THREE DIMENSIONAL LIFEBOAT LAUNCHING

A Thesis Presented for
the Degree of Bachelor
of Science in Naval
Architecture and Marine
Engineering

By

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Class of 1935

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
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Mass. Institute of Technology
Cambridge, Massachusetts
May 16, 1935

Professor James R. Jack
Mass. Institute of Technology
Cambridge, Massachusetts

Dear Sir:

In accordance with the requirements in application for the degree of Bachelor of Science in Naval Architecture and Marine Engineering, we are presenting a thesis which we have truly entitled "Three Dimensional Lifeboat Launching."

We wish to call your attention to the radical change in method of launching whinh the title indicates, the complications of design which it involves, and the difficulties in lifeboat launching which it overcomes.

Though we claim no credit for the fundemental idea, the three dimensional principle, the design, when taken up as thesis work, had no foundation but its fundemental idea and the principles of engineering mechaniss. The degree of fineness to which the design is carried was subject to restriction due to limited time, though the attempt was made to complete all working plans.

Sincerely yours,

ACKNOWLEDGEMENTS

We are grateful to the following men for aid given during the preparation of this thesis:

To

Mr. F. Forrest Pease for the fundamental principle on which the thesis is based,

Mr. C.B. Palmer for making available the clippings from the Boston Transcript files on Marine disasters,

Professor Evers Burtner for miscellaneous technical information.

A.L.H.

R.P.I.

INTRODUCTION

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PURPOSE. The purpose of this thesis is to introduce to the naval architectural world the value, principle, and design of the Pease Lifeboat Launching Crane.

METHOD PERSUED. Herein is briefly outlined the method used in the working out of the design. A more detailed explanation is found in the section under Design.

Before starting the actual design work, it was necessary to determine how much had been done before us. This proved to be nil. Therefore, to have a practical basis , a thorough perusal of all clippings from the Boston Transcript files on marine disasters was undertaken. From these were taken the faults of the present equipment which were listed as factors to be eliminated in the new design.

As this type of lifeboat launching equipment cannot be a stock unit, but must be custom built, it next remained to choose a boat for which to design the apparatus. The S.S. Colombia, a Ferris design, was chosen both because of its common type and because of the adaptability of the apparatus to it.

In order to give the initiate to this new principle a clear idea of how it works, a drawing was made of the mid-ship portion of the Colombia, showing the launching of a lifeboat in three views, accompanied by a fourth supplementary diagram. (See Plate I)

Due to the extreme complications in direction of cables and arms, it was considered advisable to draw in four views

the complete motion of a single arm, and as far as possible, obtain all data graphically from these drawings (See Plate II). This plan was carried out in obtaining all the forces and moments inthe arms and cables.

The bending moments were taken or resolved into two perpendicular planes and combined.into a resultant bending moment, which was in turn combined with the twisting moment by Rankine's theory. The result, along with the total compression and an assumed fiber stress, was used to get the varying diameters of the arms.

United States Standard tables in Mark's handbook were used to estimate the size of cables.

HISTORY

HISTORY

The idea behind the Pease Lifeboat Launching Crane is not a brainstorm brought on by recent marine disasters. It followed six years of sea experience from 1909 to 1915. In the latter year the idea was presented to leaders in marine fields and received considerable encouragement. Dr. Alfred Williams Anthony of Lewiston, Maine was desirous of backing it and had the money to do so. To make sure of the ground on which he trod, Dr. Anthony requested Professor Peabody of the Massachusetts Institute of Technology to look into the idea. Professor Peabody's report showed the idea to be sound and extremely worthy of development.

With Mr. Pease as engineer and Dr. Anthony as manager, the idea grew. A steamer at Portland was chosen as a suitable subject, and designs were started for installation when Mr. Pease was forced to abandon the work due to illness.

By 1916, Mr. Pease was again on the trail of a backer for the idea. This led him to the Fore River plant of the Bethlehem Steel Corporation. Rush of war construction prevented any immediate action on the lifeboat launching crane, but Mr. Pease was taken on as superintendant of apprentices in the yard. Soon after, the United States Shipping Board ordered him to Philadelphia, not to return until after the War.

In 1921, Fore River again attacked the lifeboat launching problem. The technical staff was instructed to examine the idea, and considerable amount of study and preliminary plans were made. The investigation showed that a great deal of money was needed for experimentation, and the management, though

realizing the value of the idea, decided that it didn't care to go into the development.

From then on,,the idea bounced from shipyard to ship owner, from ship owner to Steamboat Inspection Service, and from Steamboat Inspection Service back again. The shipyard couldn't develop the apparatus for lack of money. They were willing to install one should the owner desire it, but only under that condition could they see the value to themselves. The shipowner was satisfied with the existing equipment as long as it was passed by the Steamboat Inspection Service, and the latter had no money or authority to do the developing. They said that they would gladly pass judgement on the apparatus if it were installed an some ship.

This leaves only two sources of backing for the idea, the government and a private investor. The latter is looking for a minimum of risk with a maximum of return, and the very nature of the apparatus fails in this qualification.

At present, the government is extremely interested. Recently, Mr. Pease demonstrated his model and showed moving pictures of this model launching model-lifeboats into an experimental tank of the Massachusetts Institute of Technology before the members of the United States Steamboat Inspection Service. They were greatly impressed, and Mr. Pease now awaits a report from the chairman, Mr. Dickerson Hoover, as to the success they have had toward getting appropriations for the project. Assisting Mr. Pease in Washington is Congressman Wigglesworth who is an enthusiastic advocate. In addition, the newspapers have been big aids in helping the idea to gain the approval of the public and professional men which it now has.

APPLICATION

APPLICATION

The Court of Inquiry Report on the Titanic disaster, made in August 1912, stated that at sea there was but one death in every 800,000 passengers carried. This does not impress us with anything but its insignificance as a figure, but its insignificance is confined to the field of mathematics and does not trespass on the human side of the question. When nine out of ten of these deaths are due to faulty machinery or inadequate machinery, it remains for the engineer to answer the cry of the indignant public.

The public cry is naturally heard the loudest immediately following a spectacular disaster. It rose in volume following the sinking of the Titanic in April 1912. Two years later, the collision of the Empress of Ireland and a Danish tramp with a consequent loss of life of more than 1000 persons gave rise to another public outburst. In May 1915, it was the sinking of the Luisitania, in 1928 it was the Vestris, today it is the recent Ward Line calamities that have roused humanity.

As engineers, we are interested in the lesson from each of these calamities. From the Titanic disaster, we learn that it is perfectly feasible to launch all boats safely under perfect conditions with the ordinary boat davits. This is important as a contrast to the success of the operation under distress conditions. When the Titanic sank, the sea was perfectly calm and the ship sank by the head without listing and slow enough to give everyone plenty of time to get away in the boats. The greatest reason for loss of life in this case was the lack of sufficient lifeboats and reluc-

tance of passengers in leaving what they were lead to believe was an unsinkable ship. Excellent conditions are abnormal rather than normal, and as we are looking for difficulties to overcome, the sinking of the Titanic is not of such importance.

Though no detailed information is available on the fate ofthe Empress of Ireland's complement, it is suspected that this disaster can be classed with the Titanic.

When the Luisitania was torpedoed off the Irish coast, the American public was too busy directing its indignation toward the Imperial German Government to ask why, when given 15 minutes to abandon ship, nearly 1200 persons found no means of safety at hand. Here we learn the lesson of speed and efficiency that is not associated with the age old block and tackle principle used in existing lifeboat davits.

In the sinking of the Vestris off the Virginia Capes in 1928, we learn several things about lifeboat launching. The weather was extremely bad. The ship had a list of about 15° as the boats were being lowered. The Vestris carried 14 boats, plenty to accomodate all 328 persons aboard from either side of the ship. The following table, showing what happened to each boat, gives an impressive picture of what this thesis is trying to eliminate.

-Port side

Boat #2 - Stayed on deck and sank with ship.

#4 and #6 - Lowered part way with women and children but never launched, and went down with ship.

#8 - Launched but sank because of damage in launching.

#10 - Successfully launched with all passengers and no crew. Tackle chopped off block because of entanglement.

#12 - Remained on deck and sank with ship.

#14 - Remained on deck and floated off as ship sank and remained afloat.

-Starboard side

Boat #1,3,5,7,11 - Successfully launched.

#9 - One end lowered before the other and sank.

#13 - Floated off deck and remained afloat.

Out of the total of 14 boats, 8 got away safely with 218 people. Of these 8, two were never launched but floated off the ship empty as it sank, and still another had to be chopped loose from the blocks as it hung loaded from davits, and was thus prevented from beigg dragged down with the ship.

The port side was the windward and high side. The effect of list is plainly shown by the table, but even on the low side, two of the boats failed to get away. It took two hours to launch the boats on the high side.

Trouble with the releasing gear was shown in a statement by the chief engineer, Mr. Johnson. He said that the releasing equipment broke in two instances and that there was no instance when it operated successfully.

Now we come to the present, the recent Ward Line calamities, of which the burning of the Morro Castle is the most striking. Out of 12 lifeboats, each with a capacity of from 50 to 80 persons, only 8 got away, some of them carrying as few as six people, mostly crew. We find here that the windward side of a burning ship is the only possible side from which to launch lifeboats. When tremendous seas hurl their force from that quarter, the problem of boat launching becomes exceedingly difficult.

These few major disasters cause but a small part of the

toll of deaths at sea, however. They are the ones we read about in the newspapers, but we do not look behind the scenes of smaller incidents of the same sort where numbers of persons are lost in one sweep. Captain Fried lost two men and five boats attempting to rescue the crew of the Antinoe in January 1926. He stood about 80 hours before daring to launch a single boat. When the Monroe and Nantucket crashed in January 1914, 49 men drowned because ten minutes were not sufficient to launch the lifeboats. A whole fleet of ships, standing by the Volturno in Mid-ocean all day, October 13, 1913, watching her burn to the water's edge, unable because of the high seas to be of any assistance. 136 lives were lost, all of which were lost in attempts to get away in the ship's boats.

And so on through volumes and volumes with always the same story -- "calm water enables all boats to get away safely," or -- "heavy sea conditions prevented proper manipulation of launching equipment." In other words, the design of launching equipment should include all the factors possible for the elimination of each of the defects in present davits. The most important of these factors are listed below:

1. Long outreach to enable launching on windward side and against a list.
2. Single motion with one man operation for efficiency and speed in operation.
3. Launching under motion for immediate steerage way.
4. Capacity to handle large boats for greater seaworthiness and for greater confidence from the passengers. (This also enables more boats to be carried on each

side of the ship where desired.)

5. No rigid fastenings to lifeboats for immediate and synchronous release when waterborne, and also to enable boat to float free of sinking ship were it not launched.

All of these factors, along with convenience, beauty, and economy have been considered in the following design, though compromises were inevitable in some instances.

DESIGN

DESIGN

THEORY. The motion described by the Pease Lifeboat Launching Crane can be outlined very simply. First, consider as a horizontal plane, the plane of rotation (Plate II), with a line perpendicular to it. Around this line, rotating in a semi-circle are two other lines in a plane perpendicular to the plane of rotation which meet the latter plane in the same point, one of which intersects the plane at an angle of 16° and the other at about 40°. The 16° line is a steel arm, and its angle with the plane of rotation is termed the "droop" of the arm. The 40° line is a fixed length cable which holds the arm from drooping further. Now, take this whole unit and place it alongside a vessel with the plane of rotation perpendicular to the side, tip it up so that the plane of rotation makes about 45° with the surface of the water, and we have the motion described by one of the three arms of the apparatus. Three of these units with parallel planes of rotation, and with the outer ends of the arms rigidly connected, make up the launching device. If the description has been followed up to this point, the picture now is that of a parallel ruler in three dimensions, where the rules are the ship and the rigid arm connection (Fig. 3, Plate I). The latter is the cradle in which the boat rests.

To control the motion of the apparatus, three operating cables in the planes of rotation are fastened from the outer end of the arms through sheaves and around a drum on deck. The drum is operated by a winch to return the apparatus after launching a boat. It is also fitted with a hand or centrifugal

brake to check the boat in launching.

To keep the cradle upright, nothing more than three parallel pins is necessary. These are perpendicular to the plane of rotation and pin the arms to the cradles. They are the outer swivels of the parallel ruler.

Instead of using pins at the ship's side, a universal joint is employed to allow the cradle to rise straight up with a heavy sea in launching rather than twist off the pins.

CHOICE OF BOAT. The S.S. Colombia was chosen as subject for the design of the equipment for several reasons. She has a small enough complement of passengers and crew to enable all to be accomodated in a single large lifeboat. Her freeboard is not so extreme but that the arms can be conveniently butted and swivelled at the base of the superstructure. This allows the arms to lie inside the outermost beam of the vessel and prevents damaging from other boats or from a wharf. Also, the farther from the load line that the arms are swivelled the longer they must be to properly place the boats on the water.

GENERAL LAYOUT. It was first desired to keep the arms for this installation perfectly straight and about 40 feet long. However, it was found that straight arms would necessitate cutting into all three of the upper decks to get the boats inboard. To eliminate this trouble the arms were curved for about 8 feet of their length so that only the boat deck had to be cut away under the lifeboat.

Were the arms limited to 40 feet in length, the angle of launching would have been excessive. Considering 45°

as a reasonable angle at which to launch, the arms were drawn to about 47 feet in length.

The droop of the arms is determined by the height of the arm butt at the ship from the load line. The higher this point is, the more droop is necessary in order to have the boat waterborne at the outermost point of the curve. A slight compromise was necessary here in that if the boat were waterborne at exactly the outermost point, an arm 60 feet long would be necessary, other factors remaining the same. The boat actually floats off the cradle at about 10° past position #4 with the 47 foot arms.

In order to keep the twisting moments inthe arm at a minimum, and angle of 45° was constructed between the plane of the curved arm and the ship's side in #1 position (See Plate II). This causes the plane of the curved arm to be perpendicular to the water in the waterborne position.

MOTION. Plate II, showing the motion of a single arm, explains itself, though it is rather complicated descriptive geometry. All N-B lines represent the fixed length cables; all O-B-A lines, the arms; all B-B lines (Fig. 4 only), the operating cables. N-O shows the slant of the cradle pins and the center-line of rotation.

LOAD ON ARMS. There are three sources of load on the arms: a static load, a dynamic load due to rolling of the ship, and a dynamic load due to checking the boat before reaching the water. These three were calculated for the worst possible conditions and a resultant vertical load of 6.41 tons was determined. The load was assumed constant, though it actually was slightly less at the extremities of

the motion. Several assumptions were made for the calculations on the dynamic forces, but the forces themselves are so small in relation to the resultant vertical load that variations in these assumtions have little effect on the result.

TENSION IN CABLES. Plate III shows the settup for obtaining the tension in the fixed length cables. It is simply a case of taking moments around the butt of the arm in position #1. The tension in these cables is constant.

The stress diagram for finding the tension in the operating cables is shown in Plate IV. This is a settup of moments in the plane of rotation, and due to the changing angle between the cable and the arm, this cable varies in tension. #7 position was eliminated at this stage because of an infinite tension in the operating cables. Stops would prevent the arms from reaching further than #6 position.

BENDING MOMENTS. With all the forces known, the next step was to figure the bending moments due to each force for different positions along the arm and for each position. This was done so that all moments obtained lay either in the horizontal or vertical plane. (The horizontal plane here referred to is the plane of rotation.) A plot of vertical and horizontal bending moments was made and a resultant bending moment diagram obtained by taking the square root of the sum of the squares of the vertical and horizontal ordinates. The force diagram used is shown by a representative sketch in Plate IV, and also in Plate III.

TWISTING MOMENTS. Due to the curve in the arm, there is a varying twisting moment set up by the load on the end. Figure 3, Plate II, shows the true arm for the twisting moment, but it was necessary to construct the auxilary

diagram to get the angles at which the load acts.

COMPRESSION IN ARMS. In Plate II, it can be seen that the triangles O-B-B are isoceles in every position. We know the length of the arm to the cable fastening and can measure the length of the cable. With these we can find the base angles and therefor the compression in the arm due to these cables.

Straightforward resolution of the tension in the fixed length cables along the arm in Plate III gives the compression from this source. An auxiliary diagram was used to obtain the compression and tension due to the load. Algebraic addition of the three forces gave the resultant compression.

DIAMETER OF ARMS. Rankine's formula for combining twisting and bending moments was used to obtain an equivalent bending moment.

$$Eq = 1/2\ M_B + 1/2 \sqrt{M_B^2 + M_T^2}$$

The theoretical value given by this equation is that simple bending moment which, if applied to a beam at the point in question, will give a fiber stress equal to that due to twisting and bending both. It was calculated for every point of probable maximum as shown by the bending moment diagrams.

Assuming a steel of yield point around 60,000 pounds per square inch and a safety factor of two, the diameter of the arm for every 15 feet of its length was calculated, using the maximum equivalent bending moment. The results are shown in Plate V.

SIZE OF CABLES. It is noted that the tension in the fixed length cables is 11.8 tons and in the operating cables, 12.1 tons. This would call for about the same size cables, but

but the fixed length cables must partially support the lifeboat as it is stowed and are not wound on any drum. For these reasons, a heat-treated steel cable is used.

The operating cables, on the other hand, take no load until the boat is being launched and must be flexible enough to wind on the winch drum. This calls for an annealed steel cable.

RESULTS

RESULTS

1. Details of Arm.

    a. Hollow steel, Y.P. 60,000 lbs./sq. in.

    b. See Plate V.

2. Fixed Length Cables.

    a. Plow steel, heat treated.

    b. Carbon content, .90.

    c. Diameter, 1 inch.

    d. Ultimate strength, 50 tons.

    e. Safety factor, 4. (Based on ultimate strength.)

3. Operating Cables.

    a. Monitor plow steel, annealed.

    b. Carbon content, .82.

    c. Diameter, 1 inch.

    d. Ultimate strength, 50 tons.

    e. Safety factor, 4. (Based on ultimate strength.)

CONCLUSIONS

CONCLUSIONS

The three factors for presentation by this thesis are, as stated in the purpose, the value, principle, and design of the Pease Lifeboat Launching Crane. It was essential to cover the first two thoroughly before attempting the last. As a result, the last was left dangling near the end.

However, the hard part of the design is done, the part that takes the "settin' and thinkin' ". With the arms defined both in size and motion, it is but a routine machine designer's job to lay out drawings for fittings, and to make up a suitable cradle.

With no basis for comparison, it is hard to define the limits of reason, but the dimensions as calculated for arms and cables seem to lie within this range despite the lack of any parent design from which to work. Where so many considerations are necessary, mistakes or omissions are not impossible either in assumptions or calculations.

There are many special considerations. One is a ram on the promenade deck at the position of the buffer as shown in Plate II to force the apparatus outboard against a list, and to hold it there in spite of the ship's rolling. Another is a catch device to take the load as the boats are stowed. In other words, there are any number of refinements that are out of place in this pioneering paper, but will develop with the idea.

It remains that the three dimensional principle applied to lifeboat launching has all the advantages necessary to

Supplementary Conclusions

In addition to the conclusions above, there are various considerations as to application and operations. To install this apparatus aboard a ship of the "Morro Castle" type with her extremely high sides, it would be necessary to build the arms down on her shell plates. This would be out of the question as the apparatus would be very easily damaged by a ship coming alongside or even by a dock when the ship comes to port. A solution for this problem might be to set these arms into the hull. This would entail breaking the contenuity of the ships strength members. When starting from scratch this could be taken into consideration by indenting the hull at the place where the apparatus is going to be enstalled. It is obvious to the naval architect what an added expense this would be. To take the life boat off a deck as high as the boat deck of a modern transoceanic liner would be highly improbable, as the arms would be in vicinity of one hundred feet in length and about two feet in diameter. Of course a more accessible place for the life boats would be on a lower deck; if this could be accomplished the lifeboat could easily carry out its mission. But even so, if the boats were fifty feet above the water, an arm of about seventy feet in length and eighteen inches in diameter would be necessary. This however is not out of proportion to a lifeboat sixty feet in length and having a capacity of approximately four hundred persons.

A curved arm was designed in the thesis so as to eliminate cutting into the decks above the butt of the arms. It added a few complications to the design, but we feel that its superiority over the straight arm is so great as to make it almost imperitive. True, the straight arm would have simple bending, tension, and compression to withstand; and the curved arm has one more bending moment and a twisting moment due to its curvature. These two added items increse the size of the arm; but this disadvantage is more than offset by making the application of the apparatus enormously simpler.

Wehave often been confronted with the question,"How are you going to pick up these boats after they are launched?" The answer is,"who wants to put lifeboats back on a disabled ship?" This statement assumes that the lifeboats are to be used in the ship's own emergency. In calm water it would be possible to pick up boats with this apparatus, but in heavy weather it would be impossible. "Can the conventional type of lifeboat launching apparatus pick up a boat in rough water,or does it wait till the seas subdue?" Another objection that has come up occasionally is that the huge arms protruding from the side of the vessel act in the same manner as did the bowsprit of the old square riggers,- under water and out. This claim completely forgets that a ship does not pitch back and forth transveresly but lells back and forth slowly. When full of water, it does not even do much of that.

The time of launching could easily be chosen at the most opportune time of the roll.

When several of these installations are put on a ship, it is possible, but not probable that one set after launching would interfere with another while it was launching. With a little foresight in the design, this trouble could easily be avoided.

When the present day apparatus fails to launch its lifeboat on the high side of the vessel, this new apparatus is able to do so with little difficulty. There is a limit, of course, as to the angle of list against which it will launch a boat. But, due to the nature of the design of this apparatus, the lifeboat is launched with it's keel always parallel to the keel of the vessel itself. So, to go to the extreme, if the ship were trimmed by the stern by an angle of forty five degrees with the horizontal, the lifeboat would never be launched- it would simply move in a horizontal plane. In other words the plane of rotation would become a horizontal plane. But does one wait till the ship is trimmed to such an excessive angle before launching a lifeboat? The answer is obviously no. It is nevertheless one of the biggest disadvantage if not the biggest. But if care is taken to launch the boat as soon as the ship is in danger of taking a permanent and dangerous trim, or even if the ship is brought back to almost normal trim by flooding compartments, it would practically eliminate this trouble. It of course requires fast and straight thinking on the part of the commander.

we have purposely left the question of weight and cost till last. It is in our opinion the least important of all the questions discussed. It goes without saying that the weight and cost of this new apparatus exceeds that of the present type; but not by an amount to make it prohibitive. therefore, since it is not so much in excess to present day equipment, and if the ultimate goal is to endeaver to make the passenger's life a little safer, then why debate the question of cost and weight?

In weighing the advantages and disadvantages of this new apparatus and comparing it with present day equipment, we have come to the conclusion that it does give the removal of passengers from a sinking ship a greater element of security than could be obtained with the conventional type of davit. It must be kept in mind, however, that we are not insinuating that it is a foolproof apparatus; but it does make the life of the passenger a little safer in case of emergency; and after all this the ultimate purpose of any lifesaving apparatus.

DATA

POSITION NO.1

C	F	R	(B.M.) FV	M	(B.M.) FH	A V	A H	T	(B.M) T	A 2	B	F B	(B.M.) B
0	4.53	47	-213	-5	23	45°	45°	11.8	213	45°	36.5	-.63	-23
10	"	37.4	-170	"	"	"	"	"	159	"	24.8	"	-15.6
18	"	29.7	-135	"	"	"	"	"	122	"	17.8	"	-11.2
26	"	21.7	-98.5	"	"	"	"	"	80.2	"	9.7	"	-6.1
34	"	14.1	-64.0	"	"	"	"	"	33.0	"	1.1	"	-.69
42	"	6.5	-29.4	"	"	"	"	"	0	"			
44.5	"	3.7	-16.8	-3.8	17.2	"	"	"					
45.7	"	2.2	-9.97	-2.5	11.3	"	"	"					
46.5	"	1.1	-4.98	-1.3	5.90	"	"	"					
47.2	"	0	0	0	0	"	"	"					

C	M	(B.M.) FH	I	S	(B.M.) S
0	19.0	-86.0	39.2	2.19	86.0
10	14.2	-64.3	29.8	"	65.2
18	10.3	-46.1	22.4	"	49.0
26	6.6	-29.9	14.8	"	32.4
34	2.7	-12.2	7.6	"	16.6
42	-1.2	5.43	0	"	0
44.5	-1.6	7.25			
45.7	-1.2	5.43			
46.5	-.6	2.72			
47.2	0	0			

C	M	(B.M.) FH	I	S	(B.M.) S
0	38.1	-173	35.0	4.91	174
10	29.8	-135	26.8	"	123
18	23.1	-105	20.0	"	98.2
26	16.4	-74.3	13.0	"	64.0
34	9.7	-44.0	6.50	"	32.0
42	3.1	-14	0	"	0
44.5	1.3	-5.9			
45.7	.7	-3.17			
46.5	.3	-1.36			
47.2	0	0			

POSITION NO.4

C	M	(B.M.) FH	I	S	(B.M.) S
0	47.0	-213	28.7	7.44	214
10	37.3	-169	21.8	"	162
18	29.6	-134	16.5	"	123
26	21.7	-98.1	12.3	"	93.3
34	14.2	-64.3	5.4	"	40.2
42	6.5	-29.4	0	"	0
44.5	3.7	-16.7			
45.7	2.2	-9.95			
46.5	1.2	-5.43			
47.2	0	0			

POSITION NO.5

C	M	(B.M.) FH	I	S	(B.M.) S
0	43.2	-196	20.2	9.64	195
10	34.9	-158	15.6	"	151
18	28.2	-128	11.7	"	113
26	21.5	-97.5	7.8	"	75.3
34	14.8	-67.0	3.8	"	36.6
42	8.2	-37.1	0	"	0
44.5	5.0	-22.6			
45.7	3.2	-14.5			
46.5	1.5	-6.8			
47.2	0	0			

POSITION NO.6

C	M	(B.M.) FH	I	S	(B.M.) S
0	27.8	-126	10.6	12.1	128
10	23.0	-104	8.0	"	96.6
18	19.1	-86.5	6.0	"	72.5
26	15.2	-68.8	4.0	"	48.4
34	11.5	-52.0	2.0	"	24.2
42	7.6	-34.4	0	"	0
44.5	5.0	-22.6			
45.7	3.3	-15.0			
46.5	1.5	-7.0			
47.2	0	0			

TABLE OF TWISTING MOMENTS

Pos.	F	(J-90)	Cos.(J 90)	FCos.(J-90)	f	FCos.(J-90)f
1	6.41	32	.866	5.56	5.0	27.8
2	"	24	.914	5.85	2.80	16.40
3	"	9	.988	6.33	.80	5.06
4	"	-11	.982	Not measurable		
5	"	-32	.848			
6	"	-52	616	3.95	1.50	5.93

TWISTING MOMENTS

C	F	B	Sin.B	F Sin. B	g	F Sin.B(g)
44.5	6.41	15	.259	1.66	3.8	6.30
45.7	"	5	.087	.558	2.5	1.40
46.5	"	0	0	0	1.3	0
47.2	"	0	0	0	0	0

44.5	6.41	17	.292	1.87	2.0	3.74
45.7	"	6	.105	.674	1.4	.944
46.5	"	3	.052	.334	.60	.201
47.2	"	0	0	0	0	0

Twisting Moments

TABLE OF TWISTING MOMENTS

C	F	B	Sin.B	FSin.B	g	FSin.B(g)
44.5	6.41	28	.469	3.01	.6	1.811
45.7	"	22	.375	2.40	.4	.960
46.5	"	17	.292	187	.2	.374
47.2	"	14	.242	1.55	0	0

C	F	B	Sin.B	FSin.B	g	FSin.B(g)
44.5	6.41	86	.998	6.40	1.2	7.68
45.7	"	80	.985	6.31	.8	5.05
46.5	"	77	.974	6.24	.4	2.50
47.2	"	76	.970	6.22	0	0

Twisting Moments

COMPRESSION & TENSION

						Due to F			
C.C.	0---42	---------		44.5	45.7	46.5	47.2		
Pos.	Due. to F F Sin(J-90)	Due to T T Cos.26	Due to S S Cos.J	F Cos.B	F Cos.B	F Cos.B	F Cos.B	B	Cos.B
11	3.20	10.6	0	6.20	6.40	6.41	6.41	15	.966
2	2.60	"	.543	6.12	6.39	6.41	6.41	17	.956
3	1.00	"	2.37	5.66	5.95	6.20	6.22	28	.883
4	-1.23	"	5.08	3.69	4.69	5.48	5.94	55	.574
5	-3.40	"	8.03	1.88	3.20	3.90	4.76	73	.292
6	-5.06	"	11.3	.45	1.12	1.46	1.55	86	.070

Compression and
Tension in arms.

TENSION & COMPRESSION

	Due to F							
C	45.7		46.5		47.2			
Pos.	B	Cos. B	B	COS.B	B	Cos. B	L	$\frac{L}{42}$:Cos.B
1	5	.996	0	1	0	1	0	0
2	6	.995	3	.999	0	1	10.4	.248
3	22	.927	17	.956	14	.970	20.3	.483
4	43	.731	32	.848	22	.927	28.7	.683
5	60	.50	53	.602	42	.743	35.0	.833
6	80	.174	77	.225	76	.242	39.2	.933

Compression and Tension in Arms.

TABLE OF TENSION IN WORKING CABLES

E	POS.	d	F	Fd	e	$S' = \frac{Fd}{-e}$	G	Sin. G	$S = \frac{S'}{Sin. G}$
0	1	-5	4.53	-23	36.5	-.63	S=S'=0 at E=6°		
30	2	19	"	186	40.5	2.12	75	.966	2.19
60	3	38	"	172	"	4.25	60	.866	4.91
90	4	47	"	213	"	526	45	.707	7.44
120	5	43	"	195	"	4.82	30	.500	9.64
150	6	28	"	127	"	3.14	15	.259	12.1

CALCULATIONS

CALCULATIONS FOR LOAD ON ARMS

~WEIGHT~

NO. OF PERSONS IN BOAT --	140
AVERAGE WT. PER. PERSON	140 LB
TOTAL WT. OF PERSONS --	19,600 LB.
WT. OF BOAT - - -	7,840 "
WT. OF DIESEL ENGINE	2,000 "
WT. OF ARMS & CRADLE	2,960 "
TOTAL WT.	32,400 LB

$\frac{32,400}{2240} = 14.8$ LONG TONS [ON THREE ARMS]

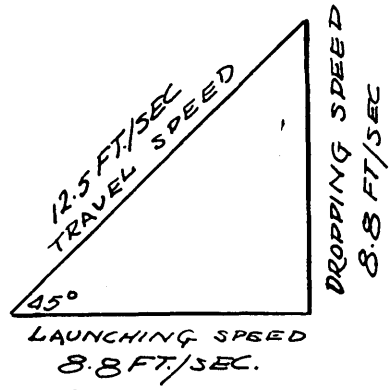
$\frac{14.8}{3} = 4.9$ TONS ON ONE ARM

ADDED FORCE DUE TO SUDDEN CHECKING.

LET r = LENGTH OF ARMS = 47 FT.

TRAVEL IN PLANE OF ROTATION = $S = \frac{2 \times 3.14 \times 47}{4}$ = 73.8 FT.

LAUNCHING SPEED = 6 M.P.H = 8.8 FT/SEC = 52 KNOTS



$s = \frac{1}{2}at^2$ $\qquad$ $v = at$

$73.8 = \frac{1}{2}at^2$ $\qquad$ $8.8 = at$

$at^2 = 147.6$ $\qquad$ $a = \frac{8.8}{t}$

$\frac{8.8}{t} \cdot t^2 = 147.6$; $t = 16.8$ SECS. [LAUNCHING TIME]

MAXIMUM FORCE DUE TO CHECKING

LET "t" = TIME OF BRAKING

" "S" = DISTANCE TRAVELLED DURING BRAKING

"v" = MAX. VELOCITY.

$v = 12.5$ FT./SEC.

$S = 20$ FT [ASSUMED]

$20 = \frac{1}{2} a t^2$ $\qquad v = at$

$40 = at^2$ $\qquad 12.5 = at$

$\therefore 40 = 12.5t$

$t = 3.2$ SECS.

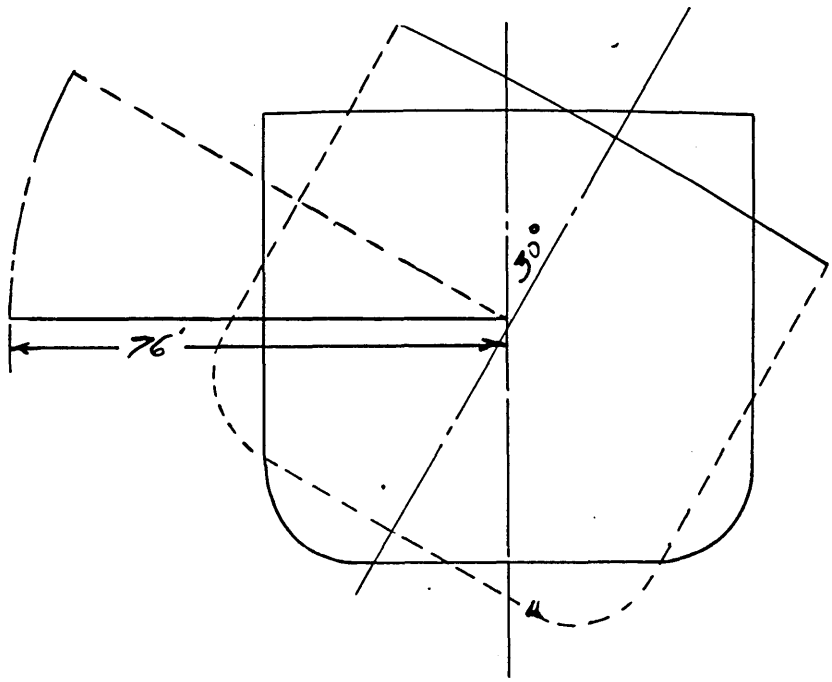
$a = -\frac{12.5}{3.2} = -3.91$ FT/SEC.^{2}

$F = Ma = \frac{14.8}{32.2} \times 3.91 = 1.80$ TONS

OR $.60$ TONS/ARM.

VERTICAL COMPONENT $= .60 \times .707 = .42$ TONS

ADDED FORCE DUE TO ROLLING OF SHIP.



ASSUME THE LIFEBOAT TO HAVE SIMPLE HARMONIC MOTION

ASSUME ROLLING PERIOD TO BE 15 SECS. BOTH WAYS.

FOR SIMPLE HARMONIC MOTION

$S = r \sin \omega t$; $V = \frac{ds}{dt} = r\omega \cos \omega t$, $a = \frac{dv}{dt} = \omega^2 r \sin \omega t$

S = DISTANCE travelled

r = RADIUS OF PATH OF TRAVEL

ω = ANGULAR VELOCITY

$\omega = 2\pi$ [NO. OF CYCLES/SEC]

$= 2\pi \times \frac{1}{20} = .314$ RAD/SEC.

$a_t = (.314)^2 \times 39.8 \times 1 = 6.92$ FT/SEC^2

[ωt ASSUMED 90° SO AS TO GET MAX. ACCEL.]

$S = \frac{3.14 \times 76 \times 2}{6} = 79.5$ FT

$F = \frac{W}{g} \times a = \frac{14.8 \times 6.92}{32.2} = 3.18$ TONS

OR 1.06 TONS/ARM

AGGLOMERATION OF FORCES FOR WORST POSSIBLE CONDITIONS

∴ TOTAL FORCE = 4.93 + 1.06 + .42 = 6.41 TONS/ARM

LET "C" = ANGLE BETWEEN THE TOTAL LOAD [6.41 TONS] AND THE PLANE OF ROTATION.

"C" = 45° FOR ALL POSITIONS.

∴ 6.41 × .707 = 4.53 TONS = "F" = FORCE NORMAL TO PLANE OF ROTATION AND ALSO IN THE PLANE OF ROTATION.

TENSION IN FIXED LENGTH CABLES

ARM = 42 FT.
LOAD = 6.41 TONS
ANGLE OF LOAD WITH PLANE OF ROTATION = 45°
∴ EFFECTIVE LOAD = 6.41 × .707 = 4.53 TONS

MOMENT DUE TO "F" = MOMENT DUE TO "T"
$4.53 \times 47.0 = T \times 18$

T = 11.8 TONS [TENSION]

CALCULATIONS FOR DIMENSION OF ARMS

POSITION NO. 1

STRESS AT A POINT 42 FT. FROM PIN.

$(BM)_V = 23$ FT. TONS

$(B.M)_H = 30$ FT. TONS

$$\text{RES. B.M.} = \sqrt{(23)^2 + (30)^2} = 37.8 \text{ FT. TONS}$$

TWISTING MOMENT = 27.8 FT. TONS

$$\text{EQUIVALENT B.M.} = \tfrac{1}{2} \times 37.2 + \tfrac{1}{2}\sqrt{(27.8)^2 + (37.8)^2}$$

$$= 42.4 \text{ FT. TONS}$$

DIRECT STRESS = 13.8 TONS IN COMPRESSION

ASSUME A HOLLOW ARM — ASSUME $f_t = 60{,}000$ #/□

OUTSIDE DIA. = 8 INCHES — " A FACTOR OF

INSIDE DIA. = 5 INCHES — SAFETY OF 2

$$f_t = \frac{My}{I} \pm \frac{P}{A} \; ; \; f_t = \frac{42.4 \times 4 \times 2240 \times 12 \times 4}{3.14(4^4 - 2.5^4)} - \frac{13.8 \times 2240}{3.14(4^2 - 2.5^2)}$$

$$= 26{,}000 \text{ \#/□}$$

THIS STRESS IS A LITTLE LOWER THAN DESIRED THEREFORE ASSUME AN INSIDE DIA. OF 6 IN.

$$f_t = \frac{42.4 \times 2240 \times 12 \times 4 \times 4}{3.14(256 - 81)} - \frac{31{,}000}{3.14(16 - 9)}$$

$$= 31{,}790 \text{ \#/□}$$

THIS STRESS IS SATISFACTORY THEREFORE THE ABOVE ARM IS USED AS A STANDARD FOR COMPUTATIONS AT OTHER POSITIONS OF ARM.

POSITION NO. 6

STRESS AT A POINT 42 FT. FROM PIN

$(BM)_V = 30$ FT. TONS

$(BM)_H = 34.4$ FT. TONS

$$\text{RES. B.M.} = \sqrt{(30)^2 + (34.4)^2}$$
$$= 45.5 \text{ FT. TONS}$$

TWISTING MOMENT $= -5.93$ FT. TONS

$$\text{EQUIVALENT B.M.} = \tfrac{1}{2} \times 45.5 + \tfrac{1}{2}\sqrt{(45.5)^2 + (5.93)^2}$$
$$= 45.9 \text{ FT. TONS}$$

DIRECT STRESS = 17.0 TONS IN COMPRESSION

ASSUME A HOLLOW SHAFT

OUTSIDE DIA. = 8 IN.

INSIDE DIA. = 6 IN.

$$f_t = \frac{45.9 \times 4 \times 2240 \times 4 \times 12}{3.14(8^4 - 3^4)} - \frac{17.0 \times 2240}{3.14(4^2 - 3^2)}$$
$$= 34{,}260 \text{ \#/□}$$

THIS EXCEEDS THE DESIRED STRESS INTENSITY SO ANOTHER APPROXIMATION OF THE ARM DIMENSIONS IS NECESSARY.

ASSUME OUTSIDE DIA. = 8.5 IN.

" INSIDE " = 6.0 IN.

$$f_t = \frac{45.9 \times 4 \times 2240 \times 12 \times 4.25}{3.14(4.25^4 - 3^4)} - \frac{38{,}200}{3.14(4.25^2 - 3^2)}$$
$$= 25{,}200 \text{ \#/□}$$

WITH THIS STRESS INTENSITY THE FACTOR OF SAFETY IS $\frac{60{,}000}{25{,}200} = 2.38$

IN ORDER TO GET A UNIFORMLY VARYING INSIDE DIAMETER – ASSUME INSIDE DIA. OF ARM AT PIN TO BE 3 INCHES, THEN BY DIRECT PROPORTION THE INSIDE DIA. AT ANY OTHER SECTION CAN BE OBTAINED. [SKETCH ILLUSTRATING THIS IS FOUND ON PREVIOUS PAGE.]

$$\frac{x}{72} = \frac{6}{84} \; ; \; x = 5.1''$$

USE A 5" INSIDE DIA.

ASSUME A 7 IN. OUTSIDE DIA.

$$f_t = \frac{33.8 \times 2240 \times 12 \times 4 \times 3.5}{3.14(3.5^4 - 2.5^4)} - \frac{15.2 \times 2240}{3.14(3.5^2 - 2.5^2)}$$

$$= 34{,}600 \, \#/\square \quad [\text{TOO HIGH}]$$

ASSUME A 7½ IN. OUTSIDE DIA.

$$f_t = \frac{33.8 \times 2240 \times 12 \times 4 \times 3.75}{3.14(3.75^4 - 2.5^4)} - \frac{34{,}000}{3.14(3.75^2 - 2.5^2)}$$

$$= 25{,}840 \, \#/\square$$

THEREFORE AN ARM OF 7½ IN. OUTSIDE DIA. AND 4.0 IN. INSIDE DIA. IS REQUIRED AT A SECTION 30 FT. FROM PIN.

POSITION NO. 6

STRESS AT A POINT 30 FT. FROM PIN.

$(BM)_V = 20.1$ FT. TONS

$(BM)_H = 22.5$ FT. TONS

RES. B.M. $= \sqrt{(20.1)^2 + (22.5)^2} = 30.2$ FT. TONS.

TWISTING MOMENT $= -5.93$ FT. TONS

EQ. B.M. $= \frac{1}{2} \times 30.2 + \frac{1}{2}\sqrt{(30.2)^2 + (5.93)^2}$

$= 30.5$ FT. TONS

DIRECT STRESS = 16.8 TONS IN COMP.

POSITION NO. 5.

STRESS AT A POINT 30 FT. FROM. PIN.

$(BM)_V = 20.1$ FT. TONS

$(BM)_H = 27.0$ " "

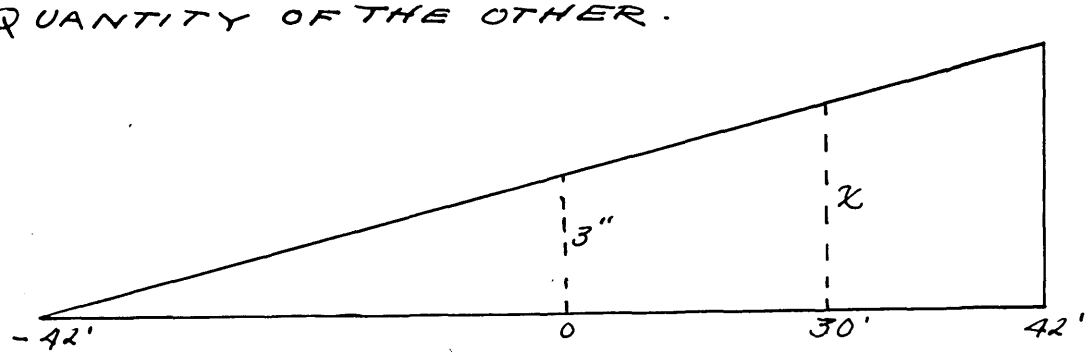
RES. B.M. $= \sqrt{(20.1)^2 + (27.0)^2} = 33.8$ FT. TONS

TWISTING MOMENT = 0

EQ. B.M. $= \frac{1}{2} \times 33.8 + \frac{1}{2} \times 33.8 = 33.8$ FT. TONS

DIRECT STRESS = 15.2 TONS IN COMP.

IT IS EVIDENT THAT THE STRESS INTENSITY WILL BE PRACTICALLY THE SAME FOR THE TWO POSITIONS ABOVE; THEREFORE IT IS NEEDLESS TO BRING BOTH COMPUTATIONS TO A FINISH SINCE ONE IS A REPRESENTATIVE QUANTITY OF THE OTHER.



POSITION NO. 1

STRESS AT A POINT 15 FT. FROM PIN.

$(BM)_V = 5.0$ FT. TONS

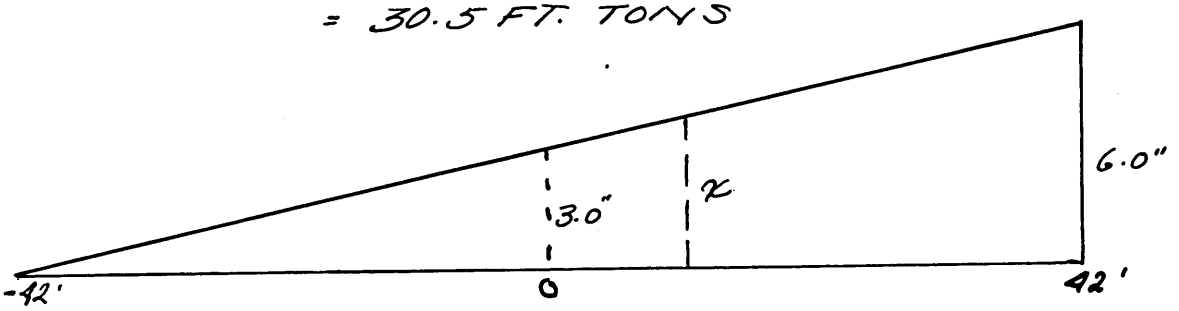
$(BM)_H = 23.0$ FT. TONS

TWISTING MOMENT = 27.8 FT. TONS

RES. B.M. $= \sqrt{(27.8)^2 + (5.0)^2} = 23.6$ TONS

EQ. B.M. $= \frac{1}{2} \times 23.6 + \frac{1}{2}\sqrt{(23.6)^2 + (27.8)^2}$

$= 30.5$ FT. TONS



$\frac{6}{x} = \frac{84}{57}$; $x = 4.07''$ USE 4" INSIDE DIA.

ASSUME OUTSIDE DIA. = 6½ IN.

$$f_t = \frac{30.5 \times 2240 \times 12 \times 4 \times 3.25}{3.14(3.25^4 - 2^4)} - \frac{13.8 \times 2240}{3.14(3.25^2 - 2^2)}$$

$= 34{,}010$ #/□ [TOO HIGH]

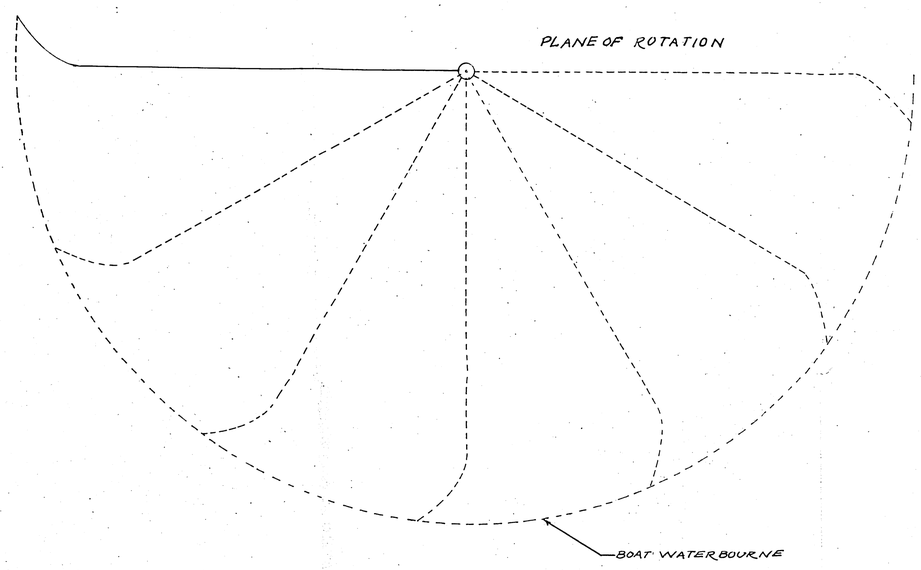
ASSUME OUTSIDE DIA. = 7 IN.

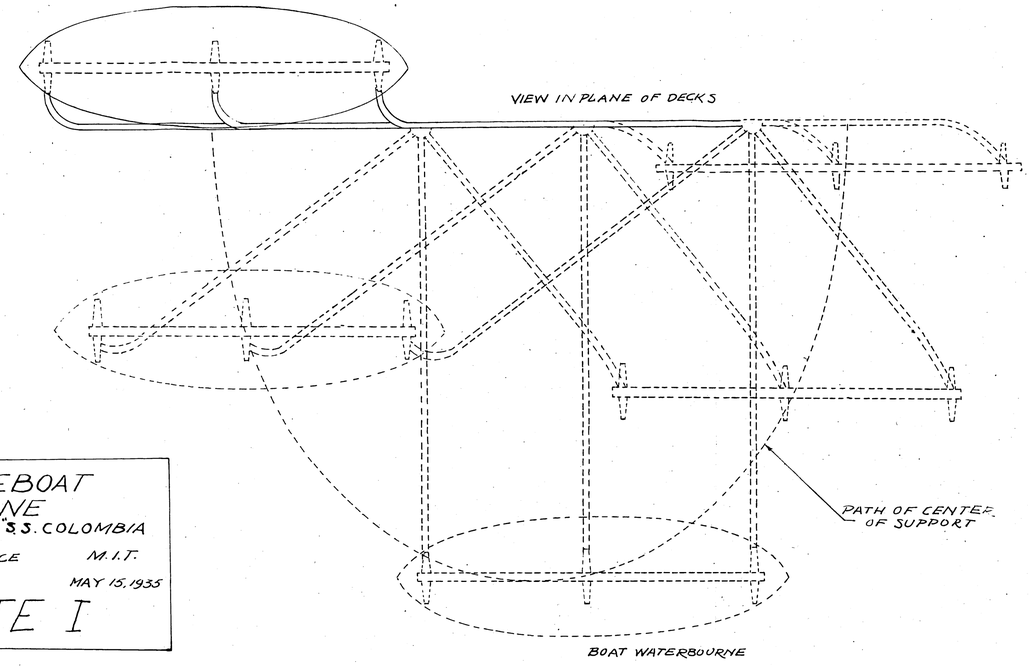
$$f_t = \frac{30.5 \times 2240 \times 12 \times 4 \times 3.5}{3.14(150 - 16)} - \frac{31{,}000}{3.14(12.6 - 4)}$$

$= 26{,}150$ #/□

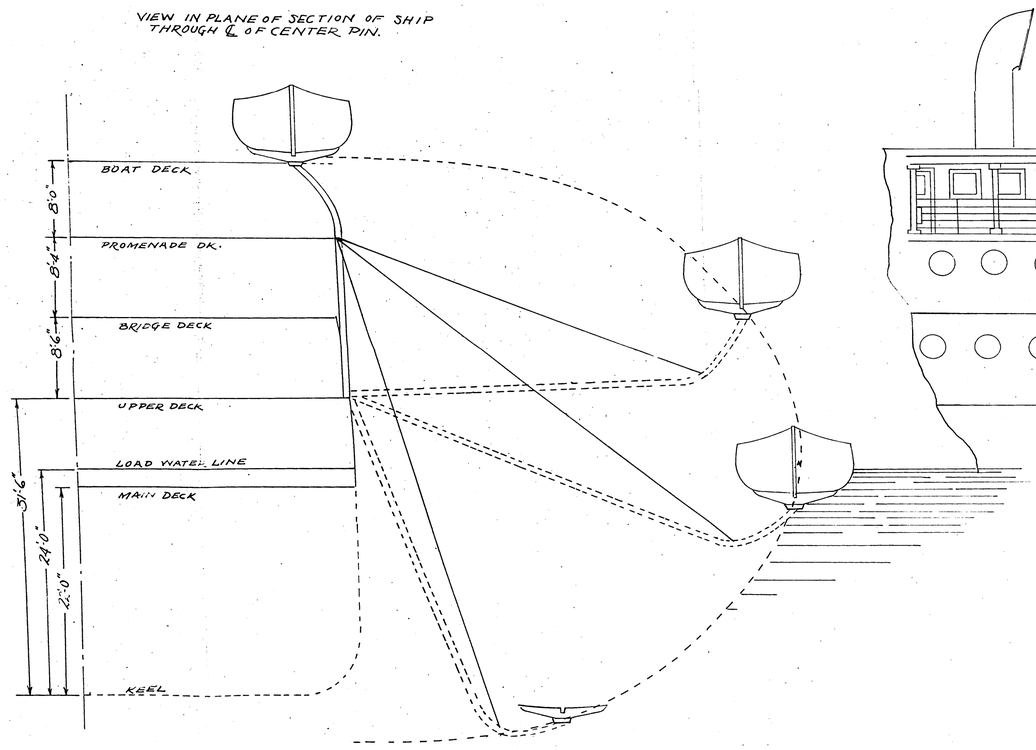
THEREFORE AN ARM OF 7 IN. OUTSIDE DIA. AND 4 IN. INSIDE DIA. IS REQUIRED AT A SECTION 15 FT. FROM THE PIN.

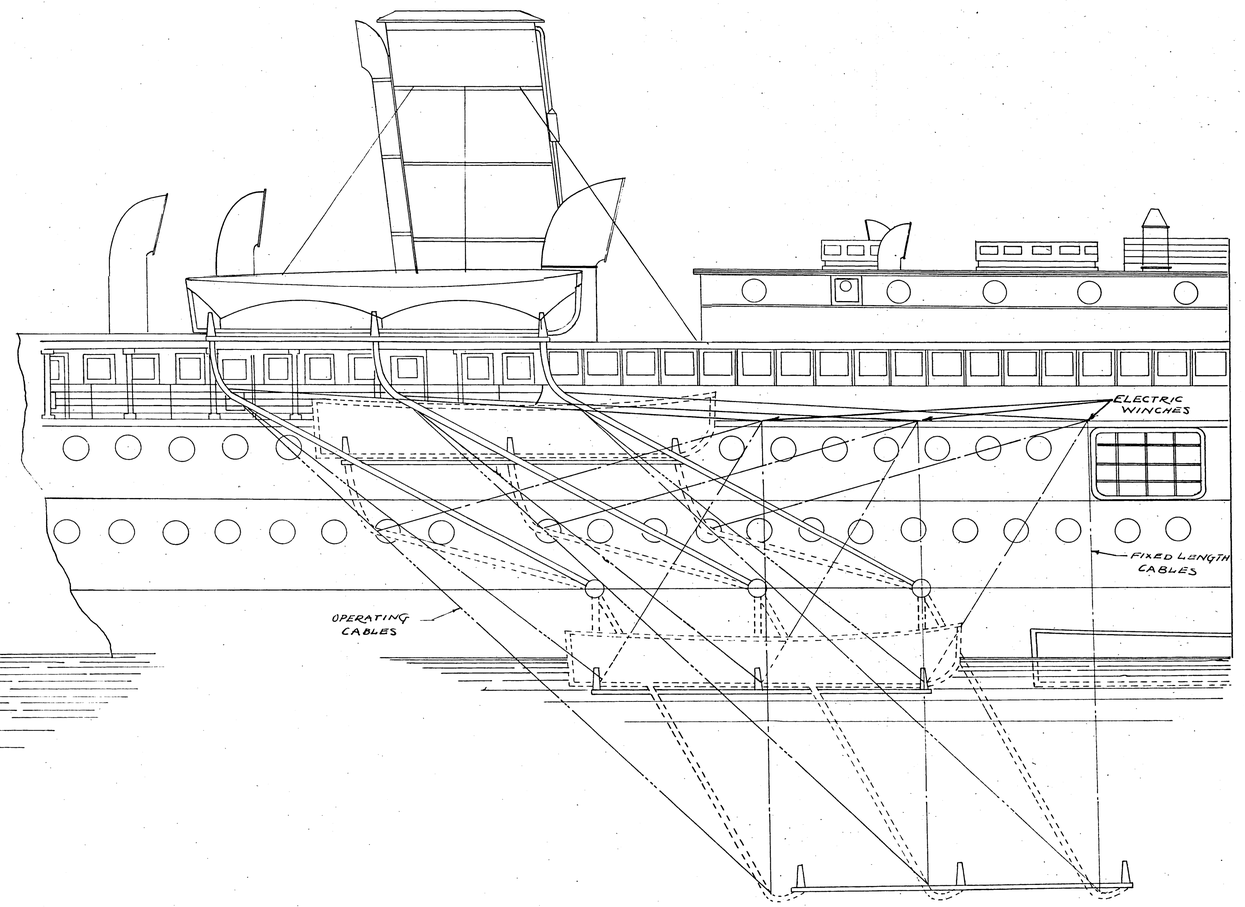
PLATES

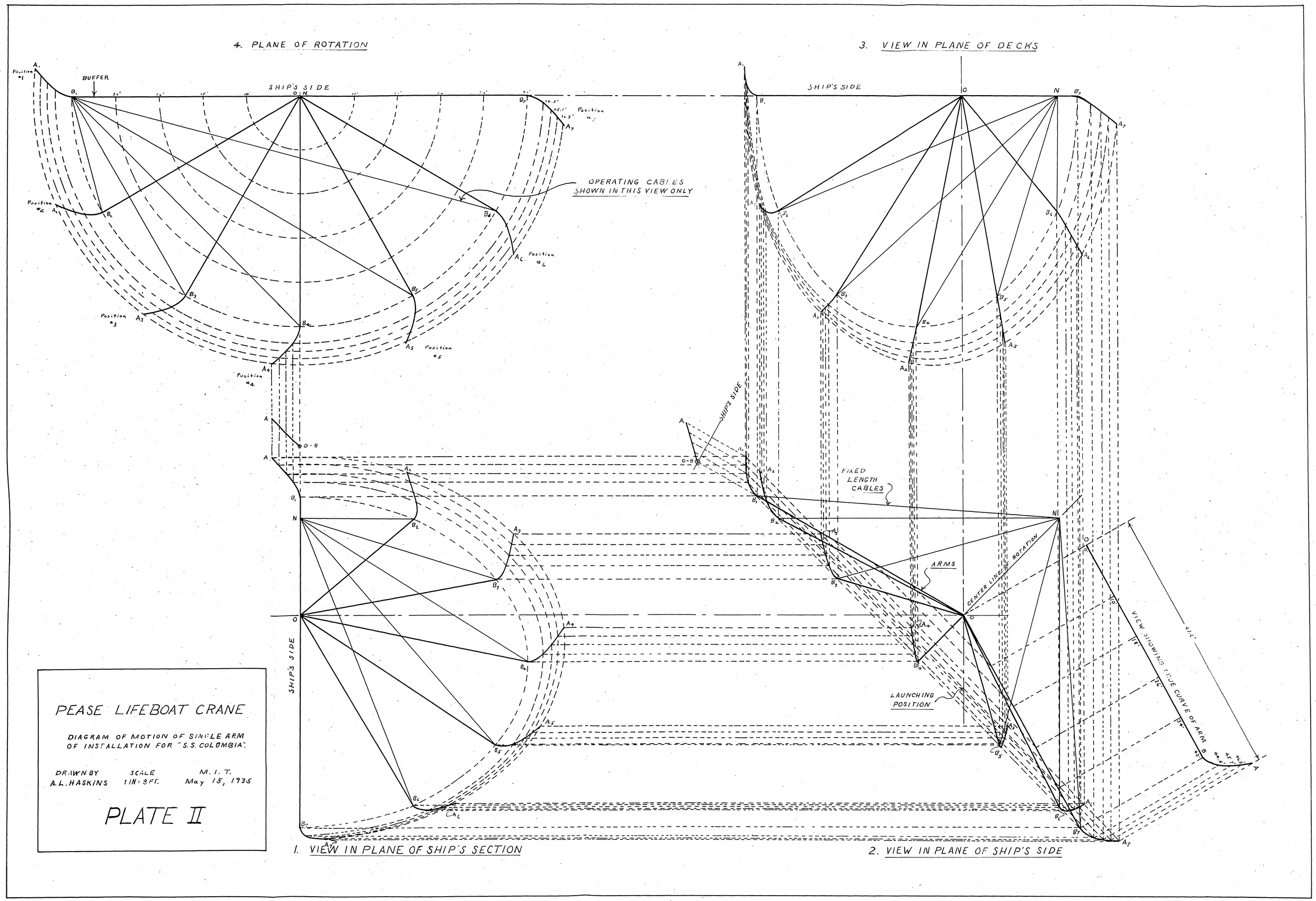




PEASE LIFEBOAT CRANE
AS INSTALLED ON "S.S. COLOMBIA"
DRAWN BY. R.P. IODICE M.I.T.
SCALE 1"=8 FEET MAY 15, 1935
PLATE I







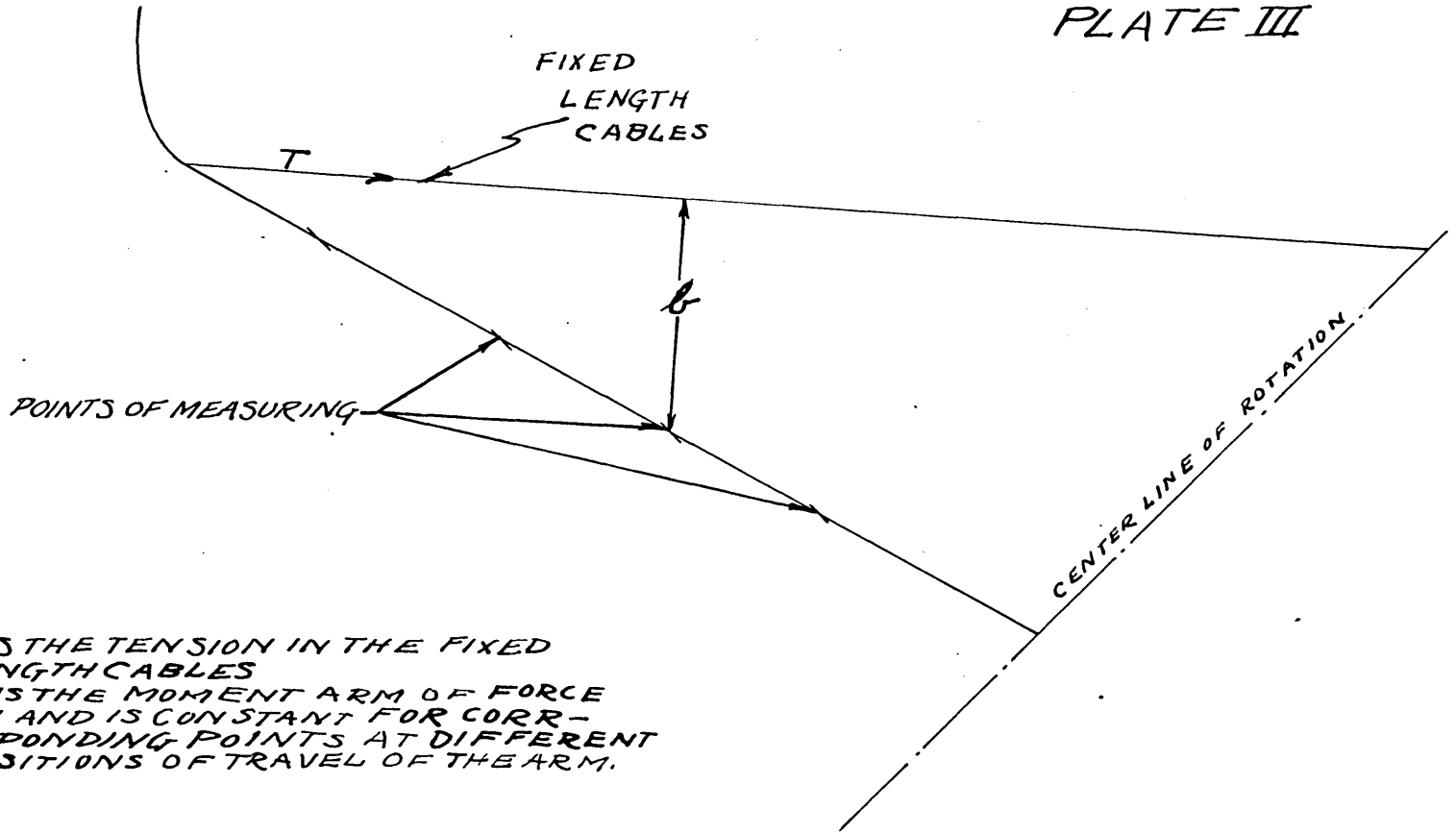
PEASE LIFEBOAT CRANE

DIAGRAM OF MOTION OF SINGLE ARM OF INSTALLATION FOR "S.S. COLOMBIA".

DRAWN BY A.L. HASKINS | SCALE 1 IN = 3 FT. | M.I.T. May 15, 1935

PLATE II

PLATE III

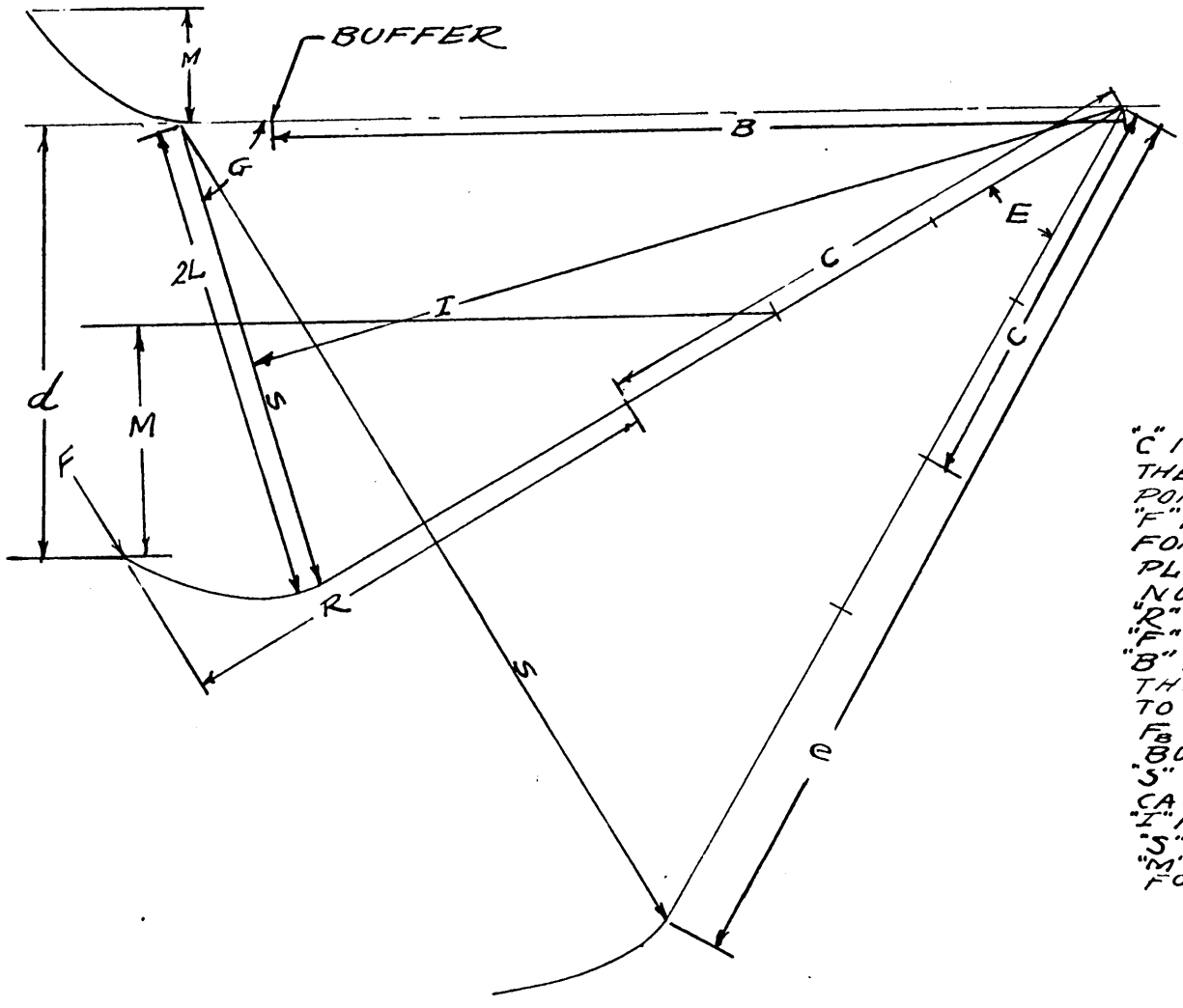


"T" IS THE TENSION IN THE FIXED LENGTH CABLES
"b" IS THE MOMENT ARM OF FORCE "T" AND IS CONSTANT FOR CORRESPONDING POINTS AT DIFFERENT POSITIONS OF TRAVEL OF THE ARM.

VIEW IN PLANE OF SHIP'S SIDE.

VIEW SHOWING PLANE OF ROTATION

PLATE IV



"C" IS A DISTANCE MEASURED ALONG THE ARM FROM THE PIN TO THE POINT IN QUESTION.
"F" IS THE COMPONENT OF THE TOTAL FORCE ACTING ON THE ARM IN THE PLANE OF ROTATION AND ALSO NORMAL TO THE PLANE OF ROTATION
"R" IS THE MOMENT ARM OF THE FORCE "F"
"B" IS THE DISTANCE MEASURED ALONG THE ARM FROM THE POINT IN QUESTION TO THE BUFFER [ONLY IN NO. 1 POS.]
F_B IS THE FORCE EXERTED BY THE BUFFER ON THE ARM [ie A NORMAL FORCE]
"S" IS THE TENSION ON THE OPERATING CABLES [ALWAYS IN THE PLANE OF ROT.]
"I" IS THE MOMENT ARM OF THE FORCE "S"
"M" IS MOMENT ARM OF COMPONENT OF FORCE "F" IN PLANE OF ROTATION.

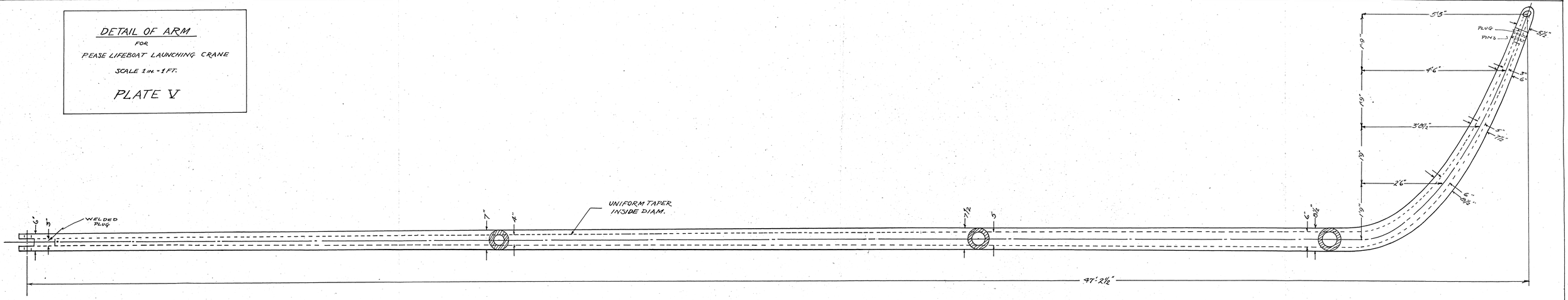
DETAIL OF ARM

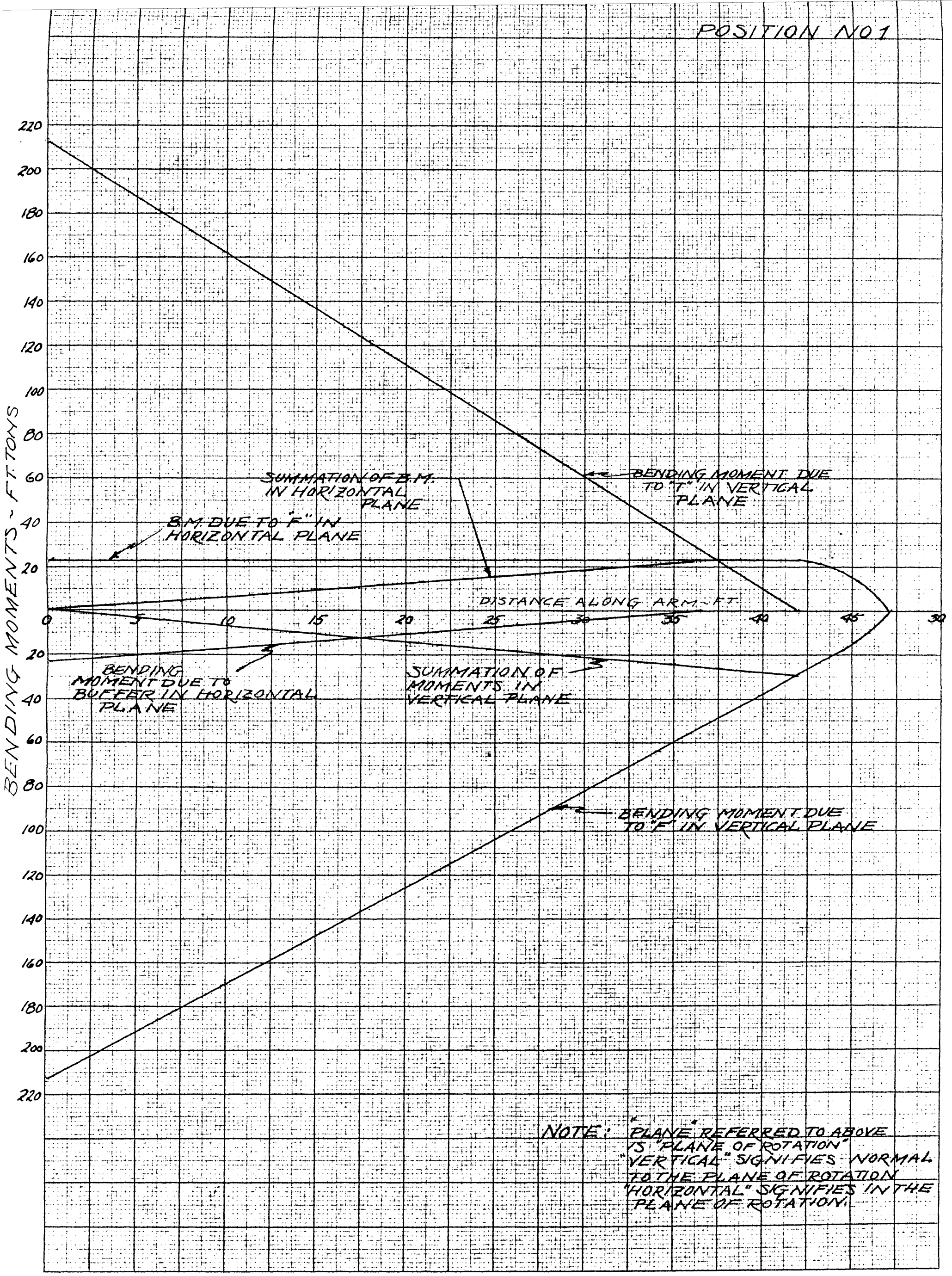
FOR

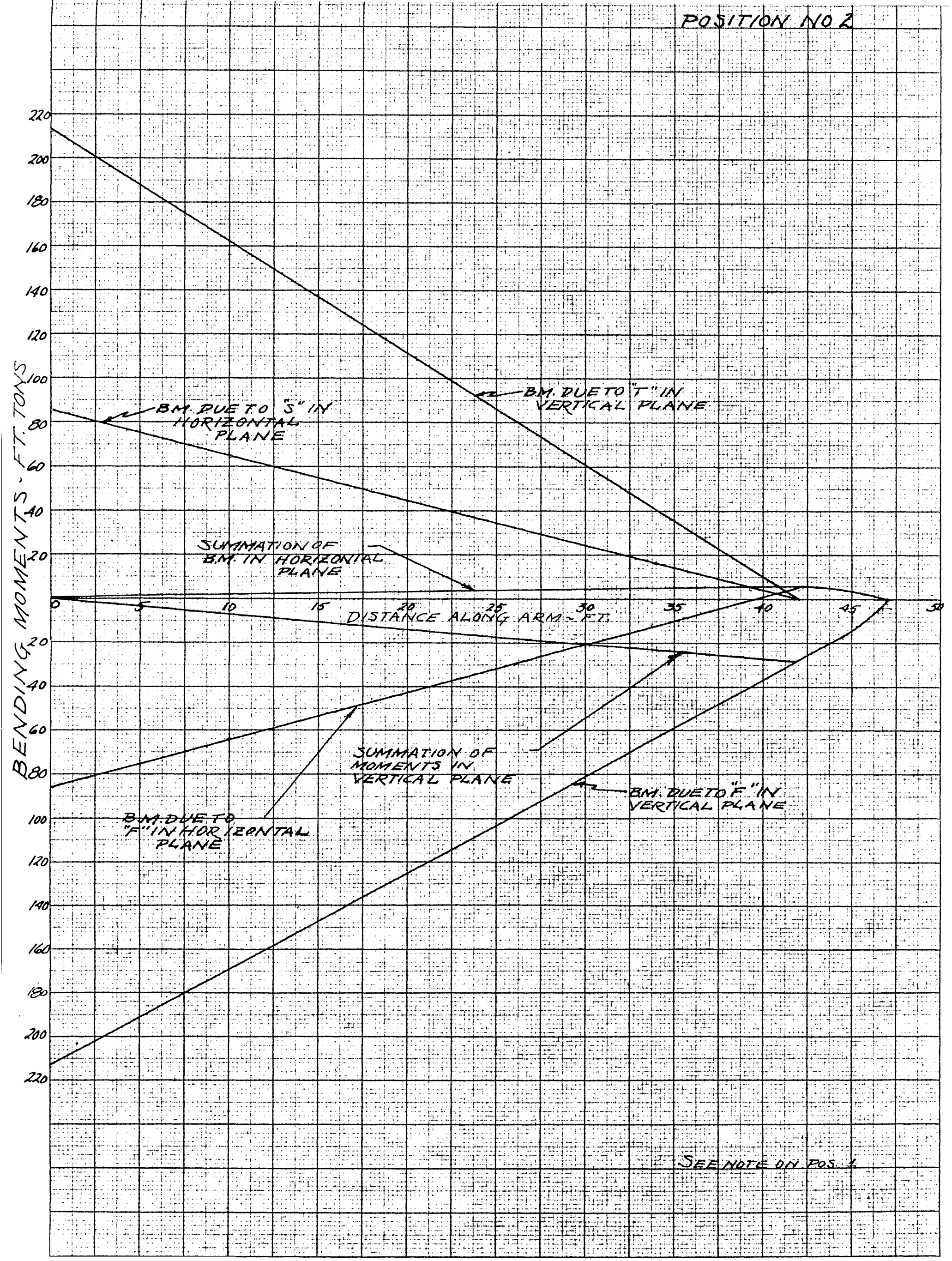
PEASE LIFEBOAT LAUNCHING CRANE

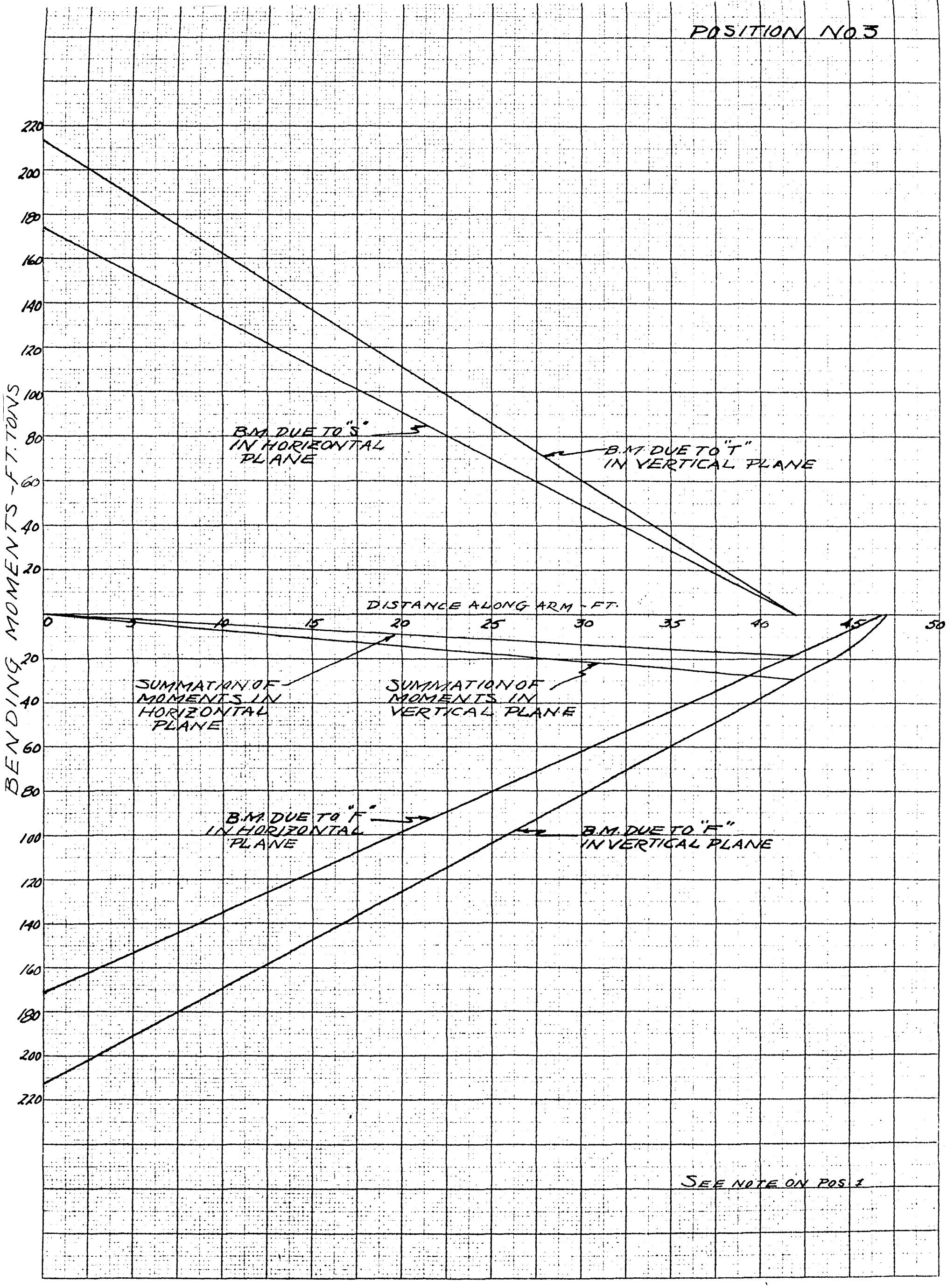
SCALE 1 IN. = 1 FT.

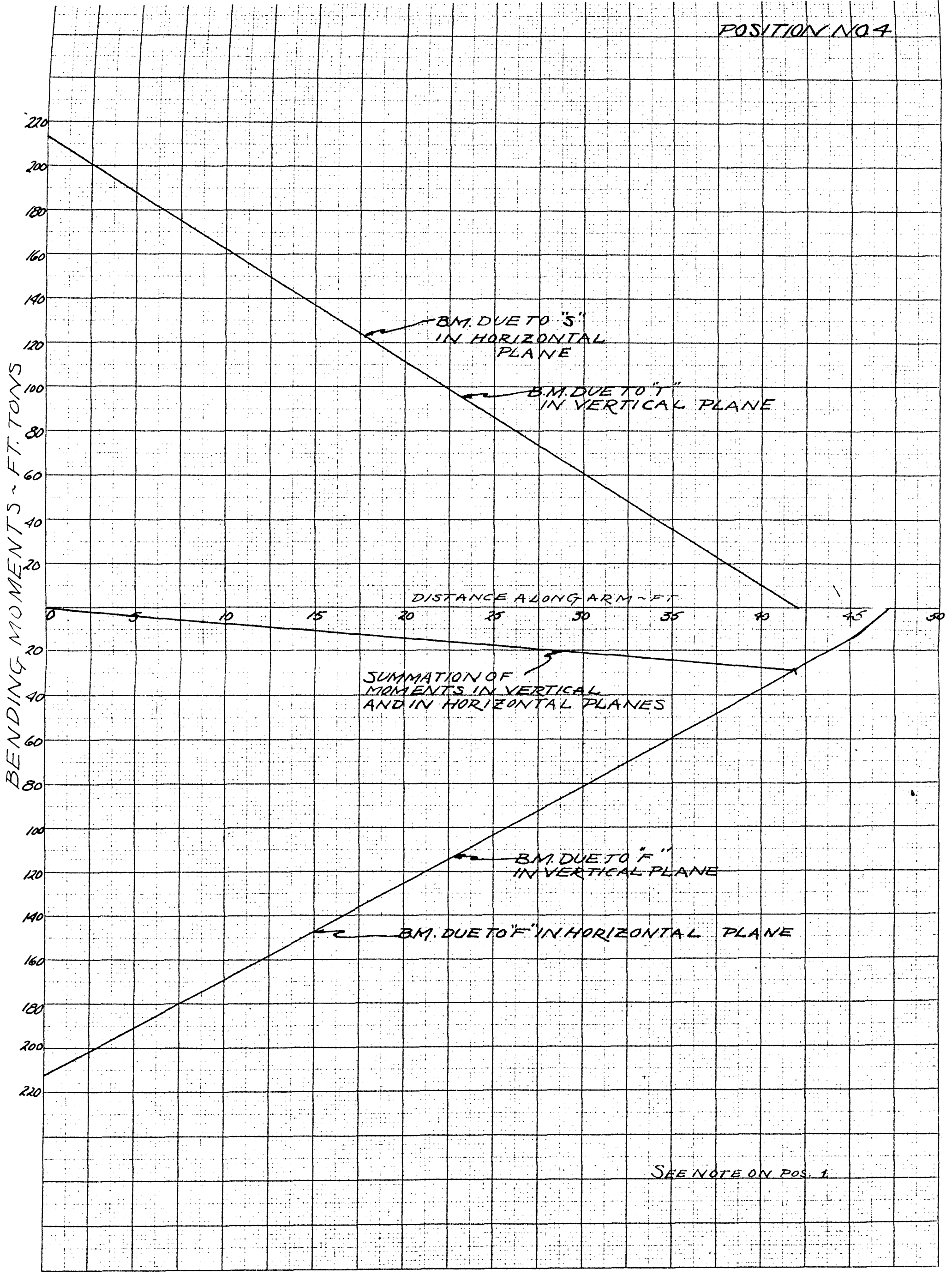
PLATE V

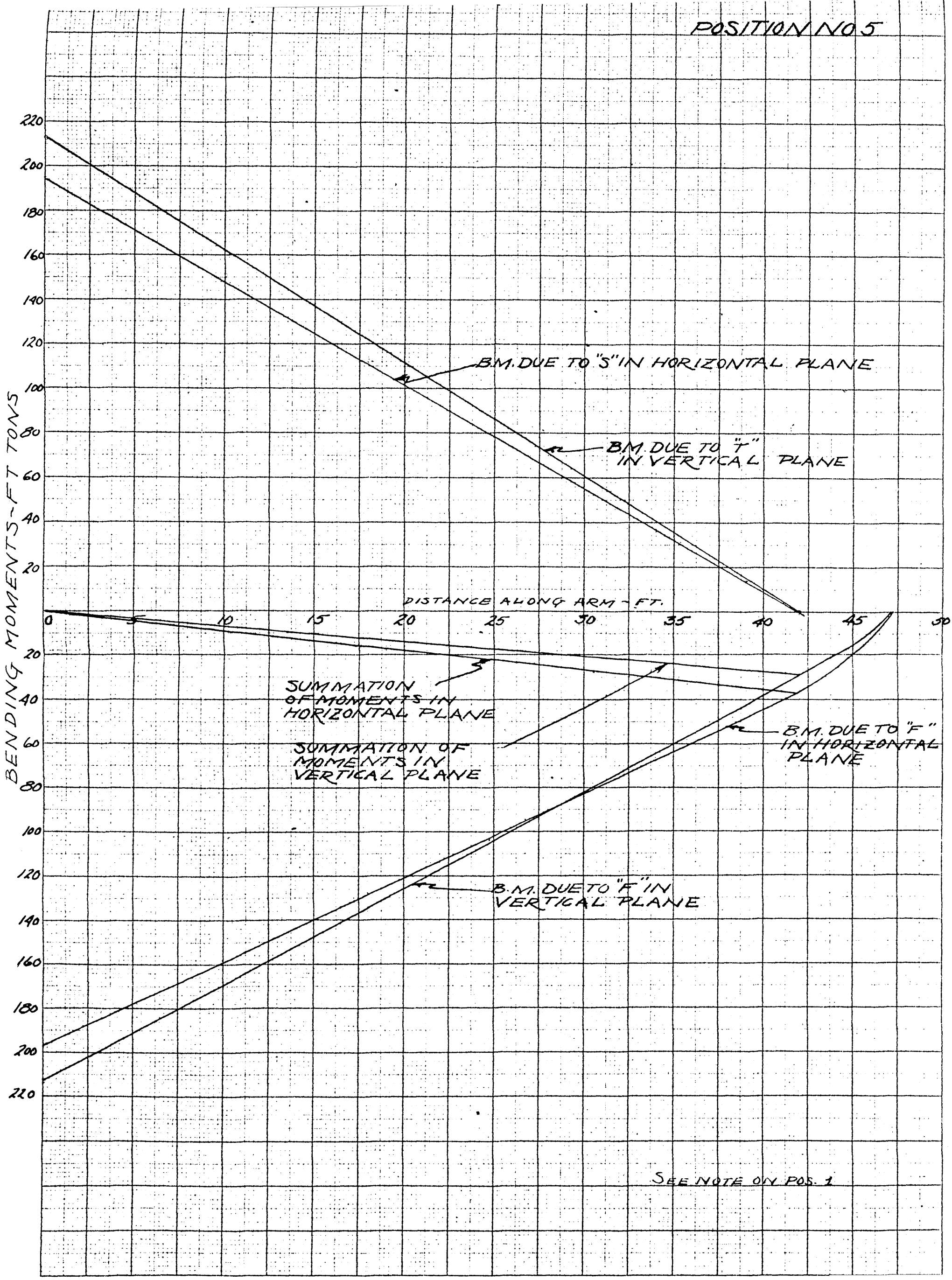



POSITION NO 1
BENDING MOMENTS ~ FT. TONS
220
200
180
160
140
120
100
80
60
40
20
0
20
40
60
80
100
120
140
160
180
200
220
0
5
10
15
20
25
30
35
40
45
50
DISTANCE ALONG ARM ~ FT.
SUMMATION OF B.M. IN HORIZONTAL PLANE
BENDING MOMENT DUE TO "T" IN VERTICAL PLANE
B.M. DUE TO "F" IN HORIZONTAL PLANE
BENDING MOMENT DUE TO BUFFER IN HORIZONTAL PLANE
SUMMATION OF MOMENTS IN VERTICAL PLANE
BENDING MOMENT DUE TO "F" IN VERTICAL PLANE
NOTE: "PLANE" REFERRED TO ABOVE IS "PLANE OF ROTATION"
"VERTICAL" SIGNIFIES NORMAL TO THE PLANE OF ROTATION
"HORIZONTAL" SIGNIFIES IN THE PLANE OF ROTATION.


POSITION NO 2
BENDING MOMENTS - FT. TONS
220
200
180
160
140
120
100
80
60
40
20
0
20
40
60
80
100
120
140
160
180
200
220
0
5
10
15
20
25
30
35
40
45
50
DISTANCE ALONG ARM - FT.
B.M. DUE TO "S" IN HORIZONTAL PLANE
B.M. DUE TO "T" IN VERTICAL PLANE
SUMMATION OF B.M. IN HORIZONTAL PLANE
SUMMATION OF MOMENTS IN VERTICAL PLANE
B.M. DUE TO "F" IN HORIZONTAL PLANE
B.M. DUE TO "F" IN VERTICAL PLANE
SEE NOTE ON POS. 1


POSITION NO 3
BENDING MOMENTS - FT. TONS
220
200
180
160
140
120
100
80
60
40
20
20
40
60
80
100
120
140
160
180
200
220
B.M. DUE TO "S" IN HORIZONTAL PLANE
B.M. DUE TO "T" IN VERTICAL PLANE
DISTANCE ALONG ARM - FT.
0
5
10
15
20
25
30
35
40
45
50
SUMMATION OF MOMENTS IN HORIZONTAL PLANE
SUMMATION OF MOMENTS IN VERTICAL PLANE
B.M. DUE TO "F" IN HORIZONTAL PLANE
B.M. DUE TO "F" IN VERTICAL PLANE
SEE NOTE ON POS 1


POSITION NO 4
BENDING MOMENTS ~ FT. TONS
220
200
180
160
140
120
100
80
60
40
20
0
20
40
60
80
100
120
140
160
180
200
220
B.M. DUE TO "S" IN HORIZONTAL PLANE
B.M. DUE TO "T" IN VERTICAL PLANE
DISTANCE ALONG ARM ~ FT
0
5
10
15
20
25
30
35
40
45
50
SUMMATION OF MOMENTS IN VERTICAL AND IN HORIZONTAL PLANES
B.M. DUE TO "F" IN VERTICAL PLANE
B.M. DUE TO "F" IN HORIZONTAL PLANE
SEE NOTE ON POS. 1


POSITION NO 5
BENDING MOMENTS - FT TONS
220
200
180
160
140
120
100
80
60
40
20
20
40
60
80
100
120
140
160
180
200
220
DISTANCE ALONG ARM - FT.
0
5
10
15
20
25
30
35
40
45
50
B.M. DUE TO "S" IN HORIZONTAL PLANE
B.M. DUE TO "T" IN VERTICAL PLANE
SUMMATION OF MOMENTS IN HORIZONTAL PLANE
SUMMATION OF MOMENTS IN VERTICAL PLANE
B.M. DUE TO "F" IN HORIZONTAL PLANE
B.M. DUE TO "F" IN VERTICAL PLANE
SEE NOTE ON POS. 1