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THE

DESIGNING OF DRAW-SPANS.

BY

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FIRST EDITION.

FIRST THOUSAND.

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CONTENTS.

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DESIGNING OF DRAW-SPANS.

PART FIRST.

Plate-girder Draw-spans.

THE following pages aim to give a clear and simple explanation of the methods used in the determination of the stresses, sections required, and of the deflections produced by the various conditions of loading assumed. The machinery necessary for operating the draw is also considered, and the designing of wedging machinery for raising the ends, latching devices for preserving perfect alignment when the draw is closed, methods of raising the rails for clearance when the draw is opened, and the designing of gears, shafting, and bearings are considered in detail. Each point as taken up is illustrated by examples, as fully as necessary to make the applications clear. The aim has been to use the simplest methods, rules, and tables that will give the desired results. Where formulæ derived from the higher mathematics have been used, full and complete explanations of how they are used and applied are given.* It is believed the work may be

* The reader is referred to the works of Professor Releaux and Unwin, from which notes have been taken. The author is also indebted for valuable information to Professor Malverd A. Howe of Rose Polytechnic Institute; the Edge Moor Bridge Works, and to the Pencoyd Bridge Works.

readily followed and understood by those not having a full knowledge of the higher mathematics, and that it will prove of value to any one wishing a practical knowledge ef draw-spans and their machinery.

Plate-girder Draws.

For spans up to about one hundred and fifty feet the deck plate girder makes the most satisfactory bridge, and is the type in most general use. The conditions under which the draw-span works are much more severe than with fixed spans, and the bridge should be correspondingly heavy and rigid. Through plate-girder or lattice spans are unsatisfactory for draw-spans, owing to the small depth usually available below the floor for the introduction of diagonal bracing necessary to resist the twisting force produced in turning the draw, and especially in suddenly stopping or starting. This force is well illustrated by taking a piece of artist's rubber in the fingers and twisting. The rubber may be turned through a considerable angle and still a cross-section at any point will be a perfect rectangle as at first. This shows that any bracing introduced to resist this twisting action must run diagonally as in Fig. 1 and 1^A^. Brace-frames at right angles to the girders do little good to resist such a force, and the same is true of bracing in the planes of the chords.

An eighty-five-foot deck plate girder (Fig. 1) will be used as an example to illustrate the methods pursued with girder draws in general. There are four conditions to be considered. 1st. The span swinging or in position to open, the end wedges being drawn and all the dead loads being carried by the centre, no live load acting. 2d. The draw closed and each arm considered as an independent span for live load; the dead load not being considered for the present. 3d. The bridge

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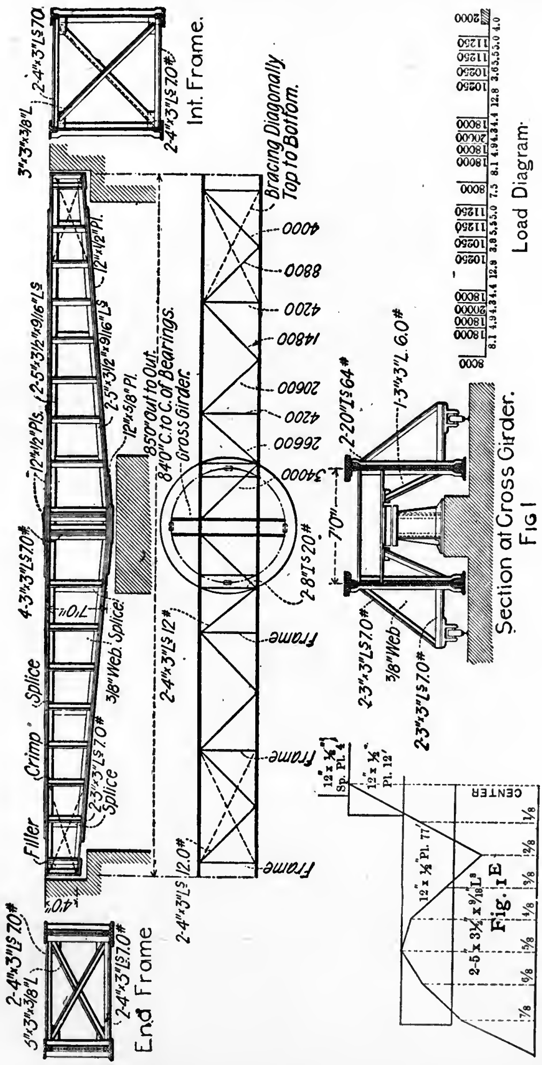
considered as two continuous spans for the live load; and 4th, considered as two continuous spans for the dead load only



Cases 1 and 2 might occur at the same time; also 1 and 3 or 3 and 4. The one of these combinations giving the greatest strains is to be used in determining the sections required.

If the end wedges are just driven to a bearing but not hard enough to *raise* the ends, the dead load would still be carried by the centre, and the span is still *swinging* so far as the *dead* load is concerned. If both arms were now loaded *equally*, the bridge is then a continuous girder of two spans so far as the *live* load is concerned. This is not true, however, if a live load comes on one arm only, unless the other arm be held down so that it does not raise up off the end support as the dead load moves over the first arm. Instead of holding the unloaded arm down, it may be raised so high by the end wedges that the deflection produced in the loaded arm will not be sufficient to raise the unloaded end off the support. Unless one or the other of these plans is followed there will be what is called 'hammer' in the draw. That is, as the load comes on one end and moves over the bridge, first one end and then the other will rise off the supports and drop back again to a bearing. This movement is very noticeable in some draws, and especially so where the rails are cut just at the clearance line and a small space left between the ends. To make sure the rails will clear as the draw turns, this space may needs be three-eighths or one-half inch. This method, or lack of method, of providing for the continuity of the rails is now almost entirely superseded by devices which do not require this clearance. Some of the methods used will be described later. The amount it is necessary to raise the ends by means of the wedges or some similar device will be explained under the deflection of draws.

To determine the strains produced by the dead load swinging, we will assume the weight of the floor (including ties, guards, rails, bolts, etc.) to be 400 lbs. per linear foot, and the weight of the span itself to be 650 lbs. per linear foot. $400 + 650 = 1050$ lbs. $= 525$ lbs. for each girder. Only one arm need be considered if the two arms are equal. If the two arms are not equal, the shorter one is counterweighted until they balance, but the strains would have to be considered separately. The moments may be determined by assuming the dead load as concentrated at several points; thus for the moment over the pier we may assume the load on one arm as concentrated at its centre of gravity, which is at the centre of the arm (see Fig. 2).

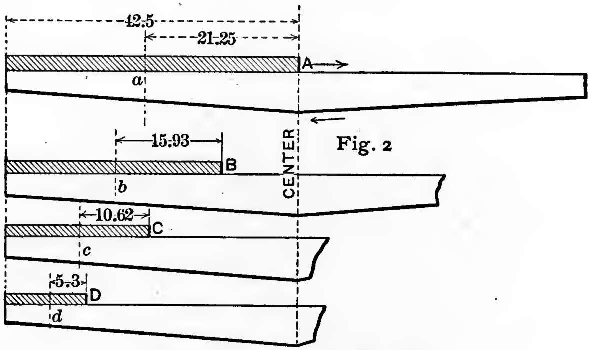


Fig. 2

Arms for Dead-load Moments.

Taking moments at *A*, we have the dead load of one arm, $525 \times 42.5 = 22,310$. Assuming this as concentrated at *a*, the moment is $22,310 \times 21.25 = 474,100$ ft.-lbs. This moment is balanced by forces represented by the arrows and acting in the flanges of the girder. One force is tension and the other compression. The depth of the girder at the centre is 7 feet and the moment $474,100 \div 7 = 67,750$, which is the tension in the upper flange and the compression in the lower.

The depth assumed (7′) should be the depth between the centres of gravity of the flanges. For the moment at *B* we have the load 525×31.86 (the distance from the end to *B*) $= 16,730$. This multiplied by the distance of the centre of this load from *B*, 15.93 feet, $= 266,500$. Dividing by the depth at this point, 6.25 feet, we have $266,500 \div 6.25 = 42,800$. At *C* the moment $= 525 \times 21.25 \times 10.62 = 118,470$. At *D* the moment $= 525 \times 10.62 \times 5.31 = 29,600$. It is not necessary to find the chord stresses at each point now. The moments may be combined with others for live load, and the areas required for both found at one operation. The moment at any point for dead load may also be found by means of a parabola drawn as follows (Fig. 3). Lay off the horizontal

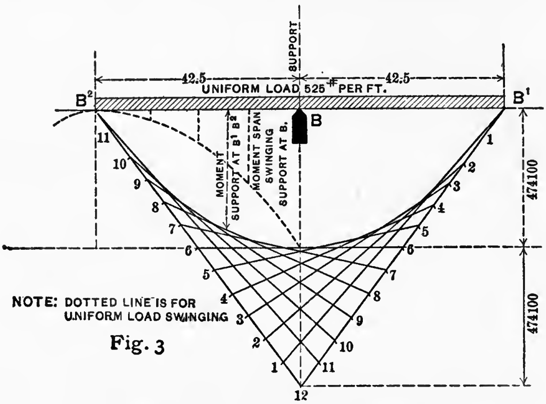


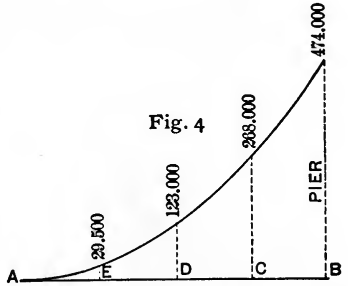
Fig. 3

METHOD OF DRAWING PARABOLA

line *BB′* equal to twice the length of one arm of the draw. From the centre of this line draw a vertical line equal to twice the moment at the pier* (in this case 474,000). Any convenient scale may be used, and the same scale need not necessarily be used for the horizontal and the vertical lines. Draw the inclined lines *B* 12 and *B′* 12, and divide each of them into any number of equal parts (12 in the figure). Connect the points 1–1, 2–2, 3–3, etc., and the lines so drawn will be tangents to the required curve, which is now readily drawn. Only one half the curve is used, as shown by the figure. The curve being drawn, the moment at any point is

* To find the centre of gravity of any number of loads from any point (as one of the end loads), multiply each load by its distance from the point, add the results, and divide by the sum of all the loads. The result will be the distance of the centre of gravity from the point assumed. Note that if there is a load at the point from which we start, this load must be included in getting the sum of all the loads.

found simply by scaling the ordinate between the line $B'B$ and the dotted curve. Having thus shown two methods for determining the dead-load moments with the draw swinging,



Curve of Moments, Dead Load Swinging. 525 lbs. per lin. ft.

we will now consider the case of the draw closed and each arm acting as a single span for *live* load.*

For the live-load moments, each arm acting as a single span, we should so arrange the loads as to get as many loads as possible on the span, and the heavier ones as near the centre as may be. Placing the loads as in Fig. 5, we find the centre of gravity to be 18.7 feet from wheel No. 1, and the wheels are shifted if need be until the centre of the span is half-way between the centre of gravity and load No. 4. We now lay off the load line AB, Fig. 5^A, assume a distance $HO = 100{,}000$ on a horizontal line drawn from any point in AB, and draw the lines AO, BO, etc., connecting the points found by laying off the loads on AB with the point O. This figure (5^A) is called the force polygon. Next, starting from A' (any point in a vertical line through A) draw the line $A'a'$ parallel to AO in the force polygon, and from a' draw the line $a'b'$ parallel to 5–O, from b' the line $b'c'$ parallel to 4–O, and so on until the last line $f'B'$ is drawn parallel to BO.

* In drawing the parabola it will be noticed that the moment over the pier must first be figured. This moment for the load uniformly distributed is $\frac{1}{2}wL$, L being the length of the arm, and w the dead load of one arm. (See first method of finding the dead-load moments.)

The line $A'a'b'c'-f'B'$ meeting the vertical lines through A and B at A' and B' is called the equilibrium polygon. If the line OR be drawn in the force polygon parallel to $A'B'$ of the equilibrium polygon, it will divide the load line AB into

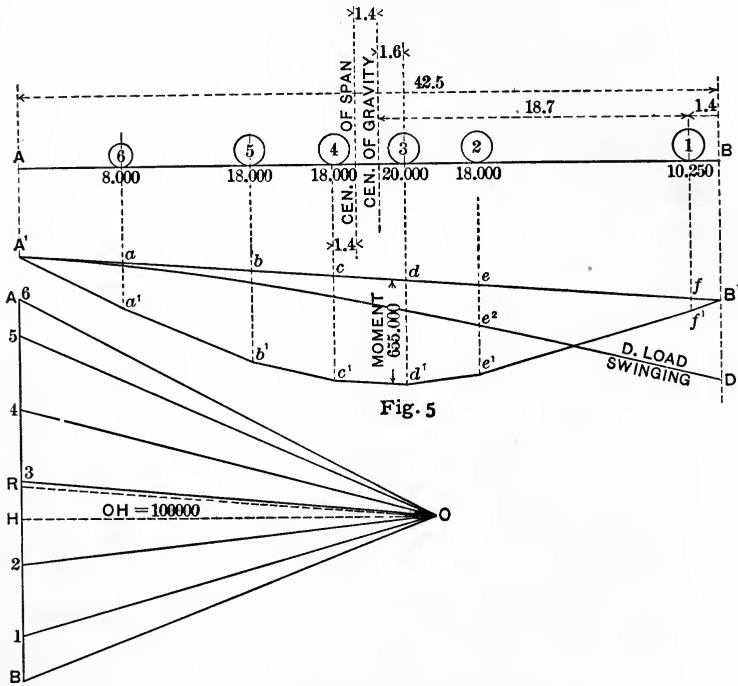


Fig. 5

Diagram for One Arm as Single Span.
Moment at any point as $CC' = CC' \times HO = CC' \times 100$.

the two parts AR and RB which represent the reactions at A and B. Having the equilibrium polygon drawn, the moment at any point is found by multiplying the ordinate between the closing line $A'B'$ and the line $A'a'b'c'$, etc., by the distance OH in the force polygon. HO being 100,000, the moment at b, for example, will be bb' multiplied by 100,000. The distance HO is made 100,000 for convenience. It should be made of such length as will give a good depth to the equilibrium curve, so that the ordinates may be accurately scaled. The distance *HO* must be measured to the same scale as the load line *AB* was laid off, and the ordinates in the equilibrium polygon must be measured to the scale used in laying off the half-length of the span (see Fig. 5). It is not necessary that the two figures be drawn to the same scale. The moments at as many points as necessary can now be determined. These moments are given in column 4 of the table of strains. In Fig. 5 the curved line *A'D* and the line *A'B'* give the dead-load moments with the span swinging, *A'D* being a parabola and the ordinate *B'D* being the moment at the centre support divided by the distance *HO* (= 100,000). The signs of the moments are determined as follows: The loads acting to the left of the centre support tend to revolve the span downward in a direction opposite to the movement of the hands of a clock. These moments are called minus (—). Considering the same arm as a single span, the reaction at the left support tends to revolve the span upward or in the direction of clock motion. These moments are called plus (+). It is immaterial which are called plus, provided all moments tending to produce rotation in the same direction are given the same sign. The total moment then at any point, as e', under the two conditions, dead load swinging and live load discontinuous, on one arm, would be the ordinate $ee' - ee^2 = e^2e'$ multiplied by the pole distance *HO*. It might be found that slightly greater moments would be obtained by placing the loads so that the centre of the span would be between the centre of gravity and load number 3, instead of between the centre of gravity and load number 4 (see Fig. 6). Both positions should be tried. Having shown how to determine the moments for the span swinging, and for the condition of one arm acting as a single span supported at the ends, with live load only acting, we will now consider the span as a continuous girder under the action of both dead and live load. It

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will be noted that in the case of dead load swinging only one arm was considered. This is sometimes confusing and the question is asked, 'Why can one arm be neglected? They must surely both produce strains over the centre.' It is the old problem of two men pulling at the ends of a rope; each man pulls one hundred pounds, but the strain on the rope is not two hundred pounds. One man cannot pull one hundred pounds unless there is a resistance of this amount opposing his pull. It makes no difference whether the resistance is given by a man or by a post at the other end of the line. In the same way an arm of the draw when open is balanced by the other arm. And the moment at the centre is the moment produced by one arm. When the span rests on three or more supports or the loads are not balanced we can no longer consider one arm only.

If a load is placed at any point on the span, a greater proportion of this load will be carried to the centre support than would be the case if the arm on which the load is placed were considered as a single span resting on two supports. Just how much more of the load is carried to the centre is given by the diagram Fig. 9. The figures at the bottom under the line 'values of k' are the distances from the left-hand support to the loaded point, in terms of the length, and the figures in the line marked 'values of D_1' give the per cent of the load going to the left-hand support. Suppose there is a load at three tenths of the length of the arm from the left support. From the figure 0.3 in line k_1 we move up until this line intersects the curve marked 'S_1 loads in first arm'; from the point where the line through 0.3 intersects this curve we go over to the left until we reach the line D_1, which is at 0.63. 63 per cent of the load then goes to the left support. If we wish for the bending moment at this point, we move up the line through 0.3 in k_1 until we meet the curve marked 'M_2 loads in first or second arm.' We intersect this curve on the horizontal line 0.685,* and so for any other point in the span. We will now place the engine-loads on the span in two or three positions and see which position will

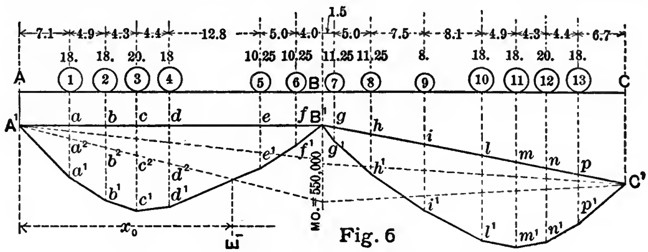


Fig. 6

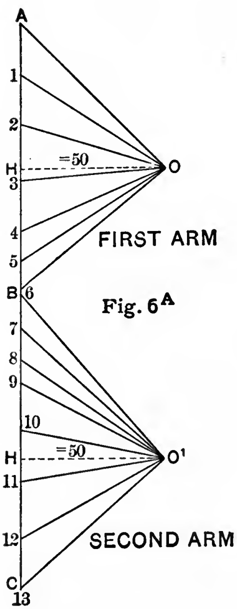


Fig. 6A

FIRST ARM.

	k.	*c.*	*CPL.*
7.1 ÷ 42.5	= .167	.0405	30,980
12.	= .282	.0645	49,340
16.3	= .384	.0819	69,610
20.7	= .487	.0927	70,900
33.5	= .788	.0745	32,450
38.5	= .906	.0400	17,420
			270,700
Moment = *CPL*			279,385
			550,085

SECOND ARM.

	k.	*c.*	*CPL.*
6.7 ÷ 42.5	= .0156	.0380	29,608
11.1	= .261	.0605	51,450
15.4	= .362	.0785	60,030
20.3	= .477	.0920	70,380
28.4	= .668	.0925	31,350
35.9	= .845	.0600	28,680
40.9	= .962	.0165	7,887
			279,385

Diagram for Two Spans Continuous. Scales, 20 and 50.

* 0.685 is the value of *C* in formula $M = CPL$ = moment at any point.

give us the greatest moment over the pier. Arranging the loads as in Fig. 6, we first find the values of k; thus for loads

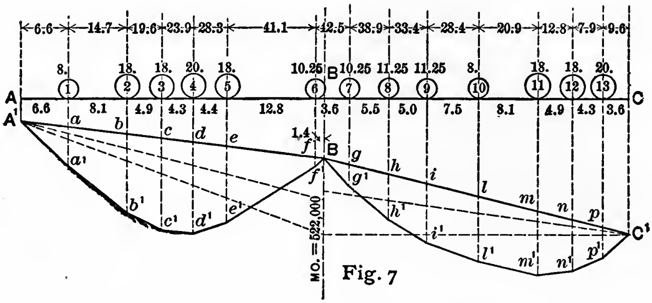


Fig. 7

FIRST ARM.

k.	c.	CPL.
6.6 ÷ 42.5 = .155	.0380	12,929
14.7 = .346	.0756	57,815
19.6 = .461	.0910	69,605
23.9 = .562	.0963	81,850
28.3 = .666	.0927	70,905
41.1 = .967	.0150	6,525
		299,629
Moment = *CPL*		233,331
		532,960

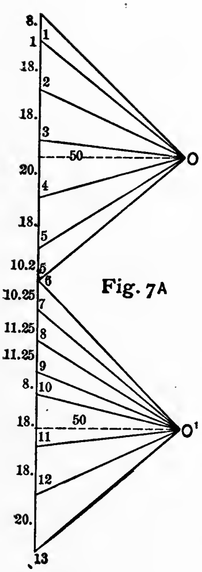


Fig. 7A

SECOND ARM.

k.	c.	CPL.
3.6 ÷ 42.5 = .084	.0215	18,274
7.9 = .186	.0450	34,422
12.8 = .301	.0685	52,395
20.9 = .492	.0933	31,720
28.4 = .668	.0927	44,760
33.4 = .786	.0750	35,862
38.9 = .915	.0365	15,898
		233.331

Diagram for Two Spans Continuous. Scales, 20 and 50.

1, 2, 3, 4, 5, and 6 we divide the distances from the left by the half-span 42.5′, and for loads 7, 8, 9, 10, 11, 12, and 13 we divide the distances of the loads from the right-hand abut-

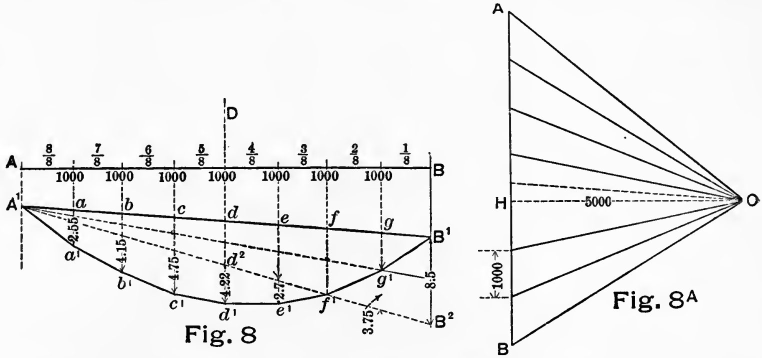
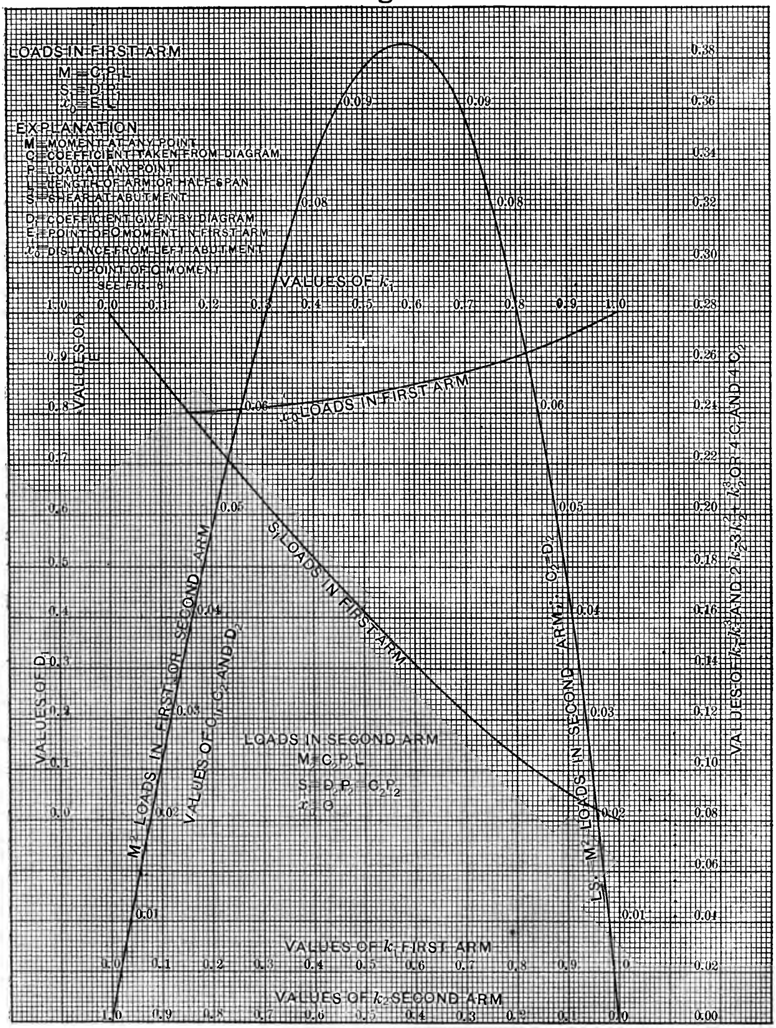


Diagram for Uniform Load Continuous. 1000 lbs. at each eighth point assumed load in diagram.

ment by the half-span 42.5. The values are given in the table: .167, .282, etc., for first arm and .0156, .261, etc., for the second arm. From diagram Fig. 9 we now find the values of C corresponding. The vertical through $k = 167$ meets the curve of moments on the horizontal line .0405, and for $k = .788$ on the line .0745. The values of k for the second arm are given from the right abutment, so we find C exactly as in the first arm. If the distances had been given from the centre pier, we could have found C in the same manner, only using the line marked k^2 in the diagram instead of line k'; for example, if a load is .8 the length of the half-span from the right abutment, it is .2 the half-arm from the centre pier. $0.1k'$ is over $k^2 = 0.9$. It is perhaps a little simpler to use the line k' all the time, and give the distances of the loads from the abutments in each case. All values of C have the same sign. Multiplying each value of C by the *load at that point*, and by the length of the half-span, gives us the moment over the centre pier for that load. $CPL =$ moment over pier for load P at any point. The values of these moments for each of the wheel-loads with the engine placed in the two positions given in Figs. 6 and 7 are given

Fig. 9



in the tables under the figures. Two or three trials will show how the engine should be placed to give the greatest moments. By referring to the diagram Fig. 9 it will be seen that C is greatest for loads near the centre of each arm, and a little nearer the centre pier than the abutments. The heavier wheels should then be placed as near these positions as possible to give the maximum moments. Adding together the moments produced by all the loads, we have the total moment. In the two cases given these total moments are 550,085 and 522,960. It is possible that the uniform train load might give a greater moment at the pier than the engines, and this moment should be found.

Before considering the uniform load we will take one more example of moment from concentrated load to make the method just described perfectly plain.

Suppose we take wheel No. 11 in Fig. 7. The distance of this wheel from the right abutment is 12.8. $k = 12.8 \div 42.5 = .301$. C for $k = .301$ is .0685, and CPL, the moment, = 52,395. $P = 18$ and $L = 425$.

Considering now the case of uniform load, span continuous, the Reading loading diagram gives 4000 lbs. per linear foot, or 2000 lbs. on one girder. $2000 \times 42.5 = 85{,}000 =$ the load on one arm. The formula for the moment at centre support with uniform load is $\frac{1}{8}wl^2$ $w =$ the load per foot, and $l =$ the length of one arm of the span In this case $w = 2000$, $l = 42.5$, $wl = 85{,}000$, $\frac{1}{8}wl^2 = 451.562$. This is considerably less than the moment from the wheel-loads, which was 550,085 for one position of the loads. It will be noticed that the moment over the pier, $\frac{1}{8}wl^2$, is just the same as the moment at the centre of a single span of length equal to one arm of the draw and covered with the same uniform load; and is also just one fourth as much as it would be over the centre support were the draw swinging and covered with the same load. Note that in moment $\frac{1}{8}wl^2$, wl is load on one arm. A convenient method of finding these moments for uniform load is to assume a load one pound or one thousand pounds per foot, find the moments for this loading, and then multiply the results by the ratio of the actual loads to the one assumed. To make us a little more familiar with the force and equilibrium polygons, we will divide each arm into eight parts and assume a load of 1000 lbs. at each of these points and one half load at the ends. The loads at the ends, coming directly over the supports, may be neglected in the computation. We lay off then on the vertical line AB, Fig. 8^A^, seven spaces representing 1000 lbs. each. Any scale may be used, say one-half inch equals 1000 lbs. Next assume the point O distant from AB 5000 to the same scale. Note that the point O may be anywhere in a vertical line which is distant 5000 from the vertical line AB, and also remember that we assumed the distance 5000; any convenient distance may be used. We next connect the point O with each of the points laid off on AB. Now going to Fig. 8, at any point on a vertical through A we draw the line $A'a'$ parallel to AO in Fig. 8^A^, and from a' the line $a'b'$ parallel to the next line in the force polygon, and so on until finally $g'B$ is drawn parallel to BO in the force polygon. Now connect A' and B' with a straight line. From B' in Fig. 8 scale off the distance B'–B^2 equal to the moment at the centre support divided by the distance $HO = 5000$ in Fig. 8^A^. The distance $B'B^2$ must be laid off to the same scale as Fig. 8 is drawn to. The moment at the centre is of course found for the same loading (1000 lbs. at each eighth point $= \frac{1}{8}wl^2$). By the diagram Fig. 9 the values of C for $k = \frac{1}{8}, \frac{2}{8}, \frac{3}{8}, \frac{4}{8}, \frac{5}{8}, \frac{6}{8}$, and $\frac{7}{8}$ are:

$K = \frac{1}{8} = .125$	$C = .0308$;	$P = 1000$,	$L = 42.5$;	CPL	$= 1309.00$
$K = \frac{2}{8} = .250$	$C = .0586$;	"	"	"	$= 2490\ 50$
$K = \frac{3}{8} = .375$	$C = .0806$;	"	"	"	$= 3425\ 50$
$K = \frac{4}{8} = .500$	$C = .0938$;	"	"	"	$= 3986.50$
$K = \frac{5}{8} = .625$	$C = .0952$;	"	"	"	$= 4046.00$
$K = \frac{6}{8} = .750$	$C = .0820$;	"	"	"	$= 3485.00$
$K = \frac{7}{8} = .875$	$C = .0513$;	"	"	"	$= 2180.25$
	.4923				20922 25

The moment $\frac{1}{8}wl$ for the same load uniformly distributed (8000 lbs. on each arm) is 42,500. The difference by the two

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methods is 655.75 or $1\frac{1}{2}$ per cent, which shows that the method is practically correct, and it is merely a question of reading the diagram correctly to obtain accurate results. Making a table of the moments (see p. 18), we have first the column of moments for dead load swinging, the moments being found by methods shown in Fig. 2 or 4. These moments are 474,000, 350,000, etc. Next we make the column of moments for dead load continuous, as shown by Fig. 8, remembering that the moment at any point is equal to the moment for the same load, considering the arm as a single span supported at the ends, less the negative moment at this point, and that this negative moment is represented by the ordinate between the lines $A'B'$ and $A'B^2$ multiplied by the pole distance HO; the ordinate B^1B^2 being the moment over pier divided by the pole distance HO. Thus the moment at D equals ordinate dd' minus ordinate dd^2 (Fig. 8) multiplied by HO ($HO = 5000$).

Having the moments tabulated, we now see which combinations will give the largest totals. The dead load swinging and live load continuous, case A, give the largest moment over the centre support, 1,024,000. The same combination also gives the greatest moment at the $\frac{1}{8}$ point. At the quarter point the dead load swinging and case A live continuous give a minus moment of 318,000, and live load discontinuous with dead load swinging give a plus moment of 187,000, and so at each of the points $\frac{3}{8}$, $\frac{4}{8}$, etc., we obtain the results given in column 8. Dividing these results by the depth of girder (centre to centre of gravity of flanges), we obtain the results given in column 10. Dividing these results by the unit stresses as allowed by the specifications (in this case 8000 lbs.), we have the areas required (column 12).* In Fig. 1^E the areas required at the several points are laid off to scale, and the lengths of the cover-plates required readily determined.

* Where the flange-areas are determined for tension, the areas after deducting rivet-holes must be used.

TABLE OF STRAINS.

Dist. from Pier.	Moment.				Live Load Continuous.		Total.	Depth.	Flange-strain.	Unit Stess.	Area.
	Dead Load Swinging.	Dead Load Continuous.	Live Load Discontinuous.	Live Load Uniform, 2000 lbs.	Fig. 6. Case A.	Fig. 7. Case B.					
1	2	3	4	5	6	7	8	9	10	11	12
Pier	− 474,000	− 118,500		− 451,600	− 550,000	− 522,000	−1,024,000	7′.	146,280	÷ 9,000	= 18.29
1/8	− 350,000	− 50,900	+ 255,000	− 194,000	− 270,000	− 250,000	− 620,000	6′.63	93,500	"	= 11.69
							− 318,000		47,000	"	= 5.9
2/8	− 268,000	− 00	+ 455,000	− 00	− 50,000	+ 45,000	+ 187,000	6′.25	29,920	"	= 3.75
							− 185,000				
3/8	− 185,000	+ 38,060	+ 600,000	+ 145,000	+ 156,000	+ 273,500	+ 415,000	5′.88	68,880	"	= 8.61
							− 123,000				
4/8	− 123,000	+ 61,510	+ 655,000	+ 226,000	+ 317,000	+ 395,000	+ 532,000	5′.50	96,720	"	= 12.1
							− 65,000				
5/8	− 65,000	+ 66,700	+ 600,000	+ 254,000	+ 417,000	+ 400,000	+ 535,000	5′.13	104,300	"	= 13.04
							− 29,500				
6/8	− 29,500	+ 59,500	+ 455,000	+ 227,000	+ 382,000	+ 300,000	+ 441,500	4′.75	92,900	"	= 11.61
							− 10,000				
7/8	− 10,000	+ 37,300	+ 255,000	+ 142,000	+ 225,000	+ 200,000	+ 262,300	4′.38	60,000	"	= 7.50

Combinations: Col. 2 with 4–6 or 7; col. 3 with 6 or 7.

The plates should extend about two feet beyond the points so determined.

The web is not considered as taking any flange-stress, and the area in top flange is made up by two $5'' \times 3\frac{1}{2}'' \times \frac{9}{16}''$ angles and two $12 \times \frac{1}{2}$ plates. One of the plates will be too long to get in one length, and a splice-plate is added to make up the section at the splice. In the bottom flange two $\frac{5}{8}''$ plates are used.

WEB-STRAINS.

We will next consider the shearing stresses in the web. The greatest shear at the abutments will be obtained by considering one arm as a single span for live load and dead load swinging, no dead reaction at abutment, as the condition of dead load *continuous* and live load discontinuous cannot occur. See combination of strains made. From a table of 'shears and bending moments' for this engine we have the end shear for a span 42.5 = 72,650 lbs. That is, 72,650 lbs. is the *upward* force exerted by the support at the abutment. Say the specifications allow 6000 lbs. per square inch shearing on webs; then $72,650 \div 6000 = 12.1$ sq. in. required; $48 \times \frac{3}{8}$-inch web plate gives 18 sq. in. At the quarter point the upward shear is 46,500 lbs. From this is to be taken the dead load between the abutment and this point. This load equals $525 \times 10.62 = 5600$ lbs. $42,500 - 5600 = 36,900$ lbs. Note that in finding the greatest live-load shears the heavy wheel at the front of the engine is placed at the point where the shear is required, and that there is no live load on the span between the abutment and the point whose shear is being determined. At the centre of the arm the live shear is 22,700 upward, and the dead-load shear downward is 11,200. $22,700 - 11,200 = 10,500$. The greatest shear at the pier will be with dead load swinging (all dead load carried to the pier) and with live either continuous or discontinuous. For discontinuous live load we have the same maximum shear at the pier as at the abutment, the engine simply headed the

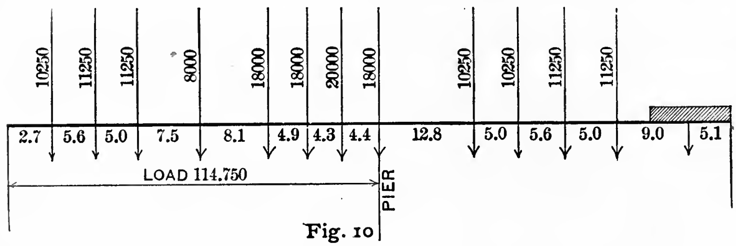

Fig. 10

	Span.	k_1	D_1	Load.	S_1
2.7	425	0.06	0.93	10,250	9,740
8.3	"	0.19	0.76	11,250	7,780
13.3	"	0.31	0.62	11,250	6,975
20.8	"	0.49	0.41	8,000	3,280
28.9	"	0.68	0.23	18,000	4,140
33.8	"	0.79	0.14	18,000	2,520
38.1	"	0.89	0.06	20,000	0,120
42.5	"	1.00	0.00	18,000	0,000
					34,355
			D_2		
12.8	425	0.30	0.089	10,250	912
17.8	"	0.42	0.096	10,250	984
23.4	"	0.54	0.091	11,250	1,024
28.4	"	0.68	0.071	11,250	782
37.4	"	0.88	0.024	20,000	481
42.5	"	1.00	0.0		000
					4,183

$114,750 - (34,355 - 4183) = 84,580 \pm.$

Shear at Centre, Girder Continuous.

other way. We have then the upward shear live $= 72,650 +$ the dead weight of one arm $= 22,300$. $72,650 + 22,300 = 94,950$.

Considering now the case of live load continuous: it is clear that a load in any position (as the centre) on one arm tends, by causing this arm to deflect, to raise the other arm off its abutment or end support. This support then has less to do or the shear is reduced at this point by the load on the other arm; it follows therefore that, as all the load on the span must be carried by the abutments and the pier, if some load is taken from the abutment it must be added to the load on the pier. A greater proportion of the load is carried by the

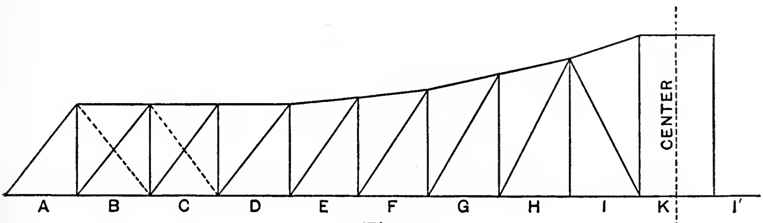

Fig. 11

centre pier considering the two arms as continuous than by considering them as independent spans. And in determining the shear at any point the loads on both arms must be considered. By means of diagram Fig. 9 the reactions caused by loads at any point in either arm are readily determined. Arranging the loads as in Fig. 10, and finding the values of k_1, k_2, and D_1, D_2, we get for the shear just to the left of the

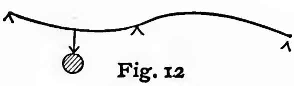

Fig. 12

pier 84,580; this added to the dead-load shear gives a total of 84,580 + 22,300 = 106,880. The area of 84 × $\frac{3}{8}$ web = 31.5 sq. in., against 106,880 ÷ 6000 = 17.8 required. Stiffeners should be at intervals of about the depth of the web apart, with 6 ft. as a maximum.

Lateral Bracing.

The laterals should be figured for a wind-load of, say, 600 lbs. per lineal foot, the point being to get sections heavy enough to render the span stiff laterally. Cross-frames should be used at intervals of ten to fifteen feet. Note that lateral bracing should be figured to carry strains to the centre, and that this force, equalling at least 300 lbs. per foot = 25,500 lbs. for both arms, should be considered in designing centre pivot and anchorage.

Cross-girders.

When the draw is closed and ready for the passage of trains the girders are supported at the centre by wedges, so the cross-girders carry only the dead weight of the span. This amounts to 44,600 at each side; as there are two cross-girders, the moment on each is 22,300 × 42 in. = 936,600 in.-lbs. Using 20-in. 64-lb. beams, with a moment of resistance of 114, gives a fibre-stress of 8200, allowing an ample margin.

Centre-post.

The load on the centre-post is about 90,000 lbs. The base of the post should be large enough to distribute this well over the masonry and to give the post stability. There should be anchor-bolts built into the masonry, and their area should be sufficient to resist the shear from wind-forces, assuming for this purpose a wind-pressure of 300 lbs. per lineal foot of bridge, and neglecting the friction of the base-plate on the masonry. This gives a force of 300 × 85 = 25,500 lbs. Four $1\frac{1}{4}$-in. bolts at 7300 lbs. each would be ample. A wrought-steel post is preferable to one of cast iron, as it is much less liable to break if the bearing on masonry becomes unequally distributed. The post should be made high enough to throw the point of suspension into the upper half of the web; the girders will then hang better and turn more easily, as there will be less weight thrown on the trailing-wheels.

Deflection.

Deflection Formulæ.

Note.—These formulæ are applicable to spans of any length if the proportions are approximately as given below

$$I = \frac{\left(\frac{1}{2}h\right)^2\left(\frac{1}{2}l + x\right)}{12}.$$

$$D \text{ for uniform load} = \frac{4.704 WL^3}{Eh^3} \quad \ldots \ldots \quad (1)$$

$$D \text{ for load at end} = \frac{13.18 PL^3}{Eh^3} \quad \ldots \ldots \quad (2)$$

$$I = \frac{1}{6.8}h^3. \qquad h = \frac{4}{7}h_1 + \frac{3}{7}h_1 . \frac{x}{l}$$

$$D \text{ for uniform load} = \frac{1.166 WL^3}{Eh_1^3} \quad \ldots \ldots \quad (3)$$

$$D \text{ for load at end} = \frac{3.377 PL^3}{Eh_1^3} \quad \ldots \ldots \quad (4)$$

$$I = \frac{1}{6}h^3. \qquad h = \frac{1}{3}h_1 + \frac{2}{3}h_1 . \frac{x}{l} = \frac{2}{3}\frac{h_1}{l}\left(\frac{l}{2} + x\right).$$

$$D \text{ for uniform load} = \frac{1.315 WL^3}{Eh_1^3} \quad \ldots \ldots \quad (5)$$

$$D \text{ for load at end} = \frac{4.248 PL^3}{Eh_1^3} \quad \ldots \ldots \quad (6)$$

D = deflection;

h_1 = height at centre; h = height for any distance x;

L = length in inches;

x = distance from left end in inches;

P = load at end; W = totol load uniformly distributed.

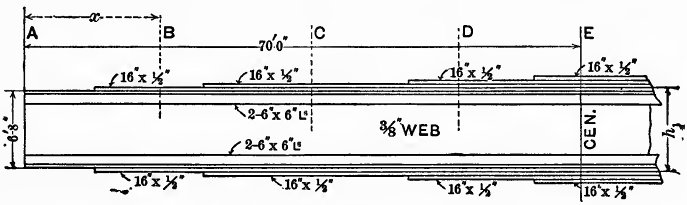

Fig. 13.

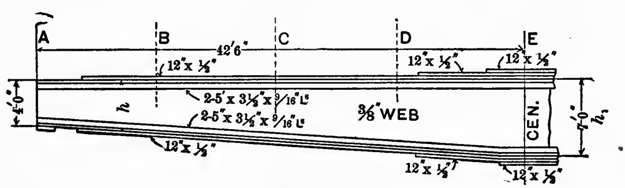

Fig. 14.

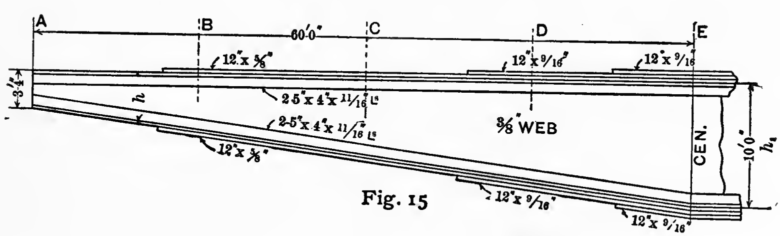

Fig. 15

The amount the girders will deflect under the various loads depends upon the length, the depth, and the arrangement of the material in the girders. If the flanges are parallel and their area of cross-section remain the same or nearly so throughout their length, the formula for deflection for constant section may be used; thus for uniform load

$$D = \frac{Wl^3}{8EI}. \qquad (7)$$

D = deflection, W = the load on the girder, l = the length in inches, E = 29,000,000, and I = the moment of inertia. For a load at the end

$$D = \frac{Pl^3}{3EI}. \qquad (8)$$

If the flanges are parallel, but the cover-plates are of several lengths and the girder have about the proportions shown in Fig. 13, the deflection for uniform load will be

$$D = \frac{4.704Wl^2}{Eh_1^2}. \qquad (1)$$

W = total load on arm, and h = depth of girder back to back of flange-angles. For load at end

$$D = \frac{13.18Pl^2}{Eh_1^2}. \qquad (2)$$

Number 1 is equal to

$$D = \frac{Wl^3}{195,500,000I}, \qquad (1a)$$

and number 2 may be written

$$D = \frac{Pl^3}{68,900,000I}. \qquad (2a)$$

W = total load on one arm in each case, and I = the moment of inertia at the centre support. For girders having approximately the proportions shown in Fig. 14, which is the span

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taken as the example in considering strains, etc., we have for uniform load

$$D = \frac{1.166\,Wl^3}{Eh_1^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$= \frac{Wl^3}{161,500,000I}, \quad . \quad . \quad . \quad . \quad . \quad (3a)$$

and for load at end

$$D = \frac{3.377Pl^3}{Eh_1^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$= \frac{Pl^3}{55,700,000I} . \quad . \quad . \quad . \quad . \quad . \quad (4a)$$

Where the girders have the proportions as given in Fig. 15, for uniform load

$$D = \frac{1.315\,Wl^3}{Eh_1^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$= \frac{Wl^3}{132,000,000I}, \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

and for a load at end

$$D = \frac{4.248Pl^3}{Eh_1^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$= \frac{Pl^3}{40,800,000I} . \quad . \quad . \quad . \quad . \quad . \quad (6a)$$

W and I as given above. Some one of the formulæ would be applicable to any case likely to occur.

Considering first the case of uniform load: the girder we have been considering is composed of $84'' \times \frac{3}{8}''$ web, four $5'' \times 3\frac{1}{2}'' \times \frac{9}{16}''$ angles, and (neglecting the short splice-plate), two $12'' \times \frac{1}{2}''$ plates in top flange and two $12'' \times \frac{5}{8}''$ in bottom flange. To simplify the calculations, we will for the present consider all cover-plates as $\frac{1}{2}$ in.; if this is not done, we should first find the centre of gravity of the section, and then the moment of inertia about this axis. Usually the flange-plates are the same, and we will obtain nearly correct results by so considering them. The moment of inertia of the web about its centre is equal to $\frac{1}{12}bh^3$. ($b = \frac{3}{8}$, and $h = 84$.) $\frac{1}{12}bh^3 = \frac{1}{12} \times \frac{3}{8} \times 84^3 = 18,522$. The moment of inertia of the cover-plates and angles about the centre of the web is found by multiplying the area of each by the square of the distance between its centre of gravity and the centre of the web. Thus the area of the four $\frac{1}{2}$-in. plates = 24 sq. in., and the square of the distance from the centre of web to their centre is $(42 + \frac{1}{2})^2$. $24 \times (42.5)^2 = 43,350$. By referring to Carnegie's Pocket-book we see that the centre of gravity of the $5 \times 3\frac{1}{2}$ angles is about 1 in. from the back of the angle, and that the area of the four angles is 17.88 sq. in. The half-depth 42 in. — 1 in. gives 41 in. as the distance from the centre of the web to the centre of gravity of the angles. $17.88 \times (41)^2 = 30,056$. To the moments of inertia thus obtained we add the moments of inertia of the cover-plates, and the angles about their own centres of gravity; for the cover-plates $\frac{1}{12}bh^3 = \frac{1}{12} \times 12 \times 1$ in. = 1 for each flange, and for the angles we have from the Pocket-book 4.2 for each angle (see page 103, edition of 1893). $4.2 \times 4 = 16.8 + 2 = 18.8$, amount to add for plates and angles. The total moment is then $18,522 + 43,350 + 30,056 + 18.8 = 91,946.8$. It will be noticed that the moments of inertia of the plates and angles about their own axis is very small, and might be neglected without seriously affecting the result.

Using our formula No. 3*a*, we have

$$D = \frac{Wl^3}{161,500,000I}.$$

$W = 22,300$, as previously found, $L = 42.5$ ft., and $I = 91,946.8$.

$$D = \frac{22,300 \times 132,651,000}{161,500,000 \times 91,946.8} = 0.19 \text{ inch} = \tfrac{3}{16} \text{ inch}.$$

If each arm is given a camber, this must be considered in determining the end deflection. Suppose the top chord be lengthened by adding $\frac{1}{4}$ in. at a web-splice near the centre of the arm. If the girder be 5 ft. 6 in. deep at this point, and the distance to the end be 21 ft., the end will drop $\frac{1}{4} \div 5.6 \times 21 = .94$, say $\frac{61}{64}$ in. Adding $0.19 + .94 = 1.13$ in. $= 1\frac{1}{8}$ in., end deflection.

MACHINERY.

For Turning.—The forces to be overcome in turning the draw are, first, the inertia of the span itself. That is, there is a certain mass which has to be revolved through a quarter of a circle or 90° of an arc in a certain time. Second, there is the friction on the centre pivot or rollers. Third, the friction of the trailing-wheels due to the overturning force of the wind, and the friction on the vertical surface of the pivot due to the wind-pressure. Fourth, there is the friction of the trailing-wheels due to any unbalanced load there may be. Fifth, the friction of the shaft-bearings, etc. Item four might be considerably increased by the rails on which the wheels bear being out of level, rough, and with wide openings at the joints. It is sometimes assumed that the draw shall turn against a wind-force acting on one arm only of the span. While this might possibly happen in the case of a long span, it could hardly occur in the short 85-foot span we are considering, and this condition will not therefore be treated at present.

Force required to Overcome Inertia.—For convenience we replace the mass of the bridge by an equivalent mass acting at the rack-circle. This mass is found as follows: Multiply the weight of the span by the square of half the length plus the square of half the width, and divide by 96.6 times the square of the radius of the rack-circle. Putting this in the form of an equation,

$$M = \frac{W(a^2 + b^2)}{96.6R^2},$$

where W = weight of span;
a = half-length of span;
b = half-width of span;
R = radius of rack-circle;
M = equivalent mass at rack-circle.

The weight of our span is 89,200 lbs. = W. a, the half-length, = 42.5 ft.; b, the half-width, = 3.5 ft.; and R, the radius of the rack, = 7.85 ft. We have therefore

$$M = \frac{89,200 \times (42.5^2 + 3.5^2)}{96.6 \times 7.85^2} = 27,224.$$

If we assume that the draw shall open in two minutes, the average velocity will be one fourth the circumference of rack divided by 120 sec. $= \frac{49.32}{4 \times 120} = 0.103$ ft. per second. But the velocity is not uniform; it increases during the first half of the turning, and then reduces to 0 again at the end. The maximum velocity at the end of 60 seconds is then twice the average, or 0.206 ft. per second. The rate of increase is $0.206 \div 60 = .0034$.

The force necessary to give a mass of 27,224 lbs. a constantly increasing velocity of .0034 ft. per second = $27,224 \times 0.0034 = 92.5$ lbs. We will call this Fm.

Force to Overcome Friction on Centre Bearing.—A Sellers centre is used so the friction from load will be rolling friction; a coefficient of .003 may be used, and this multiplied by the load gives $89,200 \times .003 = 267.6$ lbs. This acts at the centre of the length of the roller, or with a leverage of 8 in. or .62 ft. $267.6 \times .62 \div 7.85 = 21.1$ lbs. the force required at rack to overcome it. This force we designate F_p.

Friction on Side of Pivot or End of Rollers for Wind-pressure.—Assuming a wind-load of 300 lbs. per lineal foot, there results a total horizontal force of $300 \times 85 = 25,500$ lbs. This, whether acting against the ends of the rollers or on the side of a pivot, will produce sliding friction. Using a coefficient of 0.1, this gives $25,500 \times 0.1 = 2550$ lbs. acting at the end of roller or at circumference of pivot (acting on vertical surface). Let the radius of end of roller be $9\frac{1}{2}$ in. or .8 ft., then $2550 \times .8 \div 7.85 = 259.8$ lbs. at rack. We will denote this by *Fw*.

Force required to Overcome an Unbalanced Condition of the Draw.—Suppose that from snow or some other cause there is an unbalanced load on one arm, acting at a point 15 ft. from the centre pivot. The force at the wheel-circle required to balance this is $15 \div 7$ (the radius of the wheel-circle) $= 2.143$ times the load. Assume the load to be 2000 lbs.; this multiplied by 2.143 gives 4286 as the pressure on the balance-wheel. The friction caused by this pressure will be rolling friction and equal to $4286 \times .003 = 13$ lbs. Thirteen pounds at the wheel-circle will require $13 \times 7 \div 7.85 = 11.6$ at the rack to overcome it (7 and 7.85 being the radii of the two circles). This force we will call *Fu*.

The centre of the surface exposed to the wind, including ties and guard-rails, is almost exactly in line with the bottom of the cross-girders, so that the moment of the wind-force tending to revolve the girders about the centre casting as a fulcrum is in this case slight and may be neglected. Suppose the centre of wind-pressure had been one foot above the point of support for cross-girders, the overturning moment would then have been $25,500 \times 1 = 25,500$ lbs.; this divided by the horizontal distance from the centre support to the centre of the trailing-wheel, 7 ft., gives the vertical force acting at wheel to resist overturning. $25,500 \div 7 = 3643$ lbs. Using coefficient of friction .003 gives 10.9 lbs. $10.9 \times 7 \div 7.85 =$, say, 9.7, force at rack necessary to overcome it. This will show how to proceed in cases where this overturning force of the wind is too great to be neglected.

Force required to Overcome the Friction of the Shaft.—There will be only one shaft required in the turning arrangement.

Assuming one man is able to turn the draw, and that he exerts a pressure of 75 lbs. horizontally against the top of the shaft; assuming for the present also that he works at the end of a five-foot lever, and that a pinion 8 in. in diameter can be used in rack, we have a horizontal pressure at foot of shaft of $75 \times 60 \div 4 = 1125$ lbs. $1125 + 75 = 1200$ lbs., total pressure on shaft-bearings. The friction caused by this will be sliding friction, for which the coefficient is 0.05 to 0.1. Multiplying $1200 \times 0.1 = 120$ lbs. as the frictional force acting at the circumference of the shaft. This we will call Fs.

We have then forces to be overcome as follows: $Fm = 92.5$, $Fp = 21.1$, $Fw = 259.8$, $Fu = 11.6$, and $Fs = 120$ lbs.

First we will see how much power is consumed in overcoming Fs. The radius of the shaft will be assumed as $1\frac{1}{4}$ in. for the present, then $120 \times 1\frac{1}{4} \div 60 = 2.5$, the power required at end of turning-lever to balance it. This leaves us $75 - 2.5 = 72.5$ lbs. as available against the other forces which all act at rack-circle. These equal $92.5 + 21.1 + 259.8 + 11.6 = 384.9$ lbs. Dividing 384.9 by 72.5 gives 5.3, which is the number of times the power must be multiplied between the turning-lever and the pinion, or by the two. We see at once that our power will be greatly in excess of the amount required. It will multiply as many times as the radius of the pinion is contained in the length of the turning-lever, $60 \div 4 = 15$ (using an 8-in. pinion). We might use a six-inch pinion and four-foot turning-lever. It is well, however, to have a good excess of power, as machinery may get out of adjustment, the track become rough, and with gaps at the joints, the span may become badly unbalanced, etc.

Time for Turning.—The man turning the draw will walk at an average velocity of, say, 3 ft. per second. If he be moving at the end of a five-foot lever, he will move in a circle of 31.6 ft. circumference. It will require $31.6 \div 3 =$ 10.5 seconds for him to make one complete revolution. The pinion of course makes one revolution in the same time. Using a pinion of 25 in. circumference on the pitch-line, and a rack of 49.3 ft. circumference, the pinion must make $\frac{591.6 \text{ in.}}{4 \times 25} = 5.9$ revolutions in moving over one fourth of the circumference of the rack, which would be necessary to open the draw. If one revolution is made in 10.5 seconds, $10.5 \times 5.9 = 62$ seconds as the time required to open or close the draw.

Size of Turning-shaft.—The man moving at the end of the turning-lever produces a twisting moment on the shaft of $75 \times 60 = 4500$ in.-lbs. In addition to this twisting there is the bending produced by the force acting on the pinion.

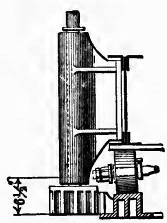
Fig. 16

Assuming an 8-in. pinion, this force equals 1125 lbs.; and assuming that the lower corner of the tooth is acting, and that the distance from this corner up to, say, 1½ in. inside the journal-bearing equals 6½ in., then the bending moment will be $1125 \times 6\frac{1}{2} = 7312$ in.-lbs. By referring to the notes on shafting we find that the strength of a shaft to resist both bending and twisting is given by the formula

$$T' = M + \sqrt{M^2 + T^2}.$$

$M =$ bending moment, and $T =$ twisting moment.

$$T' = 7312 + \sqrt{53465344 + 20250000} = 7312 + 8585 = 15897.$$

Adding 50 per cent to this to allow for contingencies, we have 23,846 in.-lbs., requiring a $2\frac{5}{8}$-in. diam. shaft. Note that the shaft is weakened by the keyways and the shoulders for turning-lever.

Proportions of Trailing-wheels.—The face of the wheel should be about 4 in., to make sure it always has bearing on the rail and to keep the bearing back from the edge. Letting w = width of face, the other proportions would be about as follows: Thickness of rim = $.4W$; thickness of solid web = $.25w$; stiffening-ribs, six in number, thickness = $.2w$; length of hub, not less than $1.5w$; diameter of hub, 1.85 times the size of axle required.

The side bearings should not be less than the diameter of the axle, giving total bearing of $2D$ or more.

In figuring the size of axle required, if a length from the centre of the wheel to the centre of bearing be used, the unit stress in bending might be assumed at 30,000 lbs. per square inch. The reason for this is that the bearings and hub practically fix the axle so that it cannot bend until it leaves the hub or the bearings.

Strength of Teeth in Rack and Pinion.—Referring to the tables and notes on the strength of teeth, we find the formula for the safe load on cast-iron teeth $P = 375t^2$. This formula is for the strength of tooth considering the load as applied at one corner. We found the pressure on the tooth to be 1125 lbs.; then $P = 1125 = 375t^2$. $t^2 = 3$, and $t = 1.73$. (See table of cast-iron teeth.) We find also from the table that the width of face must be $2\frac{1}{2}$ in. to give the same strength, assuming the load as uniformly spread over the length of face. As the speed is slow, we use the value of P_1 for 100 ft. per minute or under. It is common practice to make the breadth of the tooth not less than two to three times the pitch.

Steel Rollers in Centre Bearing.—Making the rollers hard steel on hard-steel bearing-plates, we can allow a pressure per lineal inch of roller of 1750 $\sqrt{d}$; d being the average diameter of roller. Calling this average diameter 2.5″, we have 2765 lbs. allowed pressure per lineal inch. The weight of the span is 89,200 lbs., and this divided by 2765 gives 32.2 lineal inches required. There are 15 rollers, 3 in. long, giving 45 in. actual.

If a centre-pin is not used, care should be taken to give the ends of the rollers an even bearing to resist the lateral pressure as explained above. The plates or rings between which the rollers move should be thick enough to distribute the pressure evenly and so that there will be no give or spring as the span revolves. For three-inch rollers the plates should not be less than 2¾ to 3 in. thick.

If a pivot with flat disks had been used (see details of this form of centre), the coefficient of friction would have been about 0.1 (see table of allowed bearing on disks of steel and bronze). The centre of pressure on pivots is at two thirds the radius from the centre.

Wedging Arrangement at Centre and Ends.—The centre roller-bearing is supposed to carry dead load only. To support the span under live load, wedges or some equivalent device are used under the girders at the centre and at the ends. The supports at the centre should be driven just hard enough to bring them to a full solid bearing, but not hard enough to take the dead load off the centre pivot or rollers. The amount the end wedges should drive is determined by the amount of deflection it is found necessary to take out of the girder so that there shall be no raising of the ends off the supports as the load passes over one arm. The gears or levers moving the wedges are easily arranged to give any desired amount of motion to either set. The amount it is necessary to raise the ends of the girder will now be considered. Placing the engine on one arm with the heavier wheels at the centre, we find the reaction at the end of unloaded arm to be 7070 lbs. (see Fig. 9.) This means that

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a force of 7070 lbs. must be applied at the end of unloaded arm to prevent its raising off the support. This force may be obtained by driving the wedges under the ends of the girder, and giving it an upward deflection until it is strained sufficiently to give the reaction required.

Our formula for the deflection from an end load and girder of varying section is, from page 26, No. 4*a*,

$$D = \frac{Wl^3}{55,700,000I}.$$

We have $W = 7070$ lbs., $l = 42.5$ ft. $= 510$ in., $I = 91,946.8$.

$$D = \frac{7070 \times 132,615,100}{55,700,000 \times 91,946.8} = .185 = \tfrac{3}{16} \text{ inch.}$$

Our wedge must then have a vertical movement of something over $\frac{3}{16}$ in. If we make the slope of the wedge 1 in 5, a horizontal throw of 12 in. will give us ample clearance for turning.

The horizontal force necessary to drive the wedge will be $\frac{7070}{6}$ ($\frac{1}{6}$ being the slope of the wedge) plus the friction of the top and bottom surfaces of the wedge on their bearings. This friction we will assume as 236 lbs. Then $\frac{7070}{6} + 236 = 1414$, which is the horizontal force to be applied. The coefficient of friction might be as high as 0.10. At this value we have $\frac{7070}{6} + 707 + 707 = 2592$ lbs. as against the 1414 lbs. we are now using. It will be noticed that the friction is an important element in determining the actual power to be derived from the wedge.

The centre wedges should not be driven hard enough to lift the span off the centre support, but just to a solid bearing. We will assume, however, for the present that all six wedges are driven with a force of 1414 lbs. each. This will give us an excess of power of about 50 per cent. One man, it was assumed, could exert a force of 75 lbs. The power must then be multiplied between the man and the wedges. $1414 \times 6 = 8484 \div 75 = 113.2$ times. Using a 60-inch lever and the worm-screw arrangement as shown in Fig. 46, in one revolution of the shaft the man moves $120 \times 3.14 = 376.8$ ft. The pitch of the screw is, say, $2\frac{1}{2}$ in., or there is a vertical motion of $2\frac{1}{2}$ in. Dividing 376.8 by $2\frac{1}{2}$ gives 150.7 as the multiplication of power, against 113.2 required. We do not then need to increase the power further, and all arms on the shafts may be of the same length. If the rods connecting centre and end shafts are on one side of the bridge only, that is, if one set only are used (sometimes one and sometimes two are employed; if the bridge is wide, there should be a set on each side), these rods will carry a strain of $1414 \times 2 = 2828$ lbs. each. Rods $\frac{5}{8}$ or $\frac{3}{4}$ in. round will be ample. The worm-shaft has a twisting moment of $75 \times 60 = 4500$ in.-lbs.; by the table on shafting we see that this requires a shaft of, say, $1\frac{7}{16}$ in. diameter. In order to make a suitable thread for the worm, the shaft ought not to be less than $5\frac{1}{2}$ or $5\frac{3}{4}$ in. diameter. So in this case the worm would determine the size of shaft to use.

The angle of repose for steel on cast iron is, say, 11°. The thread of the worm should then have a slope not exceeding 10° or 12°. If the pitch is $2\frac{1}{2}$ in., the thread rises $1\frac{1}{4}$ in. in one half-revolution, and the angle is found by dividing this rise ($1\frac{1}{4}$ in.) by the diameter of screw on the pitch-line. Assuming this to be 5.8, we have $1.25 \div 5.8 = .21 =$ tangent of 12°. Rather than use a shaft of this diameter, it would be better to make the worm in the form of a sleeve, and key it to a $2\frac{1}{4}$ or $2\frac{1}{2}$ in. shaft. Or a shaft $3\frac{1}{2}$ or $3\frac{3}{4}$ in. in diameter might be used with a worm of $1\frac{1}{4}$ or $1\frac{1}{2}$ in. pitch. The objection to this arrangement for such a light span is that the time required to operate the machinery is made unnecessarily great. We will assume that the worm is made in the form of a sleeve and has a diameter at the centre of the thread (or pitch-line) of 5.8 in.

Horizontal Shafts.—We found that we multiplied our power between the end of the turning-lever and the sliding-or worm-nut on the vertical shaft 150 times. The force exerted by one man at the end of the turning-lever was assumed as 75 lbs. Then $75 \times 150 = 11,250$ would be the force exerted upon the sliding-nut, were not a portion of this used in overcoming the friction of the various parts. We will first determine what these frictional forces are, up to the point where the nut-lever acts on the horizontal shaft. These forces being found and subtracted from 11,250 will give us the force that the horizontal shaft must carry on to the wedges.

We have, first, the friction of the bearings of the vertical shaft; second, the friction on the collars from the thrust of the vertical shaft; third, the friction of the sliding-nut in its guides; and fourth, the friction of the sliding-nut on the thread of the worm-shaft.

These are all sliding frictions for which the coefficient would be between 0.05 and 0.1, depending upon the smoothness of the surfaces and the amount and character of the lubrication. We will use 0.06.

The horizontal pressure on the journals is the 75 lbs. exerted by the man at the lever increased by the leverage due

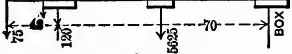

Fig. 17

to the bearing being some distance below the lever. Suppose the lever to be 42 in. above the box, and that the play in the box is sufficient so that the lower box might be assumed as resisting this bending; then we have $75 \times 42 \div 70 = 45$, as bearing on lower box. There is also the horizontal pressure from the worm-nut in its guides. This is equal to $75 \times 60 \div 8 = 562.5$ lbs. (Eight inches being the distance from the centre of the shaft to the centre of bearing of the nut on its guides.) Forces causing friction on the bearings are then $120 + 45 + 562.5 = 727.5$ lbs. If we use a $2\frac{1}{2}$-in. shaft,

$$727.5 \times .06 \times 1.25 \div 60 = .91, \quad . \quad . \quad . \quad (1)$$

the force at end of lever to overcome this friction (1.25 being the radius of the shaft, and 60 the length of the hand-lever).

Friction of the Collars.—The vertical thrust on the shaft we found to be 11,250 lbs. This acts on the collars with a leverage (the distance to the centre of gravity of the ring) of, say, $1\frac{3}{4}$ in.; then $11,250 \times .06 \times 1\frac{3}{4} \div 60 = 19.7$, the force at end of hand-lever. This is excessive, and the friction should be reduced by using a ball bearing in the collars (see detail of this arrangement in cuts). This reduces the friction to rolling instead of sliding friction, and the coefficient to .003; we have then

$$11,250 \times .003 \times 2 \div 60 = 1.12. \quad . \quad . \quad . \quad (2)$$

Friction of Worm-nut Sliding in its Guides.—The horizontal pressure of the nut we found to be $75 \times 60 \div 8 = 562.5$. Then

$$562.5 \times .06 \div 150 = 0.22. \quad . \quad . \quad . \quad . \quad (3)$$

(The number 150 is the number of times the power is multiplied between the hand-lever and the nut.)

Friction on the Worm-thread.—The vertical pressure is 11,250; and if the slope of the thread is 12°, this gives a force in the direction perpendicular to the screw-thread of $11,250 \div 1.022 = 11,008$. $11,008 \times .06 = 660.48$.

We will assume that the force at end of hand-lever necessary to overcome this friction is 4.4 lbs. This force is equal to the friction multiplied by the radius of the worm-thread, divided by the length of the hand-lever. In some cases the friction may reduce the efficiency of the worm 40 to 50 per cent. (See page 86.)

The sum of these frictions is $.91 + 1.12 + 0.22 + 4.4 = 6.65$ lbs. Subtracting this from 75 gives $75 - 6.65 = 68.35$, the available power at hand-lever. 68.35×150 (the number of times power multiplies) $= 10,253$, the power transferred by worm-nut to the arms on the horizontal shaft.

The horizontal shafts have, in addition to the twisting moment, the bending due to the distances between the bearings and the various levers which are keyed to the shafts. On the centre shaft we have the levers or arms working the struts which draw the centre wedges, the arms driving the rods to the end wedges, and the arms working into the worm-nut.

On the end shaft we have the arms working the end wedges, arms worked by long rods running to centre, and the cranks which work the rail-lifts. The twisting moment extends nearly uniformly through the centre shaft if the centre wedges are only driven to a bearing, and there are rods running to the end shafts on each side of the bridge. If the rods are on one side only, the moments of the twisting force will be greatest between the worm-nut lever and the end of shaft carrying the rod-arms.

In the end shaft, with one set of driving-rods, the moment is greatest between the arms driven by the long rods and the strut driving the end wedge. Then it is reduced by the amount of the moment on the wedge strut-arm. It is again reduced by the amount of rail-lift moment when this point has been passed, and so on to the other end.

With two sets of the driving-rods the moment at the centre would be 0, and increase each way to the ends. For the bending moments the portion of shaft between bearings will be considered as a single span, and the bending moments in each portion combined with the twisting moment (see table and formulæ for shafts).

The distance from one arm or prong of the lever working in the worm-nut to the nearest bearing is, say, 8 in., and as each prong carries half the load, the bending moment will be 10,253 ÷ 2 × 8 in. = 41,012 in.-lbs. The twisting moment is, if there are driving-rods on each side, 5126.5 × 11 = 56,391 in.-lbs. (11 being the length of the arm or prong from the centre of the shaft). If the driving-rods are on one side only of the bridge and run from the centre to the end on opposite sides, for opposite ends as in Fig. 19, the moments

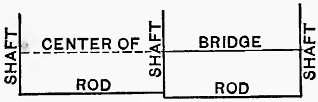

Fig. 18

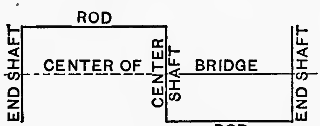

Fig. 19

are the same; but if the rods are on the same side, as in Fig. 18, the moment will be 10,253 × 11 = 112,783.

The first arrangement should of course be used, and we have bending moment = 41,012 and twisting moment 56,391. Our formula (see notes on shafting) is

$$T' = M + \sqrt{M^2 + T^2}.$$

$$T' = 41{,}012 + \sqrt{4{,}841{,}928{,}825} = 41{,}012 + 69{,}584 = 110{,}596.$$

By the table a shaft of 4 in. diameter is required for this moment.

The bearings should be placed as near the points of loading as possible.

The bearings of the horizontal shaft at the centre of the bridge carry a pressure of twice the vertical force at nut-lever, or 20,506 lbs. Using a coefficient of 0.06, and remembering that the lengths of all levers on this shaft are 11 in., we have power lost in friction on this shaft 20,506 × .06 × 2 ÷ 11 = 223.7 lbs., and the shafts at the ends of bridge, including rail-lifts, have about the same amount (it would be figured in precisely the same manner), 10,253 − (224 + 224) = 9805 lbs., available for driving wedges, or 1634 lbs. to a wedge against 1414 required. As the machinery is liable to get out of adjustment and the bearings to become dry, there should be at least 100 per cent excess of power. The wedges will

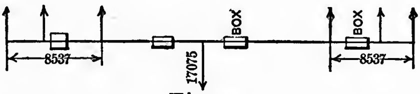

Fig. 20

stick and more power will be required to start them than will be necessary to move them when once started.

Special care should be taken to provide ample means for lubricating the wedges. The surest method is perhaps to make several deep grooves diagonally across the bearing-surfaces. These grooves will retain a large amount of oil and, as the wedges move, spread it over the surfaces. All oil-holes should be easy of access and provided with means for excluding dirt.

The Levers.—The lever-arms and the wings on the sliding-nut should be figured as beams fixed at one end and

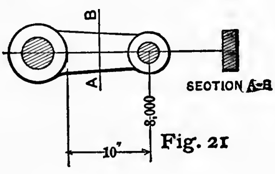

Fig. 21

loaded at the other. For cast iron the fibre-stress should be about 4000 lbs., and for cast steel 15,000 lbs. The bending moment divided by the fibre-stress gives the momennt of resistance required; thus $R = \frac{M}{f}$. $M =$ bending moment, $f =$ fibre-stress, and $R =$ moment of resistance. Say we have a pull of 8000 lbs. at the end of an arm, and the distance to the points where the arm joins the hub is 10 in.; the moment (M) is then 80,000 in.-lbs. if the arm is cast iron. $M \div f = 80,000 \div 4000 = 20$. Using a rectangular section, $R = \frac{1}{6}bh^2$ (see any table on moments of resistance and inertia). Assuming $b = 1.25$, then $20 = \frac{1}{6} \times 1.25 \times h^2$. $h = 2.1$. If a rectangular section is used, it should be stiffened by ribs on the sides if the length exceeds six or eight inches. It must be remembered that these levers are subject to sudden jars, and should be made amply strong. The hubs are weakened by the keyways, and should not be less than $1\frac{1}{4}$ to $1\frac{1}{2}$ in. thick. The keyways should be cut in the side of hub next the arm where there is the greatest excess of metal. A table giving the common sizes of keys used in shafts of different diameters is given below.

Elbow-joint.—We have found that our power has been amply multiplied by the turning-lever and the worm. It may be, and in fact the arrangement of crank on horizontal shaft and the strut driving-wedge should be, such that they increase the power two or more times. When the wedges are driven the crank and strut should be in the same straight line, or nearly so. As the force on the crank acts tangentially to the circle described by its end pin, when the crank and strut are nearly in line this tangential force is capable of exerting an enormous pressure in the direction of the strut. As the angle between the strut and crank increases this force decreases. It will be noticed that when the crank and strut are nearly in line there may be a movement of the crank through a considerable arc and very little movement in the direction of the strut, so that to get the necessary amount of action in the wedge the crank must move to a position where it is not acting to the best advantage. Assuming that the power necessary to drive the wedge increases regularly from 0 at the point where the wedge just takes a bearing to a maximum when the wedge is fully driven, then Figs. 11 to 14 and the explanation below them show how a few trials with the wedge driven to different positions will determine in which one of them the greatest tangential force is required. And this greatest force is the one to be used in determining the moments on the shaft and the power required to turn.

If, when the wedges are driven, the crank and strut stand at a considerable angle, there may be danger of the wedge working loose under the action of live load, especially if the angle of the wedge is steep.

Example of Elbow- or Toggle-joint.—Assume that the horizontal force necessary to raise the end of the girder the required amount be 2000 lbs. moved through a distance *AB*. The horizontal force is zero when the wedge is drawn out, so that the point *A* coincides with the point *B* and increases as the distance between *A* and *B* increases. Assume that when

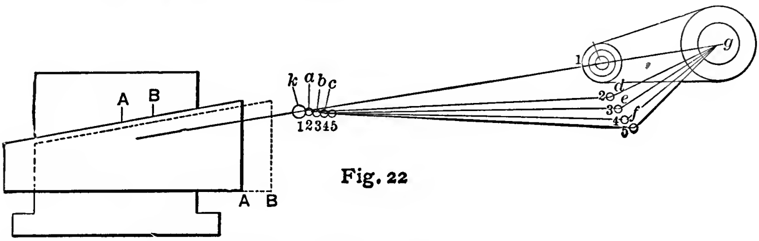

Fig. 22

the wedge is driven the line of the crank and strut will be *ghk*. With wedge moved $\frac{1}{4}$ of *AB* line of lever and strut, assume line *adg*. Moved $\frac{1}{2}$ of *AB*, the line becomes *beg*.

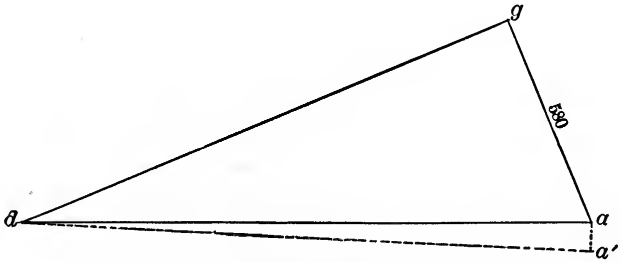

Fig. 23.

Moved $\frac{3}{4}$ of *AB*, the line becomes *cfg*. To find tangential force at end of crank, with end of crank at *d*, lay off from *d*

(Fig. 23) a line parallel with *ad*, also a horizontal line on which lay off the force on wedge at this point = $\frac{3}{4}$ of 2000 = 1500. From *a′* draw a perpendicular to *da′*, intersecting *da* at *a*; through *d* draw *dg*, parallel with *dg* in Fig. 22. Through

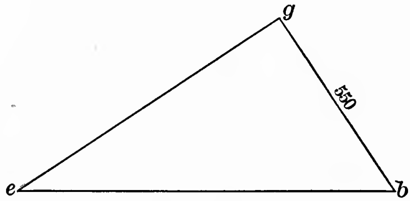

Fig. 24.

a draw *ag* at right angles to *dg*. Line *ag* equals the tangential force at *d*. Figs. 24 and 25 are drawn in the same way and give the tangential force at *e* and *f*.

The Latch.—Several styles of latch are given in the cuts. The object of the latch being to hold the bridge in exact line, it should fit close when driven to place, and it must be strong

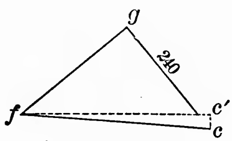

Fig. 25.

enough to hold the bridge against the wind; and if it acts automatically, it must resist the shock of stopping the bridge suddenly as it swings in position. Sometimes the latch drives at the same time as the wedges are driven, but a latch working independently is more satisfactory.

Rail-splices. — The latch should not be relied upon entirely to keep the rails in line, but a sleeve of some sort, slipping over the ends of the rails both on the draw and the abutment, should be used.

Signals.—The levers working the latch or the wedges may also throw danger-signals placed on the abutments, or the span as it revolves may be made to throw them. If there are many attachments to the same set of machinery, some of them are pretty sure to be out of adjustment most of the time. And in general the simpler the machinery of a draw-span is the better. A few heavy, amply strong parts are infinitely better than a great mass of light, complicated pieces; the one will be satisfactory in service, the other never will be.

Set-screws.—While set-screws may be used in places where there is little stress, they are not satisfactory in most places on draw-span machinery. When most needed they can only be relied upon *to fail.* Where used they should be not less than $\frac{3}{4}$ or $\frac{7}{8}$ in. diameter. If two are used at one connection, they should not be placed opposite to each other, but at right angles.

Care of Draw-spans.—To give satisfaction, the best designed draw must have constant care and attention. Many complaints of spans not working satisfactorily are due to gross neglect in their treatment. The writer once went to a draw that was giving trouble, and found that a coil of old rope left on the pier some months previously by bridge-carpenters had become wedged in between the rack and the pinion and wrapped around the shaft, rendering it almost impossible to turn the draw. How often had this part of the machinery been examined in that time? Not once. In fact, some parts out of sight and not easy of access had not been oiled in a year or more. The surest way to insure care in this respect is to have as few parts as possible, and these easy to be seen and reached. Other things being equal, the best design is the one with the fewest parts to keep in repair.

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TABLES AND GENERAL DATA.

Notes on Spur- and Bevel-gears.

PROPORTIONS OF TEETH.

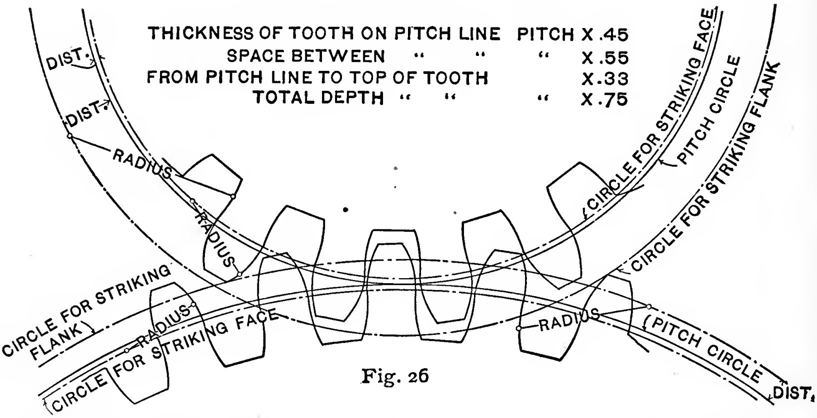


Fig. 26

GRANT'S ODONTOGRAPH TABLE FOR EPICYCLOIDAL TEETH.

Number of Teeth.		For One Diametric Pitch.				For 1" Circular Pitch.			
		For any other pitch diameter divide by that pitch.				For any other pitch multiply by that pitch.			
Exact.	Interval.	Face.		Flank.		Face.		Flank.	
12	12	2.01	.06	oo	oo	.64	.02	oo	oo
$13\frac{1}{2}$	13 to 14	2.04	.07	15.10	9.43	.65	.02	4.80	3.00
$15\frac{1}{2}$	15 " 16	2.10	.09	7.86	3.46	.67	.03	2.50	1.10
$17\frac{1}{2}$	17 " 18	2.14	.11	6.13	2.20	.68	.04	1.95	.70
20	19 " 21	2.20	.13	5.12	1.57	.70	.04	1.63	.50
23	22 " 24	2.26	.15	4.50	1.13	.72	.05	1.43	.36
27	25 " 29	2.33	.16	4.10	.96	.74	.05	1.30	.29
33	30 " 36	2.40	.19	3.80	.72	.76	.06	1.20	.23
42	37 " 48	2.48	.22	3.52	.63	.79	.07	1.12	.20
58	49 " 72	2.60	.25	3.33	.54	.83	.08	1.06	.17
97	73 " 144	2.83	.28	3.14	.44	.90	.09	1.00	.14
290	145 " rack	2.92	.31	3.00	.38	.93	.10	.95	.12
		Rads.	Dist.	Rads.	Dist.	Rads.	Dist.	Rads.	Dist.

SPUR GEAR.

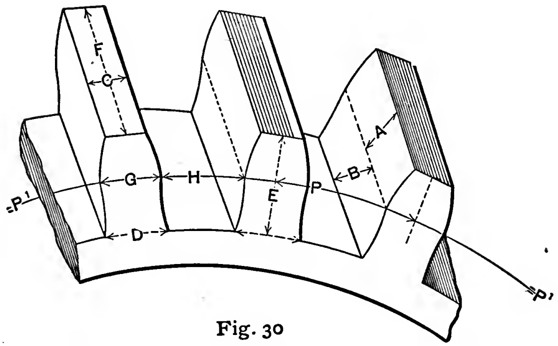


Fig. 30

$P'P'$ = pitch-circle;
A = face;
B = flank;
C = point;
D = root;
E = height;
F = breadth;
G = thickness;
H = space;
P = circular pitch.

RACK.

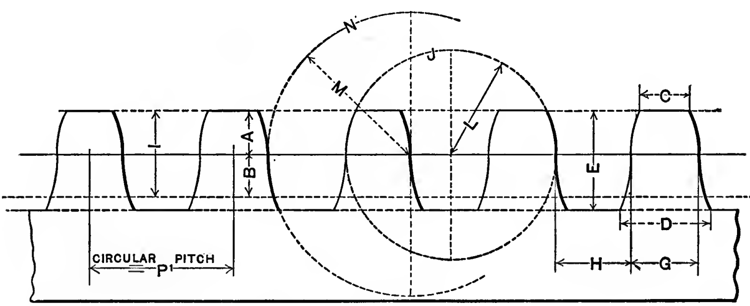


Fig. 31

Double-curve Teeth for Racks and Wheels.
Circle *J* for face-radius $L = P' - \frac{1}{2}$ of G. Circle *N* for flank-radius $M = P'$.

BEVEL AND MITER GEARS.

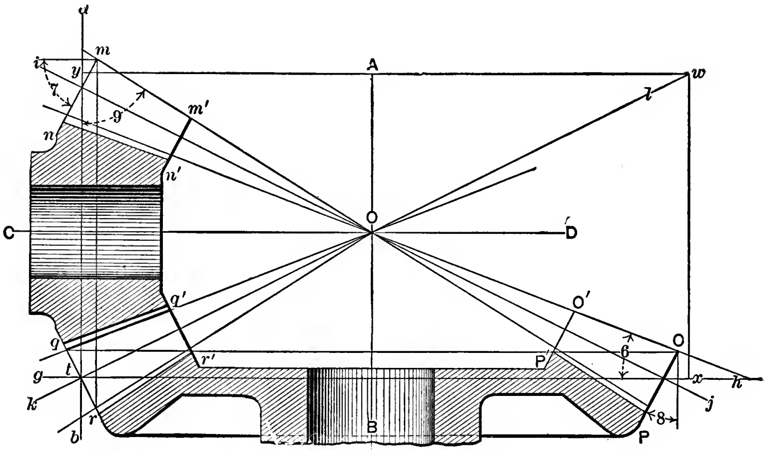
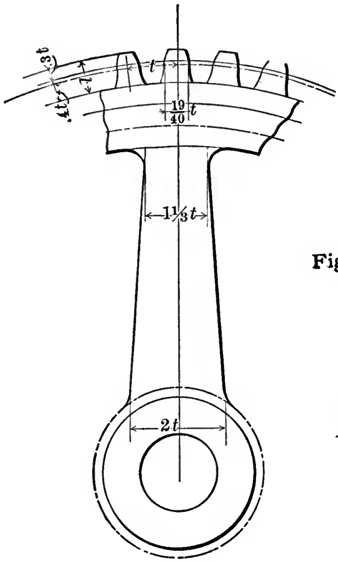


Fig. 32.

AOB = centre of wheel;
COD = " " pinion;
ab = largest pitch diameter of pinion;
gh = " " " " wheel;
OiC and OkC = angles of cone pitch-line of pinion;
OjB and OkB = " " " " " " wheel;
mr = whole diameter of pinion;
qO = " " " wheel;
wt and iO = working depths of tooth.
$\frac{1}{10}$ of $wy + ab = mr$;
$\frac{1}{10}$ of $wx + gh = qO$.
angle 9 = angle of face of pinion;
angle 6 = angle of face of wheel.



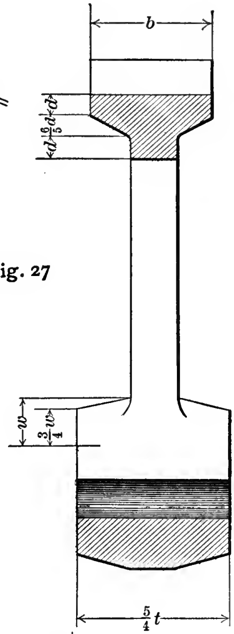


Fig. 27

$d = 0.4t + 0.125''$
$w = 0.4h + 0.4''$

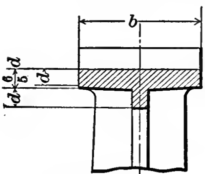


Fig. 27[a]

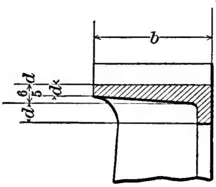


Fig. 27[b]

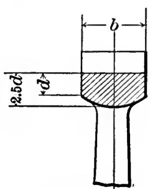


Fig. 27[c]

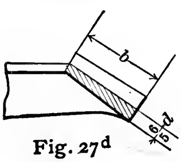


Fig. 27[d]

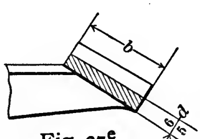


Fig. 27[e]

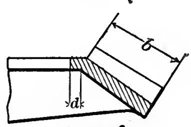


Fig. 27[f]

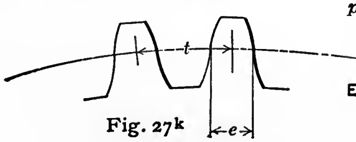


Fig. 27[k]

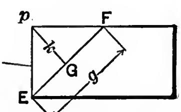


Fig. 28

Gear Teeth.—Cast teeth should be made sufficiently strong to resist the whole force transmitted by a pair of wheels acting on corner of one tooth, and pitch is determined as below (see Fig. 28):

Let e = thickness of tooth $= \frac{19}{40}t$; $EF = g = .99t$; $PG = k = .495t$; P = force at point p; moment of flexure $= Pk$; and greatest stress produced by moment of flexure on section EGF is

$$S = \frac{\text{moment of flexure}}{\text{moment of resistance}} = \frac{6Pk}{ge^2},$$

which is a maximum when angle $PEF = 45°$ and $g = 2k$. Having then the value $S = \frac{3P}{e^2}$, consequently the proper thickness for tooth is given by the equation

$$e = \sqrt{\frac{3P}{S}},$$

in which S may be taken at the values given in the table. e may be assumed to be thickness on pitch-line $= \frac{19}{40}t$; then

$$t = \frac{40}{19}\sqrt{3\frac{P}{S}}, \quad \text{when} \quad h = \frac{19}{40}t.$$

The above method of figuring the tooth is independent of the face of the tooth, and should generally be used when there is a liability of inaccuracies in the teeth.

If the face of the tooth is to be considered, as in machine-cut teeth, the pitch can be assumed and the face (b) obtained from the following.

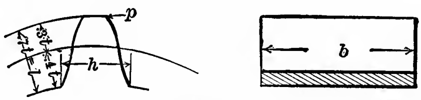


Fig. 29.

STRENGTH OF GEAR-TEETH.

Values of P_1 and P_s = the safe load assumed as applied at the corner of tooth:

$S_1 = 4500$ lbs.
$S_s = 15000$ lbs.
$P_1 = 375t^2$.
$P_s = 1250t^2$.
t = pitch.

Values of P_1 and P_s = the safe load on teeth per lineal inch of face, assuming the load as uniformly distributed over the face at outer edge, for iron and steel at varying velocities as per the table below. V = velocity in feet per minute; S_1 = unit stress for iron; S_s = unit stress for steel.

V	100 Feet or under.	200 Feet.	400 Feet.	600 Feet.	800 Feet.	1000 Feet.	1500 Feet.	2000 Feet.	2500 Feet.
S_1	4500	4310	4020	3730	3440	3150	2860	2470	2180
S_s	15000	14350	13385	12420	11450	10490	9524	8224	7260

VELOCITY IN FEET PER MINUTE.

Pitch.	Safe Load at Corner of Tooth.		100 Feet or under.		200 Feet.		400 Feet.		600 Feet.		800 Feet.		1000 Feet.		1500 Feet.		2000 Feet.		2500 Feet.	
t	P_1	P_s	P_1	P_s	P_1	P_s	P_1	P_s	P_1	P_s	P_1	s	P_1	P_s	P_1	P_s	P_1	P_s	P_1	P_s
1	375	1250	269	896	258	858	240	800	223	743	205	685	188	627	171	569	147	492	129	433
1¼	585	1948	337	1122	323	1075	301	1002	279	930	257	858	236	785	214	713	185	616	163	542
1½	844	2811	404	1345	387	1288	361	1201	335	1115	309	1028	283	942	257	855	222	738	195	650
1¾	1148	3823	470	1565	450	1499	420	1398	390	1297	359	1196	329	1095	299	904	258	859	227	756
2	1500	4995	538	1791	515	1715	480	1600	446	1485	411	1369	376	1254	342	1138	295	983	260	866
2¼	1898	6320	605	2015	579	1930	540	1800	501	1670	462	1540	423	1410	384	1280	332	1105	292	974
2½	2350	7825	673	2241	644	2146	601	2002	558	1857	514	1713	471	1568	428	1424	369	1230	325	1084
2¾	2836	9440	740	2464	709	2360	661	2201	613	2043	565	1883	518	1725	470	1566	406	1353	358	1191
3	3375	11238	807	2687	773	2573	721	2401	669	2228	617	2054	565	1881	513	1708	443	1475	390	1299
3¼	3960	13185	875	2913	838	2790	781	2602	725	2415	669	2227	612	2039	556	1851	480	1599	423	1408
3½	4594	15300	941	3133	901	3000	840	2799	780	2597	719	2395	658	2193	598	1991	516	1720	455	1515
3¾	5271	17555	1008	3358	966	3216	900	3000	836	2783	771	2567	705	2350	641	2134	553	1843	487	1623
4	6000	19980	1076	3583	1030	3432	961	3201	892	2970	822	2739	753	2508	684	2277	591	1967	520	1732

For a pitch t, face b, length of teeth l, and base thickness of tooth h, we have for a tooth-pressure p and fibre-stress S the general formula

$$bt = 6\frac{P}{S}\left(\frac{l}{t}\right)\left(\frac{t}{h}\right)^2$$

and for proportions of teeth given, h being assumed at $\frac{1}{2}t$,

$$bt = 16.8\frac{P}{S}, \quad P = \frac{btS}{16.8}.$$ (See Table, page 51

In any case the breadth of face should not be made less than $1\frac{1}{2}t$, and is generally made from $2t$ to $3t$.

It is found that the breadth of face of the tooth should increase with the increase of p. As the wear on the tooth depends on the breadth, the tooth should be proportioned so that $\frac{pn}{b}$ should not exceed a given amount. For iron $\frac{pn}{b} =$ not more than 28,000. $n =$ number of revolutions per minute.

For small forces this constant may be made as low as 12000 or 6000 without obtaining inconvenient dimensions.

For Hoisting Gears, linear velocity at pitch-circle not exceeding 100 ft. per minute, S may be taken at 42,000.

For Transmission Gears, velocity exceeding 100 ft. per minute, take S from table on page 51, in which $S = \frac{9600000}{v + 2164}$ for cast iron. For steel S may be taken $3\frac{1}{3}S$ for cast iron. $v =$ lineal velocity in feet per minute.

Arms of Gears.—A good proportion for the arms is obtained when their number A is made as follows: *

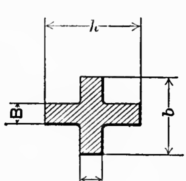


Fig. 27g

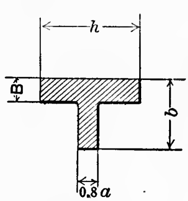


Fig. 27h

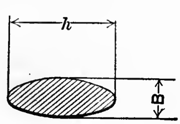


Fig. 27i

* From Releaux.

$A = 0.53 \sqrt{Z} \sqrt[4]{t}$; $Z =$ number of teeth;

$A = 0.73 \sqrt{Z} \sqrt[4]{\frac{t}{\pi}}$. $t =$ pitch.

$A =$	3	4	5	6	7	8	10	12
$Z\sqrt{T} =$	30	53	83	119	162	211	330	475
$Z\sqrt{\frac{t}{\pi}} =$	11	23	36	52	71	93	146	209

Width of arm $h = 2$ to $2.5t$.

For thickness $\frac{B}{b} = 0.07\frac{Z}{A}\left(\frac{t}{h}\right)^2$.

TABLE OF GEAR-WHEEL ARMS.

$\frac{h}{t}$	Value of $\frac{B}{b}$ when								
	$\frac{Z}{A} = 7$	9	12	16	20	25	30	35	40
1.50	0.20	0.28	0.37	0.50	0.62	0.78	0.93	1.08	1.24
1.75	0.16	0.21	0.27	0.37	0.46	0.57	0.69	0.80	0.91
2.00	0.12	0.16	0.21	0.28	0.35	0.44	0.53	0.61	0.70
2.25	0.10	0.12	0.17	0.22	0.28	0.35	0.41	0.48	0.55
2.50	0.08	0.10	0.13	0.18	0.22	0.28	0.34	0.39	0.45
2.75	0.06	0.08	0.11	0.15	0.18	0.23	0.28	0.32	0.37
3.00	0.05	0.07	0.09	0.12	0.16	0.19	0.23	0.27	0.31

Weight of Gears.

The approximate weight G of gear-wheels proportioned according to the preceding rules may be obtained from the following:

$$G = 0.0357bt^2(6.25Z + 0.04Z^2).$$

The following table will facilitate the application of the formula, as it gives the value of $\frac{G}{bt^2}$ for the number of teeth which may be given, and the weight can at once be found by multiplying the value in the table by bt^2.

Z	0	2	4	6	8
Number of Teeth. 20	5.04	5.60	6.18	6.77	7.38
30	7.99	8.61	9.24	9.89	10.52
40	11.09	1.90	12.59	13.30	14.02
50	14.74	15.48	16.23	17.00	17.77
60	18.55	19.35	20.15	20.97	21.80
70	22.65	23.50	24.36	25.24	26.12
80	27.02	27.93	28.85	29.79	30.73
90	31.69	32.66	33.63	34.62	35.63
100	36.63	37.67	38.70	39.75	40.81
120	47.40	48.54	49.69	50.85	52.03
140	59.30	60.56	61.82	63.10	64.27
160	72.35	73.73	75.10	76.39	77.90
180	86.54	88.03	89.52	91.02	92.54
200	101.88	103.48	104.98	106.70	108.34
320	118.36	120.08	122.15	123.52	125.27

For weight of gear-wheels with number of teeth between figures given in left-hand column use weight given on horizontal line through nearest ten below the given number of teeth and under the figure in top line nearest last figure in number of teeth given; thus, 46 teeth = 13.30.

Shafting.

Shafting.—The formulæ and tables given below will be sufficient to enable the size of shaft required for any case likely to occur in the consideration of draw-spans to be readily determined. When the shaft is long and works through a limited number of revolutions the diameter should be large, in order that the angular deflection may not be excessive. The use of too small shafting has been one of the most common faults in draw-span design, and in many cases has led to the renewal of machinery that in other respects would have given satisfactory service.

A deflection of one degree in a length of twenty diameters is considered good practice in millwork, but for drawbridge machinery, if the shaft be long and there are many attachments to it, an angular deflection as great as this may cause the whole arrangement to work badly. The angular deflection for any twisting moment may be determined by the following

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formulæ: A = the angular deflection in parts of one revolution, M = the twisting moment in foot-pounds, L = length of shaft in feet, d = diameter of shaft in inches; then for wrought iron $A = \frac{ML}{30000d^4}$, and for steel $A = \frac{ML}{36000d^4}$.

If the twisting moment M does not exceed $M = 50d^3$ for wrought iron and $M = 60d^3$ for steel, the angle of deflection will not exceed one degree for a length of shaft equal to 20 diameters. Thus if a 3-inch steel shaft have a twisting moment of $M = 60d^3 = 1620$ ft.-lbs., then

$$A = \frac{1620L}{36000 \times 81};$$

and if the length of shaft be 60 ft., then $A = 0.033$.

$360° \times 0.033 = 12°$. A deflection of one degree in 20 diameters = 12°.

Friction of Shaft-bearings.—For the slow motion of a hand-turning draw the friction of the shafts, if well oiled, would probably be about .025 of the pressure; but as the conditions of lubrication as well as the state of adjustment are uncertain, a coefficient of .06 has been used in the example considered. As the speed increases the coefficient will increase, and for higher speeds we may use

$$F = \frac{dl\sqrt{v}}{3.3}.$$

F = coefficient of friction, d = diameter of shaft, l = length of bearing, v = velocity in feet per second. It has been found that for loads up to 600 or 700 lbs. per square inch the friction depends upon the diameter, length of bearing, and velocity, and is independent of the pressure. With heavy loads and high speeds a coefficient of 0.11 should be used.

Collar Friction.—For the coefficient of friction on the collars, 0.06 to 0.1 (depending upon the method of oiling, etc.) should be used. This friction should be considered as acting at the centre of gravity of the ring. For method of reducing friction of collar, where the thrust is heavy, see cut of ball-bearings.

*General Formulæ.**

$$T = .196d^3s \text{ for round shafts; } \quad (a)$$

$$T = .28d^3s \text{ for square shafts. } \quad (b)$$

d = diameter of the shaft in inches;
s = shearing strength in pounds per square inch;
T = the torsional moment in inch-pounds; that is, the force in pounds multiplied by the length in inches of the lever through which the force acts, taking s at 40,000 and 50,000 lbs.; working value = 9000 and 11,200 lbs.

$$T = 1760d^3 \text{ for round iron shafts; } \quad (c)$$

$$T = 2200d^3 \text{ for round steel shafts; } \quad (d)$$

$$T = 2520d^3 \text{ for square iron shafts; } \quad (e)$$

$$T = 3150d^3 \text{ for square steel shafts; } \quad (f)$$

$$d = \sqrt[3]{\frac{T}{1760}} \text{ for round iron shafts; } \quad (g)$$

$$d = \sqrt[3]{\frac{T}{2200}} \text{ for round steel shafts; } \quad (h)$$

$$d = \sqrt[3]{\frac{T}{2520}} \text{ for square iron shafts; } \quad (i)$$

$$d = \sqrt[3]{\frac{T}{3150}} \text{ for square steel shafts. } \quad (k)$$

* Following tables on Strength of Shafting are from Pencoyd Pocket-book.

WORKING PROPORTIONS FOR CONTINUOUS SHAFTING, IRON OR STEEL.

No Bending Action except its Own Weight.

Diameter of Shaft in Inches.	Maximum Safe Torsional Moment in Inch-pounds.	Revolutions per Minute.					Minimum Distance in Feet between Bearings.
		100	150	200	250	300	
		H. P.	H. P.	H. P.	H. P.	H. P.	
1½	5,940	7	10	14	17	20	11.7
1⅝	7,552	9	13	17	21	26	12.4
1¾	9,432	11	16	21	26	32	13.0
1⅞	11,602	13	20	26	33	40	13.6
2	14,080	16	24	32	40	48	14.2
2⅛	16,892	19	29	38	48	58	14.8
2¼	20,048	23	34	46	57	68	15.4
2⅜	23,580	27	40	54	67	80	16.0
2½	27,500	31	47	63	78	94	16.5
2¾	36,603	42	62	83	102	124	17.6
3	47,520	54	81	108	134	162	18.6
3¼	60 417	69	103	137	172	206	19.7
3½	75,460	86	129	172	215	258	20.7
3¾	92,812	105	158	211	264	316	21.6
4	112,640	128	192	256	320	384	22.6

WORKING PROPORTIONS FOR CONTINUOUS SHAFTING, IRON OR STEEL.

Transmitting Power and subject to Bending Action of Pulleys, Belting, etc.

Diameter of Shaft in Inches.	Maximum Safe Torsional Moment in Inch-pounds.	Revolutions per Minute.					Maximum Distance in Feet between Bearings.
		100	150	200	250	300	
		H. P.	H. P.	H. P.	H. P.	H. P.	
1½	5,940	5	7	10	12	14	6.8
1⅝	7,552	6	9	12	15	18	7.2
1¾	9,432	8	11	15	18	22	7.5
1⅞	11,602	9	14	19	23	28	7.9
2	14,080	11	17	23	28	34	8.2
2⅛	16,892	14	21	27	34	42	8 6
2¼	20,048	16	24	33	41	48	8.9
2⅜	23,580	19	29	38	48	58	9.2
2½	27,500	22	33	45	55	66	9.6
2¾	36,603	24	36	48	60	72	10.2
3	47,520	39	58	77	96	116	10.8
3¼	60,417	49	74	98	123	148	11.4
3½	75,460	61	92	123	153	184	12.0
3¾	92,812	75	113	151	188	226	12.5
4	112,640	91	137	183	228	274	13.1

Shafts having Both Bending and Twisting.

$$T' = M + \sqrt{M^2 + T^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (l)$$

M = bending moment in inch-pounds;

T = twisting moments in inch-pounds;

T' = a new twisting moment which, substituted for T in equations g to k, will give the desired proportions for the shaft.

Ratio of M to T.	Factor of Safety.	Divisor in Formulæ.	
		(g) for Iron.	(h) for Steel.
$M = .3T$ or less	$4\frac{1}{2}$	1760	2200
$M = .6T$ " "	5	1570	1960
$M = T$ " "	$5\frac{1}{2}$	1430	1790
M = greater than T	6	1310	1640

Formulæ for Horse-power.

V = revolutions per minute;

HP = 396,000 inch-pounds per minute.

$$HP = \frac{6.28 \times T \times V}{396,000}, \quad T = \frac{63,057\ HP}{V}, \quad d = \sqrt[3]{\frac{36\ HP}{V}} \quad . \quad (o)$$

Deflection of Shafting.

$$l = \sqrt[8]{873 d^2} \text{ for bare shafts; } . \quad . \quad . \quad . \quad . \quad . \quad . \quad (p)$$

$$l = \sqrt[8]{175 d^2} \text{ for shafts carrying pulleys, etc.; } \quad . \quad (r)$$

which would be the maximum distance in feet between bearings for continuous shafting subjected to bending stress alone.

If the length is fixed and we desire the diameter of the shaft, we have

$$d = \sqrt{\frac{l^3}{873}} \text{ for bare shafting; } . \quad . \quad . \quad . \quad . \quad . \quad (s)$$

$$d=\sqrt{\frac{l^3}{175}}$$ for shafting carrying pulleys, etc. . (t)

Working Formulæ.

$$d=\sqrt[3]{\frac{50\,HP}{V}}$$ for bare shafts; . . . . (u)

$$d=\sqrt[3]{\frac{70\,HP}{V}}$$ for shafts carrying pulleys, etc.; (v)

$$l=\sqrt[3]{720d^2}$$ for bare shafts; . . . . (w)

$$l=\sqrt[3]{140d^2}$$ for shafts carrying pulleys, etc. . . (x)

Shafting-keys.

$$k = 0.16 \div \tfrac{1}{5}d; \qquad k' = 0.16 + \tfrac{1}{10}d.$$

Taper of key, .04 in. to .08 in. in 4 in.

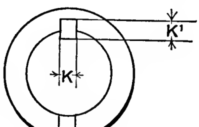

Fig. 33

Shaft	1/2"	5/8"	3/4"	1"	1½"	2"	2½"	3"	3½"	4"	4½"	5
□ Key	3/32"	1/8"	5/32"	7/32"	5/16"	7/16"	1/2"	9/16"	9/16"	5/8"	3/4"	7/8"

From Releaux.

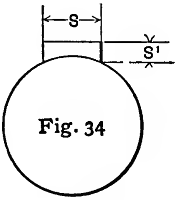

Fig. 34

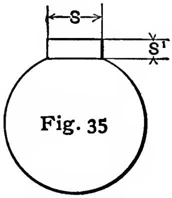

Fig. 35

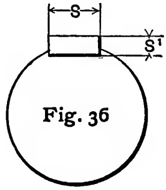

Fig. 36

If we call the diameter of the shaft D, the breadth of the key S, and the middle depth of the key S', we have:

For draft keys, $S = 0.24'' + \frac{D}{7}$; $S' = 0.16'' + \frac{D}{12}$.

For torsion keys, $S = 0.16'' + \frac{D}{5}$; $S' = 0.16'' + \frac{D}{10}$.

The taper of such keys is made about $\frac{1}{100}$.

For the more commonly occurring diameters we have the following proportions:

$D =$	1	2	3	4	5	6	7	8	9	10
					FOR DRAFT KEYS.					
$S =$	3/8″	1/2″	5/8″	13/16″	1″	$1\frac{1}{8}''$	$1\frac{1}{4}''$	$1\frac{3}{8}''$	$1\frac{1}{2}''$	$1\frac{5}{8}''$
$S' =$	1/4″	5/16″	7/16″	1/2″	9/16″	5/8″	3/4″	13/16″	7/8″	1″
					FOR TORSION KEYS.					
$S =$	3/8″	9/16″	3/4″	1″	$1\frac{3}{16}''$	$1\frac{3}{8}''$	$1\frac{9}{16}''$	$1\frac{3}{4}''$	2″	$2\frac{1}{8}''$
$S' =$	1/4″	3/8″	1/2″	9/16″	11/16″	3/4″	7/8″	1″	$1\frac{1}{16}''$	$1\frac{3}{16}''$

For shafts of less diameter than 1 in. we may make

$$S = \frac{D}{3}, \quad S' = \frac{D}{5}.$$

If several keys are used, they may be made the same dimensions as single keys. For hubs which have been forced on, and hence would be secure without any key, the dimensions for draft-keys may be used.

BEARINGS AND PIVOTS, SPRINGS, CAMS, ETC.

Bearings.—The bearings for shafts should be placed as near the points of loading as possible, and for low speeds and small loads the length of bearing should be once and one half to twice the diameter of the shaft. Where the load is heavy or speed great, the bearings are given a length of twice to four times the diameter. Where the bearing simply carries the weight of shaft, a length of once to once and one quarter the diameter is sufficient. Bearings of brass or a composition of metals are used at important points. A bushing of Babbitt metal is found to give excellent results. The friction is low and the wearing properties of this metal are good. Two bearings made in this manner are shown in the cuts. As the speed increases, the length of the bearing should be increased about in the ratio given in table below.

N	=	100	150	200	250	400	750	1000
$l \div d$	=	1.25	1.5	1.75	2.0	2.5	3.5	4.0

N = number of revolutions per minute; l = the length of bearing in inches; d = diameter of shaft in inches.

Ample provision should be made for keeping the bearing well oiled, and all oil-holes should be easy of access. To aid in spreading the oil over the whole bearing-surface small grooves are often cut spirally around the bearing.

For thickness of metal and proportion of the various parts see cuts 39 and 40.

Load on Rollers.—*Seller's Centre.*—The rotating load per lineal inch on steel roller should not exceed that given by the following formula for steel rollers on steel plates:

$$P = 2625 \sqrt{d}.^*$$

P = pressure per lineal inch of roller;

d = mean diameter of roller in inches.

Load on Wheels.—The load per lineal inch of face of wheel, while span is turning, should not exceed that given by the following formulæ, viz.:

$P = 705 \sqrt{d}$ for a cast-iron wheel on a cast-iron track;

$P = 900 \sqrt{d}$ " " " " " wrought-iron track.

For steel wheels use the following formulæ as to limit of pressure per lineal inch of wheel-face while the span is turning, viz.:

$P = 1905 \sqrt{d}$ for a steel wheel on a cast-iron track;

$P = 1515 \sqrt{d}$ " " " " wrought-iron track;

$P = 1750 \sqrt{d}$ " " " " steel track.

---

* It is often specified that the load shall not exceed $P = 1750 \sqrt{d}$.

In which formulæ

$P =$ allowed pressure per lineal inch of face of wheel;
$d =$ diameter of wheel in inches.

Pivots.

Formulæ for Pivots.

Wrought Iron or Steel on Bronze.

Slow-moving pivots $\begin{cases} p = 1422. \\ d = 0.035 \sqrt{P}. \end{cases}$

$n =$ or < 150 $\begin{cases} p = 700. \\ d = 0.05 \sqrt{P}. \end{cases}$

$n > 150$ $\begin{cases} a = 75. \\ d = 0.004 \sqrt{Pn}. \end{cases}$

Cast Iron on Bronze.

Slow-moving pivots $\begin{cases} p = 700. \\ d = 0.05 \sqrt{P}. \end{cases}$

$n =$ or < 150 $\begin{cases} p = 350. \\ d = 0.07 \sqrt{P}. \end{cases}$

$n > 150$ $\begin{cases} a = 75 \\ d = 0.006 \sqrt{Pn}. \end{cases}$

Iron or Steel on Lignum Vitæ.

Slow-moving pivots $\begin{cases} p = 2844 \\ d = 0.017 \sqrt{P}. \end{cases}$

$n =$ or < 150 $\begin{cases} p = 1422. \\ d = 0.035 \sqrt{P}. \end{cases}$

$n > 150$ $\begin{cases} p = 1422. \\ d = 0.035 \sqrt{P}. \end{cases}$

Table of Safe Load for Steel on Bronze.

d.	$0.035 \sqrt{P}$ Slow.	$0.05 \sqrt{P}$. Under 150R.	$0.07 \sqrt{P}$. Over 150R.
	Load.	Load.	Load.
1	816	398	204
1.25	1,275	622	319
1.50	1,836	895	459
1.75	2,500	1,219	625
2.00	3,265	1,592	816
2.25	4,132	2,016	1,033
2.50	5,102	2,488	1,275
2.75	6,173	3,011	1,543
3.00	7,347	3,494	1,836
3.25	8,622	4,205	2,155
3.50	10,000	4,877	2,500
3.75	11,479	5,599	2,869
4.00	13,061	6,370	3,265
4.25	14,745	7,192	3,686
4.50	16,530	8,063	4,132
4.75	18,418	8,983	4,604
5.00	20,498	9,954	5,102
5.25	22,140	10,974	5,535
5.50	24,694	12,044	6,673
5.75	26,990	13,164	6,747
6.00	29,388	14,334	7,344
6.25	31,890	15,630	7,972
6.50	34,490	16,900	8,623
6.75	37,190	18,220	9,298
7.00	41,690	19,600	10,000

The above table is made from the formula $P = 816d^2$ for slow speeds, and $P = 816d^2 \dfrac{a}{n}$ for high speeds. For cast iron on bronze use one half the above values and for steel or iron on lignum vitæ use double the values given in the table. $n =$ the number of revolutions per minute, $p =$ the pressure per square inch, $P =$ total pressure, $d =$ diameter of pivot, and the constant $a = 75$.

Formulas for Springs.

By GEORGE R. HENDERSON, Mechanical Engineer, N. & W. R. R

For Elliptic Springs. — P = maximum static load in pounds; S = corresponding fibre-strain in leaves taken at 80,000 lbs.; N = number of leaves (in full elliptic), half the total leaves; B = width of leaves in inches H = thickness of leaves in inches; L = span (or length) of spring in inches when loaded; F = deflection of spring under load P in inches; E = modulus of elasticity taken at 30,000,000. Then

$$P = \frac{2SNBH^2}{3L}, \text{ and reducing } P = \frac{53333NBH^2}{L}.$$

For half elliptic $F = \frac{55PL^3}{16ENBH^3}$, and reducing $F = .000611\frac{L^2}{H}$.

For full elliptic $F = \frac{12PL^3}{16ENBH^3}$, and reducing $F = .00133\frac{L^2}{H}$.

For Helical Springs.—P = load when spring is down solid, in pounds; S = maximum shearing fibre-strain in bar taken at 80,000; D = diameter of steel in inches; R = radius of centre of coil in inches; L = length of bar before coiling in inches; G = modulus of shearing elasticity taken at 12,600,000; F = deflection of spring under load, in inches; H = height of spring free in inches; h = height of spring solid in inches; $\pi = 3.1416$. Then

$$P = \frac{S\pi D^3}{16R};\ F = \frac{32PR^2L}{G\pi D^4};\ H = \frac{LD}{2\pi R};\ H = h + F;$$

and substituting proper constant,

$$F = .08\frac{R^2H}{D^2};\ H = h(1 + .08\frac{R^2}{D^2};\ P = 15.714\frac{D^3}{R}.$$

The most generally preferred ratio for size is $D = 5d$, where D = outside diameter of coil. It is customary to make the static load about one half the solid load.

Helical Springs.

By D. K. CLARK.

$$E = \frac{d^3 \times w}{D^4 \times C}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$D = \sqrt[3]{\frac{w \times d}{3}} \text{ for round steel;} \quad . \quad . \quad . \quad (2)$$

$$D = \sqrt[3]{\frac{w \times d}{4.29}} \text{ for square steel.} \quad . \quad . \quad . \quad (3)$$

E = compression or extension of one coil, in inches; d = diameter from centre to centre of steel bar composing the spring, in inches; w = the weight applied, in pounds; D = the diameter, or the side of square, of the steel bar of which the spring is made, in sixteenths of an inch; C = a constant which, from experiments made, may be taken as 22 for round steel and 30 for square steel.

ECCENTRICS.

Eccentrics.—An eccentric is nothing more than a crank in which (if the crank-arm is R and the shaft diameter D) the crank-pin diameter d' is made so great that it exceeds $D + 2R$, or is greater than the shaft and twice the throw. The simpler forms of eccentric construction are shown in the illustrations. The most practical of these is that shown in Fig. 37*b*, the flanges on the strap, as shown in the section, serving to retain the oil and insure good lubrication.

The breadth of the eccentric is $1\frac{1}{2}d$ to $3d$, the same as that of the equivalent overhung journal subjected to the same pressure. For the depth of flange a we have

$$a = 1.5e = 0.07l + 0.2$$

From which the other dimensions can be determined as in the illustrations

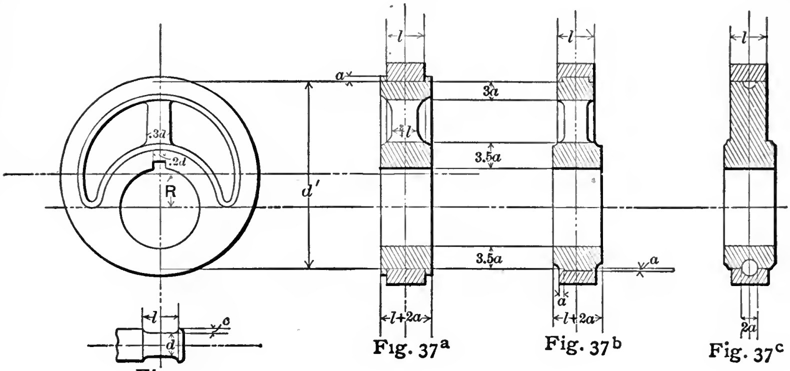

Fig. 37 Fig. 37[a] Fig. 37[b] Fig. 37[c]

Hooks.

Formulas prepared by the YALE & TOWNE MANUFACTURING CO.

Δ = capacity of hook in tons of 2000 lbs.

$D = .5\Delta + 1.25$	$G = .75D$;
$E = .64\Delta + 1.60$	$O = .363\Delta + .66$;
$F = .33\Delta + .85$;	$Q = .64\Delta + 1.60$;
$H = 1.08A$;	$L = 1.05A$;
$I = 1.33A$;	$M = .50A$;
$J = 1.20A$;	$N = .85B - 16$
$K = 1.13A$;	$U = .866A$.

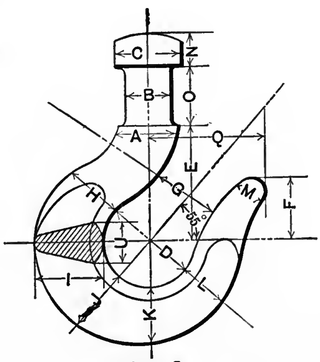

Fig. 38

Capacity of hook......	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	1	$1\frac{1}{2}$	2	3	4	5	6	8	10	tons.
Dimension A.........	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{7}{8}$	$3\frac{1}{4}$	in.

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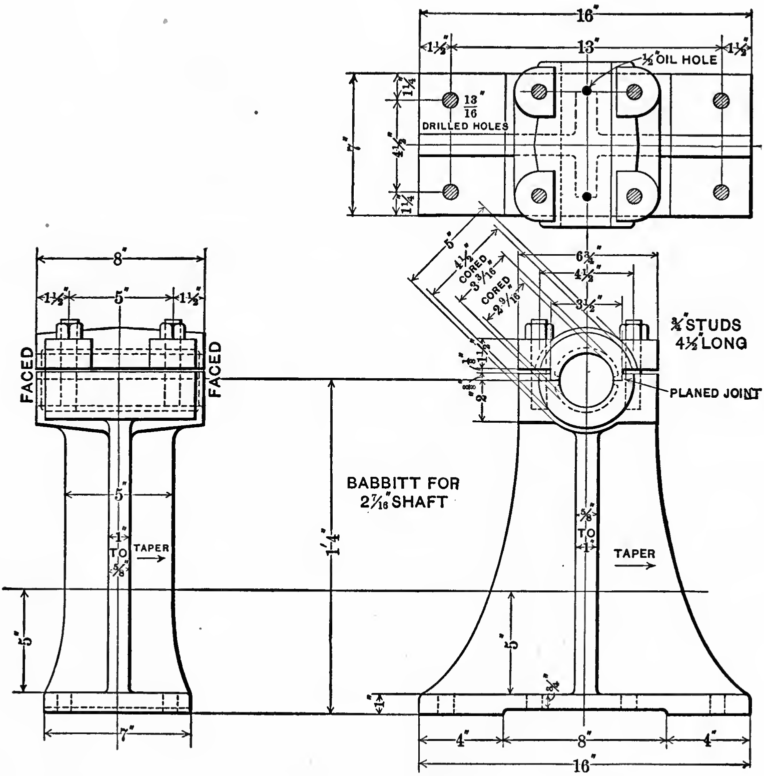


Fig 39. Fig. 39a.

SHAFT-BEARING.

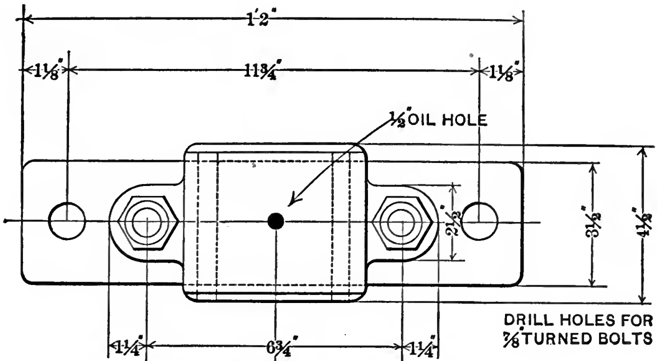


Fig. 40a

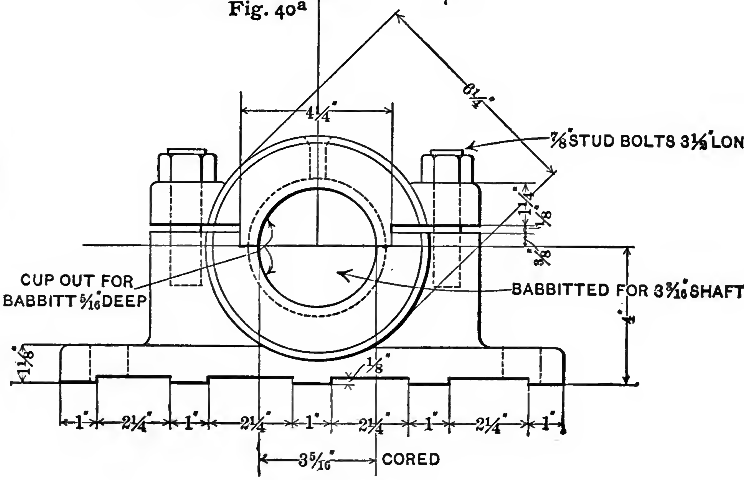
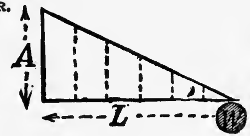
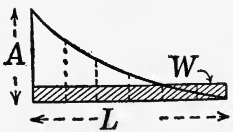
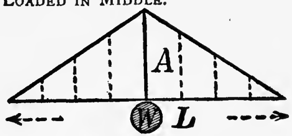
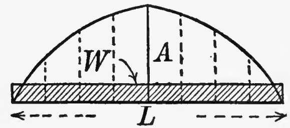
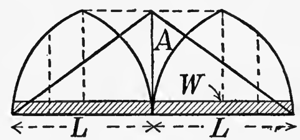


Fig. 40.

SHAFT-BEARING.

BENDING MOMENTS AND DEFLECTIONS FOR BEAMS OF UNIFORM SECTION.

Form of Beam and Position of Load.	Max. Bending Moment.	Max. Shearing Stress.	Deflection.
(1) BEAM FIXED AT ONE END, LOADED AT OTHER.  Draw triangle having $A = W.L.$ Vertical lines give bending moments at corresponding points on beam.	At point of support $= WL.$	At point of support $= W.$	At end of beam $= \frac{WL^3}{3EI}.$
(2) BEAM FIXED AT ONE END, LOAD UNIFORMLY DISTRIBUTED.  Draw parabola having $A = \frac{W.L}{2}.$ Ordinates give bending moments at corresponding points on beam.	At point of support $= \frac{WL}{2}.$	At point of support $= W.$	At end of beam $= \frac{WL^3}{8EI}.$
(3) BEAM SUPPORTED AT BOTH ENDS, LOADED IN MIDDLE.  Draw triangle having $A = \frac{W.L}{4}.$ Vertical lines give bending moments at corresponding points	At middle of beam $= \frac{WL}{4}.$	At point of support $= \frac{W}{2}.$	At middle of beam $= \frac{WL^3}{48EI}.$
(4) BEAM SUPPORTED AT BOTH ENDS, LOAD UNIFORMLY DISTRIBUTED.  Draw parabola having $A = \frac{WL}{8}.$ Ordinates give bending moment at corresponding points on beam.	At middle of beam $= \frac{WL}{8}.$	At point of support $= \frac{W}{2}.$	At middle of beam $= \frac{WL^3}{76.8EI}$
(5) CONTINUOUS BEAM ON THREE SUPPORTS. UNIFORM LOAD.  Draw two parabolas having $A = \frac{WL}{8}.$ Ordinates give bending moments.	At middle of support $= \frac{WL}{8}.$	At middle support $= \frac{5}{8}W \times 2.$	At centre of span $= \frac{WL^3}{31.9EI}.$

NOTE.—In case Fig. 45, W = total load on one arm of beam.

W = total load;
L = length of beam;
E = modulus of elasticity;
I = moment of inertia.

DRAW-SPAN MOMENTS AND SHEARS.

(See Fig. 11.)

COEFFICIENTS C' FOR LOADS IN FIRST ARM AND COEFFICIENTS C^2 AND D^2 FOR LOADS IN SECOND ARM.

Number or Panels in Half-span.	*B* *B'*	*C* *C'*	*D* *D'*	*E* *E'*	*F* *F'*	*G* *G'*	*H* *H'*	*I* *I'*	Totals.
4	.0586	.0938	.0820						.2344
5	.048	.084	.096	.072					.300
6	.0406	.0740	.0937	.0925	.0637				.3645
7	.0350	.0656	.0875	.0962	.0875	.0568			.4285
8	.0308	.0586	.0806	.0938	.0952	.0820	.0513		.4923
9	.0274	.0527	.0740	.0891	0960	.0925	.0767	.0466	.5550

COEFFICIENTS D' FOR LOADS IN FIRST ARM.

4	.691	.406	.168						1.265
5	.752	.516	.304	.128					1.700
6	.792	.592	.406	.241	.103				2.134
7	.822	.649	.484	.332	.198	.086			2.571
8	.844	.691	.544	.406	.280	.168	.074		3.007
9	.861	.725	.592	.466	.348	.241	.146	.065	3.444

VALUES OF E' FOR LOADS IN FIRST ARM.

4	.810	.842	.900						
5	.807	.827	.862	.916					
6	.805	.818	.842	.879	.929				
7	.803	.813	.830	.856	.893	.943			
8	.803	.810	.824	.842	.868	.900	.943		
9	.801	.809	.820	.832	.852	.879	.910	.950	

LOADS FOR MAXIMUM NEGATIVE MOMENTS—FIRST ARM.

4	—	—	—	—	—	—	—	—	—
5	—	—	—	—	—	—	—	—	—
6	*B*	*C*					For maximum at *F*		
7	*B*	*C*	*D*	*E*			" " " *G*		
8	*B*	*C*	*D*	*E*	*F*		" " " *H*		
9	*B*	*C*	*D*	*E*	*F*	*G*	" " " *I*		

All loads on second arm in each case. All loads cause negative moments over pier.

LOADS FOR MAXIMUM POSITIVE MOMENTS—FIRST ARM.

									Max. at
4	*B*	*C*	*D*						*B* to *D*
5	*B*	*C*	*D*	*E*					*B* to *E*
6	*B*	*C*	*D*	*E*	*F*				*B* to *E*
6			*D*	*E*	*F*				*F*
7	*B*	*C*	*D*	*E*	*F*	*G*			*B* to *F*
7					*F*	*G*			*G*
8	*B*	*C*	*D*	*E*	*F*	*G*	*H*		*B* to *G*
8						*G*	*H*		*H*
9	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*B* to *H*
9							*H*	*I*	*I*

SHEARS: All loads on second arm cause negative shear in first arm.
Loads moving *A* towards *Z* cause negative shear in first arm
Loads moving *Z* towards *A* cause positive shear in first arm.

P_1 = any load in first arm.
P_2 = any load in second arm.
S_1 = reaction at *A* from P_1 or P_2.
M_2 = moment at pier from P or P_2.
X_0 = distance from *A* to point of zero moment in first arm.
L = length of half-span.

$M_2 = C'P_1L$ or C^2P_2L.
$S_3 = D'P_1$ or D^2P_2.
$X_0 = E'L$.

WEB-STRESSES: Max. stress in any {mem., web,} load moving *A* to *Z*, is when load extends from *A* to piece in question.

Max. stress in any {mem., web,} load moving *Z* to *A*, is when load extends from *Z* to piece in question.

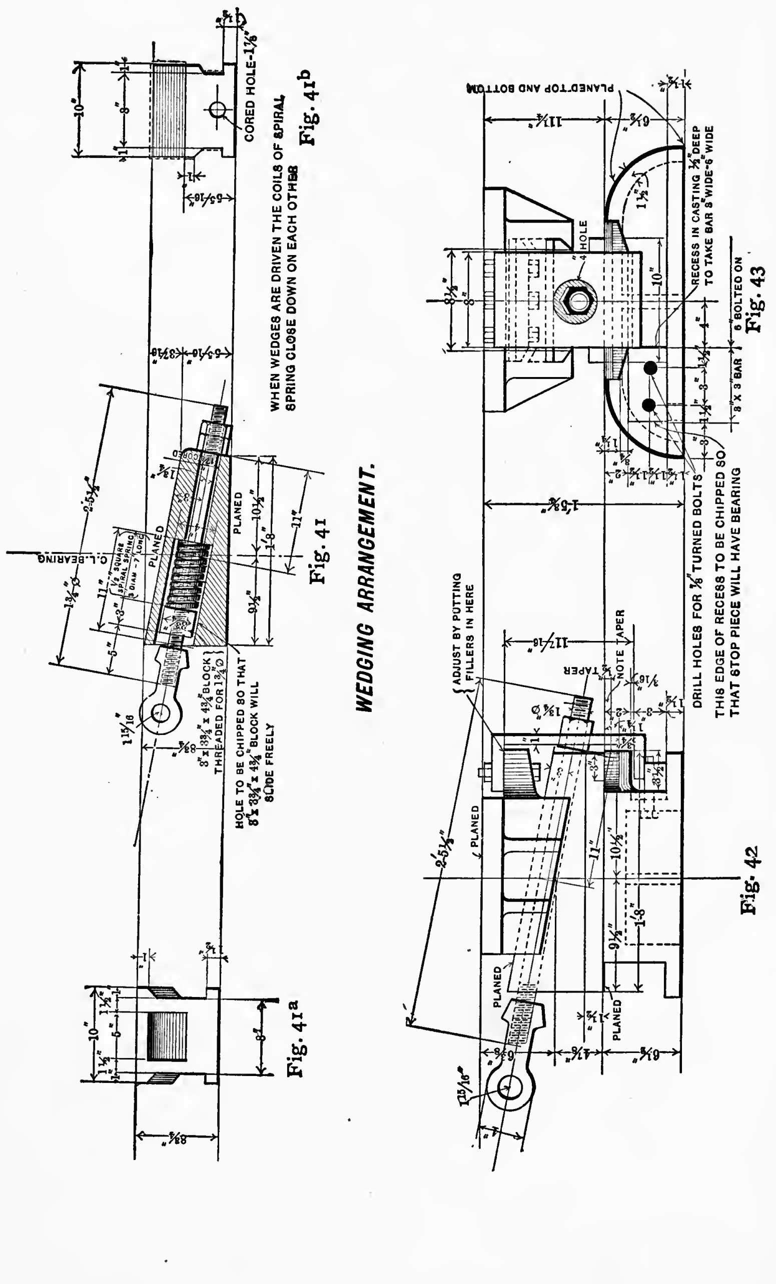


Fig. 41ᵇ

Fig. 41

Fig. 41ᵃ

WEDGING ARRANGEMENT.

Fig. 42

Fig. 43

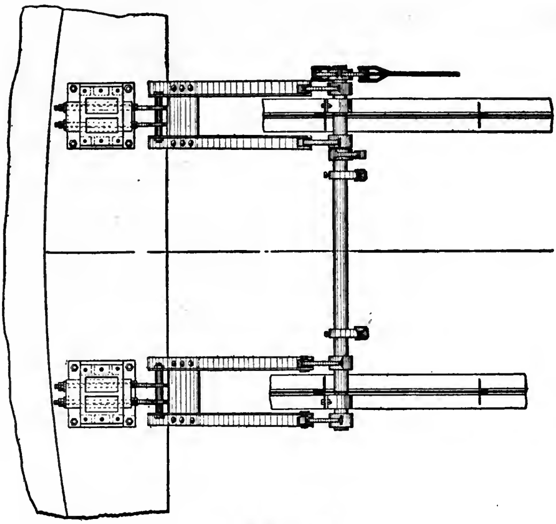


Fig. 44.

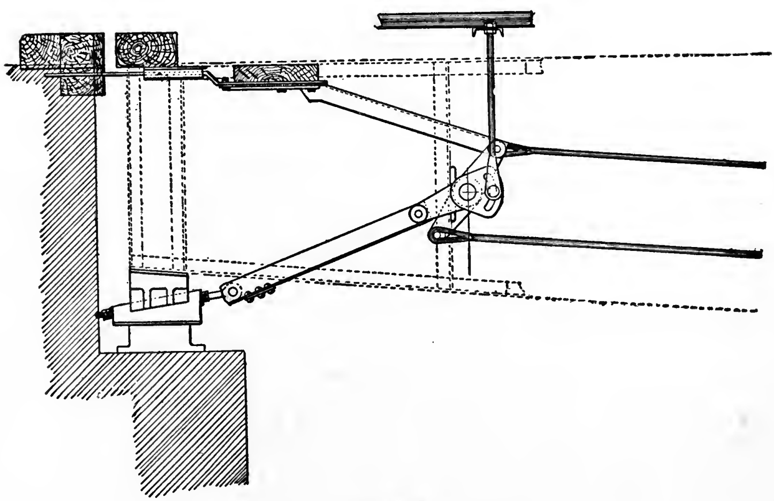


Fig. 45.

END MACHINERY.

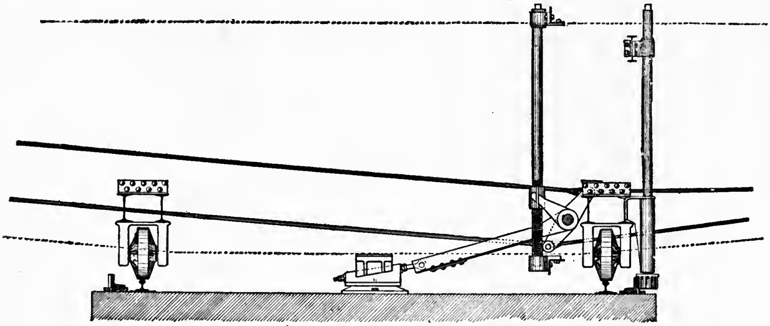


Fig. 46.

CENTRE MACHINERY.

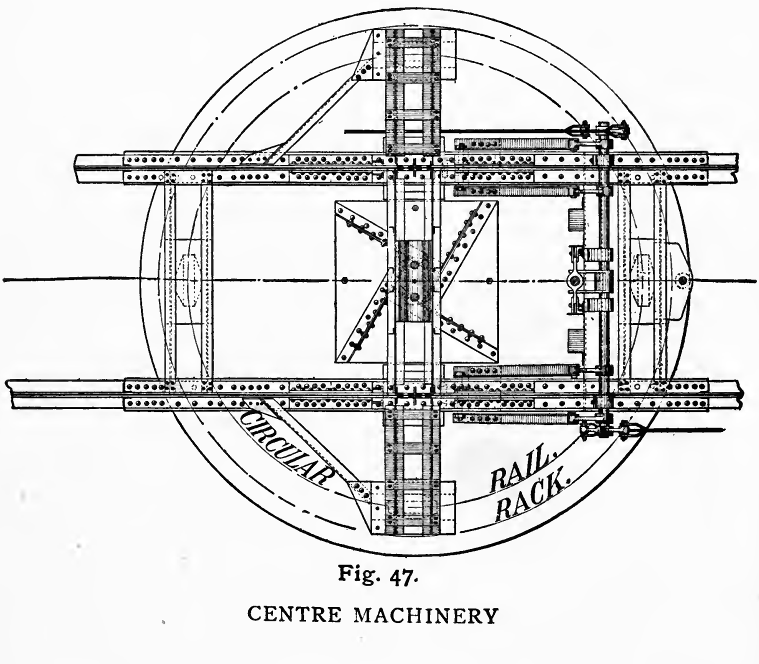


Fig. 47.

CENTRE MACHINERY

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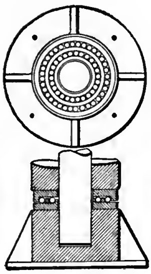

Fig. 48.
BALL-BEARING CENTRE.

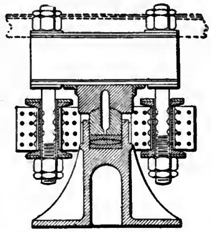

Fig. 49.
PIVOT CENTRE.

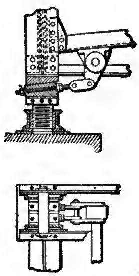

Fig. 50.
ADJUSTABLE END WEDGE.

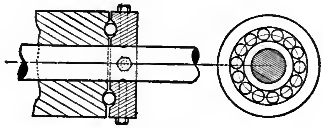

Fig. 51.
SHAFT BALL-BEARING.

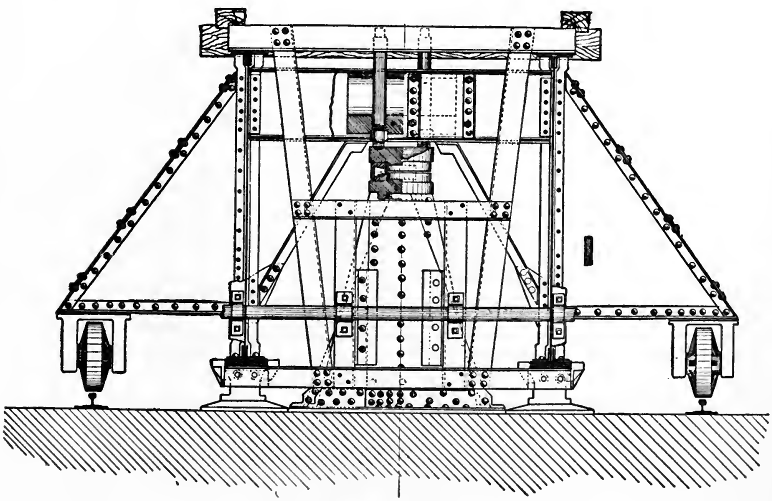

Fig. 52.
CENTRE ON CONICAL ROLLERS.

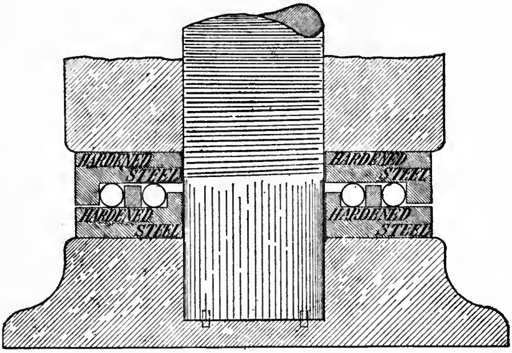

Fig. 53.

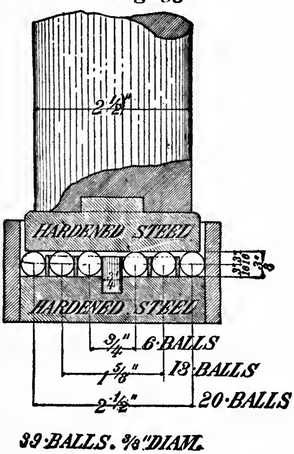

Fig. 54.

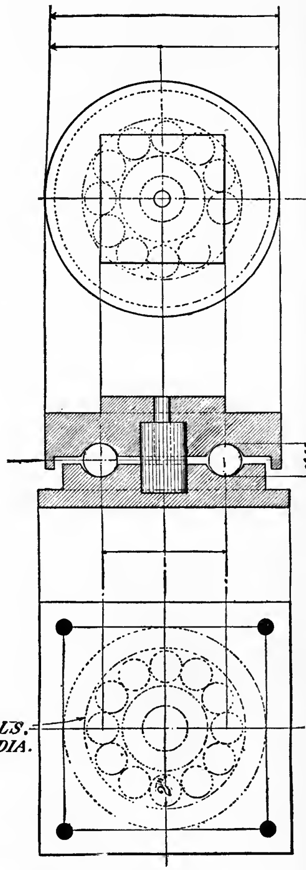

Fig. 55.

BALL-BEARINGS.

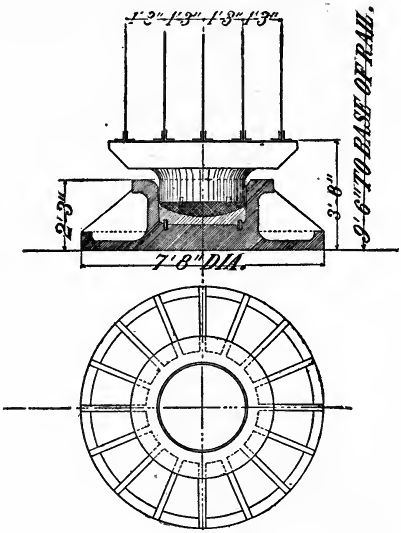

Fig. 56.
CENTRE PIVOT.

Pivot 33″ diam. to be forged in steel. Friction disks turned and ground spherically to a 36″ radius. Upper part steel. Lower part phosphor-bronze. Base of cast iron, to be faced top and bottom, turned inside.

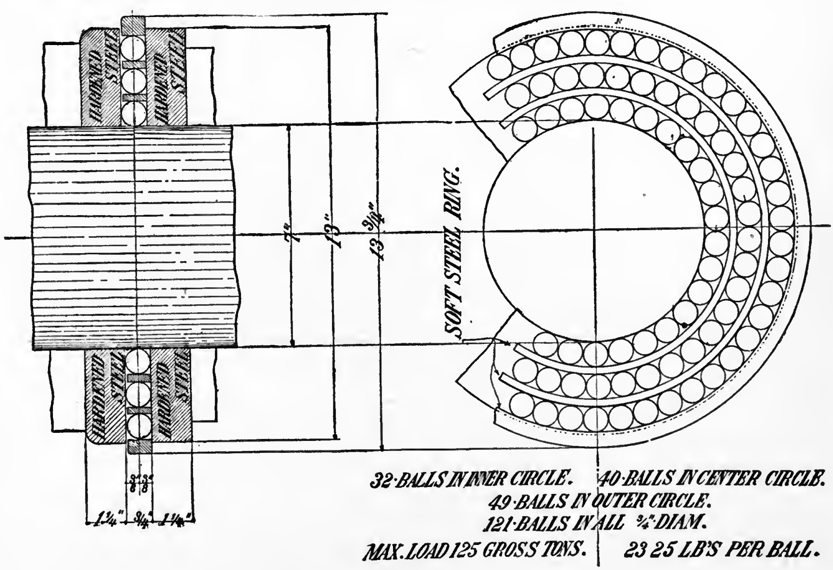

Fig. 57.
BALL-BEARING.

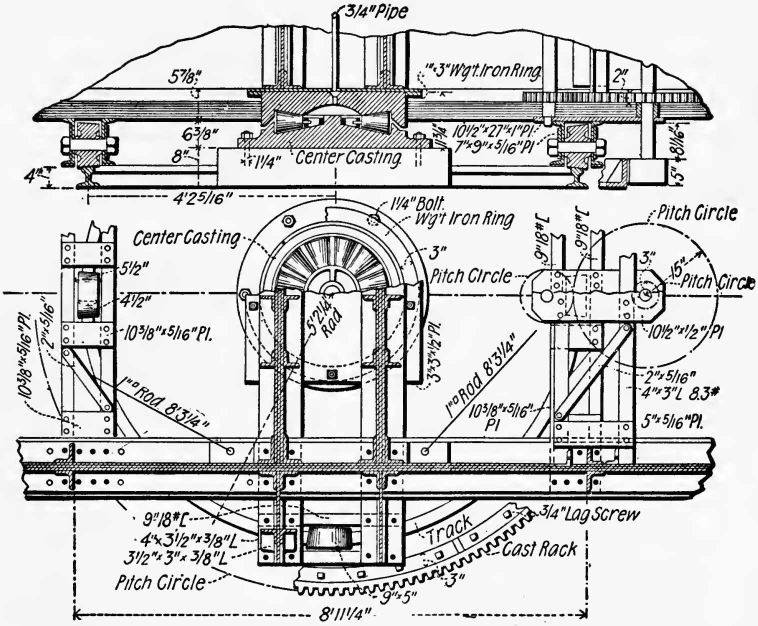

Fig. 58.

CENTRE FOR SMALL DRAW.

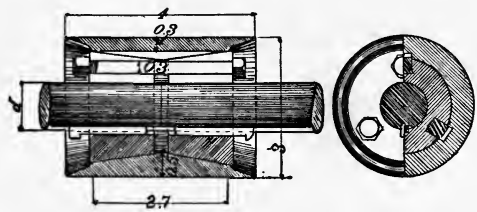

Fig. 59

SELLERS COUPLING FOR SHAFTING.

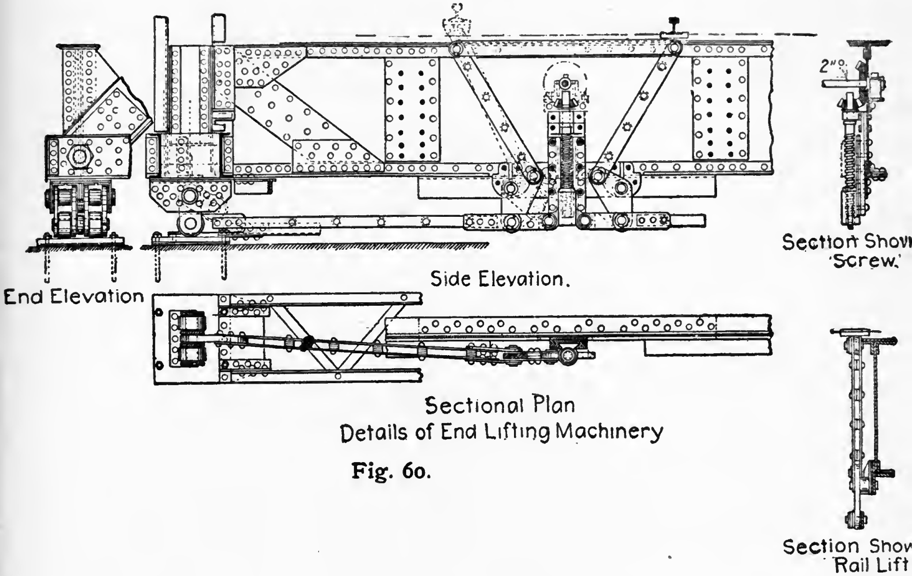
2"⌀
Section Show
Screw.
End Elevation
Side Elevation.
Sectional Plan
Details of End Lifting Machinery
Section Show
Rail Lift

Fig. 60.

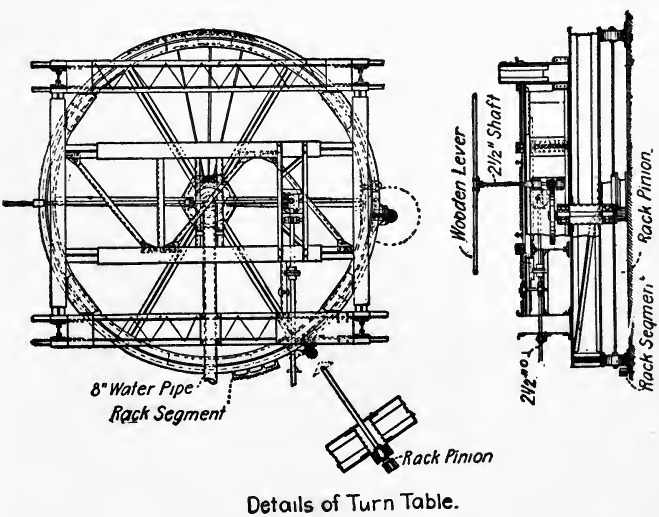
Wooden Lever
2½" Shaft
Rack Pinion.
Rack Segment
2½"⌀
8" Water Pipe
Rack Segment
Rack Pinion
Details of Turn Table.

Fig. 61.

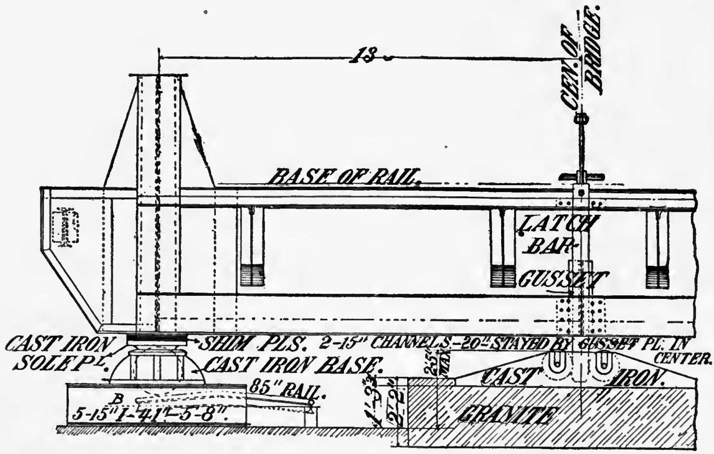

Fig. 62.

END SUPPORTS, LATCH, ETC.

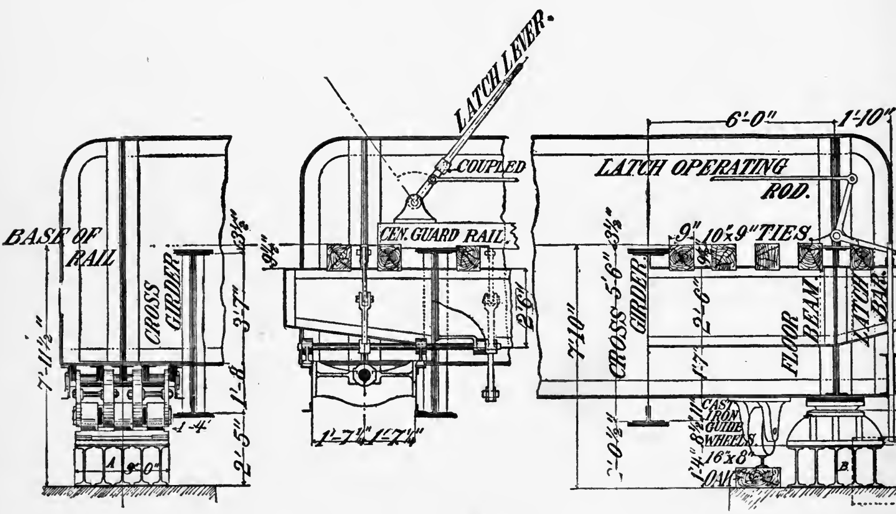

Fig. 63.

END LIFT, LATCH MACHINERY, ETC.

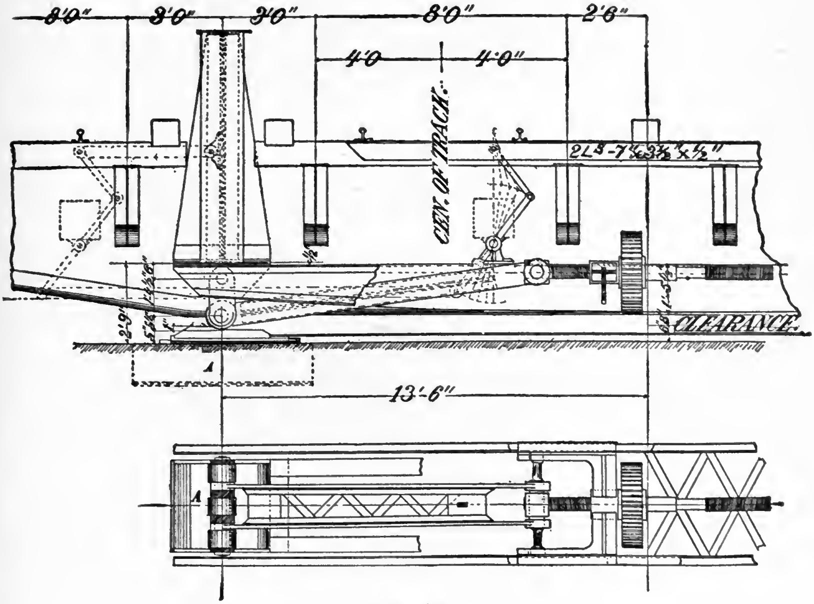

Fig. 64.

END LIFT.

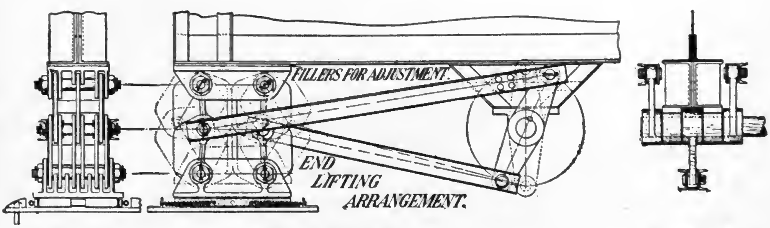

Fig. 65b. Fig. 65. Fig. 65a.

END LIFT.

When draw is closed and ends are raised the middle pins of toggle-joint stand ½" inside of vertical line through top and bottom pins to prevent the toggle from opening In above position castings bear against each other.

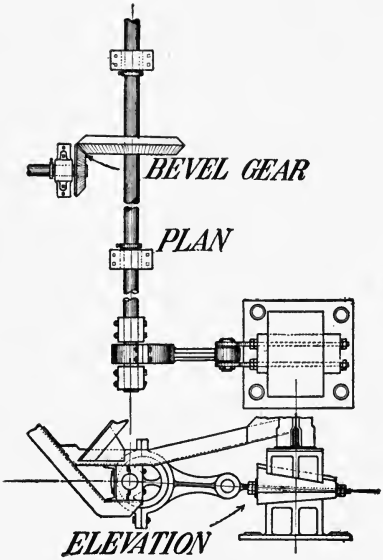

Fig. 66.

WEDGING GEAR.

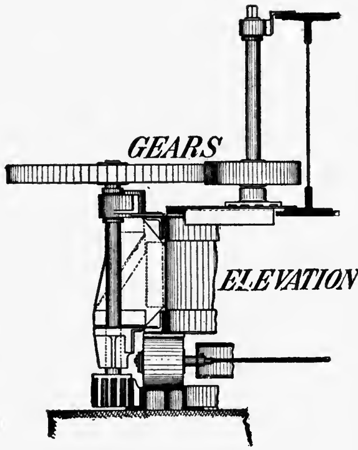

Fig. 67.

TURNING GEAR.

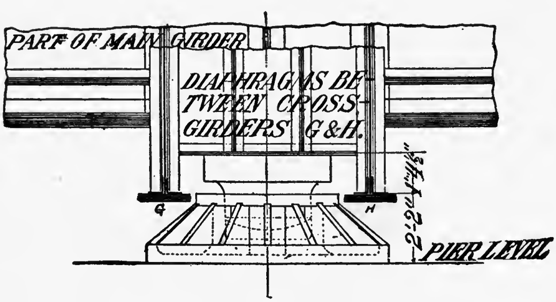

Fig. 68.

PIVOT CENTRE.

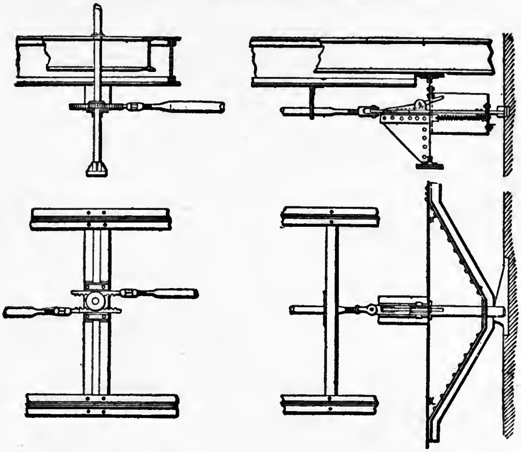

Fig. 69.

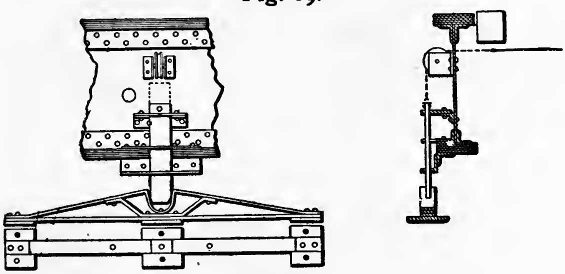

Fig. 70.
LATCHING DEVICES.

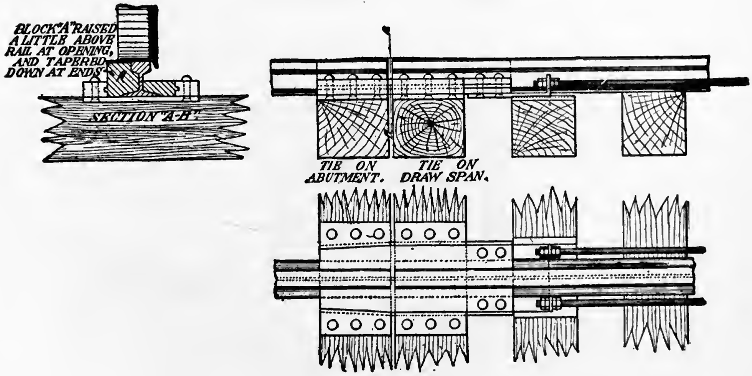

Fig. 71.
SLEEVE FOR CLAMPING RAILS.

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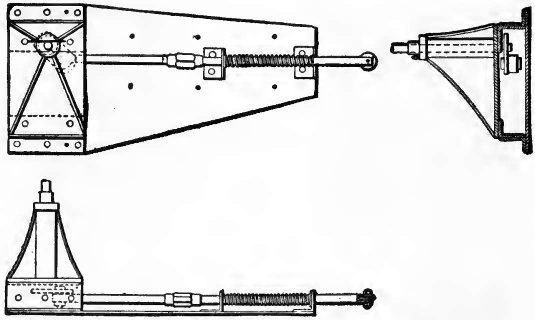


Fig. 72.

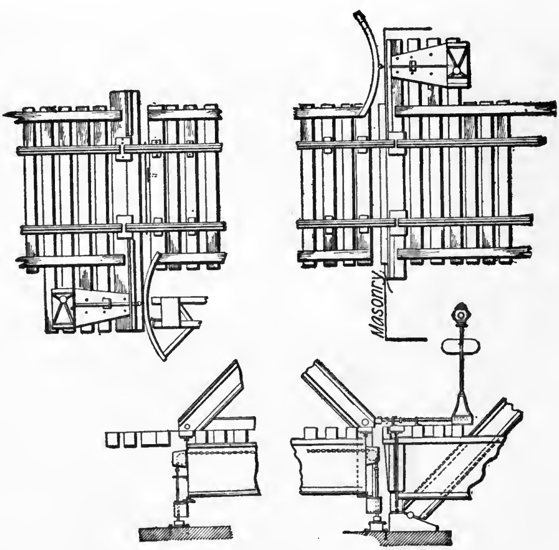
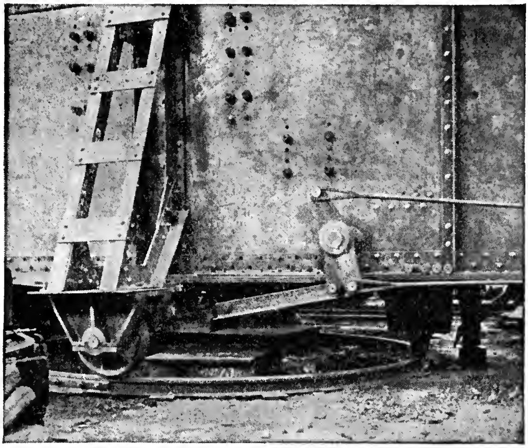


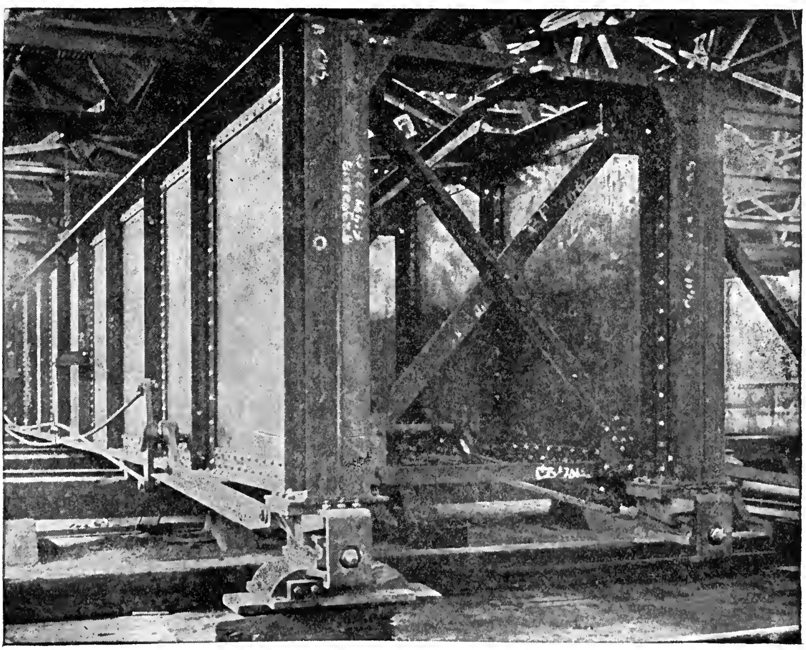
Fig. 73.

MACHINERY FOR OPERATING SAFETY-SIGNALS.

VIEWS SHOWING PLATE-GIRDER DRAW IN PROCESS OF CONSTRUCTION.

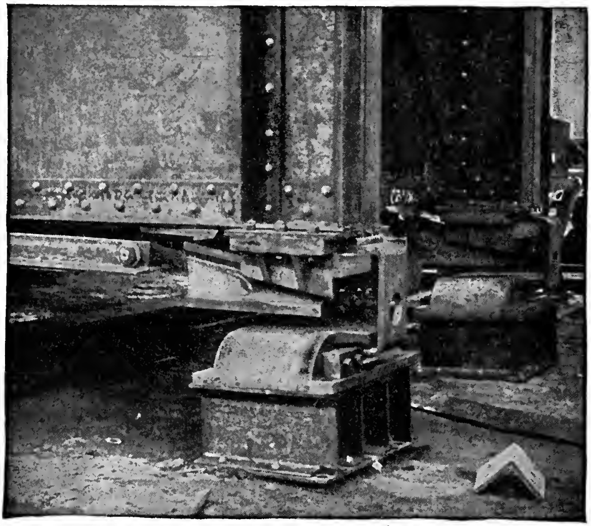


Balance-wheel and Centre Wedge.

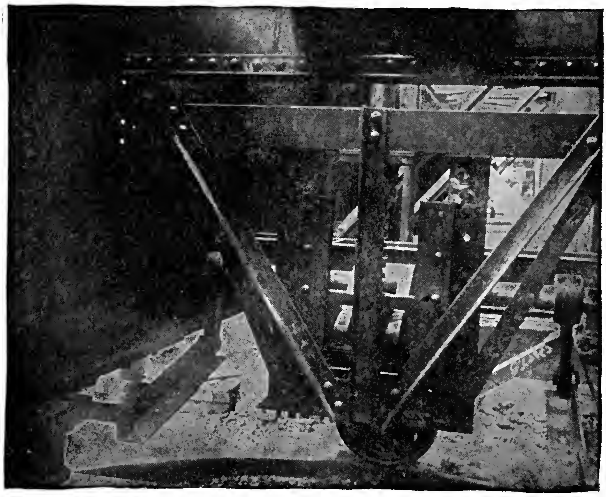


End Wedges and Portion of Machinery in Position.

VIEWS SHOWING PLATE-GIRDER DRAW IN PROCESS OF CONSTRUCTION.



End Wedging Arrangement.



Portion of Machinery at Centre.

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EXPLANATORY NOTES.

Where the term "moment of resistance" and the letter *R* designating the same have been employed in this work, they are used as indicating the moment of resistance for a fibre-stress of 1; or the term indicates the "section modulus" as given by some authors.

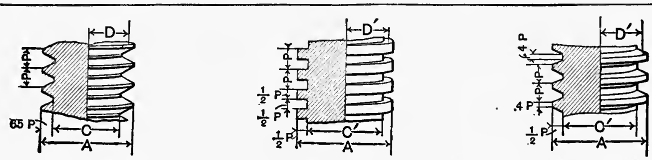
2. In Case 5, page 68, for continuous beams on three supports, note that the moments are obtained by scaling the ordinates between the curve and the inclined line, and not by scaling between the curve and the horizontal line as in the other cases.

3. On page 16 it will be noticed that the centre moments have been given for the loads on one arm only. The moments for the loads on the other arm are the same, and have been included in obtaining the total moment.

Friction of Worm-thread. (See page 38.)—The efficiency of the worm is very much reduced by the friction. In many cases a coefficient as high as 0.15 would be nearer correct than 0.10. The formula for the available vertical force is $W = \frac{r(F + F_1)}{\frac{P}{6.28} + cD}$, where W = vertical force, r = radius of turning lever, F = force at end of turning lever to overcome the vertical force W, F_1 = force at end of turning lever to overcome the friction produced by W, P = pitch of the worm-thread, D = the distance from centre of shaft to the centre of the worm-thread, c = the coefficient of friction. A force of 1 lb. at the end of a 6-ft. lever gives an available vertical force on the worm-nut, after deducting the friction of the thread and of the guides, as follows:

Diameter of Shaft.	Pitch.	Size of Thread.	*W*, in Pounds.
3½″	1½″	¾ in. sq.	161
3¼″	1¼″	⅝ "	182
3″	1″	½ "	207
2¾″	¾″	⅜ "	242

WORKING VALUES FOR WORM-SHAFTS.



V THREAD　　SQUARE THREAD　　EMERY THREAD

A	*B*	*P*	*C*	Area of *A*.	Area of *C*.	Safe Tensile Strain Iron at 10.000.	Safe Tensile Strain Steel at 12,500.	*D*	*D′*	*W*	*W′*
1½	6	.167	1.284	1.767	1.230	12,300	15.375	.696	.708	91.7	90.5
1⅝	5½	.182	1.389	2.073	1.496	14,960	18,700	.754	.767	84.5	83.2
1¾	5	.200	1 491	2.405	1.750	17,500	21,875	.810	.825	78.3	77.1
1⅞	5	.200	1.616	2.761	2.000	20,000	25,000	.873	.888	73.7	72.7
2	4½	222	1.712	3 141	2.300	23,000	28,750	.928	.944	68.8	67.8
2¼	4½	.222	1.962	3.976	2 990	29,900	37,375	1.053	1.070	62.1	61.3
2½	4	.250	2.176	4.908	3 640	36,400	45,500	1.169	1.188	55.8	55.0
2¾	4	.250	2.426	5.939	4.806	48,060	60,070	1.294	1.313	51.3	50.7
3	3½	.286	2 629	7 068	5.411	54,110	67.638	1.407	1.420	46.8	46.2
3¼	3½	.286	2.879	8.295	6.491	64.910	81,137	1.530	1.554	43 6	42.6
3½	3¼	.308	3.100	9.621	7.080	70,800	88.500	1.650	1.673	40 5	40.0
3¾	3	.333	3.317	11.044	8.395	83,950	104,930	1.767	1.792	37.7	37.3
4	3	.333	3.567	12.566	9.970	99,700	124.620	1.892	1.917	35.6	35.2
4¼	2⅞	348	3 798	14.186	11.144	111,440	139,300	2.012	2.038	33.5	33.2
4½	2¾	364	4.028	15.904	12.567	125,670	157,080	2.132	2.159	31 8	31.4

Number of threads per inch on above bolts is the number given in the Sellers System.

A = external diameter; *B* = number of threads per inch; *C* = diameter at root of thread; *D*, *D′* = radius of centre of thread; *W* (for V thread), *W′* (for square thread) = the weight which can be raised by a force of 1 lb. with a leverage of 1 foot. Coefficient of friction = .15.

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