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No. 47

A Dollar's Worth of Condensed Information

Design of Electric Overhead Cranes

CRABS, GEARING AND BRAKE MECHANISM

By R. B. Brown

Second Edition

Price 25 Cents

CONTENTS

The Industrial Press, 49-55 Lafayette Street, New York
Publishers of MACHINERY

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CHAPTER I

PRELIMINARY CONSIDERATIONS*†

The introduction and development of the electric motor, which has revolutionized so many of the methods of manufacture and transportation, has, perhaps, influenced the design of no other single auxiliary apparatus in the productive industries more than that of cranes and hoists. The present treatise on the subject, therefore, has been written with the intention of placing on record the present practice in the design of overhead cranes, electrically operated, and of presenting such data as will aid the designer of such apparatus to properly calculate and proportion the various details, and supervise their construction.

Overhead Travelers

The overhead traveler in its various forms is probably in greater demand than any other type of electric crane on the market, a fact which has induced many firms to specialize in this particular branch of crane building. As a result of the continued and growing demand for these cranes, many attempts have been made, with more or less success, to standardize, as far as possible, the various details of construction.

Electric travelers represent a type of crane which under ordinary conditions is in almost continuous service, and, as with other constantly working machines, it is essential that rapidity of operation, together with economy in current consumption, be pre-eminent factors to the purchaser and manufacturer alike. The requirements of the former should be based on the results of general experience gained during the past few years, while these results depend entirely on the skill of the designer, and the workmanship.

The three types of ordinary travelers in use are the one-motor, three-motor, and four-motor cranes. The three-motor, and for medium and heavy cranes, the four-motor types, have been found to be by far the most efficient, and are, practically speaking, the only types now used for modern workshops, warehouses, and similar places. Until quite recently the single-motor type was considered preferable, on account of its cheapness, for engine rooms and similar places where a crane is only required occasionally. The present price of motors and their connections, and the fact that single-motor cranes require more gearing than the three-motor type, is in favor of the universal adoption of the latter, more especially since several makers manufacture the crabs of this type in quantities and keep them in stock, and can therefore give a quicker delivery.

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* The present treatise deals exclusively with the design and construction of the crabs, gearing, and brake mechanism. The subject of girders for electric overhead cranes is treated in MACHINERY'S Reference Series No. 49.

† MACHINERY, January, 1909.

One of the principal obstacles that has been placed in the way of standardizing electric cranes is the widely varying opinions of engineers on the question of speeds. Except for travelers which are required for work of a special nature, there is no reason why all cranes of this type should not be worked at practically the same speeds. In order to consider the conditions affecting the speed of each motion, they must be dealt with separately.

When inquiring for a crane of any type, it is usual to state the speed at which the maximum load has to be lifted, and in selecting this speed the fact should not be overlooked that, excepting the case of small-powered cranes and those required for special service, the normal load is seldom more than about 20 per cent of the full capacity of the crane. It is better, therefore, to consider the highest and safest speed at which this normal load can be worked, and then select a full load speed which will give the same foot-tonnage of work done. By the use of crane-rated series-wound motors, a variation above the rated speed of about 50 per cent, increasing in proportion to the load, can be obtained, and this fact makes the use of change gears on the main lift unnecessary. If, however, a crane is to be used in a shop where a great deal of small material has to be constantly handled, but where a full load only occurs occasionally, as, for instance, in a fitting shop, it is more economical to have an auxiliary barrel fitted onto cranes of from ten to twenty tons capacity, and worked by the main lift motor. When a light lift is required, it should be one-fifth of the full capacity of the crane, and the speed specified should be such as will give the same foot-tonnage as the main lift. Auxiliary barrels are generally placed on the main barrel pinion shaft, and so arranged that either the main pinion or the auxiliary barrel may be driven from this shaft by means of a clutch.

For cranes of twenty-five tons and upward, that are to be in constant use, the best practice now demands an independent motor for the auxiliary barrel, the capacity of which is generally five or six tons, and the speed from twenty to thirty feet per minute.

The conditions concerning the acceleration of speed under lighter loads apply in a similar manner to traversing and traveling, and it is never worth while having a change of gear applied to these motions. The traveling speeds are a somewhat variable quantity, and cases often occur where small-powered cranes have to travel at a very high speed, as, for example, where cranes are used over pig-casting beds or stock yards; when engaged in such work they may travel at a speed of 500 feet per minute, or more. For ordinary shop and similar practice the various speeds given in Table I are deduced from modern requirements and represent an average of the speeds which have been standardized by leading makers.

In connection with the speeds in Table I, it will be necessary to explain how the horsepower required in each case has been arrived at. The horsepower of the lifting motor depends purely on the work done on the load, and the power absorbed in the resulting friction of the gearing, journals, and pulleys. This quantity varies to some extent with the number of reductions and the type of gearing. The efficiency of a crane is generally lowest at the test, improving somewhat as the journals and teeth get bedded down. The efficiency of the first or motor reduction with well-made machine cut spur gears running in an oil bath, has been found by trial to reach as much as 97 per cent, and may be taken at 95 per cent under ordinary practical condtions.

The average efficiency of one reduction of cut spur gears, running dry, is 92 or 93 per cent, and of cast spur gears running dry, 90 per cent. The loss due to journal friction is generally about 2 per cent for each axle when properly lubricated. The only other loss in efficiency of any importance is in the snatch block, if there is one fitted to the crane. This quantity is always reduced by using large pulleys

TABLE I. SPEEDS OF ELECTRIC OVERHEAD CRANES

Power of Crane, tons.	Lifting.		Traversing.			Traveling.						
						Weight of Crane.				B. H. P. Required.		
	Speed, feet per min.	B. H. P.	Weight of Crab, tons.	Speed, feet per min.	B. H. P	30′ 0″ Span, tons	50′ 0″ Span, tons.	70′ 0″ Span, tons.	Speed, feet per min	30′ 0″ Span.	50′ 0″ Span.	70′ 0″ Span.
3	33	10	2	120	2	7	10	14	300	5	6	7
5	20	10	2½	120	2	10	13	16	300	7	8	9
7½	20	15	3	120	3	11	14	17	300	8	9	10
10	15	15	4	100	4	12	15	18	250	9	10	11
15	12	18	4½	100	5	14	16	21	250	12	13	14
20	10	20	5	100	6	16	17	23	250	14	15	16
25	10	25	5½	80	6	19	21	26	200	14	15	16
30	10	30	6	80	8	21	25	31	200	15	18	20
40	7½	30	8	80	10	25	34	43	200	20	23	25
50	6	30	12	60	10	32	39	48	150	20	23	25
60	5	30	16	60	12	....	44	58	150	...	26	28
75	5	38	20	60	12	....	50	70	150	....	32	36
100	5	50	28	60	16	....	70	80	150	.	40	42
120	5	60	32	60	20	. ..	80	95	150	....	45	50
150	5	75	38	40	20	....	90	105	150	....	55	60

and, preferably, small hardened pins, the pulleys being bushed with gun-metal, under which condition the efficiency works out to about 97 per cent.

From the above results a very fair idea of the over-all mechanical efficiency of a crane can be determined if the number of reductions and the other particulars are known. It will be found that small high-speed cranes have a higher efficiency than the larger ones, owing to there being less gearing; thus, in the case of a crab lifting three tons on a single rope and having two reductions of machine-cut gearing, the first of which runs in oil, the over-all efficiency will be about

$$\frac{95 \times 92 \times 98 \times 98}{100} = 84 \text{ per cent.}$$

For a contrary example, take a 50-ton crab having four reductions, the first three of which are machine cut, the motor reduction running in oil. Then the over-all efficiency will be about

$$\frac{95 \times 93 \times 93 \times 90 \times 98^4 \times 97}{100} = 66 \text{ per cent.}$$

A very common rule in practice is to allow ten foot-tons of work done at the hook per brake horsepower, this factor being equivalent to a mechanical efficiency of about 66 per cent. This constant is practically correct for medium and large cranes, but for small sizes it allows for a slightly larger motor than is really necessary, which is perhaps a good fault, since small cranes are generally in constant use. The above calculations are not, generally speaking, necessary in practice, but have been made in order to show how and where the power due to friction is principally absorbed, and it will be seen that the results agree very closely with the diagrams shown in Fig. 1, which are made

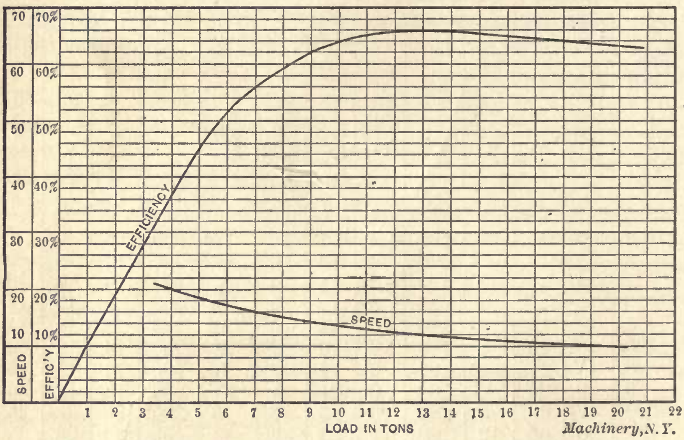


Fig. 1. Lifting Diagram for 20-ton Crane; Full Load Speed, 10 feet per Minute

from trials taken from overhead cranes, representing the best class of design and workmanship. It is usual and more instructive to speak of the gross efficiency of a crane—that is, the combined electrical efficiency of motor and wiring—and the mechanical efficiency of the gearing, or, in other words, the ratio between current consumed at the switchboard and work done at the hook. This is the efficiency as shown by the diagrams.

The electrical efficiency of a lifting motor may generally be taken at 80 per cent, covering motor and wiring, so that in the case of the 3-ton crane exemplified above, the gross efficiency would be about $\frac{84 \times 80}{100} =$ 67 per cent. Similarly, the 50-ton crab would give a gross efficiency of $\frac{66 \times 80}{100} = 53$ per cent. Both of these quantities agree with actual results.

The horsepower, or current required for each motion, as given in Table I, is the calculated power based on the above-described coefficients; for practical purposes, however, the nearest manufactured size of motor would be used. The power required for traversing and traveling must be sufficient to overcome the rolling and axle friction, and the friction of the intermediate driving gear. The horsepower of these motors is generally based upon a certain tractive resistance, usually expressed in so many pounds per ton of rolling load. This is a very variable quantity, even the results of tests showing a remarkable latitude. Although it is best to work with data obtained from experiments, it will be well to show the calculations which most nearly agree with actual results.

The axle friction μ depends to some extent on the lubricating arrangements, but in the calculations it is assumed that these conditions are

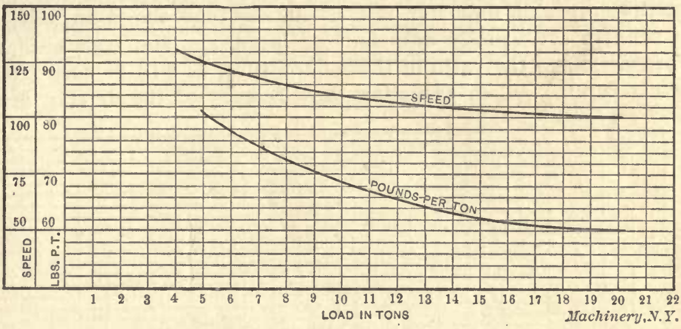


Fig. 2. Traversing Diagram for 20-ton Crane; Full Load Speed, 100 feet per Minute

well provided for, both in traversing and traveling. The power required is figured from the formula

$$P = (\mu r) \frac{W}{R}$$

where W = load in pounds,
R = radius of wheel in inches,
r = radius of axle in inches,
μ = coefficient of friction = 0.10.

The rolling friction of metal wheels on steel rails is considered to be equal to $0.002 \frac{W}{R}$. This quantity may, therefore, be combined with the above quantity and the result obtained direct, thus

$$(\mu r + 0.002) \frac{W}{R}.$$

Take for example a 30-ton crab weighing six tons, having runners 18 inches in diameter and axles 4½ inches in diameter. The combined axle and rolling friction will be

$$(0.1 \times 2.25 + 0.002) \times \frac{36 \times 2240}{9} = 2034 \text{ pounds.}$$

The efficiency of the driving gear can be found in the same manner as described for the lifting gear, when the number of reductions are known. In the present example there would be three reductions of machine-cut gears, all running dry, the total efficiency of which would be about

$$\frac{92 \times 92 \times 92 \times 98 \times 98}{100} = 75 \text{ per cent.}$$

Taking the full load speed at 60 feet per minute, it will be found that the brake horsepower required will be

$$\frac{2034 \times 60 \times 100}{33{,}000 \times 75} = 5 \text{ B.H.P.}$$

The axle friction of the traveling gear will always be found to be considerably less than that of the traversing motion, due to the fact that the crab axle diameter is often larger than the main axle, while the runners are usually only half as large, and the resistance varies in proportion to the ratio of these quantities, as will be seen from the above formula. When assuming the efficiency of the driving gear for

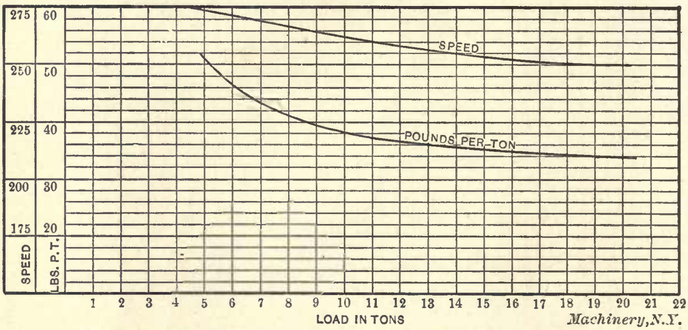


Fig. 3. Traveling Diagram for 20-ton Crane; Full Load Speed, 250 feet per Minute

the traveling motion, some special allowance should be made for the loss of power due to the cross shaft. This shaft is carried by several bearings, and it is probably the deflection of the girders, and the consequent slight bending of the shaft, that causes the drive to be rather inefficient.

No direct results concerning this shaft are available, but its efficiency will probably be about 90 per cent for cranes of moderate span. Allowing for two reductions of machine-cut gears running dry, the efficiency of this drive will, therefore, be about $\frac{92 \times 92 \times 90}{100} = 76$ per cent. Suppose, for example, that a 30-ton crane traveler weighs 25 tons and runs upon wheels 30 inches in diameter having 4-inch axles; then the combined rolling and axle friction will be, as in the case of the crab:

$$(0.1 \times 2 + 0.002) \times \frac{55 \times 2240}{15} = 1659 \text{ pounds.}$$

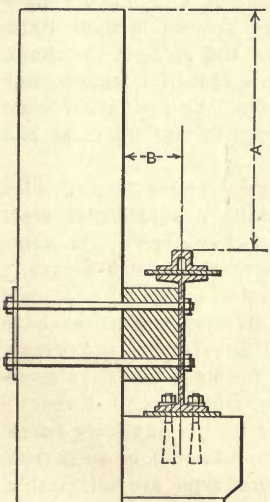
Taking the traveling speed at 150 feet per minute, and neglecting acceleration, which may, for ordinary speeds, be assumed as taken care of by the coefficient of friction and the overload of the motor permissible, the brake horsepower required will be

$$\frac{1659 \times 150 \times 100}{33{,}000 \times 76} = 10 \text{ B.H.P.}$$

The resistance to traction, as nominally referred to, which this power covers, will be

$$\frac{10 \times 33{,}000}{150 \times 55} = 40 \text{ pounds per ton.}$$

Similarly the 5 B.H.P. motor for traversing allows for 76 pounds per ton. For practical purposes 40 to 50 and 60 to 70 pounds per ton have been allowed for the best class of travelers having large diameter wheels and machine-cut gears. Some tests have shown that a higher factor than 70 pounds per ton is required for traversing, such results



Machinery, N.Y.

TABLE II. CLEARANCE SPACE REQUIRED FOR OVERHEAD CRANES.

Size of Crane	A		B
Tons.	Ft	Ins	Inches.
3	5	0	7
5	5	6	7
7½	5	6	7
10	5	9	7½
15	6	0	7½
20	6	0	8
25	6	6	8
30	7	0	9
40	8	0	9
50	8	6	10
60	9	0	10
75	9	6	10
100	10	0	12
120	12	0	12
150	12	0	12

possibly being due to the fact that the lubrication was inefficient; the traveling wheels also are often too small. The diagrams in Figs. 2 and 3 show actual results obtained in the traversing and traveling motions of cranes.

Before concluding this chapter on preliminary considerations, it is advisable to call the architect's attention to the importance of allowing an adequate working space for travelers when designing shops, etc. Due to overlooking this fact, the first cost is often considerably increased, while the crane is generally of inefficient design and unsightly appearance. The dimensions given in Table II have been taken from actual practice, and will be a guide to those designing new buildings. The headroom given in this table is the least possible with an ordinary type of crab.

CHAPTER II

FRAMING AND GENERAL DESIGN OF CRABS*

The great difference in appearance of electric crabs lies chiefly in the design of the framing. Some very different opinions seem to exist on the primary qustion of material, for while certain makers have standard crabs with steel frames for all sizes, others manufacture crabs with cast-iron frames up to a considerable weight. Both types are undoubtedly substantial enough, but there is a limit to their adoption. One of the principal objects in designing an electric crab should be to make the various component parts as accessible as possible for renewal and repair, and in this direction the steel frames undoubtedly have the advantage. Another point in favor of steel frames is their lightness, and the consequent saving of material in the girders. Perhaps, in the case of small crabs, say up to five tons, cast-iron frames may be cheapest, if large quantities are being made, but for larger sizes the steel frames are certainly more advantageous to manufacturer and purchaser alike.

Frames made up of double steel plates, or steel plates framed with angles, are much used, but although they make a substantial crab, nothing special can be said in their favor, and they present the same disadvantages as cast-iron frames, in having several closed bearings.

Fig. 4 shows the crab for a 20-ton, three-motor crane with a steel frame of modern design and similar to which large quantities have been made with satisfactory results. Fig. 5 illustrates the same crab, slightly altered to receive an auxiliary barrel for light loads, as sometimes found advantageous. The crab for a four-motor crane is usually of similar construction to the one just referred to, the auxiliary barrel being arranged to suit the framing of the crab. As will be seen from Figs. 4 and 5, all the bearings in crabs of this type are adjustable, except those for the barrels and perhaps the running wheels, which are bushed; since these latter have slow-running shafts, and ample bearing surface can be provided, there is no necessity for adjustable bearings. In the designs shown it will be seen that the main barrel shaft is so situated that by jacking up the crab it can be drawn out clear of the rails, and the barrel and wheel lowered direct to the ground for rebushing or other attention.

Owing to the variety of designs of crab frames in general use, it is practically impossible to give any definite calculations or details which would be of general value. In the case of small crabs, the smallest convenient sections which can be used are generally strong enough to resist all the strains with a large margin of safety, but with the larger sizes the strength of the sections should be calculated and the stress

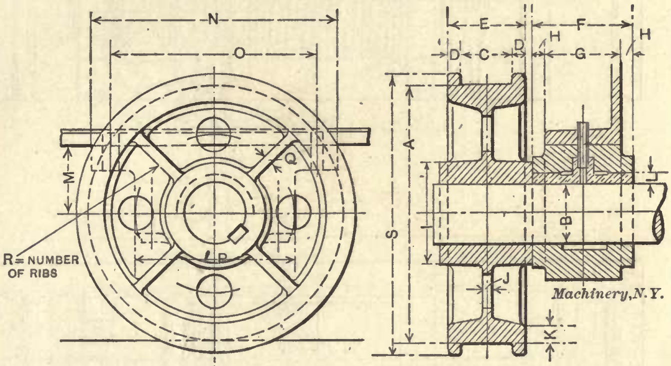
* Machinery, February, 1909.

limited to 4½ or 5 tons per square inch in order to avoid possible deflection and subsequent binding of shafts and attendant disadvantages.

Running Wheels and Axles

The first question to be considered concerning the crab running wheels is the material, and since they are subject to considerable wear their durability ought to be considered. For the best class of work, cast-iron runners should not be used for cranes above 10 tons; other-

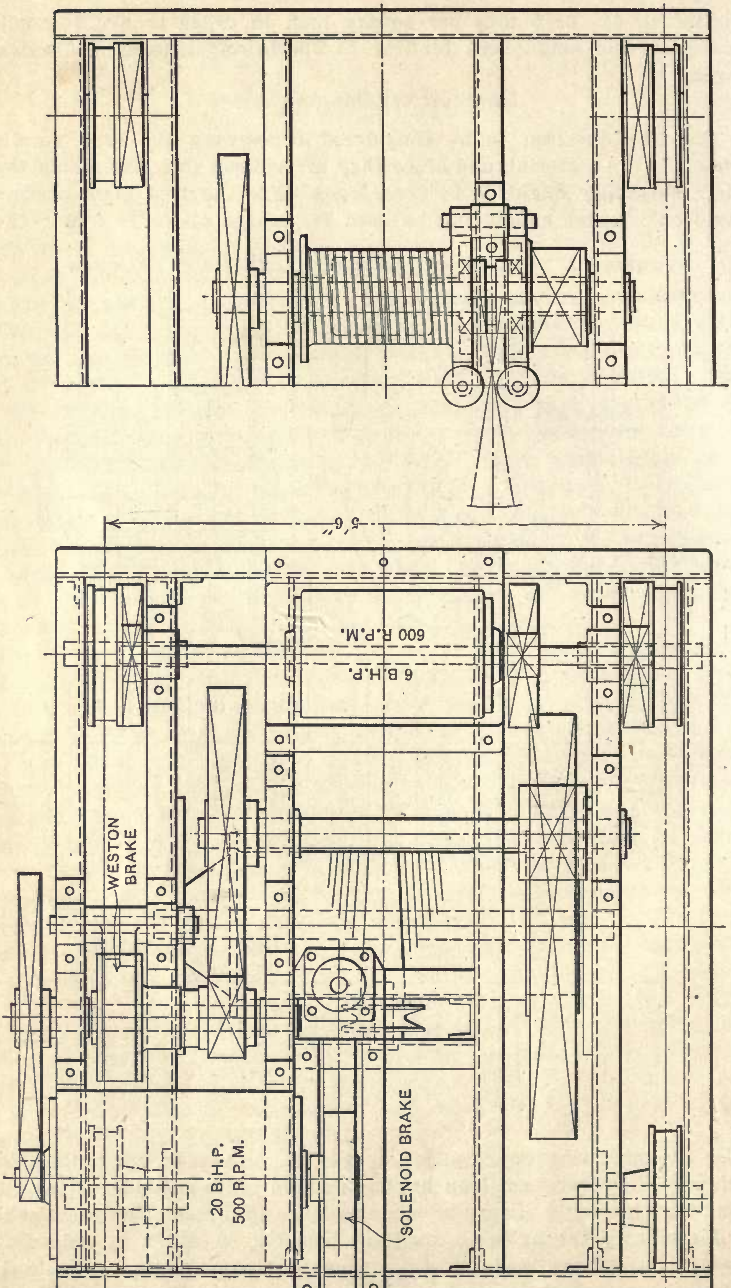
TABLE III. DIMENSIONS OF RUNNERS, AXLES AND BEARINGS



Size of Crane, tons.	A	B	C	D	E	F	G	H	I	J
5–7½	12	3¼	2½	⅝	3¾	4½	3½	½	6	⅝
10–15	15	3¾	2½	¾	4	4½	3½	½	6	⅝
20–25	15	4	2¾	¾	4¼	5	3½	¾	6¾	⅝
30–40	18	5¼	2¾	⅞	4½	5½	4	¾	8¾	¾
50–60	21	5¾	2¾	1	4¾	6	4	1	9¼	⅞

K	L	M	N	O	P	Q	R	S	Material.
¾	$\frac{5}{16}$	3	14½	11¼	8½	⅝	4	13½	Cast iron
¾	⅜	3½	15½	12½	9½	⅝	4	16½	Cast steel
⅞	⅜	4	16½	13½	10¼	⅝	4	16½	Cast steel
1	$\frac{7}{16}$	5	20	16¾	13¼	¾	5	20	Cast steel
1¼	½	6	21¼	17½	14½	⅞	5	23	Cast steel

wise the tread may wear quickly, due to the pressure. For crabs below this size, however, cast iron has been found quite suitable. Crab running wheels should always be made as large in diameter as practicable, and similarly the axles as small as possible, in order to reduce the tractive resistance and consequent current consumption. Table III gives the principal dimensions of runners as usually made, and also the sizes of axles and bearings, which have been calculated as follows:


5' 6"
600 R.P.M.
6 B.H.P.
WESTON BRAKE
20 B.H.P.
500 R.P.M.
SOLENOID BRAKE

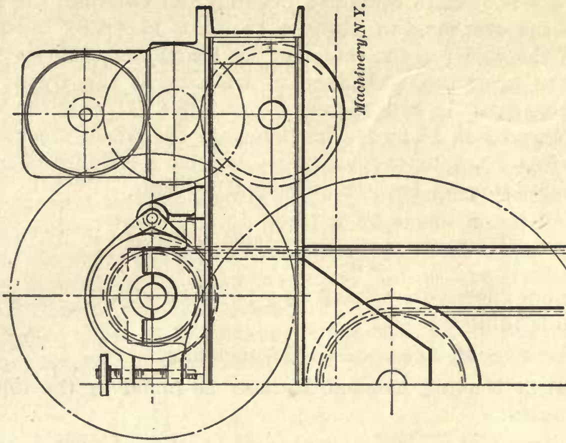


Fig. 5. Crab, similar to the one shown in Fig. 4, slightly altered to accommodate Auxiliary Barrel for Light Loads

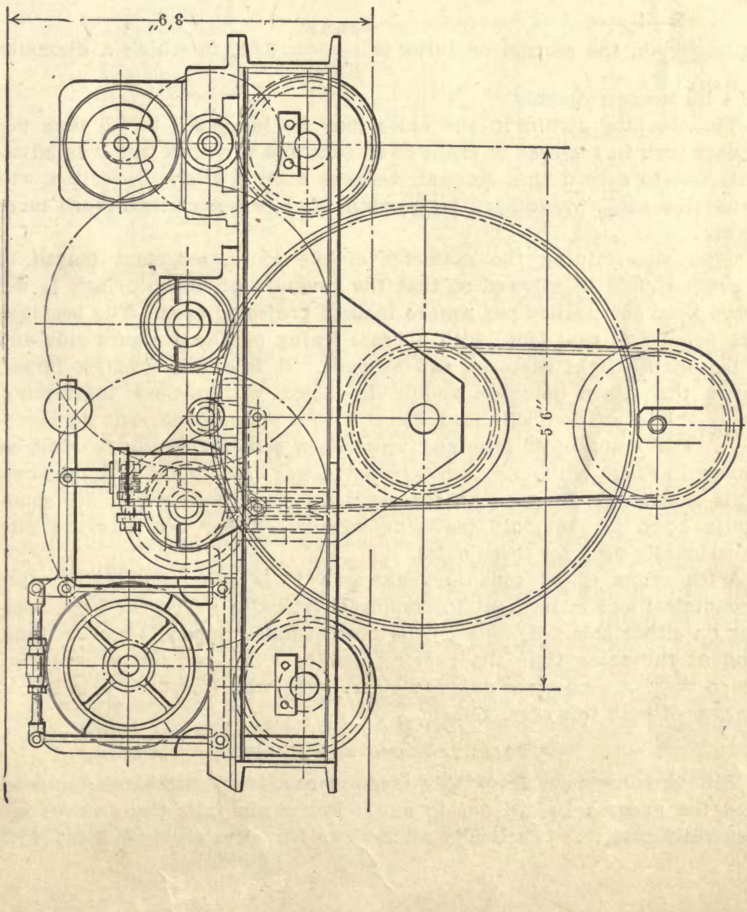


Fig. 4. Crab for 20-ton Three-motor Electric Overhead Crane with Steel Frame

The axles are subject to combined bending and twisting, the former being due to the overhanging distance, which is generally taken from the center of the wheel to the center of the bearing to ensure stiffness, and the latter from the resistance to traction at the tread of the wheel. For example, it will be seen from Table III that the runner for a 20-ton traveler is 15 inches in diameter; the wheel pressure will be about 6½ tons. Taking the maximum possible resistance to traction at 90 pounds per ton, we have the torsional moment,

$$M_t = \frac{90 \times 6.5 \times 7.5}{2240} = 1.95 \text{ inch-ton.}$$

The maximum effective overhang may be taken at 5 inches, and consequently the bending moment

$$M_b = 6.5 \times 5 = 32.5 \text{ inch-tons.}$$

The equivalent bending moment M_e may be found by the following common formulas:

$M_e = \frac{1}{2} M_b + \frac{1}{2} \sqrt{M_b^2 + M_t^2} = 16.25 + \frac{1}{2} \sqrt{32.5^2 + 1.95^2} = 16.25 + 16.3 = 32.55$ inch-tons. Assuming a stress on the material of 5 tons per square inch, the section modulus is $\frac{32.55}{5} = 6.51$, to which a diameter of 4 inches corresponds.

The working strain in the axles may be increased to 5.5 tons per square inch in the case of crabs of 25 tons and upwards, but it is advisable not to exceed this amount, because even a slight deflection will cause the shaft to bind in the bearings and consequently absorb more power.

After determining the diameter of the axles, sufficient length of journal should be allowed so that the pressure on the bearings is not more than 900 pounds per square inch of projected area. The bearings are usually of cast iron, with a brass lining on the pressure side and fitted with a light cast-iron cap beneath. It is of the greatest importance that these journals should be fitted with proper lubricators, since this provision will lead directly to a reduced current consumption. For crabs of 40 tons and upwards a self-lubricating bearing, as shown in Fig. 6, although more expensive, has been found advantageous. This bearing is designed on the same lines as those which are sometimes fitted to the main traveling wheels. Roller bearings are also occasionally used for these axles.

With crabs of 60 tons and upwards it is generally found more economical and convenient to employ short axles supported by a bearing on either side. By this means much smaller journals can be used, and at the same time the gearing can be more compactly arranged, since there are no cross axles to clear. One wheel on each side has to be geared with this arrangement.

Barrels, Ropes and Chains

Lifting barrels are invariably made of cast iron, cast blank for rope, and the grooves turned out to suit. For chain falls the grooves are generally cast in. Practically all modern travelers are now fitted with wire rope falls, the alternatives being ordinary chain and pitch link or Gallé chain, the latter being used mostly on the European continent, or for special cranes. Table V gives particulars of steel ropes suitable for lifting purposes and drums or barrels for same. The factor of safety usually adopted for lifting rope is eight, which allows a good margin of strength even after a few strands have broken. The life and value of the rope depends to a great extent on the size of the barrel and pulleys around which it has to pass. Some rope makers recommend a barrel diameter of six and a half times the circumference of the rope. This is quite suitable and convenient for cranes using ropes under 3½ inches circumference, but above this size it has been found satisfactory to make the barrel and pulleys from five and a half to six times the circumference.

Another point of importance is the spacing of the ropes or centers of the grooves. For all sizes up to 4 inches circumference it is neces-

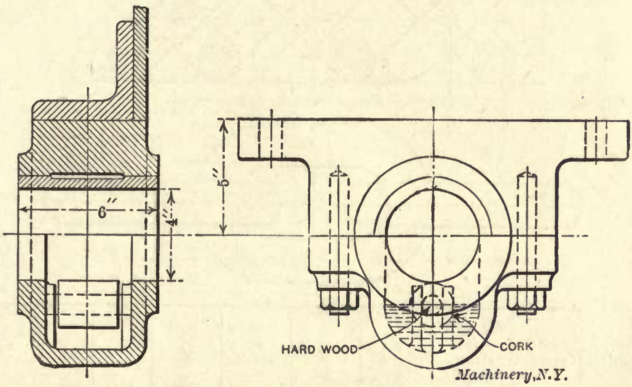


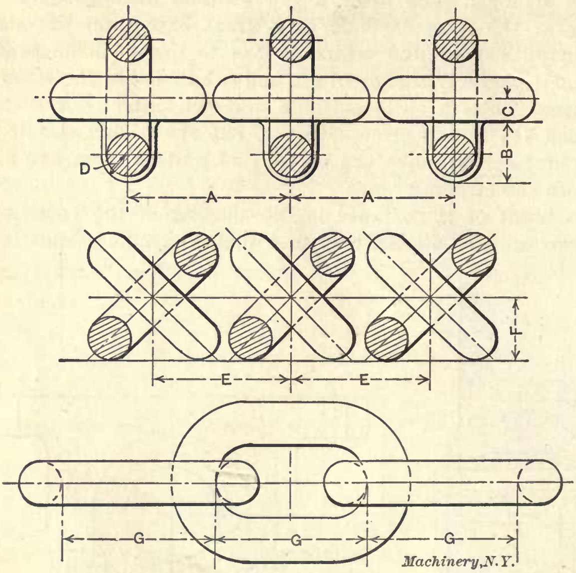
Fig. 6. Self-lubricating Bearing for Shafts in Crabs of 40-ton Capacity and Larger

sary to allow one-eighth inch between the ropes, but above that size three-sixteenths inch gap should be provided, if possible. This allowance is due to the fact that the ropes flatten out slightly under the load, and if there is not sufficient side clearance they grind against each other and are likely to break some of the strands.

When chain is used it is of the type known as short-link crane chain, having a breaking strain of about 23 tons per square inch, and is usually stressed from four and a half to five tons per square inch under the working load. Table IV gives working loads and dimensions of standard crane chains; and shows the spacing and leading dimensions of suitable chain barrels. The length of the lifting barrel is a quantity which naturally depends on the height of lift, but when designing a standard traveler it is advisable to allow for sufficient rope to give a vertical lift of thirty feet with two spare coils on either end, this being the maximum height required under ordinary conditions.

Chain falls should be avoided for longer lifts than twenty feet, otherwise the barrels become inconveniently large. The principal reason for providing the spare coils referred to is to reduce the strain on the anchors, which, however, should always be strong enough to take the full load. Several methods of anchoring the rope are in use, of which probably the two methods shown in Fig. 7 are the best. In either case,

TABLE IV. DIMENSIONS OF CRANE CHAINS

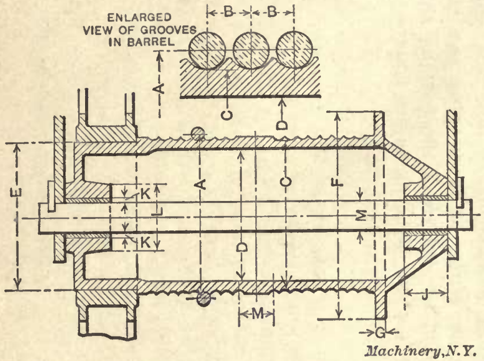


Size, inches.	Load, tons.	Weight per Foot, pounds.	A	B	C	D	E	F	G
$\frac{3}{8}$	1	1.5	$1\frac{7}{16}$	$\frac{17}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	$1\frac{1}{8}$	$\frac{1}{2}$	1
$\frac{7}{16}$	1.3	2.0	$1\frac{5}{8}$	$\frac{19}{32}$	$\frac{7}{32}$	$\frac{9}{32}$	$1\frac{9}{32}$	$\frac{37}{64}$	$1\frac{1}{8}$
$\frac{1}{2}$	1.7	2.7	$1\frac{13}{16}$	$\frac{21}{32}$	$\frac{1}{4}$	$\frac{5}{16}$	$1\frac{1}{2}$	$\frac{11}{16}$	$1\frac{3}{8}$
$\frac{9}{16}$	2.1	3.3	$2\frac{1}{16}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{11}{32}$	$1\frac{5}{8}$	$\frac{3}{4}$	$1\frac{9}{16}$
$\frac{5}{8}$	2.7	4.1	$2\frac{1}{4}$	$\frac{13}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$1\frac{25}{32}$	$\frac{53}{64}$	$1\frac{5}{8}$
$\frac{11}{16}$	3.3	4.8	$2\frac{1}{2}$	$\frac{29}{32}$	$\frac{11}{32}$	$\frac{13}{32}$	$1\frac{15}{16}$	$\frac{29}{32}$	$1\frac{7}{8}$
$\frac{3}{4}$	3.9	5.9	$2\frac{11}{16}$	$\frac{31}{32}$	$\frac{3}{8}$	$\frac{7}{16}$	$2\frac{1}{8}$	1	$2\frac{1}{8}$
$\frac{13}{16}$	4.6	6.8	$2\frac{7}{8}$	$1\frac{1}{32}$	$\frac{13}{32}$	$\frac{15}{32}$	$2\frac{1}{4}$	$1\frac{1}{16}$	$2\frac{1}{8}$
$\frac{7}{8}$	5.4	8.1	$3\frac{1}{16}$	$1\frac{3}{32}$	$\frac{7}{16}$	$\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{3}{16}$	$2\frac{3}{8}$
$\frac{15}{16}$	6.2	9.0	$3\frac{1}{4}$	$1\frac{5}{32}$	$\frac{15}{32}$	$\frac{17}{32}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$2\frac{1}{2}$
1	7.0	10.3	$3\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{9}{16}$	$2\frac{3}{4}$	$1\frac{5}{16}$	$2\frac{1}{2}$
$1\frac{1}{16}$	8.0	11.3	$3\frac{5}{8}$	$1\frac{9}{32}$	$\frac{17}{32}$	$\frac{19}{32}$	$2\frac{15}{16}$	$1\frac{13}{32}$	$2\frac{9}{16}$
$1\frac{1}{8}$	8.9	12.5	$3\frac{3}{4}$	$1\frac{5}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$3\frac{1}{16}$	$1\frac{15}{32}$	$2\frac{15}{16}$
$1\frac{3}{16}$	9.9	14.3	4	$1\frac{13}{32}$	$\frac{19}{32}$	$\frac{21}{32}$	$3\frac{1}{4}$	$1\frac{9}{16}$	$3\frac{5}{16}$
$1\frac{1}{4}$	10.9	15.8	$4\frac{1}{4}$	$1\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$	$3\frac{7}{16}$	$1\frac{21}{32}$	$3\frac{3}{8}$

a solid cast-iron eye is woven into the end of the rope, and a hole drilled in the eye for a turned pin or stud. This makes a very substantial connection.

It has become the universal practice to so arrange the lifting ropes or chains that the load will be lifted centrally and thereby be dis-

TABLE V. DIMENSIONS OF LIFTING BARRELS



Load, tons.	Number of Ropes.	Size, Circumference, inches.	A	B	C	D	E	F	G	H	J	K	L	M
3	2	$1\frac{3}{4}$	12	$\frac{5}{8}$	11.45	$10\frac{1}{8}$	$9\frac{7}{8}$	$12\frac{1}{2}$	$\frac{5}{8}$	$2\frac{1}{2}$	$3\frac{1}{2}$	$\frac{5}{16}$	5	$2\frac{1}{2}$
5	2	$2\frac{1}{4}$	15	$\frac{3}{4}$	14.30	13	$12\frac{3}{4}$	$15\frac{3}{4}$	$\frac{3}{4}$	$2\frac{3}{4}$	$3\frac{1}{2}$	$\frac{5}{16}$	$5\frac{1}{2}$	$2\frac{1}{2}$
7.5	4	2	12	$\frac{11}{16}$	11.37	$10\frac{1}{8}$	$9\frac{7}{8}$	$12\frac{5}{8}$	$\frac{5}{8}$	$2\frac{3}{4}$	$3\frac{1}{2}$	$\frac{5}{16}$	$5\frac{1}{2}$	$2\frac{1}{2}$
10	4	$2\frac{1}{4}$	15	$\frac{3}{4}$	14.30	$12\frac{3}{4}$	$12\frac{1}{2}$	$15\frac{3}{4}$	$\frac{3}{4}$	$2\frac{3}{4}$	$3\frac{1}{2}$	$\frac{5}{16}$	$5\frac{1}{2}$	$2\frac{1}{2}$
15	4	$2\frac{3}{4}$	$16\frac{1}{2}$	1	15.62	$14\frac{1}{8}$	$13\frac{3}{4}$	$18\frac{1}{2}$	$\frac{7}{8}$	$3\frac{1}{8}$	4	$\frac{3}{8}$	$6\frac{1}{4}$	3
20	4	$3\frac{1}{4}$	18	$1\frac{1}{8}$	17.00	$15\frac{1}{4}$	$14\frac{3}{4}$	$19\frac{1}{2}$	1	$3\frac{1}{4}$	4	$\frac{3}{8}$	$6\frac{1}{2}$	3
25	4	$3\frac{1}{2}$	21	$1\frac{1}{4}$	19.90	18	$17\frac{1}{2}$	$22\frac{1}{4}$	$1\frac{1}{8}$	4	$4\frac{1}{2}$	$\frac{7}{16}$	8	4
30	4	4	24	$1\frac{3}{8}$	22.70	$20\frac{1}{2}$	20	$25\frac{1}{2}$	$1\frac{1}{4}$	$4\frac{3}{4}$	$4\frac{1}{2}$	$\frac{7}{16}$	$8\frac{3}{4}$	4
40	4	$4\frac{1}{2}$	27	$1\frac{1}{2}$	25.50	23	$22\frac{1}{2}$	$28\frac{3}{4}$	$1\frac{1}{2}$	5	5	$\frac{7}{16}$	$9\frac{1}{4}$	$4\frac{1}{2}$
40	4	5	30	$1\frac{3}{4}$	28.40	30	$29\frac{1}{2}$	32	$1\frac{1}{2}$	$5\frac{1}{2}$	$5\frac{1}{2}$	$\frac{7}{16}$	10	$4\frac{1}{2}$

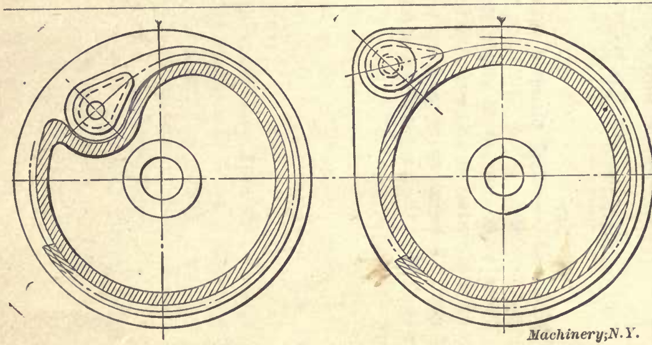


Fig. 7. Two Methods in use for Anchoring the Lifting Ropes

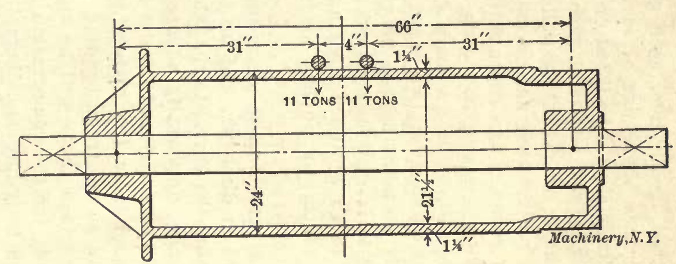


Fig. 8. Barrel for a 40-ton Overhead Electric Crane

tributed equally over each girder. This arrangement is necessary for all cranes above five or seven tons, from which size up to fifty tons the loads should be lifted on four parts of rope, two parts being coiled in right- and left-hand grooves on the barrel, the other end passing around a compensating pulley, as will be seen by referring to Fig. 4. This pulley need not be more in diameter than about twice the circumference of the rope, since it is not subjected to any motion. For cranes up to and including three tons the load should be lifted on a single fall, while for loads of five, six, and seven tons it is more convenient to employ two parts, one coiling on the barrel. There is practically no advantage in providing a central lift for these light cranes, since it is generally impracticable to make the girders light enough to give an ordinary working strain, and the little extra weight that will be thrown on one girder will not destroy the economy of construction.

The barrels used for ordinary overhead travelers seldom exceed 30 inches in diameter, and up to this size, the method shown in Table V of fixing the barrel to the wheel has been found to be the cheapest, and is almost universally adopted. The thickness of metal in the barrel should be sufficient to resist the maximum bending which occurs when the load is in the highest position and the ropes are in the center. If the stress from bending is not greater than about 0.75 ton per square inch there will be an ample margin of strength left to safely provide for the compression from the ropes themselves. For example, take a forty-ton crane barrel as shown in Fig. 8. The maximum span over which bending takes place is from center to center of the bearings, and the maximum bending moment $M_b = 11 \times 31 = 341$ inch-tons.

$$\text{Section modulus of annulus} = \frac{\pi}{4} \times \frac{R^4 - r^4}{R} = \frac{\pi(12^4 - 10.75^4)}{4 \times 12} = 483.$$

$$\text{Stress} = \frac{341}{483} = 0.7 \text{ ton per square inch.}$$

The size of the barrel shaft should be sufficient to take the bending from the center of the support to the center of the bearing at a stress of about 5 tons per square inch, and the length of the bearing should be such that the pressure on same is about 900 pounds per square inch.

CHAPTER III

GEARING*

The gearing may be regarded as one of the most important details of crane design, since on its merits rest the efficiency and safety of the crane. A few years ago all lifting machinery was worked by wheels having cast teeth, but with the introduction of electric driving, and higher speeds, machine-cut gearing became necessary. The adoption of cut gears has grown to such an extent that a large quantity of cranes are now supplied with machine-cut gears throughout. In deciding how many of the gears of a crane ought to have the teeth machine cut, some consideration should be given to the work the crane is likely to have to do. If the speeds are high, or the crane is going to be in continual use, it will pay to have as many of the gears machine cut as possible, because the subsequent saving in electric current will soon pay for the extra first cost.

For constantly-working high-speed cranes up to seven tons load, it will be found most satisfactory to have machine-cut gears throughout, including the barrel wheel and pinion. For all sizes above seven tons, it is not advisable to machine cut the barrel gear and pinion even for the best class of work, partly owing to the fact that it is desirable to shroud the pinion of this pair of wheels; as the speed is low, the loss in power due to friction is not worth considering. For cheap cranes, and those required for intermittent working, as in engine rooms, for instance, it is necessary to have only the first reduction for each motion machine cut.

Rawhide or buffoline motor pinions have been found suitable, and are to be recommended for the first reduction for cranes up to twenty tons load. The gear with which this pinion meshes should be of cast iron, since it has been found that the strength of rawhide and cast-iron teeth are about the same. The principal advantages of rawhide pinions are that they run almost noiselessly and do not require any lubricant. A gear case need not be provided for this reduction unless the crane is working in a very damp location. For cranes above twenty tons the motor pinion ought to be of machine steel, running with a cast-iron or steel gear, preferably in an oil bath, the latter taking the form of a cast-iron gear case.

It is generally found advisable to make small machine-cut pinions of machine steel, the blanks usually being cut from ordinary rolled bar, or forged to size. When this is done, the pinion teeth are considerably stronger than those of the gear, if this is made of cast iron, and about equal in strength to those of a cast-steel gear.

For cranes up to twenty tons the barrel gear and pinion can be of cast iron; above this size cast steel is preferable, and in all cases

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* MACHINERY, March, 1909.

above seven tons the pinion should be full shrouded, therby making the teeth about equal in strength to those of the gear. For cranes above twenty tons it is considered good practice to make all pinions of steel, whether they are machine cut or not; while for cranes which are very severely handled and constantly on full load, as, for instance, steel works and forge cranes, the purchaser will probably find it most satisfactory in the end to have steel gears throughout for all motions. It will be found satisfactory for ordinary work to have cast-iron gears for cranes up to twenty tons, and above this size and up to forty tons, to have all pinions and the barrel gear of steel, and the remainder cast iron. Cranes above forty tons should preferably have steel gears throughout.

In order to keep down the size of the crab in every way, the pinions should be kept as small as possible consistent with smooth running, but at the same time no pinion should ever have less than twelve teeth, since below this size pinions run badly and are very weak, and cutters for a smaller number of teeth are not usually available.

Strength of Teeth in the Gears

The most important question concerning the gearing lies in the strength of the teeth, and a great variation is found in practice in the stress to which the material is subjected. Upon examining the practice of various firms, one finds that while some stress the cast-iron barrel gear teeth to one ton per square inch, and the steel gear teeth to three tons per square inch, others work to as much as three and six tons per square inch, respectively. Now the average ultimate strength of cast iron and cast steel subject to bending, as in a tooth, is eighteen and thirty tons, respectively, but it is possible for either of these values, and particularly that of cast iron, to be considerably reduced by a lack of homogeneity in the metal, which may never be detected, even if it is not wholly internal. With due consideration to the above fact, one is justified in allowing a factor of safety of 8 for cast iron and 6 for steel for slow running.

Convenient tables may be compiled for rapidly arriving at the pitch and width of the teeth, but even when referring to these it is advisable to know how to calculate the strength. The teeth of gearing used for crane work are of the involute pattern with radial flanks which gives a fairly short tooth with a broad root; these teeth are consequently of the greatest strength.

Prof. Unwin states that the load which falls upon one tooth lies between one-half the full load and the total load, and is generally taken at two-thirds the full load; but if the pinion is small, it is common practice to consider the full load as taken by one tooth. When the pinion is of steel and the gear of cast iron, or the pinion is full shrouded, the gear teeth can be considered weaker or equal to those of the pinion, and the strength calculated accordingly to suit the shape of the teeth in the gear. Prof. Unwin also states that the agreed percentage of the full load will act, at one particular period, on the full length of the tooth, and this should be allowed for.

In calculating the strength of a tooth, it must be regarded as a cantilever, and made strong enough to resist the consequent bending moment. As an example, find the stress in the teeth of a steel barrel gear having 108 teeth of 1¾-inch circular pitch by 5 inches wide, and keyed onto the barrel of a 30-ton crane; diameter of barrel, two feet, fifteen tons load on the barrel. The diameter of the gear is approximately 5 feet.

$$\text{Load on tooth} = \frac{15 \times 2}{5} = 6 \text{ tons.}$$

$$\text{Maximum load at full length of tooth} = \frac{2}{3} \times 6 = 4 \text{ tons.}$$

Full length of tooth = 1.09 inch.

Bending moment = 4 × 1.09 = 4.36 inch-tons.

Modulus of tooth at root = 1.03.

$$\text{Stress} = \frac{4.36}{1.03} = 4.2 \text{ tons per square inch,}$$ which is a suitable working stress for slow running cast-steel gears.

Some makers shroud the barrel gear and pinion to the pitch line, but if the strength has been calculated as shown above, this form of shrouding does not add to the strength required to resist bending, because if the bending moment and modulus are taken at the pitch line, they will be in similar proportion to those found at the root. The principal value of half shrouding is in minimizing the tendency the teeth have to break across the corner, especially with cast gears where the teeth may not bear evenly together, or may not be parallel to each other.

For heavy cranes, say 30 tons and upwards, double helical gears for the barrel gear and pinion have been used with advantage, since they insure a freedom from shock. There is no doubt that helical gearing is much stronger than spur gearing, but there are many opinions as to their relative values. Since the points of contact on a well-made tooth of this type at any moment are distributed over the whole of the working face, from root to point, the average leverage of the whole load is only half that of ordinary spur gears. Then, again, the developed width of these teeth is more for a certain width of wheel, and in calculating the strength this can be taken into account. It is not safe, however, to allow in full for all these advantages over ordinary spur gears, owing to the fact that in practice it has been found difficult to make the apices of each pair of teeth run in the same plane, and hence the load may be thrown on one side of the gear only. This difficulty may be avoided to some extent by allowing a little lateral play on the pinion shaft whereby the pinion will adjust its position to suit the gear, and consequently equalize the stresses somewhat. In the absence of any reliable data on the strength of these gears, they are seldom considered as more than one and one-half times as strong as ordinary spur gears.

In calculating the pressure on the teeth of any pair of gears, with the exception of the barrel gear and pinion, it is more correct to allow for the load due to the maximum torque of the motor, than to take the reaction from the load, since this latter does not allow for the resistance due to the friction of the intermediate gearing.

The stresses to which gear teeth may be subjected depend principally on the pitch line speed and the nature of the work to be done; *i. e.*, whether running under a steady load in one direction, or subject to varying loads and quick reversal, as in cranes.

The most convenient way for calculating the strength of gear teeth is by the Lewis formula. This formula is:

$$W = SPFY,$$

in which

W = force at pitch line in pounds,
P = circular pitch,
S = allowable fiber stress for the material used, in pounds per square inch (see Table VI),
F = width of face of gear,
Y = the factor known as the Lewis outline factor which varies with the number of teeth and the form of gear tooth.*

TABLE VI. ALLOWABLE FIBER STRESS IN GEAR TEETH.

Material.	Speed at Pitch Line in Feet per Minute.							
	100	200	300	600	900	1200	1800	2400
Cast iron........	4800	4200	3800	3200	2400	1920	1600	1360
Cast steel........	12000	10500	9600	8000	6000	4800	4000	3400
Gun-metal.......	7200	6300	5760	4800	3600	2880	2400	2040
Machine steel....	19200	14410	11200	9600	7200	5760	4800	4080

Applying this formula to the previous example, we have: $S = 12{,}000$ (from Table VI); $P = 1.75$; $F = 5$; and $Y = 0.118$, for 108 teeth and 15-degree involute gearing. Consequently,

$$W = 12{,}000 \times 1.75 \times 5 \times 0.118 = 12{,}390 \text{ pounds.}$$

Dimensions of Arms, Rim and Hub

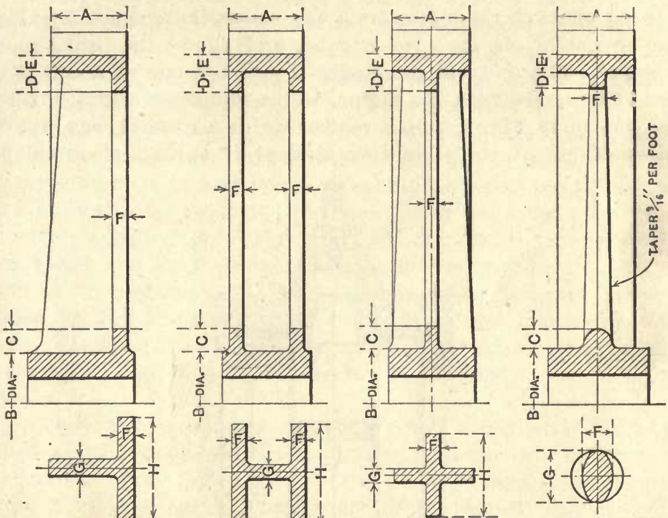
The proportions of the arms, rim and boss of a gear are often determined by practical considerations of casting, but at the same time the strength of the arms may be calculated, and it is well to do this for large wheels transmitting heavy loads. There are several different kinds of sections of arms in use. The I-section is probably the most extensively used for machine molded gears in England, but it is not as commonly used in the United States; it is very strong, but it is the most expensive, since the space between the arms must be cored. The T-section and +-section are often used, but are not as strong as the I-section. Oval and rectangular sections are coming more into use, and possess the advantages of strength, cheapness in patterns, and simplicity in molding—an advantage which, in turn, gives good castings.

* The subject of the strength of gear teeth is thoroughly treated in MACHINERY'S Reference Series No. 15, Spur Gearing, second edition, Chapter IV.

To calculate the strength of the arms, it is necessary to find the bending moment at the root; it is generally agreed that each arm takes its share of the peripheral load, therefore, the bending moment in each arm equals the load at the pitch line in tons multiplied by the distance from the pitch line to the root of the arm in inches, and divided by the number of arms. For small gears it is not necessary to make this calculation, but proportion each part of the arms, rim or

TABLE VII. DIMENSIONS OF GEAR RIMS AND ARMS.

P = circular pitch ; b = bore.



Machinery, N.Y.

Form of Arm.	A	B	C	D	E	F	G	H (at rim)
T section..........	$3P$	$1\frac{7}{8}b$	$\frac{5}{8}P$	$\frac{9}{16}P$	$\frac{1}{2}P$	$\frac{1}{2}P$	$\frac{1}{8}P$	$2\frac{1}{4}P$
I-section..........	$3P$	$1\frac{7}{8}b$	$\frac{5}{8}P$	$\frac{9}{16}P$	$\frac{1}{2}P$	$\frac{1}{2}P$	$\frac{1}{8}P$	$2\frac{1}{4}P$
+-section........	$3P$	$1\frac{7}{8}b$	$\frac{5}{8}P$	$\frac{9}{16}P$	$\frac{1}{2}P$	$\frac{1}{2}P$	$\frac{1}{8}P$	$2\frac{1}{4}P$
Oval section......	$3P$	$1\frac{7}{8}b$	$\frac{5}{8}P$	$\frac{9}{16}P$	$\frac{1}{2}P$	$\frac{1}{2}G$	$2\frac{1}{8}P$	.....

boss to some function of the pitch. There are several tables of these proportions used in practice, and Table VII may be taken to represent a fair average.

The number of arms generally adopted in crane practice is:

Four for gears up to 3 feet diameter.

Six for gears from 3 feet to 6 feet diameter.

Eight for gears above 6 feet diameter.

Gears below one foot diameter are generally made with a solid plate web, possibly relieved with some holes. The thickness of this web should be about one-half the pitch, and the proportion of rim and boss should remain practically the same as for the armed wheels.

CHAPTER IV

BRAKES AND BRAKE MECHANISM*

Brakes for electric cranes may be divided into two types, *viz.*, solenoid, or magnetic brakes, and mechanical brakes. Electrical or magnetically operated brakes are generally ordinary strap or clamp brakes, which are held off by the action of a magnet or solenoid, electrically connected with the motor in such a manner that when the current is cut off from the motor from any cause, the magnet releases the spring or weight, as the case may be, and allows the brake to come into action. The solenoid commonly in use consists of a coil of wire connected in series with the motor, and a plunger working inside the coil as shown in Fig. 9, which represents in a general way, the form of solenoids manufactured in several sizes by various electrical firms.

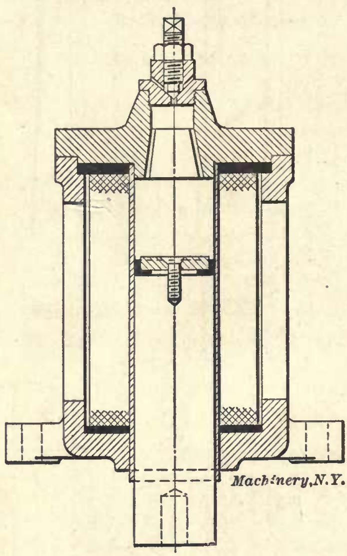


Fig. 9. Solenoid used to Operate Brake for Electric Cranes

Solenoids should be so proportioned that their action is not delayed when the current has been cut off, due to residual magnetism. On the other hand, a too rapid application of the brake is to be avoided, since it has occasionally bent armature shafts. To effect this end the solenoid forms in itself a dashpot, the air being throttled in the small hole at the top of the body. Arrangements are usually made in the winding of the solenoid to enable it to lift off the brake when the controller is on the first contact, otherwise the motor would drive against the brake.

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* MACHINERY, April, 1909.

For cranes above five tons capacity the solenoid brake is applied principally as a means of stopping the motor quickly, to facilitate rapid reversal, and it should, of course, always be powerful enough to hold the full load in the event of the mechanical brake failing. The disadvantages of relying entirely on a solenoid brake are due to the fact that it does not permit of steady lowering, since when the motor is reversed, the brake lifts entirely off and consequently allows the load to run down unchecked until the current is cut off again. This arrangement has been found fairly satisfactory for cranes under five tons (and is, in fact, often used on larger cranes), one reason being that the friction of the crab itself helps to retard a light load to some extent. A better method of lowering for small cranes is shown in Fig. 10, where an ordinary solenoid brake is used which can be released and controlled when lowering the load.

When a crane is fitted with both a solenoid brake and an automatic mechanical brake, the principal function of the solenoid brake, as already stated, is to absorb the momentum of the armature and gear and stop the motor rapidly, and since this action must take place with the motor running in either direction, the clamp type of brake has been found the most suitable, although the ordinary type of strap brake is frequently used. The solenoid brake is most conveniently applied on the armature shaft itself, since the momentum is more readily absorbed at that point and a smaller solenoid can consequently be employed; but it is a mistake to cut the size of the solenoids too close.

For convenience, motors are sometimes made with the shaft extended at both ends, so that the driving pinion can be fixed on one end and the brake pulley on the other; this arrangement obviates the necessity of coupling a short shaft to the motor and providing an extra and somewhat expensive outer bearing. Brake pulleys are made of cast iron, and, while they should be as large as possible, in order to reduce the tangential force, at the same time the peripheral speed should not, if possible, be more than 2000 to 2500 feet per minute, or an inconvenient amount of heat will be produced. In any case, they will, of necessity, become more or less heated, a fact which makes timber-lined brakes preferable to leather-lined ones, because the timber absorbs more oil and consequently does not dry up and wear so quickly as does leather. The following sizes of brake pulleys have been found convenient for the sizes of motors given below, the full load speed being limited to 750 revolutions per minute for motors up to 20 B. H. P., and to 500 revolutions for those above.

Brake Horsepower	Diameter, Inches	Brake Horsepower	Diameter, Inches
5	10	30	18
10	12	35	18
15	12	40	21
20	15	45	24
25	15	50	24

A compact and typical design of a clamp brake is shown in Fig. 11.

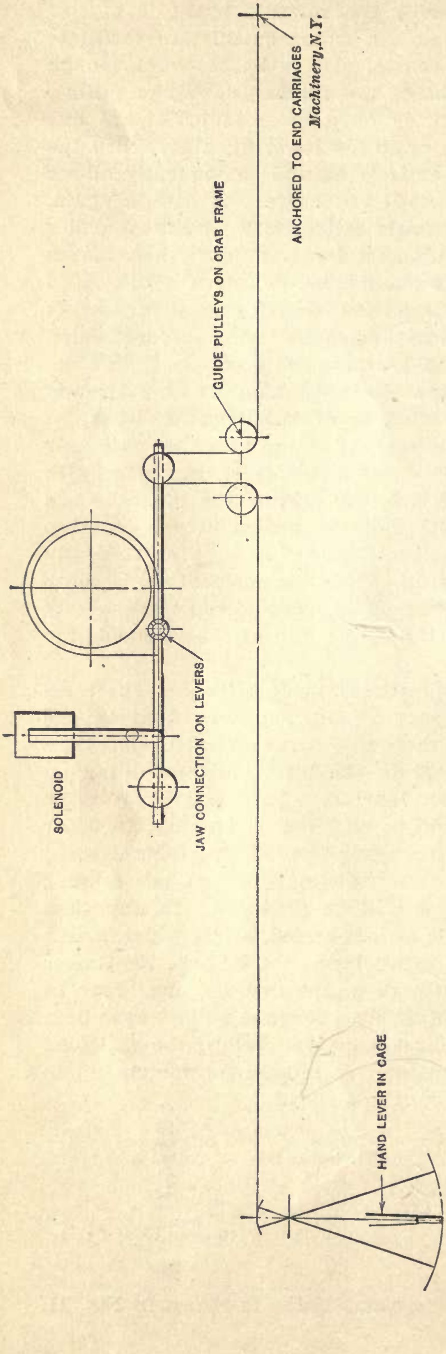


Fig. 10. Solenoid Brake, which can be Released and Controlled when Lowering the Load

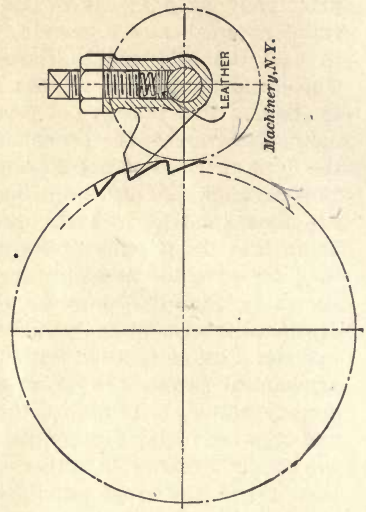


Fig. 12. Arrangement for Releasing Ratchet when Load is Lifted

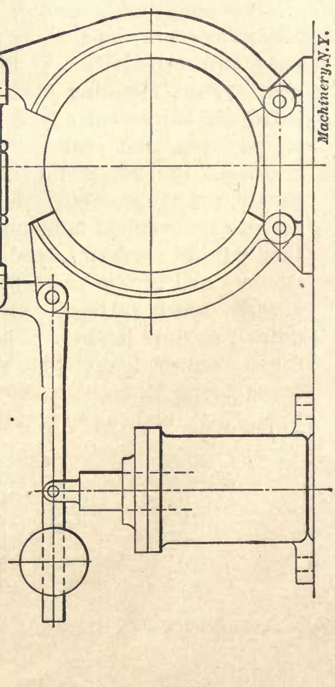


Fig. 11. Typical Design of Compact Clamp Brake for Electric Cranes

The calculations for all types of clamp brakes are identical, and the horsepower of the motor being given, it is necessary in the first place to find the size of the solenoid. Take for example a 20 B. H. P. motor running at 500 revolutions per minute. If the brake pulley is 15 inches diameter, the tangential effort will be

$$\frac{20 \times 33{,}000 \times 12}{500 \times \pi \times 15} = 335 \text{ pounds.}$$

The safe coefficient of friction for greasy wood on iron is 0.3; therefore, the force required at the center of the blocks will be $335 \div 0.3 = 1116$ pounds.

The ratio of the levers depends on the stroke of the magnet. Suppose the stroke to be limited to A inches, and the blocks adjusted to lift off B inches; then the ratio will be $A : 2B$, and, consequently, the weight required to stop the motor will be $1116 \div \frac{A}{2B}$ pounds. It is always advisable to allow for a little extra weight, say 25 per cent, in practice, to cover the momentum, etc. A small adjusting weight is added to make up the difference between the weight of solenoid core and the actual weight required.

Several electrical firms supply an armature brake complete and self-contained on the motor, in which case the weight is generally replaced by a spiral spring.

When a strap brake is more suitable, as in the case shown in Fig. 10, the calculations are somewhat different. The size of the pulley remains the same as given in the previous table, and as in the case of the clamp type, wood-lined straps are the most serviceable. Having found the tangential effort on the pulley, as before, it is necessary to proportion the levers to give the required pull on the strap. The quantity depends on the proportion of the pulley enclosed by the arc of contract of the strap, the value of the coefficient of which will be found in Table VIII. In the diagram over this table it will be seen that if the tangential effort on the brake due to the load is acting in the direction shown by the arrow, the fixed end of the strap will be at F and the slack end, that is the end attached to the lever, at x. To find the pull on the slack end it is only necessary to multiply the tangential effort P by the value opposite the angle θ and under the coefficient μ. For example, the brake for a 10 H. P. motor running at 500 revolutions is 12 inches diameter. The tangential effort

$$P = \frac{10 \times 33{,}000}{\pi \times 1 \times 500} = 210 \text{ pounds.}$$

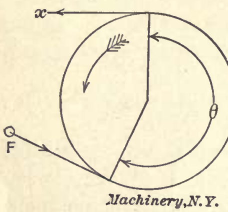
If $\theta = 210$ degrees, the value of $x = 0.5$ for wood blocks on an iron pulley. Therefore the pull at $x = 210 \times 0.5 = 105$ pounds. The pull on the fixed end can be found from the table in a similar manner, although a table is hardly necessary, since it will readily be seen that the pull at F will equal $x + P$, or the pull on the slack end plus the tangential effort.

The above coefficient may be found independently of the table by calculating the ratio of the tension $\frac{T}{T_1}$ in fast and slack ends of strap from the following formulas:

$$\frac{T}{T_1} = 2.718\,\mu \frac{L}{R}$$

where μ = coefficient (0.3 for wood on iron).
L = length of contact in inches.
R = radius of pulley in inches.

TABLE VIII.



μ = coefficient of friction = 0.2 for iron to iron, 0.3 for wood to iron, 0.4 for leather to iron. P = pull on brake rim. x = strain on loose end of brake strap. F = strain on fast end. θ = arc embraced by strap (in degrees). Table gives tension on brake straps when $P = 1$.

θ	Value of x.			Value of F.		
	$\mu = 0.2$	$\mu = 0.3$	$\mu = 0.4$	$\mu = 0.2$	$\mu = 0.3$	$\mu = 0.4$
30°	9.09	5.89	4.29	10.09	6.89	5.29
45	5.89	3.76	2.71	6.89	4.76	3.71
60	4.29	2.71	1.92	5.29	3.71	2.92
75	3.35	2.08	1.45	4.35	3.08	2.45
90	2.71	1.66	1.14	3.71	2.66	2.14
105	2.26	1.37	0.93	3.26	2.37	1.93
120	1.92	1.14	0.77	2.92	2.14	1.77
135	1.66	0.98	0.64	2.66	1.98	1.64
150	1.45	0.84	0.54	2.45	1.84	1.54
165	1.29	0.73	0.47	2.29	1.73	1.47
180	1.14	0.64	0.40	2.14	1.64	1.40
195	1.03	0.56	0.35	2.03	1.56	1.35
210	0.93	0.50	0.30	1.93	1.50	1.30
240	0.76	0.40	0.23	1.76	1.40	1.23
270	0.64	0.32	0.18	1.64	1.32	1.18
300	0.54	0.26	0.14	1.54	1.26	1.14

For example, take the preceding case; then

$$2.718 \times 0.3 \times \frac{22}{6} = 2.99 = \frac{T}{T_1}.$$

$$P = T - T_1;\text{ but } T = 2.99\,T_1.$$

$$\text{Therefore } P = 2.99\,T_1 - T_1 = 1.99\,T_1,\text{ or } T_1 = \frac{P}{1.99}$$

$$\text{Since } P = 210 \text{ pounds, } T_1 = \frac{210}{1.99} = 105 \text{ pounds.}$$

$$\text{And } T = P + T_1 = 210 + 105 = 315 \text{ pounds.}$$

The short lever should be given about half an inch travel, to allow the strap to lift as clear as possible of the pulley, therefore, if the magnet has a stroke of two inches, the ratio of leverage will be 4 to 1, and the pull required 105 ÷ 4 = 26 pounds.

Brake straps should be made of a good quality mild steel or wrought iron; they can then be stressed up to 5 tons per square inch in the net section, and consequently kept light and pliable.

There are several forms of automatic mechanical brakes used on overhead travelers, foremost among which is the screw brake, as shown in Fig. 13. This brake consists of a pinion of phosphor-bronze, or steel, bushed with gunmetal, mounted on a thread which is cut in one of the intermediate driving shafts *B*. The steel or iron casting *C*

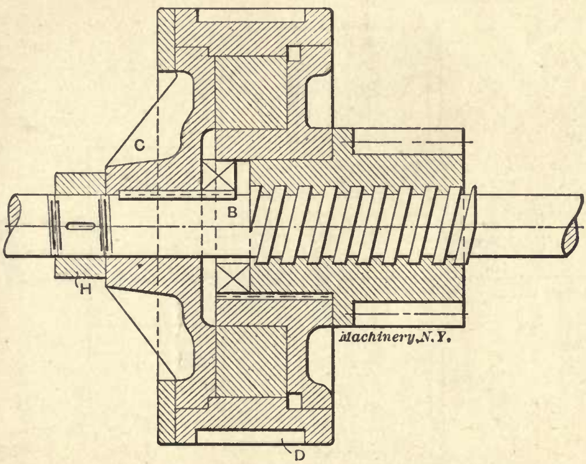


Fig. 13. Automatic Screw Brake for Overhead Electric Cranes

is fixed to the driving shaft by means of a key which allows the necessary lateral movement for the adjustment, regulated by the nut *H*. The ratchet *D*, which is cored out to hold oil, and bushed with gunmetal, runs loose on the boss formed by the jaws. All working faces are preferably lined with gunmetal. One or two pawls, according to the size of the brake, engage with the ratchet, being thrown out of gear by a suitable arrangement.

The action of the brake is as follows: When lifting the load, the resulting pressure on the pinion due to the screw holds it hard up against the ratchet face and, the pawls having been lifted out of gear, the whole brake revolves together without resistance. As soon as the lifting ceases and a slight reverse has taken place, the pawls fall into gear, and the load is held secure by friction. In order to lower the load, the motor must be reversed and run on light power, this having the effect of reducing the pressure on the friction faces and allowing the load to slip steadily.

Some makers employ several friction disks in place of the two faces in the form just described. This design is shown in Fig. 14. This type is more powerful in proportion to its size than the other design, but is preferably enclosed in an oil bath to ensure complete lubrication between all the faces. The two jaws in both types are necessary to prevent the pinion backing from the face when the load is too light to keep them together, as when lowering the empty hook.

To calculate the size of the brake shown in Fig. 14, it is necessary to make the diameter such that, allowing for the lowest possible coefficient of friction, it is always in excess of the reaction from the pinion at the radius to the center of the pressure of the frictional surface.

Let r = radius of pitch circle of pinion,

W = load on teeth,

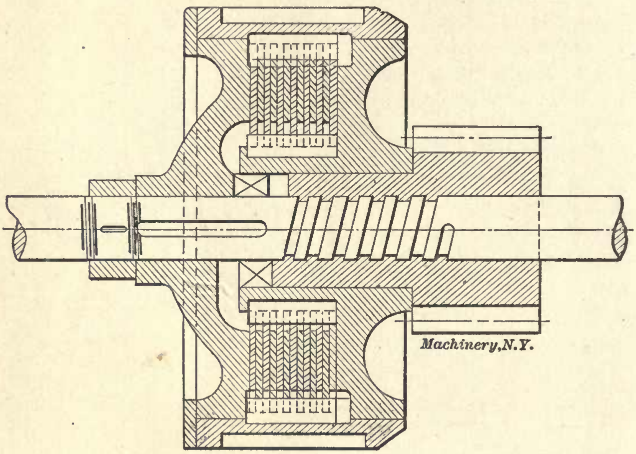


Fig. 14. Friction Brake with a Number of Friction Disks

R = radius to center of pressure of disks.

p = pitch of thread,

μ = minimum possible coefficient of friction = 0.04,

N = number of frictional faces.

$$\text{Then } \frac{W\,r}{R} \text{ must equal } \frac{N\,W \times 2\,r\,\pi}{p} \times \mu.$$

Some arrangements must be adopted for the pawls in order to release the ratchet when the load is being lifted, otherwise an objectionable noise will be made. One common arrangement is shown in Fig. 12 in which there are two pawls, one of which is set at half the pitch of the ratchet and mounted on a shaft which is driven through light spur gearing from the brake shaft, as shown. The pawls are moved by this shaft through the friction due to the leather pads which are pressed onto the shaft by a spring.

Another, and somewhat simpler, construction is that shown in Fig. 15. In this arrangement the weighted pawl A is thrown in and out of gear by the hinged friction clips *B* which are adjusted by means of the spring bolt *C*. In either of the above designs the pawls should always be so constructed that in the event of the friction drive failing they will fall into gear with the ratchet.

Shafts and Bearings

Very little more than a general idea can be given about the shafts and bearings for overhead cranes, since they possess no individual difference to those used for any other type of machinery. Although the calculation of the shafts is a simple matter in itself, it is no uncommon thing to find shafts stressed abnormally high, due to the fact that the size has been guessed at and not checked by calculation. A weak shaft is an annoyance, because quite apart from the

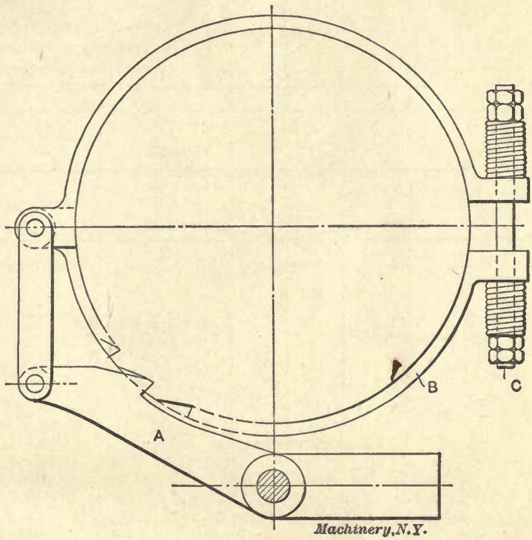


Fig. 15. Another Arrangement for Releasing Ratchet when Load is Lifted

fact that breakage may take place, the deflection causes heating and binding and a consequent heavy loss of power. The forces due to the combined bending and torsion should always be considered in the ordinary way, and the diameters so proportioned that the stress does not exceed 6 tons per square inch for large shafts and axles, or 5 tons for small shafts.

No definite rule can be given for the limiting stress, it being rather a question of practical consideration and discretion. For example, if a heavy shaft is carrying its load near the bearing, it is safe to subject it to a stress that would not be permissible for a light shaft carrying a gear at some distance from the bearing, as for instance a shaft carrying a number of gears.

Overhung wheels should be avoided where possible, but where such have to be adopted the stress in the shaft ought to be kept low, more especially if subjected to constant reversal, as for instance, the pinion on the end of a motor shaft. Double keys placed at right angles are preferable for fastening gears on high speed reversing shafts.

Ordinary cast-iron plummer blocks are generally used for steel-framed crabs, and should as far as possible be standardized, in order to permit manufacturing to stock. Bearings should be fitted with split brasses and adjustable caps where practicable, in order to allow ready inspection and repair. One of the principal objections to the plate-sided crab lies in the fact that several of the shafts have to be carried in solid bearings. Large and substantial grease or oil lubricators should be fitted to all bearings.

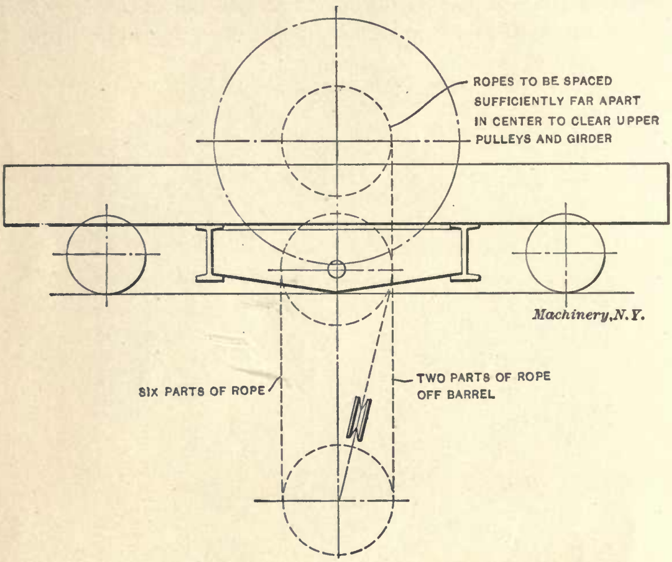


Fig. 16. Modification of Crane Crab when the Number of Ropes exceeds Four

The calculations and general details given above apply to all types of crabs, irrespective of the frame formation, or general design. The principal variations are in the type of brakes, due to the fact that certain makers have patents or particular designs of their own, the bulk of which are, nevertheless, only modifications of the two principal types mentioned.

The type of crab shown in Fig. 4 which, it may be remarked, has only been shown for reference purposes, is subject to considerable modification when the number of ropes exceeds four, when it generally takes the form shown in the outline diagram Fig. 16.

Many makers use four ropes for cranes up to 50 tons, six up to 75 tons and eight up to 100 tons, or in other words, limit the load off the barrel to 25 tons, and these quantities may be taken as a maximum.

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The Industrial Press, Publishers of MACHINERY,

49-55 Lafayette Street, New York City, U. S. A.