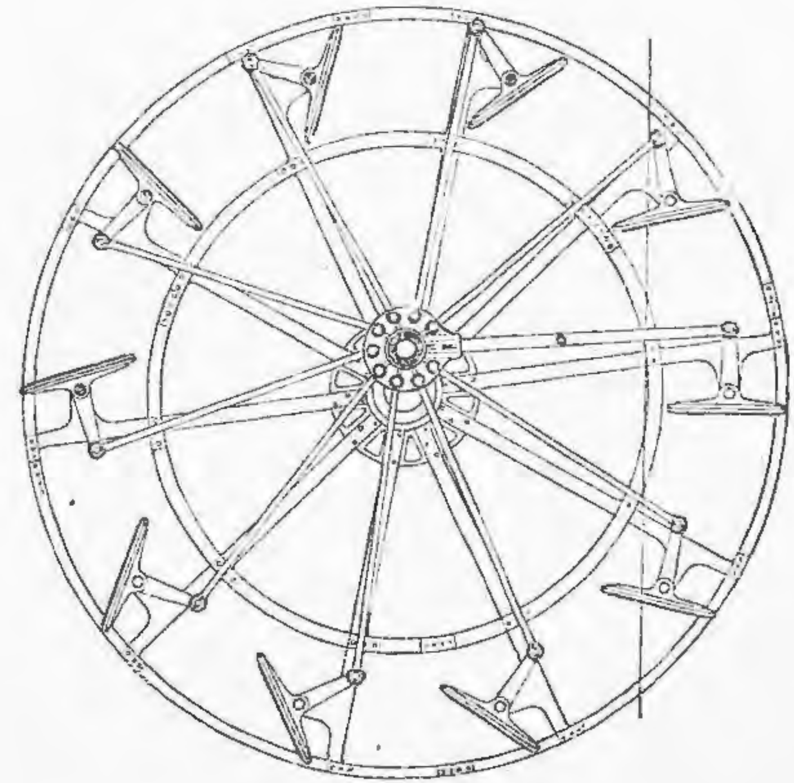

Fig. 284.

they are exposed, without undue straining, for damage to any part of the feathering apparatus is liable to paralyse the action of the entire wheel. These wheels are consequently made much heavier than the radial wheel, and are more difficult to properly support.

This complication and liability to serious injury might possibly have tended to prevent their being extensively used for long sea voyages, in preference to the simpler radial wheels, which, if damaged, could be so much more easily repaired. As, however, the paddle-wheel for ocean navigation has been entirely superseded by the screw-propeller, this point need not be further discussed, and there can be no doubt that for short voyages, river navigation, and towing purposes, for which alone paddle-wheels are now used, feathering floats possess very great advantages, enabling the wheels to be made of less diameter and width, and in consequence of their increased efficiency the indicated horse-power of the engines may be proportionately reduced for a given speed.

Dimensions and pitch of floats.—The floats in feathering paddle-wheels are generally placed about twice as far apart as the floats in the radial wheel; that is, the pitch of the floats is usually about six feet. They are also made deeper, say about twice the depth of the common float, for in this case the area of the race, or stream driven back on either side of the ship, is equal to the width multiplied by the depth of the float instead of the width of float multiplied by the depth of immersion, as is assumed to be the case with the radial paddle-wheel.

Eccentricity of feathering apparatus.—The method of determining the throw and position of the eccentric necessary to produce the proper action of the floats in the water may be easily explained by means of a skeleton diagram. In Fig. 286 let A represent the centre of the paddle-shaft, and K the centre of the eccentric pin or sheave that produces the necessary movement of the paddle-floats, the correct position of which is required to be found. For simplicity, the floats are supposed to be jointed at their centres. In practice this is not exactly the case, the joint being just behind the float, and as close to it as possible. In an actual design, this would render necessary a slight modification in the details of the following method of determining the eccentricity, but the deviation is small, and there will be no difficulty in making the required correction when the principles involved are understood. The circle B C D E F G, drawn with A as centre, through the centres, or joints, of the floats, may be taken to represent the paddle-wheel circle. Let W W represent the water-line, B and D being the points in which it is cut by the paddle-wheel circle.

Consider three floats in the positions shown by B, C, and D, one just entering the water, the second at its lowest point, and the third just leaving the water. In order that the motion of the floats through the water should be correct, moving as nearly as possible edgewise, relatively to the water in the paddle race, the directions of the faces of these floats produced, should meet at the point F, at the top of the paddle-wheel circle. If, therefore, from F, the highest point of the circle, straight lines, F B, F C, and F D are drawn, these will represent the directions of the faces of the paddle floats at these respective points.

From the centres of these three floats, B, C, and D, draw the float-levers, B*b*, C*c*, D*d*. These are usually at right angles to the float, and their lengths are about three-fifths of the depth of float. These values are

arbitrary, and subject to convenience in any particular design; but the angle seldom deviates much from a right angle, and the proportionate length of lever given above is generally suitable. Having thus determined the points, *b*, *c*, and *d*, to which the radius rods from the eccentric have to be jointed, it is only necessary to find by plane geometry the centre, K, of the circle passing through them. K will then be the centre, and A K the throw, of the eccentric necessary to produce the required motion of the floats.

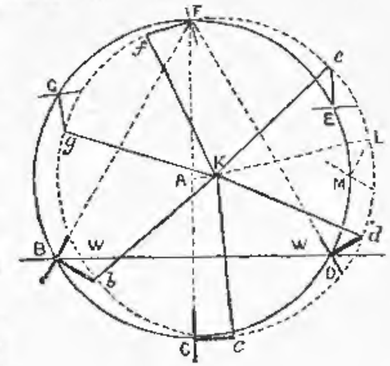

FIG. 286.

The velocity of the propeller race is clearly represented by the circumferential velocity of the circle B D F G, and the effect of the motion produced by the action of the eccentric, thus determined, will be to cause the floats, while *in the water*, to move as nearly as possible edgewise, relatively to the propeller race, and thus prevent loss from oblique motion. By drawing floats in other positions, it will be seen that their action when *out of the water* is far from being free from vertical reactions, but these, operating only on the air, may be neglected.

Paddle-shaft bearings.—The shaft carrying the paddle-wheel is called the paddle-shaft, and is sometimes supported by two bearings, one on the ship's side, and the other on a beam, called the *sponson* or *spring beam*, on the outside of the paddle-box. In this case, the feathering apparatus has to be worked by a large eccentric on the paddle-shaft, to which the radius rods are attached.

Overhung wheels.—The most general arrangement, however, is that shown in Figs. 284 and 285, in which the paddle-wheel is overhung and supported by a single bearing on the ship's side, the outer bearing being dispensed with. In this case the feathering motion is produced by attaching the radius rods to a sheave working on a pin carried by a bracket fixed to the outer side of the paddle-box, in the proper position, eccentric to the wheel, to produce the required movements of the floats.

Driving and radius rods.—In the feathering apparatus, one of the guide or radius rods, called the driving rod, is rigidly fixed to the eccentric, to make it rotate about the axis K. The remainder of the rods are simply jointed to the eccentric, as well as to the float-levers, with pins. In Fig. 284 the driving rod is marked D. All the joints in the feathering apparatus should be bushed either with gunmetal, white metal, or lignum-vitæ.

Details of paddle bearings.—The outer bearings of paddle-wheels, when they are so fitted, cannot be examined when the engines are at work. Guide-boards or troughs are therefore fitted on the side of the paddle-box, so that the water carried up by the wheel is caused to constantly run on these bearings to prevent their overheating. This splashing and churning action of the wheel on the water is also often utilised for the purpose of keeping a small tank, fitted inside the paddle-box, always full of water, to be used, if necessary, on the bearings of the paddle and intermediate shafts which are above the water-line. The water-service pipes for these journals are also, in general, connected with the delivery-pipe from one of the auxiliary pumping engines of the ship.

When the paddle-wheels are overhung, and carried by a single bearing on the ship's side, the journal should be made of larger diameter, and considerably longer than is necessary when an outside bearing is fitted, to resist the increased pressure and strains. There should also be thrust collars on the journal, to prevent end motion when the ship rolls. The bearings for paddle-shafts in the Royal Navy are generally made of gunmetal, though they are sometimes made of lignum-vitæ strips, as in the case of bearings for screw-shafts. When so fitted the shafts should be cased with gunmetal.

Stuffing-box on ship's side.—The hole in the ship's side through which the paddle-shaft passes is either fitted with an ordinary stuffing-box, or covered with a leather disc to prevent the passage into the ship of water carried round with the paddle-wheel.

Disconnecting apparatus.—In paddle-wheel tug-boats, gear is usually fitted to enable the wheels to be disconnected from each other, and each engine worked independently, to facilitate the manœuvring of the vessel. In many cases an ordinary disconnecting clutch is fitted on the intermediate shaft for this purpose.

Another plan consists in fitting a cast-iron disc on the intermediate shaft, in lieu of a crank-arm. This is driven by feathers on the shaft, over which it may be drawn back, clear from the crank-pin, when the engines are required to be worked independently. Engines of this class, of large power, should be fitted with auxiliary steam starting-engines and starting-valves to facilitate handling.

In the more recent paddle-wheel tug-boats in Her Majesty's service a pair of cylinders, forming a compound engine, has been attached to each crank. The shafts for each wheel may either be connected by a clutch coupling, or left quite independent of each other, for the engines will be entirely under control whether they are coupled or not.

CHAPTER XXV.

SCREW-PROPELLERS.

Each blade of a screw-propeller may be regarded as a small portion of the thread of a screw of great pitch, and of considerable depth relatively to the pitch. The generation of the surface of a propeller blade of uniform pitch may be conceived from the following geometrical construction.

Let AA', Fig. 287, represent the axis of the screw. Suppose a line AD, perpendicular to AA', to move uniformly along AA', and at the same time to revolve uniformly around it. It is clear that the extremity B of the arm AB will travel on the surface of a cylinder, and will trace out a spiral curve BD' on that cylinder. The same will be true of every point in the line AD; the point C, for example, traces out the curve CC', therefore the surface swept out or developed by the line AD will be a spiral or screw surface.

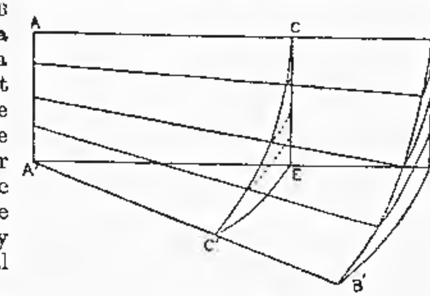

Fig. 287.

Pitch of the screw.—If the line AB made a complete revolution around AA', the distance of A' from A at the end of the revolution would be the *pitch of the screw*. It is the distance between two consecutive threads measured parallel to the axis.

Length of screw.—An actual screw-blade consists only of a portion of a complete convolution; and the extreme dimension of the blade, measured parallel to the axis, is called the *length of the screw*. In Fig. 287 this is represented by the line AA'. The aggregate length of all the screw-blades is equal to the length of the screw multiplied by the number of blades.

Angle of the screw.—The angle $BD'D$, between the curve and the plane $A'DB'$ perpendicular to the axis, is called the *angle of the screw*, at the radius AB. It is evident, if the pitch be constant throughout, that the smaller the radius the greater will be the angle of the screw, the angle $CC'E$, for example, being considerably greater than the angle $BB'D$. The relations between the pitch, circumference, and angle of the screw may be shown by means of a right-angled triangle, having the pitch as perpendicular and the circumference as base, the tangent of the angle of the screw being equal to the pitch divided by