necessarily show a decrease in the actual production; it is probably an error in the estimates.

The above figures were taken from the *Petroleum Reporter*. From reports made to me during the past week, the Warren district is said to be producing nearly 500 barrels, while the Bradford daily production approaches 8500 barrels. Of the Bradford production, about four-fifths come from the regular Bradford sand, and one-fifth from the higher horizons in the "slush oil" wells.

CRANK PINS AND JOURNALS.¹

By WILLIAM D. MARKS, Whitney Professor of Dynamical Engineering, University of Pennsylvania.

(15) THE CRANK PIN.—The crank pin has ever been one of the most troublesome parts of the steam engine to the mechanical engineer. The mere determination of its proportions, so that it will not break under the strain put upon it by the pressure of the steam upon the piston head, does not suffice, and often results in trouble from heating when the engine is at work. It therefore becomes a first consideration to so proportion crank pins as to prevent heating, their strength being a matter of secondary importance to be afterwards investigated, if it is deemed necessary to do it.

Before taking up the mathematical part of our consideration, it will be of practical value to quote, from the writings of General Morin, the following remarks:

"But it is proper to observe, that from the form itself of the rubbing body (cylindrical), the pressure is exerted upon a less extent of surface, according to the smallness of the diameter of the journal, and that unguents are more easily expelled with small than with large journals. This circumstance has a great influence upon the intensity of friction, and upon the value of its ratio to the pressure.

"The motion of rotation tends of itself to expel certain unguents, and to bring the surfaces to a simply unctuous state. The old mode of greasing, still used in many cases, consisted simply in turning on

¹ An extract from a course of lectures on "The Relative Proportions of the Steam Engine."

the oil, or spreading the lard or tallow upon the surface of the rubbing, and in renewing the operation several times in a day.

"We may thus, with care, prevent the rapid wear of journals and their boxes; but with an imperfect renewal of the unguent, the friction may attain $\cdot 07$, $\cdot 08$, or even $\cdot 10$ of the pressure.

"If, on the other hand, we use contrivances which renew the unguent, without cessation, in sufficient quantities, the rubbing surfaces are maintained in a perfect and constant state of lubrication, and the friction falls as low as $\cdot 05$ or $\cdot 03$ of the pressure, and probably still lower.

"The polished surfaces, operated in these favorable conditions, became more and more perfect, and it is not surprising that the friction should fall far below the limits above indicated." (Bennett's *Morin*, pp. 307-8.

If the unguents are expelled by extreme pressure, so that the surfaces are simply unctuous, the friction increases rapidly, and the surfaces begin to heat and wear immediately.

These statements apply with equal force to cast iron and cast iron; cast iron and wrought iron; cast iron and brass or Babbitt's metal; or with steel or wrought iron in the place of cast iron.

The supposed superiority of brass or Babbitt's metal lined boxes over iron boxes, in positions very liable to heating, lies in their greater softness and conductivity for heat. Brass will conduct heat away from two to four times as rapidly as iron. However, the film of unguent interposed may render the conductivity of brass of less avail than is generally supposed, and the advantage lies only in the fact that being a softer metal, in case of heating, the surface of the softer metal receives the principal damage.

It is, perhaps, good practice to use brass or soft metal wherever the pressure exceeds 125 lbs. per square inch of projected area. At lower pressures an ordinary good lubricating oil may be relied upon to form a film, and run without breaking at ordinary speeds. (Winter strained lard oil was the lubricant used, diameter of journal 3 inches, making 120 revolutions per minute. But a few experiments were made. The continuity of the film of lubricant is affected by so many different conditions, that it is impossible to fix any exact limit of pressure.)

With soft metal or brass bearings, good results can be obtained at pressures of 1000 lbs. or more per square inch of projected area; if,

however, the film of unguent does break at these higher pressures, heating begins almost instantly, and if the surfaces in contact are both of hard metal, as iron and iron, injury to both at once results, while if the boxes are brass or some of the softer metals, the continuity of the surface film may be restored by increased lubrication, or by stopping and cooling as soon as heating is observed.

Several expedients are used to keep bearings, which have a tendency to heat, cool.

The introduction of rotten stone, or sulphur with oil, is perhaps the best. Quicksilver, or lead-filings, introduced with oil, coat the rubbing surfaces, and diminish the heating where the rubbing surfaces are very much scored.

A great increase of the velocity of the rubbing surfaces renders bearings more liable to heat than a great increase in pressure, although the total amount of work done by friction is the same in both cases, and is probably accounted for by the more rapid expulsion of the lubricant.

As the cause of the heating of bearings, when they are of tolerably good workmanship, is the transformation of the work of friction into heat, we see that it is necessary to reduce the friction, as much as possible, by the perfect smoothness of surfaces in contact, the interposition of lubricants, and the reduction of the speed and pressure upon the rubbing surfaces.

In all machines there is a limit, below which we cannot reduce the speed and pressure of the rubbing surfaces, and we must, therefore, so proportion journal bearings as to cause no more work due to friction, *i. e.* heat, to be produced than can be conveyed away by the unguents, the atmosphere, and the conductivity of the metals, without raising the temperature of the bearing appreciably.

From the statistics of the working of the crank pins of four screw propellers in the United States Navy (Van Buren, "Strength of Iron Parts of Steam Machinery," page 24), we take the following statement and table:

"The crank pins of these vessels worked cool, giving but little trouble, which is the exception rather than the rule for screw engines."

The projected area of crank pin journal, given in column 7, is that rectangular area formed by a central section of the crank pin journal, in the direction of its length.

Columns 1, 2, 4, 5, and 6, are given. Columns 3, 7, 8, 9, 10, and 11, are calculated from them.

In calculating column 9 from columns 3 and 8, we have assumed the coefficient of friction at .05, which is the highest value given by Gen'l Morin for constant lubrication, and the most probable in the present cases.

Column 11 is derived from columns 3 and 7. Column 10 is derived from columns 7 and 9.

Name of Vessel.	1. Diameter of steam cyl.	2. Pressure per sq. inch.	3. Total force of steam pressure on piston head.	4. No. of strokes per minute.	5. Length of crank pin.	6. Diameter of crank pin.	7. Projected area of crank pin.	8. Velocity of rubbing surfaces, in feet, per minute.	9. Total work of friction in foot pounds per minute.	10. Work of friction per sq. inch of projected area in foot-pounds per minute.	11. Pressure per sq. inch of projected area.
	in.	lbs.	lbs.		in.	in.	sq. in.				
Swatara.....	36	40	40716	160	12	8.5	102.	178.	362372	3552.6	399.5
Saco.....	30	40	28274	180	9	7.5	67.5	176.7	249801	3700.7	419.
Wampanoag.....	100	40	314160	62	27	16.	432.	129.8	2038898	4719.7	727.2
Wabash.....	72	28	114002	100	16	15.	240.	196.3	1118910	4662.1	475.
Averages										4159.	505.

From column 10 of the table, we find the average amount of work per square inch of projected area of crank pin journal, which, in the cases cited, has been borne without heating, to be 4159 ft. lbs. per minute, and in making use of this quantity in our subsequent calculations, we are on the safe side, if the coefficient of friction (assumed at .05) has not been taken too small.

Let d = the diameter of the piston head in inches.

Let P = the mean pressure in the steam cylinder in pounds per square inch.

Then, $7854 d^2 P$ = the mean pressure on the piston head in pounds.

Let f = the coefficient of friction.

Let l_3 = the length of the crank pin journal in inches.

Let d_3 = the diameter of the crank pin journal in inches.

The mean *force* of friction at the rubbing surfaces of any crank pin journal, per square inch of projected area, is

$$= .7854 f \frac{P d^2}{l_3 d_3}.$$

Let N = the number of strokes per minute (= twice the number of revolutions).

Let w = the work of friction per minute.

The space passed over by the force due to friction, in one minute,

$$= \frac{\pi}{2} N d_3 = 1.5708 N d_3 \text{ inches,}$$

and we have for the work of friction, *i. e.*, heat, per minute,

$$w = 1.5708 \times .7854 f \frac{P N d^2}{l_3} \text{ inch lbs.}$$

From this formula the diameter of the crank pin journal (d_3) has vanished. Why it has vanished will be understood when we observe that the force per square inch of projected area, due to friction, is inversely as the diameter of the journal; while the space passed over by this force is directly as its diameter.

Replacing w in the last formula by the mean value derived from column 10 of the table, = 4159 ft. lbs. = 49908 inch lbs., we have

$$49908 = 1.2337 f \frac{P N d^2}{l_3}.$$

Therefore, $l_3 = .0000247 f P N d^2$, (20).

Considering formula (20), we see that the length of the crank pin increases and decreases with the coefficient of friction, the mean steam pressure per square inch, the number of strokes per minute, and with the square of the diameter of the steam cylinder.

A consideration of the component formulæ of (20), shows that as the crank pin journal decreases in size, the pressure per square inch becomes greater; but if this reduction in size is obtained by a diminution of the diameter (d_3) of the crank pin journal, the work per square inch of projected area is not increased, for the velocity of the rubbing surfaces, by this means, is decreased in the same ratio as the pressure is increased.

Within reasonable limits, as to pressure and speed of rubbing surfaces, the general statement may be made:

The longer any bearing, which has a given number of revolutions and a given pressure to sustain, is made, the cooler it will work, and its diameter has no effect upon its heating.

Example :

Let $d = 30''$, $N = 180$, and $P = 40$ lbs. per square inch.

We have, by substitution in formula (20),

$$l_3 = .0000247 \times f \times 40 \times 180 \times 900 = 160 \cdot f.$$

If in this we take $f = .03$ to $.05$ for perfect lubrication, we have

$$l_3 = 4 \cdot 8'' \text{ to } 8''.$$

If we take $f = .08$ to $.10$ for imperfect lubrication, we have

$$l_3 = 12 \cdot 8'' \text{ to } 16''.$$

The results show the great advantages arising from constant oiling of bearings.

As a practical application under the most trying conditions, we find that Messrs. Wm. Sellers & Co. are able to double the speed of their bolt cutters when an oil pump is used, without injury to the dies.

NOTE.—If for locomotive crank pin journals we assume N the number of strokes per minute = 600.

P the pressure per square inch in lbs. = 150, the formula (20) being changed to

$$l_3 = .00000247 f P N d^2,$$

by removing the decimal point one place to the left, will give the length of journal commonly assumed in successful practice, if we assume the coefficient of friction at $.06$.

The above formula then becomes

$$l_3 = .013 d^2.$$

This formula would show the amount of heat per square inch of projected area conveyed away from the crank pins of locomotives to be ten times greater than in the case of marine engines, did not the variations of speed and frequent stoppages of a locomotive prevent comparison.

Example :

Let $d = 18$ inches.

We have,

$$l_3 = .013 \times 324 = 4 \cdot 21 \text{ inches.}$$

We will next take up the diameter of the crank pin journal.

It is necessary, first, to determine its length by formula (20), and with this length to determine the proper diameter.

Let a = the deflection of pin under stress in inches.

Let S = the stress on pin in lbs.

Let E = the modulus of elasticity of wrought iron = 28000000 lbs.

Let W = the measure of the moment of flexure of the pin.

Let l_3 = the length of journal in inches.

The deflection of a beam fixed at one end, and loaded at the other ("Weisbach's Mechs. of Eng.," Sec. IV, Art. 217), is

$$a = \frac{S l^3}{3 W E}$$

If, again, the beam be supposed to be uniformly loaded and fixed at one end, its deflection will be (W. M. E., Sec. IV, Art. 223)

$$a_2 = \frac{1}{8} \frac{S l^3}{W E},$$

and if for the load at the end we concede a deflection of $\frac{1}{100}$ of an inch, we have for the same load under the two cases above mentioned,

$$a_1 = .01 \text{ inch,}$$

$$a_2 = .0038 \text{ inch.}$$

Then taking the most unfavorable case, *i. e.*, the load at the end, we have,

$$\text{Letting } S = \frac{\pi d^2}{4} P_b,$$

Letting P_b = the greatest pressure of steam in cylinder = the boiler pressure,

$$\text{Letting } W = \frac{\pi d_3^4}{64},$$

$$\frac{1}{100} = \frac{1}{8} \frac{\frac{\pi d^2}{4} P_b l_3^3}{\frac{\pi d_3^4}{64} 28000000} = \frac{16 P_b l_3^3 d^2}{84000000 d_3^4}$$

$$d_3^4 = \frac{1600}{84000000} P_b l_3^3 d^2 \quad . \quad . \quad . \quad . \quad . \quad (21).$$

$$d_3 = .066 \sqrt[4]{P_b l_3^3 d^2} \quad . \quad . \quad . \quad . \quad . \quad (22).$$

Example:

Let $P_b = 60$ lbs. per square inch.

Let $l_3 = 8$ inches.

Let $d = 30$ inches.

Substituting in formula (22) we have

$$d_3 = .066 \sqrt[4]{60 \times 512 \times 900} = 4.79 \text{ inches.}$$

There is no need of an investigation of the strength of a crank pin, as the condition of rigidity gives a great excess of strength.

The length of a steel crank pin is just the same as that of a wrought iron pin, and the modulus of elasticity of steel is so nearly equal to that of wrought iron, as to make formula (22) serviceable for both steel and wrought iron alike.

Referring to the table, we find the average pressure per square inch of projected area to be about 500 lbs.

If now we divide the whole pressure upon the crank pin by 500, we obtain the projected area required when this limit is not to be exceeded.

$$\text{The equation } d_3 l_3 = \frac{\pi d^2 P}{4 \times 500} \text{ gives } d_3 = .00157 \frac{d^2 P}{l_3}.$$

Example:

Let $P = 40$ lbs., $l_3 = 8''$, $d = 30''$,

$$d_3 = .00157 \frac{900 \times 40}{8} = 7.06 \text{ inches.}$$

This latter method is perhaps the most practical, and has the advantage of limiting the pressure. It will almost always give larger results than the preceding method.

For journals in general,

Let R = force on bearing in pounds.

Let l = length of bearing in inches.

Let d = diameter of bearing in inches.

Let T = the number of turns per minute.

Let f = the coefficient of friction.

And we have

$$49908 = \pi T d \cdot f \frac{R}{l d},$$

$$l = \frac{\pi}{49908} f T R = .000063 f T R,$$

giving the minimum length of bearing allowable, which should always be exceeded where circumstances will permit.