DESIGN AND CONSTRUCTION OF MODERN ELECTRIC TRAVELING CRANES

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THESIS

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INTRODUCTION

Since time immemorial, devices and appliances to raise and lower loads have been a necessity in various industries, construction work and processes of manufacture. Many of these were crude and inefficient, the primary end in view being to move loads at any cost.

In analyzing the contributing causes which have made the electric traveler such a necessity there is probably no one factor so prominent as the application of electric power to various crane requirements. Other forms of power have been used, yet there is no other form which is so adaptable to the requirments of traveling cranes. In view of this, many improvements have taken place which redound to the benefit of the crane manufacturer as well as purchaser.

Contrasting the earlier devices and appliances for the moving of loads with those of today, we find the conditions and demands different and more far reaching. Where the crane was used exclusively as a mover of loads irrespective of cost, it is now, in addition, a conveyer to a large extent, not only lifting objects but transporting them from place to place. Where formerly manual labor was employed to transport material, the electric traveler, bucket crane and lifting magnet carry large quantities, reducing time and labor costs to a minimum.

It is the purpose of this thesis to consider the actual design as well as the commercial construction of the electric traveler. It is not the purpose to consider the electric traveler from a purely theoretical standpoint but to

blend scientific principles with practical requirements; to satisfy the customer's demand without losing sight of the manufacturer's interests. It is only with due consideration and proper co-ordination of these various factors that the interests of all concerned are best served.

II

CLASSIFICATION OF CRANES

In general, cranes may be classified according to their motions, or their source of motive power. Under the first classification a crane would be rotary or rectilinear, while under the second we would have the following sub-divisions:

Cranes having their power attached, electric, pneumatic and steam.

Locomotive cranes, which supply their own power.

Hand--where manual labor is used for operation.

All electric travelers can be grouped under one of five heads. So far as industrial requirements are concerned it seems more logical to classify the electric traveling crane according to the service it is to perform. Thus we would have the single drum crane which hoists and lowers its load by means of attaching the load to an ordinary crane hook. This is the type of crane in greatest demand and has a wide application in the industrial world. The average shop would require nothing further than the hook. A structural shop, for instance, would require a wider gauged trolley which would permit the suspension of a lifting beam from the ropes. This may result in two drums on the same shaft, yet in principle it is nothing but a single drum extended. Foundries, machine shops, ware-houses, boiler-shops, assembly floors, locomotive shops, etc., can use the same type of crane. Some details must be altered to suit the

shop, the chief one being, in most cases, the application of the power to the load.

A second class would be the bucket crane. This class of crane must be very rugged and substantial since the service is extremely severe and frequently rough. Where it formerly required a dozen laborers to unload and convey a barge of coal, for instance, one bucket crane will dispense with nearly all of them. There is a saving of time and money. The operating expenses and power consumption are small items when compared to the labor item, to say nothing of the time gained which is a very important consideration in most enterprises. Coal, sand, coke, cinders, slag, etc., offer a wide application for the bucket trolley.

A third class of electric traveler is the gantry crane. This may be an ordinary gantry, two or one-legged, it may be a single cantilever or a double cantilever gantry. It may have a hook trolley or a bucket trolley. In this class as in any other it is impossible to conceive of all the possible industrial demands and subsequent designs that could be produced.

The ladle crane enables the carrying and pouring of large quantities of metal heretofore impossible. The service is severe at all times and every part must be reliable and not fail its purpose at critical moments.

In the electric stripper we have a fifth class which is used to remove the moulds from ingots.

III

PRINCIPAL PARTS OF ELECTRIC TRAVELING CRANE

Every crane is composed of two parts and these are known as the trolley and bridge. The trolley comprises the entire hoisting and traversing mechanism, in fact everything above the bridge rail. Every part below the bridge rail down to the runway rail is included in the bridge, the machinery for longitudinal travel not excepted.

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IV

PROPORTIONING OF TROLLEY UNITS.

Trolley Sides.

The trolley is the most important part of the crane for reasons which are presented farther on in our discussion. It is the unit of the crane upon whose successful operation more is dependent than upon the bridge. It receives the most careful attention both from the designer's and the manufacturer's standpoint. The success or failure of any crane depends primarily upon the reliability and serviceability of the trolley.

The sub-structure of the trolley usually consists of two cast frames known as the front and rear trolley sides, looking at the front bridge platform on a line parallel to the runway rails. The front and rear trolley sides are securely bolted together by a box shaped casting known as the machinery girt. This girt rests upon machined ledges of the trolley sides, thus insuring a firm foundation for the parts of machinery it is designed to carry. Incidentally this form of connection also obviates the necessity of having all reamed bolts.

The trolley sides should be rigid both vertically and laterally. In many cases design and foundry considerations necessitate a depth of casting greater than that which is required to carry the vertical loads imposed. The designer must always bear in mind that it is impossible to obtain as perfect a casting as the pattern. If this is lost sight of figures may show a safe section while in reality the actual casting is not safe. This is not speaking of faulty

design but simply those irregularities due to unavoidable variations. In one place very little metal may fulfill all the requirements of strength, yet the molten metal must flow and internal shrinkage strains must be reduced to a minimum, hence it will be advisable to add more metal.

Another condition frequently encountered in the design of machinery is where figures will show that a section is amply strong yet to the eye it seems insufficient and out of proportion. In this case it is frequently advisable to strengthen the section until a more substantial appearance is produced. Each part must not only be amply strong for the required service, but the machine must also present a pleasing effect to the eye that will help sell it. It must be remembered that any part of a machine may be satisfactory from the designer's point of view, and yet be a failure commercially. Ultimately the machine is made to sell and must meet competition successfully.

Care should be exercised in laying out the trolley sides so that cores are reduced to a minimum. Very often a little forethought will do this. A rib placed in one position may not allow the sand to "stand up", yet it was probably quite feasible to place the rib so that the sand would leave its own core. Casting after casting will be made from these trolley side patterns and any saving effected, whatever the form, will be in exact in proportion to the number of castings produced.

In a similar manner the ease or difficulties that may be encountered in the machine shops should be anticipated.

To form a connection or to accomplish some feature is frequently the easiest part; but to attain this in a way which will be practical and without special or cumbersome machining may be quite another problem. Likewise the machine operations should be held down to a minimum. The careful observer will find examples of unnecessarily complicated machining. Not only does this add to the cost but also is there greater possibility of such machining not being done satisfactorily.

A simple example of how machining operations and cost may be reduced is found in a bracket. We will assume a bracket having four base bolts. In one case the bracket will have narrow finishing pads the entire length of the base. This will necessitate a planer or shaper operation. Suppose now, other things being equal, this bracket had been provided with round bosses instead of the finishing strips. The bracket is fully as rigid, possibly requiring a little heavier base. The bracket is set up on the drill and all four holes drilled. Next the four bosses are spot-faced instead of doing this on the planer or shaper. Thus the following saving has been effected:

1. One routing.
2. One setting up.
3. Cheaper labor.
4. In most cases the spot-facing requires less time.

If the bracket goes on some structural or others unfinished work the bosses have the additional advantage

of being more easily fitted to structural irregularities.

Most of the foregoing points apply to all castings, however they are mentioned under this head because of the very nature of the trolley sides these points are more easily overlooked and not found until too late.

Attempts have been made to use but one pattern for the front and rear trolley sides. The difficulties are obvious. While there are advantages in having but one pattern yet the disadvantages decidedly outweigh the advantages. One good reason for having two different patterns for the average crane is the fact that castings will be made in large quantities, and by having two patterns they should last twice as long as the single pattern.

Structural Versus Cast Members.

For the standard line of cranes the cast trolley sides have the following advantages:

1. Deflection, as compared to structural frames, is eliminated.
2. The casting is easily adapted to make connections.
3. Bearings and sub-structure are one unit.
4. Castings can be jigged thus saving time and insuring interchangeability and accuracy.
5. To assemble cast frames requires less skilled labor.
6. Quicker delivery to a customer.

The structural frame is ideal for special trolleys. By its use the pattern requirements are comparatively simple and reduced to a minimum. In laying out a ma-

chine of this kind the matter of deflections is probably more important than any other one thing, especially where the service is severe and continuous. Clearances should not be taken so closely that it will require unnecessarily accurate work to hold to given limits.

Machinery Girt.

The machinery girt usually carries the hoist motor and the mechanical load brake or, when dynamic braking is employed, the electric brake. Sometimes the machinery girt is also used to support the upper block sheaves. While this reduces the length of trolley, yet it is objectionable inasmuch as the sheave leads may influence the machinery mounted on the girt. In case of excessive load, shock or some accident, the machinery girt would be affected and not as easy to replace as a separate load girt. Its section is box shaped with ribs at the ends and intermediate points.

Upper Block Girt

When a separate girt is used to support the upper sheaves two channels supporting an idler sheave will answer for the smaller capacities. For heavier loads a section built up of plates and angles must be used. With the system of having two ropes winding on the drum the load girt is called upon to carry heavy loads. Take, for instance, a fifty ton crane having twelve parts of rope. The load girt must support a vertical load of

83.33 pounds. This load is not rigid and produces lateral forces for which provision must be made. Occasionally a crane is used to pull loads horizontally, thus inducing heavy lateral forces. Perhaps the severest duty of all is a suddenly applied load. If the crane operator is not careful to tighten his rope gradually, abnormal strains will be induced proportional to the speed and the amount of slack rope. Another condition which must be considered is the sway of the load. Usually the load is still ascending when the bridge or trolley travel or traverse, respectively. It is obvious that, primarily, the sway of the load is dependent upon the load, the distance of suspension and the rate of acceleration.

Upper and Lower Block Sheaves.

The average pitch diameter of sheaves is usually about twenty-four to twenty-six times the diameter of rope. The throat is extended well beyond the pitch diameter to preclude all possibility of the rope jumping out. Formerly most all sheaves were made with arms, but of late the tendency is to use a web with cored holes. This simplifies the pattern making and when the web is reenforced by ribs, almost any amount of lateral strength may be obtained with less metal. The sheaves are turned to give a true bearing for the rope and to make the pitch diameter concentric with the bore.

Lower Block.

A simple design for a lower block consists of a plate reenforced by a narrower plate, the two forming one side. At the upper end is the shaft supporting the sheaves while the lower part supports the cross-head which in turn carries the hook. To facilitate the turning of the hook it is mounted on a ball bearing.

Rope Drum.

In most all cases rope drums are made of cast iron since it lends itself much better to the turning of grooves Where a light drum is necessary cast steel is sometimes used. In determining the inside drum diameter it should be remembered that the core may not be exactly concentric hence a liberal allowance should be made above any theoretical determination. The allowance necessary will vary with different foundries.

Different formulas have been derived for the determination of drum section, some of them being too cumbersome for practical and rapid calculations.

Let $P =$ pull on one rope in pounds.

" $t =$ thickness of drum at bottom of groove.

" $p =$ pitch of grooves.

" $l =$ distance in inches between drum bearings.

" $D =$ drum diameter at bottom of groove.

" $d =$ inside diameter of drum.

" $Bm =$ bending moment.

" $S =$ section modulus of drum.

Let B = fibre stress in bending.

" C = " " " compression.

Then $S = \frac{D^4 - d^4}{10d}$

" $B = \frac{Bm}{S}$

" $C = \frac{P}{pt}$

Combined stress $= \sqrt{B^2 + C^2}$

For cast iron $\sqrt{B^2 + C^2}$ should not exceed 6000 pounds per square inch.

Strong extra flexible wire rope has taken the place of chains. For hoisting drums the plow steel rope is used having a hemp center, six strands and thirty-seven wires to the strand.

A feature of the modern drum is that two ropes are wound on it simultaneously. On the earlier cranes it was considered unsafe to wind more than one rope on a drum, yet at the present time cranes having a capacity of one-hundred tons are employing the two-rope scheme with good success. When one rope per drum was used, two drums opposite each other wound the ropes. This had the undesirable feature of causing the hook to ascend in a spiral like manner. The two-rope single drum scheme effects a truly vertical rise of the hook so long as the sheaves are placed in proper relation to one another.

Gearing.

The gears in the hoisting train are usually made of cast steel, and pinions of steel forgings. All gears and pinions are cut from solid blanks.

Considerable variation exists in the matter of determining the faces of gears. Some builders will use a greater face than given by the Lewis formula while others use a smaller face than allowed by Lewis. So far as hoisting machinery is concerned this seems to be more a matter of individual choice rather than expediency. Gear faces proportioned by the Lewis formula are conservative and reliable for crane service.

In the past many overhung gears have been tolerated chiefly because in the design it was the path of least resistance. These, however, are a thing of the past and the near future will see the elimination of all overhung gearing for standard work.

Shafting.

Medium steel with a carbon content of about .25 to .30 is extensively used in crane construction. Cold rolled steel is also prominent where the shaft diameter can be used unchanged thus avoiding machine work. For steel mill service Open Hearth steel having .40 to .50 carbon is specified. Where a light shaft or exceptional strength is required special steel is employed.

Bearings.

Very little can be said in regard to the bearings since they possess no individual difference as compared to other types of slow moving machinery. In some isolated cases ring oiled bearings are specified. While ring oiled bearings have obvious advantages, yet is has not been proven that they are desirable on slow moving

crane journals.

Lifting Magnets.

For certain classes of work the lifting magnets offer many advantages. For example, in the foundry it may be used to clean the sand, gather the flasks and transport the castings. Since it is used as an auxiliary its use in no way precludes the service of the regular crane hook the magnet being suspended from the hook. Provision must be made to wind and unwind the magnet cable as the magnet is raised or lowered. This may be accomplished by driving a drum from one of the shafts or by using a cable reel which automatically winds and unwinds the cable. When the magnet is not in use the cable is wound and fastened on its drum enabling the trolley to be used as any ordinary hook trolley.

Crane Wheels.

In general, cast iron wheels with chilled treads are used, ground true and to uniform circumference. Cast steel and steel tired wheels are frequently specified. A very special and seldomly used wheel is made of manganese steel. This metal being so hard that it is impossible to machine, it becomes necessary to cast a soft steel center integral with the wheel. In this way the wheel can be bored, keyseated and the hubs finished. Where the center of softer material is cast with the wheel it must not become loose. One scheme to accomplish this is to provide the center with a series of "V" shaped cylinders turned on the circumference, the theory being that in casting the ends of the "V" will fuse and effect a union with the surrounding molten metal.

Crane Hooks.

Crane hooks made from a high grade of wrought iron are preferable to steel hooks since defects, produced by wrong heat treatment, are more easily detected. Defects in a steel hook are invisible. In the case of a dangerous overload the iron hook will give warning by a visible, slow deflection before actual failure occurs.

For comparatively light loads drop forged hooks find extensive application, while the larger capacity hooks are forgings made from tough refined wrought iron.

V

PROPORTIONING OF BRIDGE UNITS

Girders.

Bridge girders are proportioned the same as any other girders, there being no distinctive features except in the requirements of lateral strength necessary in the upper flange. As a rule the front girder carries the platform, motor and cage, thus loading the front girder heavier. When the cage is placed at or near the center of span special care must be taken to have the girder strong enough to resist the torsional stresses produced by the cage in starting and stopping the crane. These stresses are an indeterminate quantity. Good judgment based on experience is the best guide in selecting the proper section.

End Trucks.

The end trucks are either cast or strutural or a combination of both, as circumstances may require. The connection between the girder and truck beam should be very rigid to insure strength and proper alignment. The wheel base should never be less than one sixth of the span otherwise the wheel flange pressure becomes too great. This is especially true where the bridge motor is not mounted in the center of the span, the tendency being for one side to overhaul the other, due to the difference in the angle of torsion in the driving shaft. Furthermore, the trolley will also be operating near the bridge extremities thus placing very unequal loads on the end trucks tending to throw the crane " out of

square" which makes a substantial wheel base a prerequisite to successful bridge travel.

Driving Shaft and Brackets.

The maximum demand is made on the driving shaft and gears when the load is near one bridge extremity since an unequal tractive effort must be transmitted which should be considered in determining the sizes of these members.

The driving shaft brackets should be securely bolted and elevated as the center of the span is approached such that, when the girders are carrying their rated loads, the deflection of the girders will bring the center line of driving shaft on a horizontal line. Good practice demands that brackets be spaced not over ten feet apart and preferably less for the smaller shafts. Babbitted bearings are extensively used because of their adaptability.

Bridge Motor Brake and Platform.

The bridge motor should preferably be located at the center of the span thus equalizing torsional deflections and a consequent tendency of keeping "in square" the end trucks. It is bolted directly to the front girders or mounted horizontally on the platform supported at this point by substantial brackets.

It is a good plan to apply the foot brake directly to the motor armature thus requiring a less powerful brake for the same braking effect. Where the brake is

applied to the driving shaft additional bending and twisting stresses must be taken by the shaft. Since a brake of this kind must be equally effective in forward or backward motion the two lever arms connecting with the band extremities must be of equal lengths.

In order to make all parts of the bridge and trolley machinery readily accessible a platform is provided the full length of the front girder with a hand rail of ample height. When the motor is mounted on a horizontal base and takes up considerable room it becomes necessary to widen the platform around the motor so that a man may pass with ease and without having to step over any parts. The platform floor is made of either wood or checkered steel plates as the case may require.

Operator's Cage.

Facing the bridge machinery the cage is usually mounted on the right hand side of the front girder, though at times the service demands that the cage be placed at the left, in the center, or even on the trolley itself. Its location should be such that the operator may have the best possible view at all times and a comfortable position. In this connection it is well to locate the hoist controller on the side of the cage nearest to the center of span to enable the operator to look down over the cage edge; likewise it is also best to so place or wire the controllers that corresponding lever operations will bear the same relation with regard to the operator. The most natural way is to have the controllers so arranged that

the cessation of any motion shall be produced by a forward movement of the controller levers. In this way the different operations will be simplified besides saving possible accidents at critical moments.

The cage contains the foot brake lever which is connected to the motor brake by means of bell cranks and rods mounted on the side of the girder. The cage floor consists of wood boards or steel plates as may be required. Also a foot gong within easy reach of the operator. The average crane has either three or four controllers, hoist, bridge and trolley traverse,--and where the trolley has an auxiliary drum a fourth controller is required. Where the controllers are mounted to the rear of the operator and connected to levers in the front of the cage there is no obstruction to the operator's view. This is a desirable feature especially where the nature of the work demands close attention of the operator.

Every cage contains a switch-board of ample size to mount the following: For a three motor crane--a magnetic switch, main switch, main line fuses, and fuses for each of the three motor circuits. The main switch should preferably be placed above and within easy reach of the operator, such, however, that accidental contact will be avoided. A lamp and its fuses are also provided.

VI

ACCESSIBILITY OF MECHANISM

Examining some of the older cranes it is evident that accessibility was not a serious consideration, however, any machine of today which will stand the scrutiny of the intelligent buyer, besides rigid competition, must not be deficient in this particular. First of all, such parts are unnecessarily cumbersome to assemble, but the great objection comes after the machine is in the hands of the customer. It may be necessary to remove some parts prior to an inspection; breakage may occur, due to any one of a number of causes, making it imperative that some parts of the machine be dismantled. When this becomes necessary it will be seen that inaccessibility may be largely responsible for making idle both men and equipment.

The introduction of half hexagon bushings on the axles of the Master Car Builder's type enable its users to remove the axle by simply releasing the weight from the wheel. The majority of trolleys at the present time are not equipped with M. C. B. bearings, yet the tendency in the newer designs is to adopt this type of bearing.

For the bridge axles the M. C. B. feature is especially desirable since they are much larger and, as a rule, more inaccessible. The driving shaft should not be in too long sections in order to facilitate the handling. Each shaft should be removable without disturbing a

second.

Where gear guards are used or where moving parts are in any way enclosed it is desirable to provide inspection covers so that their operation may be observed without removing or loosening any part. Frequently minor adjustments may be made in this way with comparative ease.

VII

FLEXIBILITY A VITAL FACTOR IN THE DESIGN OF ANY HOISTING MECHANISM

Aside from correct scientific and practical construction of a hoisting mechanism in all its phases, the success or failure of such mechanism will, in a very large measure, be determined by its flexibility, i. e., the adaptability of the greatest number of parts of said machine, without change, to the greatest number of demands which may be required of that particular class of machinery. When one considers the variables which must be met or the combination of these it is apparent that it embraces many features all of which require careful consideration.

First of all, there is the great variety of hoisting speeds that must be accommodated which immediately affects the hoisting gears and the motor, as well as the gear guards. Now there are a great many different makes of motors. One customer prefers this make of motor and another some other make, etc. Each manufacturer has motors which differ from those of his competitors mechanically and electrically. It is of little avail to tell a prospective customer that a certain make of motors will answer his requirements, for he will at once conclude that the manufacturer is trying to give him something inferior at perhaps a greater profit. Added to this there is the choice of magnetic brake for the hoist motor. Not infrequently it becomes necessary to employ

a motor and brake manufactured by two different concerns, thus further adding to the already complex problem of flexibility.

What has been said of the hoist mechanism also applies to the traverse, although deviation is not of as much consequence. Sometimes the span is so short that it is absurd to specify any speed at all since the motor would have no chance to accellerate up to full speed. Hence it is obvious that so far as actual results are concerned the traverse motor speed and gearing could be higher than specified.

The question of flexibility manifests itself with equal force in the pattern shop, the foundry, and finally in the machining of the castings. If, in view of any of the aforesaid conditions, it becomes necessary to make alterations in existing patterns, or perhaps new ones, valuable time is consumed and , in reality,it becomes semi-special work. But special work means one, or, at best, a limited number which in turn spells production at reduced profits. Apart from the ability to make a quick delivery when these variable features have been anticipated, there is the added advantage that parts can be placed in stock in quantity, completely machined, thus reducing production costs to a minimum.

Aside from the obvious difficulties probably the greatest single deterrent in flexibility has been the greatest initial expense involved which, however, in the end pays big dividends. It is safe to predict that the

future trolley will show a very marked improvement as regards flexibility and simplicity.

The question of flexibility also embraces the shape of parts. Unnecessary curves or angular lines can usually be reduced to a minimum and thus greatly facilitate the placing of adjoining parts and the determination of clearances.

In this connection it is well to remember the flexibility of mechanical load brake. All hoisting mechanism has a load brake or its equivalent. For special work structural steel frames are extensively used and if this is kept in mind in the design of standard work, a brake should be produced which can be removed as a unit and easily mounted on a structural frame of a similar capacity.

VIII

INDUSTRIAL DEMANDS WHICH TAX THE DESINGER'S INGENUITY.

Probably an actual case will best illustrate some of the difficulties which must be met. A plant, we will say, was built five years ago adequate for the demands at that time. Possible growth and expansion had been considered. At that time a two ton jib crane was sufficient to handle all loads. The need of a five ton electric traveling crane was thought necessary at a later date so a runway was placed with this in view at the time the building was erected.

Five years later the jib crane is no longer satisfactory. The firm finds that, in justice to their business, a ten ton crane with five ton auxiliary is required. But the runway is already in place. The building can be extended but it is out of the question to make any change in the height of the existing building. Furthermore, physical shop conditions demand that the hook approach the runway on each side within a fixed distance. Thus we have the anomalous condition of placing a ten ton crane, with auxiliary, within the same clearance dimensions required by a five ton crane without auxiliary. Above the roof trusses limit, while any lowering of the machine simultaneously decreases the vertical hook movement. Thus it will be seen that a machine must be designed within narrow limits and without making the cost of same exceed estimated figures.

Other examples are: the combination of hoist motor and magnetic brake each of which is manufactured by another firm; hoist motor with magnetic brake so arranged that a load may be hoisted electrically or manually, all control being effected from the floor by pendent cords; an adjustable automatic device for a cantilever gantry so arranged that it will gradually retard the boom speed in the last few feet of travel, both raising or lowering, and finally stopping the boom at a given point should the operator fail to shut off the power at the proper time. This device to be so designed that it will not prevent the operator from stopping the boom through the controller at any desired point in its travel.

Many other cases could be cited, however, these serve to illustrate the diversity of demands.

IX

POWER SUPPLY OF BRIDGE AND TROLLEY MOTORS.

The majority of crane motors are of the direct current type and series wound. The main wires are usually placed beneath and to the inside of the runway girders, on one side only. The bridge truck beam carries a bracket supporting an arm to which are fastened the main collectors supplying the power to all motors. For obvious reasons this collector bracket should preferably not be placed between the girders. The bridge motor takes its supply direct from these main collectors. Since the trolley moves along the bridge it is necessary to provide conductors along the bridge girders from which the trolley motors draw their power. Where there is no platform on the rear of the bridge this is the ideal place to locate them. When, however, it is not convenient to so locate them they must be placed on the inside of one or both of the girders. When this becomes necessary particular care must be taken that the hoisting ropes or any part of the trolley shall not come in accidental contact with the conductors. In fact the electrical and mechanical clearances as well as insulation should be liberal, and preclude the possibility of short circuits. It is well to place the trolley conductors on the girders in such a position that when the pull of the hoisting ropes is such as to bring them to touch the bottom plate of the girder the conductors will

still have ample clearance. This is an extreme condition. Occasionally, however, a crane is used to drag a load or move cars and when this is done it is imperative to have the conductors out of reach.

Both wires and plain flat bars are used as conductors. Wires have the advantage of lightness as well as being easily insulated. Where flat bars are used supports must be provided at frequent intervals which is not necessary with wires. Should a wire break from some cause or another it must necessarily fall with the possible result of serious accident. Where flat bars are used there is no danger from this source.

For the average three motor direct current crane nine trolley conductors are required: four each for the hoist and traverse motors and one for the limit switch.

X

DETERMINATION OF POWER UNITS

The electric crane motor is serviceable and reliable even under extreme conditions of loading. Its marked adaptability to various loads makes it an ideal machine. Heavy overloads will be carried without any serious consequences so long as the overload duration of time is short.

For electric travelers no motor should be overloaded when carrying its rated load. Trolley traverse and bridge motors should have some excess horse-power to enable them to accelerate quickly. Cranes of the same capacity will frequently have motors of different horse-powers. This discrepancy can be accounted for in three different ways or a combination of any: Different efficiencies, difference in the builders rating, and individual practice in the excess of horse-power allowed.

The hoist gear ratio for a D. C. series motor may be determined in two ways, as follows:

Let C = circumference of drum in feet.

" r.p.m = revolutions of motor per minute.

" H = hook speed in feet per minute.

" R = hoist gear ratio

Then $\frac{r.p.m.}{R}$ = Revolutions of drum per minute.

Then $\frac{C \times r.p.m.}{R} = H$

1) or $R = \frac{C \times r.p.m.}{H}$ for single part of rope off drum.

Since the speed of the series motor varies through wide limits, for reliable results, it becomes necessary to ascertain the exact motor torque exerted and the corresponding speed. The normal speed rating cannot be used in the above formula unless the motor is actually carrying full load.

A more direct solution is to equate the load torque to the motor torque and compute the required ratio therefrom.

Let L = load on drum in pounds.

" r = pitch radius of drum in inches.

" T = motor torque in inch pounds.

" E = efficiency of hoisting mechanism.

Then $Lr = TER$ or

(2) $$R = \frac{Lr}{TE}$$

The size of hoist motor may also be determined in two ways.

Let P = load on hook in pounds.

Neglecting weight of parts then,

(3) $$\frac{PH}{33000\ E} = \text{actual required H. P.}$$

Again neglecting weight of parts,

(4) $$\frac{Lr}{RE} = \text{actual required motor torque.}$$

Comparing the two formulas it is evident that (3) does not consider the diameter nor the load on the drum. In consequence thereof it would be possible to compute the same H. P. for two different machines of the same

capacity although the rope reeving and drum diameters may not be the same. This would necessitate two different gear ratios for the two cases. Where the size of motor is determined by formula (3) it may be found necessary to add another reduction in order to use the propesd motor, which may eliminate the intended machine and the design or substitution of another having a greater gear reduction. An actual example will make this clearer.

Let P = 40,000 pounds.

" H = 15 ft. p. m.

" E = 76 %

Then by (3) $\frac{40000 \times 15}{33000 \times .76} = 23.9$ required H. P.

We will select a Westinghouse 8K, D.C. crane motor, 220 volts, 25 H. P. @ 575 r. p. m., whose torque is 2740 inch pounds.

To use (4) we will further take following conditions:

Let L = 13333 pounds. r = 12" - six parts of rope.

Then by (4) $\frac{13333 \times 12}{59 \times .76} = 3570$ inch pounds, required motor torque.

But this result is greater by 830 inch pounds, neglecting any excess H. P., and hence cannot carry its rated load without overloading, in which event we should not get a hoist of fifteen feet per minute.

From (1) $R = \frac{24 \times 11 \times 575}{12 \times 15 \times 3} = 80.5$ required hoist ratio for Westinghouse 8K motor. Therefore we should either have to provide a larger motor or a greater gear

ratio to obtain the required hoisting speed.

Again, if we had computed the size of motor according to (4) we obtain the following:

$$\frac{13333 \times 12}{3570 \times .76} = 59$$ hoist gear ratio required, which our proposed machine can accommodate.

Traverse Motor.

Let T = required tractive effort in pounds.

" G = dead weight plus live weight.

" D = diameter of trolley wheel in inches.

" d = diameter of axle in inches.

" f = coefficient of friction--about .07.

" E = efficiency in per cent.

" S = linear speed of trolley.

Then $T = \frac{.07Gd}{DE}$ and

Required H. P. of traverse motor is $\frac{TS}{33000}$

While this gives the actual horse-power required, yet seldomly, if ever, is the traverse motor developing its full horse-power. In consequence, if it is essential that the traverse speed be adhered too closely, it becomes necessary to ascertain the exact torque exerted by the motor and base the gear ratio on the r. p. m. at this torque.

Bridge Motor.

The size of bridge motor may be determined in a similar manner as the traverse motor. In practice a more

preferable way is to compare load torque and motor torque and proportion the gear ratio accordingly. A liberal excess of H. P. should be allowed on the bridge motor since the position of load on the bridge may demand excessive power especially on long span cranes. Where, for instance, the trolley is near one bridge extremity and carrying full load, there is a tendency for the bridge to get "out of square" while traveling.

Furthermore, there is the deflection of girders which cannot be eliminated. While it is possible to compute the deflection of the girders at different points and elevate the driving shaft brackets accordingly, yet it is not so easy in practice to locate them with mathematical precision, considering various variables in the process of manufacture. Thus it is evident that excessive friction may be produced due to irregularities, all of which demands a liberal excess of H. P. to provide for these contingencies.

XI

SAFETY APPLIANCES.

So long as the human element is a factor with which we must reckon, it is necessary to provide means that will prevent the wrecking of machinery, injury to workmen and possible death. The fundamental principle of all safety appliances is to take action, either suddenly or gradually, before the danger zone is reached without in any way interfering with the regular operations. Such devices must, of necessity, be automatic, i. e., they must act regardless of any action or inaction of the operator.

The overtravel of hook or the hoisting of the load until it interferes with parts of the trolley is a source of trouble unless there is provision made to prevent this. Where it becomes necessary to utilize the full hoist continually it is obvious that such provisions for safety become of considerable moment. Sometimes it is required to limit the hook travel in the lowering as well as in the hoisting direction.

Manufacturers have produced devides of considerable diversity to limit hook travel and the merit of any one of them depends upon a number of things taken collectively. A device may answer every requirement of safety and yet be undesirable from the crane builder's point of view. The device must not be too expensive and the external proportions such that it can be attached readily

to any hoisting mechanism. Where the attachment of the device depends upon a certain construction, very frequently considerable expense is involved in attaching such a safety device which in reality adds to the cost of such appliance. Whatever the construction, it should never diminish the vertical hook travel.

In response to the demand of a positive safety device to prevent overtravel of hook which would be satisfactory to the purchaser of cranes as well as the manufacturer the Shaw Electric Crane Co., developed a new device which the writer had the pleasure of desinging and which is now used on all the cranes manufactured by this Company. In the design of this mechanism the following points were kept in mind, and the apparatus developed accordingly:

1. Reliability at all times and positive action.

2. Simplicity consistent with the requirements making for easy and speedy manufacture.

3. To make the apparatus a complete standard unit in itself which could be adapted to various hoists without any alteration or substitution of parts.

4. Flexibility--to enable its attachment to the various classes of cranes without the necessity of long and tedious layouts.

5. To employ the same device for open or closed circuit, single or double acting.

6. To make the outside dimensions as small as possible.

7. The ultimate cost to be low consistent with the above requirements.

Through the medium of a crank connection is made with the drum shaft, the crank being on a worm and driving a worm wheel which in turn has a contact lug. As the worm is driven by the drum the contact lug on the worm wheel is brought opposite a similar lug on a trip lever, which opens the motor circuit through a magnetic switch in the cage. The limit switch will not allow the motor to get any current in the hoisting direction, but any time that the controller is reversed to the lowering direction everything resumes its normal course. Furthermore, the device keeps the hoist circuit open for several revolutions thus eliminating the danger existing in an instantaneous opening--that of continuing in the hoisting direction.

The range of limit switch is ample for all hoists required, the adjustment being effected by the bringing nearer or farther the point of contact as the case may require.

The limit switch uses the same parts for open or closed circuits, the change being accomplished by different relative positions of parts. Generally the closed circuit is used because the breaking of a circuit is positive, but not the making of a circuit. In other words, the electric circuit is complete until interrupted by the device.

The entire mechanism is enclosed in a rectangular box whose external dimensions are $4\frac{1}{2}'' \times 6\frac{1}{2}'' \times 5''$. The box

has two lugs by means of which it is bolted to the end of the drum bearing.

Since the device is adaptable to any hoist without change, it can be manufactured in large quantities thus materially reducing the cost of manufacture.

XII

AVERAGE VS. STEEL MILL CRANES

The steel mill crane is a product in response to a demand for a crane thoroughly reliable at all times under most exacting conditions and equipped with every safety device. It must be berne in mind that the steel industry represents an investment of enormous capital and the process of manufacture is one which, perhaps more than any other, places the life and limb of those engaged in jeopardy. Delays of minutes may make inoperative an amount of equipment whose value totals thousands of dollars. Thus it will be seen that steel mills must have the best possible and most modern equipment irrespective of cost.

While such cranes are primarily used for steel mills yet there are other industries to which these remarks apply, in a measure, with the result that a good many steel mill specifications are incorporated in buying a crane. In fact, the steel mill specifications are manifesting themselves in various phases of the crane industry.

The mill crane immediately gives the impression of massiveness, strength and liberal proportions. In many parts the factor of safety is greater. All couplings are of the safety flange type. All parts are made of steel except drums and collecter shoes. All pinions are made from forged steel and no pitch finer than four diametral pitch; no overhung pinions are used. No wood is

used for any construction, conductor insulators not excepted. Trolley shafting is usually made of Open Hearth steel and the bridge driving shaft of C. R. S. No trolley wheels are less than 12" in diameter and bridge wheels not less than 24". All treads are so proportioned that with a wear of 1 1/4" on the diameter they will still be serviceable. All through bolts are used and preferably no reamed bolts.

The hoist motor magnetic brake is designed to hold the load independently whereas on standard cranes the brake is proportioned to hold about one-half the motor torque.

The bridge motor and driving shaft brackets are mounted horizontally on substantial brackets while standard cranes have the motor and brackets bolted to the side of the girder. All axle bearings are of the M. C. B. type. Drum and sheave diameters not less than thirty rope diameters, whereas standard cranes run under thirty. No babbitted bearings.

The preeminent feature on the mill trolley is the almost exclusive use of the dynamic braking hoist controller, which has been perfected within the last three or four years. As has been pointed out, the steel mill service is unusually severe and in consequence thereof mechanical load brakes have been the source of delays and heavy repair costs. A brief analysis of the mechanical load brake will show why this is so.

When a load is lifted a certain number of foot pounds of work must be done. Energy has been stored which is available for useful work or it must be dissipated in some form or other. The principle of the mechanical load brake is to dissipate this energy in the form of friction. In other words the load brake is a means for transforming potential energy into heat which must be dissipated rapidly if the brake shall not become excessively hot. When we multiply the hook load by the hoist, neglecting harmful resistances, it becomes clear that even light loads store considerable energy, and which is proportionately greater for heavier loads and longer hoists.

Where dynamic braking is used the control in lowering is effected by converting the motor into a generator thus returning useful electrical energy to the main line. If the resisting force, due to friction, is greater than the load the motor lowers the load by power; but if the frictional forces are exceeded by the load the motor immediately becomes a generator diminishing the speed of lowering which may be varied by the insertion or removal of resistance. The prominent feature of dynamic braking is that useful electrical energy is returned to the main lines, and if not then a part of it appears in the form of heat through resistances especially designed for this purpose. This cannot be said of the mechanical load brake.

While dynamic braking actually controls the speed

of lowering, yet it will not hold a load in any position. For this reason it becomes necessary to add a second electric brake on the motor gear shaft. For mill cranes this brake as well as the motor brake is made powerful enough so that each will hold the load independently. Such a combination is very powerful and quite instantaneous and there is a question if this action is not too powerful for ultimate satisfactory performance.

In dynamic braking the worst eventuality which could happen would be the simultaneous failure of the supply voltage, magnetic brake and the controller wiring yet, in that event, because of safety devices used in dynamic braking, the load would actually descend so slowly that the danger of any damage would be remote.

Automatic Magnetic Switch Control.

Another very important feature of large mill cranes is the application of the automatic magnetic switch control. Where large motors are used it has been found that manually operated controllers have a high maintenance charge and are cumbersome to operate.

The numerous advantages obtained through the application of this control all reduce delays, and it is this feature more than any other which recommended this form of control.

Switch Board.

On the mill switch board we usually find overload relays for all motor circuits instead of fuses, thus eliminating the expense of blowing fuses; push button for opening or closing quickly the circuit breaker switch; a safety plug which the operator can remove and lock giving certainty that the crane will not be traveled until he has replaced the plug. It also has the main line switch, lamp and its fuses.

XIII

PRINCIPAL CALCULATIONS IN THE DESIGN OF TWENTY TON TROLLEY.

On the following pages are given the requirements and principal calculations of the twenty ton two motor trolley shown on the enclosed drawing, 3D142.

220 Volt direct current.

Hoist motor--Westinghouse No.8, Type KG, 25 H. P., @ 575 r. p. m., torque 2740 inch pounds.

Traverse motor--Westinghouse No. 4, Type K, 7½ H. P., @ 750 r. p. m.

Hoisting speed 14 feet per minute.

Traverse speed 130 feet per minute.

Eight parts of 3/4" rope.

Vertical hook movement 28 feet.

All gears are to be cast steel--pinions forged steel.

Shaw limit switch.

Fibre stresses in shafting not to exceed 9000 pounds per square inch, except sheave pins which may be stressed up to 12000 pounds per square inch.

Dynamic braking to be used instead of a mechanical load brake.

Westinghouse, type KT, magnetic brake for the hoist motor.

Cage to be attached to right hand end of bridge.

Master Car Builder's type of axle bearings.

JA

Hoist gears will be enclosed with No. 10 U. S. G. sheet steel guards.

The traverse gears will be enclosed with cast iron guards.

No overhung gears, motor pinions excepted.

Hoist Motor.

Following efficiencies will be used:

Total number parts of rope	Per cent efficiency			
	Lower block to drum	No. of gear reductions		
		First	Second	Third
8	87	80	74	68

$$H = \frac{H.\ P.\ x\ 33000\ x\ E}{P} = \frac{25\ x\ 33000\ x\ .68}{40000} = 14.02'$$

per minute hoist.

$$H.\ P. = \frac{H\ P}{33000\ E} = \frac{14.02\ x\ 40000}{33000\ x\ \ .68} = 25\ H.\ P.\ \text{required.}$$

$$R = \frac{L\ r}{E\ T} = \frac{10000\ x\ 10}{.68\ x\ 2740} = 53.5$$ required hoist ratio.

$$\frac{79}{23} \times \frac{84}{21} \times \frac{85}{23} = 51$$ Actual hoist ratio used.

Traverse Motor.

We will use 85 % efficiency for two reductions and one idler.

13 x .261 = 3.4' circumference of 13" wheel.

$R = \frac{C \times r.p.m.}{S} = \frac{3.4 \times 750}{130} = 19.6$ required traverse ratio.

$\frac{52}{14} \times \frac{83}{16} = 19.25$ actual traverse ratio used.

Estimated weight of trolley	13500 pounds
Live load	40000 pounds
Total load	53500 pounds

$T = \frac{G f d}{D E} = \frac{53500 \times .07 \times 3.25}{13 \times .85} = 1100$ pounds total tractive effort required.

$H.P. = \frac{T S}{33000} = \frac{1100 \times 130}{33000 \times} = \cancel{4.34}$ H. P. required.

Since the No. 3 frame is a 5 H. P. the next largest, a No. 4 frame, $7\frac{1}{2}$ H. P., will be used.

Hoist Gearing.

Live load on drum $= \frac{40000}{4} = 10000$ pounds.

P. D. of drum gear = 26.333"

P. D. of drum pinion = 7.666"

P. D. of intermediate gear = 24.000"

P. D. of intermediate pinion = 6.000"

P. D. of motor gear = 21.250"

P. D. of motor pinion = 5.750"

$\frac{79}{23} = 3.43$ Drum gear ratio.

$\frac{84}{21} = 4$ Intermediate gear ratio.

$\frac{85}{23} = 3.69$ Motor gear ratio.

Where the dead weights are comparatively small they will be neglected in calculating the strength of parts.

$\frac{10000 \times 20}{26.33 \times .80} = 9500$ pounds, tooth load on drum gears.

$\frac{575}{4 \times 3.69} = 39$ r. p. m. of drum pinion.

$39 \times .261 \times 7.66 = 78'$ P. m. , lineal velocity of drum pinion.

Value of factor "Y" (Lewis)

Teeth	20° Involute	Teeth	20° Involute
12	.078	19	.100
13	.083	20	.102
14	.088	21	.104
15	.092	23	.106
16	.094	25	..108
17	.096	27	.111
18	.098	30	.114

Allowable fibre stress in gear teeth (Lewis)

Material	Speed at pitch line in feet per minute					
	100	200	300	600	900	1200
Cast St.	12000	10500	9600	8000	6000	4800

Then by the Lewis formula for the strength of gear teeth.

$W = S\,P\,F\,Y$

$F = \frac{W}{S.\ P\ Y} = \frac{9500}{12000 \times 1.047 \times .1} = 7.58"$ say 7 3/4"

face of drum pinion and gear.

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$$\frac{9500 \times 7.66}{24 \times .92} = 3300$$ pounds teeth load on intermediate gears.

$$\frac{575}{3.69} = 156$$ revolutions P. M. of intermediate pinion.

$156 \times .261 \times 6 = 244'$ lineal velocity of intermediate pinion.

$$F = \frac{3300}{10000 \times .897 \times .1} = 3.68''$$ say 4" face of intermediate gears.

$$\frac{3300 \times 6}{21.25 \times .92} = 10\ 12$$ pounds teeth load on motor gears.

$575 \times .261 \times 5.75 = 863'$ p. m. lineal velocity of motor pinion.

$$F = \frac{1012}{6000 \times .785 \times .1} = 2.14''$$ say 2½" face of motor gear, pinion 2 5/8" face.

Traverse Gearing.

P. D. of driver gear = 13"

P. D. of driver pinion = 3.5"

P. D. of motor gear = 16.6"

P. D. of motor pinion = 3.2"

$$\frac{1100 \times 13}{13} = 1100$$ pounds teeth load on driver gears.

$$\frac{750 \times 16}{83} = 145$$ r. p. m., of driver pinion.

$145 \times .261 \times 3.5 = 132'$ p. m. lineal velocity of driver pinion.

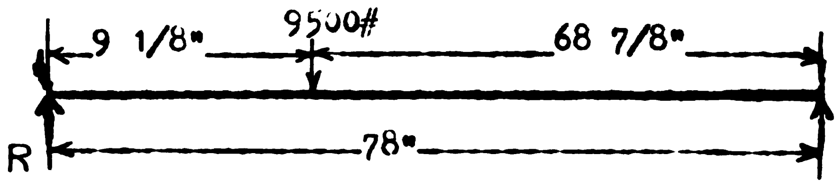
$$F = \frac{1100}{11000 \times .785 \times .088} = 1.45''$$ face required for driver gears. We will use 2½" face.

On general principles we will make the motor gear

2 1/8" face, pinion 2 1/4" face. Since this face is larger than the face required for the driver gears, further calculations will be unnecessary.

The increase of face is to take care of any excess loads which the motor may be called upon to carry. Furthermore, the actual faces required would appear weak and out of proportion for a machine of this size.

Drum Shaft.



Live load on drum 10000 pounds.

Drum gear keyed to drum thus eliminating all twisting on the drum shaft.

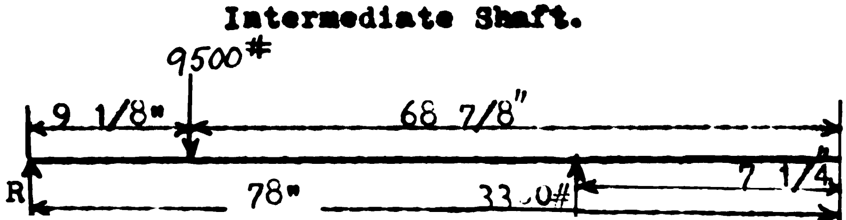
$R = \frac{9500 \times 68\ 7/8}{78} =$ 8380 pounds reaction on drum bearing due to gear load.

Since the hoisting ropes are central on the drum each bearing will take one half of the rope loads. Hence $8380 + 5000 = 13380$ pounds, total reaction on drum bearing.

We will take the lever arm 1/4" inside of drum hub, hence $4.25 \times 13380 = 57000$ inch pounds bending moment.

$\frac{57000}{9000} = 6.32$ section modulus required. This corresponds to a 4" shaft. We will use a drum shaft 4 1/8" diameter.

Intermediate Shaft.



$$R = \frac{9500 \times 68\ 7/8 - 3300 \times 7.25}{78} = \frac{654000 - 24000}{78}$$

R = 8700 pounds, reaction on drum pinion bearing.

8700 x 9.12 = 79300 inch pounds bending moment.

9500 x 3.83 = 36400 inch pounds twisting moment.

$\frac{36400}{9000}$ = 4.03 section modulus, which corresponds to a 2 3/4" shaft for twisting only.

$\frac{79300}{36400}$ = 2.18 = K----N = 1.66.

1.66 x 2.75 = 4.57" say 4 5/8" diameter of intermediate shaft.

Note:— All shafting computations involving bending and twisting are based on the formula

$$Te = B_m + \sqrt{B_m^2 + T_m^2}$$

Brake and Brake Shaft.

$\frac{79}{23} \times \frac{84}{21}$ = 13.75 Ratio between drum and brake.

$\frac{10000 \times 10 \times .74}{13.75}$ = 5380 inch pounds retarding torque for which the second shaft brake must be designed.

$\frac{5380}{7.5}$ = 715 pounds retarding force necessary on brake drum.

Let T = pull on tight side of brake band.

" t = pull on loose side of brake band.

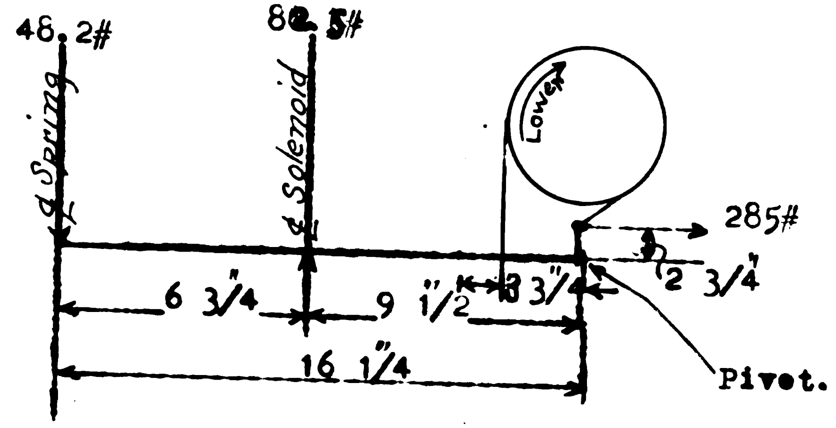
" x = angle enclosed by brake band, say 240°.

" f = coefficient of friction (.3).

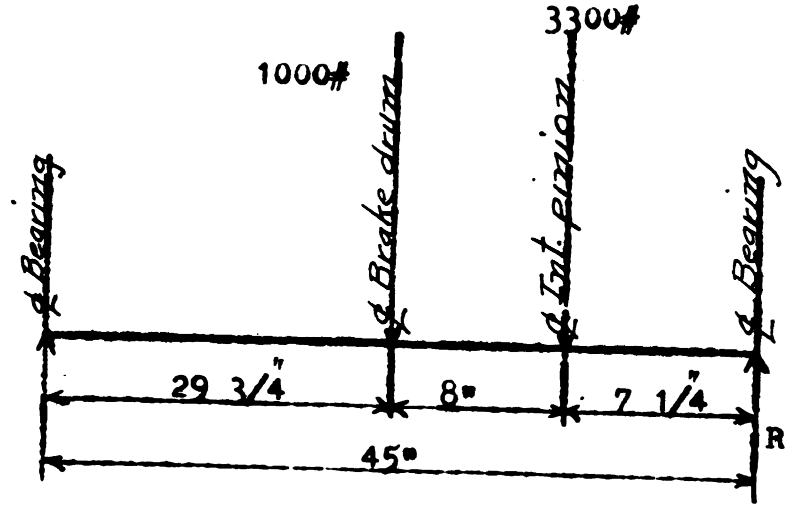
Then E^{fx} = $2.718^{.3 \times 4.188}$ = 3.514.

$$T = \frac{715 \times 3.514}{3.514 - 1} = 1000 \text{ pounds pull.}$$

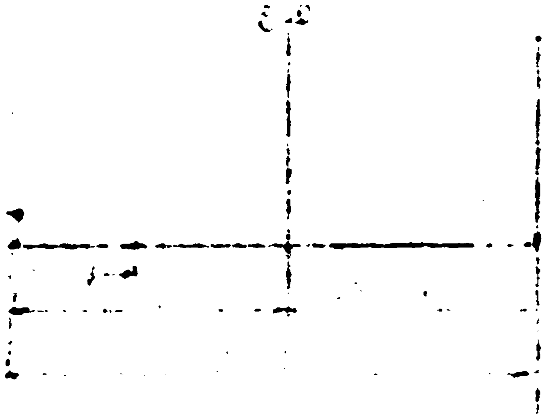
$$t = \frac{715}{3.514 - 1} = 285 \text{ pounds pull.}$$

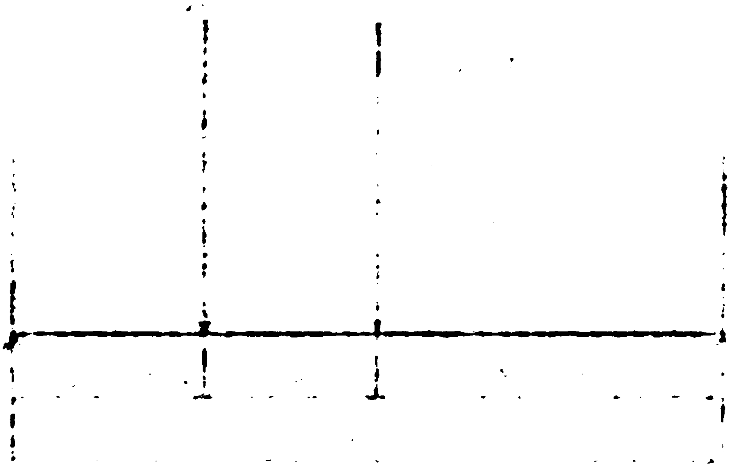


$\frac{285 \times 2.75}{16.25} = 48.2$ pounds, force to be exerted by spring neglecting weight of parts. In detailing the brake parts the effect of the weight of different parts must be considered and proper allowance made for the spring, and solenoid winding. Since the spring is adjustable the brake action can be varied through wide limits. It will be noted that the tight side of the band is fixed, thus making an economical and powerful brake.



We shall neglect the fact of continuous beam and





consider brake drum and intermediate pinion supported by only two bearings.

For practical purposes it will be sufficiently accurate to consider the resultant of the band pulls acting vertically and equal to T, or 1000 pounds.

$R = \dfrac{29.75 \times 1000 + 37.75 \times 3300}{45} = 3440$ pounds reaction on brake shaft bearing.

$7.25 \times 3440 = 25000$ inch pounds bending moment.

$3300 \times 3 = 9900$ inch pounds twisting moment.

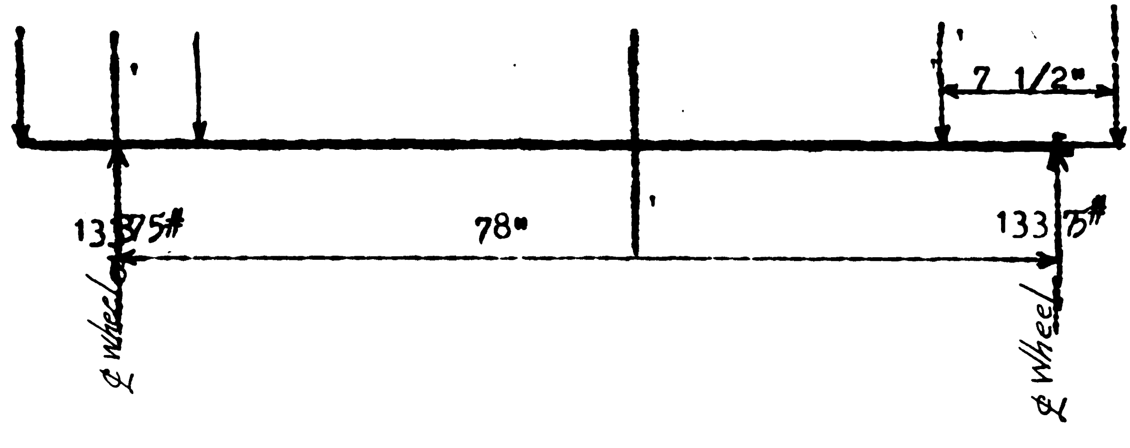
$\dfrac{9900}{9000} = 1.1$ section modulus, which corresponds to a 1 13/16" shaft for twisting only.

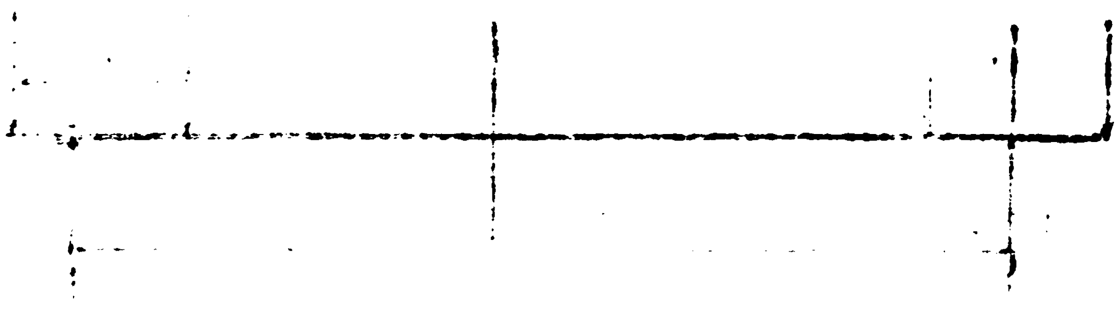
$\dfrac{25000}{9900} = 2.53 = K\text{—}N = 1.75.$

$1.75 \times 1\ 13/16 = 3.18$ say 3 3/16" diameter brake shaft required. We will use 3 1/4 diameter at this point which is practically determined by the back bearings on the motor, due allowances having been made for the assembling of parts.

Since it is apparent that this shaft meets all requirements, further calculations will be unnecessary.

Driver Axle.





We shall neglect the continuous beam feature.

Assuming that all wheels are equally loaded, we would have $\frac{40000 + 13500}{4}$ = 13375 pounds carried by each wheel.

Teeth load on driver pinion 1100 pounds.

The maximum moment occurs at the wheel and is equal to $\frac{13375 \times 7.5}{4}$ = 25000, inch pounds bending moment.

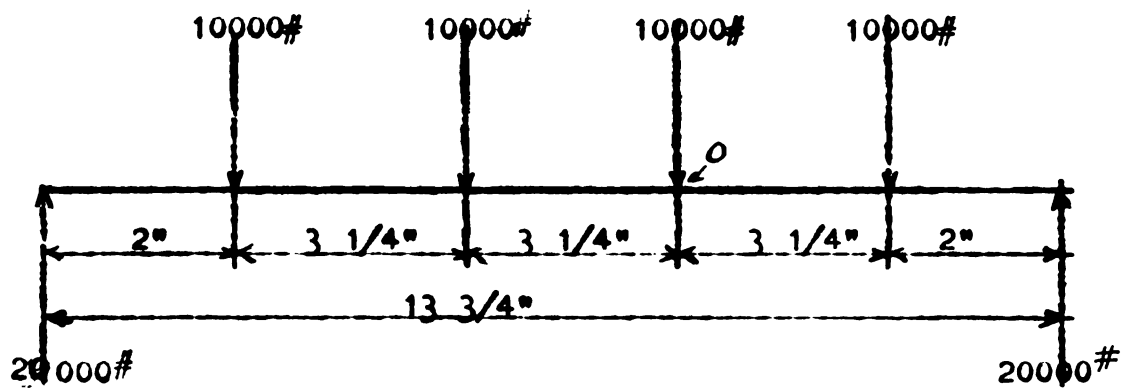
1100 x 6.5 = 7150 inch pounds twisting moment.

$\frac{7150}{9000}$ = .795 section modulus which corresponds to a 1 9/16" diameter shaft for twisting only.

$\frac{25000}{7150}$ = 3.49 = K — N = 1.93

1 9/16 x 1.93 = 3.01" say 3 1/4" diameter of axle.

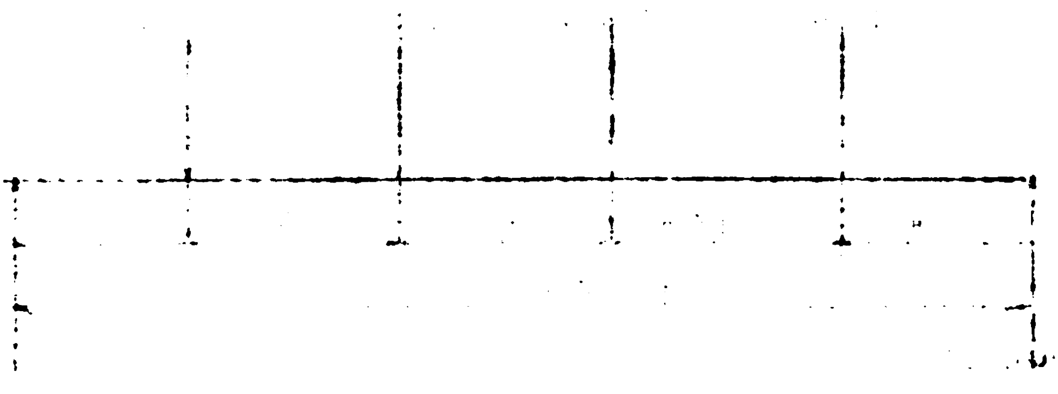
Upper and Lower Block Sheave Pins.



This diagram shows the loading of the lower block sheave pin.

Taking moments about O where the maximum moment occurs, we have

—10000 x 3.25 + 20000 x 5.25 = 72500 inch pounds bending moment.



We will use a 4" sheave pin.

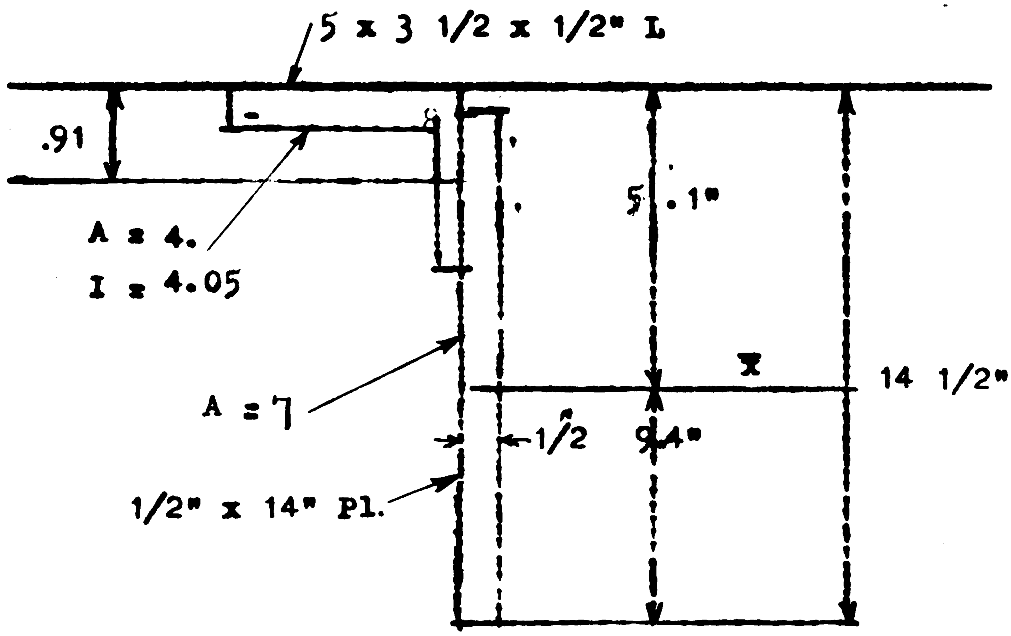
$\frac{72500}{6.4}$ = 11300 pounds per square inch on 4" sheave pin.

For the upper block pin we will also use 4" so that the sheave bushings for the upper and lower blocks will be interchangeable.

Since the upper pin is shorter and not loaded as heavily as the lower one, any calculations will be superfluous.

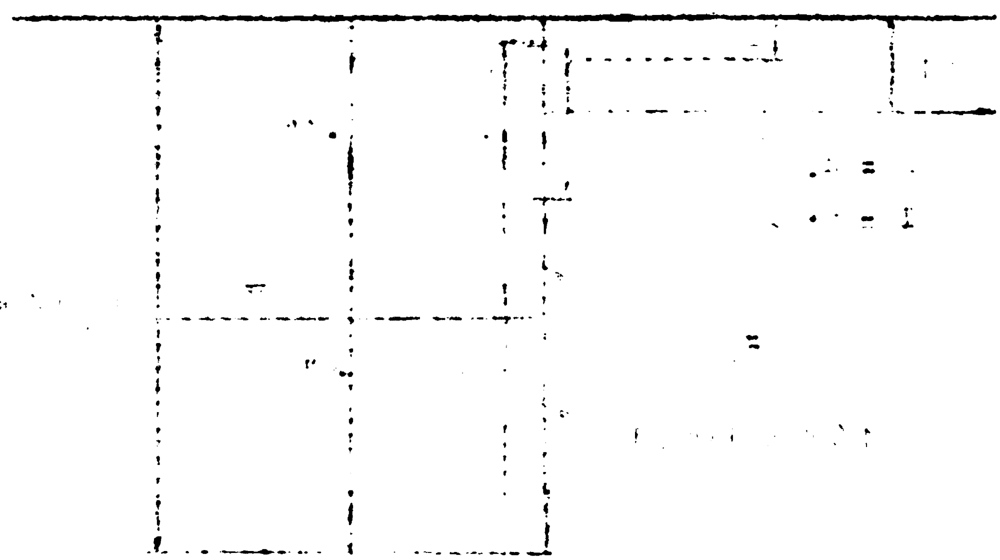
$\frac{10000}{3.25 \times 4}$ = 768 pounds per square inch. bearing pressure on sheaves.

Upper Block Girt.



$$\bar{X} = \frac{7 \times 7 + 4 \times 13.59}{7 + 4} = 9.4''$$

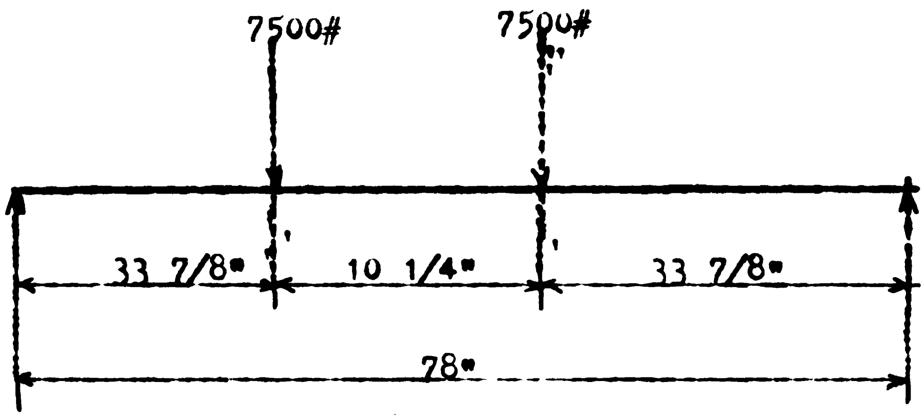
$$I_x = \frac{1}{12} \times \frac{1}{2} \times 14^3 + 7 \times 2.4^2 + 4.05 + 4 \times 4.19^2$$



$I_x = 114.2 + 40.25 + 4.05 + 70 = 228.50.$

$\frac{228.50}{5.1} = 44.3$ compression modulus.

$\frac{228.50}{9.4} = 24.3$ tension modulus.



The vertical loading of one half of the upper block girt is as shown above.

7500 x 33 7/8 = 254000 inch pounds maximum bending moment.

$\frac{254000}{44.3} = 5740$ pounds per square inch compression.

$\frac{254000}{24.3} = 10500$ pounds per square inch tension.

The compression is kept low so that the angles are enabled to resist any normal lateral forces they may be called upon to carry. To take care of extreme lateral forces we will anchor the lead girt to the machinery girt by means of a tension bolt, as shown on the drawing. This bolt will be fixed in the machinery girt, but free in a slot in the lead girt, so that deflection by the lead girt will place no load on the bolt.

20% of the vertical live load is a conservative estimate of the lateral force that the lead girt must be

able to resist, hence

$$20\% \text{ of } 30000 = 6000 \text{ pounds.}$$

At about 10000 pounds per square inch tension this will require a 1" bolt. Any excessive forces in the opposite direction will be transmitted to the machinery girt through the interposed washer.

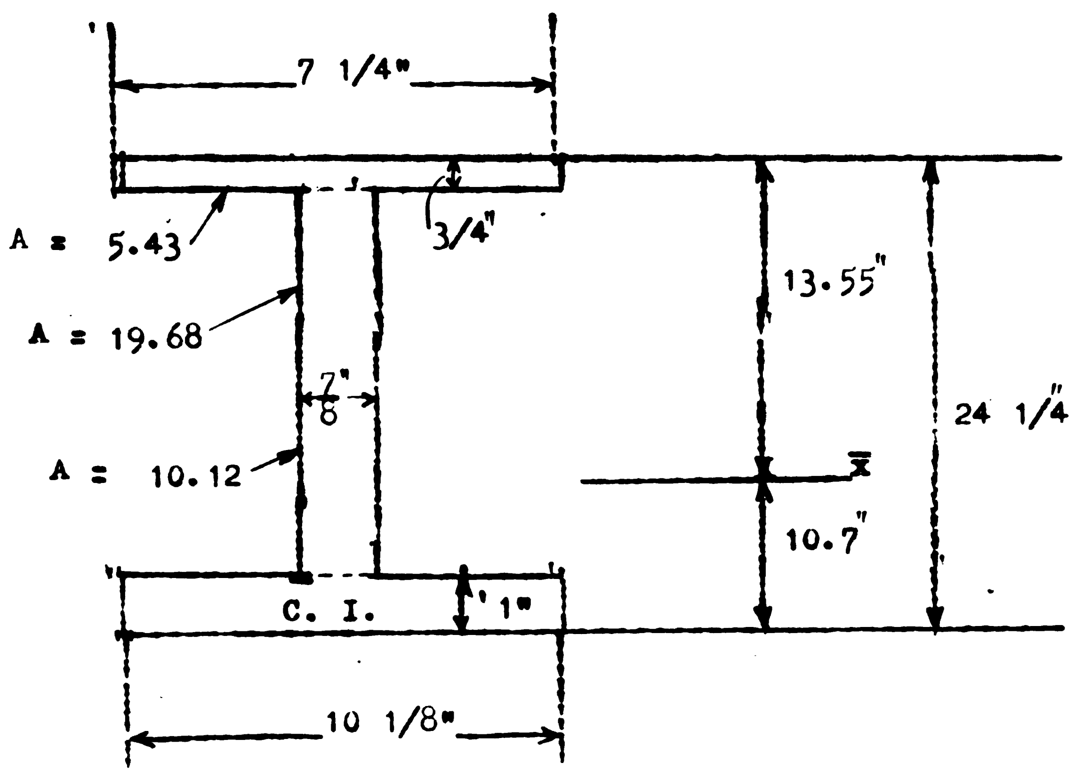
$$\frac{L}{r} = \frac{\frac{78}{2}}{.75} = 52.$$

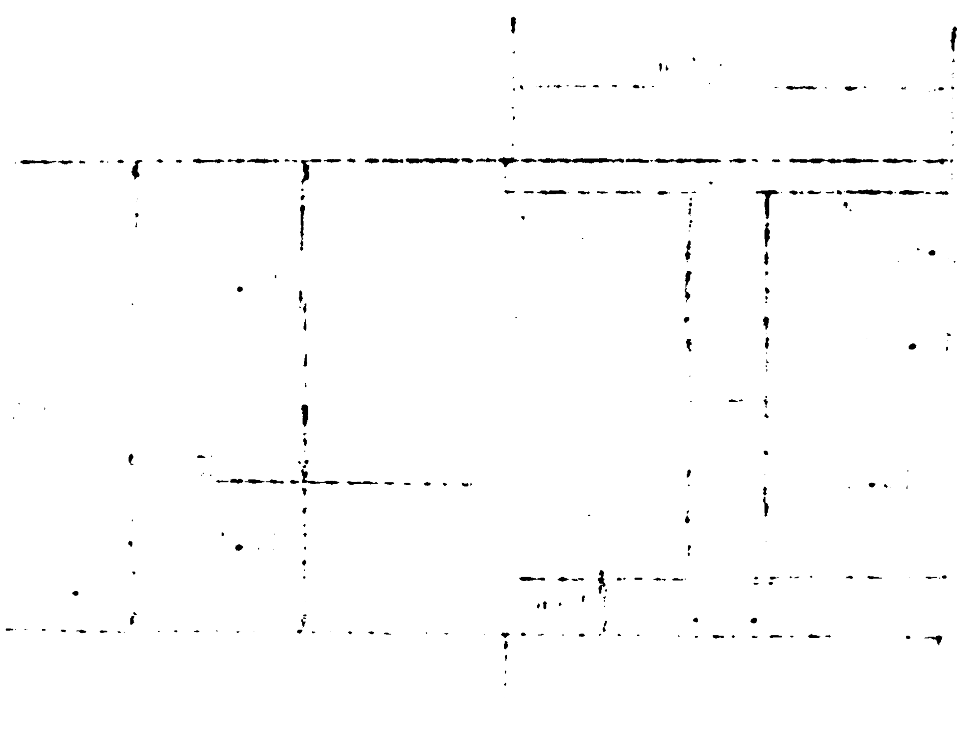
According to Gordon's formula the allowable compressive stress for a slenderness ratio of 52 is as follows:

$$\frac{12500}{1 + \frac{39^2}{36000 \times (.75)^2}} = 11600 \text{ pounds per square inch.}$$

This would indicate that the girt will not buckle under the combined vertical and lateral forces, the compressive stress due to vertical load being 5740 pounds per square inch.

Trolley Side Section.





$$\bar{X} = \frac{10.12\times1/2 + 19.68\times12.25 + 5.43\times23.87}{10.12 + 19.68 + 5.43}$$

$$\bar{X} = \frac{5.06 + 241 + 130}{35.23} = \frac{376.06}{35.23} = 10.7''$$

$$I_x = 10.12\times10.2^2 + 1/12\times7/8\times22.5^3 + 19.68\times1.55^2 + 5.43\times13.18^2$$

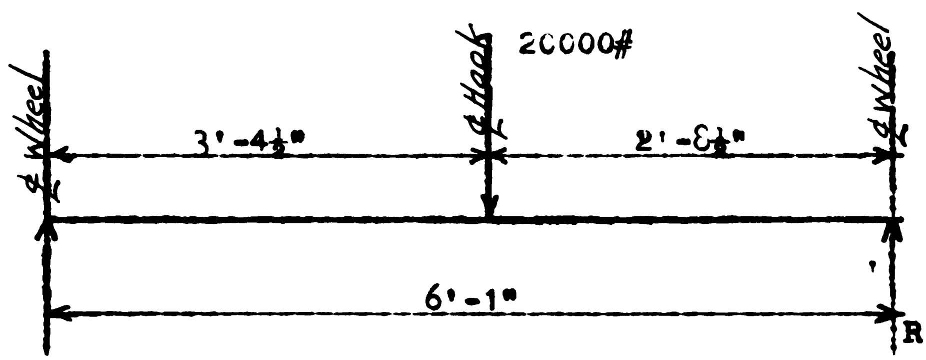
$$I_x = 1050+832+47+942 = 2871$$

The inertia of the upper and lower flanges about their own axis has been neglected since this is very small.

$\frac{2871}{13.55}$ = 212 compression modulus.

$\frac{2871}{10.7}$ = 268 tension modulus.

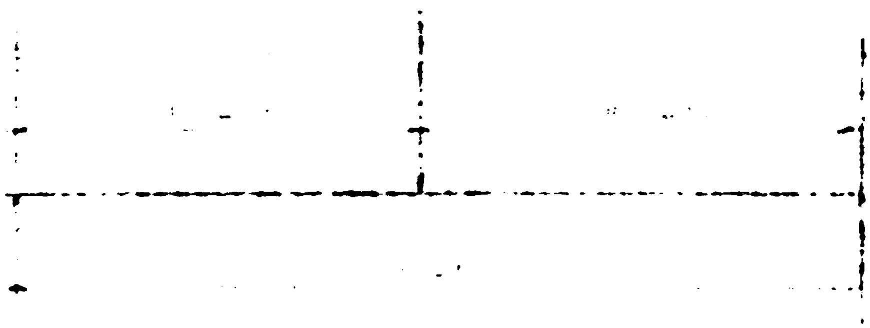
We will now take moments to determine the maximum moment on each of the trolley sides.



R = $\frac{40.5\times20000}{73}$ = 11090 pounds maximum load carried by one wheel.

11090x32.5 = 361000 inch pounds bending moment due to live load.

The dead load will be considered as uniformly dis-



tributed, hence

$$\frac{13500 \times 73}{2 \times 8} = 61600$$ inch pounds bending moment due to dead load.

$361000 + 61600 = 422600$ inch pounds, total bending moment on one trolley side.

$$\frac{422600}{212} = 2000$$ pounds per square inch compression.

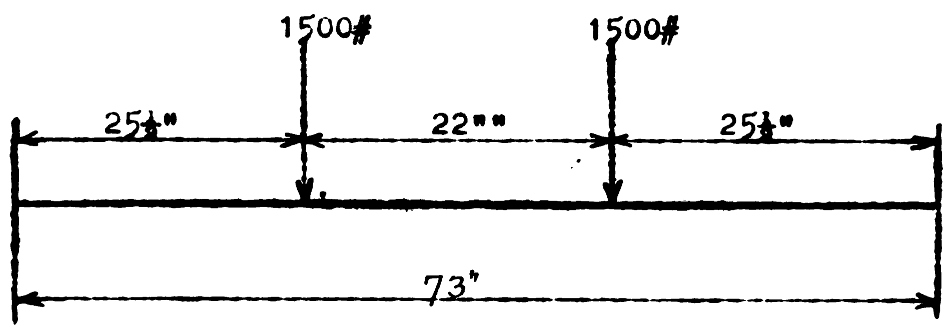
$$\frac{422600}{268} = 1580$$ pounds per square inch tension.

We shall assume that the trolley sides may be called upon to carry a lateral load of 20% of the live load on the load girt.

$20\% \times 3000 = 6000''$ pounds.

$$\frac{6000}{2} = 3000$$ pounds lateral force per trolley side.

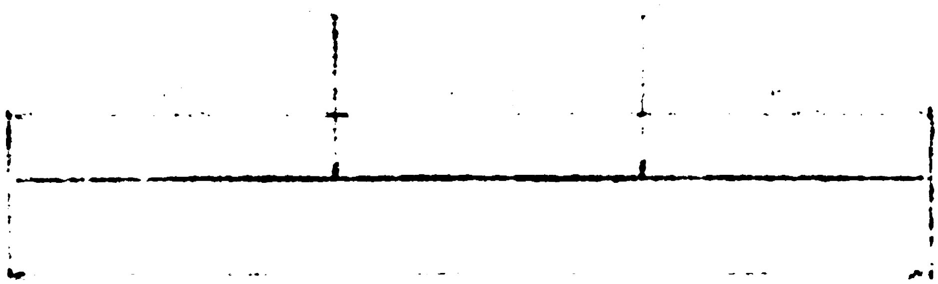
For convenience we shall assume this load as distributed equidistant between the wheels, thus



$1500 \times 25.5 = 38300$ inch pounds lateral bending moment.

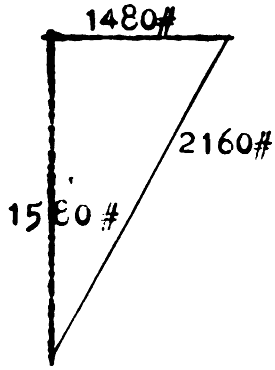
It is apparent that a part of this bending moment is transmitted to the upper flange, we will say 1/3. Therefore

$38300 \times 2/3 = 25533$ inch pounds lateral bending moment



on lower flange.

$$\frac{2533}{\frac{1\times 10\ 1/8^2}{6}} = 1480^{\#}$$ lateral tension on lower flange.



Combining the vertical and horizontal tensions, as shown, we get a maximum tensile stress in the lower flange of 2160 pounds per square inch, which is satisfactory.

Bearing Pressures.

The sheave bearing presures have been determined under a previous heading.

We have found the maximum live load on one wheel to be 11090 pounds. Assuming that the dead load is equally distributed over the four wheels, we have

$\frac{13500}{4} = 3375$ pounds per wheel.

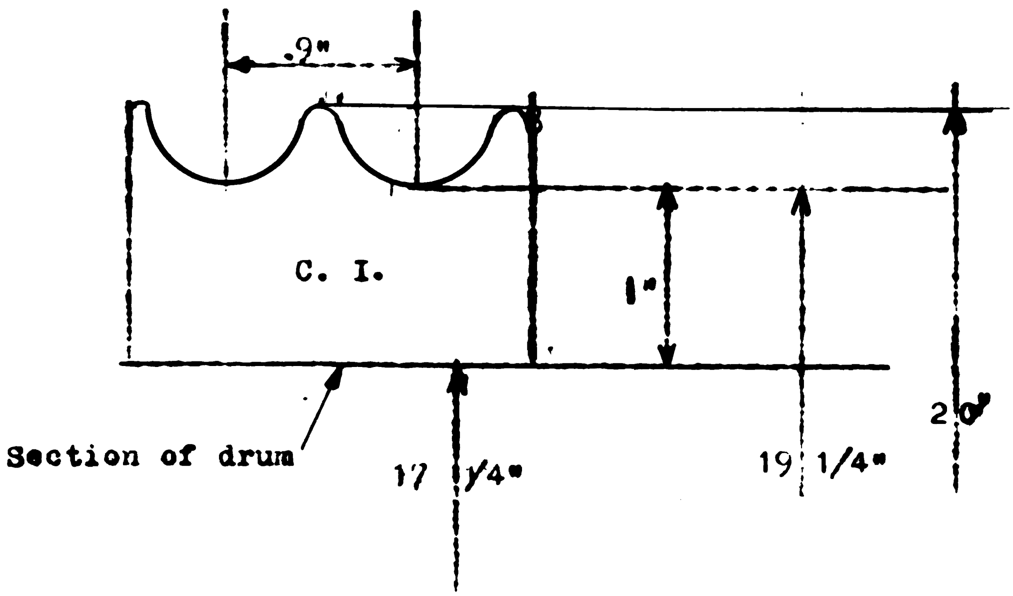
$11090+3375 = 14465$ pounds, maximum wheel load. Hence the maximum bearing pressure is

$\frac{14465}{2\times 3.75\times 3.25} = 593$ pounds per square inch.

It is apparent that the bearing pressures in the hoisting train are low. They are both safe and practicable

and hence further calculations in that direction will be superfluous.

Rope Drum.



$S = \frac{D^4 - d^4}{10d} = \frac{19\ 1/4^4 - 17\ 1/4^4}{10 \times 17\ 1/4} = \frac{138000 - 88500}{172.5} = 287$ section modulus of drum.

$\frac{10000 \times 70}{4} = 175000$ inch pounds, bending moment on drum.

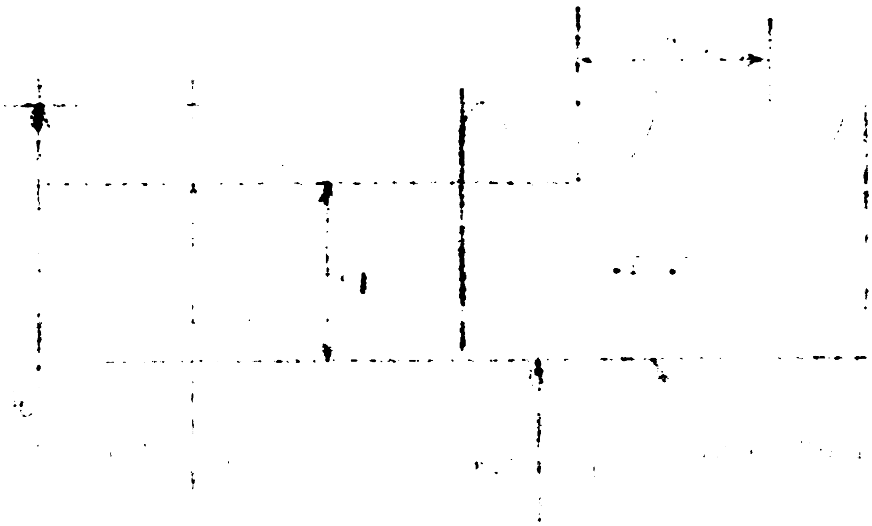
$B = \frac{Bm}{S} = \frac{175000}{287} = 610$ pounds per square inch bending.

$C = \frac{P}{pt} = \frac{5000}{.9 \times 1} = 5555$ pounds per square inch compression.

$\sqrt{B^2 + C^2} = \sqrt{610^2 + 5555^2} = 5600$ pounds per square inch combined tension and compression.

The wire rope will be extra flexible plow steel, six strands, thirty-seven wires to the strand, hemp core, having an ultimate capacity of 42000 pounds.

$\frac{42000}{5000} = 8.4$ factor of safety in rope, neglecting dead weight.



This facter of safety is a little high, since, however, the 3/4" repe is the nearest commercial size answering our requirements it has been selected.

The drum has a total of twenty-four grooves per side. We will allow two turns for wrap and stretch of repe, leaving twenty-two greoves available for hoisting.

Circumference of drum in feet is .261x20 = 5.22.

Hoist per turn $= \frac{5.22}{4} = 1.30$ feet.

1.3x22 = 26.6 feet hoist, which meets our requirements.

Pocket ms: 1 Plan

[redacted]