# STEAM-ENGINE DESIGN.

FOR THE USE OF

MECHANICAL ENGINEERS, STUDENTS, AND DRAUGHTSMEN.

BY

JAY M. WHITHAM,

MEMBER OF AMERICAN SOCIETY OF MECHANICAL ENGINEERS;
PROFESSOR OF ENGINEERING, ARKANSAS INDUSTRIAL UNIVERSITY;
LATE ASSISTANT-ENGINEER, U. S. NAVY.

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*"Practice varies; but principles are eternal."*

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*The method of procedure* in using equation (2) is to compute the value of 4 $\sin^2 \alpha$ and $\sin^4 \alpha$ for each water-line, and, taking their mean value, find the coefficient of augmentation. Then, in the body plan, measure the immersed girth of a series of cross-sections and take their mean value. This mean immersed girth multiplied by the length of the ship between perpendiculars is the wetted surface. Multiply the wetted surface by the coefficient of augmentation, and the product is the augmented surface, $A_s$.

In the deduction of equation (2) the friction of one square foot of wetted surface is taken as 0.0036 lb. for wrought-iron (Weisbach), and the I. H. P. developed in the cylinders is assumed to be 1.63 of the power actually used in propulsion.

EXAMPLE.—An iron ship 250 ft. long, having a mean immersed girth of 50 ft., is to be propelled at a speed of 15 knots per hour. The measured angles for the several water-lines are as below. Find the I. H. P.

| Water-lines. | Angle $\alpha$. | Sin $\alpha$. | Sin² $\alpha$. | Sin⁴ $\alpha$. |
|---|---|---|---|---|
| Load, | 23° 16′ | 0.4 | 0.16 | 0.0256 |
| 2d, | 17° 28 | 0.3 | 0.09 | 0.0081 |
| 3d, | 11 33 | 0.2 | 0.04 | 0.0016 |
| 4th, | 5 45 | 0.1 | 0.01 | 0.0001 |
| Keel, | 0 0 | 0.0 | 0.00 | 0.0000 |
| Sum, | | | 0.30 | 0.0354 |
| Mean value, | | | 0.06 | 0.0071 |

$$\text{Coefficient of augmentation} = 1 + 4 \sin^2 \alpha + \sin^4 \alpha$$
$$= 1 + 4 \times 0.06 + 0.0071 = 1.2471.$$

$$\text{Wetted surface} = 250 \times 50 = 12500 \text{ sq. ft.}$$
$$\text{Augmented surface} = 12500 \times 1.2471.$$

$$\text{I. H. P.} = \frac{(15)^3 \times 12500 \times 1.2471}{20000} = 2630.$$

**127. Radial Paddle-wheel.**—As in § 125, let

$V$ = linear velocity of the centre of pressure of the paddle-float in feet per second;

$v$ = speed of the ship in feet per second;

$s$ = speed of the stream of water moved by the floats, in feet per second, relatively to still water;

$A$ = area of two floats (two being supposed immersed at the same instant, one on each side of the vessel) in square feet;

$D$ = effective diameter of the wheel in feet;

$A_s$ = augmented surface of the ship (see § 126);

$N$ = speed of the ship in knots per hour $= \dfrac{60 \times 60 \times v}{6080}$;

$f$ = friction of one square foot of the wetted surface of the vessel in pounds = 0.0036 for iron (Weisbach);

$W$ = weight of a cubic foot of water = 64.4 lbs. for sea and 62.5 for fresh water.

Then

$$fWA_s\frac{v^2}{2g}$$

is the resistance of the ship, which is also equal to

$$\frac{AWVs}{g} = \frac{AWV(V-v)}{g} \text{ (from § 125).}$$

Hence

$$fWA_s\frac{v^2}{2g} = \frac{AWV(V-v)}{g},$$

or

$$\textit{Area of one float} = \frac{A}{2} = \frac{fA_s v^2}{4V(V-v)} = \frac{A_s v^2}{1110V(V-v)} \text{ sq. ft.} \quad (1)$$

Seaton gives

$$\text{Area of one float, square feet,} = \frac{(\text{I. H. P.})}{D} \times C \quad . \quad . \quad (2)$$

where $C$ = 0.25 for tugs and 0.175 for fast-running light steamers.

*The slip*, $V - v = s$, is from 15 to 30 per cent of $V$, the velocity of the paddle. The slip varies inversely as the area of the float. The *centre of pressure* will vary with the depth of immersion and the form of the float. It is usually taken at the centre of the float. The (*length* $\div$ *breadth*) of a float $= 4$ in general practice. The *number of floats* is about equal to the diameter of the wheel in feet.

*The floats* are beams supported at the ends and loaded with a uniform load whose amount is, from § 125, equation (3), $AV(V - v)$ pounds. The *thickness* may be computed for strength, but it is usual to make it 0.125 of the breadth. This will give sufficient rigidity and strength for a float made of any tough, strong wood, such as oak or elm. The floats are secured to radial wrought-iron arms which are bolted to the cast-iron or wrought-iron hub. There are two or more arms for each float. The arms are cross-stayed, and bound by one or two wrought-iron rims.

**128. Design of a Feathering Paddle-wheel.**

CASE I. *Action of the Feathering Paddle.*—The following description is taken from § 247 of Rankine's *Machinery and Millwork;* (Consult Fig. 205.)

" Each of the paddles is supported by a pair of journals, so as to be capable of turning about a moving axis parallel to the axis of the paddle-wheel, while the position relatively to that moving axis is regulated by means of a lever and rod, connecting it with another fixed axis. Thus, in Fig. 205, $A$ is the axis of the paddle-wheel; $K$ the other fixed axis, or eccentric axis; $B$, $E$, $N$, $C$, $P$, $M$, $D$, the axis of a paddle at various points of its revolution round the axis $A$ of the wheel; $BF$, $EH$, $NQ$, $CR$, $PS$, $ML$, $DG$, the *stem-lever* of the paddle in various positions; $KF$, $KH$, $KQ$, $KR$, $KS$, $KL$, $KG$, various positions of the guide-rod which connects the stem-lever with the eccentric axis."

" When the end of the paddle-shaft *overhangs*, and has no outside bearing, the eccentric axis may be occupied by a pin fixed to the paddle-box framing; but if the paddle-shaft has an outside as well as an inside bearing, the inner edges of the guide-rods are attached to an *eccentric collar*, large enough to contain the paddle-shaft and its bearing within it, and represent the small dotted circle that is described about $K$. One of the rods, called the *driving-rod*, is rigidly fixed to the collar, in order to make it rotate about the axis of $K$; the remainder of the rods are jointed to the collar with pins.

"The object of the combination is to make the paddles, so long as they are immersed, move as nearly as possible edgewise relatively to the water in the paddle-race. The paddle-race is assumed to be a uniform current moving horizontally,

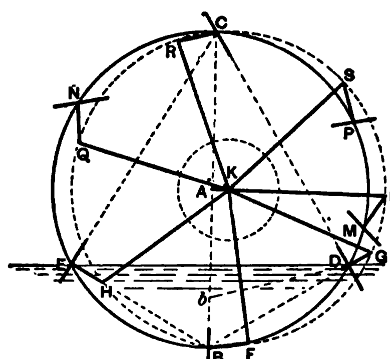



Fig. 205.

relatively to the axis $A$, with a velocity equal to that with which the axes $B$, etc., of the paddle-journals revolve round $A$. Let $E$ be the position of a paddle-journal axis at any given instant; conceive the velocity of the point $E$ in its revolution round $A$ to be resolved into two components—a normal component perpendicular, and a tangential component parallel, to the face of the paddle. Conceive the velocity of the particles of water in the paddle-race to be resolved in the same way. Then, in order that the paddle may move as nearly as possible edgewise relatively to the water, the normal components of the velocities of the journal $E$ and the particles of water should be identical.

"Let $B$ be the lowest point of the circle described by the paddle-journal axes; that is, let $AB$ be vertical. Draw the chord $EB$. Then it is evident that the component velocities of the points $B$ and $E$ along $EB$ are identical. But the velocity of $B$ is identical in amount and direction with that of the water in the paddle-race. Therefore the face of a paddle at $E$

should be normal to the chord $EB$, or as nearly so as possible. Another way of stating the same principle is to say that a tangent, $EC$, to the face of the paddle should pass through the *highest point*, $C$, of the circle described by the paddle-journal axes, $CAB$ being the vertical diameter of that circle.

"It is impossible to fulfil this condition exactly by means of the combination shown in the figure; but it is fulfilled with an approximation sufficient for practical purposes, so long as the paddles are in the water, by means of the following construction: Let $D$ and $E$ be the two points where the circle described by the paddle-journals cuts the surface of the water. From the uppermost point, $C$, of that circle draw the straight lines $CE$, $CD$, to represent tangents to the face of a paddle at the instant when its journals are entering and leaving the water. Draw also the vertical diameter $CAB$ to represent a tangent to the face of a paddle at the instant when it is most deeply immersed. Then draw a stem-lever projecting from the paddle in its three positions, $DG$, $BF$, $EH$. In the figure, that lever is drawn at right angles to the face of the paddle; but the angle at which it is placed is to a certain extent arbitrary, though it seldom deviates much from a right angle. The length of the stem-lever is a matter of convenience; it is usually about $\frac{3}{5}$ of the depth of a face of a paddle. Then, by plane geometry, find the centre, $K$, of the circle traversing the three points, $G$, $F$, and $H$; $K$ will mark the proper position for the eccentric axis; and a circle described about $K$, with a radius $KF$, will traverse all the positions of the joints of the stem-levers.

"From the time of entering to the time of leaving the water paddles fitted with this feathering gear move almost exactly as required by the theory; but their motion when above the surface of the water is very different, as the figure indicates.

"To find whether, and to what extent, it may be necessary to notch the edges of the paddles in order to prevent them from touching the guide-rods, produce $AK$ till it cuts the circle $GFH$ in $L$; from the point $L$ lay off the length, $LM$, of the stem-lever to the circle $BDE$, and draw a transverse section of a paddle with the axis of its journals at $M$, its stem-lever in the position $ML$, and its guide-rod in the position $LK$. This will show the position of the parts when the guide-rod approaches most closely to the paddle.

"Some engineers prefer to treat the paddle-race as undergoing a gradual acceleration from the point where the paddle enters the water to the point of deepest immersion. The following is the consequent modification in the process of designing the gear: Let the final velocity of the paddle-race be, as before, equal to that of the point $B$ in the wheel, and let the initial velocity be equal to that of the point $b$, at the end of a shorter vertical radius, $Ab$. Let $D$ be the axis of a paddle-journal in the act of entering the water, and $E$ the same axis in the act of leaving the water. Join $bD$ and $BE$, draw the face of the paddle at $D$ normal to $Db$, the face of the paddle at $B$ vertical, as before, and the face of the paddle at $E$ normal to $EB$. Then draw the stem-lever in its three positions, making a convenient constant angle with the paddle-face; and find the centre of a circle traversing the three positions of the end of the stem-lever; that centre will, as before, mark the proper position for the eccentric axis." *

CASE II. *Floats.—The area of one float* in square feet may be found by using formulæ (1) and (2) of § 127, by giving the quantity $V - v = s$, in (1), a value of from 12 to 20 per cent of $V$, and the constant $C$, in (2), a value of from 0.3 to 0.35. Seaton gives the following rules:

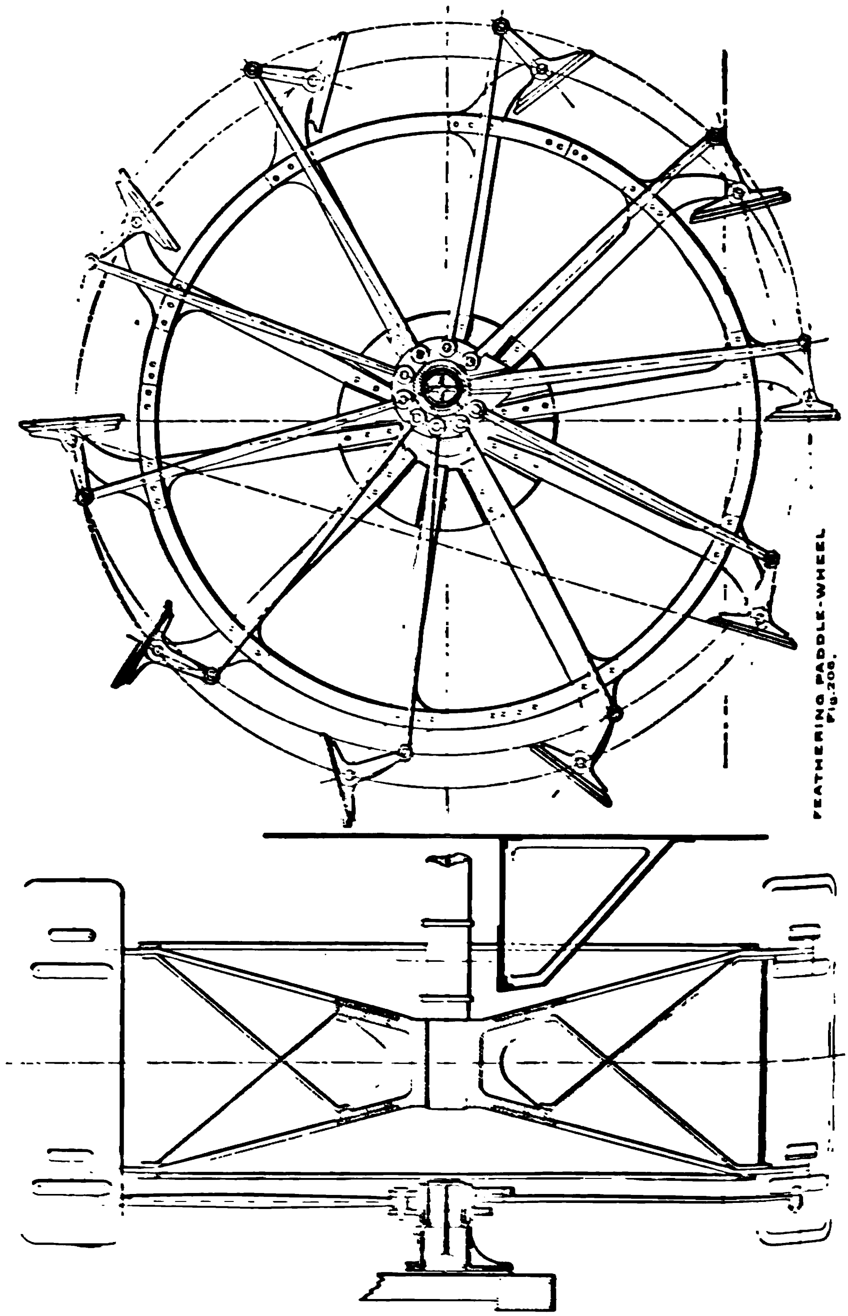
$$\text{Number of floats or paddles} = \frac{D+2}{2};$$

$$\text{Breadth of a float} = 0.35 \times \text{its length};$$

$$\text{Thickness of a float} = \tfrac{1}{12} \times \text{its breadth}.$$

CASE III. *Diameter of a Feathering Paddle-wheel.*—From § 127, $V = \pi D \times$ number of revolutions per second. The centre of pressure is, on account of the varying depth of im-

* Fig. 206 is from Donaldson's *Drawing and Rough Sketching for Marine Engineers.*



FEATHERING PADDLE-WHEEL
Fig. 206.

mersion and the disturbing influence of eddies, usually assumed to be at the centre of the float. Hence $D$ is the distance $BC$ in Fig. 205. From § 125, $V = s + v$, so that

$$D = \frac{v + s}{\pi \times \text{number of revolutions per second}}.$$

As in § 126, let

$N$ = velocity of the ship in knots per hour;
$s$ = percentage of slip;
$R$ = number of revolutions of the paddle per minute.

Then

$$D = \text{diameter of wheel at centres of pressure} = \frac{N(100 \times s)}{3.1R}. \quad (1)$$

From this it is seen that $D$ varies inversely as $R$. In case the vessel requires a large or small value for $D$, $R$ must be changed accordingly.

The depth of immersion must not be great, for if so the efficiency of the wheel is decreased. For a laden cargo steamer it is well to have the greatest immersion of a float about equal to its breadth. so that when carrying ballast only the floats will have two or three inches of immersion. For river service the greatest immersion of a float should be $\frac{\text{breadth}}{8}$, and for sea service, ordinarily, $\frac{\text{breadth}}{2}$ (Seaton).

**129. Design of Parts of a Paddle-wheel.**—From equation (3) of § 125 it is seen that the force exerted by the engine in propelling the vessel at a constant speed is

$$R = 2AV(V - v).$$

When the steamer is intended for river service the stress exerted through each wheel is $\frac{R}{2}$; but if she is used at sea or on the lakes the entire force may be transmitted through one wheel as the vessel rolls.

The radius rods, or arms, of the wheel are beams loaded at the centre of pressure of the float with a force $R$ or $\frac{R}{2}$, according to the service of the steamer. We will suppose the portion of the wheel out of the water to be rigid, and that the loaded radial arms are attached to it by either inner and outer, or by inner rims only. The beams are then loaded by forces acting in the opposite direction to $R$, whose values are the transverse strength of the sections of the rims. The radial arms should be as strong as the shaft at the outer bearing. In Chapter XII the method of finding the torsional moment equivalent to the combined twisting and bending moments on the shaft was discussed. Let $T$ be this moment for the paddle-wheel shaft in inch-pounds (this discussion will also apply to a radial paddle), and $\frac{D}{2}$ be half of the effective diameter of the paddle-wheel in inches; then, taking moments about the axis of the shaft,

$$T = \frac{RD}{2},$$

or

$$R = \frac{2T}{D}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

From equation (1) of § 91 we have

$$T = \frac{fd^3}{5.1},$$

where $f = 9000$ for wrought-iron shafts, and $d =$ diameter of the shaft in inches. By mechanics, the strength of a beam of rectangular cross-section varies as $tb^2$, where $t =$ the thickness of the radius arm parallel to the shaft, and $b =$ the breadth of the arm, both in inches. Hence, since there are $n$ of these arms to each wheel,

$$R \propto ntb^2 \propto T \propto d^3. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

As no dimensions are given for the breadth and thickness of each rim and the arm in various parts, it is impossible to give an exact solution. The following approximate solution, given by Seaton,* is sufficiently accurate for designing the paddle-wheel: From (2)

$$tb^2 = \frac{Cd^2}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$C$ is 0.7 for a section of the arm near the hub, and 0.45 for a section near the inner rim when there are two rims used. When an inner rim *only* is used the arms project beyond it, and the load comes on these projections. Hence, in this case, the arms must be stronger than when two rims are used. Assuming that the centre of pressure of a float is $\frac{1}{10}D$ from the inner rim,

$$tb^2 = C'd^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

for this case, and $C' = 0.1$.

The following are usual values of the ratio $\frac{b}{t}$:

*For the arms—*

Near the hub, $\frac{b}{t} = 5$;

Near the rim, $\frac{b}{t} = 3.5$ to 4.

When two rims are used, $\frac{b}{t} = 5$ throughout the length of the arm.

*For the rims—*

Inner rim, $\frac{b}{t} = 5$, when two rims are used;

Inner rim only, $\frac{b}{t} = 4$, when one rim is used;

Outer rim, $\frac{b}{t} = 4$, two rims being used.

* *A Manual of Marine Engineering*, p. 281.

When two rims are used the section of the inner is 0.8 of the arm near it, and the section of the outer rim is equal to that of the arm near it.

When an inner rim only is used its section is equal to that of the arm near the hub.

*The stay-rods* are bolted to the inner rim between the floats and to the sides of the hub opposite, so that they are diagonals between the arms. The diameter of a stay-rod is equal to twice the thickness of the rim.

*All pins* used in a feathering-wheel are made of iron cased in brass, and they work in lignum-vitæ bearings. When the wheel is used in sandy water white-metal bearings and iron pins are preferred.

**130. The Screw-propeller** is made in various forms. The best propeller for one ship is not necessarily the best one for a similar ship having the same-powered engines. Ordinarily a four-bladed propeller is best for sea service, and a two-bladed one for smooth water. From equation (3), § 125, we see that the reactive force on the propeller (the thrust on the shaft) is

$$2AV(V-v), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which $A$ = area of the disk of the propeller minus the area of a transverse section of the hub in square feet; $V$ = velocity of the screw in feet per second = pitch of the screw in feet multiplied by the number of turns per second; $v$ = the speed of the ship in feet per second; and $s = V - v$ = the slip of the screw.

There is always *real slip* when a vessel is propelled by any instrument. The *apparent slip* is the difference between the speed of the propeller and the speed of the ship. This is not always positive on account of the shape of the "run" of the vessel and the disturbed condition of the water around the propeller.

Formula (1) may be changed * to

$$\text{Thrust in pounds} = 5.66AS(S-s), \quad . \quad . \quad . \quad (2)$$

* See Rankine's *Rules and Tables*, p. 275.