MOUNT

Design of a
Jib Crane for a Foundry

Mechanical Engineering
B. S.

1902

204
211

DESIGN OF A JIB CRANE FOR A FOUNDRY

BY

MADISON HOGE MOUNT

THESIS FOR DEGREE OF BACHELOR OF SCIENCE

IN MECHANICAL ENGINEERING

COLLEGE OF ENGINEERING

UNIVERSITY OF ILLINOIS

PRESENTED JUNE 1902

190

UNIVERSITY OF ILLINOIS

May 29, 1902 190

THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Madison Hoge Mount

ENTITLED Design of Jib Crane for a Foundry

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE

OF Bachelor of Science in Mechanical Engineering

L. P. Breckenridge

HEAD OF DEPARTMENT OF Mechanical Engineering

1902
M84

DESIGN FOR A THREE TON JIB CRANE FOR THE FOUNDRY OF THE UNIVERSITY OF ILLINOIS.

Due to the increased railroad industry,shipping, and manufacture of iron and steel in the past few years, the demand for cranes of various designs has become so great that the attention of several large concerns is now devoted solely to the manufacture of hoisting machinery.

The books desriptive of cranes are few and by reason of improvements which have been effected they do not contain information concerning the most recent practice. The best information on modern practice which I was able to obtain was from various catalogues published by American manufacturers of cranes and hoisting machinery as a means of advertising their products, also from articles in the recent technical periodicals.

TYPE OF CRANE.

In deciding upon the type of crane required, the nature of its duties must of course receive the first consideration, also the character of the power to use. The crane in consideration is to be placed in the foundry of the University of Illinois. The foundry room is not large,the maximum load that the crane will ever be required to lift will not be more than five or six thousand pounds and usually much less. The work is done by students,usually of large classes, thus help is abundant,therefore hand power is the most economical and best adapted to the conditions. Another equally important feature considered in this design is neatness and general appearance.

8

Plate II in the complete assembled drawing showing the

final result after carefully working out all the details required to meet as completely as possible all the requirements.

GENERAL DESCRIPTION OF CRANE.

Each member of the frame consists of two parts, separated 12" so as to permit the chain and block to pass between them, so that the load can be moved close in to the mast. The hoisting mechanism is attached to the mast near its foot, and the running block, which carries the load, is suspended from a trolley travelling on the jib and capable of movement in and out by means of independent gearing attached to the jib at its intersection with the mast.

The frame consists of steel channel beams, each of the members of the frame being composed of two such channels. The dimensions are such as to give a high factor of safety, and the several parts are very securely connected at their intersections by riveting.

Hoisting is effected through a train of spur gearing, operated by two cranks. It is provided with an automatic rachet and a safety friction clutch. Thus arranged the machine is self sustaining and can be left at any time with the load in suspension without danger of the load running down or handles flying back. The hoist is provided with two changes of speed, the change from fast to slow speed or vice versa being controlled by simply shifting the crank shaft endwise.

The load is lowered by simply turning the handles backward which releases the friction discs and as soon as the backward motion of the handles stops the discs again become engaged and the load is held.

Rotation is easily effected by pulling or pushing the suspended load. Motion of the trolly on the jib, in either direction, is effected by gearing operated from below, and by an endless hand

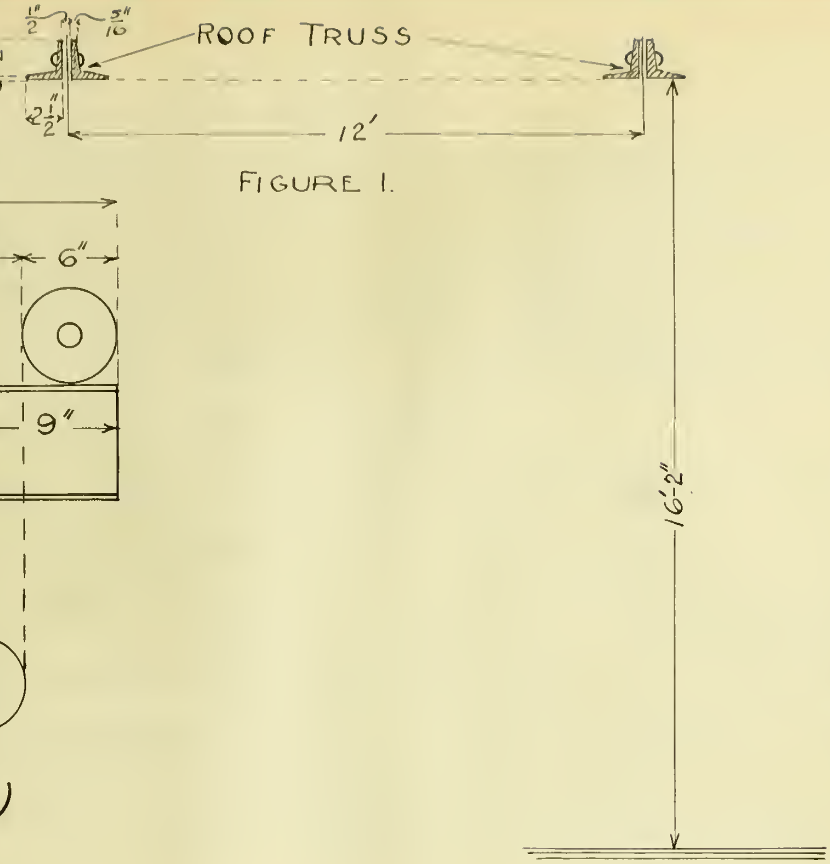

FIGURE 1.

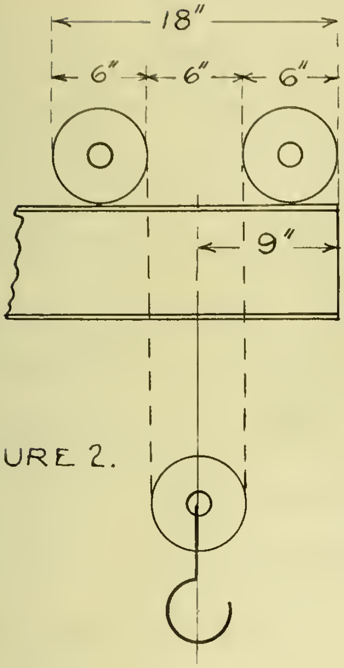

FIGURE 2.

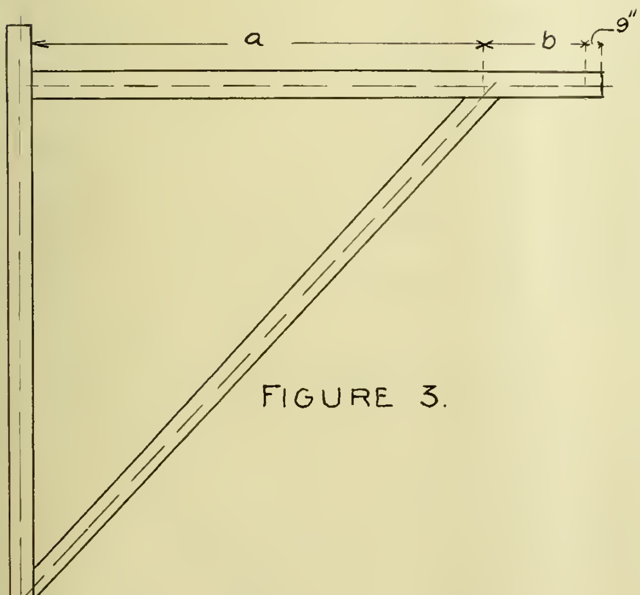

FIGURE 3.

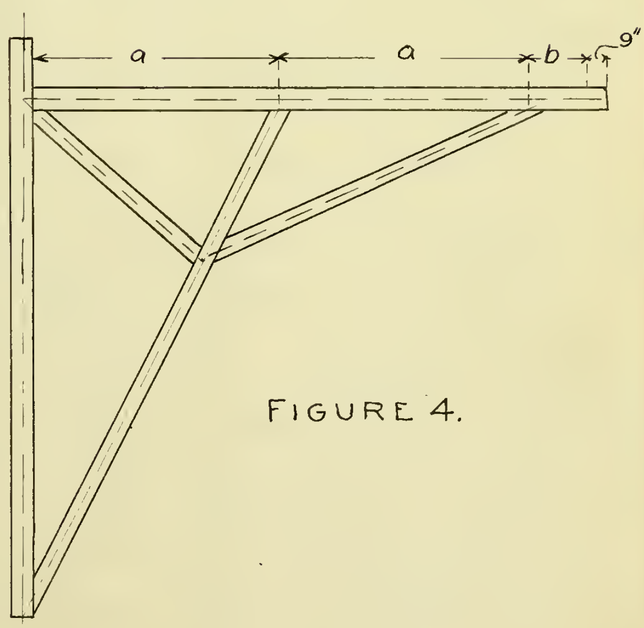

FIGURE 4.

chain.

THE DESIGN.

Before beginning the design, I examined the foundry in which the crane was to be located and took such measurements as were essential in determining the principle dimensions. The crane is to be placed near the north wall so as to occurp as little room as possible and not to be in the way.

The lower members of the roof trusses are 16'-2" above the floor of the foundry since the jib should be 16' or 18' in length I decided that the jib should be supported by inclined struts, and that the upper bearing should be supported by the roof trusses while the lower bearing rests on a solid foundation. Figure I is a sketch showing the location and proportions of the roof trusses.

As the hoisting chain will be supported by a trolley containing four rollers, this will prevent the load from coming entirely to the end of the jib, and the weight of the trolley, hoisting chain, block and hook will add to the weight carried by the jib. The the maximum weight to be lifted is three tons, the maximum load on the crane is three tons plus weight of trolley, plus weight of hoisting chain, plus weight of hook and block. Considering the trolley rollers to be six inches in diameter and 12" from centers of rollers, 1/2" hoisting chain and chain sheaves 6" in diameter, a rough estimate of the weight is about 300 pounds. Thus, the maximum load on the crane is 6300 pounds. Then the load cannot approach closer to the end of the jib than 9". Figure 2.

THE FRAME.

After making several investigations on the positions and number of supports for the jib, I decided that the neatest arrange-

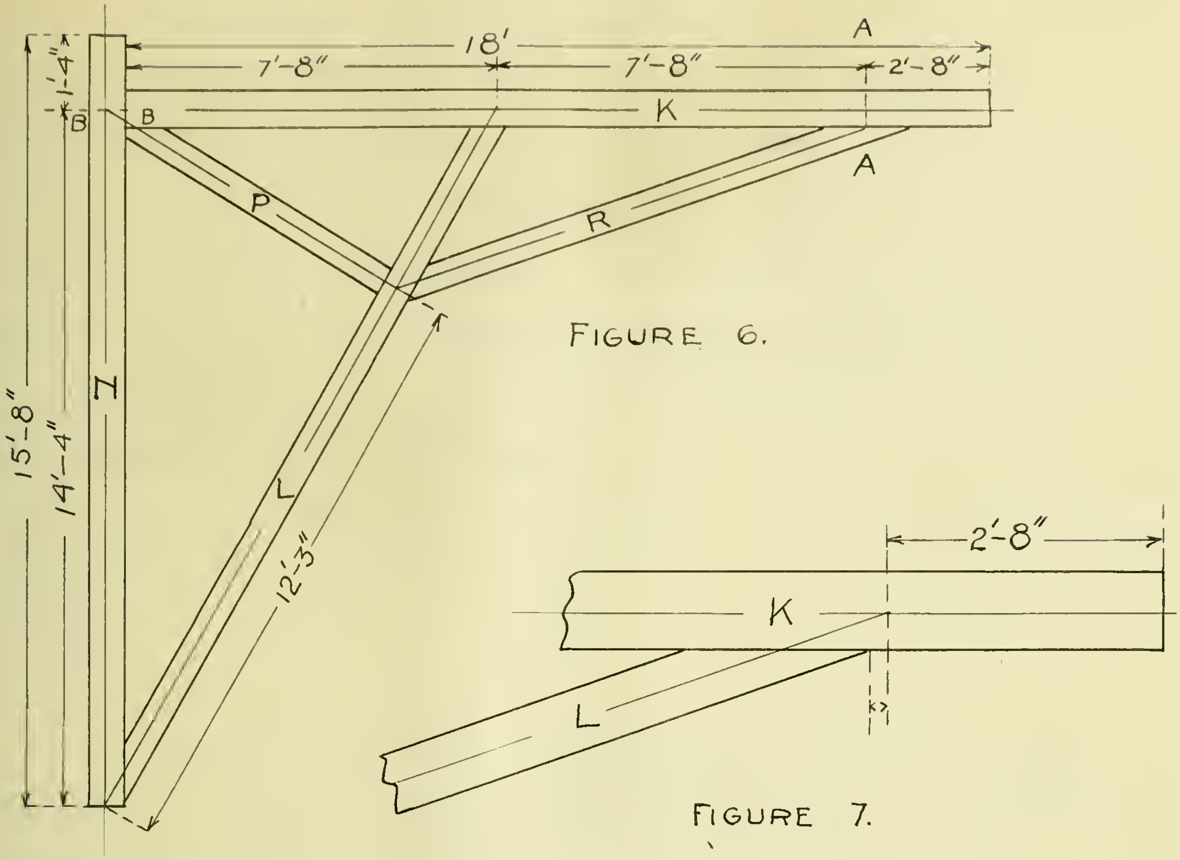

FIGURE 6.

FIGURE 7.

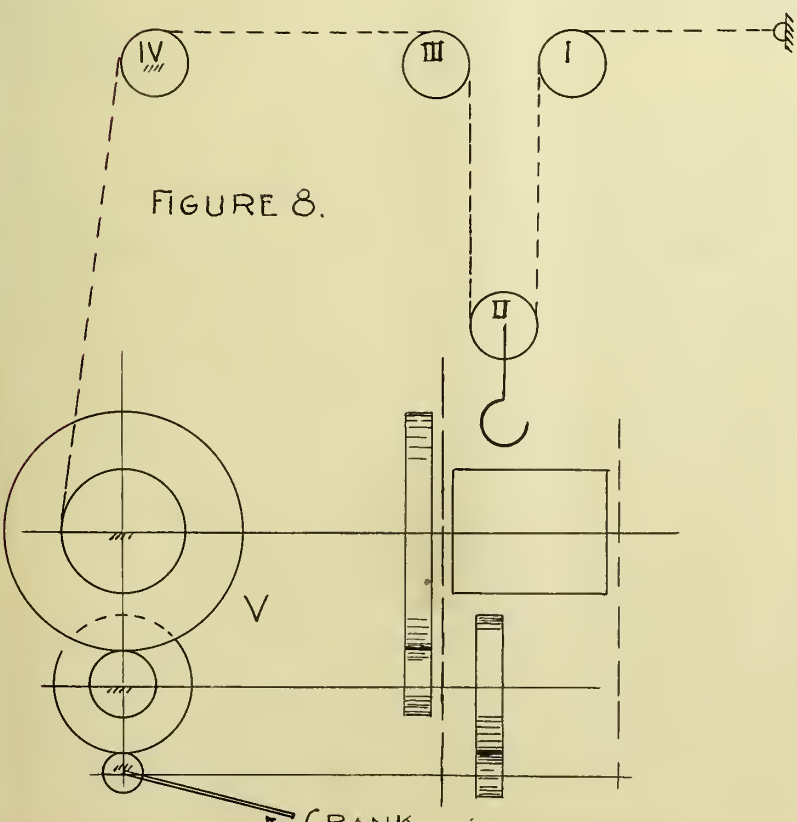

FIGURE 8.

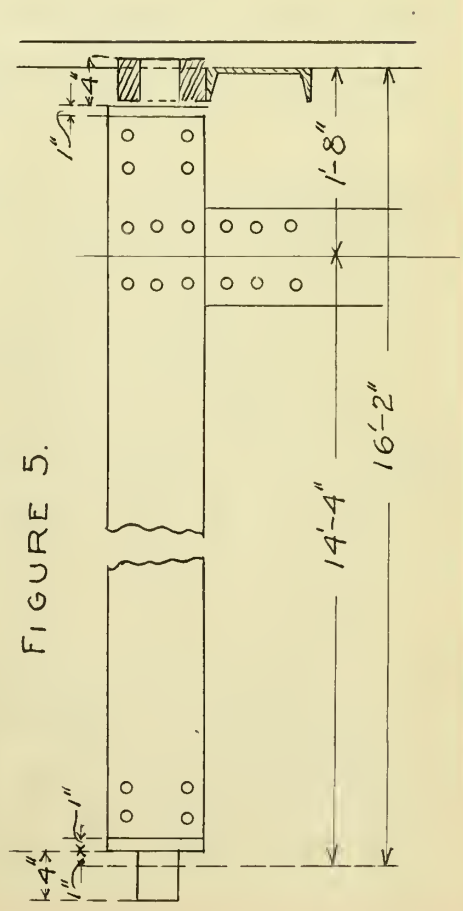

FIGURE 5.

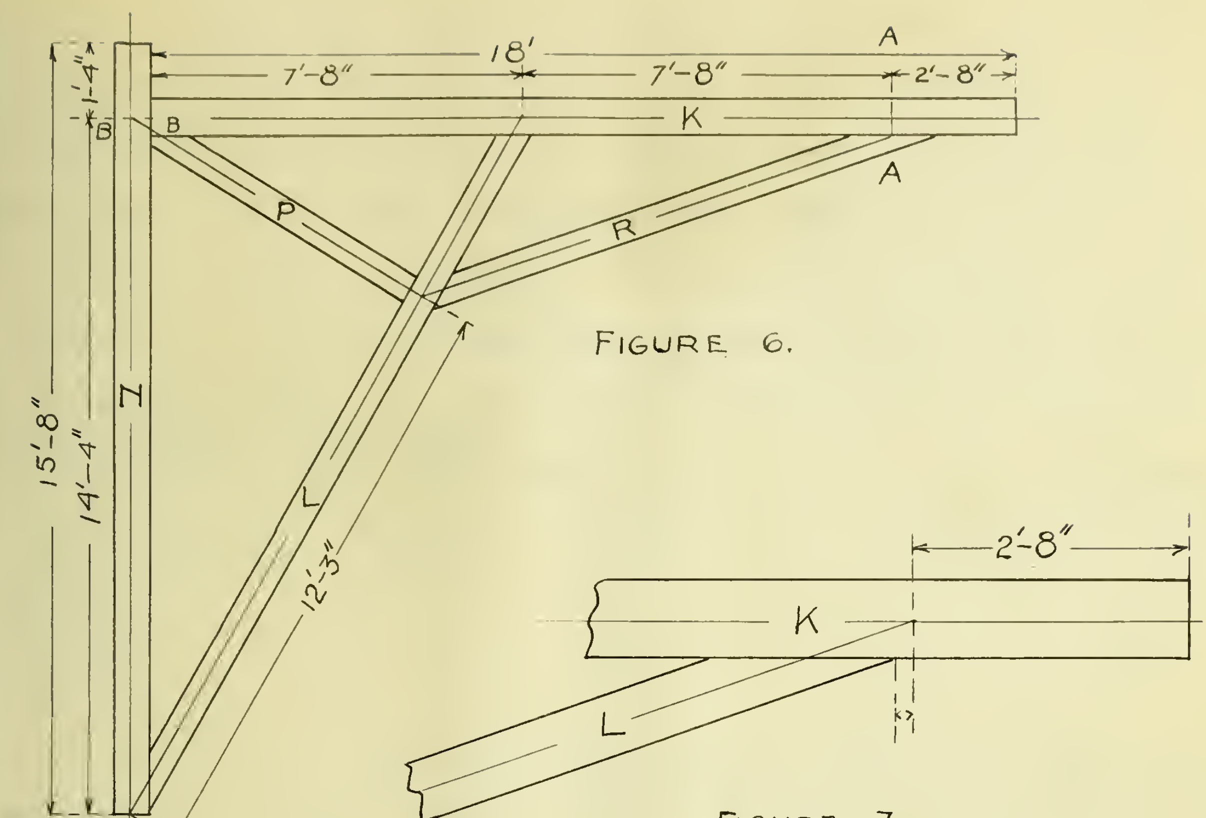

FIGURE 6.

FIGURE 7.

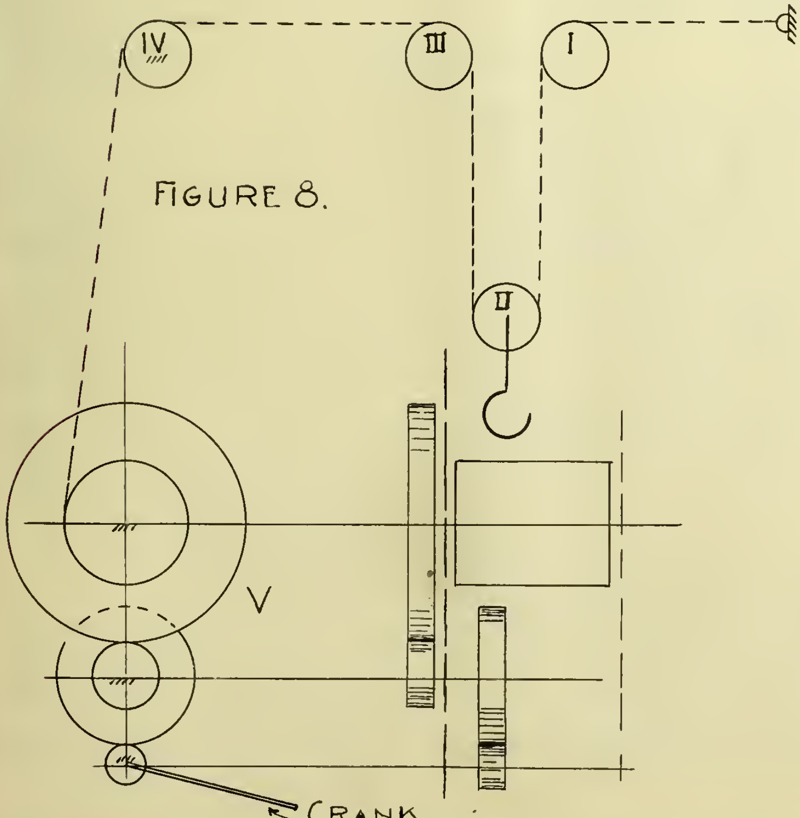

FIGURE 8.

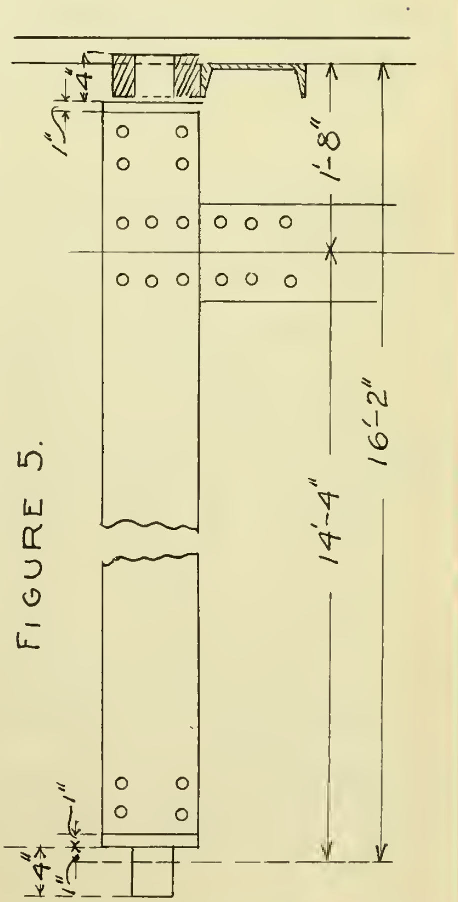

FIGURE 5.

ment would be to support the jib in three places. One end supported at the mast and the other two supports at the most advantageous points, which are found, as follows. In case one strut is used, see Figure 3, the most advantageous point to place the strut would be to make b= 1/5 a; but an arrangement like this requires [illegible] channels in the jib than look well, and a better arrangement is shown in Figure 4. This requires equal stresses in the jib at the dangerous points. Thus a = a and. a + a = 8b or b = 1/9 of(18' - 9".).

$b = 1/9 \times (18' - 9") = \frac{207"}{9} = 23"$

$a + a = 8 \times 23" = 184"$.

Therefore $a = 92"$

$b = 23"$

In order to give ample room for all parts above the jib, I decided to place the center line of the jib 20 " below the lower edge of roof trusses. Then I made a sketch showing the approximate positions of pivots and cross piece to support the upper bearing of the crane. See Figure 5.

I had now determined the linear dimensions of the mast and jib. I next laid out a drawing similar to Figure 6 and made some graphical tests so as to determine the positions of the member R on L, so as to approach a uniform stress in the two members. After several trials I decided to place them as shown in the Figure, the center line of R intersects the center line of L at a point 12' 3" from the lower end.

I now made a complete graphical solution of all the stresses in the crane with the maximum load at the greatest distance from the mast. I considered each joint as being held by one pin and the line of action of the forces acting along the center line of

the pieces composing the members. See Plate I. By placing R so that the center line passes through A...A, the point where the mast should be supported, the points A...A did not come over the support, as R was so oblique. See Figure 7. This caused the moment arm of the end of the jib to become greater than was first intended so I changed the position of R so that the end of the center line came directly under the point A...A. as shown in Figure 6.

The lightest weight channels were chosen rather than smaller and heavier ones, as the broad channels make a better appearance than smaller ones of equal strength. The lengths of the members were figured accurately from the intersection of the center lines except at the joint between K and R which was calculated from the intersection of the center line R. and the lower edge of K. for reasons given before. I now determined the real lengths that the braces should be cut and the shape of the ends by the following method. I laid off very carefully the entire plan of the center lines of the crane to a reduced scale, seeing that all dimensions checked with the calculations. These lines at the intersections gave the relative positions of the members. Then I laid off carefully to full scale the exact widths of the channels to be used on each of their center lines. The intersections of the edges of the channels laid off gave the exact form of the joint to full scale. From this figure I measured carefully the form of the ends of the members also the amount to be taken from the calculated values to get the lengths which the pieces should be cut. I now designed each joint to full scale in order to more accurately obtain a strong and good looking joint.

References to articles and books on cranes.

Die Hebezenge von Ad Ernst.

Towne on Cranes.

The Design of Cranes and Crane Details.
M.W. Jan.3, Jan. 11, Feb. 14, March 21, 1902.

Twenty Ton Wreck Crane, R.R. Gaz. Nov.10, 1893, p. 813.

Compressed Air Shop. " " " Feb.10, 1893, p. 106.

Trans A.S.C.E. XV p 369.

Eng. News Jan. 15, 1887.

" " Aug. 22, 1891.

" " Nov. 15, 1890.

" " Apr. 5, 1890.

" " Nov. 5, 1881.

R.R.Gaz. Aug. 14, 1891.

Lon. Eng. Jan. 13, 1888.

R.R. Gaz. July 10, 1891.

Eng. News Feb. 7, 1891.

R.R.Gaz. July 31, 1891.

HOISTING GEARING.

Efficiency.

Before designing the hoisting machinery it is essesntial to know something about whatefficiency the apparatus will have. In order to do this something must be decided upon as to the size and arrangement of the parts. Let Figure 8, represent roughly the arrangement of the mechanism. I, II, III, ans IV are chain sheaves and V is the hoisting windless. When the load is raised the chain passes over II, III, IV and is wound on the drum, consequently the movement of these parts causes friction and absorbs some of

the work applied at the crank.

From Professor Goodenough's notes on Mechanics of Machinery page 25 and 48, I obtained the following efficiencies.

Efficiency of fixed chain pulley.

Arc of contact 180° efficiency .958.

" " " 90° " .964.

Efficiency of hoisting drum and winches from .90 to .97. From this I considered that the different parts of the hoisting apparatus would have efficiencies as follows:-

Sheave II an efficiency of .95.

" III " " " .96.

" IV " " " .96.

Hoisting windless an efficiency of .90.

Then the entire efficiency of the apparatus would be $.95 \times .96 \times .96 \times .90 = .788$. For use in the design. I considered the efficiency to be 75 per cent.

The maximum load to be lifted is 6000 pounds.this gives a load of 3000 poinds acting on each side of the chain supporting the load. Then the power applied at the cranks must be sufficient to pull 3000 pounds on the hoisting chain and to over come the friction, or the same as to pull $3000 \div .75 = 4000$ pounds.

The following notes regarding hoisting drums are taken from Professor Goodenough's "Notes on Mechanics of Machinery" and originally from "Des Ingenieures Taschenbuch."

"HOISTING BY HAND POWER".

"Number of workmen 1,2,or 4; force applied by each workman, 15 kg (33 lbs.) for an eight hour working period. Crank velocity 0.8 to 1.0 metres (2 1/4 to 3 1/4 ft.) per second. Crank arm 35 to 40 cm. (14 to 16 inches) long;the crank shaft should be about

3 1/4 from the floor. Pairs o gears in reducing train 2, at most 3; if greater reduction is needed a block and tackle is interposed between drum and load . Greatest single reduction 1:8. The reduction bewteen crank and drum is to be so designed as to give gears of maximum diameters. The gear train should be arranged so that one or more pairs may be thrown out in order that light loads may be lifted at higher speed."

"The smallest number of teeth in gear or pinion = 10; the pitch is determined from the pressure between teeth; breadth of tooth = pitch X 2. The efficiency of each pair of train is about 0.92, taking account of tooth and journal friction. The speed of the load may be 0.10 to 0.35 metres (4 to 14 inches) per second."

From the above the following conclusions are drawn: In this case one man will not continue to work at the handle very long, then it is safe to consider that one man can exert a pressure of 35 pounds on the crank handles. Let the greatest number of workmen = 2, and let there be a crank for each workman. Let the crank radius = 16 inches and let the revolutions per minute of the crank = 20. Then the velocity of the crank handles will be

$$\frac{3.1416 \times 2 \times 16}{60 \times 12} \times 20 = 2.79 \text{ ft. per second.}$$

The power exerted per minute on the cranks is

$$\frac{2 \times 16 \times 3.1416 \times 2 \times 35 \times 20}{12} = 11732 \text{ ft.lbs.}$$

The maximum resistance due to load and friction is equal to a pull of 4000 lbs. on the circumference of the hoisting drum. Then to wind in one foot of chain on the hoisting drum per minute requires 4000 ft. lbs work, but 11732 ft lbs., of work per minute can be done at the cranks. Then the number of feet of chain that can be wound on the drum per minute = $\frac{11732}{4000}$ = 2.933 feet.

Let the circumference of the drum equal 4 feet and the diameter is $\frac{4 \times 12}{3.1416} = 15.36''$. The revolutions per minute of the drum = $\frac{2.933}{4} = 0.73325$. The revolutions per minute of the crank = 20. Then the velocity ratio between the crank and drum = $\frac{20}{0.73325} = 27.2$. With this velocity it will require two pairs of gear wheels.

Referring to Plate IV, the wheel A1 turns with the drum, wheels A2 and A3 turn together and pinion A4 turns with the cranks.

Let X_1 = R.P.M. of wheel A1 and drum.

" X_2 = " " " A2 and A3

" X_3 = " " " A4 and cranks

Let a = number of teeth in A1

" b = " " " " A2

" c = " " " " A3

" d = " " " " A4

Then $\frac{X_2}{X_1} = \frac{a}{b}$ and $\frac{X_3}{X_2} = \frac{c}{d}$

and $\frac{X_2}{X_1} \times \frac{X_3}{X_2} = \frac{a}{b} \times \frac{c}{d}$.

Therefore $\frac{X_3}{X_1} = \frac{a}{b} \times \frac{c}{d} = \frac{20}{0.73325} = 27.2$

Make the reduction between A1 ans A2 = 5 or $\frac{a}{b} = 5$

Then $\frac{X_3}{X_1} = 5 \times \frac{c}{d} = 27.2$

$\frac{c}{d} = \frac{27.2}{5} = 5.45$

Therefore $\frac{X_3}{X_1} = 5 \times 5.45$

Let b = 15, and d = 12

Then $\frac{a}{b} = \frac{a}{15} = 5$

$a = 15 \times 5 = 75$ and

$$\frac{c}{d} = \frac{c}{12} = 5.45$$

$$c = 12 \times 5.45 = 66.$$

Thus the number of teeth in each wheel is determined.

a = 75 number of teeth for wheel A1.

b = 15 " " " " " A2.

c = 66 " " " " " A3

d = 12 " " " " " A4.

By choosing the proper pitch so as to give a face of the tooth 2 to 3 times the circular pitch and of sufficient strength, I obtained the following dimensions for the wheels.

Wheel	Circular pitch inches	Face inches.	Diameter inches.
A1	1.25	3	29.84
A2	1.25	3	5.971
A3	.75	2	15.756
A4	.75	2	2.865

Several trials were made before these results were obtained. Some difficulty was experienced in obtaining the proper sized wheels, so that A3 would not come in contact with the drum.

In order that there may be a change in the speed of hoisting, a second pair of wheels must be thrown in the place of one pair of the wheels. Let this pair of wheels be thrown in, in place of wheels A3 and A4. This pair of wheels must work on the same shafts, with A3 and A4, therefore, they must have the same distances between centers.

Let this extra pair of gears be known as A5 and A6. The wheel A6 is located on the same shaft with A4, and A5 is placed on the same shaft with A3. The distance between the centers

of A3 and A4 = 9.31". The velocity ratio of A1 and A4 = $\frac{X_3}{X_1}$ = 27.2 and when A5 and A6 are thrown in the velocity ratio should be about one half of 27.2 or 13.6.

Let X_5 and X_6 = respectively the R.P.M. of A5 and A6.

Then $\frac{X_6}{X_1} = 13.6$

and $\frac{X_6}{X_5} \times \frac{X_2}{X_1} = 13.6$

but -- $\frac{X_2}{X_1} = 5$ and $\frac{X_6}{X_5} \times 5 = 13.6$

Therefore $\frac{X_6}{X_5} = \frac{13.6}{5} = 2.7$ or the reduction between X_5 and X_6 is 2.7.

Let c_5 = number of teeth of A5

" d_6 = " " " " A6

Then $\frac{c_5}{d_6} = 2.7$

Let r_5 = radius of A5

" r_6 = " " A6

Then $\frac{r_5}{r_6} = \frac{c_5}{d_6} = 2.7$

$r_6 + r_5 = 9.31"$ (1)

$\frac{r_5}{r_6} = 2.7$

$r_5 = r_6 \times 2.7$

$r_5 - 2.7\, r_6 = 0$ (2)

Subtract (2) from (1)

$3.7 r_6 = 9.31"$

$r_6 = 2.52"$

Then $r_5 = 9.31" - 2.52" = 6.79"$

Therefore

The diameter of A5 = 6.79" x 2 = 13.58"

The diameter of A6 = 2.52" x 2 = 5.04"

But we have not yet determined the pitch of these wheels which is liable to change the diameter slightly. Give each wheel a circular pitch of 3/4 inches; multiply the diameter found by 4/3 to change them to the diameters of wheels of 1 inch circular pitch having the same number of teeth. On page 889 in Kent is a table giving the diameters of spur wheels from 10 to 100 teeth of 1 inch circular pitch. Find the pair of wheels that gives the closest values to the diameters found and change them back to gear wheels having 3/4 inch circular pitch by multiplying by 3/4.

The following is the calculation according to the above method.

13.58" x 4/3 = 18.104 = dia. of A5 if circular pitch was 1 inch.
5.04 x 4/3 = 6.72 = " " A6 " " " " "

A wheel of 57 teeth 1 inch circular pitch has a diameter of 18.144"
" " " 21 " " " " " " " " " 6.685"

18.144" x 3/4 = 13.608"
6.685" x 3/4 = 5.013"

Then the diameter of the pitch circules of gear wheels A5 and A6 equals 13.608" and 5.013" respectively.

The radius of A5 = 6.804
" " " A6 = 2.506.

Then the distance between centers of A5 and A6 = 6.809 + 2.5165 = 9.32".

The following is a table of dimensions of all the gear wheels in the hoisting windless.

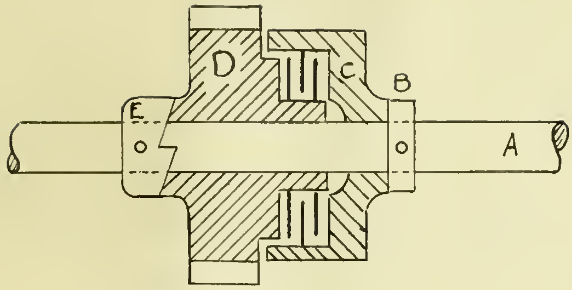

FIGURE 9.

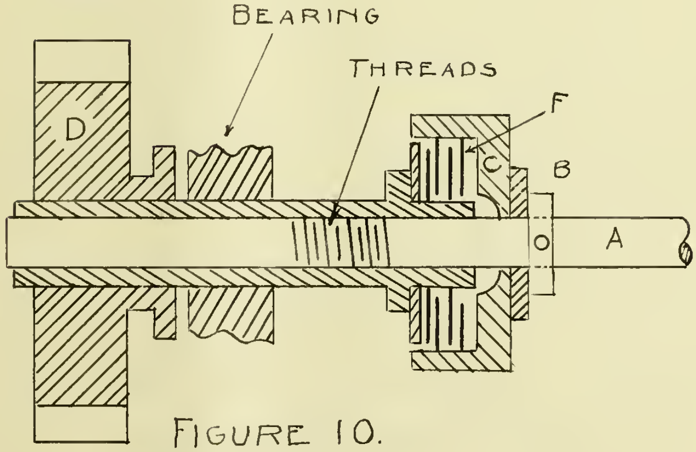

FIGURE 10.

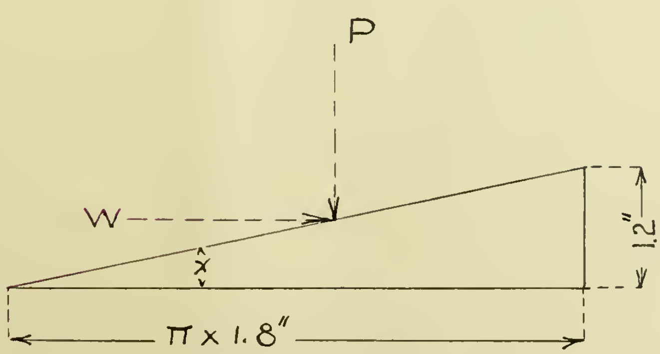

FIGURE 11.

Gear wheel	No. of teeth	Diameter of pitch circle inches.	Face inches.	Circular pitch inches.	Diametial pitch.
A1[illegible]	75	29.84	3	1 1/4	2.5133
A3	66	15.756	3	1 1/4	2.5133
A2	15	5.971	2	3/4	4.1888
A4	12	2.865	2	3/4	4.1888
A5	57	13.608	1 3/4	3/4	4.1888
A6	21	5.013	1 3/4	3/4	4.1888

THE FRICTIONAL SAFETY RATCHET.

Reference, Towne on Cranes.

" Although somewhat complicated to describe, the action of this machine is exceedingly simple and absolutely reliable under all conditions." The safety frictional ratchet I have designed is a madification of the form here described but the action is the same and the description will serve to establish the principles involved.

In Figure 9,"A is a primary shaft of a hoist,D a spur pinion carried by said shaft and gearing into a proper spur wheel upon the second shaft. C is a ratchet wheel having teeth on its periphery which engage with an ordinary pawl pivoted to the frame of the machine thereby preventing its rotation except in one direction. E is a collar screwed and pinned fast to the shaft A so that it rotates with it, and B a similar collar at the opposite side. The collar E has a helix formed upon its side which adjoins the pinion D, and the hub of the latter has a corresponding helix, so that the two when in coincidence appear as shown in the figure. F consists of a set of brass friction discs. the discs being alternately attached to the spur pinion and the ratchet wheel.

The pinion D and the ratchet C are loose upon the shaft A. Assuming that to effect hoisting the shaft A must be revolved so that its upper surface moves toward the eye, in which case the resistance due to the load tends to retard or hold backthe pinion D, the shoulder upon the collar E, as the latter revolves with the shaftA, will move away from the corresponding shoulder upon the hub of the pinion D, the effect of which is to cause the two helics to to mount upon each other thereby pushing the pinion D to the right upon the shaft and forcing it into frictional engagement with the ratchet wheel C. The latter being supported by the fixed collar B, the several parts are thus locked together and will thereupon rotate simultaneously, the teeth of the ratchet C being inclined so as to permit of motion in this direction. If at any time the rotation of the shaft A be discontinued the pressure due to the load will tend to rotate the pinion D backwards, but all the parts being locked together, as already explained, backward motion is prevented by the action of the ratchet wheel C and the load will thus remain suspended."

"When it is desired to effect lowering, the shaft A must be positively rotated backwards, the effect of which will be to relax the longitudinal engagement of the several parts by the rotation of the collar E, the helix upon which will thus move forward into coincidence with the corresponding helix on the hub of the pinion D. As soon as this movement of the collar E is sufficient to relax the longitudinal pressure, the pull of the load will cause the pinion D to follow the rotation of the shaft and to over take the collar E, thereby again applying the longitudinal pressure unless the continued backward motion of the shaft again releases it. This alternate releasing and re-engagement

will then continue so long as the shaft is revolved backward.

The safety friction ratchet as shown in Figure 9, is not well adapted in this form to suit the conditions required in the present design. The best place to put the device is on the shaft with the pinion A2 as previously mentioned. This pinion should be on the outside of the bracket supporting the shaft, but if both the ratchet and pinion were set outside it would not look well, and there is not room enough for both of them between the brackets. This can better be seen in Plate IV. In order to make this arrangement it is necessary to connect the pinion and friction discs by means of a sleeve which fits over the main shaft and passes through the bearing of the bracket supporting the shaft. Instead of a spiral collar, I have threaded the shaft and sleefe with square threads. This method gives a positive release of the discs as well as a positive action in the opposite direction. thus for backward motion the lowering of the load is sure to be steady as the discs can in no way fail to open. In order to prevent too great a thrust on the sleeve or tension in the shaft for maximum it is necessary to give the threads tripple pitch. See Figure 10.

The following are the mathematical principlaes and calcultions of the safety frictional ratchet. When the shaft is rotated so that upper surface of pinion is moving from the eye, Figure 10, the threads push the sleeve toward the right, thus the discs become engaged, and the pinion rotates in the same way as in the previous example. The friction between the discs is caused by their being pressed together or the thrust of the sleeve. The thrust of the sleeve is proportional to the load on the teeth of the pinion and pitch of the threads. The maximum load on the teeth of the pinion

is about 2000 lbs. Diameter of pitch circle of pinion 5.96".
Diameter of shaft = 2 in, depth of threads = 0.2 in. Then mean diameter of thread = 1.8 in.

Then the maximum tangential load on the threads =

$$= \frac{2000 \times 2.98}{.9} = 6622 \text{ lbs.}$$

Length of mean circumference of thread = 3.1416 x 1.8 in.
In one turn of the shaft a point on the shaft will tend to advance 1.2 in. as the thickness of one thread is 0.2 in. See Figure 11, Let P = the thrust on the sleeve.

Then $P = W \cot X$

$$\cot X = \frac{3.1416 \times 1.8}{1.2}$$

$$\text{Therefore} \quad P = \frac{6622 \times 3.1416 \times 1.8}{1.2} = 31204 \text{ lbs.}$$

This gives the maximum pressure that will ever come on the friction discs. But in this value for P loss due to friction in the screw has not been considered. With poor lubrication the efficiency may run as low as 60 per cent, or the maximum pressure on friction discs would be about 18700 lbs.

These discs act the same as pivots and the friction between them can be figured by the same method. For friction of pivots see Professor Goodenough's Notes on Mechanics of Machinery, page 11.

$$M = uP \frac{r_2 + r_1}{2}$$

$$\frac{r_2 + r_1}{2} = \text{mean radius of disc.}$$

$$P = 18700 \text{ lbs.}$$

Let μ = .05 for coefficient of friction of brass on brass.

M = the moment of resistance of the friction discs at the mean radius of two rubbing surfaces in contact.

Let $\frac{r_2 - r_1}{2} = 2\ 1/2"$

Then $M = .05 \times 18700 \times 2.5 = 2337.5$.

The moment of the force acting at the pitch circle whcih M resists = 2.98 x 2000 = 5960 lbs.

Then the number of rubbing surfaces required to resist this moment $= \frac{5960}{2337} = 2 +$ say 3.

But for greater safety, I made 5 rubbing surfaces, this requires 3 pans of discs alternately attached to the sleeve and ratchet. See Plate.VII.

THE TROLLEY.

The trolley travells on the jib see Plate II, which shows its construction. In oder to design the mechanism to move the trolley when loaded it is necessary to know about what force will be required to do the work. The load is neither raised nor lowered by the movement of the trolley, therefore the entire resistance due to its movement is due to the frictional resistance in the moving parts.When the trolley moves the hoisting chain runs over the two sheaves with the trolley and the sheave in the block supporting the hook. Thus there is friction in the sheaves, there is also rolling and journal friction caused by the movement of the four rollers.

We saw in the design of the hoisting gearing that the efficiency of the sheaves or chain pulleys, see Figure 8, is as follows: efficiency of sheaves I and III = .96, and II = .95, or this means that sheaves I and III requires .04 of the work required to move the supported load to move the sheave and, sheave III requires .05 of the work done on load. Then the maximum resistance of the trolley due to friction in chain sheaves $= \frac{W}{2}(.04 + .04 + .05) =$

3000 x .13 = 390 lbs.

From notes on rolling and axle friction by Professor Talbot

$$P = \frac{a}{r} W\mu + \frac{b}{r} W$$

a = radius of axle = 1.125"

r = " " wheel= 3"

W = load on wheel = $\frac{6000}{4}$ for maximum.

μ = coefficient of friction = .06

b = .03

$$P = \left(\frac{1.125}{3} \times \frac{6060}{4} \times .06 + \frac{.03}{3} \times \frac{6000}{4}\right) 4 = 195 \text{ lbs.}$$

Then the entire pull required to move trolley = 390 lb.+ 195lb.=585lb.

This force must be exerted by a pull on the hand chain. For maximum laods on trolley considering losses due to friction in the gearing to move the trolley it will require a pull of about 60 lbs..on the hand chain. This pull can easily be exerted by either one or two men.

The pivots and bearings at the upper and lower end of the mast was desiged with an ample factor of safety and the principles envolving their designs is simple and does not require mentioning. The hook is designed according to principles given in Professor Goodenough's Notes on Mechanics of Machinery.