CR 72.003

NAVAL CIVIL ENGINEERING LABORATORY
Port Hueneme, California

Sponsored by
NAVAL FACILITIES ENGINEERING COMMAND

EXPEDITIONARY LOGISTIC FACILITY
A CONCEPTUAL DESIGN OF MATERIALS HANDLING SYSTEM FOR NAVAL ADVANCED BASE PORTS

September 15, 1971

An Investigation Conducted by

J. J. HENRY CO., INC.
New York, New York 10006

N62399-71-C-0011

Unclassified

Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author): J. J. Henry Co., Inc. New York, New York 10006

2a. REPORT SECURITY CLASSIFICATION: Unclassified

2b. GROUP:

3. REPORT TITLE: Expeditionary Logistic Facility, A Conceptual Design of Materials Handling System for Naval Advanced Base Ports

4. DESCRIPTIVE NOTES (Type of report and inclusive dates): Final: 15 March - 15 September 1971

5. AUTHOR(S) (First name, middle initial, last name): Charles D. Hanley

6. REPORT DATE: 15 September 1971

7a. TOTAL NO. OF PAGES:

7b. NO. OF REFS:

8a. CONTRACT OR GRANT NO.: N62399-71-C-0011

b. PROJECT NO.: YF 38.536.005.01.010

c.

d.

9a. ORIGINATOR'S REPORT NUMBER(S): J.J.H. Report No. 5834

9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report): CR 72.003

10. DISTRIBUTION STATEMENT: Approved for public release; distribution unlimited

11. SUPPLEMENTARY NOTES: Contracting Activity: Naval Civil Engineering Laboratory Port Hueneme, California 93043

12. SPONSORING MILITARY ACTIVITY: Naval Facilities Engineering Command Washington, D. C. 20390

13. ABSTRACT

This report presents a unique concept for off-loading nonself-sustaining container ships at naval advanced bases. The concept is described with graphic representations, narrative discussions, performance estimates and cost data. System operations and performance are discussed. All auxiliary operations such as delivery, maneuvering, erection, positioning and repositioning are covered. The design analyses for the concept are also included. A program for the development through prototype is outlined. (U)

DD FORM 1 NOV 65 1473 (PAGE 1)

0102-014-6600

Unclassified

Security Classification

Unclassified
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Containership Offloading						
Nonself-sustaining Containerships						
Naval Advanced Bases						
Mobile Floating Cranes						

DD FORM 1 NOV 65 1473 (BACK)
(PAGE 2)

Unclassified
Security Classification

EXPEDITIONARY LOGISTIC FACILITY

A CONCEPTUAL DESIGN
OF
MATERIALS HANDLING SYSTEM
FOR
NAVAL ADVANCED BASE PORTS

PREPARED BY

J. J. HENRY CO., INC.

WITH CONSULTANT
ENESS RESEARCH & DEVELOPMENT CO.

FOR

U. S. NAVAL CIVIL ENGINEERING LABORATORY

PORT HUENEME, CALIFORNIA

Under Contract No. N62399-71-C-0011

September 15, 1971

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0.0 GENERAL

0.1 INTRODUCTION

With the ever-increasing use of containers for military cargo and with modern containerships not being self-sustaining, the U. S. Naval Civil Engineering Laboratory has established the need for an Expeditionary Logistic Facility (ELF). The ELF system is to provide for the establishment of port facilities to handle not only containers but all types of cargo from all types of ships where no port facility exists. The ELF must be portable and must be capable of erection in a short time frame.

0.2 OBJECTIVES

The purpose of the present study has been to develop the preliminary design of a concept originally suggested by Eness Research & Development Corporation to insure that it could meet the requirements of the ELF system.

0.3 ORGANIZATION

In the development of the preliminary design of this ELF system, the design of the crane features has been undertaken by Eness Research & Development Corporation and the design of the supporting platform has been the responsibility of the J. J. Henry Company, Inc. Close liaison between the two companies has led to an orderly development of the design.

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SECTION 1

1. HULL

1.0 GENERAL

The platform for the crane is essentially a barge. In this concept, however, the hull is split down the middle so that when the facility has reached the advanced base port, the hulls can be separated to provide a broad, stable platform for the crane.

Since the general and specific requirements for constructing and outfitting a barge are defined in the Rules of the American Bureau of Shipping and the regulations of the U. S. Coast Guard, we will only describe the features which are peculiar to this concept.

1.1 DIMENSIONS

The minimum dimensions of length and beam of the hull are determined by the geometry of the crane structure. As shown on the General Arrangement, Figure 1-1 in the sea condition of transport mode, the crane structure is lowered and the hulls are together. This lowers the center of gravity and makes for greater stability. Also, with the hulls together, resistance for towing or propulsion are decreased.

The minimum depth of the hull is prescribed by the A.B.S. Rules which call for a depth not less than one-fifteenth of the length.

Since the hull serves only as a platform for the crane, minimum dimensions were held in order to keep hull costs down.

The principal dimensions selected are as follows:

Length on load waterline	330'-0"
Beam (hulls together)	60'-0"
Depth of hull	27'-0"
Draft, design	14'-0"
Light Ship weight, no ballast or fuel	2760 tons

1.2 LINES

The lines as developed and as shown in Figure 1-2 were based on the design of an ocean-going barge for which we had model test data.

After making a preliminary weight estimate we found that a reduced block would give adequate displacement. The lines were therefore made finer which reduced hull resistance thus improving speed.

Before finalizing the lines for the prototype we would consider it imperative to prove the lines by model testing.

The hydrostatic characteristics or curves of form are given in Figure 1-3.

1.3 ARRANGEMENTS

The compartmentation of the hull is dictated primarily by the needs for supporting structure for the crane posts. Also, the requirements for subdivision, tankage, auxiliary machinery and quarters must be considered.

1.4 SUBDIVISION

Although it is not yet a requirement of the regulatory bodies or the U. S. Coast Guard for this type of vessel, it is considered desirable to have at least a one compartment standard of subdivision for this vessel. This means that any one compartment can be flooded and the vessel will maintain its stability without excessive list or trim.

Figure 1-4 shows the summary of damaged stability calculations indicating that this hull is satisfactory with adequate stability remaining with any hold damaged.

1.5 STABILITY

Unlike cargo ships which may have many different conditions of loading and, consequently, a variety of stability conditions, we have here basically two conditions to consider. The first, or at-sea condition, is the condition of the facility while enroute to its destination. For this condition the two hulls are brought together and the upper crane structure is lowered to the deck. Figures 1-5A, 1-5B, and 1-5C show the stability for this condition at three possible drafts.

In the working condition with the hulls spread apart, the transverse stability is increased. With the crane raised and the hook load added, however, the combined center of gravity is raised and the stability is slightly reduced, however, since the GM is so large the effect is negligible. Figures 1-6A, 1-6B, and 1-6C illustrate this condition at three possible operating drafts.

1.6 STRUCTURE

The design of the basic hull structure is based on the Rules of the American Bureau of Shipping. Special consideration has been given to seating the crane kingposts so that imposed loads are properly distributed in the hulls.

Drawing 5834-2, Figure 1-7, Scantling Plan shows the basic structural requirements of the hulls.

As a verification of the structure or scantlings, the American Bureau of Shipping requires a still water bending moment calculation. Figures 1-8A, 1-8B, and 1-8C show the bending moment and shear forces at three different drafts. Figure 1-9 shows the calculation of the section modulus of the barge hull and Figure 1-10 gives the calculation of the section modulus required by the A.B.S. The section as designed is satisfactory.

1.7 PIPING SYSTEMS

1.7.1 GENERAL

All piping systems would be required to meet the requirements of the American Bureau of Shipping and the U. S. Coast Guard.

1.7.2 BILGE AND BALLAST SYSTEM

A pumping system with associated piping is required which is capable of draining all compartments and to fill or empty ballast tanks used to correct list or trim of the vessel. Two pumps are required in each hull.

1.7.3 VENT, SOUNDING AND OVERFLOW PIPES

All tanks are to be fitted with vent pipes and overflow pipes.

All tanks and all hold compartments which are not at all times accessible are to be fitted with sounding pipes.

1.7.4 FUEL OIL PIPING SYSTEMS

A piping and pumping system is to be provided which will permit filling the fuel tanks from a convenient location on deck and then transferring the fuel to the diesel generators and to the propulsion units if fitted.

1.7.5 FRESH AND SALT WATER SYSTEMS

A fresh water tank installed in accordance with the requirements of the U. S. Public Health Service would hold water for drinking, washing and cooking purposes. A pressure tank system is used to transfer from the storage tank to the outlets.

A saltwater system, taking water from one or more sea chests, would supply water for engine cooling and sanitary purposes. Modern practice on commercial vessels is to use freshwater for sanitary needs but, since we have no ready source of steam to operate a distilling plant, we would call for saltwater for this purpose.

A general service pump which could serve as one of the bilge and ballast pumps would be used for this service.

1.8 ELECTRICAL SYSTEMS

1.8.1 POWER FOR CRANE

Two sets of diesel-generators of about 250 KW each in the barge supply power for the operation of the crane. One set is adequate for normal operation and the other provides 100 percent standby.

These generators would also supply power for the anchor windlasses, the constant tension winches and the crane lifting winches. From the switchboards in the diesel-generator room, cables carrying power to the cranes would be led under deck and then up one of the kingposts.

1.8.2 POWER FOR THE BARGE

One diesel-generator of about 75 KW would supply power requirements of the barge. In the event of failure, power could be taken from either of the crane generator sets.

Switchboard in diesel-generator room would distribute power to:

Lighting, including navigation lights.
Power for service pumps.
Air conditioning and ventilation/heating.
Galley ranges and appliances.
Communication equipment.

1.9 MOORING AND ANCHORING

Windlasses, anchors and chain are to be provided in each hull to suit A.B.S. requirements: One anchor is sufficient to hold the barge. Both anchors would be used when transferring to lighterage offshore to insure sufficient holding power for both the facility and the lighterage.

Constant tension winches are provided to secure to a pier, another vessel or to lighterage. The winches are arranged to be led to either side of the separated hulls.

1.10 FENDERING

Shock-absorbing fenders are to be provided all around each hull to protect the facility from other vessels either outboard or between the hulls.

1.11 SECURING THE HULLS

In the transport mode with the hulls together, the hulls are secured by hydraulically operated pins acting in fender-shaped pieces at the centerline as shown on Figure 1-11. The securing lugs are wedge-shaped to adjust for any misalignment when the hulls are brought together.

Also shown on Figure 1-11 is the T-shaped slot in the hulls to accommodate the cross connecting members, shown on the Eness drawings, which hold the separated hulls together.

1.12 MANNING AND QUARTERS

The U. S. Coast Guard, which has jurisdiction over manning of vessels, has no definite rules on the manning of a vessel such as this facility but considers each case individually.

The crane specifications indicate a need for a minimum of two crane operators plus a maintenance electrician and a maintenance mechanic. Since this is a small facility with basically two modes of operations, the at-sea mode and the operating mode, it is believed that the crew could be assigned dual duties. Our estimate of required manning, using arbitrary designations is as follows:

Captain		1
Mates		2
Seamen		6
Machinists		3
Electricians		2
	Total	14

Since we were advised that the crew would be military we have provided two-man rooms except for the upper ratings. The quarters would be insulated and sheathed and provided with metal furniture. Mess room, galley, reefer and dry stores space are provided. Provision is made for venting, heating, and air-conditioning the quarters.

1.13 SAFETY EQUIPMENT

As required by the U. S. Coast Guard, the facility would be provided with one diesel-propelled work/life boat and inflatable life rafts and other lifesaving gear as required.

Fire stations tied into the ship's salt water systems and portable extinguishers are to be provided.

1.14 PROPULSION

In the original concept by Eness, the crane is mounted on a towed barge. Since the Naval Civil Engineering Laboratory (NCEL) was also interested in a self-propelled facility we have also investigated the possibility of using Murray and Tregurtha type outboard units which involve minimum changes in the basic design.

Figure 1-12 shows a curve of Effective Horsepower (EHP) versus speed in knots for the hull.

Figure 1-13 shows two curves of horsepower (BHP of tug) versus speed, one for barge with skegs and one for barge without skegs but with two 1000 HP outboard units which are used for self-propulsion or to supplement the tugs and control the barge. No skegs would be fitted on the barge with propulsion units as the units could be sterred to overcome yawing.

From these curves we find that the self-propelled version with no assistance from a tug would have a speed of about 10 knots.

We found little published data on speed and horsepower of tugs. We have, therefore, constructed a series of curves for a range of tug sizes and horsepower based on published data to determine the size of tugs necessary to tow the facility at various speeds. These are given in Figures 1-14A through 1-14H.

From Figure 1-13 we find that to achieve a speed of 14 knots a tug of about 10,000 HP is required. If propulsion units on the facility are also used, the same speed of 14 knots could be obtained with a tug of about 6000 HP. The reason for this is that the hull resistance is reduced with no skegs and, of course, the added thrust from the outboards.

We would recommend that the facility be fitted with the outboard units, not only because of improved towing speed, but also because we believe they would be useful for positioning the facility at the site.

1.15 TRANSPORT MODE

For the movement of the facility to its destination, it is important that a decision be made whether the barge is to be towed only or if it is to have some degree of self-propulsion. If it is only to be towed, it will be necessary to fit skegs to give the tow directional stability. As noted in Article 1.14 (PROPULSION) we have recommended propulsion units even if towed, to eliminate the added resistance of the skegs and to supplement the tug.

Because of its fine entrance and lines, and its wide beam, the barge can be expected to perform well at sea. Of course, the normal precautions of reducing speed and heading into the seas would be recommended at higher sea states.

With the crane lowered to the deck for the transport mode there is ample transverse stability. If rolling does start at higher sea states, it is recommended that ballast in the form of seawater be pumped into two or more of the hull tanks to reduce the transverse metacenter or GM and thus increase the roll period. With too large a GM there is danger of quick snap rolls which would be uncomfortable for the crew and possibly dangerous to the equipment.

1.16 OPERATING MODE

With the hulls spread apart and the crane ready for operation, the transverse metacenter is quite large and heeling due to transfer of loads is small.

The facility can be used to straddle a finger pier, can be moored alongside a pier, or can be anchored in a river or in open water.

From a stability standpoint there are no problems as the sea state increases. The difficulties are in the crane structure and in the cross ties. This is fully discussed in Section 2 on the crane. Beam seas would, of course, tend to push the hulls together, especially if the barge is moored alongside a pier. This is considered in the crane design and provision is made for cross members at the bottom of the barge. Head seas or quartering seas tend to rack the crane structure and this is considered in the crane design, up to the limits prescribed in the crane design calculations. At higher sea states it is recommended that the crane be lowered and the barge halves brought together so that the facility could ride out the storm at anchor. In this condition we believe the barge could survive very heavy weather.

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SECTION 2

2.0 CRANE

2.1 GENERAL DESCRIPTION OF CRANE

The crane consists of several basic components; these include a trolley which houses the hoist and transverse propelling machinery and supports a control cab on each of two (2) sides, extensible booms which move transversely over the ship, dock or lighters and serve as supports for the trolley, and a bridge frame which completes the crane unit by supporting the extensible booms. The crane unit is supported at two (2) points by wheeled assemblies which permit horizontal and vertical pivoting; rack and pinion drives are provided to move the crane along the fore and aft links. The ends of the fore and aft links are integrally connected to collars which are able to move vertically on the kingposts. Additionally, there are transverse links which are pinned to the collars and serve to maintain the transverse distances required by acting as a system of parallel links. The collar, as already noted, serves as a means of raising the system along the kingposts and also houses mechanical locks to maintain the desired elevation and to relieve the load from the crane hoist tackles. Four collars are provided, one for each kingpost. The kingposts provide a means of supporting and raising and lowering the crane unit and also serve to take out the various loads resulting from sea motion.

The material presented herein describes the basic design of two (2) cranes, each having particular advantages and disadvantages which will be discussed in detail. Design Nos. 1 and 2 have the same basic kingposts, fore and aft links, transverse links and hoist machinery but differ in the extensible boom arrangement. Design No. 1 has two (2) extensible booms which are extended and set to each side of the desired outreach. In comparison, Design No. 2 has one extensible boom which moves transversely concurrent with the trolley; the trolley travels twice as far as the boom. The trolley of Design No. 1 is propelled by traction along the extensible boom and its motion is independent of the motion of the boom. The trolley of Design No. 2 is propelled by four (4) fixed lengths of wire rope each of which have one end attached to the trolley, then led through a sheave on the extensible boom end and then have the other end attached to the bridge structure. Design No. 1 is depicted on Eness Dwg. No. 71031-3 and Design No. 2 is depicted on Eness Dwg. No. 71031-13.

2.2 GENERAL PERFORMANCE CHARACTERISTICS

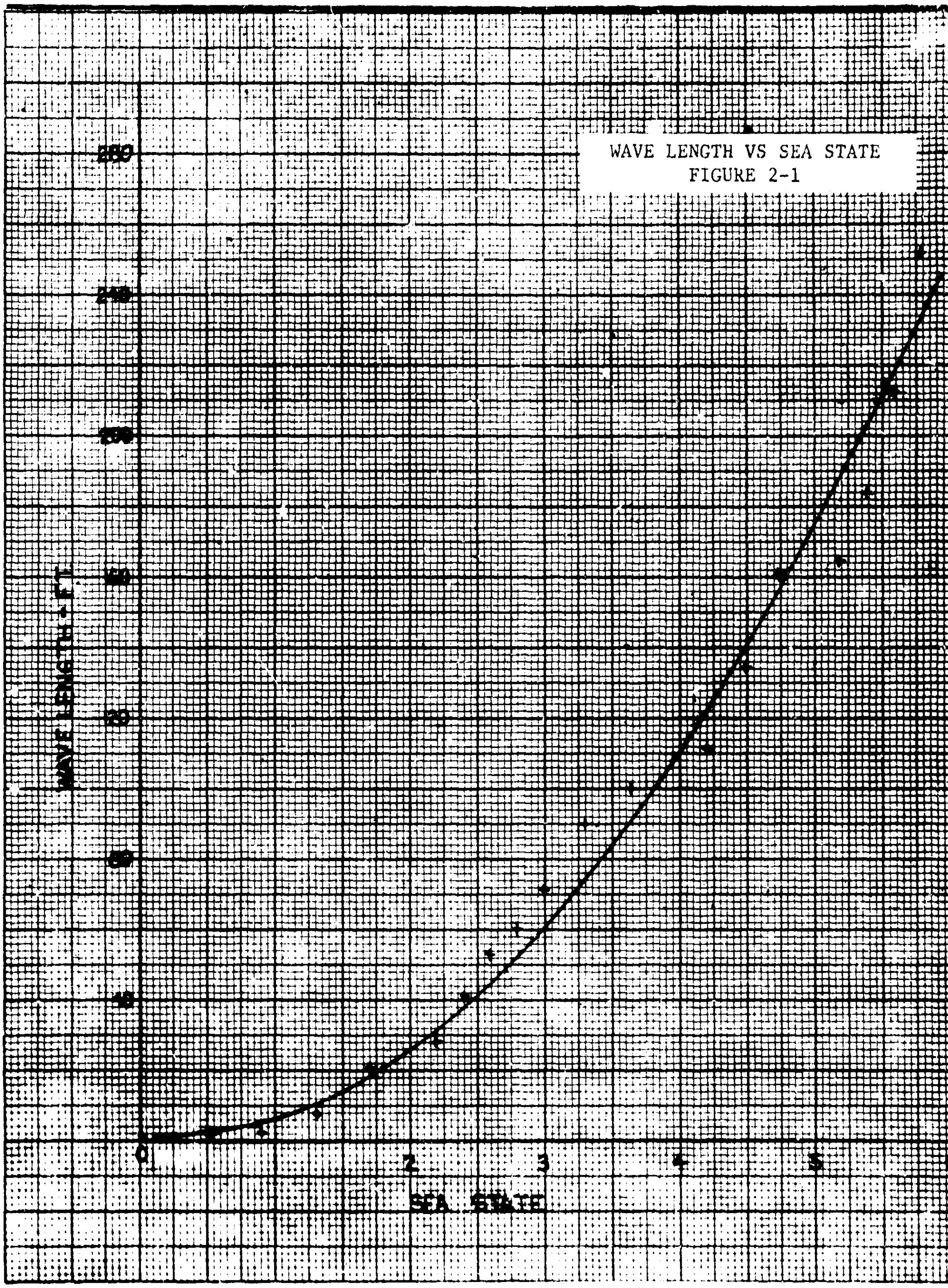
The crane is capable of projecting transversely over the hatch of a container ship, roll-on/roll-off ship, and general break-bulk type of cargo ship so as to spot over the load point and transfer loads such as containers, military vehicles, and palletized cargo to lighters, docks or causeways, as may be required. The cargo path is generally as indicated on Eness Dwg. No. 71031-1, which shows a distance of 70 feet 6 inches from the underside of the container to the waterline. The maximum outreach is 96 feet and the transverse span is 342 feet between extreme hoist points. A 44,000 pound load can be lifted, transferred, and deposited between these points in less than five (5) minutes. For the purpose of handling military vehicles weighing 70,000 pounds, the crane is capable of an outreach of 70 feet and a transverse space of 290 feet.

The following criteria has been used in establishing the concept:

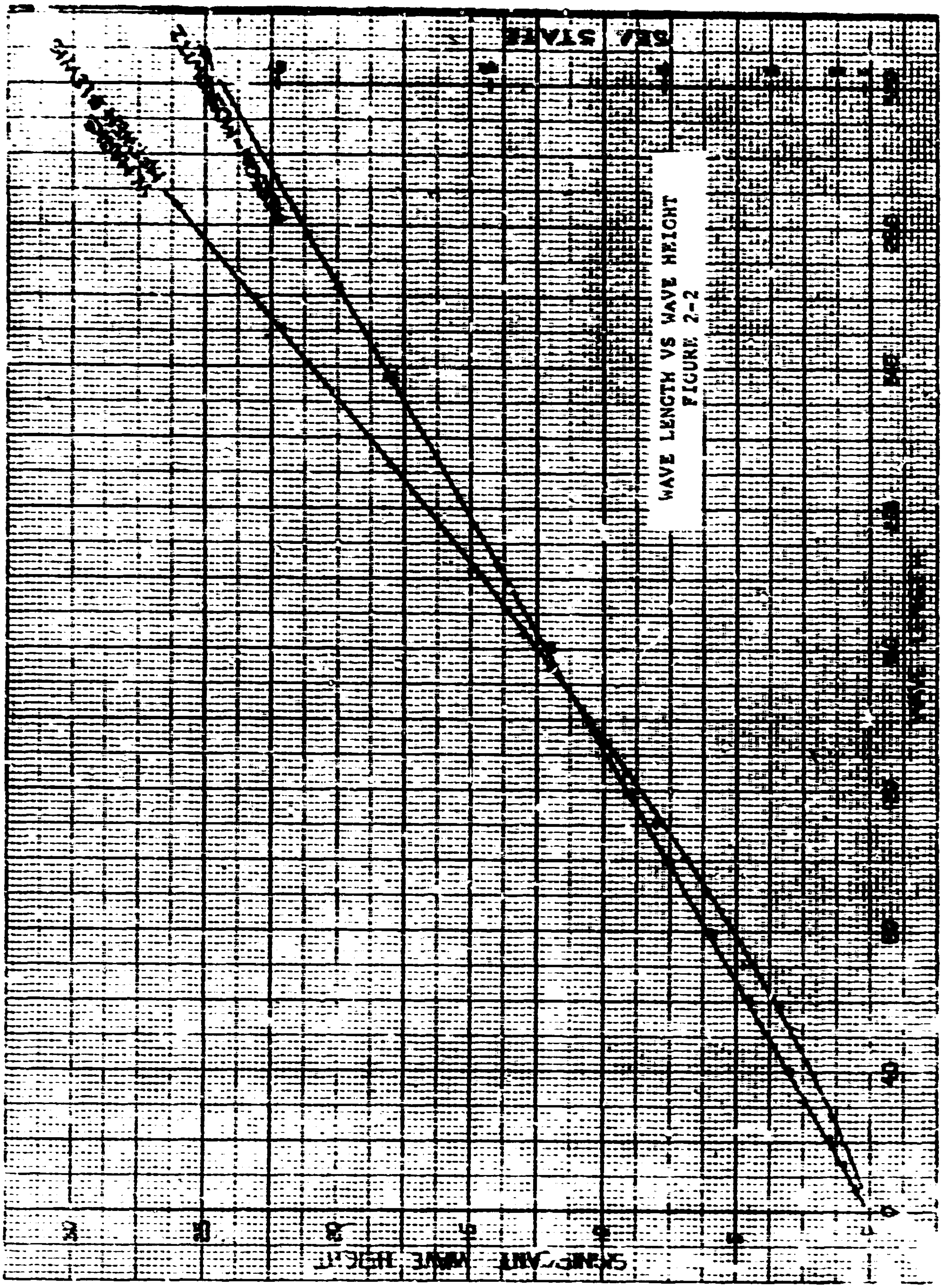
Hoist - 44,000 pounds at 115 FPM
Transverse Travel - Up 1 degree incline at 400 FPM
Boom Extension (Design No. 1) - Up 1 degree incline at 50 FPM
Fore and Aft Travel - Up 1/2 degree incline at 70 FPM

The hoist and transverse travel speeds are dictated by the requirement for 20 containers per hour through the average path. The boom speed for Design No. 1 is arbitrary and can be increased or decreased; however, the boom speed for Design No. 2 must be related to the trolley speed required to handle the 20 containers per hour. The fore and aft travel speed is also arbitrary and may be varied with subsequent change in horsepower of the drive machinery. The fore and aft travel is 76 feet from center to center of the containers and will permit serving four (4) 20 feet cells without shifting the barge relative to the container ship. The hoisting of the crane to operational position is accomplished with four (4) 50 HP winches; the rate of hoisting is predicated on the structural design and the maximum sea state at which operations will take place.

Review of the sea state data as defined by W. A. McEwen and A. H. Lewis, Vine and Volkmann, Wilbur Marks and Pierson-Moskowitz indicates that Marks is in agreement with McEwen and Lewis. Vine and Volkmann are also in good agreement for sea states less than three (3) but rely upon the "average wave height" for the "equivalent wave length". Comparing Pierson-Moskowitz wave heights and wave lengths with corresponding values given by Marks, the "significant wave height" of the former exceeds the "average 1/10 highest" for sea states less than high four (4), but, at low five (5), they are comparable with Marks' "significant wave height". At mid sea state six (6) the "significant wave height" of Pierson-Moskowitz becomes slightly less than Marks' for the respective wave length and decreases further as sea states increase. For the purpose of the crane designs discussed hereafter, the effect of sea state is predicated upon Marks and McEwen and Lewis. Figure 2-1 presents a plot of wave length versus sea state; Figure 2-2 shows significant wave height plotted against wave length. In order to relate the design to a common reference, wave length will be used and from Figure 2-2 corresponding wave heights can be determined. When wave heights and lengths are known, the design can be related to the desired source of reference for sea state.



WAVE LENGTH VS SEA STATE
FIGURE 2-1



WAVE LENGTH VS WAVE HEIGHT
FIGURE 2-2

2.3 COMPONENTS

Eness Dwg. No. 71031-12 shows the crane of Design No. 1 in the stowed condition and identifies the various components which comprise the assembly. Further, the drawings which detail the various components are also given for ready reference. Design No. 2 is very similar to Design No. 1; Dwg. No. 71031-13 illustrates the areas of difference between the two (2) designs.

2.3.1 TROLLEY

The trolley consists of an enclosure which houses the hoist equipment and travels transversely on the extensible boom. On each end of the trolley is a control cab. In operation, the cab most convenient for viewing may be used. The controls in each cab will be tied together; however, it will not be possible to arbitrarily override an active control by remote control unless a change in selection is made.

The main support structure will be ASTM A441 low alloy steel; the remaining structure will be aluminum. Where aluminum and steel are in contact, effective means of isolation will be required. Various compounds and tapes are available for this purpose.

The trolleys for Design Nos. 1 and 2 differ in that Design No. 1 has incorporated, within the trolley, the propulsion equipment to move the trolley transversely, while Design No. 2 does not require this type of equipment.

2.3.2 TROLLEY DRIVE

The two (2) designs presented herein differ in the trolley drive. Design No. 1 has an independent trolley drive which is illustrated on Eness Dwg. No. 71031-4; this drive consists of two (2) motors and reducers driving high traction, solid rubber industrial tires such as General size 10 1/2 x 6 x 5, having a friction coefficient of 0.5 when wet to 0.85 when dry. Each motor, fitted to a reducer, has a disc-type brake; in the event of a motor failure, release of the brake will permit operation at a reduced rate.

The trolley weight, including the hoist machinery, cab, controls and spreader is estimated to be 40,000 pounds for Design No. 1 with drive machinery, and 32,000 pounds for Design No. 2 without drive equipment. The total design load for Design No. 1 is 110,000 pounds, while for Design No. 2 it is 102,000 pounds. The traction force required to move the trolley will depend upon the list and the wind force. A sustained list of 1 degree will be used. The wind force, F_W, is a function of the wind velocity, V, in knots and the exposed surface area, A = 170 square feet as follows:

$F_W = 0.004\ AV^2$ (U. S. Navy General Specifications)

The force due to list $F_L = T_L \times \text{Sin (1 degree)}$

The force due to rolling contact, where R ≈ 6" for a 12" diameter wheel is $F_R = T_L \times \frac{0.005}{6}$

The total force is $F_T = F_W + F_L + F_R$

Assuming a mechanical drive efficiency of 0.6, the horsepower is:

$$HP = \frac{F_T \times 400}{33{,}000 \times 0.6}$$

Based upon appropriate wind velocities for wave lengths of 20, 40, 71, 99, and 160 feet, the total forces and horsepowers are plotted on Figure 2-3. On the basis of this information, two (2) 25 HP 1750 RPM motors were selected; each of these is coupled to a Falk GHB 2050 reducer having a 2.76: 1 speed ratio. Further reduction is accomplished through a chain drive having a ratio of 4:1. The transversing drive will have two (2) speeds, full and one-quarter.

Manufacturer's data on the traction wheel indicates a maximum load of 2200 pounds per wheel with footprint of 10.4 square inches. To satisfy the maximum required transverse force of 2350 pounds from two (2) traction wheels, the required friction coefficient will be:

$$F = \frac{2350}{4400} = .535$$

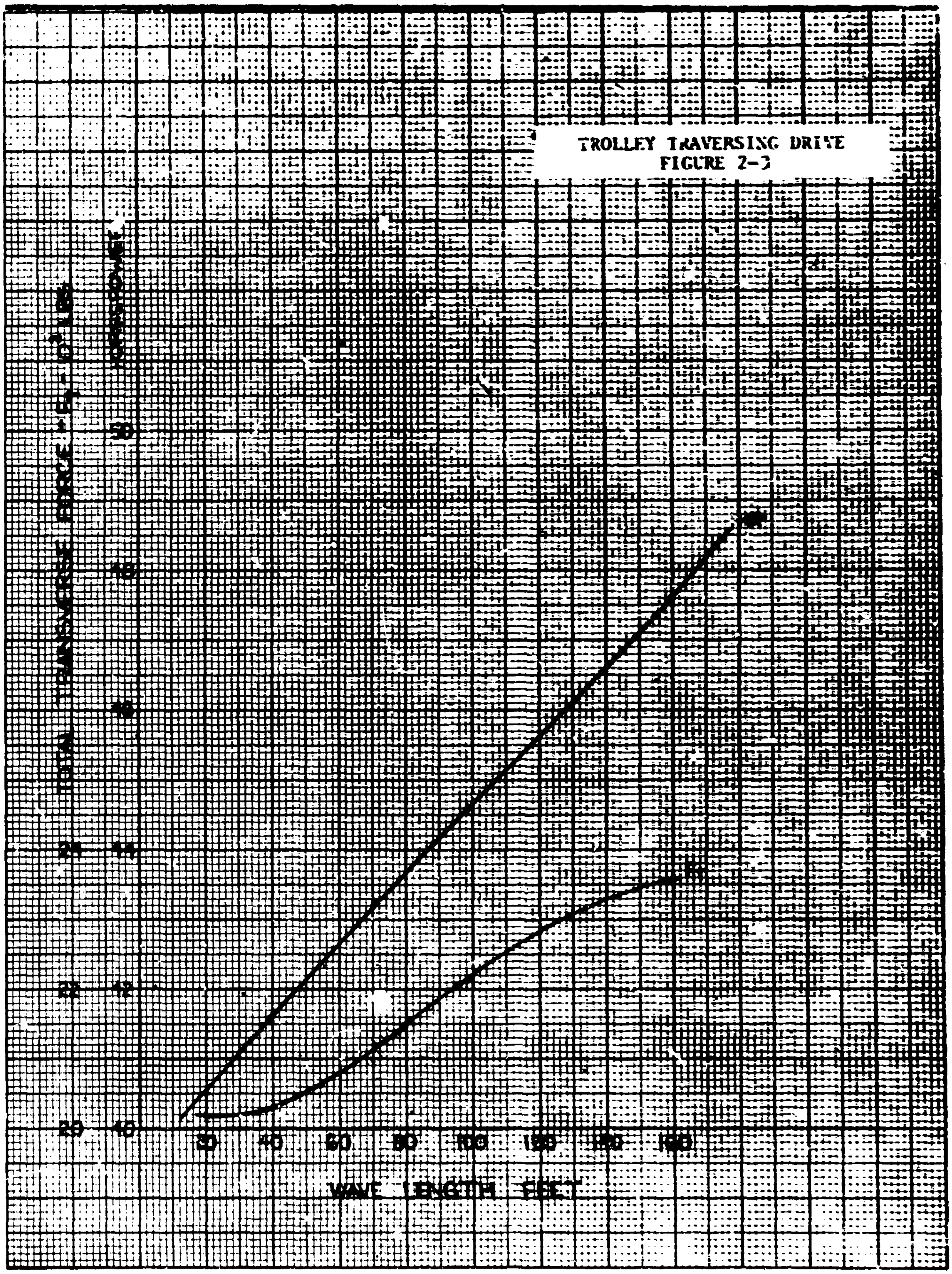
The required friction should be satisfied since contact is made on the underside of the extensible boom so that there is a high probability that it will be dry.

The trolley drive for Design No. 2 is covered in the discussion of the extensible boom for that design.

2.3.3 MAIN HOIST DRIVE

The main hoist drives are located in the trolley and consist of two (2) each of motors, gear reducers and drums. The hoisting is accomplished by using a pair of 3/4 inch 6 x 37 high strength wire ropes with double part reeving to sheaves at the four (4) points of the spreader. In the event that one (1) motor fails, the operation may continue at a reduced rate provided the brake is set on the inoperative motor.

The distance over which a container will move through its average path, for the purpose of the time cycle, is taken as the sum of the mean maximum vertical and horizontal distances. Mean maximum vertical distances of 54 feet over the vessel and 70 feet over the lighter or dock and a horizontal distance of 170 feet were used.



TROLLEY TRAVERSING DRIVE
FIGURE 2-3

Tables IA and IB indicate the breakdown of each element of the cycle time starting with the hoist in the ship, depositing the load on the dock and returning to the start. The hoist and lowering speeds are derived from Figure 2-4 using two parts in the hoist.

The ideal cycle time is one which can be obtained under ideal conditions such as in a sea state of one. It should be noted that twenty-four (24) containers of 44,000 pounds can be handled in one (1) hour. This quantity, when reduced to twenty (20) to meet requirements, allows approximately 15 percent of each hour for operator break or change. The manning discussed on page requires two (2) operators for each eight (8) hours of operation so that fatigue should not be a problem. Also recommended is a training period for the operators. In the training, sufficient experience must be had before the operator is permitted to encounter actual operations.

Relative to degradation of performance as affected by sea states, there is considerable variance with respect to cycle time, so that parameters are difficult to establish. Based upon present experience, one can say that the ideal cycle time can be related to sea state as shown in Figure 2-4A and is obtained during a sea state of zero (0) to one (1).

TABLE IA. BREAKDOWN OF IDEAL CYCLE TIME FOR CONTAINER HANDLING (44,000 POUNDS)

Ship to Shore

Operation	Time (sec.)
Hook On	4
Acceleration (hoist)	3
Hoist	28
Dwell	1
Acceleration (transversing)	3
Travel (transverse)	25
Braking	4
Dwell	1
Acceleration (lower)	1
Lower	26
Brake	2
Unhook	1
	99

TABLE IA. BREAKDOWN OF IDEAL CYCLE TIME FOR CONTAINER HANDLING (44,000 POUNDS) (con't)

Shore to Ship (empty spreader)	Time (sec.)
Acceleration (hoist)	2
Hoist	7
Dwell	1
Acceleration (transversing)	2
Travel (transverse)	25
Braking	2
Dwell	1
Acceleration (lower)	1
Lower	9
	50

Total time = 2 1/2 minutes = 24 containers per hour.

TABLE IB. BREAKDOWN OF IDEAL CYCLE TIME FOR VEHICLE HANDLING (70,000 POUNDS)

Ship to Shore

Operation	Time (sec.)
Hook On	20
Acceleration (hoist)	3
Hoist	39
Dwell	1
Acceleration (transversing)	3
Travel (transverse)	37
Braking	5
Dwell	1
Acceleration (lower)	1

TABLE IB. **BREAKDOWN OF IDEAL CYCLE TIME FOR VEHICLE HANDLING (70,000 POUNDS) (con't)**

Operation	Time (sec.)
Lower	35
Braking	3
Unhook	8
	156
Shore to Ship (empty spreader	50
	206

Total time = 3 1/2 minutes = 17 vehicles per hour.

The following calculations establish the minimum horsepower for the hoist drive:

$$\text{Horsepower, HP} = \frac{(\text{Live Load} + \text{Spreader})}{33{,}000 \times .7 \times 2} \times \text{Speed} \quad \text{(Per Motor)}$$

$$\text{For Containers: HP} = \frac{54{,}000 \times 115}{33{,}000 \times .7 \times 2} = 135 \text{ HP each Motor}$$

$$\text{For military vehicles: HP} = \frac{80{,}000 \times 83}{33{,}000 \times .7 \times 2} = 144 \text{ HP each Motor}$$

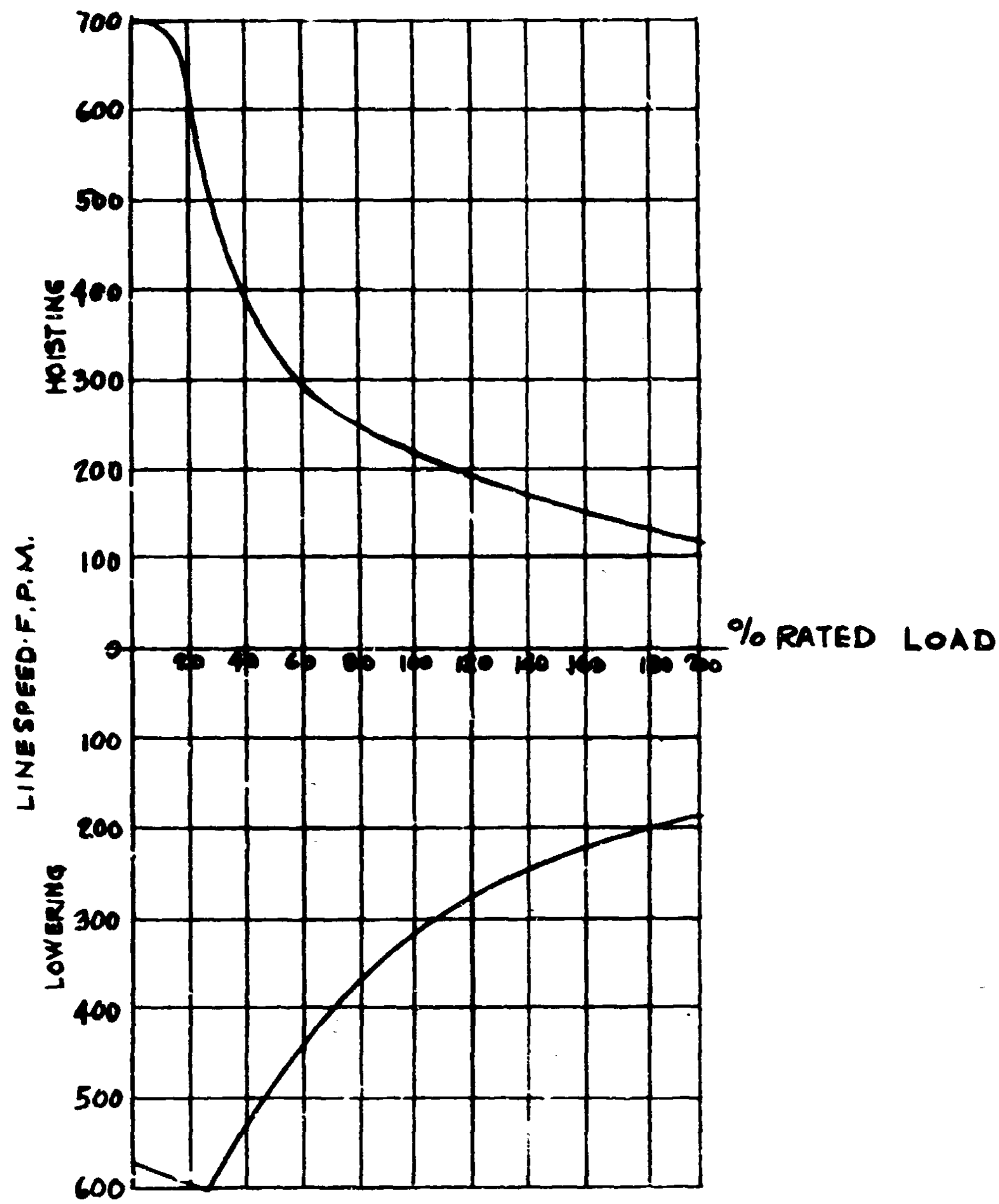
As the container operation takes primary consideration, selection of 150 HP motors will provide more than adequate margin for safety and reliability.

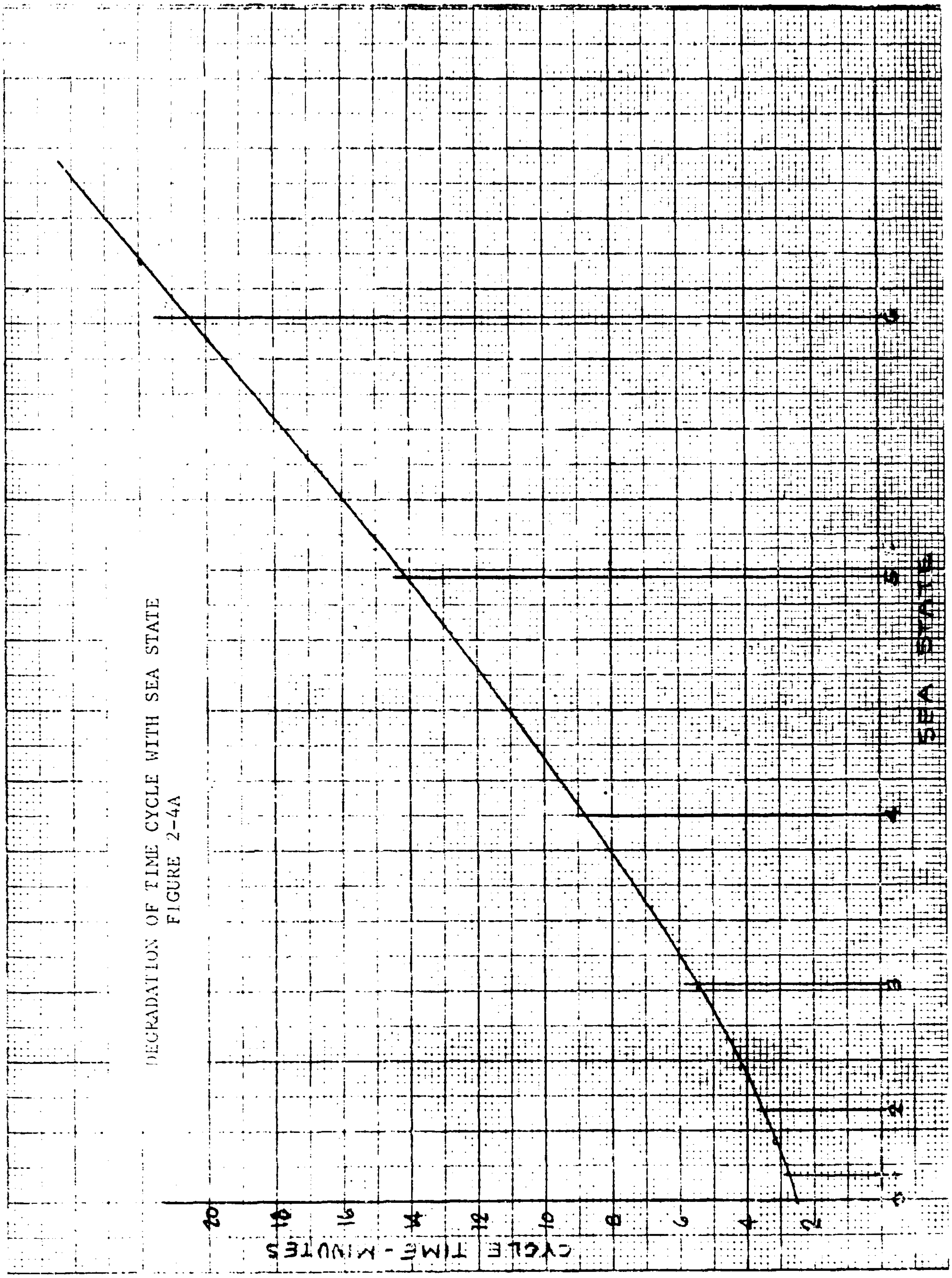
Motors are rated 150 HP at 850 RPM, coupled through a G. E. Brake A 104 to a Falk YB 2100 reducer with a ratio of 13:95. The drive will be a stepless type.

2.3.4 EXTENSIBLE BOOM. The booms of the two (2) designs differ in concept and operation. The basic structure and strength are the same and are of ASTM A441 - low alloy steel. See 2.10 for discussion on material.

2.3.4.1 DESIGN NO. 1. The boom structures are of the box girder type with top and bottom plates acting as flanges extending beyond each side of the web plates which serve as a means of supporting trackways for the trolley as well as for its wheel supports and guides. The booms are shown on Eness Dwg. No. 71031-3, Sheets 1 and 2. Referring to this drawing, it will be observed that the starboard boom is narrower than the port boom and is arranged for support from wheels attached to a bridge which spans the fore and aft links between the kingposts. The port boom has a similar but opposite configuration and is wider so that the starboard boom rests within the port boom when both are retracted. As shown in elevation 1-A, the trolley travels

LINE SPEED VS PERCENT OF RATED LOAD
MAIN HOIST DRIVE
FIGURE 2-4





DEGRADATION OF TIME CYCLE WITH SEA STATE
FIGURE 2-4A

along the lower flanges of the booms, while the upper flanges are used for support of the boom. Support of the boom is accomplished on wheels on the outboard extremities of the bridge structure which are shown in Section 2-A of Sheet 2 of referenced drawing. The inboard extremities of the booms are also supported in the vertical direction by additional wheels on the cross structure of the bridge shown in Section 2-B of Sheet 2 of referenced drawing. Each boom can be extended to one side of the centerline so that the trolley can travel transversely 342 feet with ransfer taking place in the area of boom overlap at the centerline. Each boom has individual extension machinery permitting any desired extension up to the maximum on any one side or on both sides.

2.3.4.2 DESIGN NO. 2. The boom for this design is similar to the port boom of Design No. 1 except that it is closed on both ends. The top and bottom plates also extend beyond the webs and serve as a means of supporting trackways for the trolley and its wheel supports. The arrangement of this boom is shown on Eness Dwg. No. 71031-13, Sheets 1 and 2. On this drawing, the boom is shown extended to one side; movement of the boom to the opposite side also causes the trolley to move simultaneously to that side. Support for the boom is also from wheels located on the cross structures of the bridge. The boom has two (2) sets of extension machinery which will permit operation in event of a motor failure if the couplings are released.

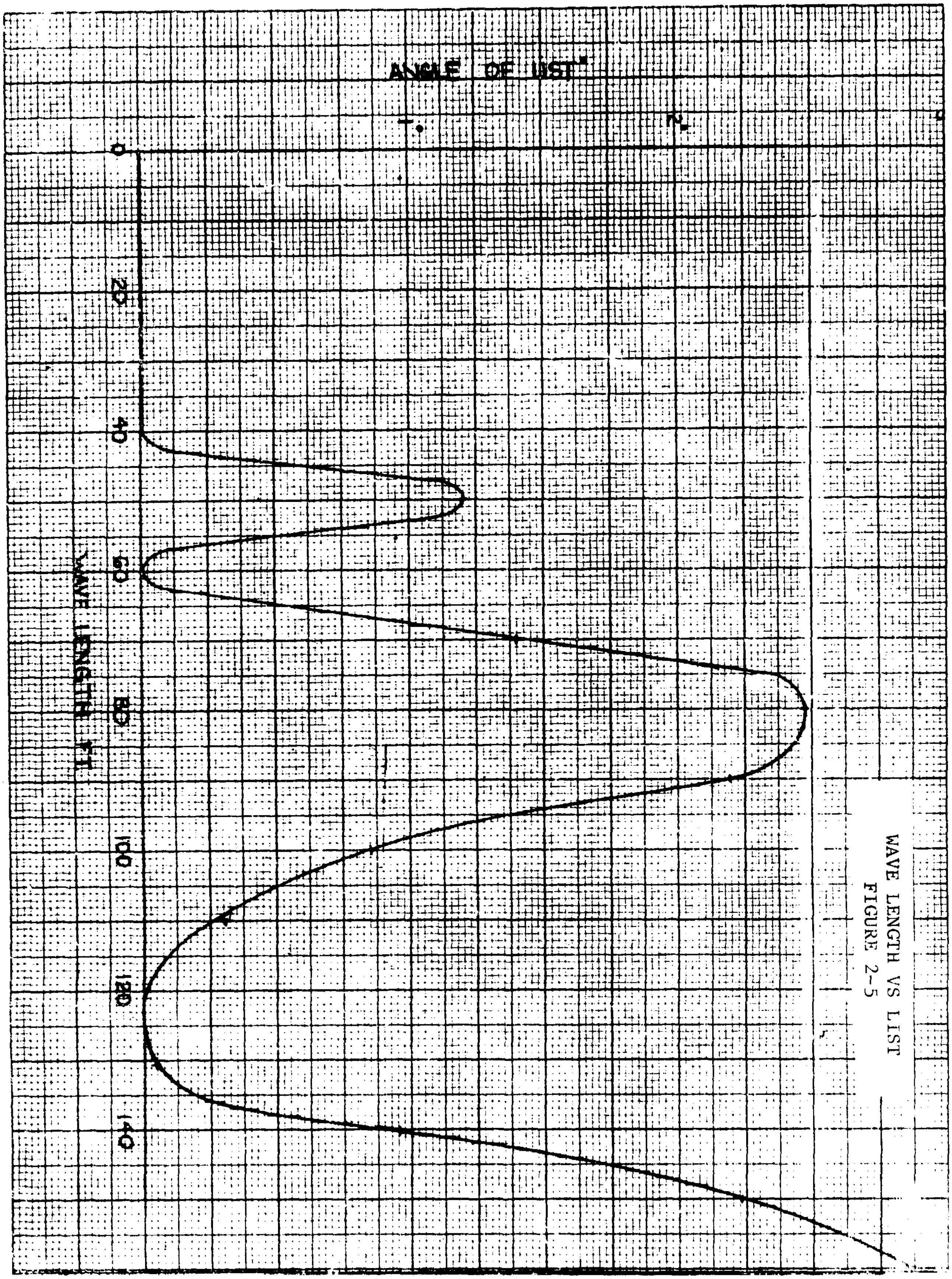
2.3.5 EXTENSIBLE BOOM DRIVES

This drive differs for Design Nos. 1 and 2. The power required for the drives will, in part, depend upon the list encountered in the various sea states which is affected by the wave length. The variation of list with wave length is shown on Figure 2-5. Listing, due to wave motion, will be for short duration and can be absorbed by the motor overload capacity for wave lengths up to 70 feet if the design is based upon a sustained list of one (1) degree.

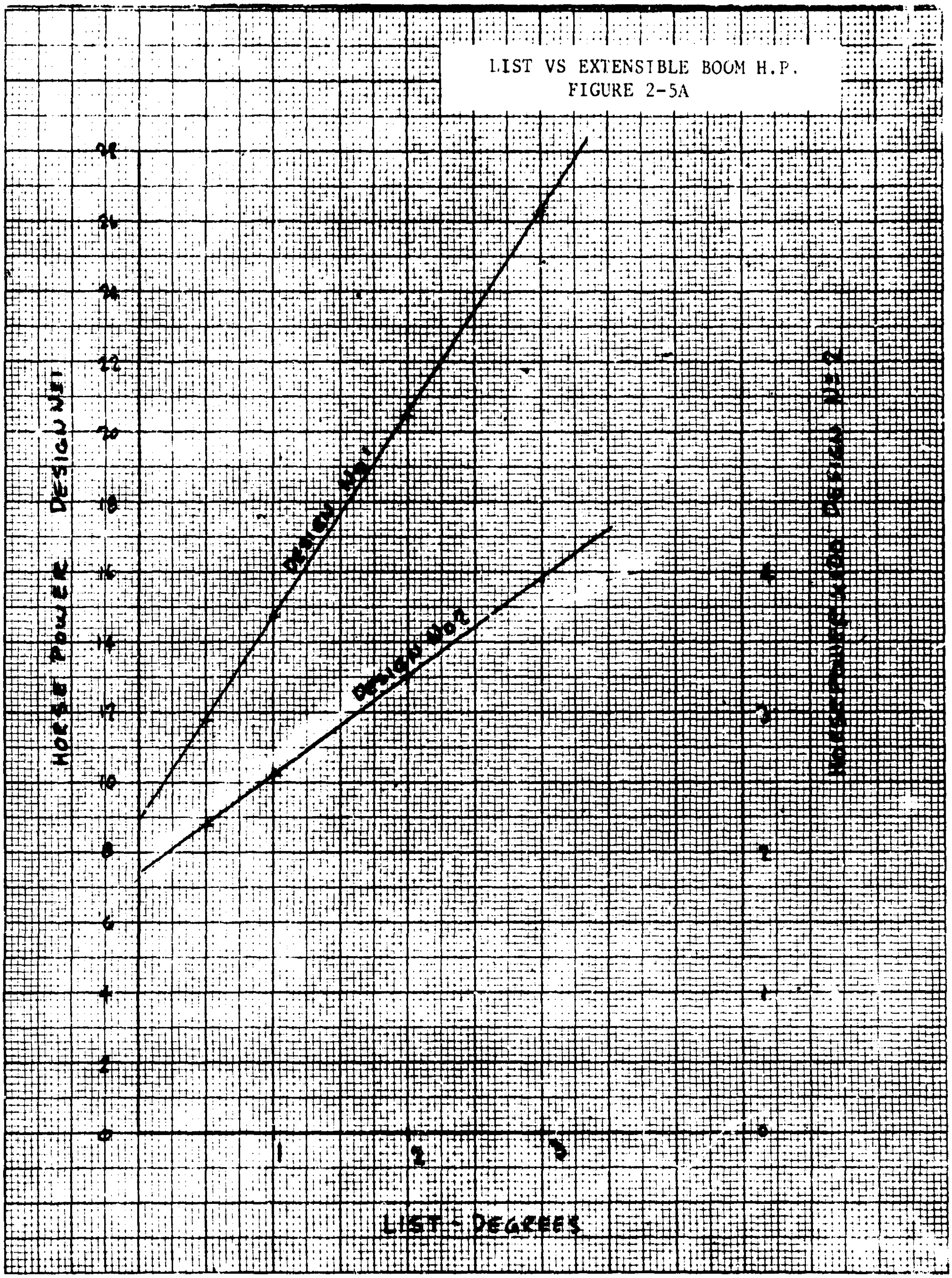
2.3.5.1 DESIGN NO. 1. The drive for each boom consists of two (2) sets of racks and pinions drives driven through a gear reducer by an electric motor having a disc-type brake. The boom drives are located on the bridge cross structure.

Figure 2-5A indicates the horsepower required for various degrees of list. The typical calculations for 1 degree list in Appendix A indicates 15 HP. Based upon usual Navy practice, this horsepower will be adequate for lists up to 2 1/3 degrees which is in excess of sea state 3.

Each of the two (2) boom drives shall consist of a 15 HP 1200 RPM motor coupled to a Falk Y2 reducer, size 2050, with a 47:08 ratio. Each motor is to be fitted with a Sterns disc brake, Model 1-087-081-X, Style 87000 (Cap. = 85 feet - pounds).



WAVE LENGTH VS LIST
FIGURE 2-5


LIST VS EXTENSIBLE BOOM H.P.
FIGURE 2-5A
HORSE POWER DESIGN NO 1
DESIGN NO 1
DESIGN NO 2
LIST - DEGREES

2.3.5.2 **DESIGN NO. 2.** The drive for this boom also consists of a rack on each girder driven by a pinion through a gear reducer and motor having a disc-type brake.

Figure 2-5A indicates that 250 HP will be suitable under a permissible overload for lists up to about 2 3/4 degrees which is also in excess of sea state 3.

Two (2) 125 HP 1200 RPM motors are selected. They are each coupled to a Falk Y2 reducer, size 2100, with a 47:08 ratio. Further, each motor is to be fitted with a G. E. brake, A 104.

2.3.6 BRIDGE

The bridges shown on Eness Dwg. Nos. 71031-3 and 71031-13 for Design Nos. 1 and 2, respectively, are structurally the same but differ in the number of wheel supports. Structurally, the bridge is a frame of boxed girder construction having two (2) main transverse members maintained parallel to each other by a cross structure at the center and at each end. The center cross structure supports the wheels and machinery for the boom propulsion; the end cross structures support the wheels for the boom as it is extended beyond the barge. Two (2) intermediate cross structures are fitted with vertical and horizontal pivots which serve as the interface between the bridge and the fore and aft links between the kingposts. The purpose of the pivots is to permit articulation when separating the barge hulls and to prevent any distortion to the bridge in event of a difference in pitching motion between the barge hulls.

2.3.7 BRIDGE SUPPORT AND DRIVE

The bridge, shown on Eness Dwg. No. 71031-5, is supported at each end on the flanges of the fore and aft links by a power driven trolley-type unit which has wheels for support of the load from the bridge and a pinion which engages a rack on each fore and aft link. The power required for these drives will in part depend upon the trim to be encountered. Based upon wave length, the trim that may be expected varies; however, it closely follows the curve indicated in Figure 2-6. For the purpose of powering, a sustained trim of 0.5 degrees will be used. The fore and aft travel is possible under load at the rate of 70 FPM because of this small trim.

Maximum load on 4 - 20" wheels = 450K.

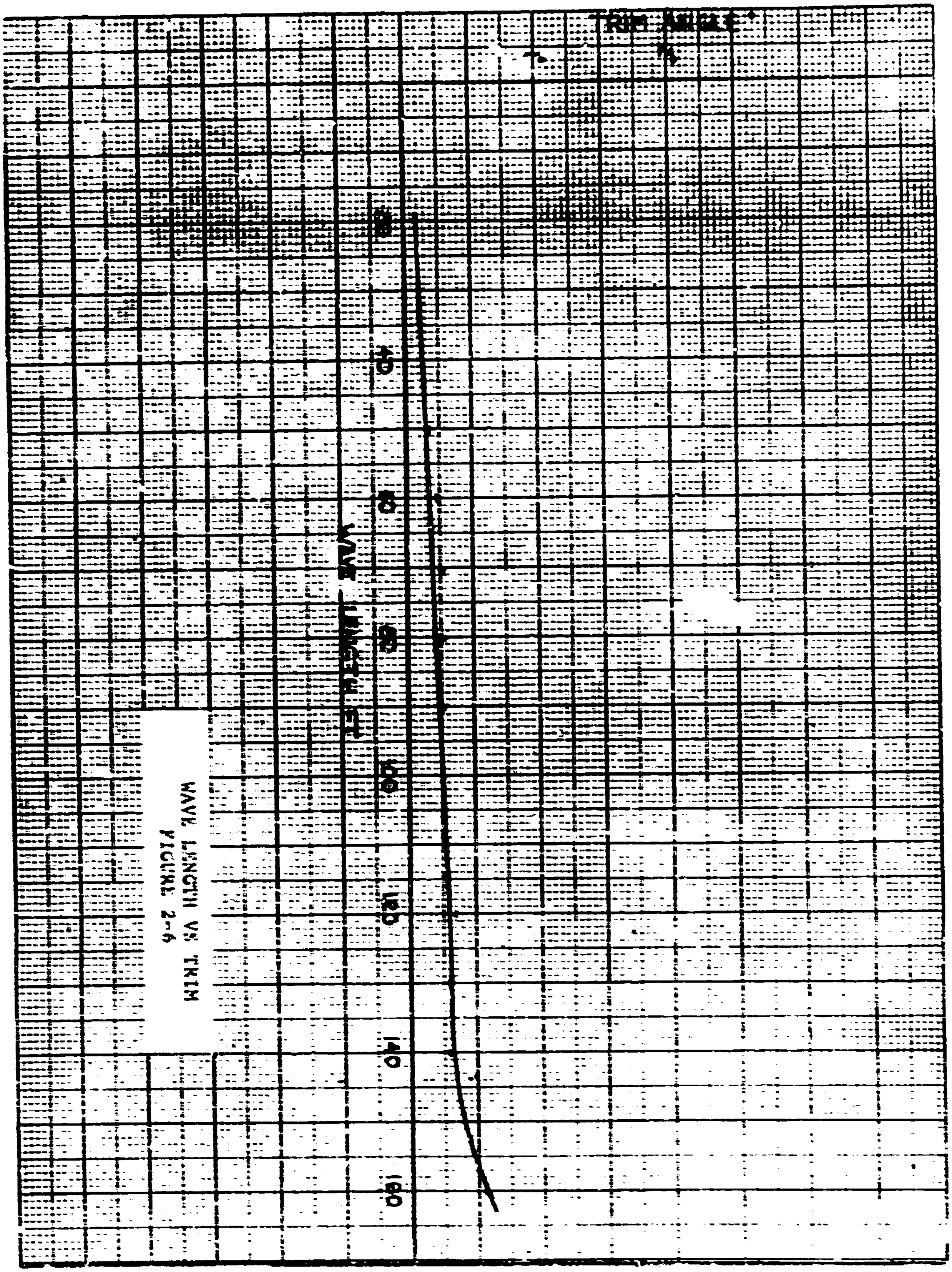
Force, F_R, due to rolling contact is determined as follows:

$$F_R = 450 \times 100 \times \frac{.005}{10} = 225\#$$

Force, F_t, to overcome a .5 degree trim:

$$F_t = 450 \times 1000 \times .0087 = 3920\#$$

$$F\ total = 4145\#$$



WAVE LENGTH VS TRIM
FIGURE 2-6

The total drive efficiency, f, is assumed to be 0.6, therefore, required HP = $\frac{4145 \times 70}{33,000 \times .6} = 14.7$. Hence, select a 15 HP, two (2) speed, 1200 RPM motor coupled to a Falk worm reducer, AU 800, having a ratio of 30:1.

Inasmuch as the maximum trim of 1/2 degree exists at wave lengths above 150 feet (Figure 2-6) the 15 HP is more than adequate.

2.3.8 FORE AND AFT LINKS

The fore and aft links are box type girders, each integral with and extending between two (2) elevating collars on the kingposts as shown on Eness Dwg. No. 71031-6, Sheets 1 and 2. The top and bottom plates of the girders extend beyond the web plates and the bottom plate supports the bridge support machinery. The collars extend beyond the kingposts to provide pivots for the transverse links. Each end of each fore and aft link in way of the collar has a sliding lock which retracts to permit changing the elevation of the links; once at the desired position, the locks are again extended so as to support the full load of the crane and links. The collar is arranged with shoe type slide bearings and moves vertically along a guide bar on the kingposts. Attached to the collar is a wire rope tackle which raises and lowers the fore and aft links; the wire rope leads are led down the inside of the kingposts to four (4) winches in the hold of the barge. The winches have split drums to accept the ends of each tackle which is endlessly rove through the sheaves and hoisting blocks shown on the above referenced plans. The total weight of the system is predicated upon the design operating sea state and corresponding wave length. The winches contemplated are 50 horsepower, so that the elevating speed will vary with the operating sea state selected.

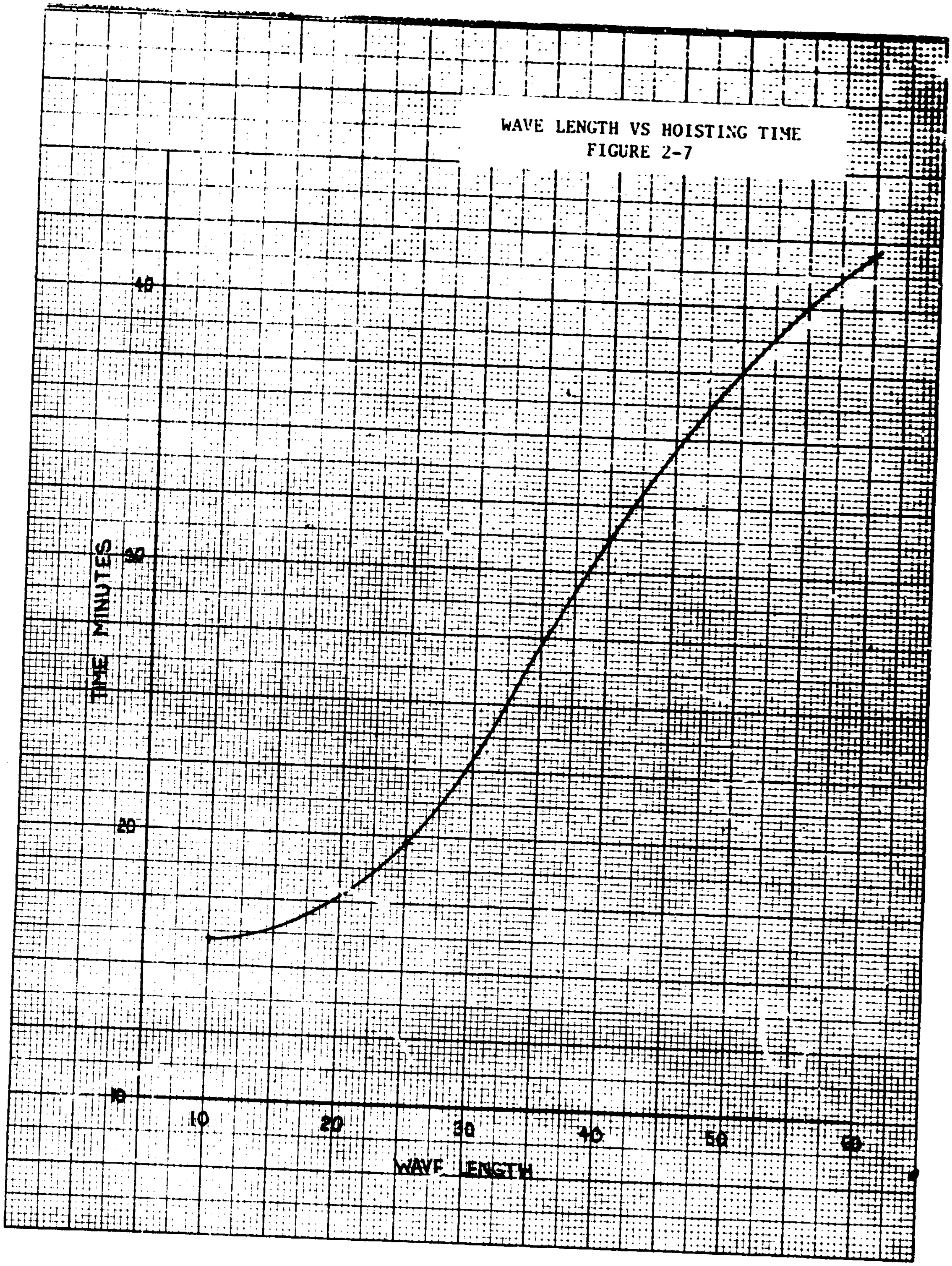
The maximum travel for the system is 84 feet along the kingposts. Depending upon the structure selected for the respective wave length, the hoist time will vary between 16 and 42 minutes, as shown on Figure 2-7.

2.3.9 TRANSVERSE LINKS

The transverse links attach to the collars at the ends of the fore and aft links and serve to maintain the parallel aspects of the two (2) hulls. These members are box-type girders following the type of construction of the fore and aft links. Eness Dwg. No. 71031-6, Sheets 1 and 2, shows the typical end fittings, pivots, and locks. These links can be of various strengths, as discussed in the section on structural design, to account for the effects of wave action. The after link contains the separation machinery.

2.3.10 KINGPOSTS

The kingposts are round tubular vertical members which support the system and serve to be a medium through which the hulls


WAVE LENGTH VS HOISTING TIME
FIGURE 2-7
TIME MINUTES
40
30
20
10
10
20
30
40
50
60
WAVE LENGTH

are separated. Dependent on the expected wave action, various strengths may be built into these posts, as will be discussed in the section on hydrodynamic considerations.

2.3.11 SEPARATION MACHINERY

The separation of the two (2) hulls is accomplished with the crane system in its lowered position; Eness Dwg. No. 71031-7 shows the machinery which causes the two (2) hulls to come together as well as to separate. The operation is accomplished by means of diagonal cables between opposite pairs of pivots of the transverse link. By reducing the length of one cable and increasing the length of the other, the parallelogram formed by the links is opened and closed. The separation drive provides a positive displacement to one diagonal through a roller chain which leads to the cable on the starboard side; the other end of the chain is secured to the cable which leads to the port side. This cable is led to a floating take-up before it is led over the port sheave. Both cables ride in floating sheaves whose positions are retained by springs such that when the tension exceeds the preset limit, the action causes the sheave to move and in turn actuates a limit switch. If the tension in the starboard wire is excessive, the starboard switch reverses the separation drive to payout until the starboard sheave is restored to the proper location. If the tension in the port cable is excessive, the port switch will cause the floating take-up to move to port until there is sufficient slack in the cable. When the starboard cable becomes slack or the tension is reduced below the preset value, slackness actuates the starboard switch so that the separation drive will haul in until the sheave is restored to its proper position. If slackness occurs in the port cable, the port switch will energize the floating drive so that it will move to the starboard side taking up the slack until the floating sheave is restored to its proper position; at this time, floating take-up will be de-energized.

The power required to separate the two (2) hulls will require an initial force to break the hulls from their lands. If both hulls are properly ballasted very little force will be required. However, if not properly ballasted, or if a lag or sag has developed, considerable force may be required to start separation. It is anticipated that hydraulic jacks may be required to start the hulls when separating and draw them together. The forces involved to perform the remaining operation involves the forces to move one hull in a current assumed to be four (4) knots. Separation is accomplished at a speed requiring five (5) minutes for complete separation. Once separated, several conditions could be encountered. These are:

1. Both hulls at anchor heading into the wind and sea and tied alongside the ship being served.

2. Moored to finger pier with ship being served along offshore side in protected waters. If wind tends to move the ship into the barge, cross truss between hulls must be used. Refer to design of truss.

3. Tied alongside of the ship being served and headed into the wind and sea with lighters tied alongside or in-between.

Condition 1. This condition would not impose any serious problem on the diagonal cables unless anchors dragged so that one would be more effective than the other. This condition would be corrected by operating the windlass until parallel orientation with the ship served has been achieved.

Condition 2. This condition imposes a problem relative to the strength of the truss since the truss would have to have strength to withstand the wind forces on the ship being served. Various trusses to withstand such forces are discussed under the truss.

Condition 3. This condition will impose forces on the hulls which will be transmitted to the diagonals. For this condition, a four (4) knot sea is assumed, the drag from one hull and any lighters secured, will provide criteria for strength of the diagonals.

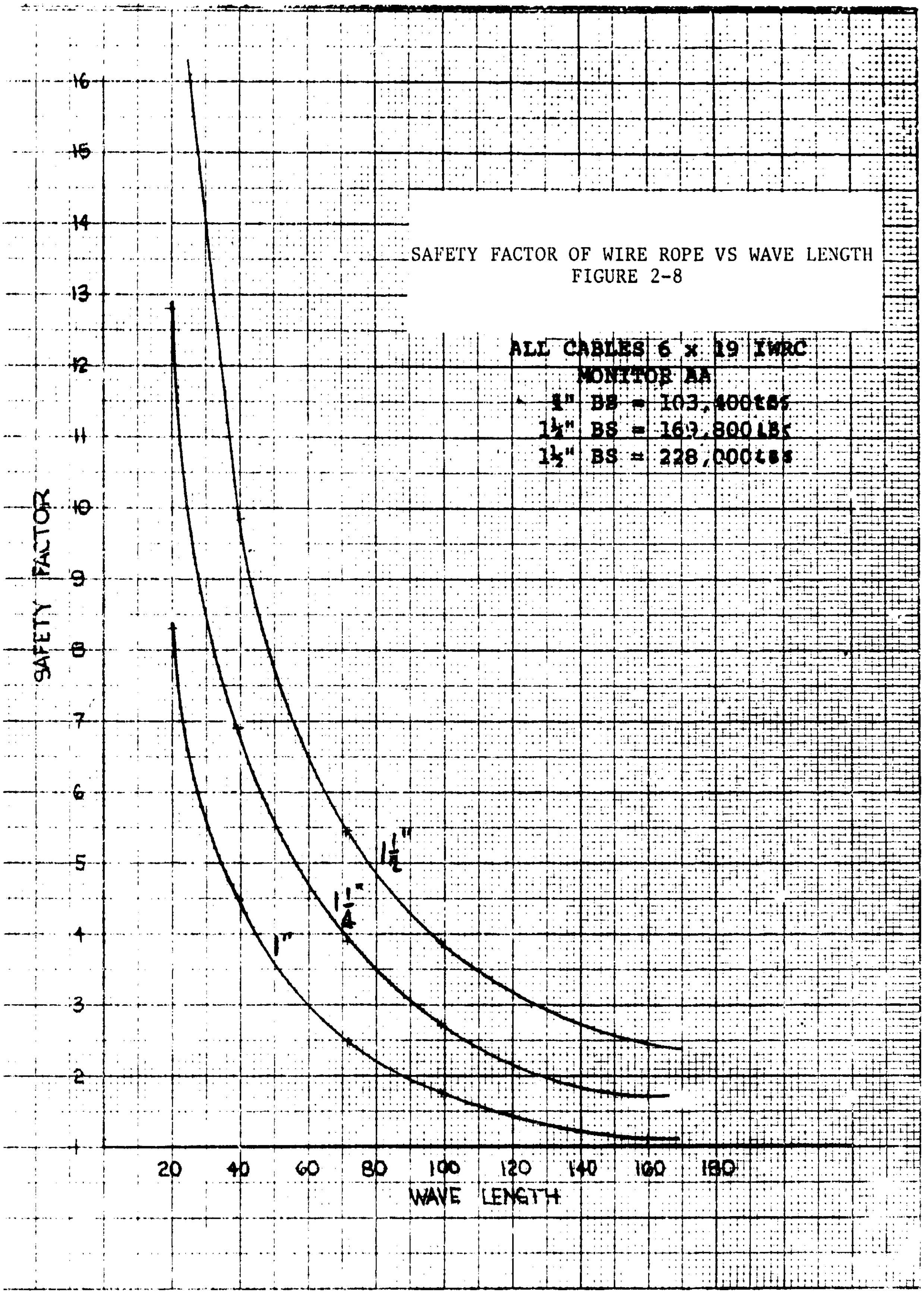
Calculations for separation forces for the various conditions are shown in Appendix B.

In separating the hulls, the cable between the aft starboard kingpost and the forward port kingpost are subject to maximum load and dictate the minimum size required to start the separation. It has been found that the tension in the cable is 7.96 times the broadside force at the center of pressure of the moving hull. The broadside wind force is the predominating force, as shown in Table I of Appendix B. The safety factors are plotted against wave length for three (3) wire sizes in Figure 2-8.

Once in the separated position, the diagonal cables will be capable of fore and aft forces without the aid of the pin lock at the transverse hinge points. Assuming a 1 1/4 inch wire rope which provides ample safety factor for 60 foot waves, the cable is capable of a safe tensile load of 40,000 pounds. Under this load, the maximum safe fore and aft force on the outboard hull that could be encountered is 28,000 pounds. The force from a 4 knot current generating a drag on the hull plus the wind load at 24 MPH will be 1383 pounds + 1880 pounds or a total of 2363 pounds. There is ample reserve to tie at least six (6) LCM-8 and six (6) Larc LX to this hull with an allowance for any pretensioning.

The force required to separate the hulls, assuming minimum wind of about 10 knots, will be approximately 20,000 pounds in the diagonal cable. The total take-up in the stressed diagonal is sixty-three (63) feet. The operation has been assumed to take five (5) minutes.

$$HP = \frac{20,000 \times 63}{33,000 \times .7 \times 5} = 10.9 \text{ HP}$$



SAFETY FACTOR OF WIRE ROPE VS WAVE LENGTH
FIGURE 2-8

For this condition, the assumption is that the wind is broadside with no protection and that the hull is encountering resistance. After initial start, the tension in the diagonal cable reduces rapidly as shown in Figure 2-9. In selecting a 10 HP motor, the tension could be increased to about 30,000 pounds. This overload will not damage any component.

Assuming the maximum tension in the opposing diagonal as unity, the transverse force in closing the hulls will vary as shown on Figure 2-9 as TSI. Based upon generating 20,000 pounds in TSI, the force available at closing will be 13,800 pounds at start and 19,600 pounds at the end of the motion.

The drive selected uses a 10 HP, 1750 RPM motor coupled to a Falk worm reducer Model AVD, size 14, having a 490:1 ratio, mounted as shown on Eness Dwg. No. 71031-7.

The 10 HP motor will be adequate for winds encountered in a mid sea state of 2. Based upon usual Navy permissible overload, this motor could be used where 15 HP may be necessary, which would be at the beginning of sea state 3. It should be kept in mind that the horsepower calculated used maximum forces at the start of motion. The variation in time will be nil based on load variations unless a DC drive is used in lieu of AC.

2.3.12 CORNER PIVOT LOCKS

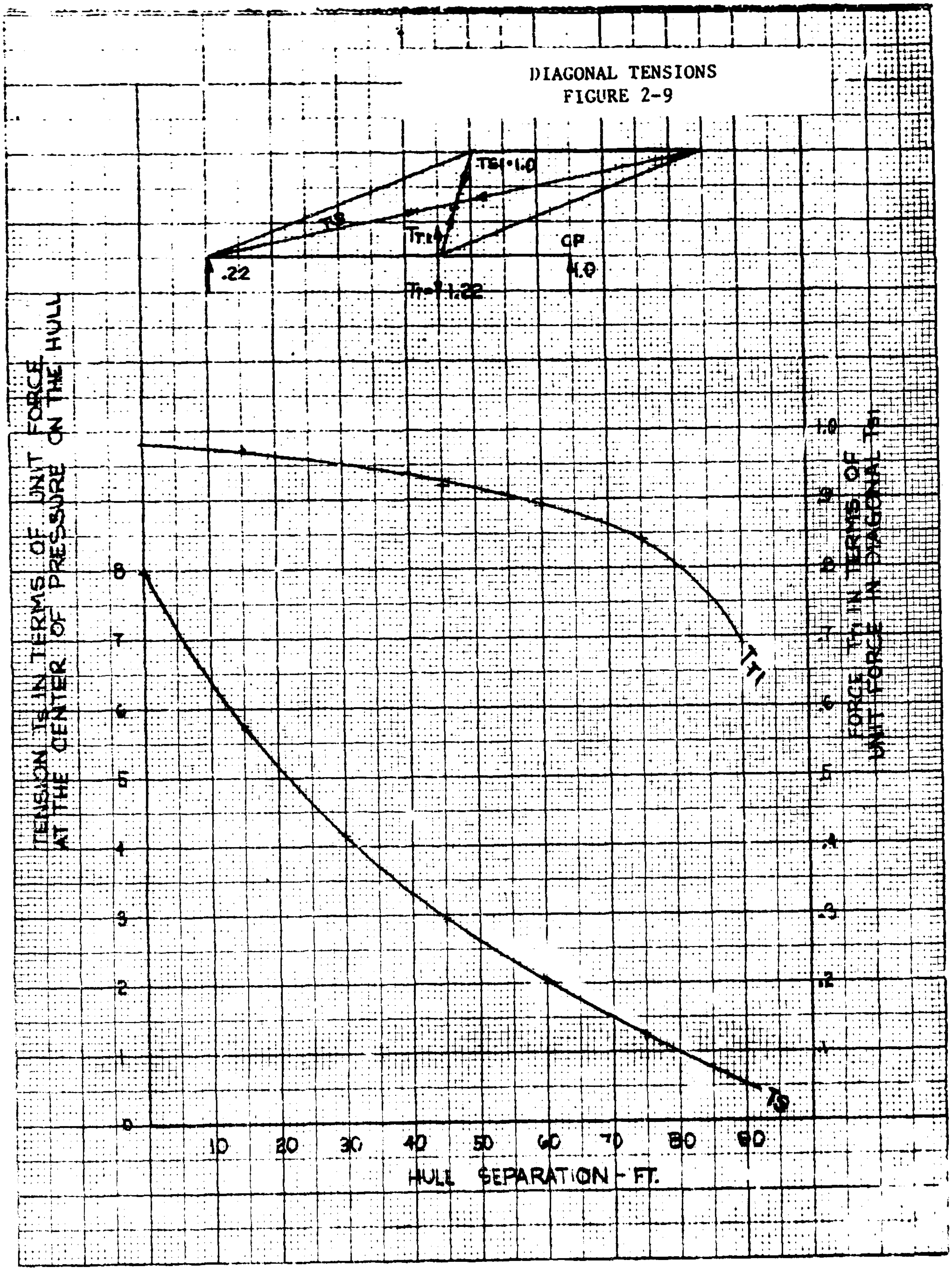
In addition to the diagonal cables, each corner pivot has a lock shown on Eness Dwg. No. 71031-6, Sheets 1 and 2.

Appendix C indicates the loads and strength of these pins. These pins will be 6 feet in diameter. Although adequate to take the entire load, they will share the loads to some degree with the diagonals.

2.3.13 SEPARATION TRUSS

The wave action on the hulls due to the various sea states exerts forces on the hull below the waterline. There is a choice as to adequate cross structure between the kingposts, however, the structure becomes considerable in size and weight, so that for sea states which have a wave length greater than 60 feet, the structure is considered to be impractical. In order to transfer forces from one hull to the other, several designs for separation trusses have been investigated covering wave lengths up to 160 feet. The truss designs will be discussed under "Structural Design".

The trusses are stowed on the underside of the transverse links as shown on Eness Dwg. No. 71031-8. Both ends of the trusses form a plate which slides in a "T" slot in the barge hull in a transverse line with the pivot of the transverse links. When brought in position under the link, a catch automatically locks the truss in stowed position. Before the crane is raised, the locks are released


DIAGONAL TENSIONS
FIGURE 2-9
TS1=1.0
TS
TT1
.22
CP
1.0
TT=1.22
TENSION TS IN TERMS OF UNIT FORCE AT THE CENTER OF PRESSURE ON THE HULL
FORCE TT1 IN TERMS OF UNIT FORCE IN DIAGONAL TS1
HULL SEPARATION - FT.
TT1
TS

and the truss is lowered by winch power. The disconnect is at the deck level where a stop is made. At this point, the truss rests at the bottom of the shell. The hoist cable is released at the disconnect and the crane is raised to operational position. For handling the truss, a simple winch with a divided drum is required. The total travel is approximately 45 feet, where more than half of the hoist takes place under water. The winch power is estimated at 10 HP which will either raise or lower the truss in approximately 2 minutes depending upon the weight of the truss selected.

The recess for the ends of the trusses will have generous clearances, and since the trusses are moved with links in the lowered position, the raising and lowering of the trusses should not present a difficult operation.

2.3.14 SPREADER

The spreader considered in the design is based upon a commercial spreader for 20-foot containers. In order to handle military vehicles, the spreader must be fitted with padeyes to accept slings used in hoisting the vehicles. The spreader must also be capable of supporting 70,000 pounds. In event that at a later date it is intended to handle the 40-foot containers, a supplementary spreader may be added and latched directly to the 20-foot spreader. Should greater flexibility be required at the start of operations, a commercial spreader of the adjusting type can be obtained. This spreader is more involved and if 40-foot containers are not contemplated, it should not be incorporated in the final design.

2.4 HYDRODYNAMIC CONSIDERATIONS

2.4.1 CRITERIA

Usual criteria[1] in determining hydrodynamic forces on catamaran hulls is to assume sinusoidal waves both athwartship and quartering on the two (2) hulls. The two conditions are analyzed under the premise that maximum loads are encountered when the center planes of the two (2) hulls are located at the point of inflexion of the wave and when the crest of the waves in a quartering sea passes through the forward quarter length of the starboard centerplane and through the after quarter length of the port centerplane. Also in computing heave forces, acceleration of ± .4"g" are used. The foregoing criteria is used on catamarans which are powered and maintain their separation using a permanently positioned structure. The analysis in this report uses the same approach, however, several modifications are made.

In beam seas the placement of the center planes to the points of inflection of the wave may be used in the catamaran application covered by this report, however, it is not considered to be

1. Dinsenbacher, A. L., Marine Technology, Vol. 7, No. 4, Oct. 1970

the only condition with respect to transverse wave action. As the barge will normally be secured to the ship served, transverse forces are not considered critical because the barge will be heading into the sea. If the barge is used alongside a wharf with the crane raised but not in service, and not sheltered from waves and wind, there is a possibility of encountering forces from waves whose points of inflection do not fall simultaneously on the center planes. The following analysis assumes one hull restrained by a wharf while the other hull is exposed to wave lengths up to 160 feet. The crest for these waves are placed so that the maximum differential of head between the two (2) sides of the hull are obtained.

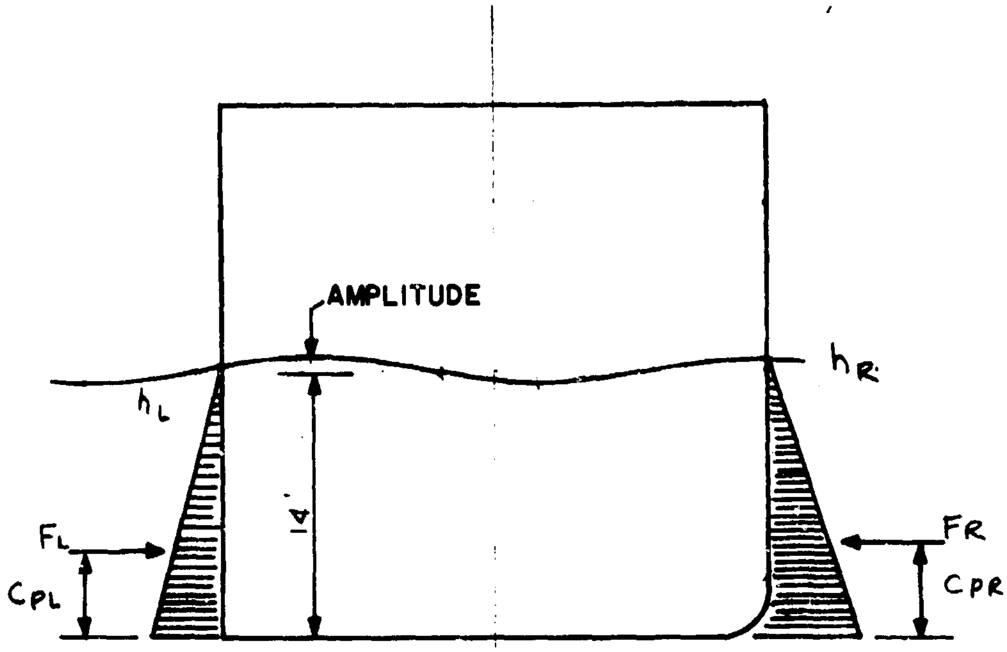
In the condition of quartering seas, the wave crests at the fore and aft quarter points of the center plane would not provide the most severe loading because the hulls, when separated, are staggered. Such seas would tend to act uniformly on both hulls, therefore, the analysis in this report assumes the separated barge to be in a direct heading sea with wave crests located to provide hull displacement out of phase with each other.

2.4.2 BEAM WAVES

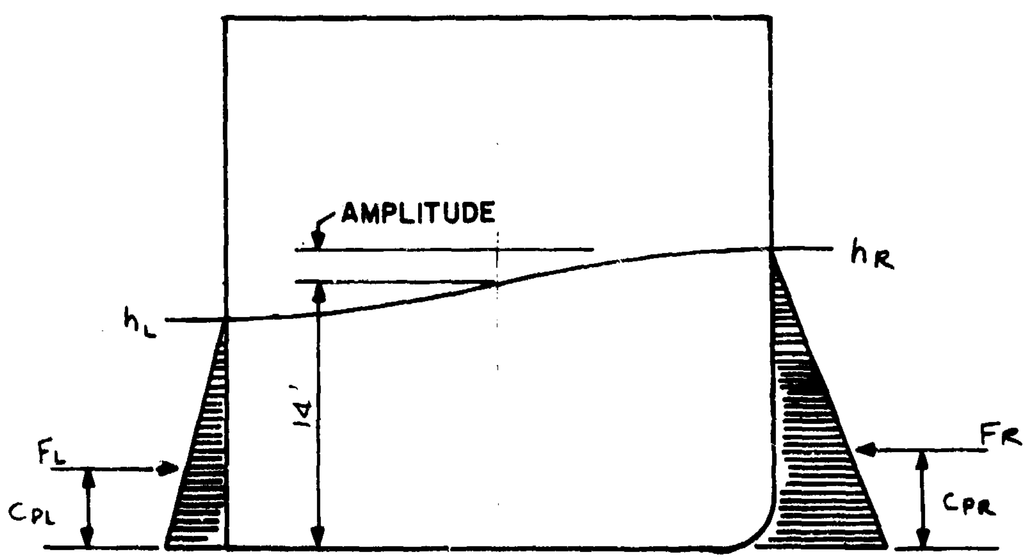
The sea characteristics used in this analysis considered wave lengths up to 180 feet, which fall into sea state 6 as determined from Figure 2-1. Because it is considered convenient to use wave lengths, the corresponding significant wave height for each wave length has been used from Figure 2-2.

As the center to center distance of the barge hulls is 120 feet, two (2) waves of 60 feet will place the center plane of each hull at the point of inflection. The amplitude of the significant wave height is taken as 3.6 feet. For wave lengths less than 60 feet shown in Figure 2-10, the waves are considered to be acting upon the two (2) hulls simultaneously and not affected by the wharf to which the barge is tied. For wave lengths greater than 60 feet the wave has been placed on the one offshore hull so that the inflection point is at the center plane and the wave acts independently of the other hull. In the absence of any test data, this is considered to be a conservative approach. It is believed that the hull adjacent to the wharf will have an interaction with the wave and tend to break it between the hulls so that the trough will not fully develop, and, therefore, the differential head will be somewhat less than used in the calculations of this report.

Where wave lengths are less than 60 feet, the crest of the wave is placed on the extreme outboard line of the shell and the various heads are determined by locating the elevation of the trough on the inside line of the shell. The head on the extreme outboard side (H_r) is determined by adding the wave amplitude to the mean draft which is 14 feet. The head at the inside line of the shell is obtained by adding 14 feet to the amplitude multiplied by the sine of the angle at this point on the sine wave. The center of pressure (C_p) is 1/3



WAVE POSITION FOR WAVE LENGTHS LESS THAN 60 FEET
FIGURE 2-10



WAVE POSITION FOR WAVE LENGTHS GREATER THAN 60 FEET
FIGURE 2-11

the head. Where the vessel is not tied to a wharf, dynamic acceleration is not considered. The barge, when alongside the wharf, will have to absorb dynamic wave forces so that the density of seawater is multiplied by 1.4 to obtain the force per lineal foot along the length of the hull; a constant (89.3) is used and the average pressure is assumed to be at the center of pressure (C_p).

The configuration is such that the longitudinal center of pressure could exert its maximum force through one transverse link, which would be alternated to the other transverse link if the mooring was reversed from one hull to the other. Therefore, the maximum hydrodynamic forces are assumed to be transferred between one set of kingposts through the transverse link.

For wave lengths of 60 feet and longer, the point of inflection at the center plane offers symmetry such as shown in Figure 2-11. The differential head is predicated upon the slope of the sine wave at the point of inflection. For wave lengths up to 150 feet the wave height is such that the slope of the sine wave at the point of inflection increases. Beyond 150 feet, the wave height is such that the slope of the wave at the point of inflection decreases. Therefore, based upon the established wave height and wave length relationship, the catamaran hulls would not encounter forces greater than those generated by the wave 150 feet in length. Figure 2-12 indicates the average force (F_R) for each foot of barge length and center of pressure (C_{PR}) on the extreme outboard shell. For the inside shell these values are indicated as F_L and C_{PL}.

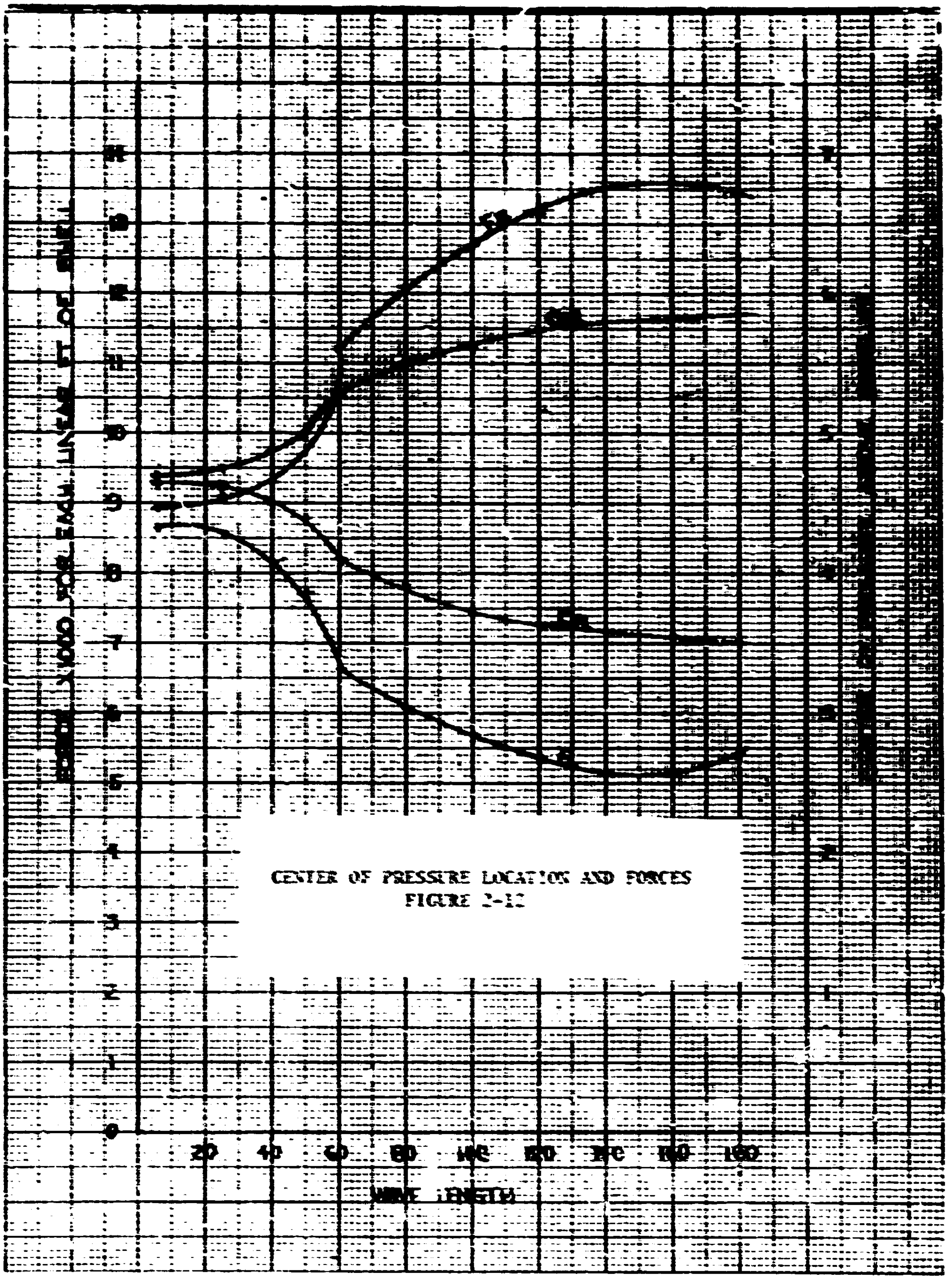
The center of bouyancy is assumed to act through the center plane at 15 feet from the side. The kingposts are located 5 feet outboard of the center of bouyancy, introducing an additional moment to the kingpost and cross structure. Considering the collar for the link to be reasonably rigid to introduce fixity, the moments at the collar are determined by assuming a 900K vertical load on each kingpost at a moment arm of 5 feet. This value would cover crane structures designed for up to 40 foot wave lengths. Additionally, an acceleration of .4"g" is assumed. The transverse links are 125 feet from the neutral axis to the base line.

The moment at the center of span of the transverse link point "P" can be obtained from the following equation:

$$\Sigma M_P = \pm \left[F_R \ (125-C_{PR}) - F_L \ (125-C_{PL}) \right] \quad 2.0$$

The moment at the collar point "Q" can be obtained by the following:

$$\Sigma M_Q = \Sigma M_P \pm 900 \times 5.0$$



CENTER OF PRESSURE LOCATION AND FORCES
FIGURE 2-12

The moment on the kingpost at the deck, point "N" can be obtained by the following:

$$2 M_N = \left[F_R (28-C_{PR}) - F_L (28-C_{PL})\right] 310 \pm 900 \times 5.0$$

Using the values from Figure 2-12, moments can be calculated selecting those combinations for maximum moments. These values have been plotted against wave length on Figure 2-13. As the magnitude of the moments are high, the effect of wind and current being very small is negligible.

2.4.3 HEADING SEAS

Conditions in heading seas have also been generated using the sinusoidal waves. These waves are placed so that one hull is in equalibrium and on an even keel. The effect on the other hull, by virtue of its position, would be acted upon by the same wave fronts, however, the center of bouyancy will change, subjecting the transverse links to torsion as a result of the tendency for the hull to encounter pitching. Figure 2-14 shows the port hull on an even keel while the starboard keel is subjected to 40-foot waves. It should be noted that the crest of the waves introduce an increase in bouyancy while the trough presents a loss. The moments (M) of each crest and trough will provide the unbalanced moment causing pitching. Using the area (A) of the sine wave above and below the mean draft and determining the lever from the stern, the new center of bouyancy (C_B) is established.

$$C_B = \frac{\Sigma M}{a}$$

The area of the center plane (A_{CP}) is the draft (14 feet) multiplied by the length (310 feet).

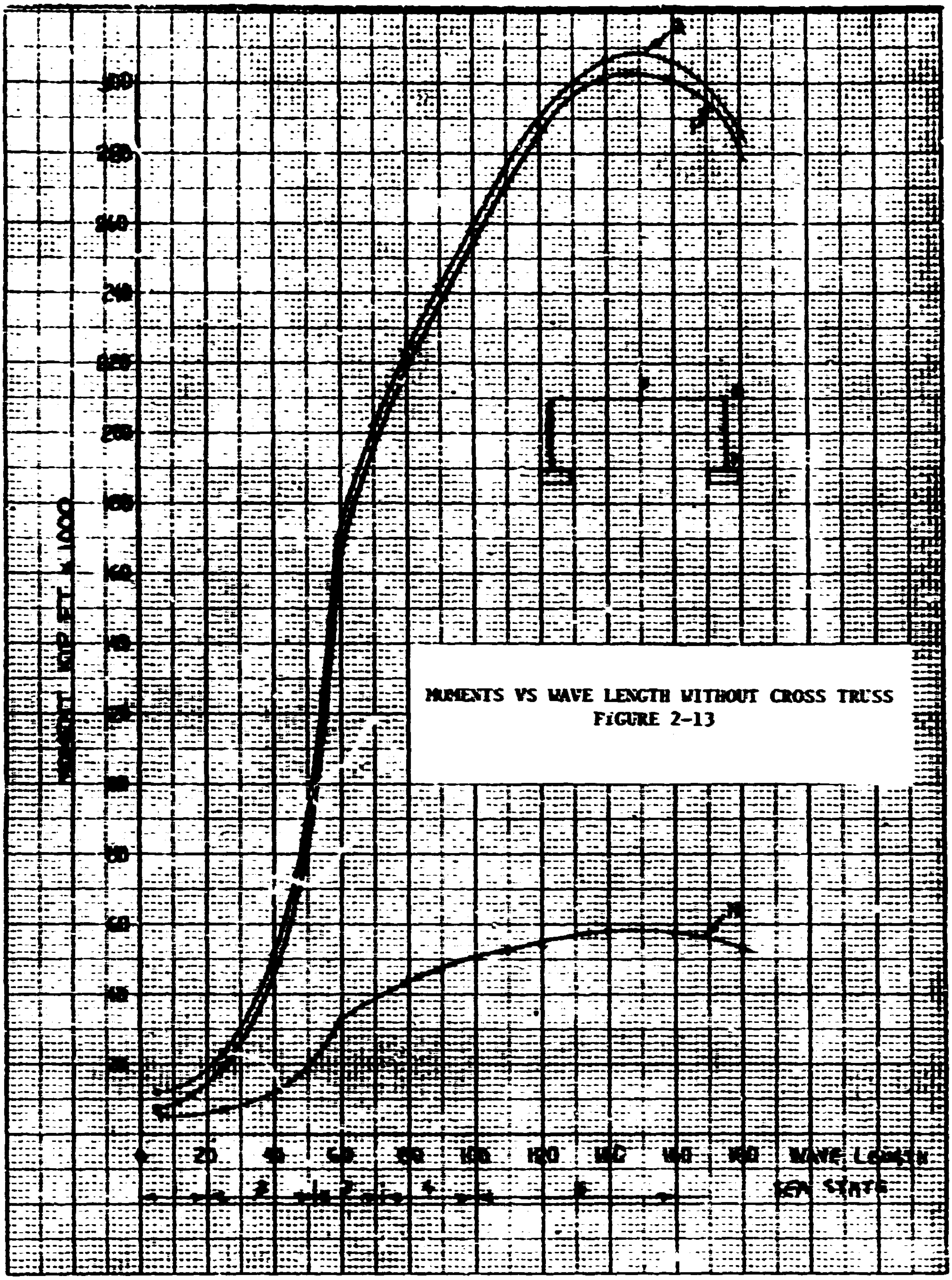
$$A_{CP} + 310 \text{ feet} \times 14 \text{ feet} = 4350 \text{ sq. ft.}$$

$$\frac{A_{CP}}{2} = 2175 \text{ sq. ft.}$$

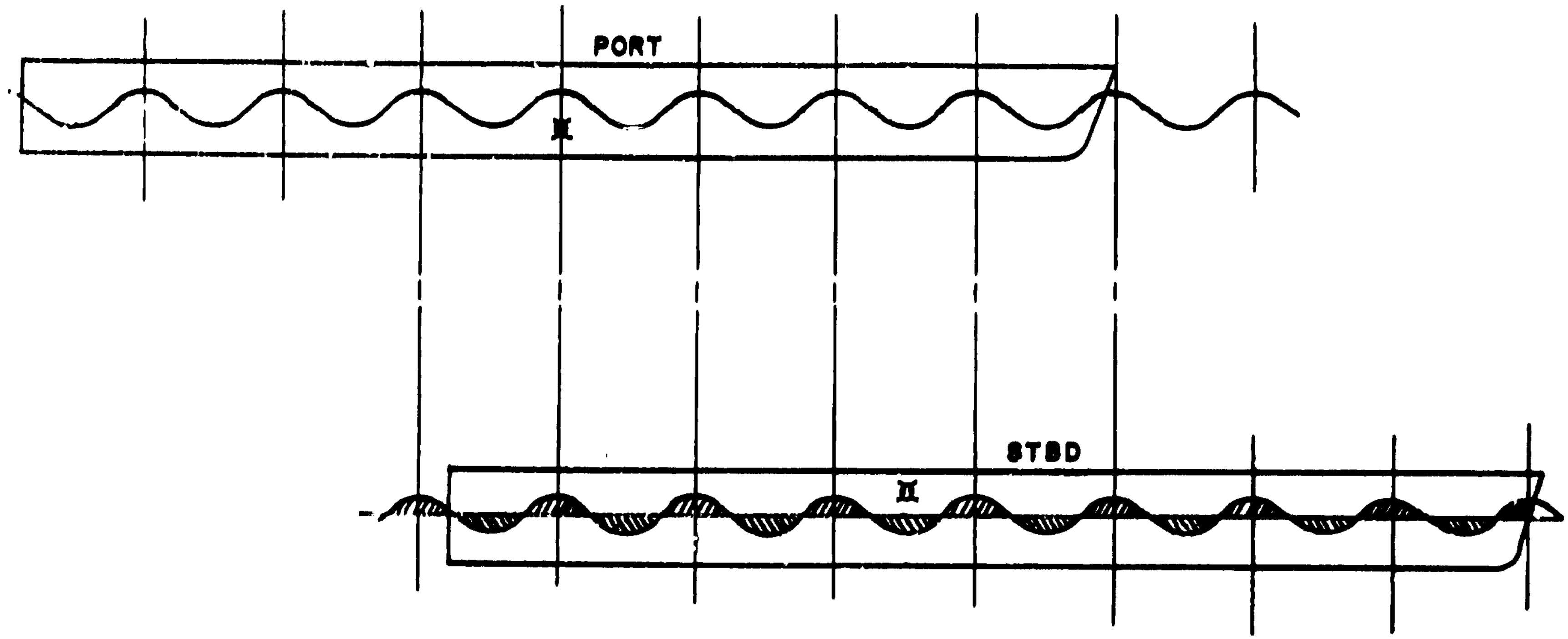
The draft aft (D_A) and forward (D_F) are determined by distributing the center plane area $\frac{A_{CP}}{2}$ equally fore and aft of the center of bouyancy. These drafts are obtained by the following:

$$\frac{D_A + 14}{2} \times C_B = 2175$$

$$\frac{D_F + 14}{2} \times C_B = 2175$$



MOMENTS VS WAVE LENGTH WITHOUT CROSS TRUSS
FIGURE 2-13



LOCATION OF WAVE CRESTS ON SEPARATED HULLS IN
A HEADING SEA
FIGURE 2-14

The draft forward and aft will provide the instantaneous trim "Θ".

$$\tan \Theta = \frac{DA - DF}{320 \text{ feet}}$$

The variation in trim has been plotted and indicated in Figure 2-16. The plot indicates that for wave lengths up to 150 feet, the trim is .3 degrees or less.

A check of the stress due to torsional deflection indicates very low stress. For this stress, a 40-foot wave structure was assumed having a depth of 96 feet and a width of 36 feet. The web is 5/8 inch and the minimum thickness of the flange was taken at 2 7/8 inches. Assuming the .3 degree trim resulted in a torsion deflection of .15 degrees on the transverse links, the maximum stress is less than 700 PSI assuming case 11 of Roarks Fourth Edition, page 196.

The barge, when subjected to broadside waves, causes listing when the wave length is such that the distance between center planes is not a module of the wave lengths for values less than 120 feet. When wave lengths are 20, 30, 40, 60, and 120 feet, the hulls are subjected to heaving with no resulting list. Other wave lengths cause a list which is determined by placing one hull on the crest of the wave and determining the location of the center plane of the other hull relative to the wave trough.

The list B may be determined from the following:

$$\tan B = W_H - \cos \frac{(120 - WL \times 360)}{WL} W_H$$

Where W_H = amplitude of the wave

W_L = length of the wave

For wave lengths up to 160 feet, the theoretical lists are plotted on Figure 2-15. The barge, when used alongside a ship, will not normally be subjected to these lists while operating. Also, it is not likely that these lists will appear during normal operations alongside a pier due to the interaction of craft surrounding the barge.

From the standpoint of hydrodynamics, the barge alongside a vessel with their bows into heading seas will be the prevailing condition. A secondary condition will be one where the barge is secured to a wharf or causeway. Under this condition, there are alternatives to the placement of the barge.

One alternative would be to have the separated barge straddle the causeway or finger pier so that one hull separates the pier from the ship being served. The barge will be protected and therefore, wave heights would be insignificant for most cases.

The second alternative would be to place the separated barge alongside a wharf between the ship and the wharf. In this particular case, the beam wind on the ship will tend to exert a closing force on the barge unless the separation trusses are of sufficient strength. Further discussion will be made under structural design of trusses.

The third alternative is with the barge tied to the wharf along a river front and the served ship outside in a current of 4 knots. A large container ship of 880 feet between perpendiculars 105.5 feet wide and 34 feet draft will impose a fore and aft force of approximately 250,000 pounds. Under such a condition, stern and head lines to the pier should be long so as to not subject the barge to large closing forces. The head lines of about 30 degrees will impose approximately 125,000 pounds towards closing and would require a truss to withstand this force.

2.4.4 OPERATIONAL LIST

Each hull of the barge, when separated, requires approximately 20 tons to displace the hull 1 inch. Relative to Design No. 1, with both booms extended equal amounts, the handling of a 44K container load at 171 feet from the center, will impose a load of 162K (72.2 T) on the near hull and -78K (-34.9 T) on the far hull. The increased immersion of the near hull will be approximately 3.6 inches and the far hull will raise approximately 1.5 inches.

$$\text{The list, } \theta = \text{ton } \frac{5.1}{12 \times 120} = 13.5 \text{ minutes } (.22 \text{ degrees})$$

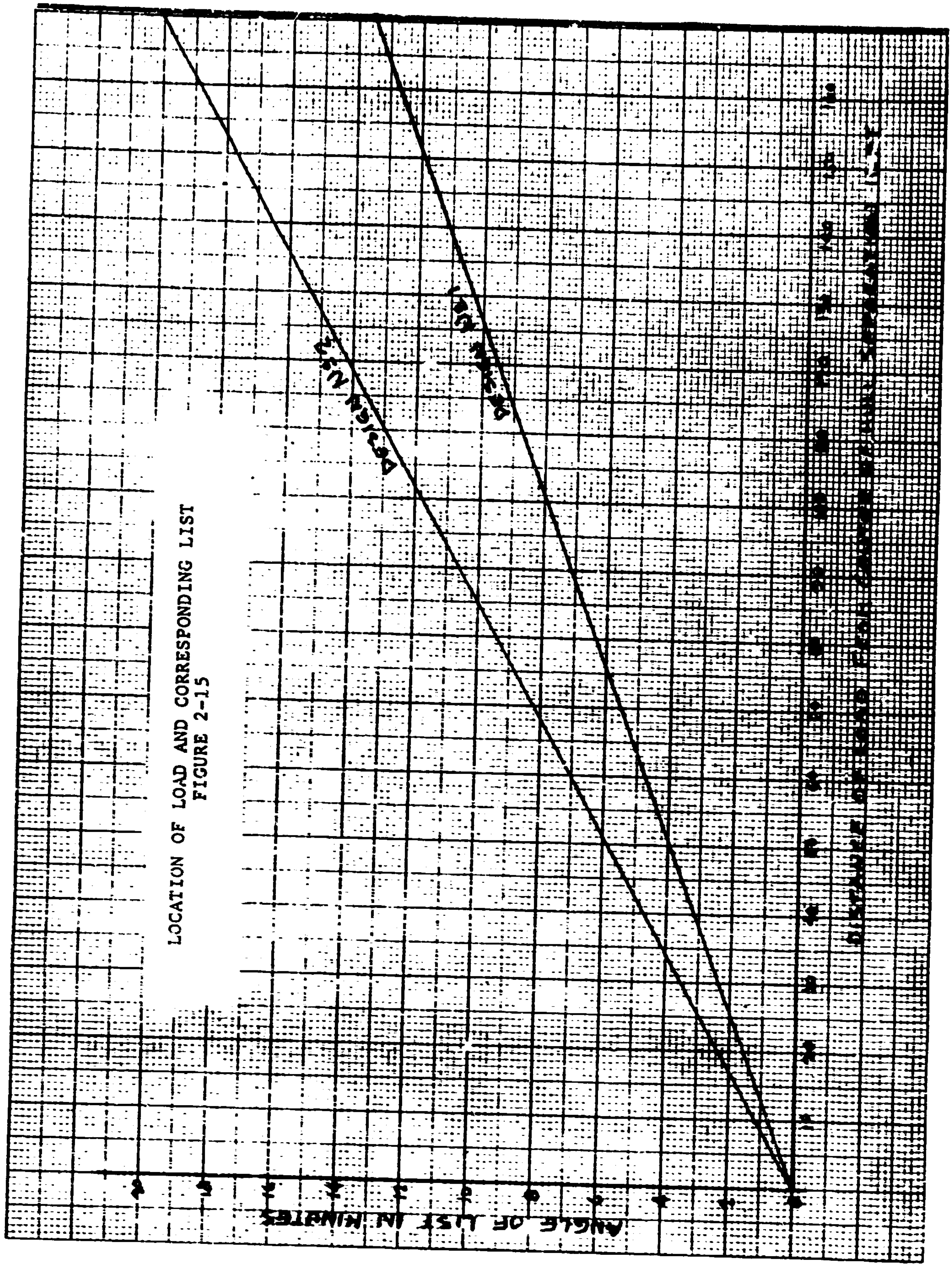
When handling 70K load at 101 feet from the center, the load on the near hull is 147.5K (66 T) and -37.5K (-1.67 T) on the far hull. The immersion of the near hull will increase approximately 3.3 inches and the far hull will raise approximately .85 inches.

$$\text{The list } \theta = \text{ton } \frac{4.15}{12 \times 120} = 10' \ (.17 \text{ degrees})$$

Design No. 2 has an increased operating list because of the moving extensible boom. The container load on the near hull is 313K (140 T) and on the far hull -115.7K (-51.6 T). The list is 23 feet (.38 degrees). When handling a 70K load at 101 feet, the load on the near hull is 298.5K (133.5 T) and on the far hull -65.2K (-29.1 T). The resultant list is just under 20 feet (.33 degrees). Figure 2-15 indicates list with respect to load location.

2.5 MECHANICAL DESIGN

The basis for mechanical design has taken into account the various aspects of conventional crane design. In a preliminary analysis, many details cannot be explored in depth; however, the design generally conforms to the publication, "A Guide for Shipboard Crane Specifications", to be published by the Society of Naval Architects and Marine



LOCATION OF LOAD AND CORRESPONDING LIST
FIGURE 2-15

Engineers. This publication provides the criteria for cranes used in the commercial marine field.

2.6 STRUCTURAL DESIGN

2.6.1 STRUCTURE

The structure covered by this report used ASTA-A514 or A517 steel having a minimum yield point of 90 KSI and a minimum ultimate tensile strength of 105 KSI. This material has been used in major strength components. The analysis has combined many known aspects of the conditions to which the structure will be exposed. Until actual experience is encountered, the conservatism or lack of conservatism will not be known. In general the allowable stress in bending is taken at 27,000 PSI with allowance made for buckling, where required. The assumptions used in the study tend to be conservative and therefore the design is expected to be on the conservative side.

2.6.2 EXTENSIBLE BOOM

The extensible boom has been designed for a 44,000 pound load at an outreach of 96 feet which permits coverage over container cells in vessels 105 feet wide. Additionally, the boom can support a 70,000 pound load at 70 feet. In Design No. 1 and No. 2 the boom is a boxed section with extending flanges and is 4 feet deep and 20 inches wide. The scantlings have been varied to provide the lightest structure in keeping with weight considerations. Eness Dwg. No. 71031-3 indicates the general configuration. Appendix D indicates the bending moment and shear diagrams, and provides means for the selection of a section having suitable properties. It should be noted that the 70,000 pounds at 70 feet provides the maximum moment on the structure and the moment diagram illustrates the moment for one member of the boom.

2.6.3 BRIDGE

Main bridge structure which travels fore and aft on the fore and aft links provides the support for the wheels on which the extensible booms ride. It is a boxed section with extended flanges and is 5 feet deep and 15 inches wide. The structure is composed of two (2) boxed members transversely oriented with cross ties for support of wheels and machinery. Appendix E indicates the bending moment and shear diagram for one member resulting from a 70,000 pound load at 70-foot outreach. As operations can be achieved on either side, it is considered that the most economical structure would be members of constant properties for the length in lieu of selecting some lighter structure where there is a slight reduction in bending moment. Appendix E indicates possible scantlings, for the purpose of this report a flange 1 1/8 inches-thick and 3/8 inch-web will be used.

2.6.4 FORE AND AFT LINKS

The fore and aft links supporting the bridge are subject to loads depending upon the location of the bridge. It is a boxed section with extending flanges. It measures 6 feet in depth and is 30 inches wide. The reactions, bending moment, and shear diagrams are indicated in Appendix F, with means to determine the required section and weight. As the crane moves fore and aft, the scantlings for these links are considered to be constant in lieu of reducing scantlings where there is a reduction in bending moment.

2.6.5 TRANSVERSE LINKS

The transverse links which maintain the barge orientation are affected by the sea states, and therefore vary in design depending upon the wave lengths for each sea state. Appendix G indicates the bending moments for various wave lengths including allowance for structural deadweight. The composite moment, required section modulus, and scantlings are also indicated. The composite moment is a combination of dead load and live load movements which provide the maximum. The estimated dead load has been assumed and found to vary approximately 1 to 3 percent. The strength of the short wave length members are in slight excess while the larger wave lengths are slightly deficient with respect to dead load. Further refinement will be required during the final design stage. The section property and weight may be obtained from Appendixes F and G.

2.6.6 KINGPOSTS

The kingposts which support the load are subject to varying loads dependent upon the sea state. Several post designs and diameters were considered. The designs considered were those of posts having a cylindrical shell with uniform thickness to withstand the loads, a cylindrical shell with "T" members inside, and a cylindrical shell with inside plate doublers on the axis of maximum bending. The outside post diameters considered are 72, 78, 84, 96, and 108 inches. Appendix H also has means for determining the properties of kingposts having a uniform thickness; having a structural "T" added inside; and properties for kingposts having a flat plate doubler where the thickness of the doubler is the same as the shell. In all cases, a 4 x 8 inch steel bar was considered to contribute to the strength although it is used primarily as the slide for the collar. The inside doubler provides for a greater efficient structure with weight savings between 1 to 7 percent. It is considered that the inside doubler provides maximum strength with respect to weight and as such has been applied to the design.

Analyzing all situations with respect to column action, the maximum L/R was found to be 10 based upon the minimum section comprising the kingposts. The selection of scantlings are such that there is ample margin in the actual stress to cover the low values of direct compression. The ratio of 10 for L/R will not materially reduce the allowable compressive stress so that these stresses can be combined directly.

For a comparison of crane loads on kingposts for various wave lengths see Figure 2-16.

For variations in kingpost weight versus wave length see Figure 2-17.

2.6.7 SEPARATION TRUSS

For operation in certain sea states, the hulls are required to have a structure to maintain the separation; two trusses are provided for this purpose. The sea state and resultant action imposes varying forces on such a structure which will absorb approximately 95 percent of the hydrodynamic forces. The separation truss is shown on Eness Dwg. No. 71031-8. It is made up of tubular members. The forward truss is subject to the higher loading; however, both trusses are made identical in scantlings as a safety measure. Figure 2-18 shows the variation of truss force with wave length. Appendix J tabulates the various pipe sizes required for chord members for different wave lengths.

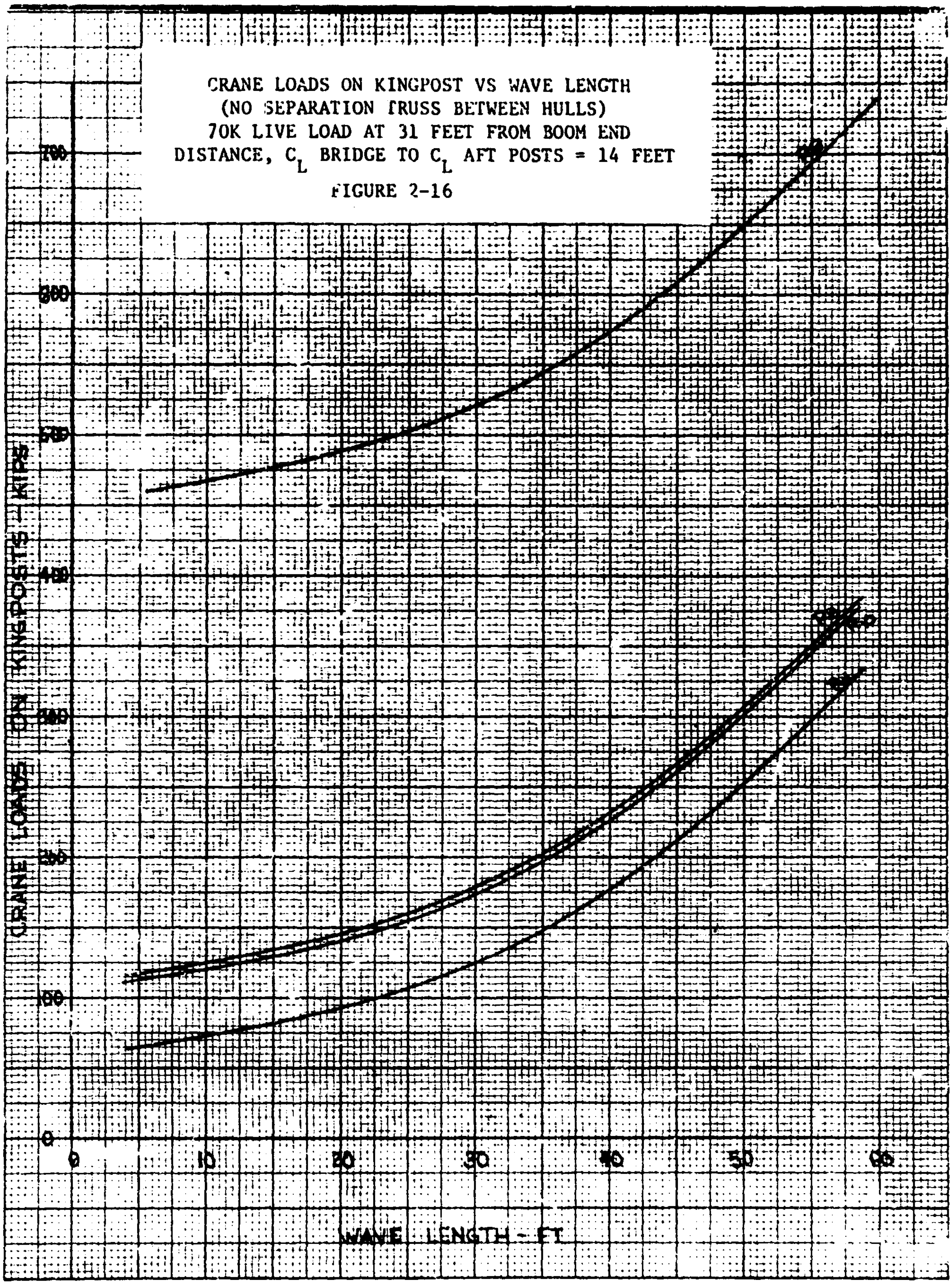
A plot of truss weight versus wave length is presented in Figure 2-19.

2.7 WEIGHTS AND CENTERS OF GRAVITY

The weight of crane Design No. 1 had been developed in depth because it is greater than that of Design No. 2. For the weight of Design No. 2, corrections will be made to the weight data of Design No. 1.

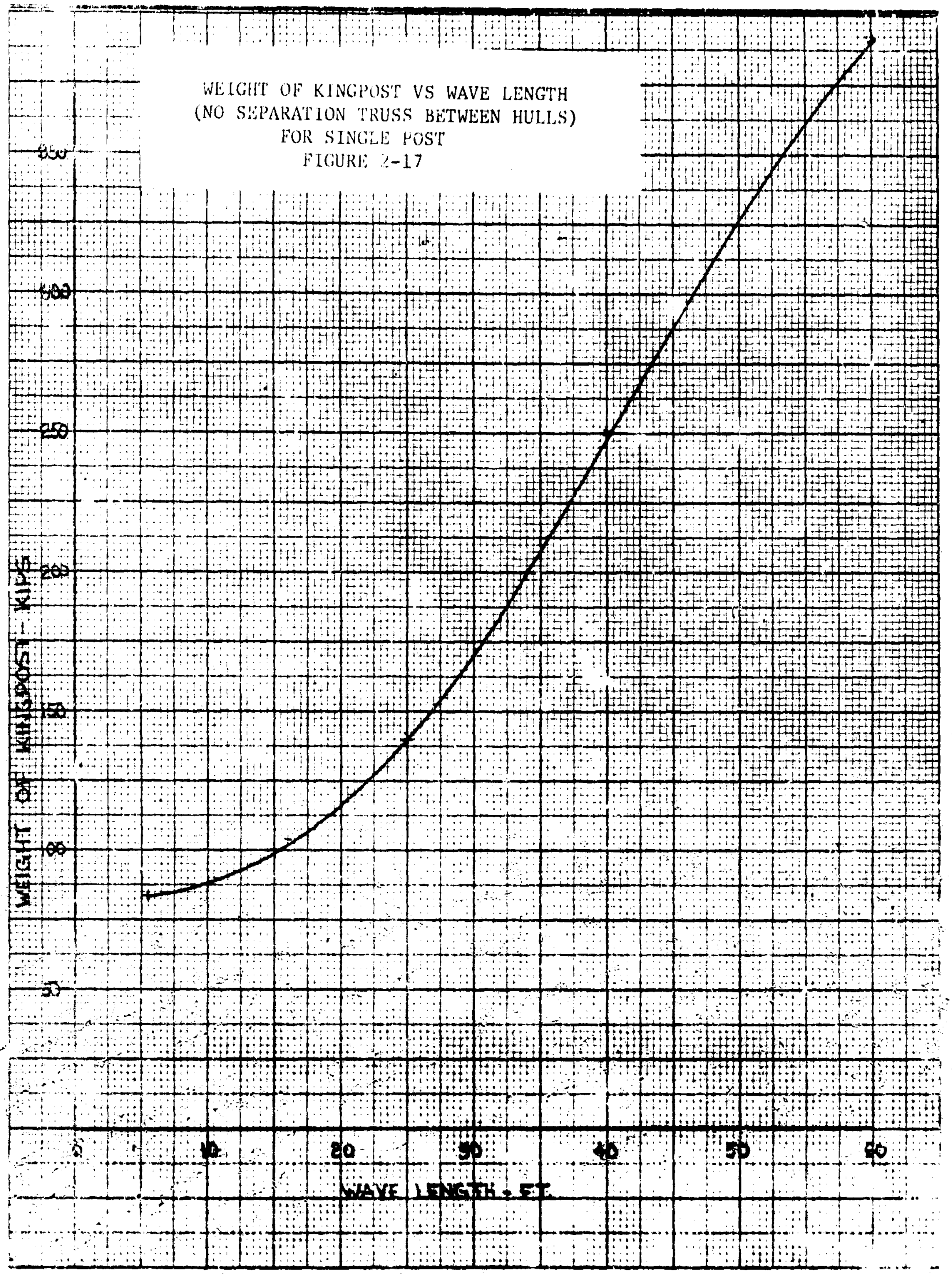
Inasmuch as the weight of the crane without a truss becomes excessive at the 60-foot wave length, further strengthening of the design for the crane for higher wave lengths is considered to be impractical. Operation of the crane in the higher wave lengths will be achieved by the selection of appropriate separation trusses. Tabulations in Tables II and III on the following pages and in Appendix K indicate the breakdown of weights and centers of gravity by component and wave length for the systems without the separation truss and with the separation truss. The purpose of these tables is to permit an evaluation of the various possibilities for discussions to follow.

The weight information has been combined with the component location to establish the center of gravity for wave lengths to 60 feet. These centers of gravities are independent of the use of the truss and are taken from the top plane of the fore and aft links; the vertical center of gravity, with respect to the barge, can be obtained by subtracting this center of gravity from the height of the plane above the selected datum. The center of gravity coordinates versus wave length are plotted on Figure 2-20.

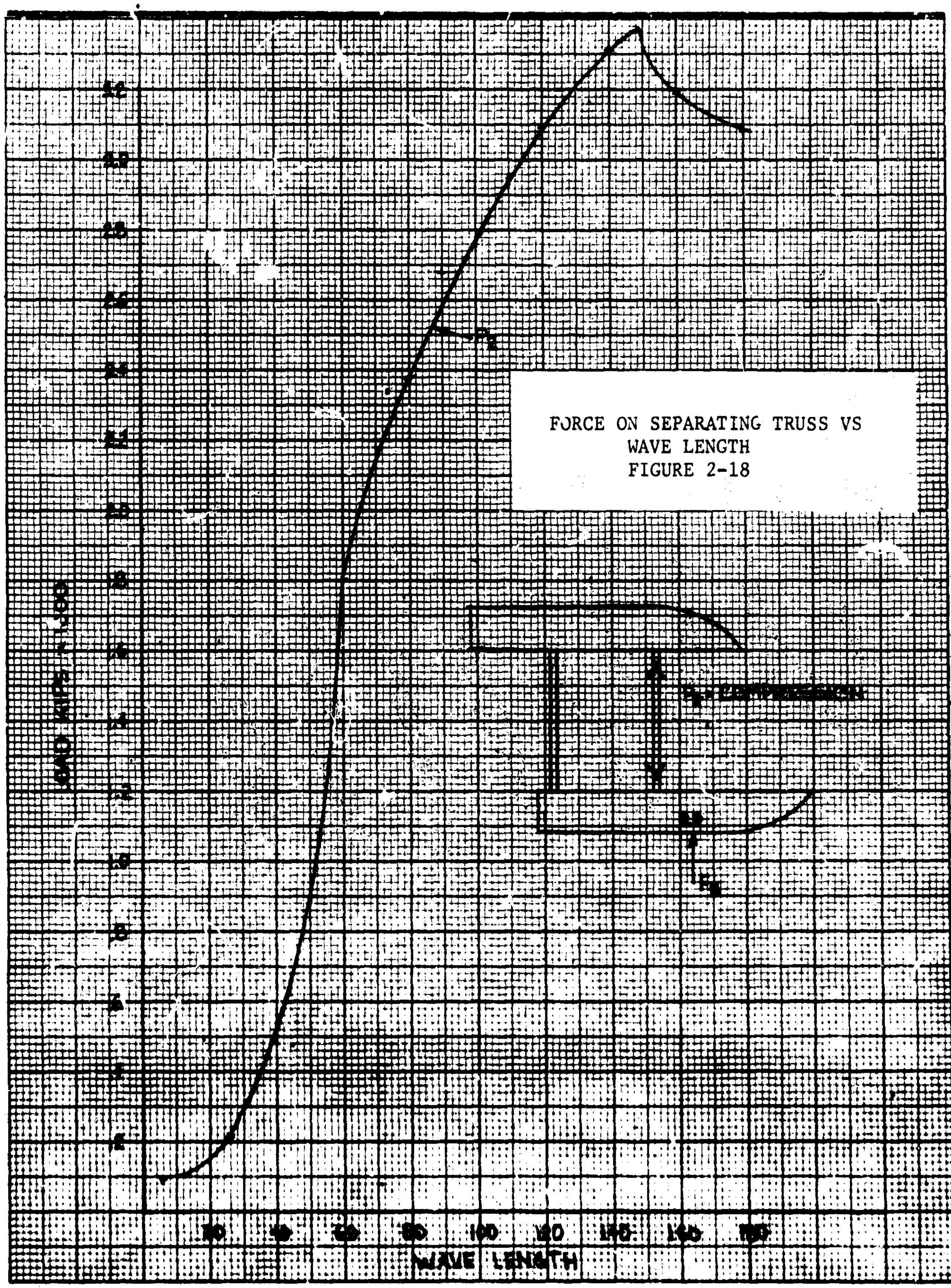


CRANE LOADS ON KINGPOST VS WAVE LENGTH
(NO SEPARATION TRUSS BETWEEN HULLS)
70K LIVE LOAD AT 31 FEET FROM BOOM END
DISTANCE, C_L BRIDGE TO C_L AFT POSTS = 14 FEET

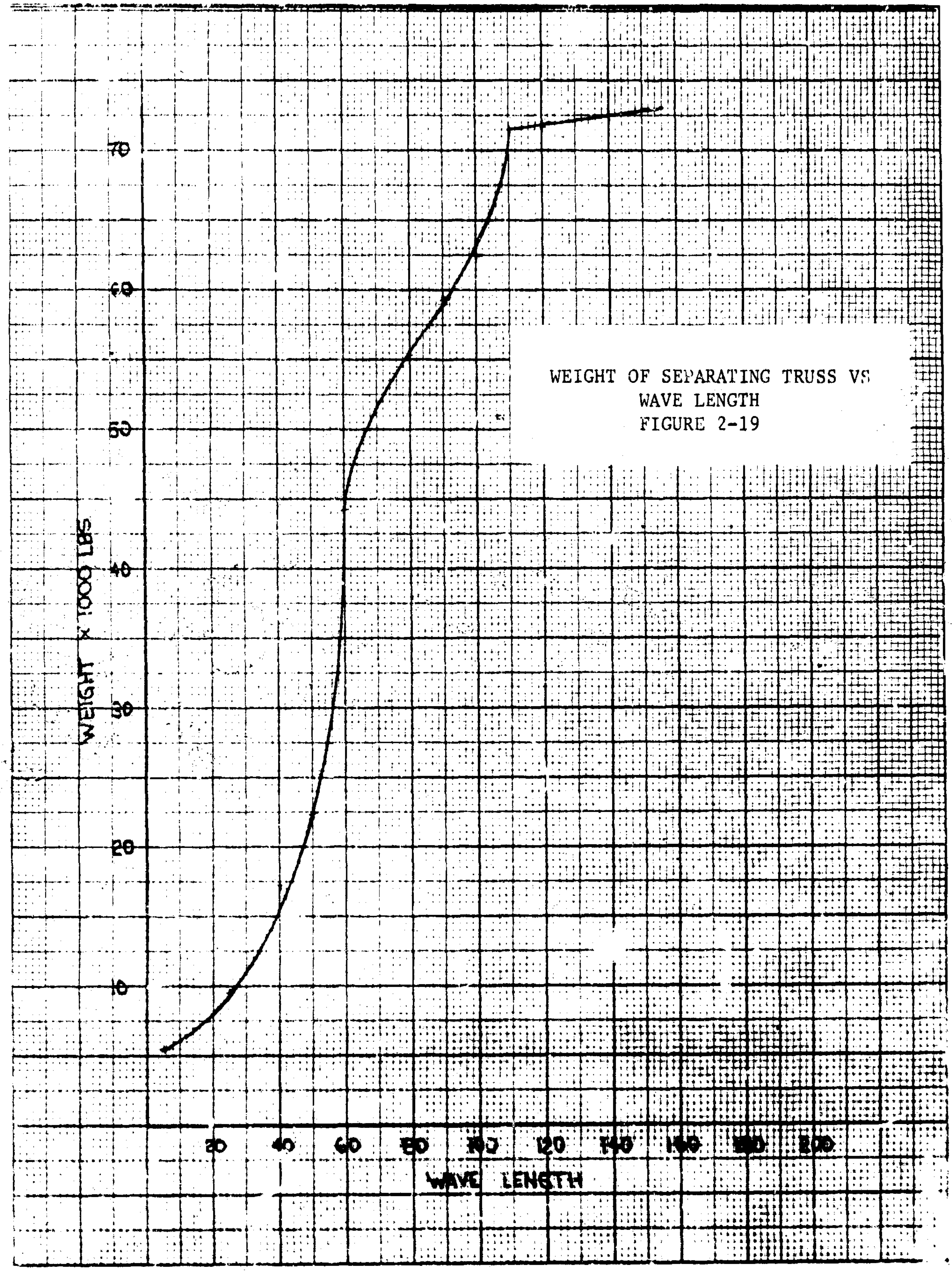
FIGURE 2-16



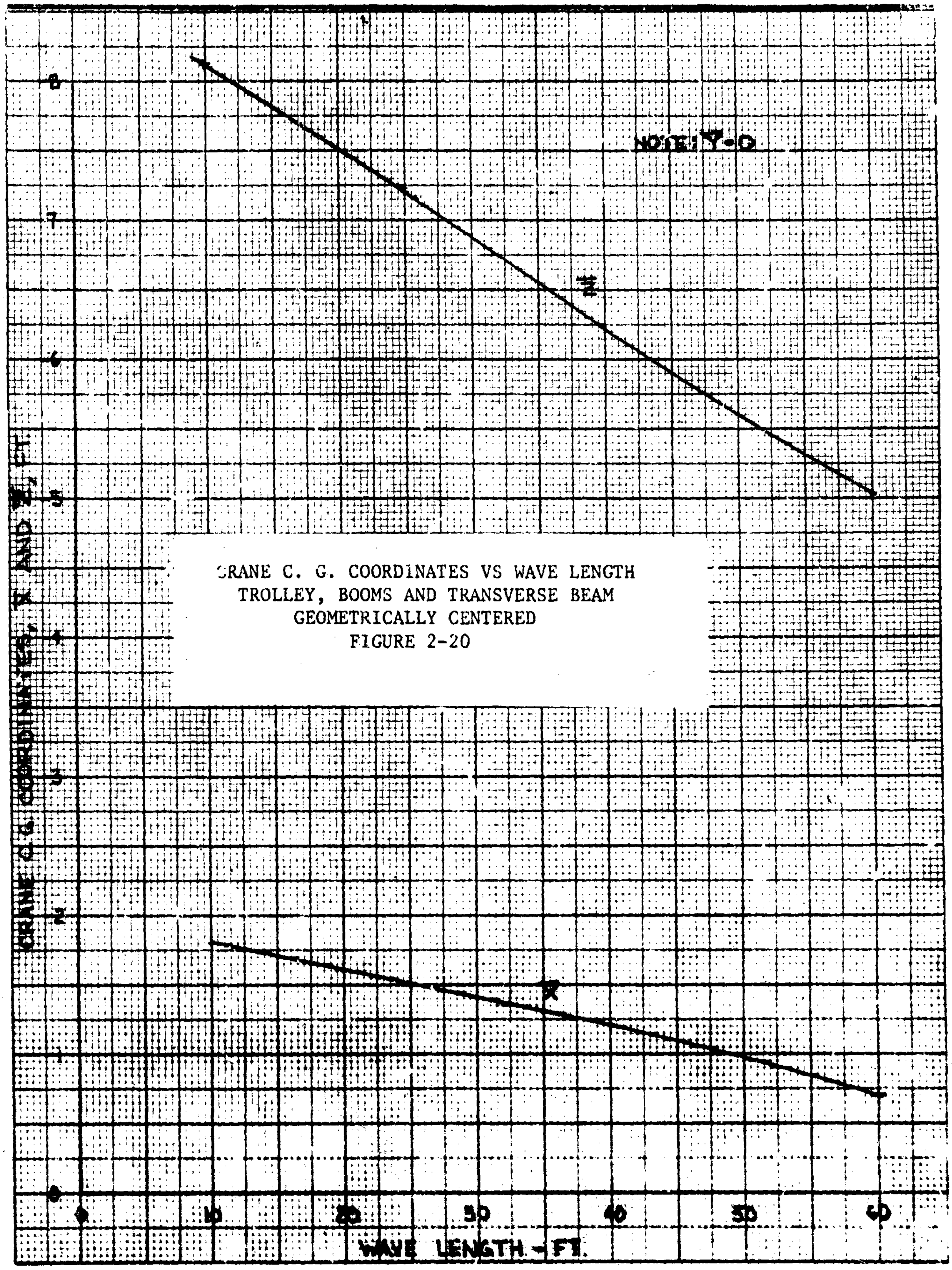
WEIGHT OF KINGPOST VS WAVE LENGTH
(NO SEPARATION TRUSS BETWEEN HULLS)
FOR SINGLE POST
FIGURE 2-17



FORCE ON SEPARATING TRUSS VS WAVE LENGTH
FIGURE 2-18



WEIGHT OF SEPARATING TRUSS VS
WAVE LENGTH
FIGURE 2-19



CRANE C. G. COORDINATES VS WAVE LENGTH
TROLLEY, BOOMS AND TRANSVERSE BEAM
GEOMETRICALLY CENTERED
FIGURE 2-20

TABLE II. CRANE AND KINGPOST WEIGHTS (NO SEPARATION TRUSS)

ITEM	MACHINERY WEIGHT‡	STRUCTURE WEIGHT* 10' WL*	25' WL	40' WL	60' WL
Trolley Machinery	28.0				
Trolley Structure		12.0	12.0	12.0	12.0
Port Boom		115.8	115.8	115.8	115.8
Starboard Boom		113.8	113.8	113.8	113.8
Boom Extending Machinery	15.3				
Boom Support Wheels	21.6				
Transverse Beam		111.2	111.2	111.2	111.2
Articulated Support Units(2)	13.0				
Port Fore and Aft Link		82.3	99.6	133.1	213.4
Starboard Fore and Aft Link		82.3	99.6	133.1	213.4
Forward Transverse Link		50.7	104.5	209.0	459.0
Aft Transverse Link		50.7	104.5	209.0	459.0
Hull Separation Machinery	20.4				
Sub-Total, Machinery/Structure	98.3	618.8	761.0	1037.0	1697.6
Sub-Total, Crane Machinery plus Structure		717.1	859.3	1135.3	1795.9
Kingpost Structure (4) and Hoist Tackle (4)	15.4 +	360.0			
or	18.4 +		568.0		
or	24.3 +			972.0	
or	38.4 +				1560.0
Hoist Winch (4), (In Hold)	72.0				
Sub-Total, Machinery/Structure	185.7	1077.1			
or	188.7		1427.3		
or	194.6			2107.3	
or	208.7				3355.9
TOTAL, Machinery/Structures		1262.8	1616.0	2301.9	3564.6

‡ All weights in KIPS

* WL = Wave Length

TABLE III. TOTAL WEIGHT OF CRANE, KINGPOSTS, AND SEPARATION TRUSSES FOR VARIOUS WAVE LENGTHS

In the following tabular summary, the weight of a pair of separation trusses designed for various wave lengths are added to the weight of the crane and four kingposts designed for wave lengths of 10, 25, 40, and 60 feet.

Wave Length	Weight of Trusses (2)	10' Wave Length Crane & Post Weight	10' Wave Length Total Weight	25' Wave Length Crane & Post Weight	25' Wave Length Total Weight	40' Wave Length Crane & Post Weight	40' Wave Length Total Weight	60' Wave Length Crane & Post Weight	60' Wave Length Total Weight
FT.	KIPS	KIPS	KIPS	KIPS	KIPS	KIPS	KIPS	KIPS	KIPS
5.5	11.0	1262.8	1273.8	1616.0	1627.0	2301.9	2312.9	3564.6	3575.6
25	18.8		1281.6		1634.8		2320.7		3583.4
40	34.4		1297.2		1650.4		2336.3		3599.0
50	44.8		1307.6		1660.8		2346.7		3609.4
60	88.4		1351.2		1704.4		2390.3		3653.0
80	110.6		1373.4		1726.6		2412.5		3675.2
90	118.6		1381.4		1734.6		2420.5		3683.2
100	125.0		1387.8		1741.0		2426.9		3689.6
110	142.8		1405.6		1758.8		2444.7		3707.4
120	143.4		1406.2		1759.4		2445.3		3708.0
140	144.6		1407.4		1760.6		2446.5		3709.2

A comparison of the weights of the components which are different in the two (2) designs is presented in Table IV below. The difference in weight obtained in going from Design No. 1 to Design No. 2 is also shown.

TABLE IV. COMPARISON AFFECTING CRANE WEIGHT

COMPONENT	DESIGN NO. 1	DESIGN NO. 2	DIFFERENCE
Extensible Boom	229.6K	123.3K	-106.3K
Boom Extending Machinery	15.3K	21.5K	+ 6.2K
Boom Support Wheels	21.6K	13.1K	- 8.5K
Miscellaneous	-	3.7K	+ 3.7K
TOTAL	226.5K	161.6K	-104.9K

2.8 OPERATIONS

Both crane designs have the capability of transferring containers between a container ship and dock, causeway, landing craft, lighter, or amphibious vehicle as shown on Enees Dwg. Nos. 71031-9 and 71031-10. Only Design No. 1 is capable of burtoning break-bulk cargo as shown on Eness Dwg. No. 71031-11. The preceeding material in this report endeavored to illustrate the many aspects of the crane and the areas in which certain factors will limit or extend the operations. For the purpose of this report, operation was assumed to be in a 40-foot wave without the use of the trusses.

Operations with lighters, landing craft, etc., will be limited when the trusses are used. In Appendix L investigation has been made in this regard using the Larc-LX and the LCM-8. Figure 2-21 indicates the variation of clearance over the separation truss with wave height for these vessels. Where trim or list develope, the limiting wave height should be reduced accordingly.

2.9 COSTS

The cost of the crane, including links, posts, and winches will be predicated upon the wave length related to the desired sea state. These costs are summarized in Table V.

CLEARANCE BETWEEN LIGHTER AND TRUSS
FIGURE 2-21

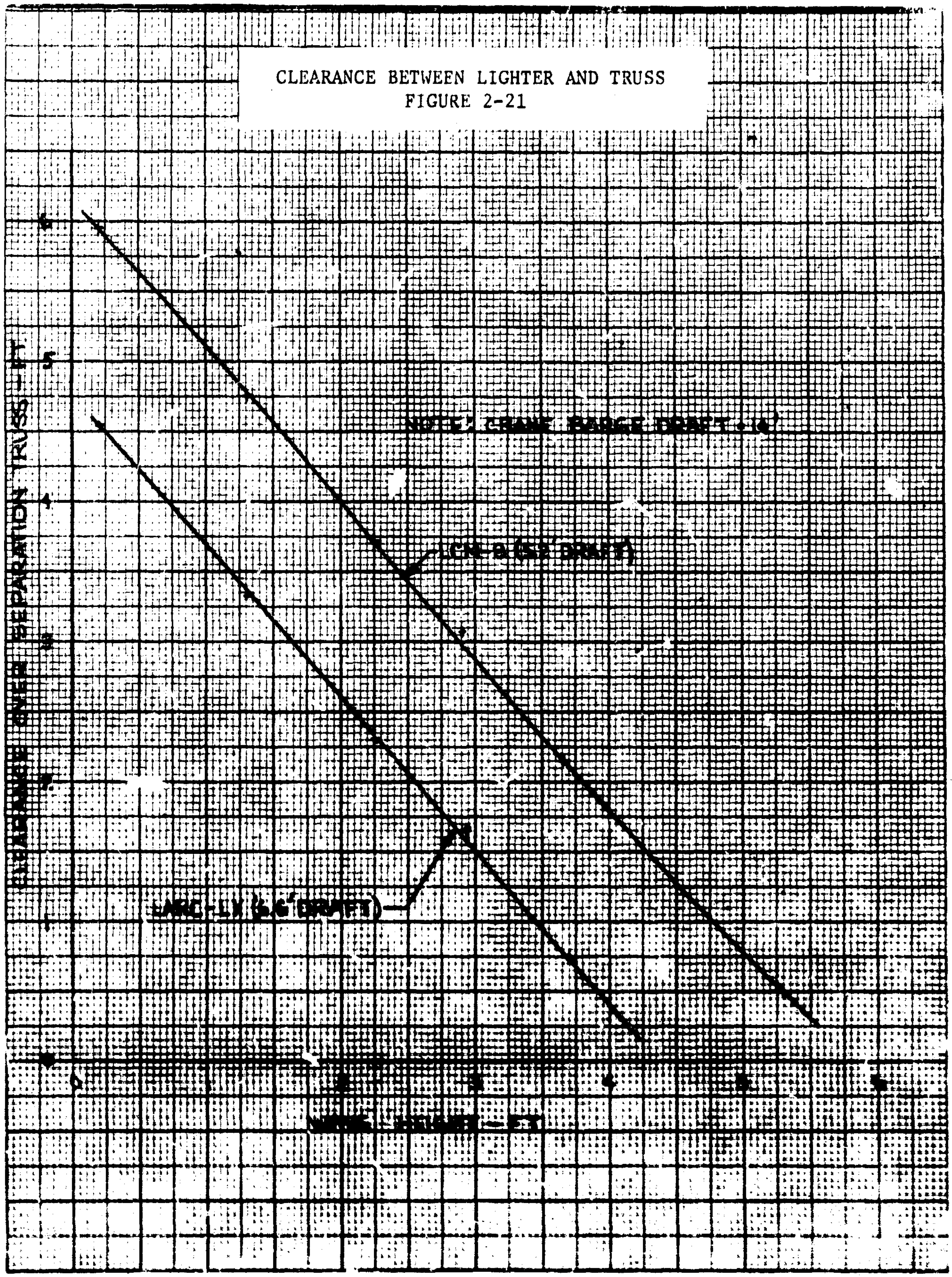


TABLE V. EFFECT OF WAVE LENGTH ON COST OF CRANE

DESIGN NO. 1

Wave Length	10'	25'	40'	60'
Machinery	$196,000	$196,000	$ 196,000	$ 196,000
Crane Structure	320,000	386,000	507,000	801,000
Kingposts	160,000	254,000	434,000	695,000
Winches	80,000	80,000	80,000	80,000
Hoist Tackle	15,000	18,000	23,000	36,000
Total Cost (no truss)	$771,000	$934,000	$1,240,000	$1,808,000

DESIGN NO. 2

Wave Length	10'	25'	40'	60'
Total Cost	$723,500	$886,500	$1,197,500	$1,761,500

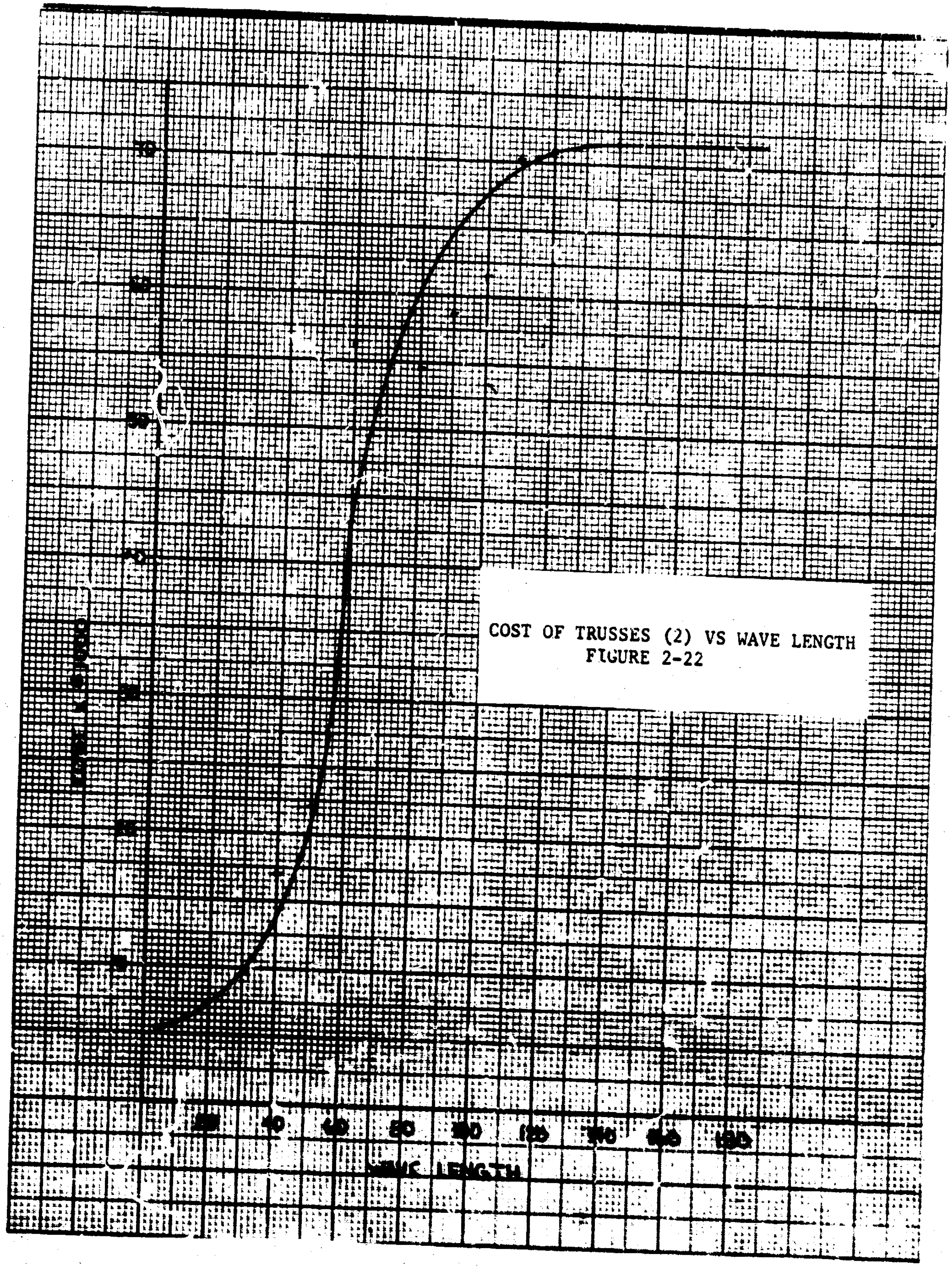
Considering that the crane designed for the 10-foot wave length could normally be used with a truss, the cost of the crane using a truss for a 180 foot wave length will be $842,000 for Design No. 1 and $794,000 for Design No. 2. Should the crane be required to operate in a fully developed sea state of 3 where the wave length is 70 feet, the cost for Design No. 1 is $824,000 and $776,000 for Design No. 2. Costs for various combinations of crane and truss can be determined using Figure 2-22, which shows a curve of truss cost versus wave length.

2.10 DISCUSSION

The selection of the crane is dependent upon the wave length as it relates to sea state. The selection of the various options presented and the costs illustrated permits one to arrive at the optimum crane design.

In summary, Design No. 1 has a greater weight and cost than Design No. 2. This is primarily because the outreach of Design No. 2 is obtained by a single extensible boom, whereas two (2) boom structures are provided in Design No. 1. Further, Design No. 1 provides extension of the boom independent of load, while Design No. 2 requires both the load and the boom be accelerated and moved each time a load is transferred.

In operation, Design No. 1 permits selective extension of the booms, while Design No. 2 does not. A desired outreach can be obtained



COST OF TRUSSES (2) VS WAVE LENGTH
FIGURE 2-22

for both designs; however, the boom of Design No. 2 will have considerable projection beyond the container except in the case of extreme outreach. In close operations this feature limits the spotting of the container.

Design No. 2 requires greater horsepower because of the boom movement, but the cost of the additional horsepower is not too significant in the overall cost of the crane.

Design No. 1 will permit extension of the boom on one side so as to counterbalance the beam on the other side. Design No. 2 does not provide this feature and, consequently, the system undergoes greater list than in Design No. 1.

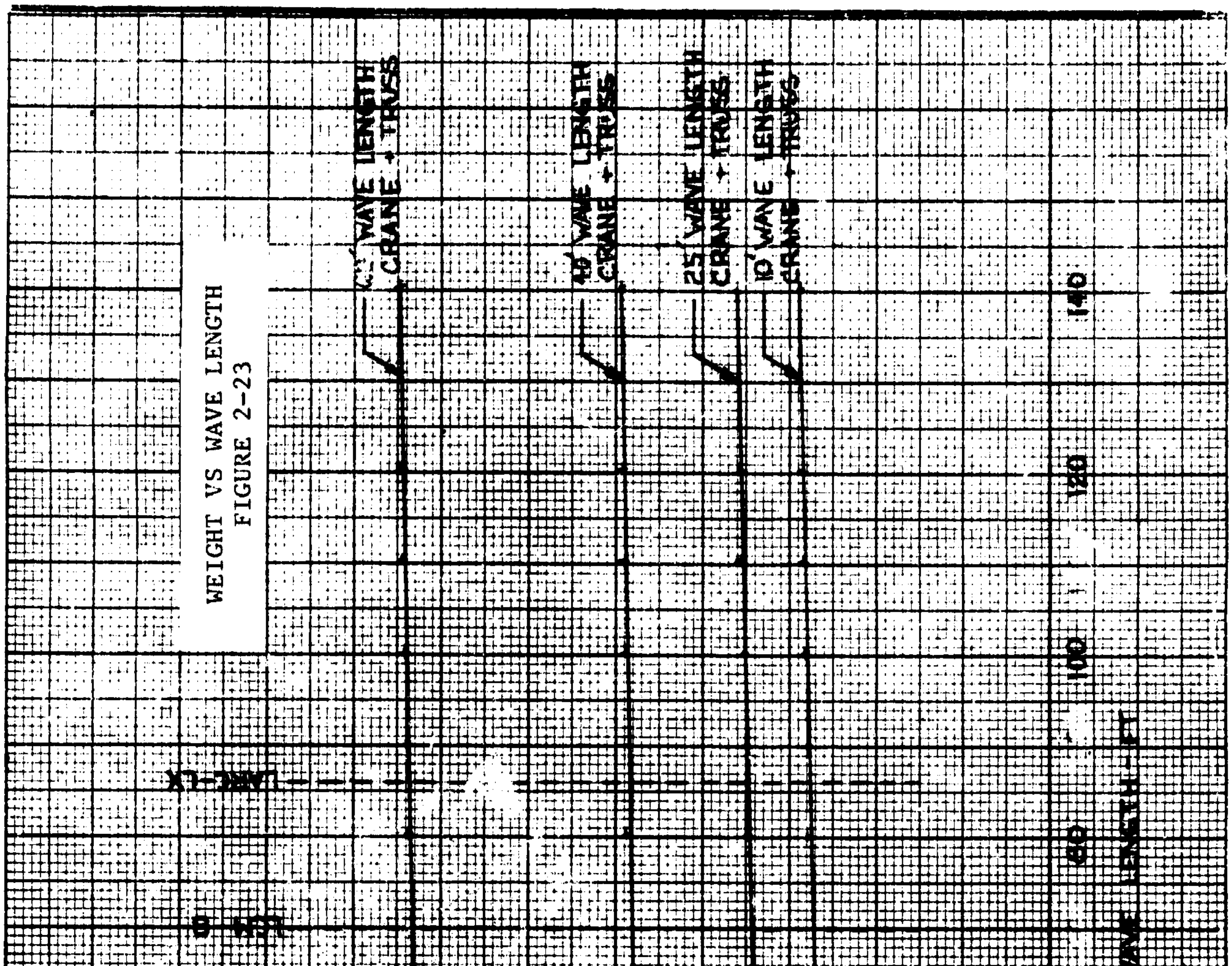
Design No. 1, with the feature of two (2) extensible booms, provides for the addition of a winch at each end so that burtoning of break-bulk is readily achieved. The system permits good spotting over the hatch or over lighters, etc. The addition of a winch in the trolley for a single hook on Design No. 2 limits the spotting unless the boom and the trolley are moved. The movement of the boom to accomplish break-bulk operation will be quite inefficient and will impose greater acceleration forces to the load. The two (2) winches of Design No. 1 must be established in the final stage in order that control can be provided in the cab.

The trolley of Design No. 2 is propelled by a system of wire rope. As compared to the traction drive, the rope drive would not permit any loss of motion, as could occur with the traction drive because of loss of friction. From the standpoint of the operator, the rope system will cause jogging and rough travel. The traction drive, with hard rubber tires, will react much like a car providing gradual acceleration and deceleration, resulting in a more comfortable ride to the operator of the crane. With respect to maintenance and reliability, the direct traction drive offers greater reliability and maintenance will be less than with the wire rope drive.

While Design No. 1 is approximately 7 percent costlier, the flexibility with some improvement in operational features appears to be worth the additional investment. The cost saving feature is the separating truss.

Figure 2-23 indicates the wave lengths for the four (4) crane sizes considered and where a truss is required, as well as the area where the LCM-8 is restricted in the operation.

The need for the truss between the hulls has been stressed because of the many circumstances in which closing forces may develop. The truss is also a means of reducing the design requirements of the crane. The hydrodynamic considerations as applied to the problem are believed to be conservative. A reduction in the design criteria can only be indicated by appropriate testing and verification can only be obtained by instrumentation on the prototype.



WEIGHT VS WAVE LENGTH
FIGURE 2-23

The problem of engaging and disengaging containers in a prevailing sea state was not considered a part of the effort required under this contract. Extensive work has been accomplished in this area, but an acceptable solution has not as yet been made available. The major factor affecting the engagement of the load is the wave height. The fact that the two (2) hulls of the barge are separated will do something to suppress the wave. There is the possibility of designing one of the trusses so that it will act as a barrier to the heading seas and result in a greater suppression of the wave so that the operation may be possible in higher sea states than presently possible. It is recommended that tank tests conducted for verification of the hydrodynamic considerations also include a barrier and study of lighter motions.

The crane design as presented incorporates a number of features with certain backup equipment which will permit operation in event of partial failure. This feature is evident in the hoist, transverse drive, and the boom extension drive, where twin motors and reducers have been provided. The dual drives (in the case of the hoist only) require the brake be set on the inoperative unit so that the operation may be continued at half its normal speed. With respect to the transverse drive, the brakes must be released; however, the boom extension for Design No. 1 would require engagement lugs between each boom so that one operating drive can move the inoperative boom. Design No. 2, having two (2) separate drives, requires the brake be released on the inoperative drive. In event of failure of the fore and aft drive, the barge may be shifted by a change in settings on the constant tension winches. Raising the crane would not be possible in event of a winch failure; however, if raised, the crane can be lowered without power by releasing the brake on the winch.

Two (2) cabs have been indicated on the trolley. This is primarily for the purpose of providing maximum visibility of the operation. The control can be transferred from one cab to another. In event that dual control is desired, an operator can be stationed in each cab; each operator will have optimum vision from his side. A means of communication and signals would be required for dual operators.

Access to the crane, when in the raised position, is via a platform on the spreader. At this platform, a control will be provided which will enable the trolley to traverse as well as raise and lower the spreader in elevator fashion. When the hulls are separated the control will provide a means of access between the two (2) hulls.

The operation and servicing of the crane will require a minimum of two (2) operators for each eight (8) hours of operation. For servicing, a competent electrician and engine maintenance man will be required. These can be normal shipboard personnel; however, for operation the selected personnel must undergo training in crane control. Under normal operation and maintenance, no outside support will be required beyond the initial familiarization period. It is anticipated that at least once a year the crane equipment will undergo thorough examination and checkout for all operational aspects as well as tested under load for a check on the mechanical and structural components of the crane. Spares will be provided for at least one (1) year of operation.

The design of the crane follows usual practice and should not present any problem areas. The transverse links and kingposts have been analyzed using the present state of the art and modified as found applicable to the concept. Their design does not present any problem except as to the severity of the conditions and circumstances under which operations will take place. There is evidence that the concept is feasible in sea states approaching 6; however, the engagement or disengagement of containers in lighters or in landing crafts is quite questionable unless relative motion of the crafts can be suppressed. In this sea state, the concept might be considered to have limited success; however, the concept has potential to suppress the movements so that anticiapted success in a fully developed sea state of 3 appears to be realistically possible.

As previously noted, the steel considered is basically one which conforms to ASTM 441. During the detail design steels of lower and higher strength than ASTM 441 may be justified considering weight reduction and economics. Also, the availability of new alloys such as ASTM 572, which is replacing ASTM 441, shall be investigated. Table VI indicates the relative cost and weight savings using ASTM A36 steel as the base.

TABLE VI. COST AND WEIGHT COMPARISON OF ALLOY STEELS

ASTM NO.	MATERIAL COST	FABRICATION COST	WEIGHT
A36	1.00	1.00	1.00
A441	1.30	1.00	.75
A572	1.13	1.00	.75
A514	2.28	2.00	.50

The ASTM 572, which has similar properties to ASTM 441, is the most economical and may be used in a majority of cases. The high strength steel A514 has been considered for use in areas where weight is critical. The final selection will depend upon the fabricator's economics in complying with the specifications.

2.11 RECOMMENDATIONS

Review of the findings in this study indicates the following recommendations:

1. Crane structure should be designed for forces resulting from a 10 foot wave length with trusses and conform to Design No. 1. Table VII presents the estimated weight and cost on the recommended system.

TABLE VII. ESTIMATED WEIGHT AND COST

ITEM	WEIGHT (LBS.)	COST
Crane for 10' Wave Length	1,262,800	$771,000
Truss for 70' Wave Length	51,500	53,000
TOTAL	1,314,300	$824,000

2. Trusses to suit various sea states can be provided; however, operations in a sea state of three (3) having a 70 foot wave length is the object. Therefore, this truss is recommended because it will more than satisfy the conditions required for the present state of the art of engaging and disengaging the spreader from the containers. In event of future improvement in the art and operations in a higher sea state become possible, the trusses can be replaced with a heavier design.

3. Model tests should be performed which will provide parameters for the structural design of the crane and sea keeping characteristics.

4. While the engagement of the spreader to the container was not part of the effort covered by this report, it is, nevertheless, a requirement for the successful operation. It is recommended that additional work be accomplished in this area. This work should consist of an engineering study employing a spreader that is capable of being lowered near to but not onto the top of the container. At this point means should be provided to orient the spreader to the container so that lowering can be achieved without outside assistance. When the study has been accomplished, the design of the spreader should be made followed by a working model of approximately 1 1/2 inches = 1 foot. The model should mechanically simulate motion at sea and the spreader should be remotely controlled.

SECTION 3

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3.1.1	CRANE TEST PROGRAM	3-2
3.1.2	HULL TEST PROGRAM	3-3
3.2	CONTRACT PLANS, SPECIFICATIONS AND POST CONTRACT WORK	3-3
3.2.1	PROPOSED WORK FOR DEVELOPMENT OF CRANE FEATURES	3-3
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SECTION 3

3.0 PROTOTYPE DEVELOPMENT

The first step in the development of a prototype will be the preparation of contract plans and specifications which will also require a program of research into known problem areas. Upon completion of this data, bids can be solicited and contracts awarded so that construction can be undertaken.

3.1 RESEARCH

Concurrent with the development of the contract plans and specifications for the facility we would recommend two model testing programs; One for the crane features and one for the hull.

3.1.1 CRANE TEST PROGRAM

Using models of a containership hull, barge crane concept and landing craft, develop wave lengths and heights in accordance with concept study. Instrument the models to provide data on the effect of the following conditions for various waves up to a maximum length of 180 feet and height of 15 feet:

Beam Seas

1. Alongside a pier without protection

2. Alongside a pier with containership on the outside

3. Straddling a causeway

Heading Seas

1. Alongside anchored containership

2. Alongside anchored containership with landing craft secured between barge hulls

3. Alongside anchored containership with landing craft secured between barge hulls with a barrier at one end

Objectives

1. Determine parameters for the loading on structural members in various sea states

2. Determine relative motions of barge with containership and landing craft with barge

3. Study problems of pendulation of loads

4. Study problems of spreader bar attachment and release

5. Study effects of relative motion on off-loading operations

6. Examine stability in survivability conditions

3.1.2 HULL TEST PROGRAM

Using model of the hull in the at-sea mode, that is with the hulls together, run towing resistance tests at three drafts to determine power requirements at speeds up to 20 knots.

Using same model with skegs at various angles determine best angle to reduce yawing.

Using self-propelled model check on sea-keeping qualities by test runs with head seas, quartering seas and following seas at various wave heights.

Objective

Determine best hull configuration and skeg alignment for easy towing and best sea-keeping qualities. Check stability in survivability conditions.

3.2 CONTRACT PLANS, SPECIFICATIONS AND POST CONTRACT WORK

Contract plans and specifications for the crane and the barge are to be prepared in sufficient detail to establish the design fully and to be able to obtain bids for construction.

Estimates of time and costs for research, preparation of plans and specifications, inspection and testing are given in the overall schedule on Page

3.2.1 PROPOSED WORK FOR DEVELOPMENT OF CRANE FEATURES

1. Prepare material for subcontracting tank testing, witness tests, and review report.

2. Relate parameters determined by testing and prepare contract drawings and specifications of crane and supports to the deck of the barge.

3. Review building plans, calculations and instruction books.

4. Provide periodic inspection and witness final testing.

3.2.2 PROPOSED WORK FOR DEVELOPMENT OF HULL FEATURES

1. Coordinate with model tank on test program, witness tests and review reports.

2. Prepare contract plans and specifications incorporating data from crane design and from model test program.

3. Review building plans, calculations and instruction books.

4. Provide periodic inspection and witness final testing.

3.3 COST OF BUILDING PROTOTYPE

Once the pre-contract engineering and testing are completed, and bids for construction are obtained, construction of the prototype can proceed.

The estimated cost and time to complete are given below for the crane and the hull.

3.3.1 CRANE COST

As noted in Article 2.9 of Section 2 where the effect of wave lengths on cost of crane is fully discussed, where a truss to connect the hulls is not used, the cost of the crane structure and kingposts goes up rapidly as the wave length increases. It is out recommendation trusses be used, in which case the cost of the crane for a 180 foot wave length is $842,000 for Design No. 1 and $794,000 for Design No. 2.

It is estimated that it would take one year after the award of contract to build the crane.

3.3.2 HULL COST

For preliminary estimating of cost the normal practice in the marine field is to use up-to-date costs per ton and apply this to the estimated weight of the vessel. We have access to current shipyard bids for barges generally similar to this hull and from these prices have developed a price per ton which includes steel and the standard outfit. To this we will add the special items applicable to this hull.

Estimated weight 2200 tons @ $1100/ton	=	$2,420,000
Crane generators and electrical equipment	=	54,000
Installation of kingposts and crane	=	100,000
Total	=	$2,574,000
Add for self-propelled 2 Murray & Tregurtha units		330,000
Total	=	$2,904,000

Prices have been escalating at a rate of about 10 percent per year.

3.3.3 DEVELOPMENT TIME AND COST SUMMARY

On Page   is a proposed schedule covering all the various phases for the development of the prototype.

3.4 OPERATING COSTS

Assuming that this is a Navy owned facility it is difficult for us to assess the operating costs. In commercial operation the biggest factors are capital costs, insurance and crew costs. This is subject to further review.

3.5 RECOMMENDATIONS

We believe the concept as developed offers the promise of being a useful and worthwhile facility which meets the demands of the ELF program.

It is recommended, therefore, that serious consideration be given to building a prototype which could prove useful at ports and facilities not now equipped with container handling capability.

SCHEDULE FOR BUILDING PROTOTYPE OF ELF MATERIALS HANDLING CONCEPT

TASK NO.	DESCRIPTION	TIME: FISCAL YEAR 1 / FISCAL YEAR 2 / F.Y. 3	ESTIMATED COST
1	MODEL TESTING SPECS. AND P.O.		$ 3,000
2	WITNESS TESTS, REVIEW DATA, COORDINATE WITH DESIGN		25,000
3	REVISE CALCULATIONS USING TEST RESULTS		3,000
4	PREPARE CONTRACT PLANS & SPECIFICATIONS FOR CRANE		15,000
5	PREPARE CONTRACT PLANS & SPECIFICATIONS FOR BARGE		29,000
6	ASSIST IN BID ANALYSIS		2,000
7	PLAN APPROVAL CRANE CONTRACTORS' DWGS., CALCS., ETC.		8,000
8	PLAN APPROVAL BARGE CONTRACTORS' DWGS. CALCS., ETC.		10,000
9	INSPECTION & TESTING OF CRANE CONSTRUCTION		5,000
10	INSPECTION & TESTING OF BARGE CONSTRUCTION		15,000
11	PREPARE STUDY OF SPREADER ENGAGEMENT WITH CONTAINER		2,000
12	DESIGN SPREADER		3,000
13	CONSTRUCT SPREADER MODEL TO VERIFY DESIGN		6,000
14	CRANE MODEL TEST		25,000
15	BARGE MODEL TEST		15,000
16	CRANE CONSTRUCTION		[illegible]
17	BARGE CONSTRUCTION		[illegible]

SECTION 4

LIST OF HULL DRAWINGS

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LIST OF HULL DRAWINGS (con't)

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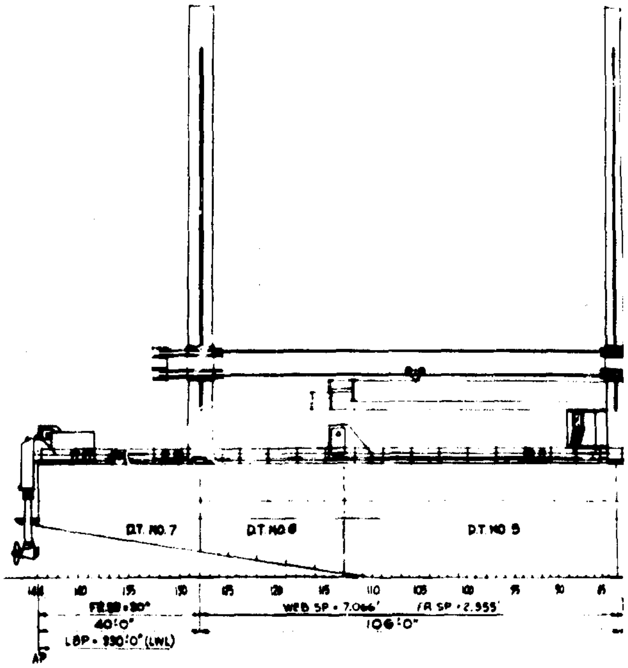
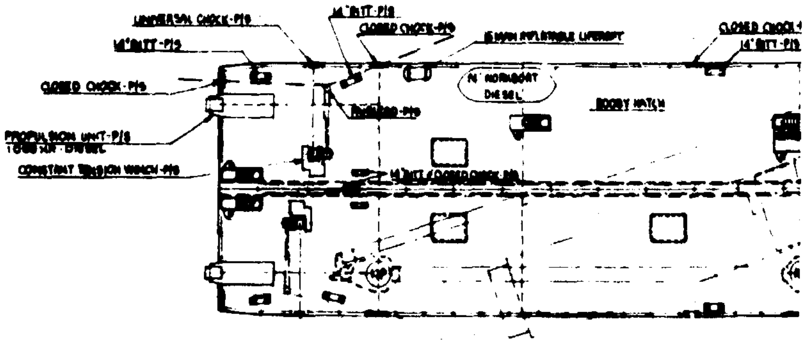
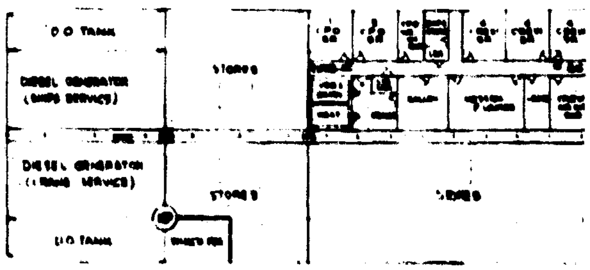

D.T. NO. 7
D.T. NO 6
D.T. NO 5
WEB SP = 7.066'
FR SP = 2.955'
40'-0"
106'-0"
LBP = 330'-0" (LWL)
AP

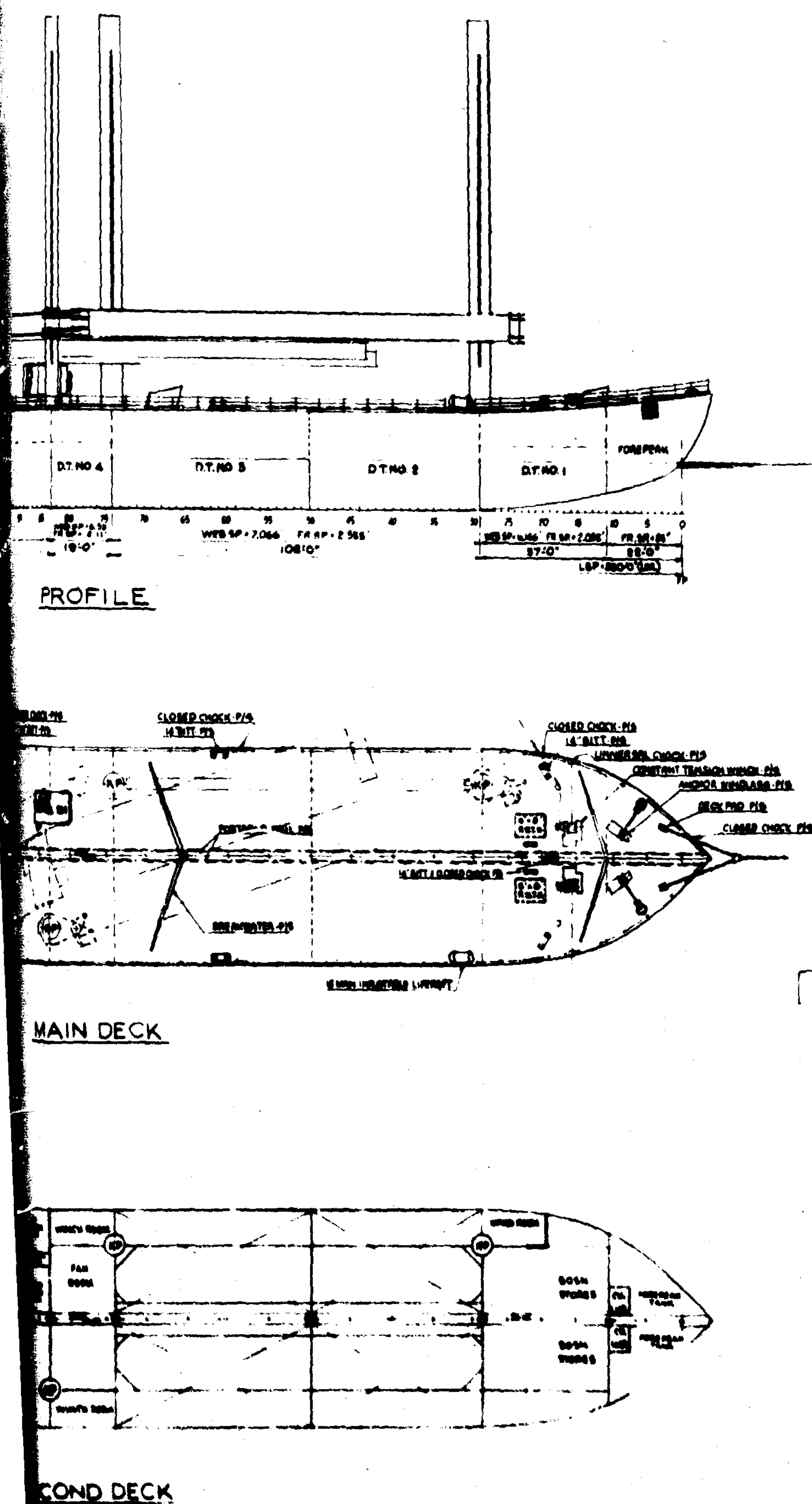
PI


UNIVERSAL CHOCK-P/S
14" BITT-P/S
CLOSED CHOCK-P/S
PROPULSION UNIT-P/S
CONSTANT TENSION WINCH-P/S
BOOBY HATCH
CLOSED CHOCK

M


D.O. TANK
DIESEL GENERATOR (SHIPS SERVICE)
DIESEL GENERATOR (CRANE SERVICE)
STORES

SE



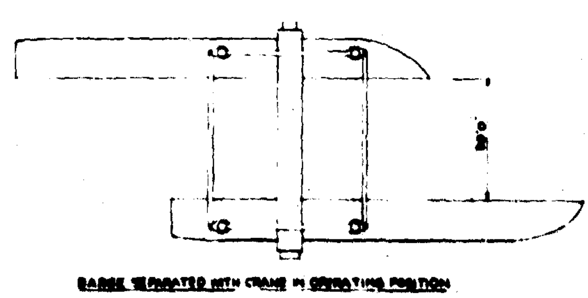
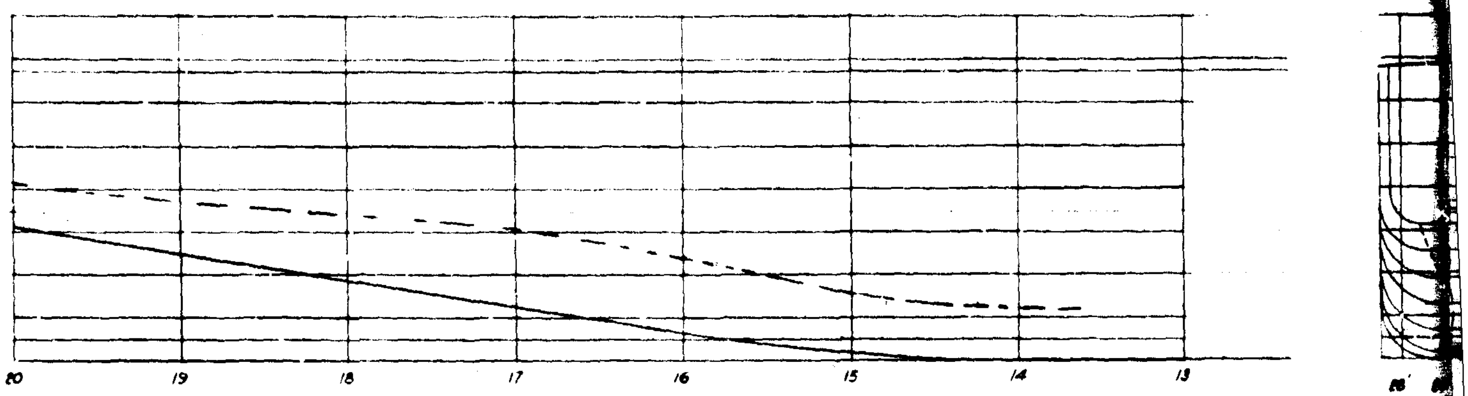


FIGURE 1-1. CONTAINER CRANE BARGE
GENERAL ARRANGMENT
(SCALE 1/16" = 1'0"; DWG. NO. 5834-1


20
19
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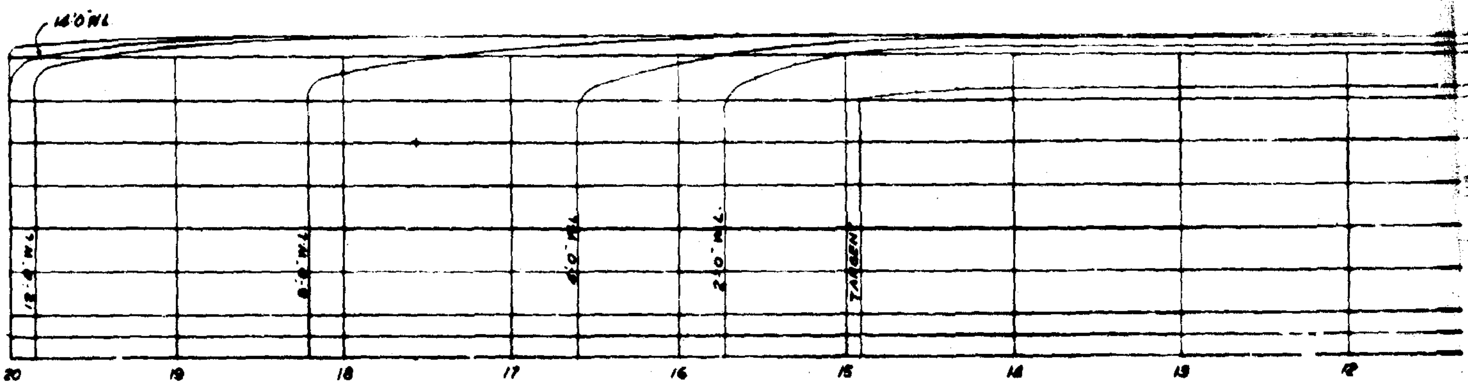
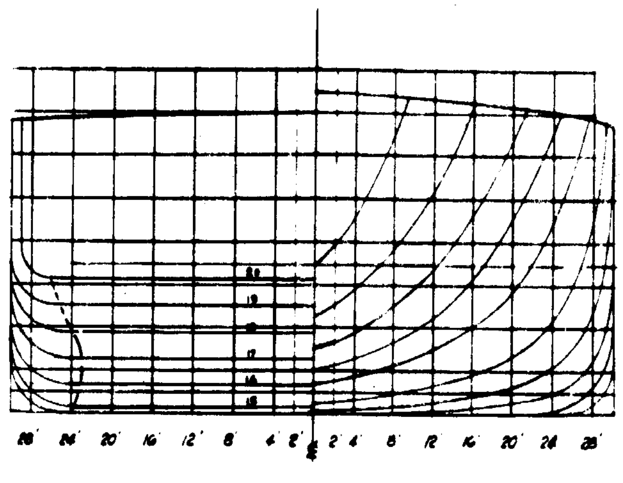
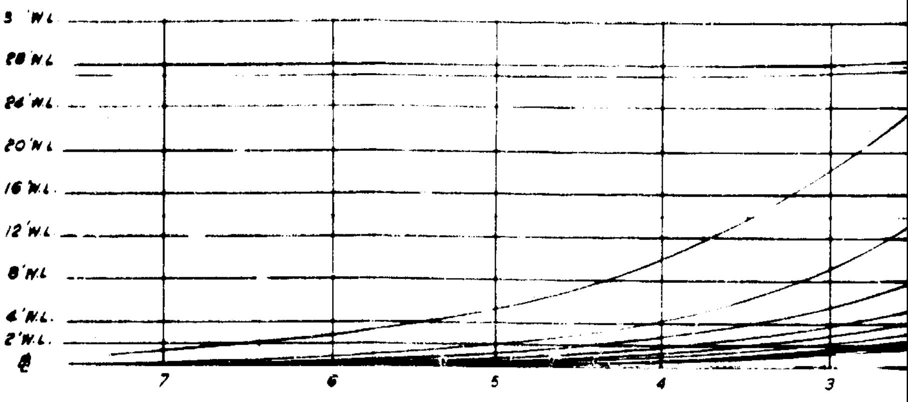
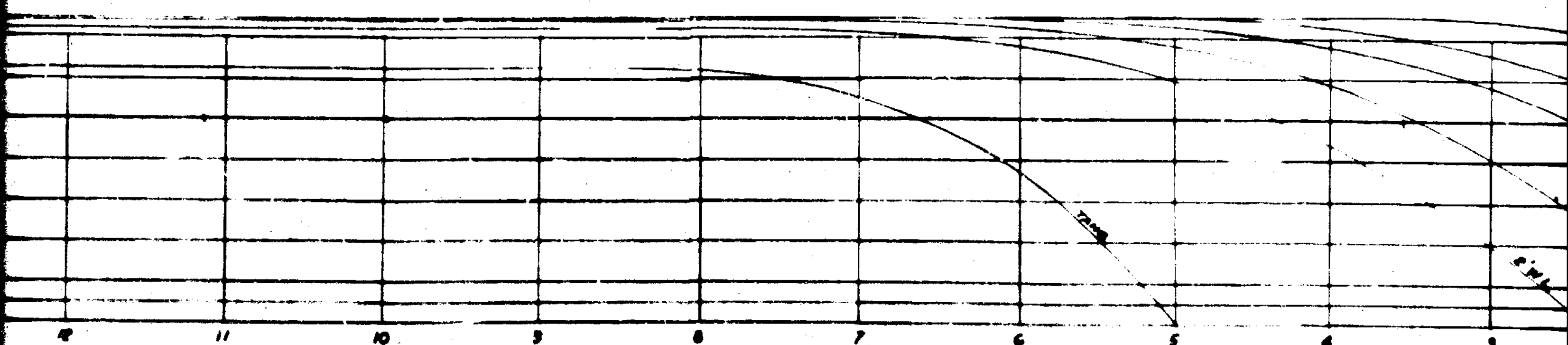

14'0" WL
12'0" WL
8'0" WL
6'0" WL
2'0" WL
TANGENT
20
19
18
17
16
15
14
13
12

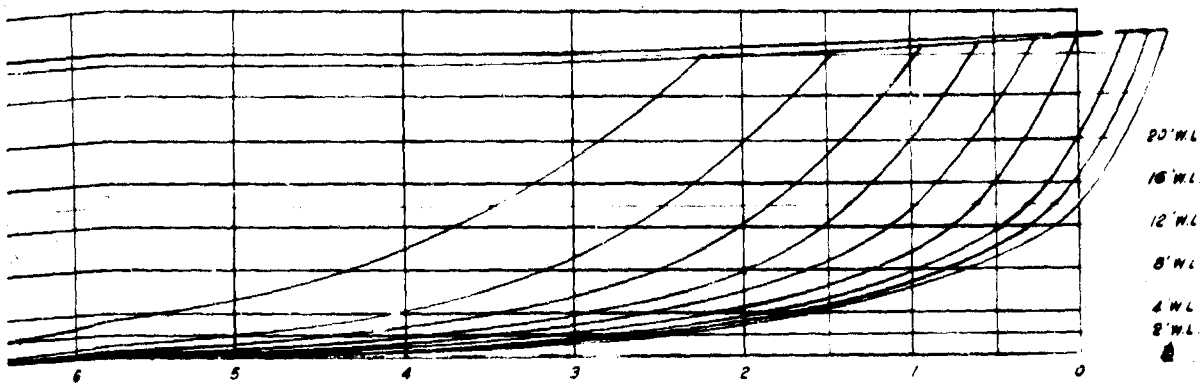




PRIN

LENGTH,W.
BEAM_ _ _
DEPTH_ _
DRAFT, DES
C_b- - - - -





PRINCIPAL DIMENSIONS

LENGTH, W.L.	330'-0"
BEAM	60'-0"
DEPTH	27'-0"
DRAFT, DESIGN	14'-0"
C_b	.75

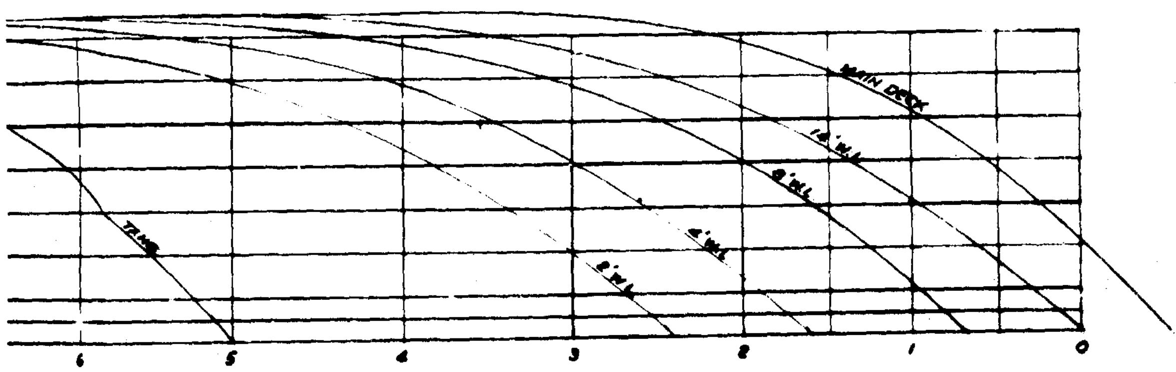
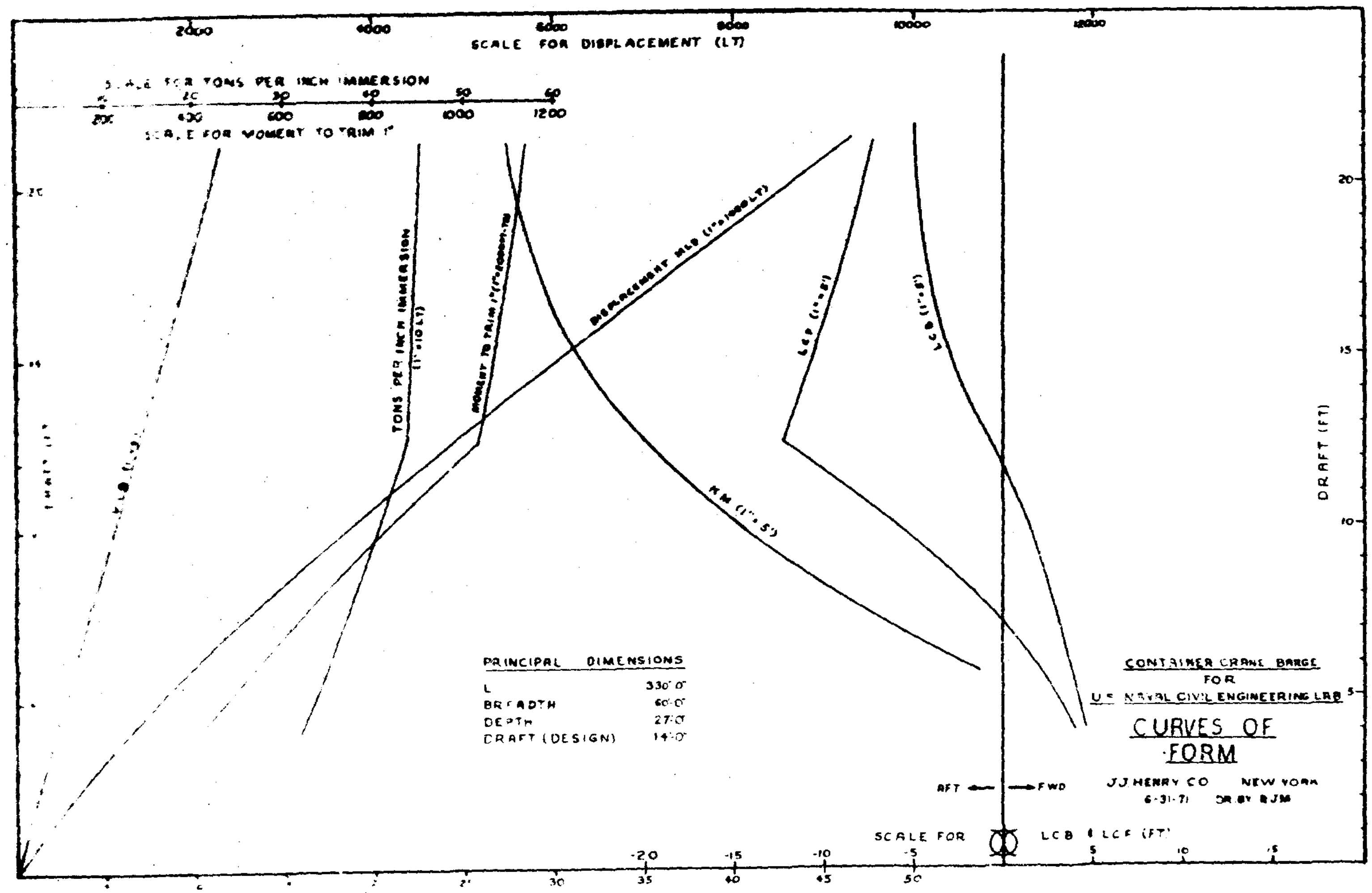
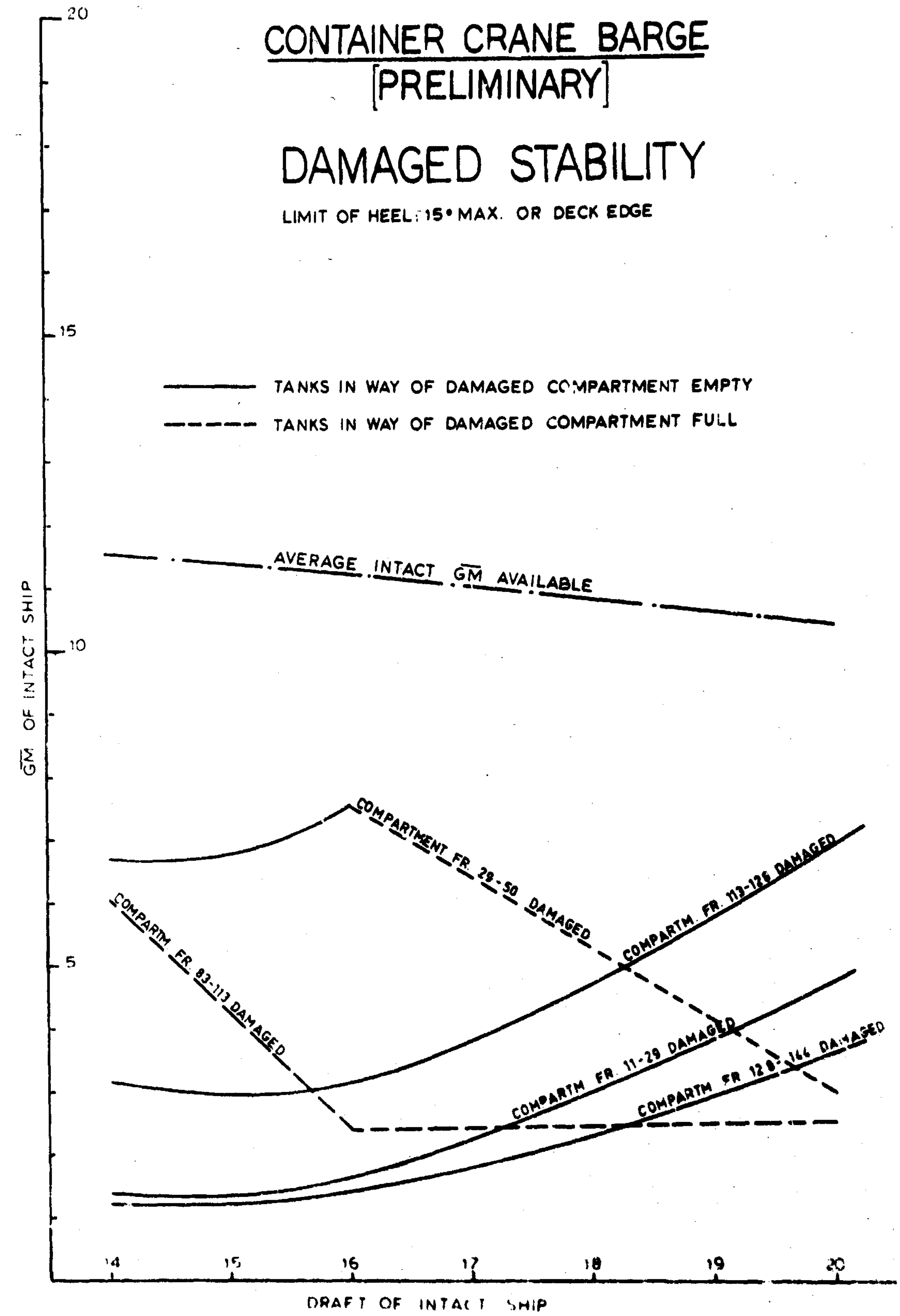


FIGURE 1-2. CONTAINER CRANE BARGE
GENERAL ARRANGEMENT
(SCALE 1/16" = 1'0"; DWG. NO. 5843-3)



CURVES OF FORM
FIGURE 1-3



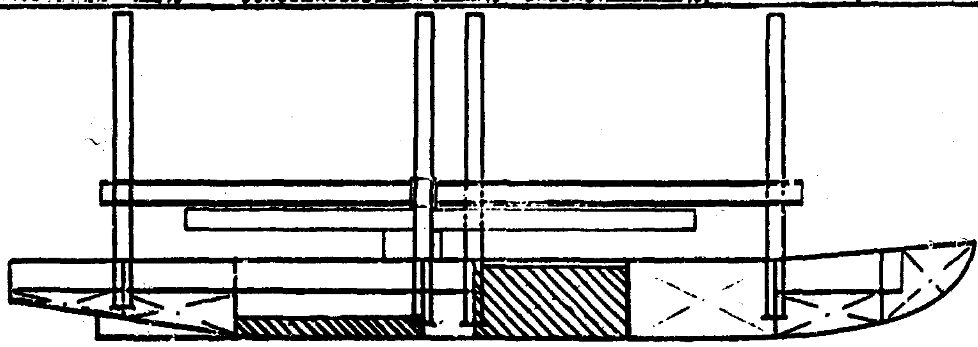
DAMAGED STABILITY
FIGURE 1-4

SYMBOLS: S.W. · F.W. · F.O. OR D.O. · OIL CARGO · BULK CARGO · DRY CARGO · REFRIG. CARGO

TRIM & STABILITY SUMMARY

CONDITION OF VESSEL: TOWING CONDITION 14' DRAFT NO. ___ DATE 07.15.71 PAGE 1

CARGO ___ % CONSUMABLES ___ % BALLAST ___ % BY G.L. JOB NO. 5934



330'x 60'x 27' CONTAINER CRANE BARGE

REF. LINE FOR V.C.G. — REF. LINE FOR L.C.G.

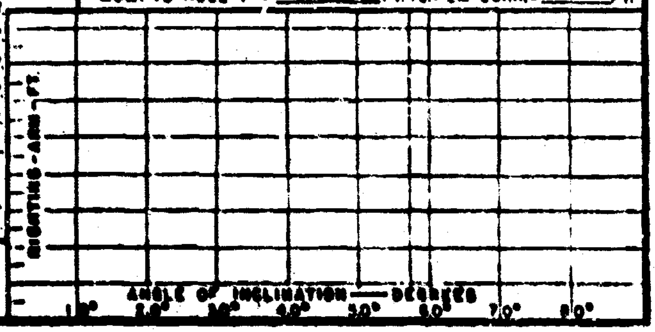
SYMBOL	COMPARTMENT	CU.FT. TON	WEIGHT TONS	V.C.G. ABV. B.L. FT.	MOMENT ABV. B.L. FT. TONS	L.C.G. ABT. F.P. FT.	MOMENT ABT. F.P. FT. TONS	VERT. MOM. OF F.S. FT. TONS
	CREW & STORES		30	22	660	290	8700.	
	DIESEL OIL		50	20	1000	310	15500.	
			(80)		(1660)		(24200)	
	DT #5 S.W.B.	34%	563	8.2	4617	217.0	122171	7300
	DT #3 S.W.B.	95%	2172	15.3	33232	138.5	300822	5840
			(2735)		(37849)		(422993)	
	DEADWEIGHT							
	LIGHT SHIP		2760	23.40	64584	176.1	486036	
	DISPLACEMENT		5575	18.67	104093	167.4	933229	13140

TRIM

DRAFT AT L.C.F. = 14.00 FT.
KM_L = ___ FT.; GM_L = ___ FT.
MLD. DISPLACEMENT = ___ CU.FT.
MOMENT TO ALTER TRIM 1" = ___ FT.TS.
L.C.B. AFT OF F.P. = 167.4 FT.
L.C.G. AFT OF F.P. = 167.4 FT.
L.C.F. AFT OF F.P. = ___ FT.
TRIMMING LEVER = ___ FT.
TRIM EVEN KEEL = ___ FT.
DRAFT AT F.P. = ___ A.P. = ___
DRAFTS AT DRAFT MARKS:
FWD. = ___ AFT = ___
MEAN = ___

STABILITY

METACENTRE ABOVE B.L. KM = 32.60 FT.
CENTRE OF GRAVITY ABV. B.L. KG = 18.67 FT.
METACENTRIC HEIGHT GM = 13.93 FT.
ALLOWANCE FOR FREE SURFACE = 2.36 FT.
MOM. TO HEEL 1° = ___ FT.TS.; GM CORR. = 11.57 FT.



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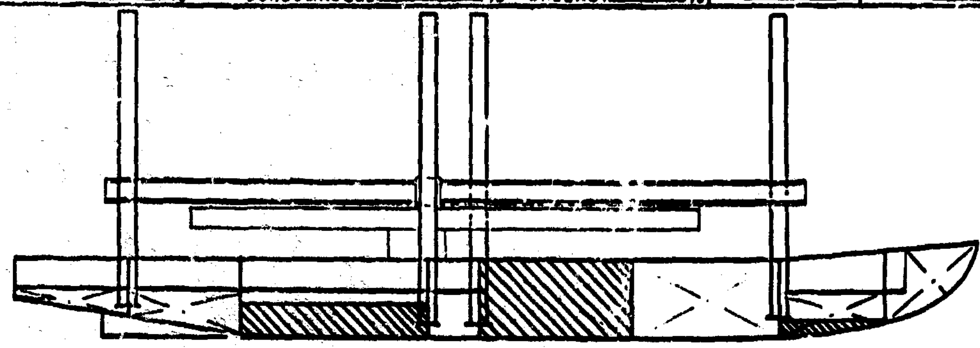
STABILITY IN TOWING CONDITION - 14 FOOT DRAFT
FIGURE 1-5A

SYMBOLS: S.W. · F.W. · F.O. OR D.O. · OIL CARGO · BULK CARGO · DRY CARGO · REFRIC. CARGO

TRIM & STABILITY SUMMARY

CONDITION OF VESSEL: TOWING CONDITION 16' DRAFT NO. ____ DATE 07 15 71 PAGE 2

CARGO ____ % CONSUMABLES ____ % BALLAST ____ % BY G.L. JOB NO. 5886



330'x 60'x 27' CONTAINER CRANE BARGE

REF. LINE FOR V.C.G. ____ REF. LINE FOR L.C.G. ____

SYMBOL	COMPARTMENT	CU.FT. TON	WEIGHT TONS	V.C.G. ABV. B.L. FT.	MOMENT ABV. B.L. FT. TONS	L.C.G. ABT. F.P. FT.	MOMENT ABT. F.P. FT. TONS	VERT. MOM. OF F.S. FT. TONS
	CREW & STORES		30	22	660	290	8700	
	DIESEL OIL		50	20	1000	310	15500	
			(80)		(1660)		(24200)	
	DT #5 S.W.B.	76%	1262	8.2	10348	217.0	273854	7300
	DT #3 S.W.B.	100%	2283	15.4	35158	138.5	316196	
	DT #1 S.W.B	38%	190	11.7	2223	43.5	8265	4100
			(3735)		(47729)		(598315)	
	DEADWEIGHT							
	LIGHT SHIP		2760	23.40	64584.	176.1	486036	
	DISPLACEMENT		6575	17.33	113973.	168.6	1108551	11400

TRIM

DRAFT AT L.C.F. = 16.00 FT.

KM_L = ____ FT.; GM_L = ____ FT.

MLD. DISPLACEMENT = ____ CU. FT.

MOMENT TO ALTER TRIM 1" = ____ FT. TS.

L.C.B. AFT OF F.P. = 168.6 FT.

L.C.G. AFT OF F.P. = 168.6 FT.

L.C.F. AFT OF F.P. = ____ FT.

TRIMMING LEVER = ____ FT.

TRIM EVEN KEEL = ____ FT.

DRAFT AT: F.P. = ____ A.P. = ____

DRAFTS AT DRAFT MARKS:

FWD.: ____ AFT: ____

MEAN: ____

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STABILITY

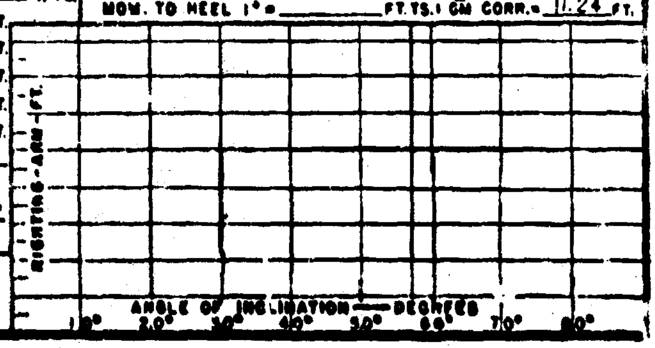
METACENTRE ABOVE B.L. $\overline{KM}$ = 30.30 FT.

CENTRE OF GRAVITY ABV. B.L. $\overline{KG}$ = 17.33 FT.

METACENTRIC HEIGHT $\overline{GM}$ = 12.97 FT.

ALLOWANCE FOR FREE SURFACE = 1.73 FT.

MOM. TO HEEL 1° = ____ FT. TS.; $\overline{GM}$ CORR. = 11.24 FT.



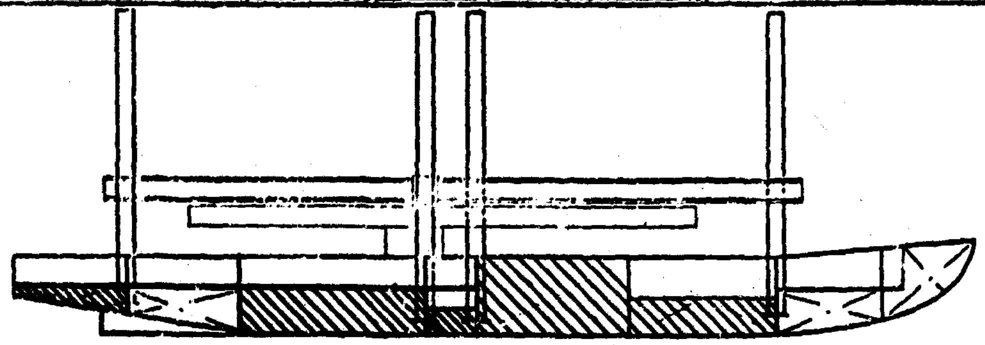
STABILITY IN TOWING CONDITION - 16 FOOT DRAFT
FIGURE 1-5B

TRIM & STABILITY SUMMARY

CONDITION OF VESSEL: TOWING CONDITION 20' DRAFT — NO. ___ DATE 07 15 71 — PAGE 3

CARGO ___ % CONSUMABLES ___ % BALLAST ___ % BY G.J. — JOB NO. 5834

SYMBOLS: S.W. — F.W. — F.O. OR D.O. — OIL CARGO — BULK CARGO — DRY CARGO — REFRIG CARGO



330'x 60'x 27' CONTAINER CRANE BARGE

REF. LINE FOR V.C.G. ___ REF. LINE FOR L.C.G. ___

SYM NO.	COMPARTMENT	CU.FT. TON	WEIGHT TONS	V.C.G. ABV. B.L. FT.	MOMENT ABV. B.L. FT. TONS	L.C.G. ABT. F.P. FT.	MOMENT ABT. F.P. FT. TONS	VERT. MOM. OF F.S. FT. TONS
	CREW & STORES		30	22.0	660	290	8700	
	DIESEL OIL		50	20.0	1000	310	15 500	
			(80)		(1660)		(24200)	
	DT #2	51%	1107	16.1	17323	85.8	94981	5840
	DT #3	100%	2283	15.4	35,158	138.5	316196	
	DT #4	60%	290	8.1	2349	174.5	50605	2094
	DT #5	100%	1670	8.2	13694	217.0	362390	
	DT #7	100%	410	15.0	6150	304.9	125829	
			(5760)		(75174)		(950001)	
	DEADWEIGHT							
	LIGHT SHIP		2760	23.40	64584	176.1	486036	
	DISPLACEMENT		8600	16.44	141418	169.8	1460237	7934

TRIM

DRAFT AT L.C.F. = 20' FT.

KM_L = ___ FT.; GM_L = ___ FT.

MLD. DISPLACEMENT = ___ CU. FT.

MOMENT TO ALTER TRIM 1" = ___ FT. TS.

L.C.B. AFT OF F.P. = 169.8 FT.

L.C.G. AFT OF F.P. = 169.8 FT.

L.C.F. AFT OF F.P. = ___ FT.

TRIMMING LEVER = ___ FT.

TRIM EVEN KEEL = ___ FT.

DRAFT AT F.P. = ___ A.P. = ___

DRAFTS AT DRAFT MARKS:

FWD. = ___ AFT = ___

MEAN = ___

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STABILITY

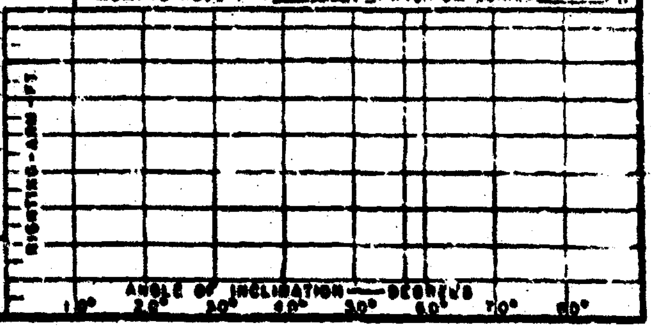
METACENTRE ABOVE B.L. KM = 27.75 FT.

CENTRE OF GRAVITY ABV. B.L. KG = 16.44 FT.

METACENTRIC HEIGHT GM = 11.31 FT.

ALLOWANCE FOR FREE SURFACE = 0.92 FT.

MOM. TO HEEL 1° = ___ FT. TS.; GM CORR. = 10.39 FT.

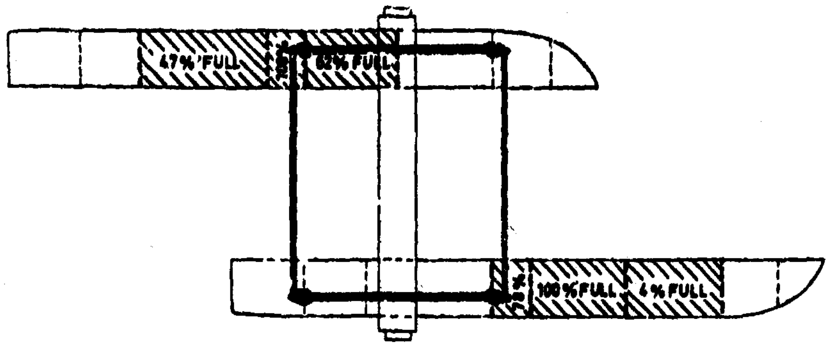


STABILITY IN TOWING CONDITION - 20 FOOT DRAFT
FIGURE 1-5C

TRIM & STABILITY SUMMARY

CONDITION OF VESSEL: OPERATING CONDITION 14' DRAFT NO. ____ DATE 08/20/71 PAGE ____

CARGO ____ % CONSUMABLES ____ % BALLAST ____ % BY G L JOB NO. FB34



REF. LINE FOR V.C.G. B.L. REF. LINE FOR L.C.G. F.P.

SYMBOL	COMPARTMENT		CU.FT. TON	WEIGHT TONS	V.C.G. ABV. B.L. FT.	MOMENT ABV. B.L. FT. TONS	L.C.G. ABT. F.P. FT.	MOMENT ABT. F.P. FT. TONS	VERT. MOM. OF F.S. FT. TONS
	CREW STORES	PORT HULL		15	22.0	330	290.0	4350	
	D.O.	PORT HULL		25	20.0	500	310.0	7750	
	LIGHT SHIP	PORT HULL		1380	42.8	59095	163.5	225663	
	DT #5 S.W.B.	PORT HULL	47%	393	8.2	3223	217.0	85281	3650
	DT #4 S.W.B.	PORT HULL	100%	240	8.1	1944	174.5	41880	1047
	DT #3 S.W.B.	PORT HULL	62%	705	15.3	10787	138.5	97643	2920
	LOAD ON CRANE	PORT HULL		30	82.0	2460	112.0	3360	
	TOTAL	PORT HULL		(2788)		(78339)	(167.1)	(465927)	
	CREW STORES	STBD. HULL		15	22.0	330	290.0	4350	
	D.O.	STBD. HULL		25	20.0	500	310.0	7750	
	LIGHT SHIP	STBD. HULL		1380	42.8	59095	188.7	260373	
	DT #4 S.W.B.	STBD. HULL	78%	140	8.1	1134	174.5	24430	1047
	DT #3 S.W.B.	STBD. HULL	100%	1151	15.3	17610	138.5	159413	2920
	DT #2 S.W.B.	STBD. HULL	4.4	47	16.1	757	85.8	4033	2920
	LOAD ON CRANE	STBD. HULL		30	82.0	2460	237.0	7110.	
				(2788)		(81886)		(467459	
DEADWEIGHT									
LIGHT SHIP									
DISPLACEMENT				5576	28.7	160225	167.4	933386	14504

TRIM

DRAFT AT L.C.F. = 14.0 FT.

KM_L = ____ FT.; GM_L = ____ FT.

MLD. DISPLACEMENT = ____ CU.FT.

MOMENT TO ALTER TRIM 1" = ____ FT.TS.

L.C.B. AFT OF F.P. = 167.4 FT.

L.C.G. AFT OF F.P. = 167.4 FT.

L.C.F. AFT OF F.P. = ____ FT.

TRIMMING LEVER = ____ FT.

TRIM EVEN KEEL = ____ FT.

DRAFT AT F.P. = ____ A.P. = ____

DRAFTS AT DRAFT MARKS:

FWD. = ____ AFT = ____

MEAN = ____

STABILITY

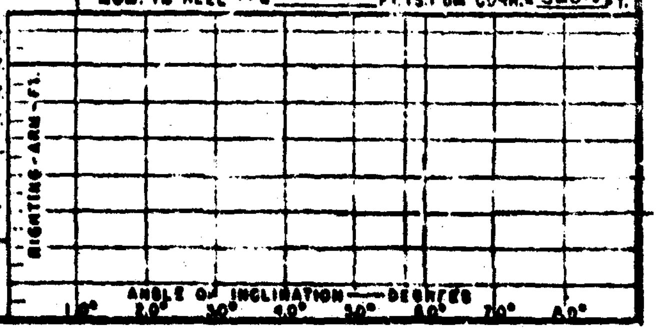
METACENTRE ABOVE B.L. KM = 357.8 FT.

CENTRE OF GRAVITY ABV. B.L. KG = 28.7 FT.

METACENTRIC HEIGHT GM = 329.1 FT.

ALLOWANCE FOR FREE SURFACE = 2.6 FT.

MOM. TO HEEL 1° = ____ FT.TS.; GM CORR. = 326.5 FT.



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SYMBOLS: S.W. · F.W. · F.O. OR D.O. · OIL CARGO · BULK CARGO · DRY CARGO · REFRIG. CARGO

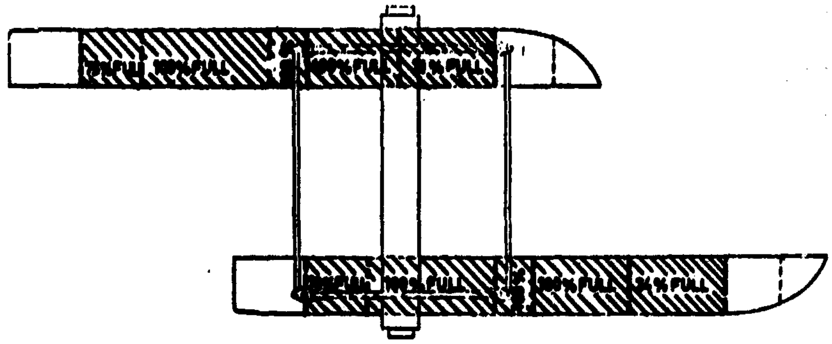
STABILITY IN OPERATING CONDITION - 14 FOOT DRAFT
FIGURE 1-6A

SYMBOLS
S.W.
F.W.
F.O. OR D.O.
OIL CARGO
BULK CARGO
DRY CARGO
REFRIG. CARGO

TRIM & STABILITY SUMMARY

CONDITION OF VESSEL: OPERATING CONDITION, 20' DRAFT NO.____ DATE 9/11/71 PAGE ____

CARGO ____ % CONSUMABLES ____ % BALLAST ____ % BY RJM JOB NO. 5434



REF. LINE FOR V.C.G. B.L. REF. LINE FOR L.C.G. F.P.

SYM. NO.	COMPARTMENT		CU.FT. TON	WEIGHT TONS	V.C.G. ABV. B.L. FT.	MOMENT ABV. B.L. FT. TONS	L.C.G. ABT. F.P. FT.	MOMENT ABT. F.P. FT. TONS	VERT. MOM. OF F.S. FT. TONS
	CREW STORES	PORT SIDE		15	22	330	290	4350	
	D.O.			25	20	500	310	7750	
	LIGHT SHIP			1380	42.8	59095	163.5	225663	
	CRANE LOAD			30	82	2460	112.0	3360	
	DT #2 S.W.B.		31%	339.6	16.1	5468	85.8	29138	2921
	DT #3 S.W.B.		100%	1142	15.3	17473	138.5	158167	2921
	DT #4 S.W.B.		100%	240	8.1	1944	174.5	41880	1047
	DT #5 S.W.B.		100%	835	8.2	6847	217.0	181195	3652
	DT #6 S.W.B.		75%	293.4	14.0	4108	268.6	78807	2191
	TOTAL			(4300)		(98225)	(169.8)	(730310)	
	STORES & DO & L.S.	STBD SIDE		1420	42.2	59925	167.4	237763	
	CRANE LOAD			30	82	2460	237.	7110	
	DT #2 S.W.B		34%	362	16.1	5828	85.8	31060	2921
	DT #3 S.W.B		100%	1142	15.3	17473	138.5	158167	2921
	DT #4 S.W.B		100%	240	8.1	1944	174.5	41880	1047
	DT #5 S.W.B		100%	835	8.2	6847	217.0	181195	3652
	DT #6 S.W.B		70%	271	14.0	3794	268.6	72791	2191
	TOTAL			(4300)		(98271)	169.8	(729966)	
	DISPLACEMENT			8600	22.8	196496	169.8	1460276	25464

TRIM

DRAFT AT L.C.F. = 20.0 FT.
KM_L = ____ FT.; GM_L = ____ FT.
MLD. DISPLACEMENT = ____ CU. FT.
MOMENT TO ALTER TRIM 1" = ____ FT. TS.
L.C.B. AFT OF F.P. = 169.8 FT.
L.C.G. AFT. OF F.P. = 169.8 FT.
L.C.F. AFT OF F.P. = ____ FT.
TRIMMING LEVER = ____ FT.
TRIM EVEN KEEL = ____ FT.
DRAFT AT: F.P. = ____ A.P. = ____
DRAFTS AT DRAFT MARKS:
FWD. = ____' ____" AFT = ____' ____"
MEAN = ____' ____"

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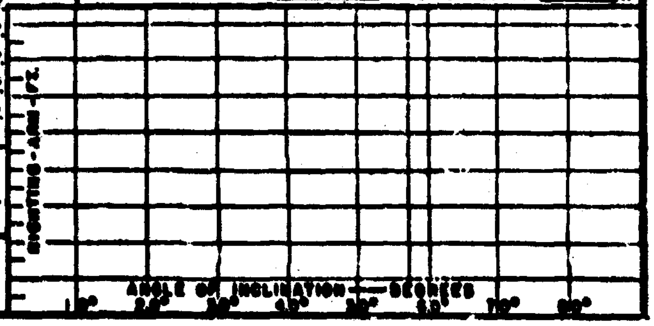
STABILITY

METACENTRE ABOVE B.L. KM = 232.8 FT.
CENTRE OF GRAVITY ABV. B.L. KG = 22.8 FT.
METACENTRIC HEIGHT GM = 210.0 FT.
ALLOWANCE FOR FREE SURFACE = 3.0 FT.
MOM. TO HEEL 1° = ____ FT. TS.; GM CORR. = 207.0 FT.



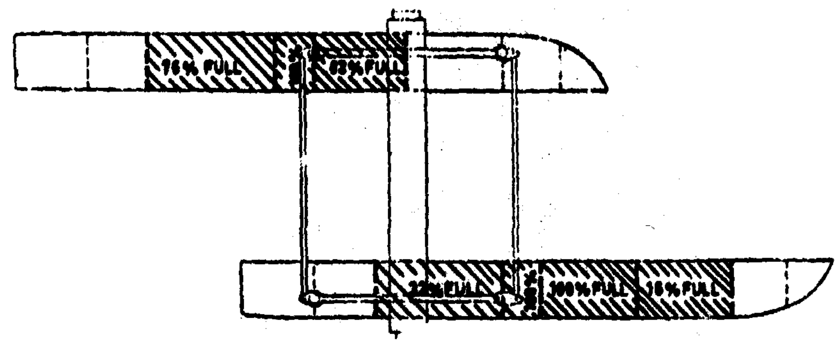
STABILITY IN OPERATING CONDITION - 16 FOOT DRAFT
FIGURE 1-6B

TRIM & STABILITY SUMMARY

SYMBOLS: S.W. · F.W. · F.O. OR D.O. · OIL CARGO · BULK CARGO · DRY CARGO · REFRIG. CARGO

CONDITION OF VESSEL: OPERATING CONDITION 16' DRAFT NO. ____ | DATE 6/11/71 | PAGE ____

CARGO ____ % CONSUMABLES ____ % BALLAST ____ % | BY RJM | JOB NO. 5834



REF. LINE FOR V.C.G. B.L. REF. LINE FOR L.C.G. F.P.

SYM. NO.	COMPARTMENT		CU.FT. / TON	WEIGHT TONS	V.C.G. ABV. B.L. FT.	MOMENT ABV. B.L. FT. TONS	L.C.G. ABT. F.P. FT.	MOMENT ABT. F.P. FT. TONS	VERT. MOM. OF F.S. FT. TONS
	CREW STORES	PORT HULL		15	22.0	330	290.0	4350	
	D. O.	PORT HULL		25	20.0	500	310.0	7750	
	LIGHT SHIP	PORT HULL		1380	42.8	59095	163.5	225[illegible]	
	DT # 5 S.W.B.	PORT HULL	76.4%	638.1	8.2	5232	217.0	138468	3650
	DT # 4 S.W.B.	PORT HULL	100.%	240	8.1	1944	174.5	41880	1047
	DT # 3 S.W.B.	PORT HULL	83.3%	958.9	15.3	14671	138.5	132807	.2920
	LOAD ON CRANE	PORT HULL		30	82.0	2460	112.0	3360	
	TOTAL	PORT HULL		(3287)		(84232)		(554278)	
	LOAD ON CRANE	STBD HULL		30	82.0	2460	237.0	7110	
	CREW STORES	STBD HULL		15	22.0	330	290.0	4350	
	D. O.	STBD HULL		25	20.0	500	310.0	7750	
	LIGHT SHIP	STBD HULL		1380	42.8	59095	188.7	260373	
	DT # 5 S.W.B.	STBD HULL	32.1	267.8	8.2	2196	217.0	58113.	3650
	DT # 4 S.W.B.	STBD HULL	100%	240	8.1	1944	174.5	41880	1047
	DT #3 S.W.B.	STBD HULL	100%	1151	15.3	17610	138.5	159413	2920
	DT #2 S.W.B.	STBD HULL	16.5%	178.2	16.1	2869	85.8	15290.	2920
	TOTAL	STBD HULL		(3287)		(87004)		(554279)	
DEADWEIGHT									
LIGHT SHIP									
DISPLACEMENT				6574	26.0	171236	168.6	1108557	18154

TRIM

DRAFT AT L.C.F. = 16.0 FT.

KM_L = ____ FT.; GM_L = ____ FT.

MLD. DISPLACEMENT = ____ CU. FT.

MOMENT TO ALTER TRIM 1" = ____ FT. TS.

L.C.B. AFT OF F.P. = 168.6 FT.

L.C.G. AFT OF F.P. = 168.6 FT.

L.C.F. AFT OF F.P. = ____ FT.

TRIMMING LEVER = ____ FT.

TRIM EVEN KEEL = ____ FT.

DRAFT AT F.P. = ____ A.P. = ____

DRAFTS AT DRAFT MARKS:

FWD. = ____' ____" AFT = ____' ____"

MEAN = ____' ____"

STABILITY

METACENTRE ABOVE B.L. $\overline{KM}$ = 292.1 FT.

CENTRE OF GRAVITY ABV. B.L. $\overline{KG}$ = 26.0 FT.

METACENTRIC HEIGHT $\overline{GM}$ = 266.1 FT.

ALLOWANCE FOR FREE SURFACE = 2.8 FT.

MOM. TO HEEL 1° = ____ FT. TS.; $\overline{GM}$ CORR. = 263.3 FT.

RIGHTING - ARM - FT.

ANGLE OF INCLINATION — DEGREES: 10° 20° 30° 40° 50° 60° 70° 80°

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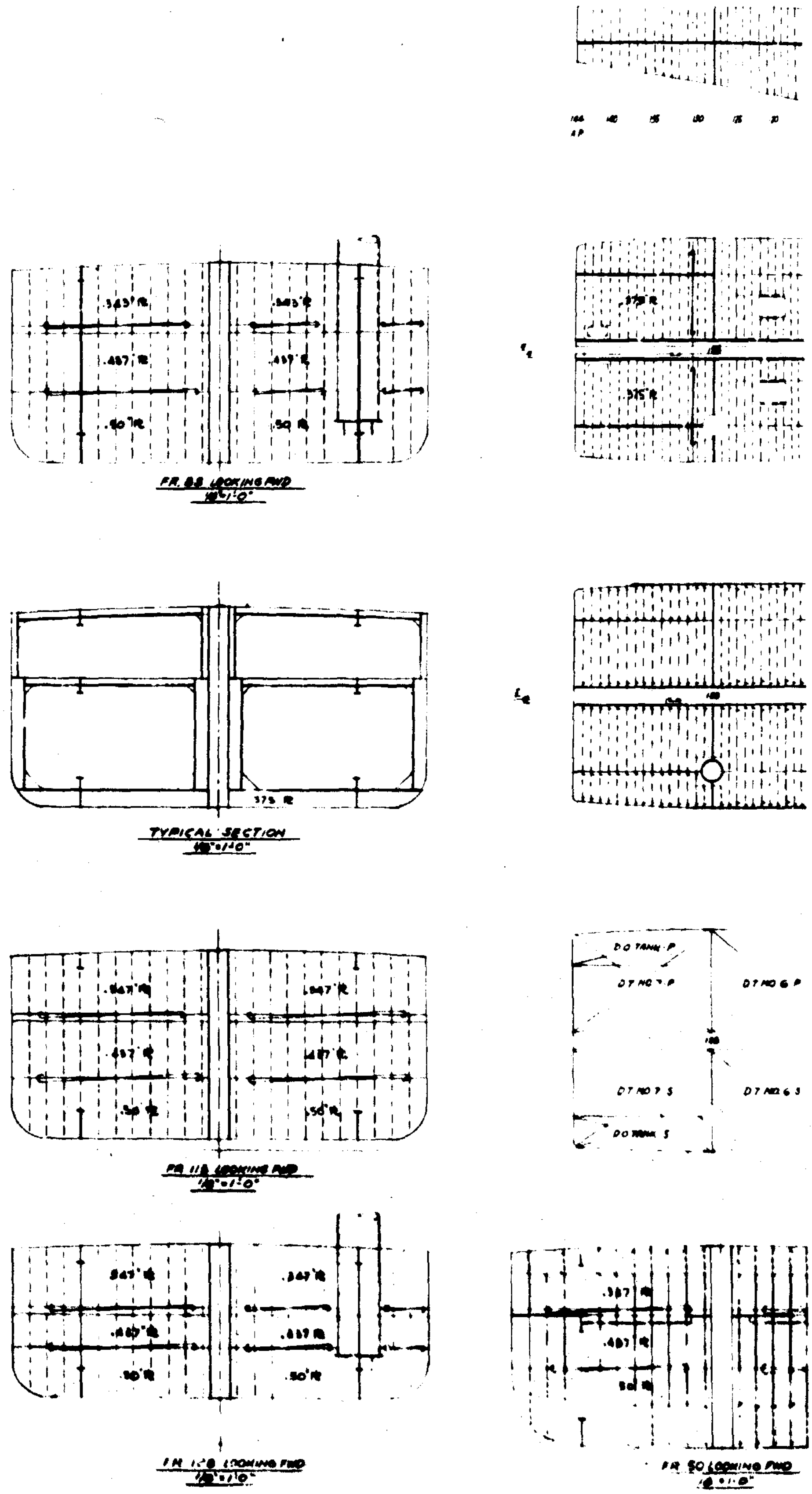
STABILITY IN OPERATING CONDITION - 20 FOOT DRAFT
FIGURE 1-6C



FR. 88 LOOKING FWD
1/8"=1'-0"
TYPICAL SECTION
1/8"=1'-0"
FR. 115 LOOKING FWD
1/8"=1'-0"
FR. 128 LOOKING FWD
1/8"=1'-0"
FR. 50 LOOKING FWD
1/8"=1'-0"
D.O. TANK P
D.T. NO. 7 P
D.T. NO. 6 P
D.T. NO. 7 S
D.T. NO. 6 S
D.O. TANK S

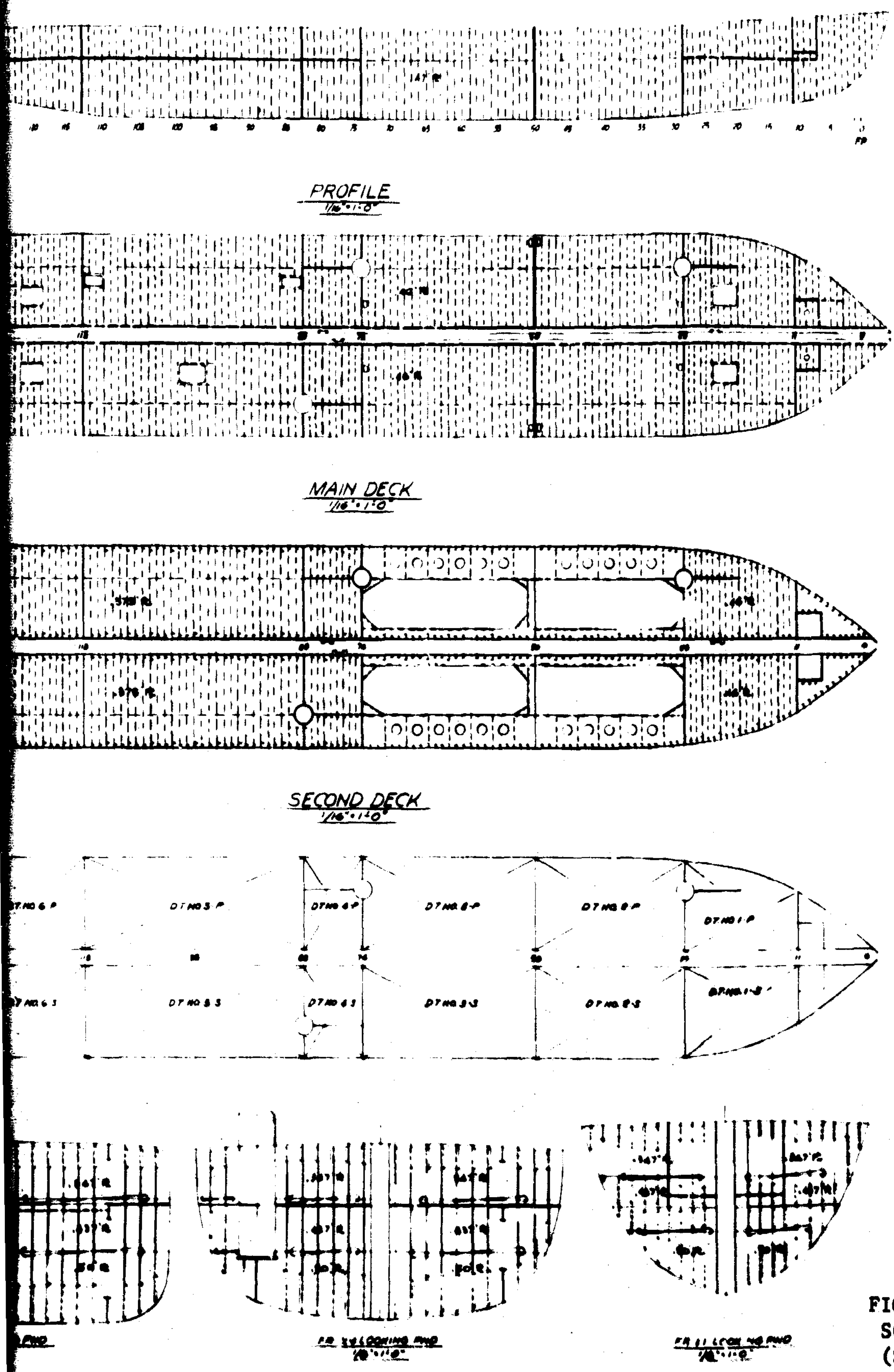
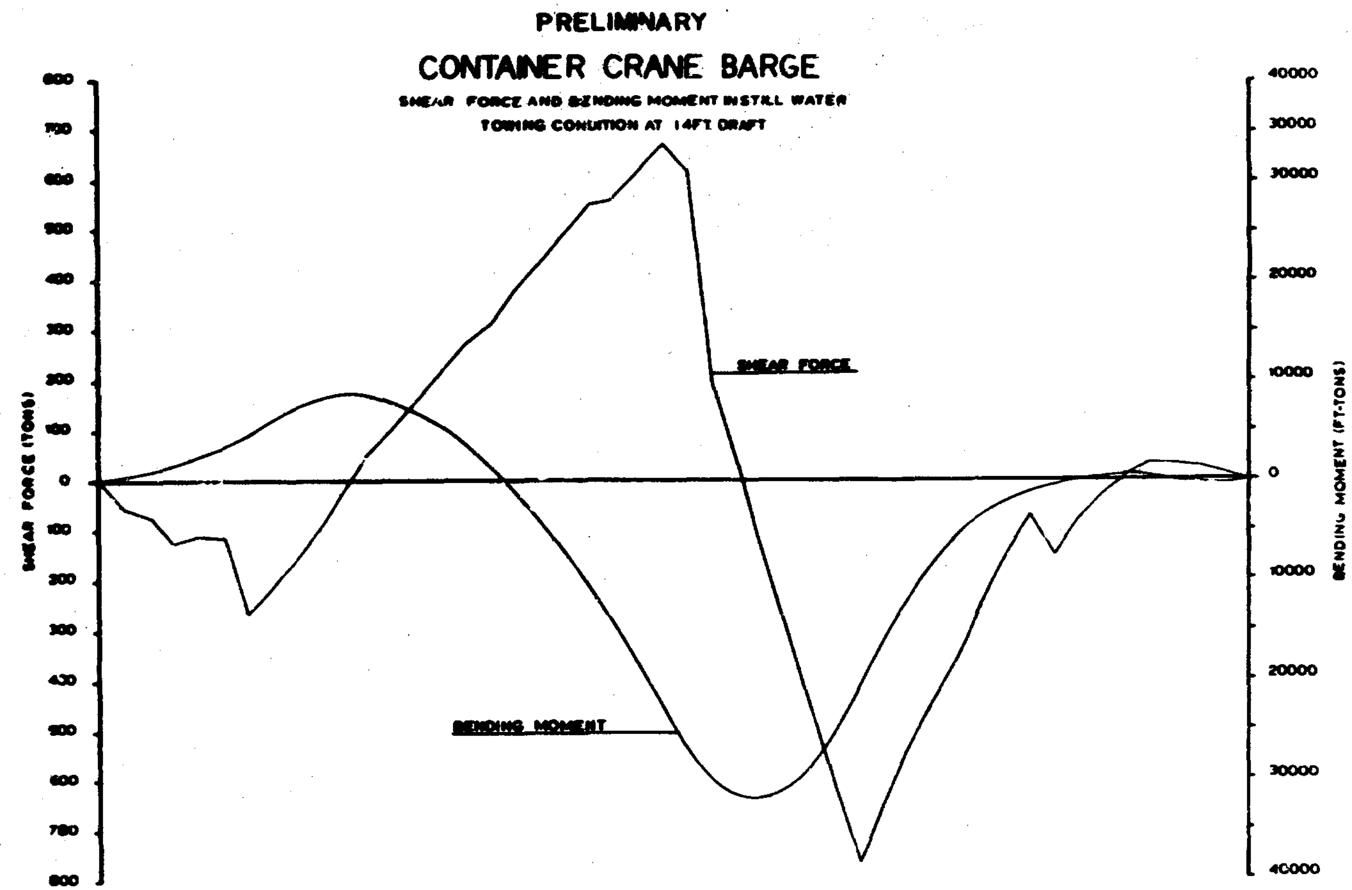
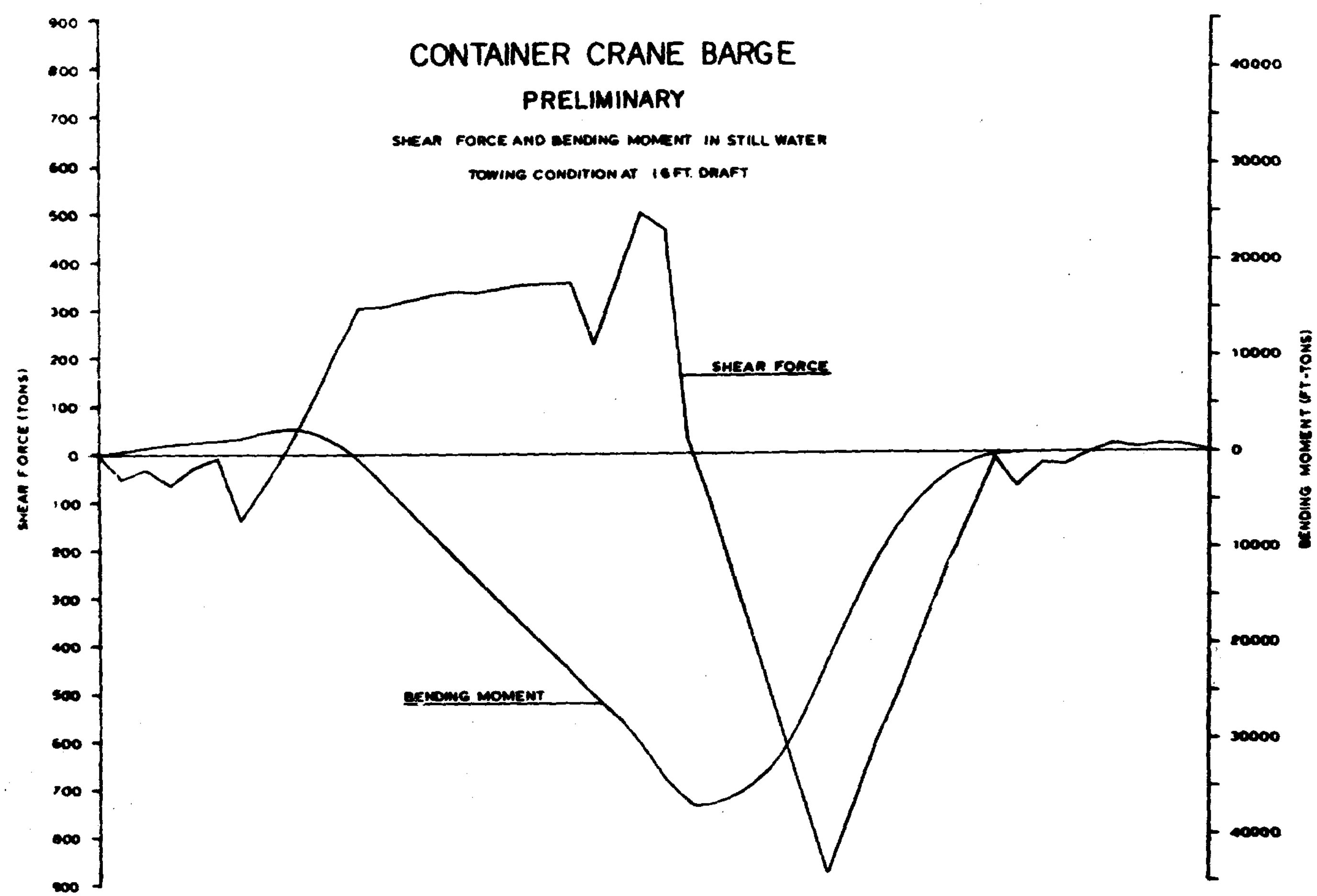


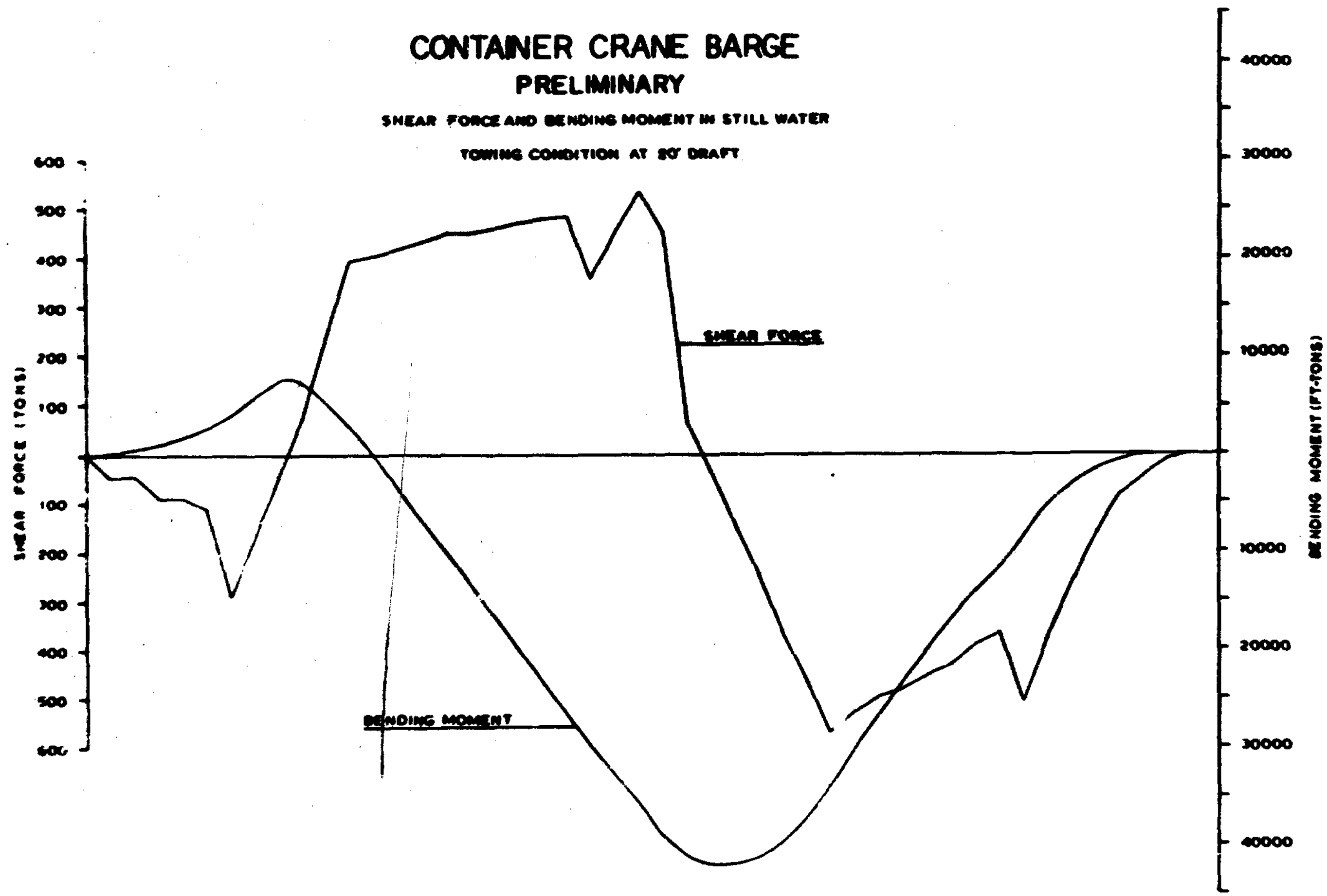
FIGURE 1-7. CONTAINER CRANE BARGE SCANTLING PLAN PROFILES AND DECKS (SCALE AS NOTED; DWG. NO. 5834-2)



STILLWATER BENDING MOMENT - 14 FOOT DRAFT
FIGURE 1-8A



STILLWATER BENDING MOMENT - 16 FOOT DRAFT
FIGURE 1-8B



STILLWATER BENDING MOMENT - 20 FOOT DRAFT
FIGURE 1-8C

LONGITUDINAL SECTION MODULUS CALCULATION

SHIP CRANE BARGE TYPE ______ JOB NO. 5884 DATE 07/06/71 BY RJM

L = 330'-0" B = 60'-0" D = 27'-0" H DESIGN MLD = ____ H SCANTLING MLD = ____

CO ____ ASSUMED NEUTRAL AXIS B.L. ABOVE B.L. (% D = ___) ______

TYPE CALCULATION (TOP BOTTOM BARE TOTAL) SHEET NO 1 OF ____

MEMBER	SCANTLING-IN.	a - IN²	d_n - FT.	ad_n FT. IN²	ad_n^2 FT.² IN²	h - FT.	i_o FT² IN²
MN DK	360 x .46	165.6	27.5	4554	125235		
2ND DK	124 x .375	46.5	18.0	837	15066		
SIDE SHELL	264 x .47	124.1	16.0	1985	31765	22	5005
BILGE RADIUS	5'R x .551	51.9	1.82	94.	172.	—	123.
BOTTOM SHELL	300 x .551	165.3	.05	8	0		
LONG. BHD.	336 x .47	157.9	14.0	2211	30952	28	10317.
MN. DK. LONG. GIRDER	24"x12"T x .360	13.3	26.5	352.	9340	2	15.
2ND. DK. LONG. GIRDER	24"x12"T x .404	14.6	17.0	248.	4219	2	15.
BOTTOM LONG. GIRDER	48"x12"T x .404	24.2	2.0	48.	97	4	78.
TOTALS		763.4	13.54	10337	216846		15573
				+	15573		
					232419		

$\frac{ad_n}{a} = \frac{10337}{763.4} = 13.54$ FT. $\quad Ad_g^2 = 763.4(13.54)^2 = 139955$

$\frac{1}{2} I = 232419 - 139955 = 92464$ $\quad I = 184928$ FT² IN²

$y = 13.46$ TOP 13.54 BOTTOM $\quad I/y = 13739$ TOP 13658 BOTTOM

REQUIRED I/y BY ( ) ____ TOP ____ BOT. % OVER/BELOW REQ'D ____ TOP ____ BOT.

REQUIRED I/y BY ( ) ____ TOP ____ BOT. % OVER/BELOW REQ'D ____ TOP ____ BOT.

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SECTION MODULUS CALCULATION
FIGURE 1-9

Longitudinal Strength (ABS 1971 Rules, Section 6)

By Section 6.3: $SM = C\,f\,B\,(Cb + .5)$

$C = 1.00$ (type 1)

$f = 152$

$B = 60$

$C_b = .77$

$SM = 1.00\,(152)\,(60)\,(.77 + .5) = 11582.4$

From still water bending moment calc.,

Max bending moment ≈ 42000 ton ft.

From Table 6.2, Case II

$.485 \times S \times SM < M < S \times SM$

$23874 < 42000 < 49225$

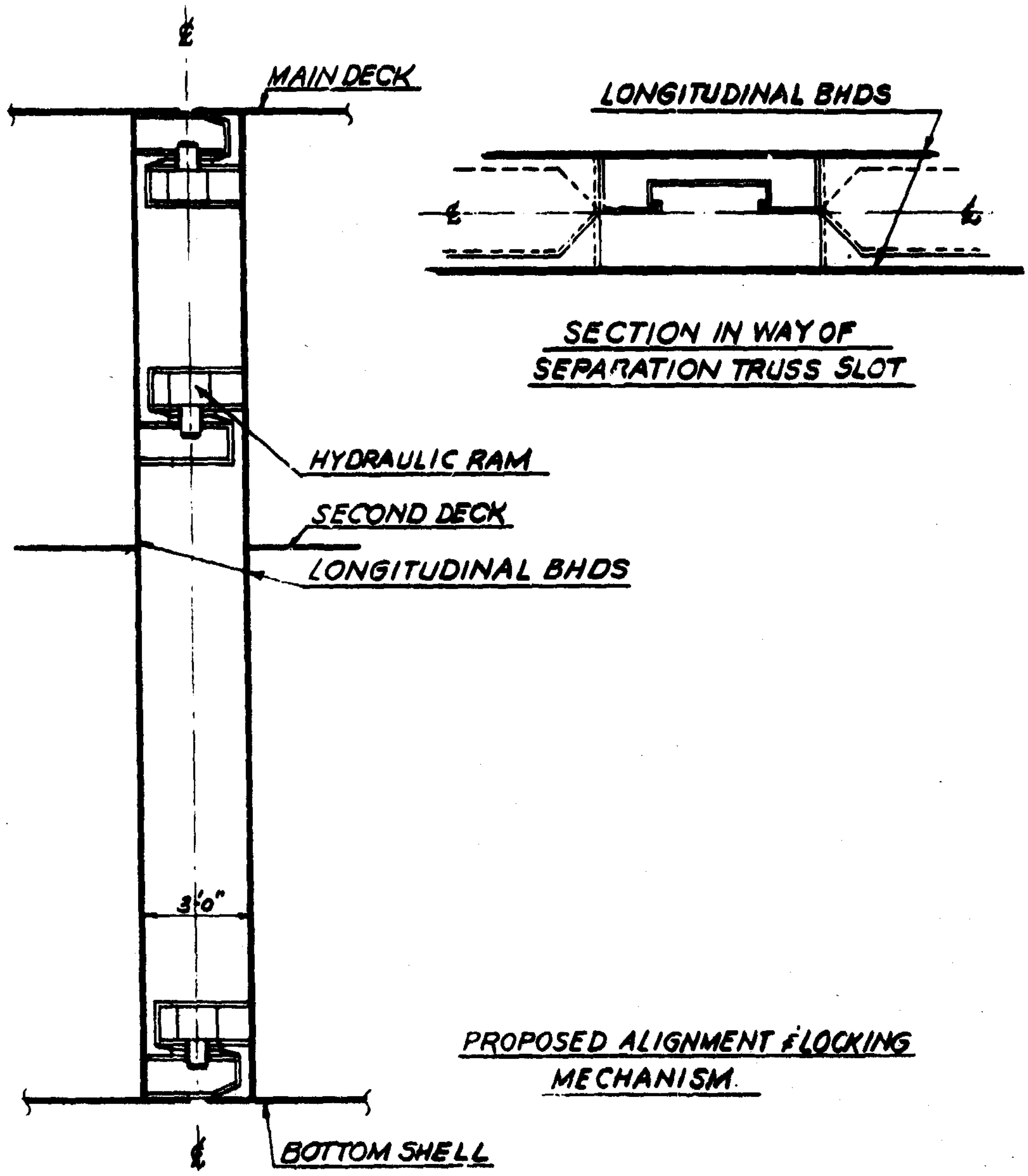
$SM_T = SM$ $\underline{SM_T \text{ (required)} = 11582}$

From Table 6.3, Case I

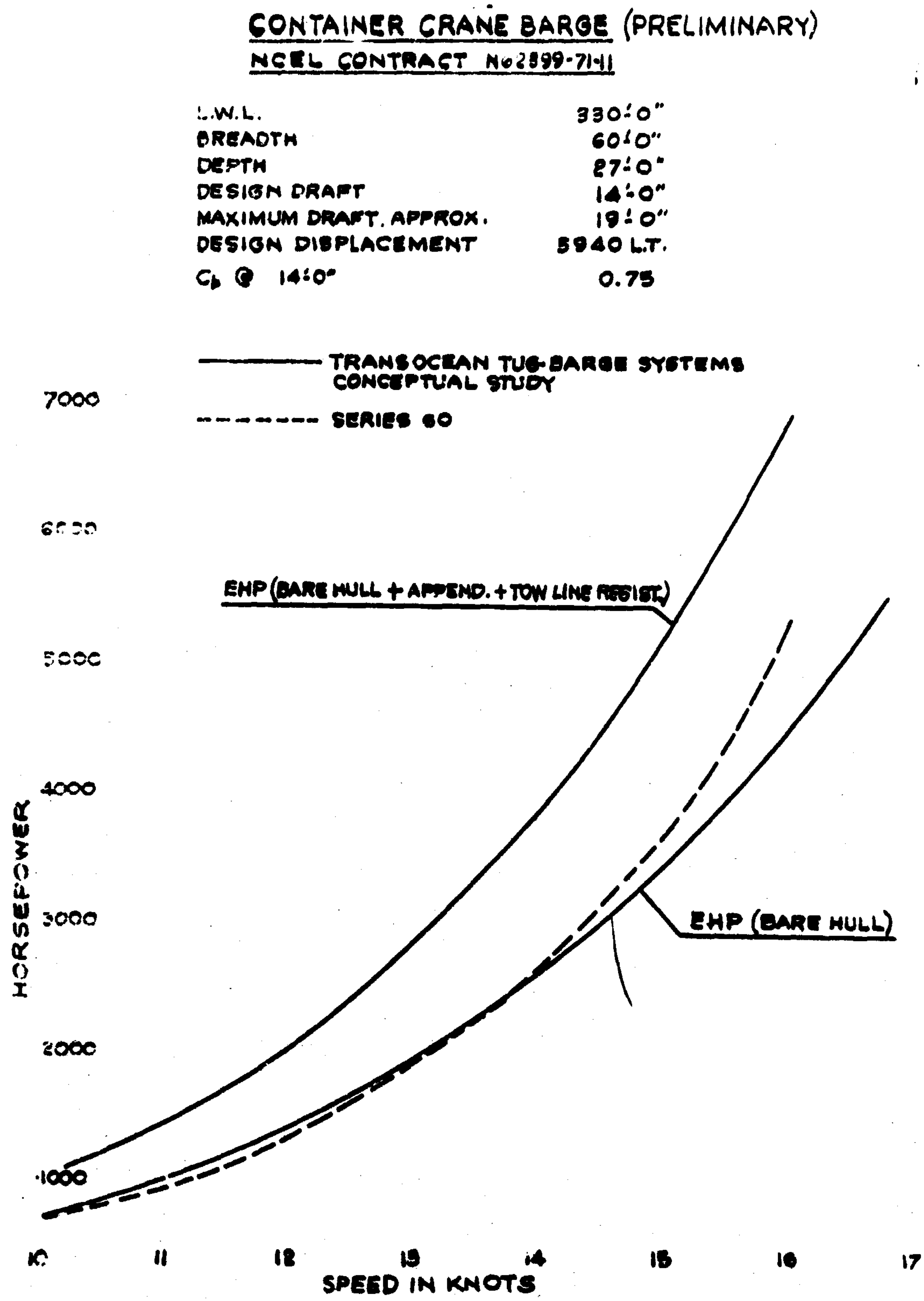
$$SM_B = \left[\frac{1000-L}{10000} + 1\right] SM_T = 1.067\ SM_T$$

$\underline{SM_B \text{ (required)} = 12358}$

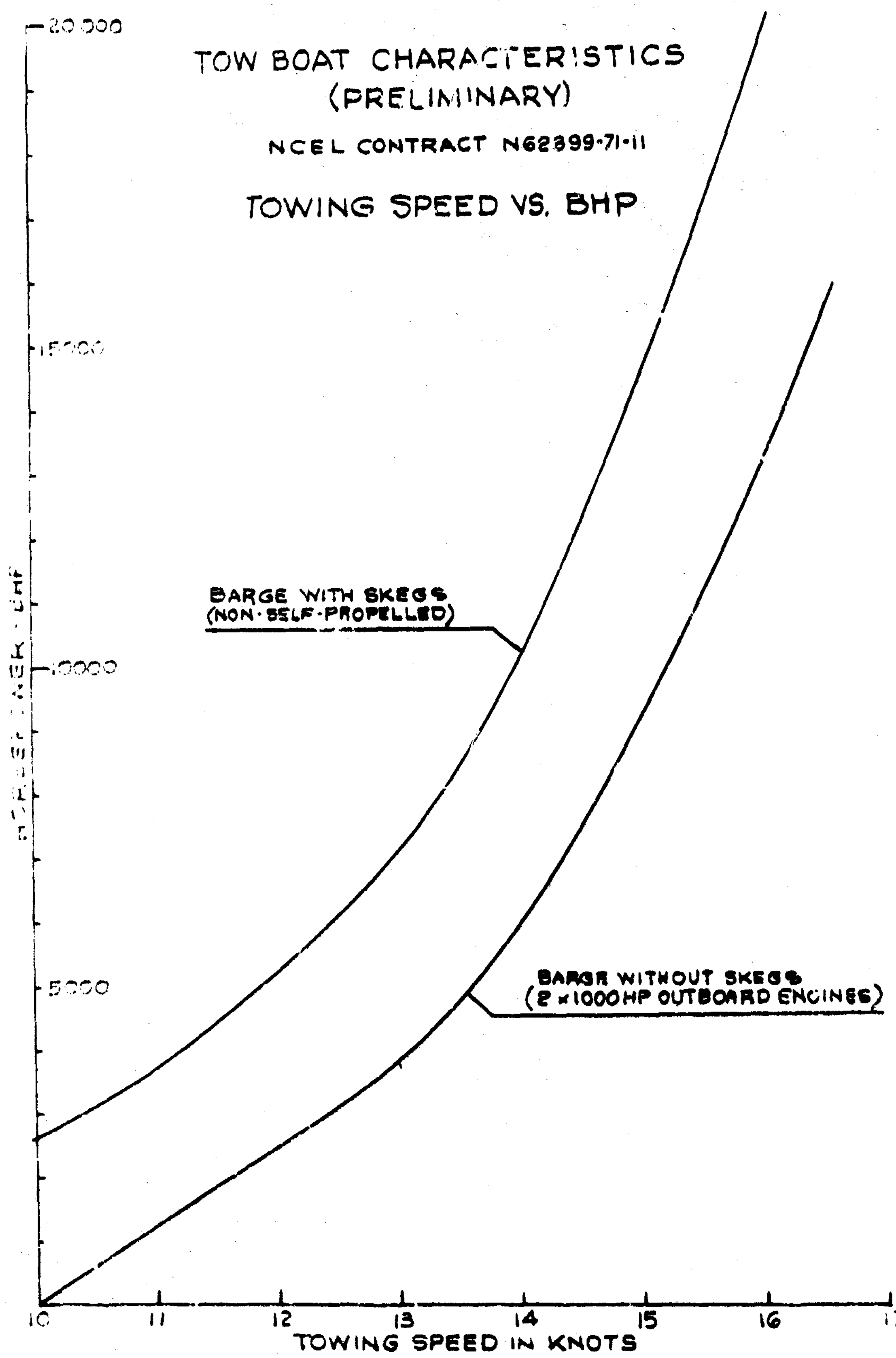
CALCULATION OF REQUIRED SECTION MODULUS
FIGURE 1-10



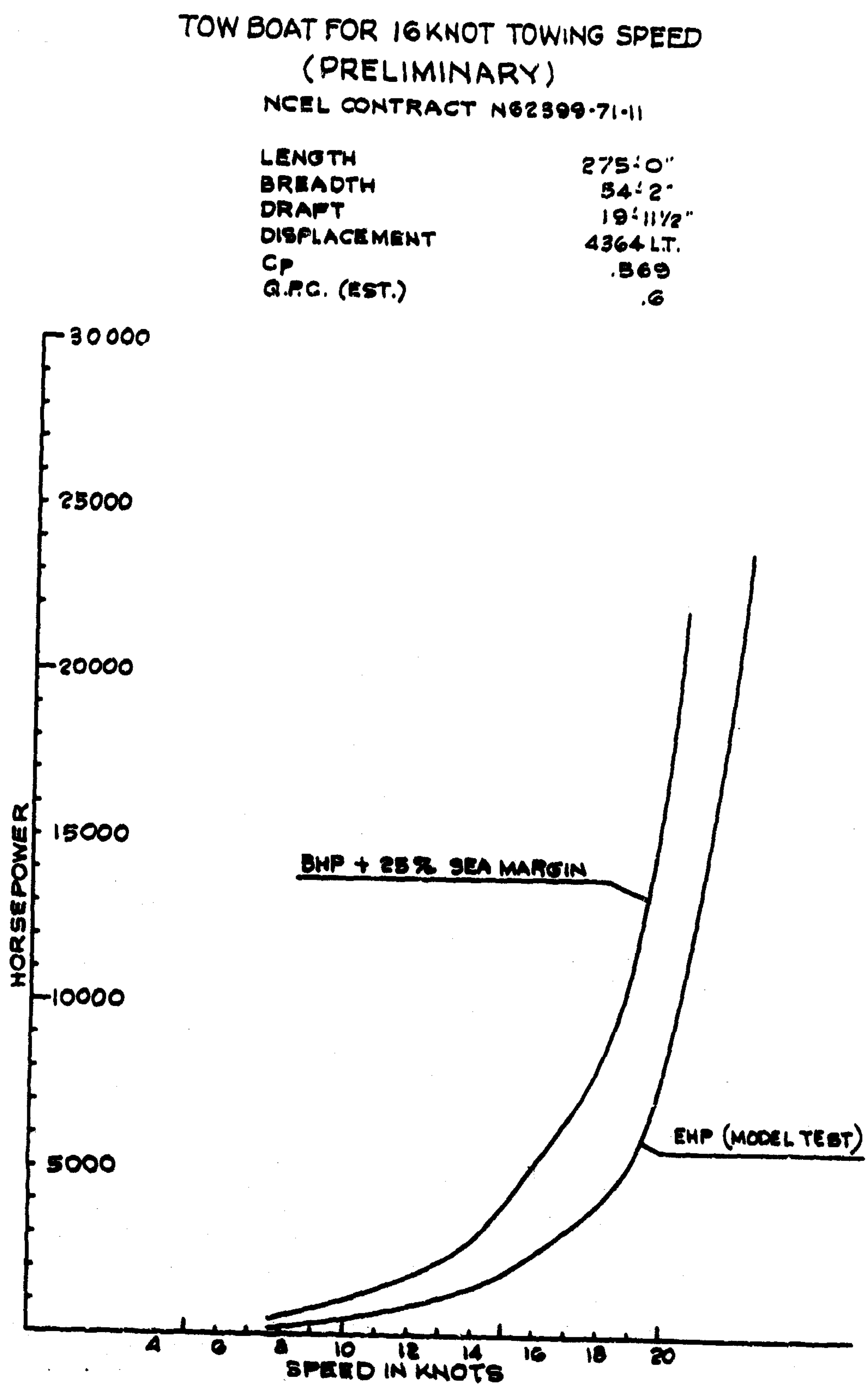
PROPOSED ALIGNMENT AND LOCKING MECHANISM
FIGURE 1-11



SPEED VS HP FOR BARGE TUG COMBINATION
FIGURE 1-12



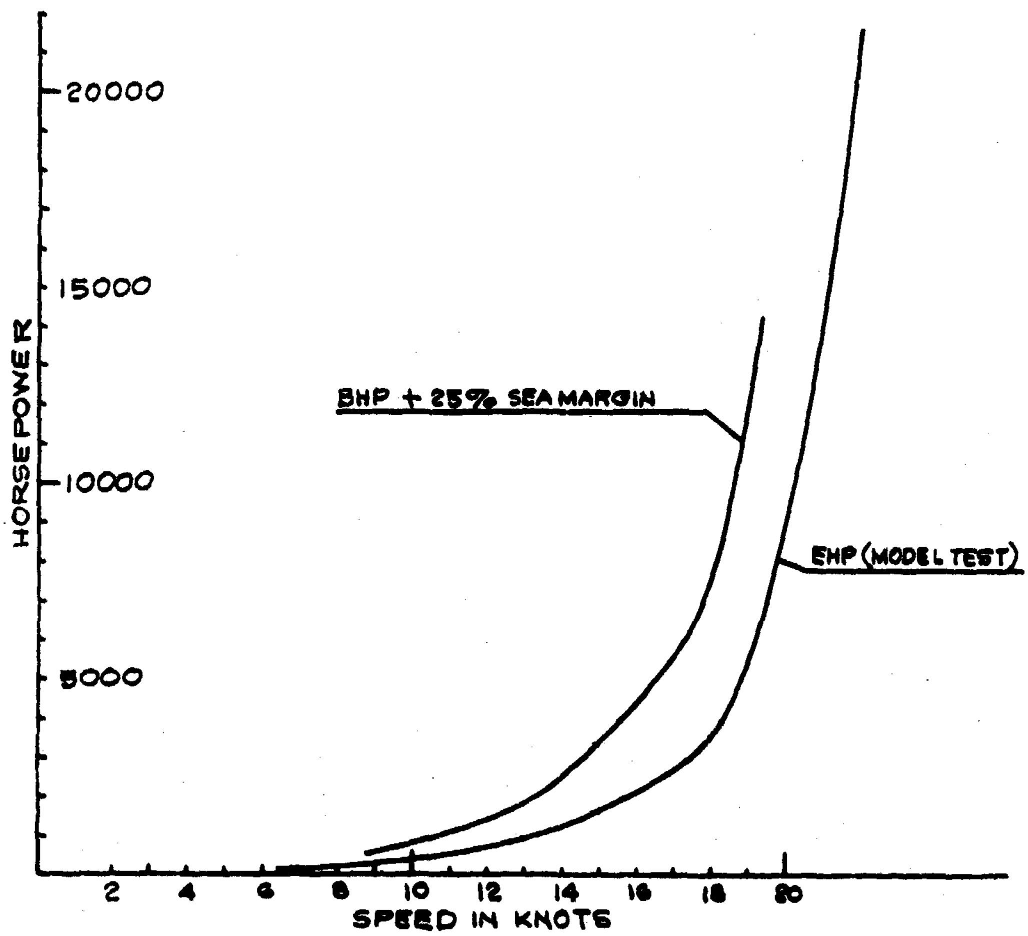
TOWING SPEED VS BHP OF TUG
FIGURE 1-13



SPEED VS HP - TUG FOR 16 KNOT TOWING
FIGURE 1-14A

TOW BOAT FOR 15 KNOT TOWING SPEED
(PRELIMINARY)
NCEL CONTRACT N62399-71-11

LENGTH	245'0"
BREADTH	48'3"
DRAFT	17'9"
DISPLACEMENT	3086 L.T.
Cp	.569
Q.P.C. (EST.)	.6

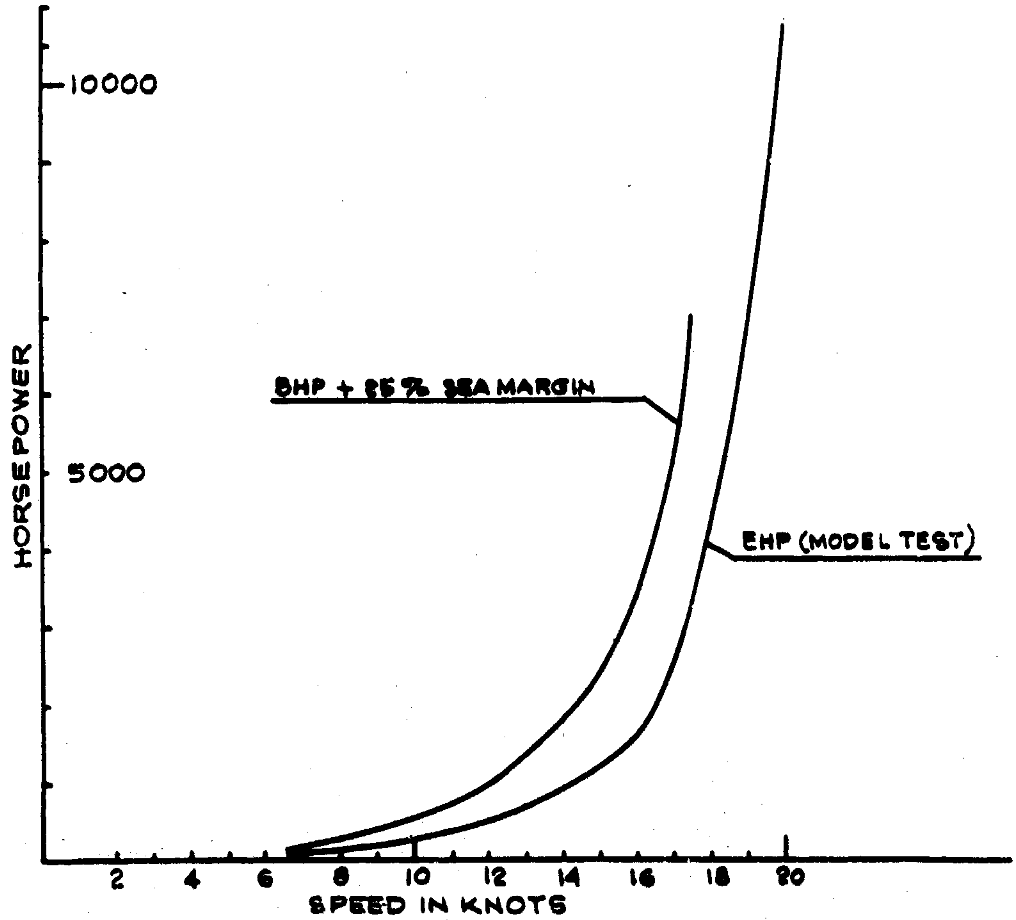


SPEED VS HP - TUG FOR 15 KNOT TOWING
FIGURE 1-14B

TOW BOAT FOR 14 KNOT TOWING SPEED
(PRELIMINARY)

NCEL CONTRACT N62399-71-11

LENGTH	200'-0"
BREADTH	39'-5"
DRAFT	14'-6"
DISPLACEMENT	1679 L.T.
Cp	.569
Q.P.C. (EST.)	.6

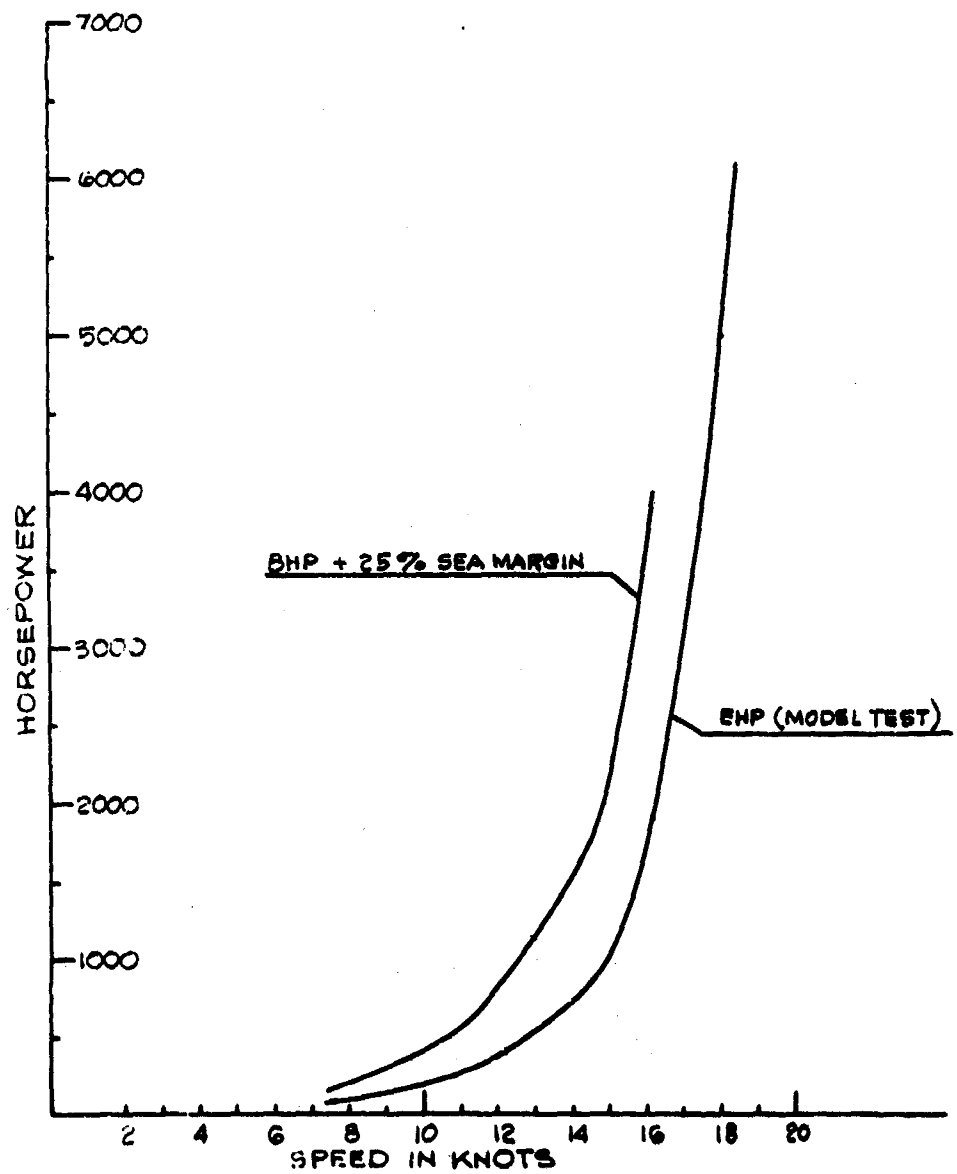


SPEED VS HP - TUG FOR 14 KNOT TOWING
FIGURE 1-14C

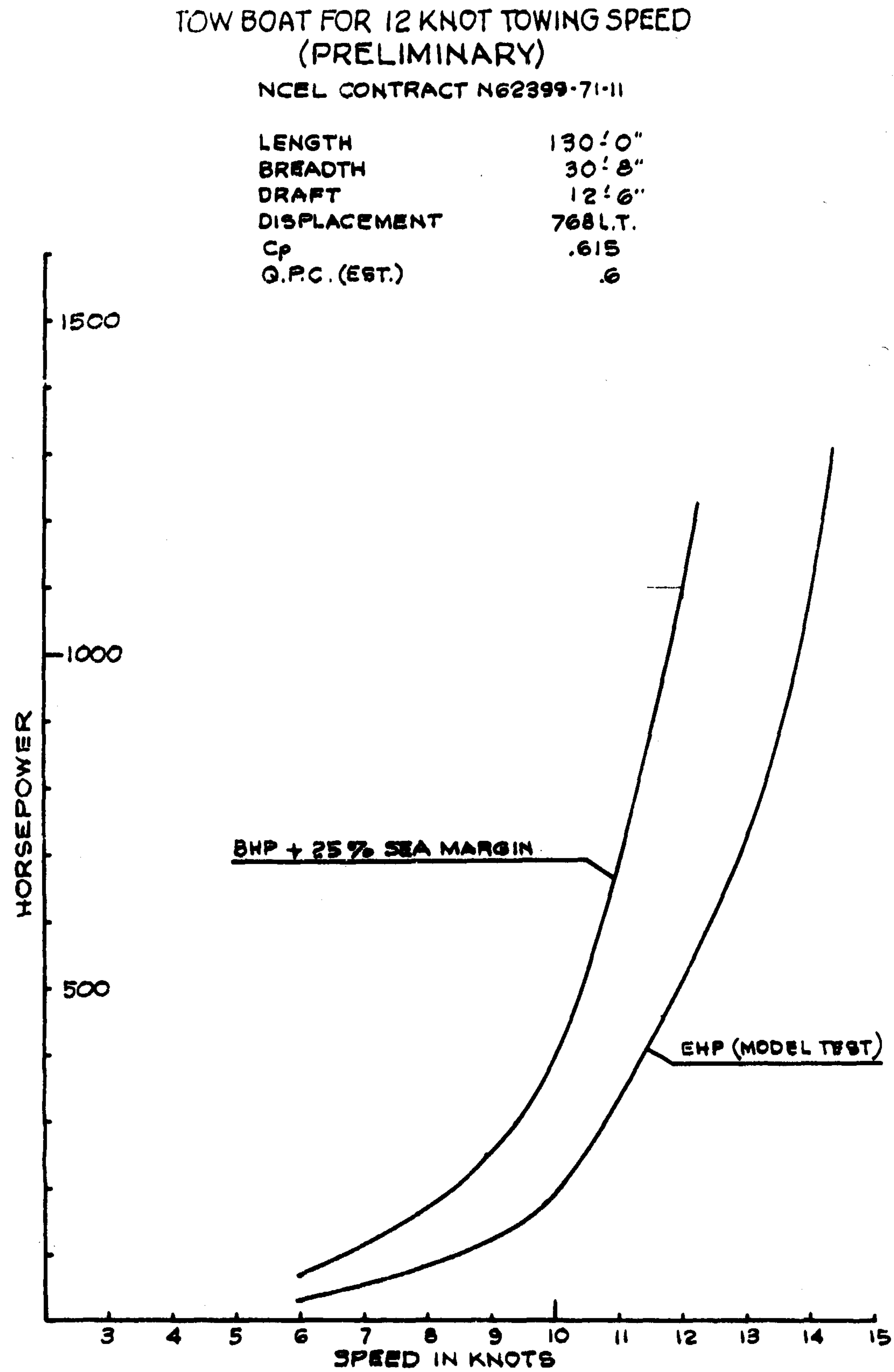
TOW BOAT FOR 13 KNOT TOWING SPEED
(PRELIMINARY)

NCEL CONTRACT N62399-71-11

LENGTH	170'-0"
BREADTH	33'-6"
DRAFT	12'-4"
DISPLACEMENT	1091 L.T.
Cp	.569
Q.P.C. (EST.)	.6



SPEED VS HP - TUG FOR 13 KNOT TOWING
FIGURE 1-14D

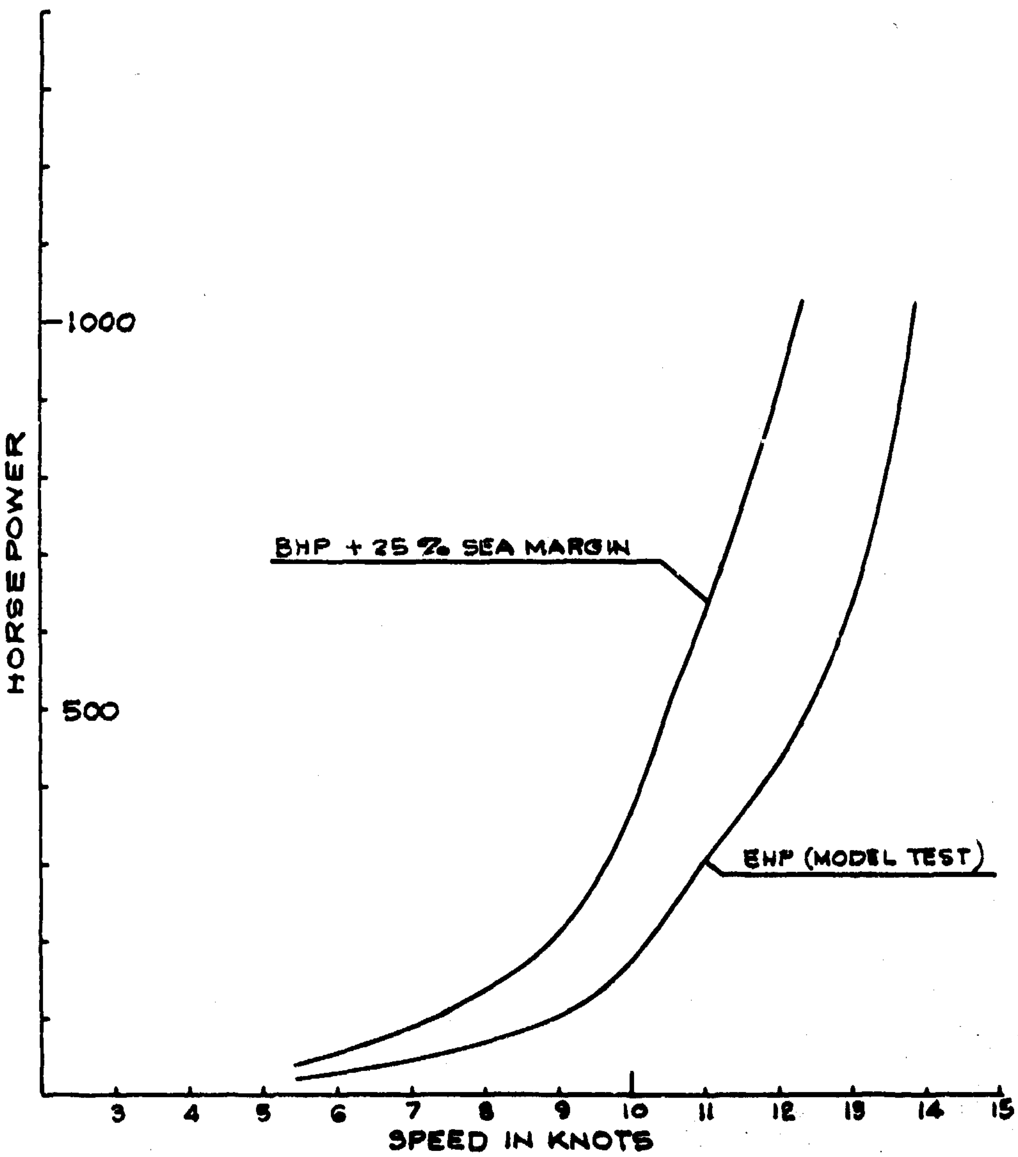


SPEED VS HP - TUG FOR 12 KNOT TOWING
FIGURE 1-14E

TOW BOAT FOR 11 KNOT TOWING SPEED (PRELIMINARY)

NCEL CONTRACT N62399-71-11

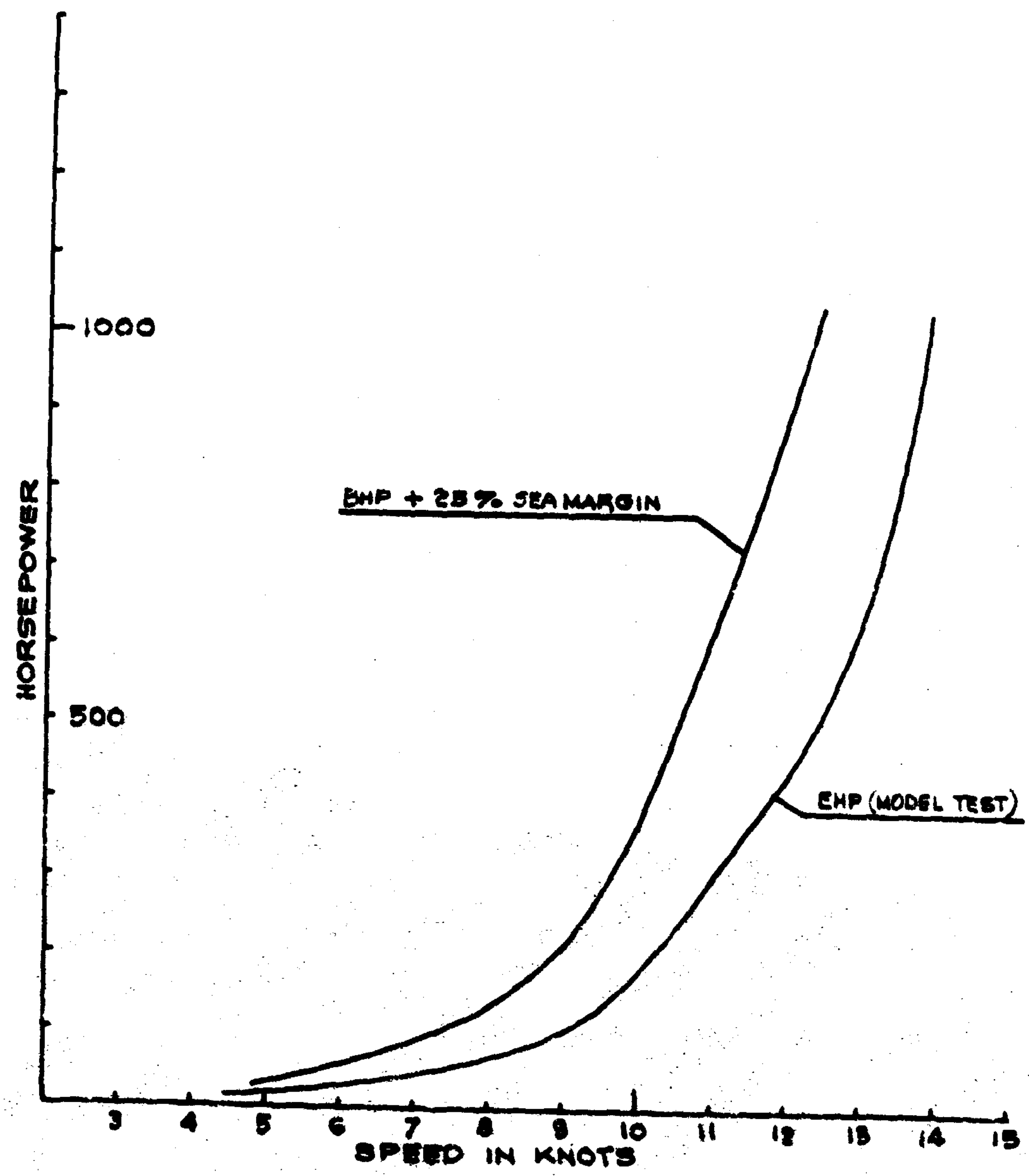
LENGTH	115'-0"
BREADTH	27'-1"
DRAFT	11'-1"
DISPLACEMENT	532 L.T.
Cp	.615
Q.P.C. (EST.)	.6



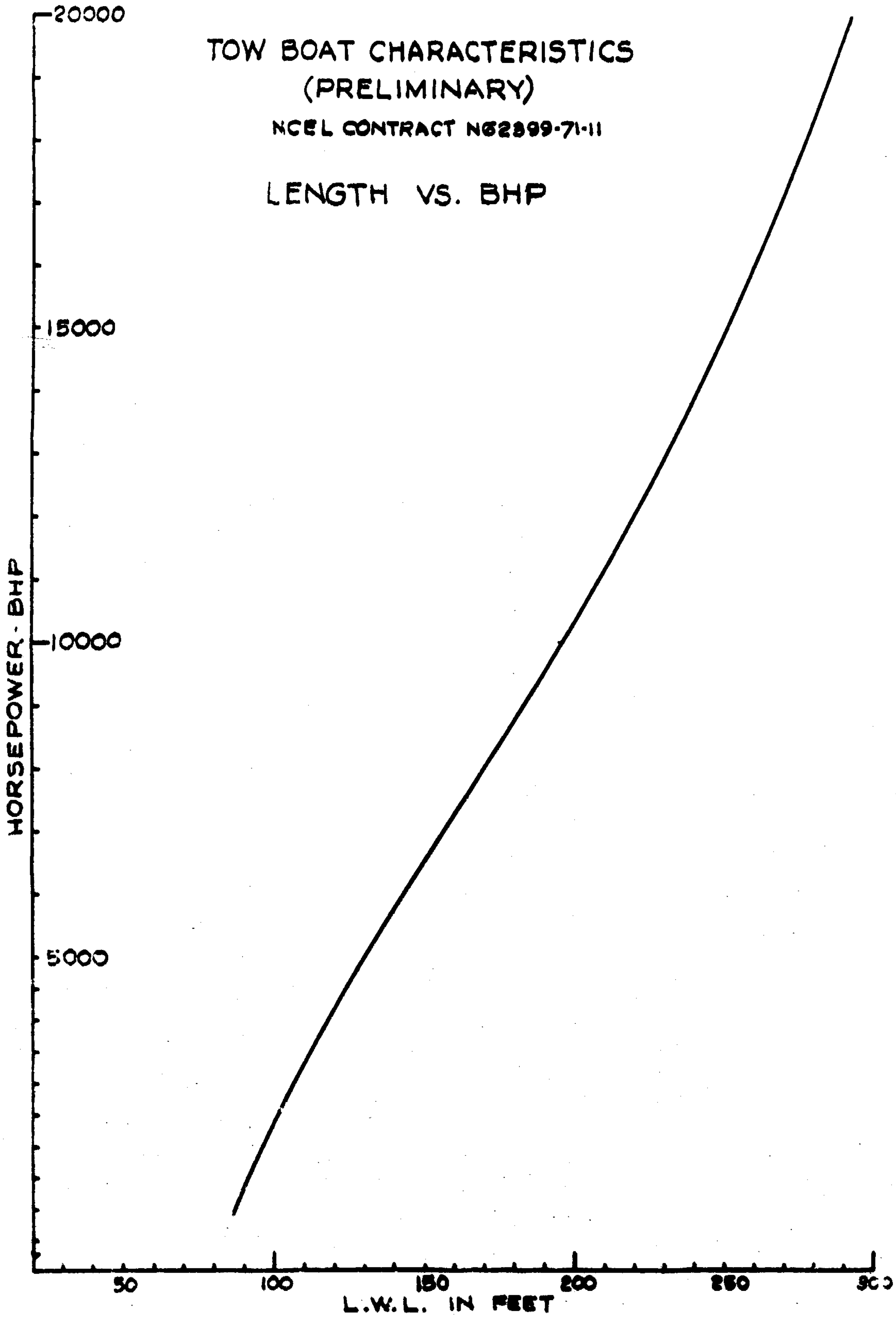
SPEED VS HP - TUG FOR 11 KNOT TOWING
FIGURE 1-14F

TOW BOAT FOR 10 KNOT TOWING SPEED
(PRELIMINARY)
NCEL CONTRACT N62399-71-11

LENGTH	115'-0"
BREADTH	27'-1"
DRAFT	11'-1"
DISPLACEMENT	532 L.T.
Cp	.615
Q.P.C. (EST.)	.6



SPEED VS HP - TUG FOR 10 KNOT TOWING
FIGURE 1-14C



HP VS LENGTH FOR TUGS CONSIDERED
FIGURE 1-14H

SECTION 5

CRANE SUPPORTING DATA AND DRAWINGS

APPENDIX	TITLE	PAGE NO.
A	BOOM STRUCTURE CALCULATIONS	
B	DERIVATION OF SEPARATION FORCES FOR VARIOUS CONDITIONS	
C	CALCULATIONS OF LOADS ON LOCK PINS	
D	ANALYSIS OF EXTENSIBLE BOOM	
E	ANALYSIS OF TRANSVERSE BEAM	
F	ANALYSIS OF FORE AND AFT LINKS	
G	ANALYSIS OF TRANSVERSE LINKS	
H	ANALYSIS OF KINGPOSTS	
J	ANALYSIS OF SEPARATION TRUSS	
K	DETERMINATION OF CRANE C.G.	
L	DETERMINATION OF CLEARANCE BETWEEN LIGHTER AND SEPARATION TRUSS	

TABLES

TABLE NO.	
I	SEPARATION FORCES, NO SHELTER, BROADSIDE WIND
II	SEPARATION FORCES, NO SHELTER, HEAD ON WIND
III	SEPARATION FORCES, WITH SHELTER, BROADSIDE WIND
IV	SIZE OF TRUSS CHORD MEMBERS

LIST OF ENESS DRAWINGS

DRAWING NO.	TITLE	PAGE NO.
71031-1	CONTAINER HANDLING PATH - DESIGN NO. 1 & 2	
71031-2	PRELIMINARY ARRANGEMENT - BARGE CRANE - DESIGN NO. 2	
71031-3 (2 sheets)	BOOM CRANE ARRANGEMENT - DESIGN NO. 1	
71031-4	TROLLEY ARRANGEMENT - DESIGN NO. 1	
71031-5	BRIDGE SUPPORT DRIVE & ARRANGEMENT - DESIGN NO. 1 & 2	
71031-6 (2 sheets)	KINGPOST FITTINGS & LINK TIES - DESIGN NO. 1 & 2	
71031-7	SEPARATION MACHINERY - DESIGN NO. 1 & 2	
71031-8	SEPARATION TRUSS - DESIGN NO. 1 & 2	
71031-9	CONTAINER OPERATION - DESIGN NO. 1	
71031-10	VEHICLE OPERATION - DESIGN NO. 1	
71031-11	BURTONING OPERATION - DESIGN NO. 1	
71031-12	STOWED CONDITION - DESIGN NO. 1	
71031-13 (2 sheets)	BOOM CRANE ARRANGEMENT - DESIGN NO. 2	

APPENDIX A

BOOM STRUCTURE CALCULATIONS

The total load on the boom structure required to be put in motion is the weight of the trolley and the boom,

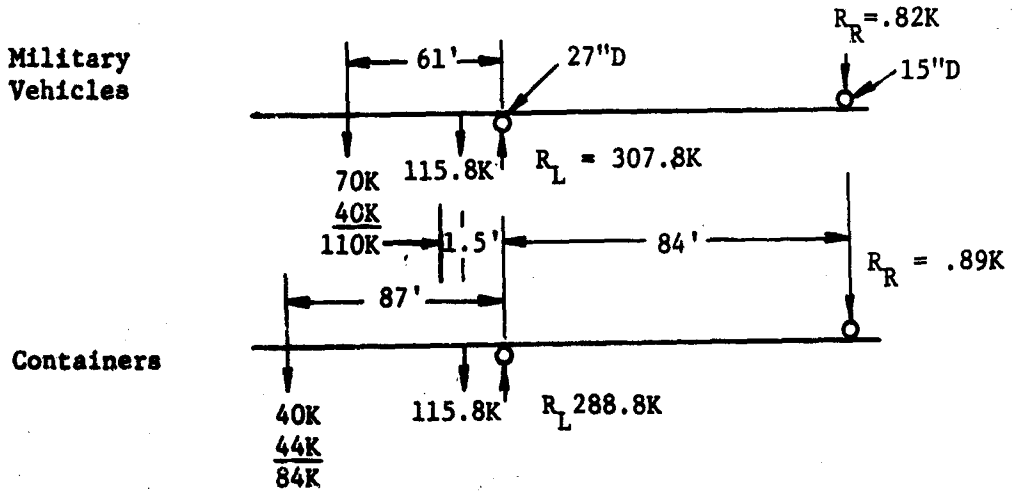
$$T_L = W_T + W_B$$

$$T_L = 40K + 116K = 156K$$

Force, F_L, due to list is:

$$F_L = T_L \times \text{Sin (1 degree)} = 2730 \text{ pounds}$$

Force, F_R, due to rolling contact is determined by the wheel diameters and the reactions.



$$F_R = R_L \times \frac{.005}{13.5} + R_R \times \frac{.005}{7.5}$$

$$F_R = 1000\left(307.8 \times \frac{.005}{13.5} \times 82 \times \frac{.005}{7.5}\right) = 114 + 54.6 = 168.6\#$$

$$F_R = 1000\left(288.8 \times \frac{.005}{13.5} + 89 \times \frac{.005}{7.5}\right) = 107 + 59.4 = 166.4\#$$

Force, F_f, to overcome bearing friction is determined from the anti-friction bearings starting from maximum outreach and assuming 1 percent loss,

$$E_f = (307.8 + 82) \times .01 \times 1000 = 3898\#$$

Force, F_L, to overcome a list of 1 degree is determined by the sum of the trolley and boom weights.

$$F_L = (40{,}000 + 115{,}800) \times .0175 = 2720\#$$

$$\text{Total Force } F_T = F_R + F_f + F_L$$

$$F_T = 169 + 3898 + 2720 = 6787\#$$

$$HP = \frac{6787 \times 50 \text{ FPM}}{33{,}000 \times .7} = 14.7$$

select 15 HP

The total load on the boom structure required to be put into motion is the weight of the load, trolley and boom:

$$T_L = W_L + W_T + W_B$$

$$T_{L1} = 44K + 32K + 123.3K = 199.3K$$

$$T_{L2} = 70K \times 32K + 123.3K = 225.3K$$

The force, F_L, due to list is:

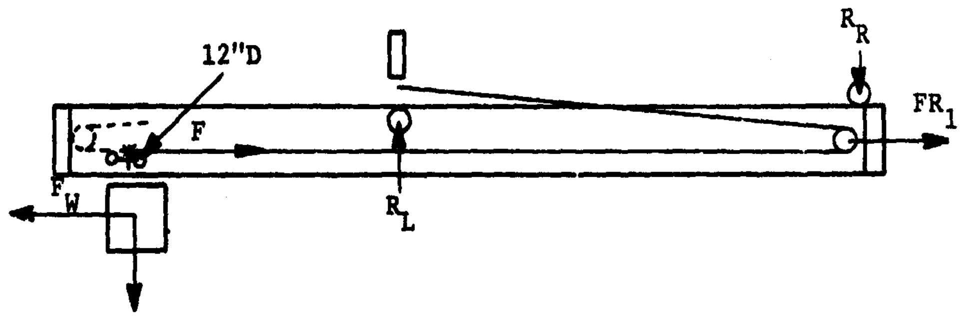
$$F_{L1} = 199.3 \times 1000 \times \text{Sin (1 degree)} = 3500\#$$

$$F_{L2} = 225.3 \times 1000 \times \text{Sin (1 degree)} = 3940\#$$

Force, F_R, due to rolling contact is determined by the wheel diameters and reactions on the boom. In addition, there is a force required to move the loaded trolley resulting from rolling contact and bearing friction.

Force, F_W, due to wind is assumed to be the maximum as used for containers handled under Design No. 1.

$$F_W = 2357\#$$



W_T = 32K

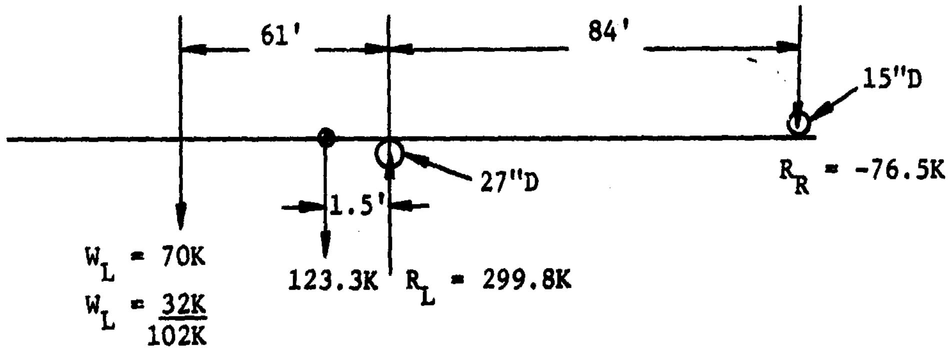
W_T = $\underline{70K}$
102K

$$F = (W_T + W_L) \times \frac{.005}{6} + (W_T + W_L)\ .01 + FW$$

$$F = (102K \times \frac{.005}{6} + 102K \times .01) \times 1000 + 2357 = 3462\#$$ (Military Vehicles)

$$F = 76K \times \frac{.005}{6} + 76K \times .01 + 2357 = 3182\#$$ (Containers)

$$FR_1 = 2F = 6924\# \text{ (max.)}$$



$$FR_2 = R_L \times \frac{.005}{13.5} \times R_R \times \frac{.005}{7.5}$$

$$FR_2 = 1000\ (299.8K \times \frac{.005}{13.5} + 76.5 \times \frac{.005}{7.5} = 111 + 51 = 162\#$$

Force, F, to overcome bearing friction, is found in a similar manner to that in Design No. 1:

$$F_F = (299.8 + 76.5) \times .01 = 3763\#$$

$$\text{Total Force } F_T = F_{L2} + F_{R1} + R_{R2} + F_F$$

$$F_T = 3940 + 6924 + 162 + 3763$$

$$F_T = 14{,}789\#$$

Based upon a mechanical drive efficiency of .7, horsepower,

$$HP = \frac{14{,}789 \times 400}{33{,}000 \times .7} = 256\ HP$$

APPENDIX B

DERIVATION OF SEPARATION FORCES FOR VARIOUS CONDITIONS

Moving hull against the sea such as a rudder - following the equation $F = KAV^2$ - where:

F = normal force on the hull in pounds

A = projected area normal to motion in square feet

K = factor for fineness of lines assumed to be .8

V = velocity of flow by the hull in knots

A = 310 feet x 14 feet = 4350 square feet

V = 90 feet in 5 minutes = 180 feet/hour or .178 knots

$F = .8 \times 4350 \times .178^2 = 110$ pounds

TABLE I

SEPARATION FORCES, NO SHELTER, BROADSIDE WIND

Broadside Wind Force with no Shelter (FWB) = $.004\ AV^2$ (U. S. Navy Specifications)

A = 318' x 14' + 106' x 6' = 5085 square feet

Wind Velocity (Knots)	8.5	12.0	16.0	19.0	24.0
Wave Length (Feet)	20	40	71	99	160
FWB	1465#	2820#	5200#	7320#	11,700#

TABLE II

SEPARATION FORCES, HEAD ON WIND

Head-on Wind Force (FWH) = $.004\ AV^2$ (U. S. Navy Specifications)

A = 30' x 14' + 30' x 6' = 600 square feet

Wind Velocity (Knots)	8.5	12.0	16.0	19.0	24.0
Wave Length (Feet)	20	40	71	99	160
FWH	174#	346#	615#	867#	1383#

TABLE III

SEPARATION FORCES WITH SHELTER, BROADSIDE WIND

Broadside Wind Force sheltered by Pier (FWP) = $.004\ AV^2$

A = 318' x 5' + 106' x 6' = 2225 square feet

Wind Velocity (Knots)	8.5	12.0	16.0	19.0	24.0
Wave Length (Feet)	20	40	71	99	160
FWP	640#	1238#	2280#	3220#	5100 #

Head-on Hull Drag Force (Fd) = $.009\ SV^{1.825}$ (U. S. Navy Specifications)

S = Wetted Area in square feet

V = 4 Knots

Average Girth = 54 feet

S = 54' x 310' = 16,700 square feet

Drag = $.009 \times 16{,}700 \times 4^{1.825}$ = 1880 pounds

Drag from one LCM-8

Wetted Area S = 1915 square feet

Drag = $.009 \times 1915 \times 4^{1.825}$ = 216 pounds

Drag from one Larc LX

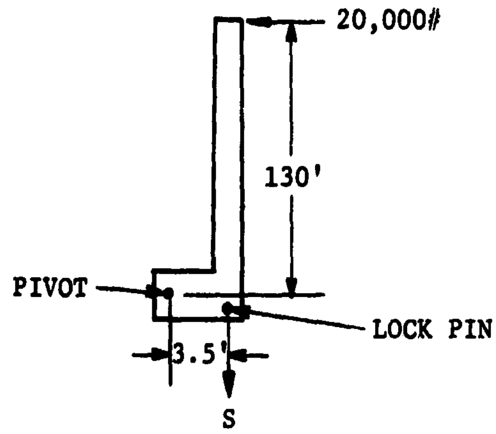
Wetted Area S = 2240 square feet

Drag = $.009 \times 2240 \times 4^{1.825}$ = 252 pounds

APPENDIX C

CALCULATIONS OF LOADS ON LOCK PINS

The load on each lock pin is as follows:

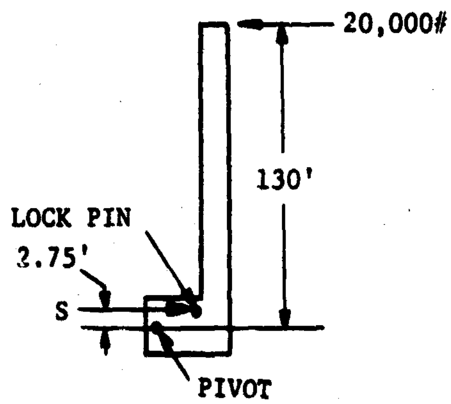


$$S = 20K \times \frac{130}{3.5} = 743K$$

Allowable shear stress = 32,000 x .6 = 19,200 PSI

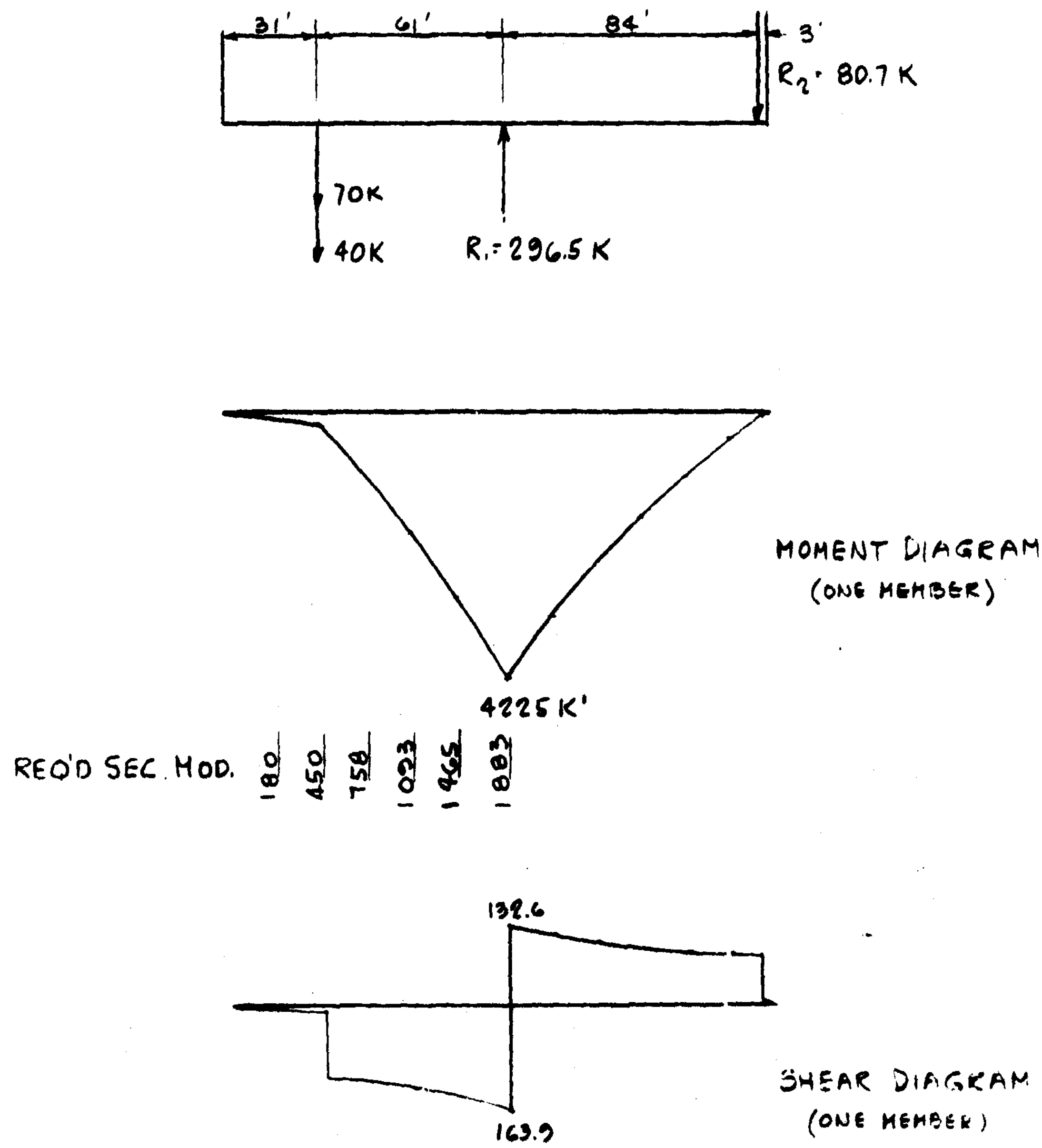
6 in. dia. pin in double shear 2 x 28.3 sq. in. = 56.6 sq. in.

$$\text{Stress} = \frac{743{,}000}{56.6} = 13{,}200 \text{ PSI} < 19{,}200 \text{ PSI}$$

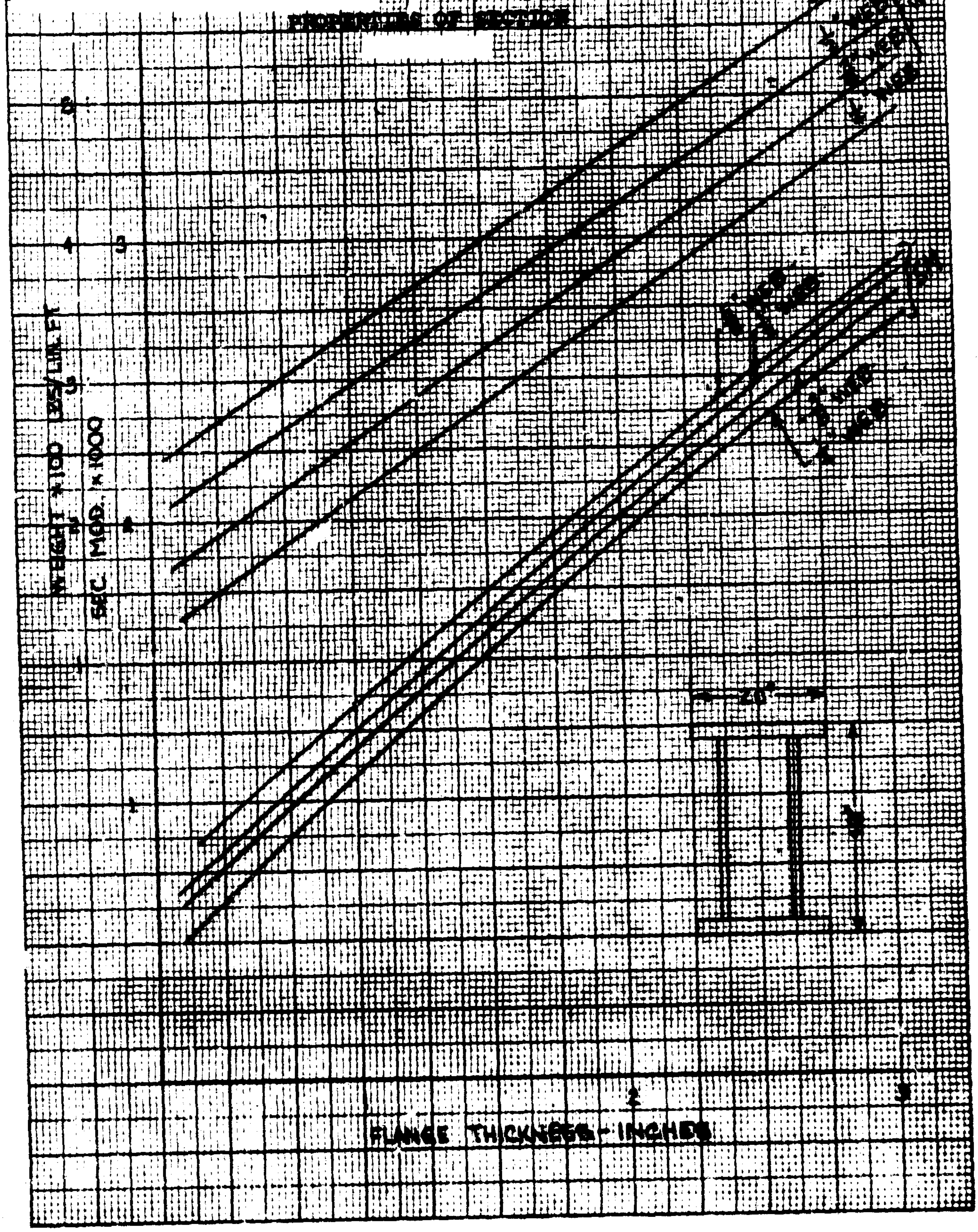


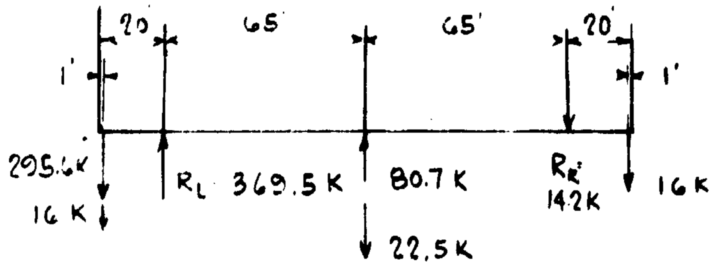
$$S = 20K \times \frac{130}{2.75} = 945K$$

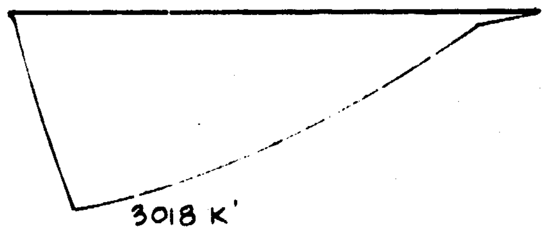
$$\text{Stress} = \frac{945{,}000}{56.6} = 16{,}700 \text{ PSI} < 19{,}200 \text{ PSI}$$



FREE BODY, SHEAR AND MOMENT DIAGRAMS

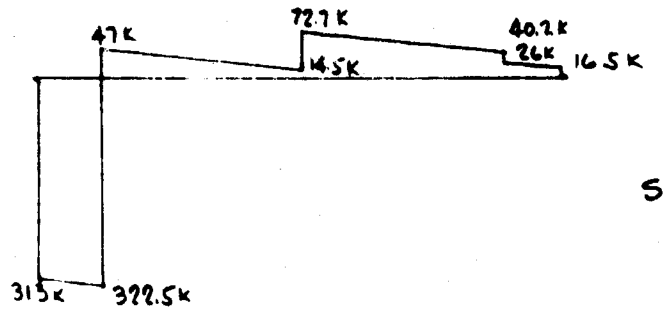

SEC. MOD. x 1000
FLANGE THICKNESS - INCHES





MOMENT DIAGRAM
(ONE MEMBER)

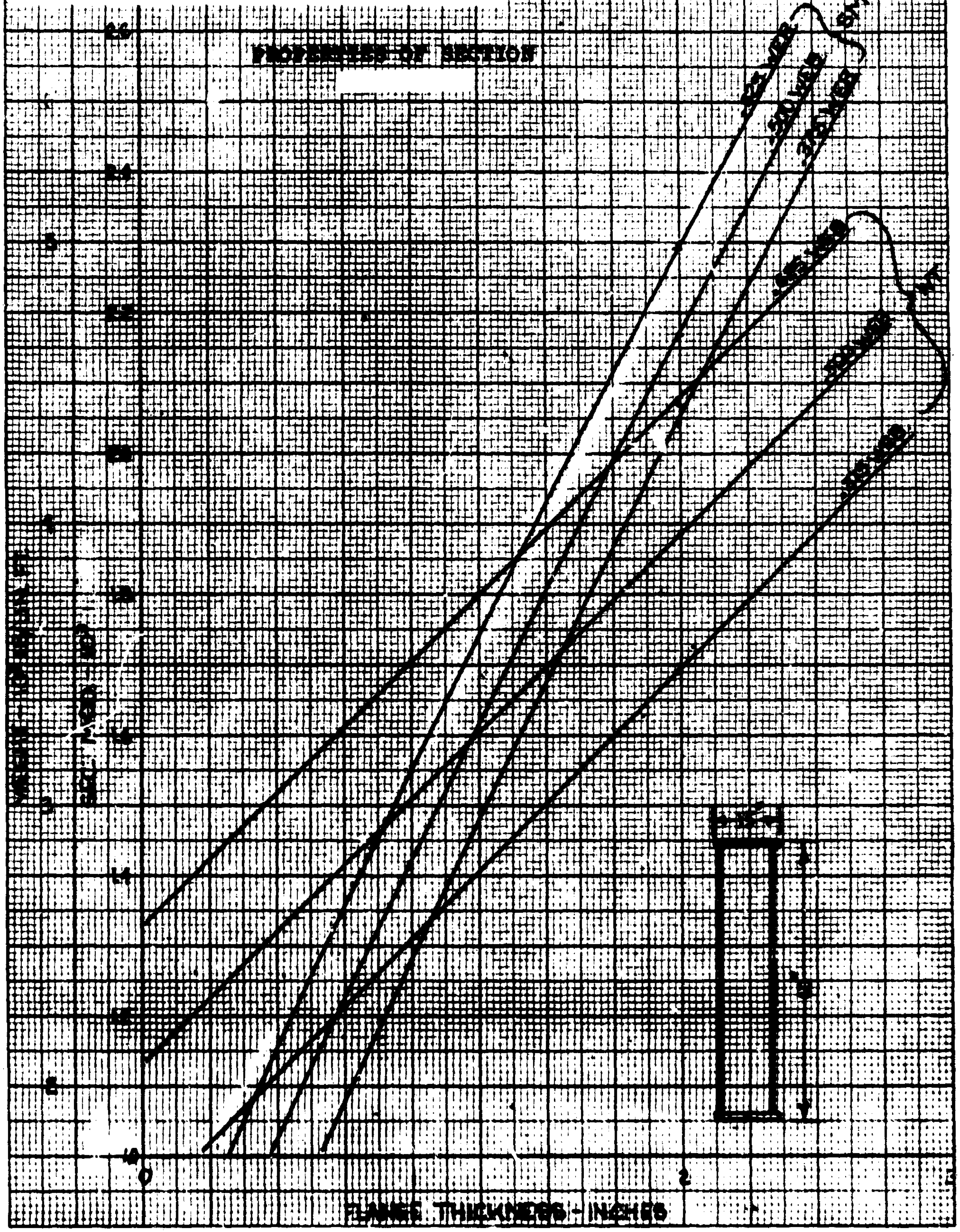
REQD. SEC MOD 1345

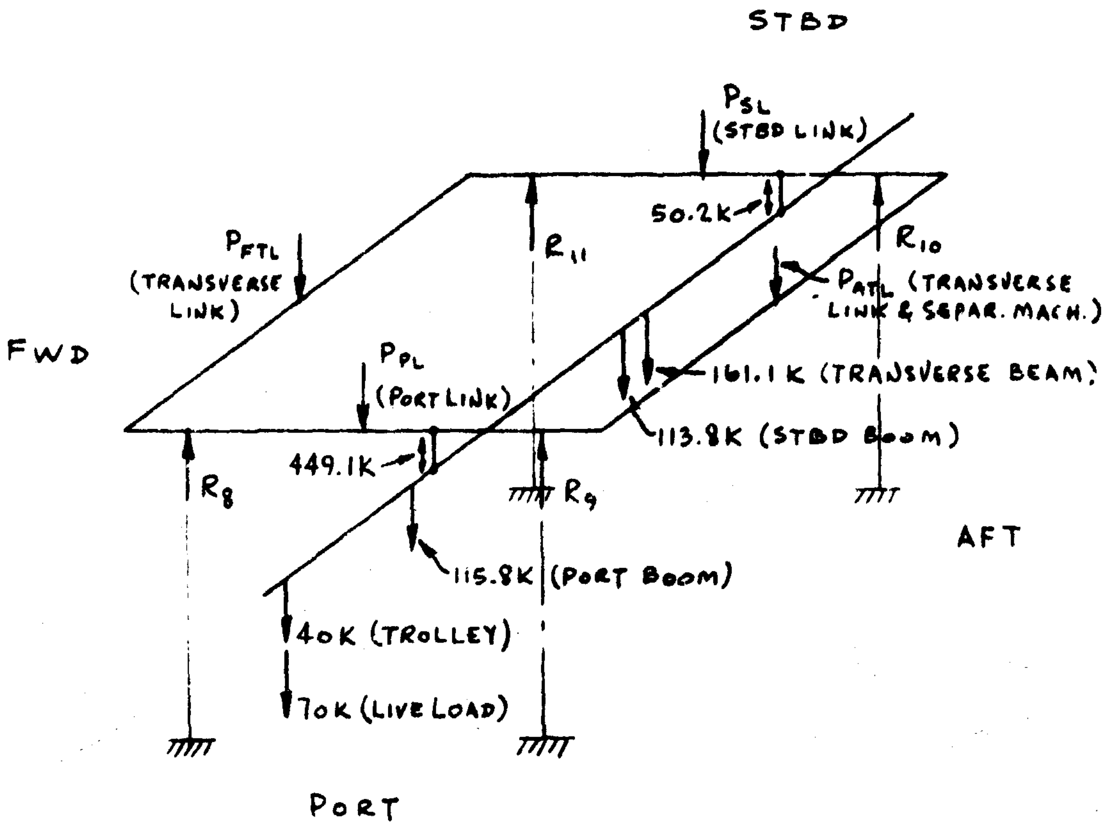


SHEAR DIAGRAM
(ONE MEMBER)

FREE BODY, SHEAR AND MOMENT DIAGRAMS

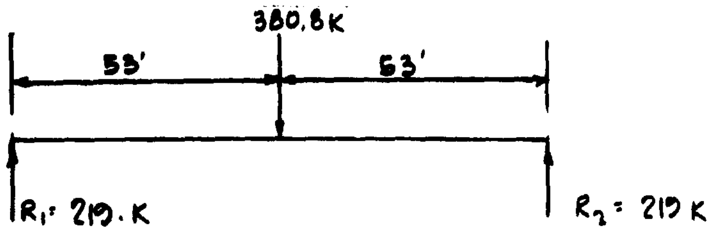
APPENDIX E - Cont'd

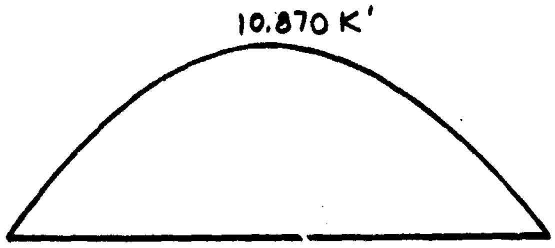




WAVE LENGTH	LOADS & REACTIONS - KIPS							
	P_{PL}	P_{SL}	P_{FTL}	P_{ATL}	R_8	R_9	R_{10}	R_{11}
10'	82.33	82.33	50.70	71.10	124.72	467.63	121.37	72.03
25'	99.65	99.65	104.50	124.90	160.28	503.20	156.93	107.59
40'	133.13	133.13	209.00	229.40	229.23	572.15	225.88	176.54
60'	213.35	213.35	459.00	479.40	394.33	737.74	340.97	341.63

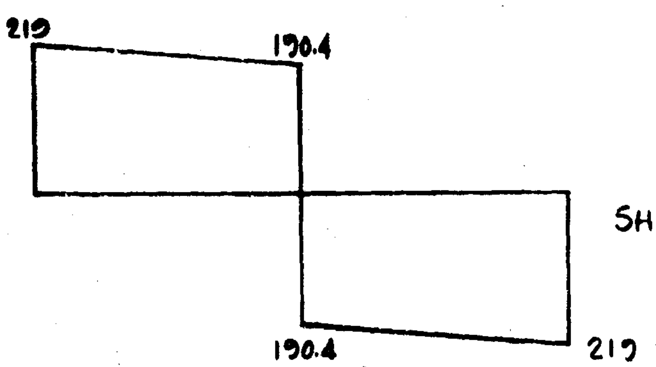
CRANE WEIGHTS AND REACTIONS AT KINGPOSTS FOR CONDITION OF NO SEPARATION TRUSS BETWEEN HULLS





MOMENT DIAGRAM

REQ'D SEC.MOD. 4910

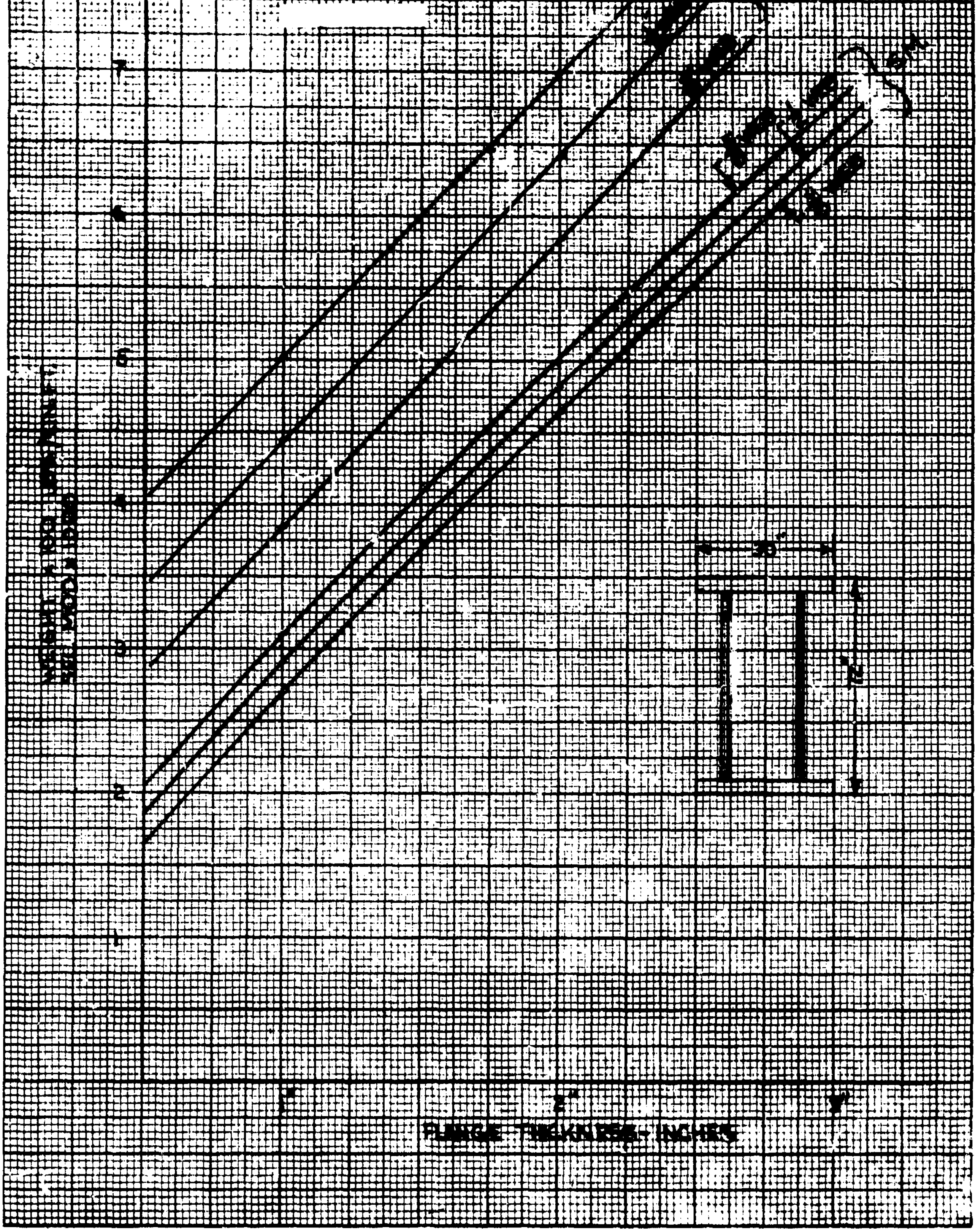


SHEAR DIAGRAM

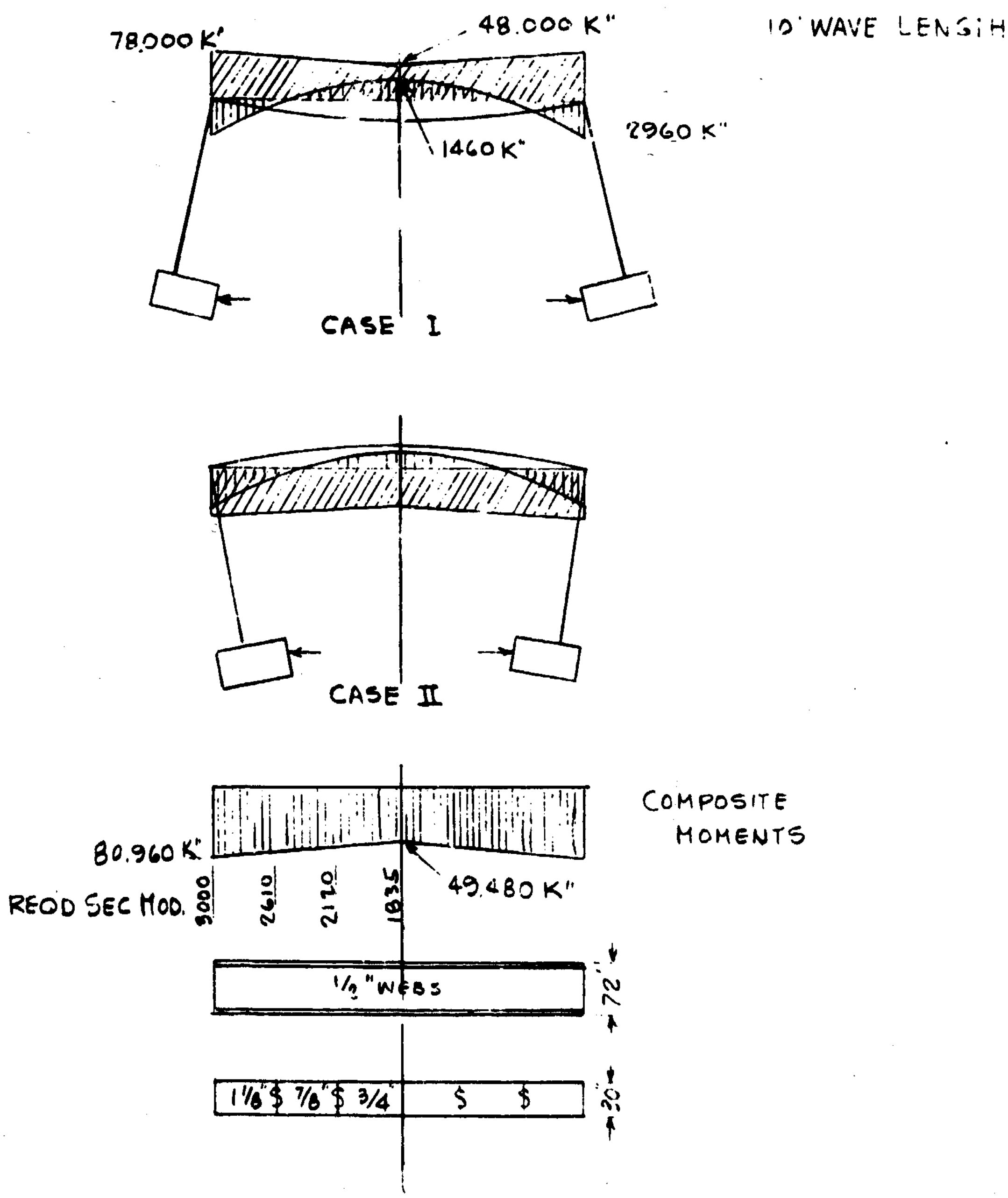
FREE BODY, SHEAR AND MOMENT DIAGRAMS

APPENDIX F - Cont'd

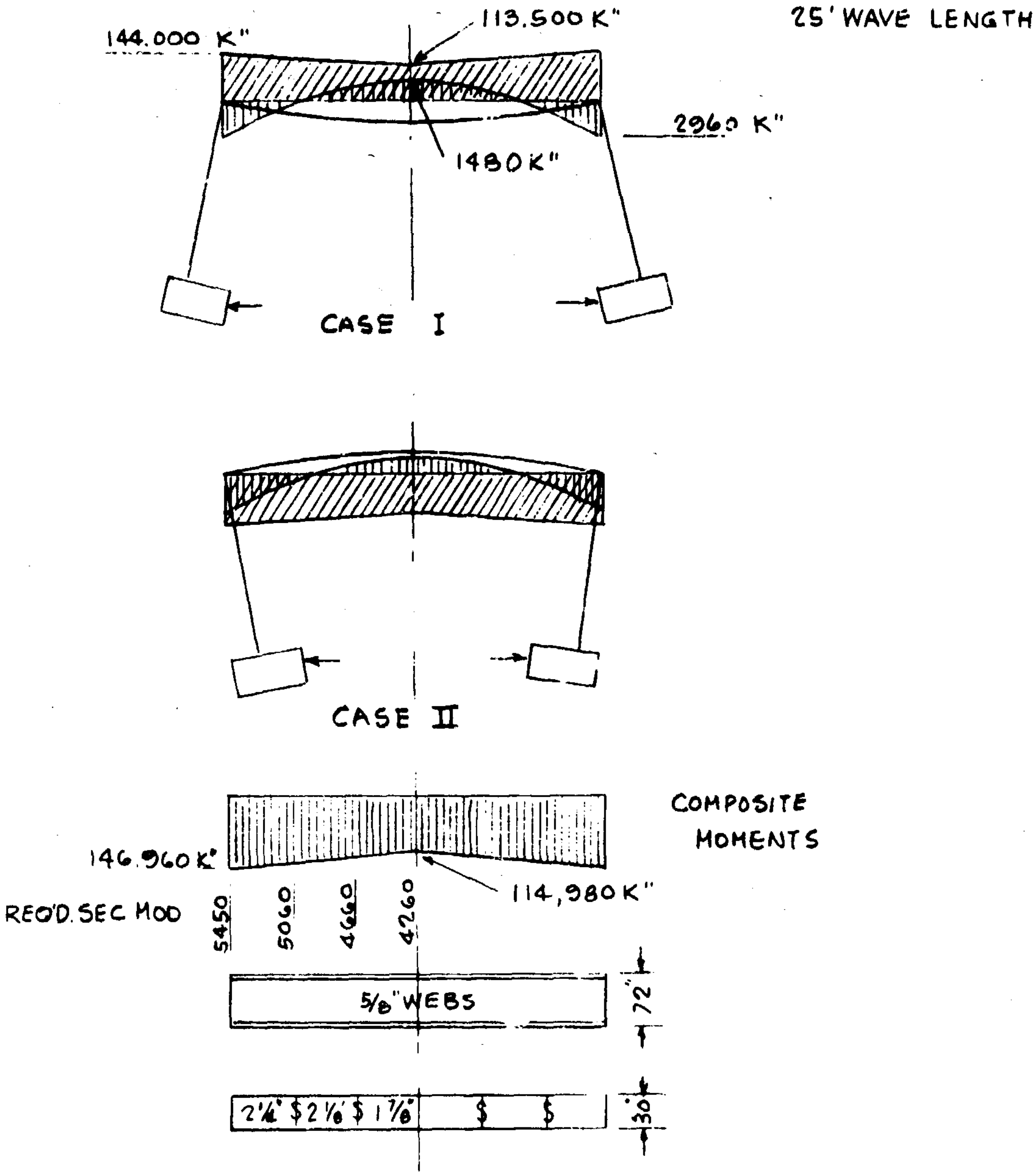
PROPERTIES OF SECTION



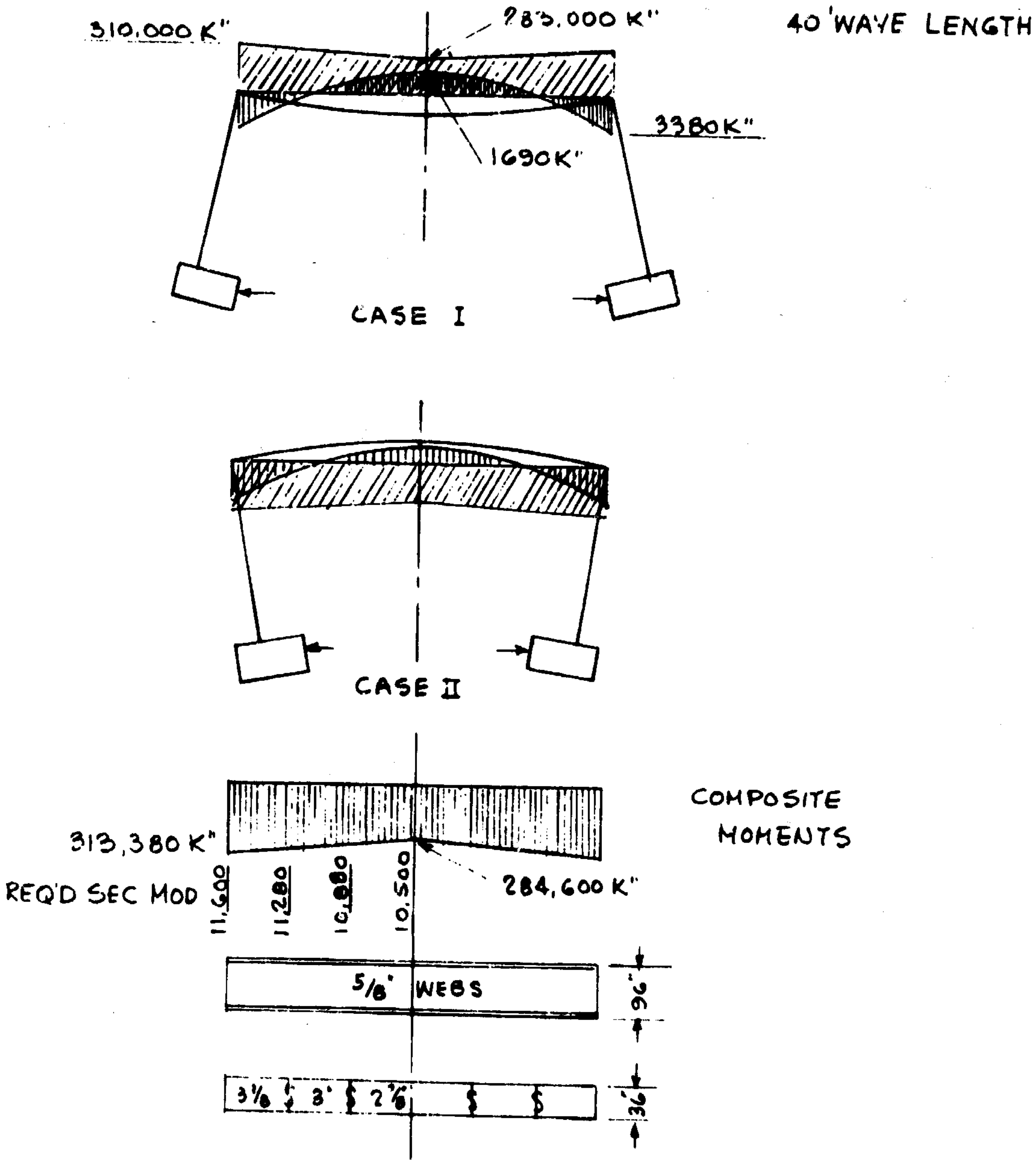
APPENDIX G - ANALYSIS OF TRANSVERSE LINKS



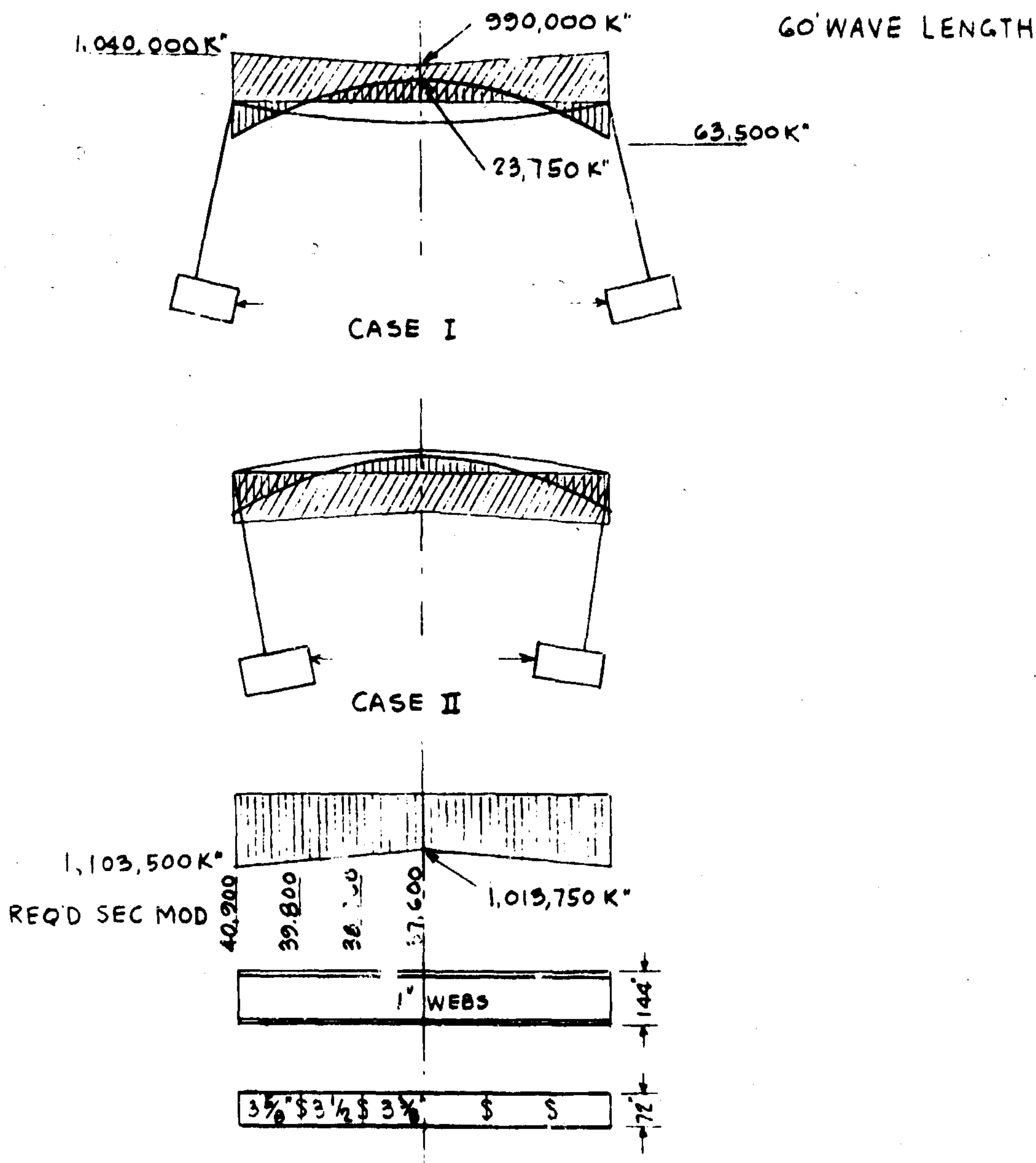
BENDING MOMENTS FOR 10' WAVE LENGTH



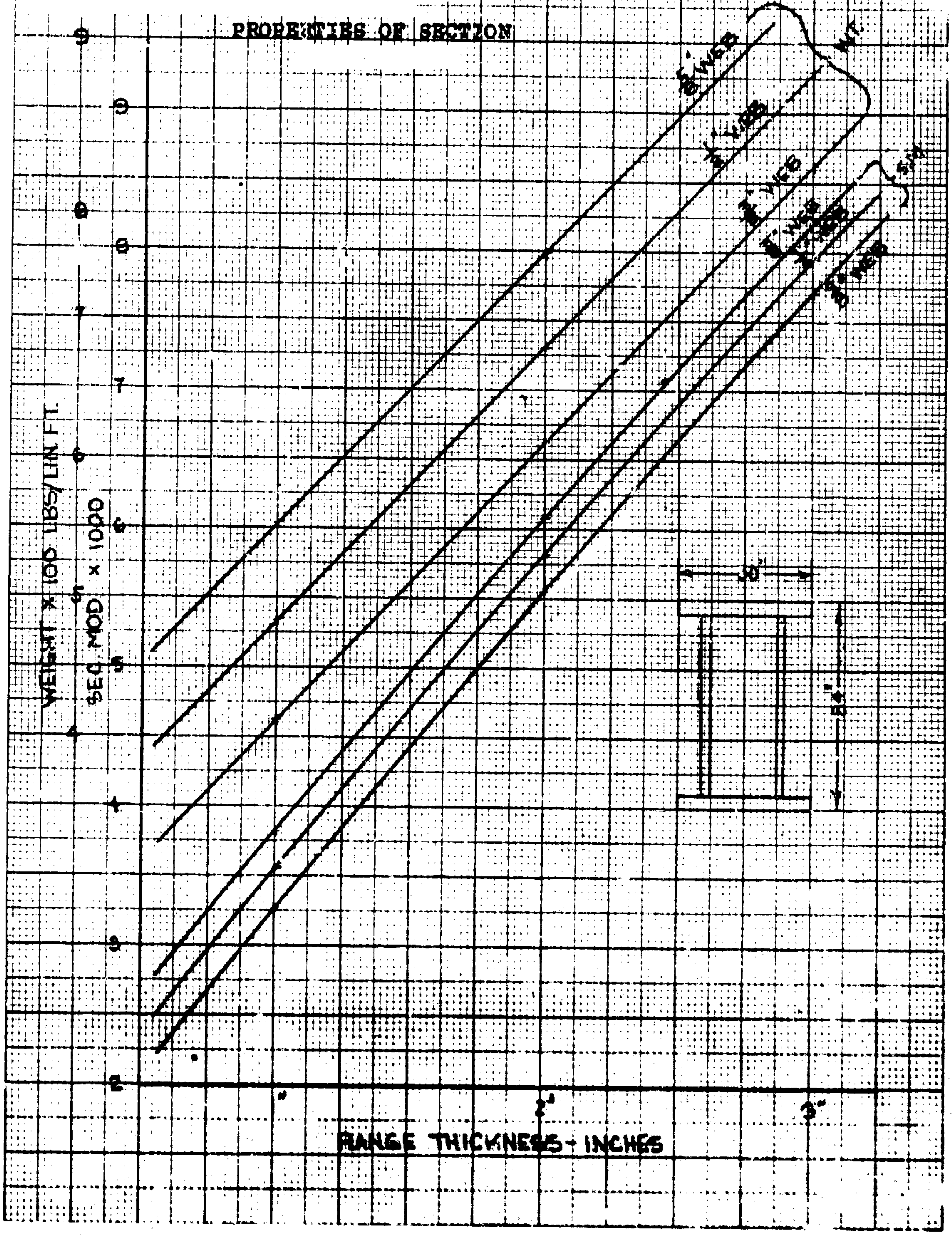
BENDING MOMENTS FOR 25' WAVE LENGTH



BENDING MOMENTS FOR 40' WAVE LENGTH

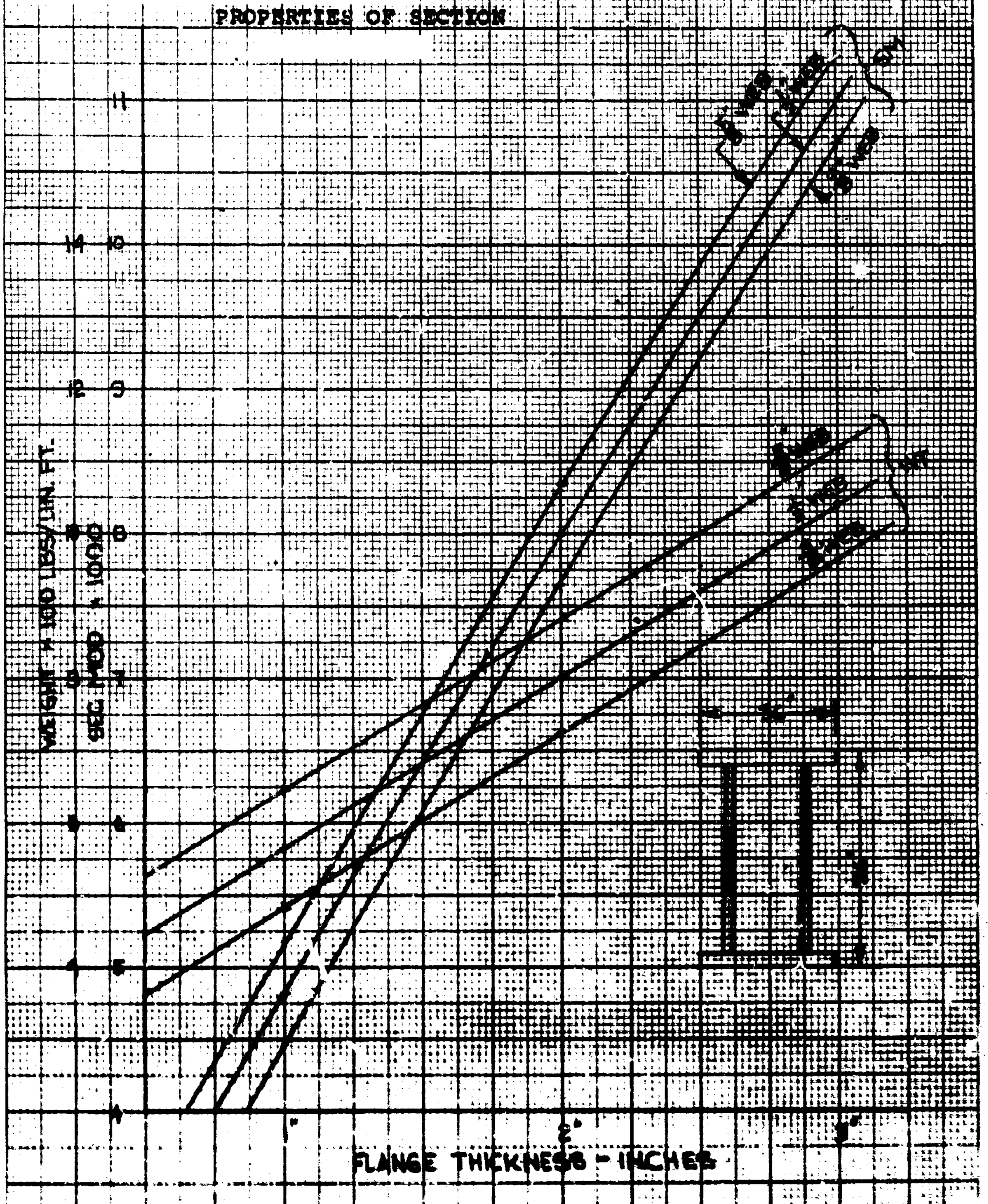


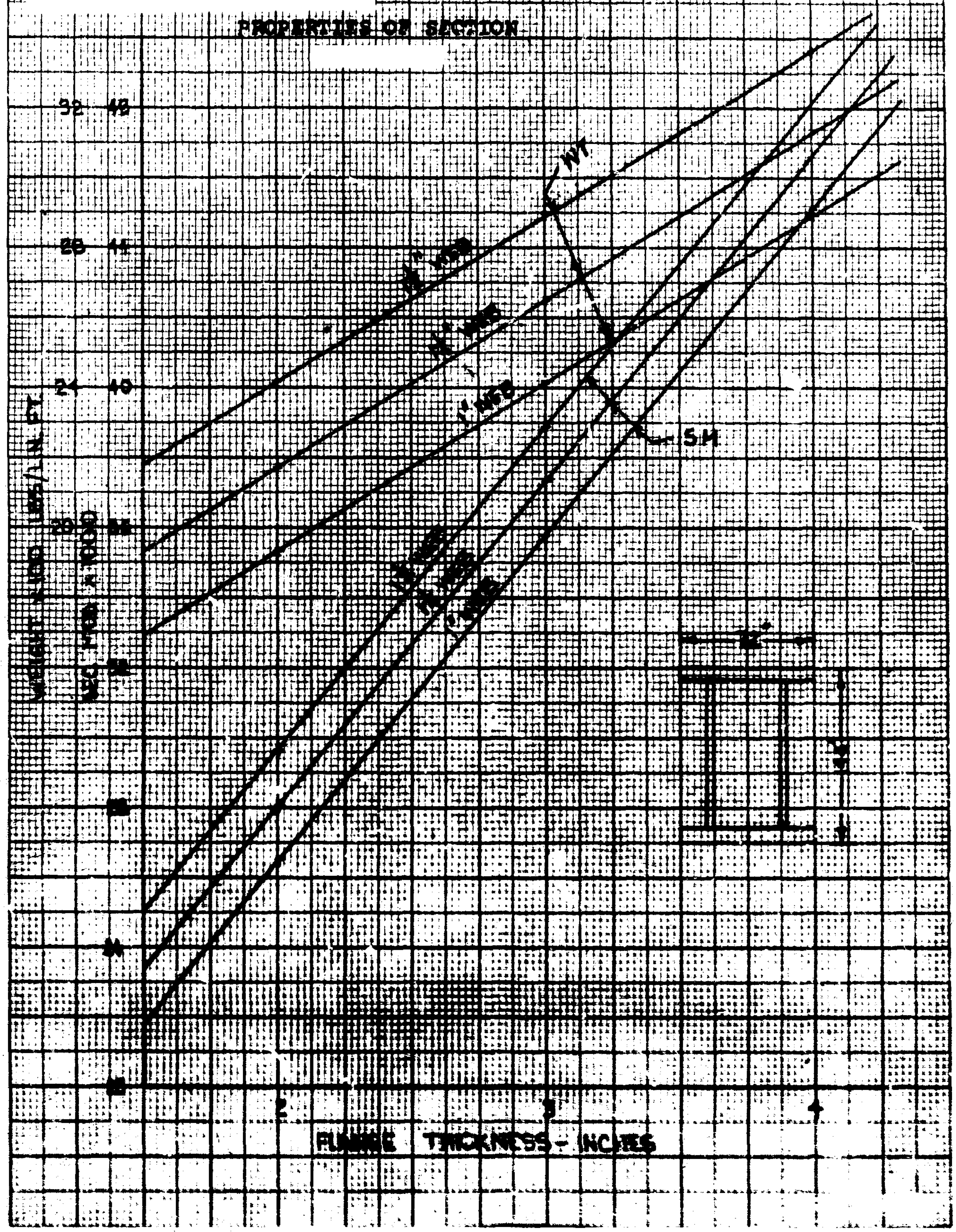
BENDING MOMENTS FOR 60' WAVE LENGTH


PROPERTIES OF SECTION
WT.
S.M.
WEB
WEIGHT × 100 LBS/LIN. FT.
SEC. MOD × 1000
1"
2"
3"
FLANGE THICKNESS - INCHES

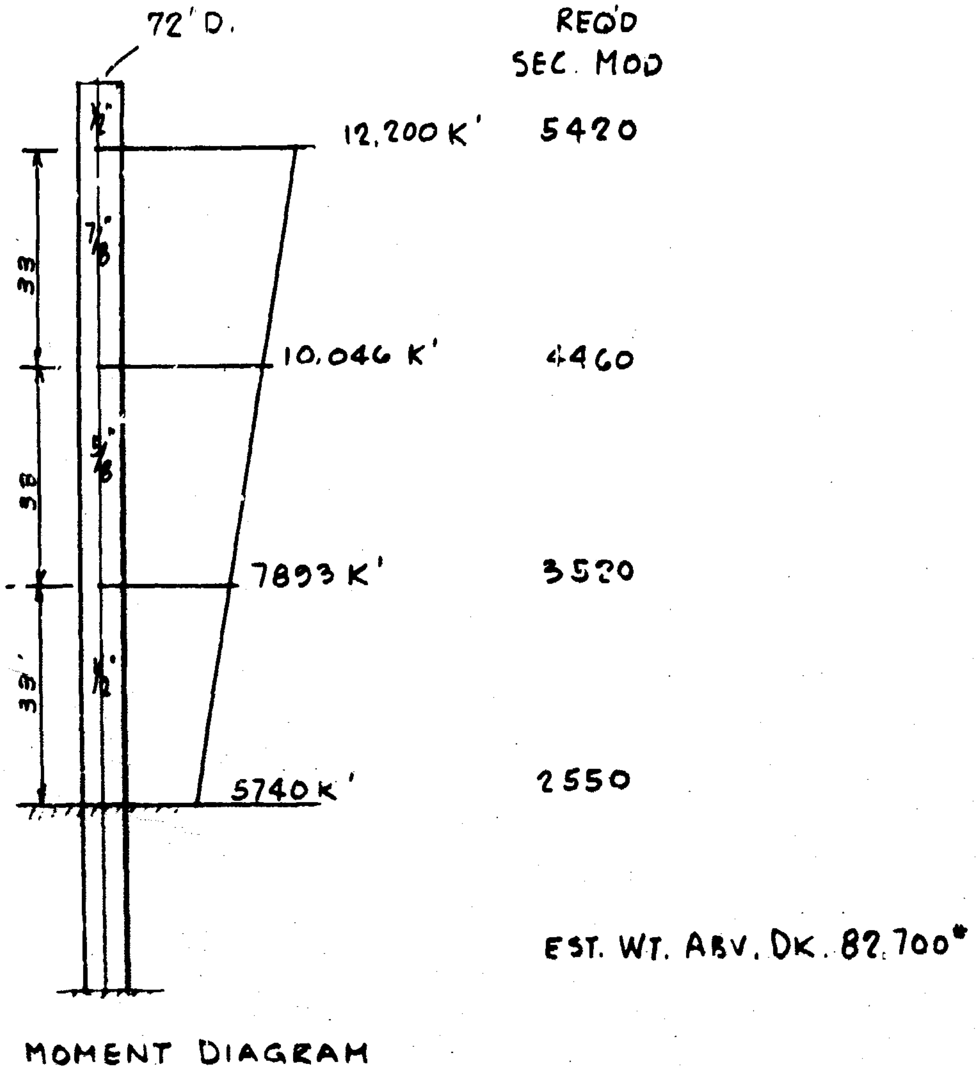
APPENDIX G - Cont'd

PROPERTIES OF SECTION



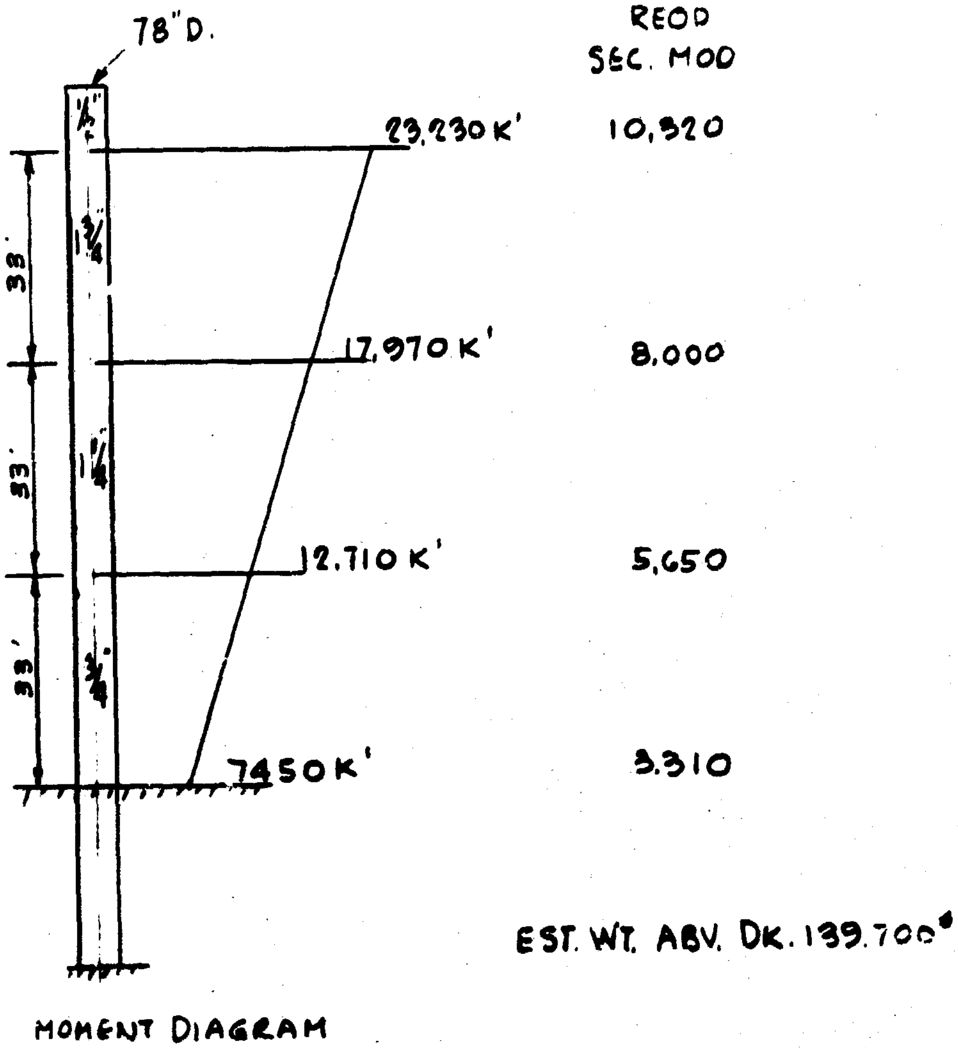

PROPERTIES OF SECTION
WT
SM
1" WEB
FLANGE THICKNESS - INCHES

5.5' WAVE LENGTH



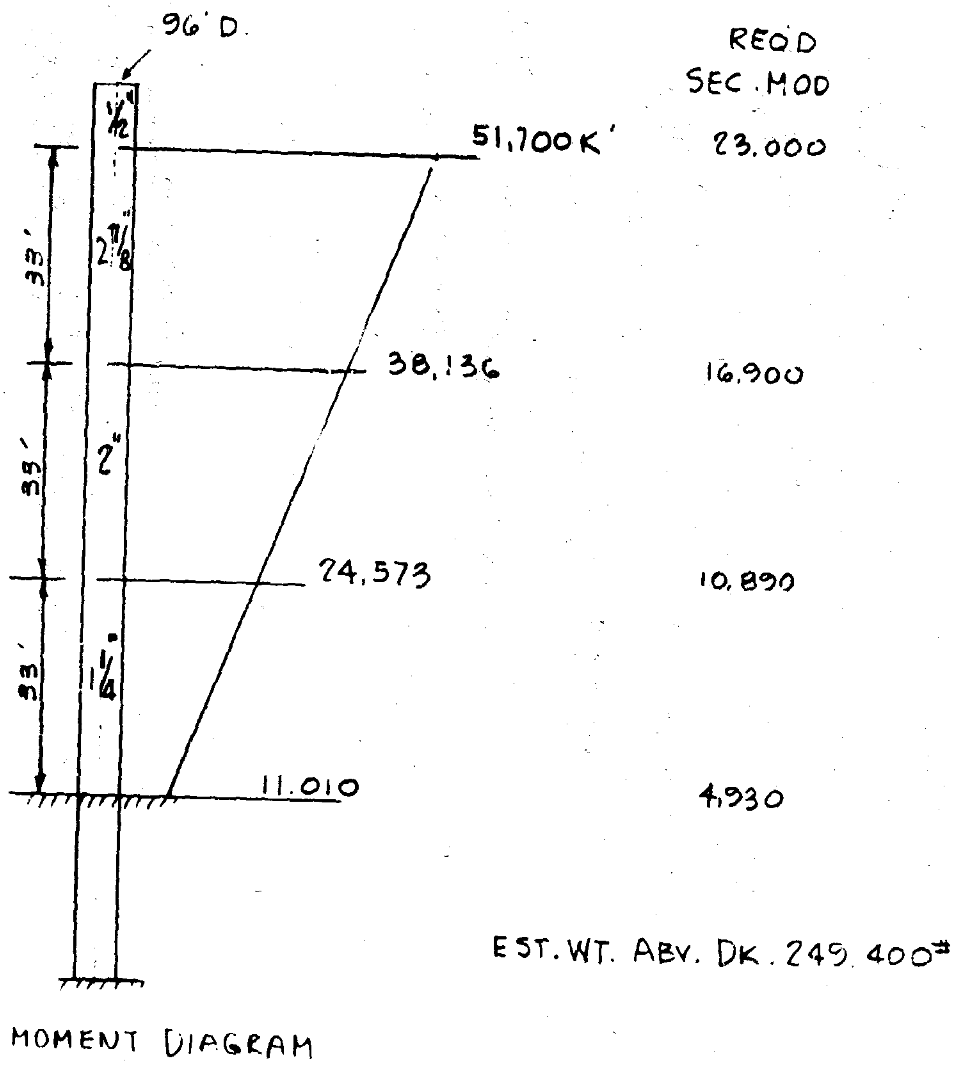
72" DIAMETER KINGPOST MOMENT DIAGRAM FOR 5.5' WAVE LENGTH

25' WAVE LENGTH



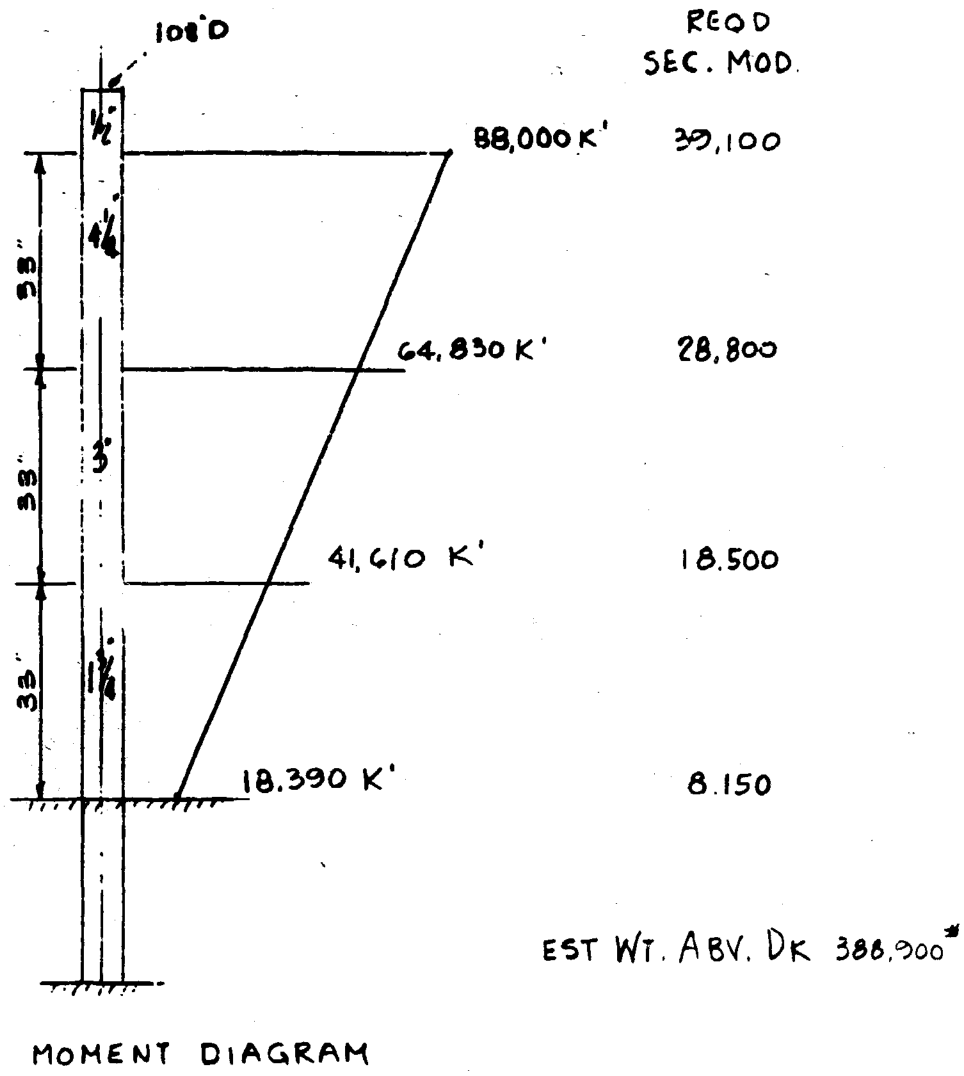
78" DIAMETER KINGPOST MOMENT DIAGRAM FOR 25' WAVE LENGTH

40' WAVE LENGTH



96" DIAMETER KINGPOST MOMENT DIAGRAM FOR 40' WAVE LENGTH

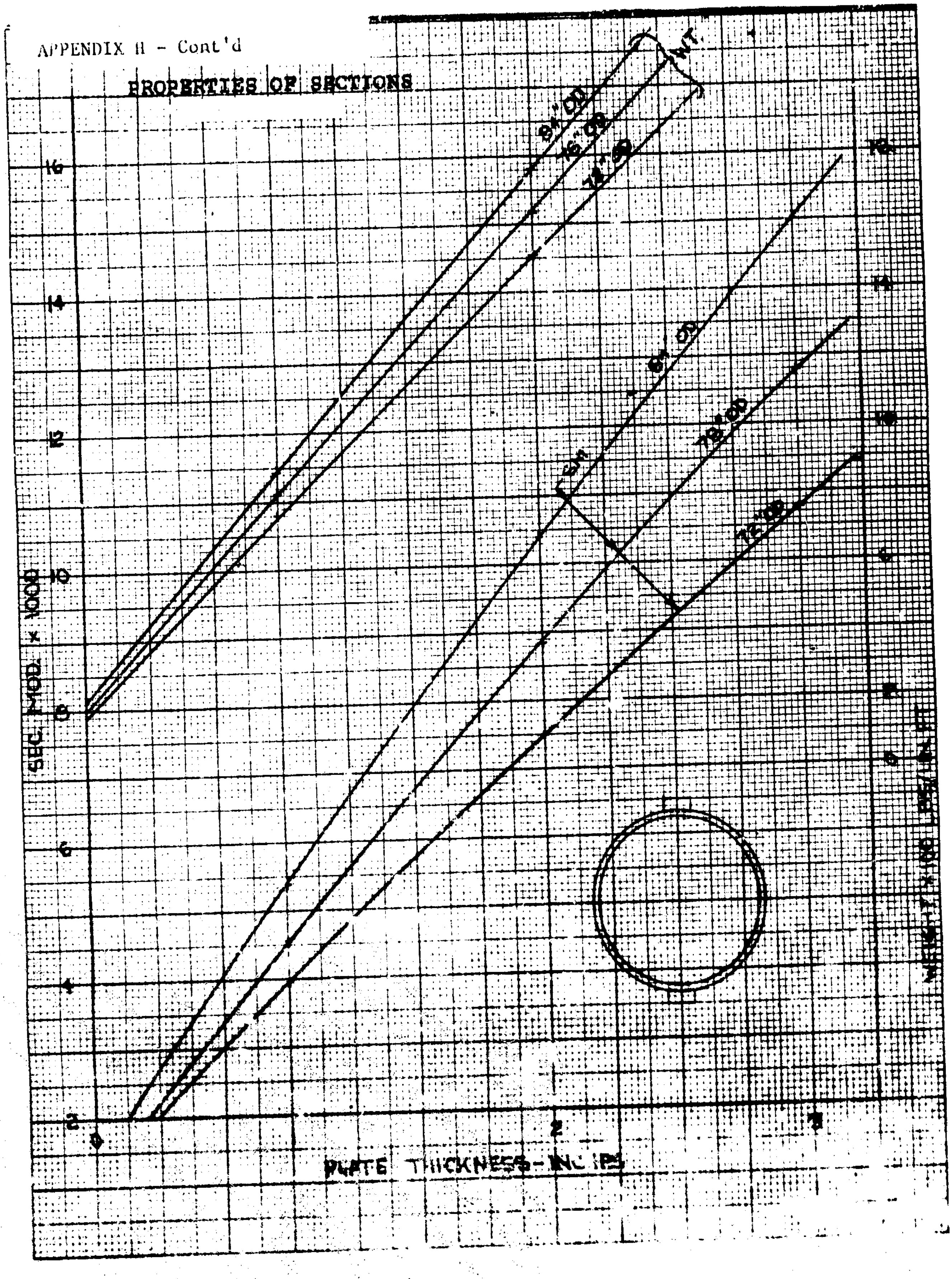
60' WAVE LENGTH



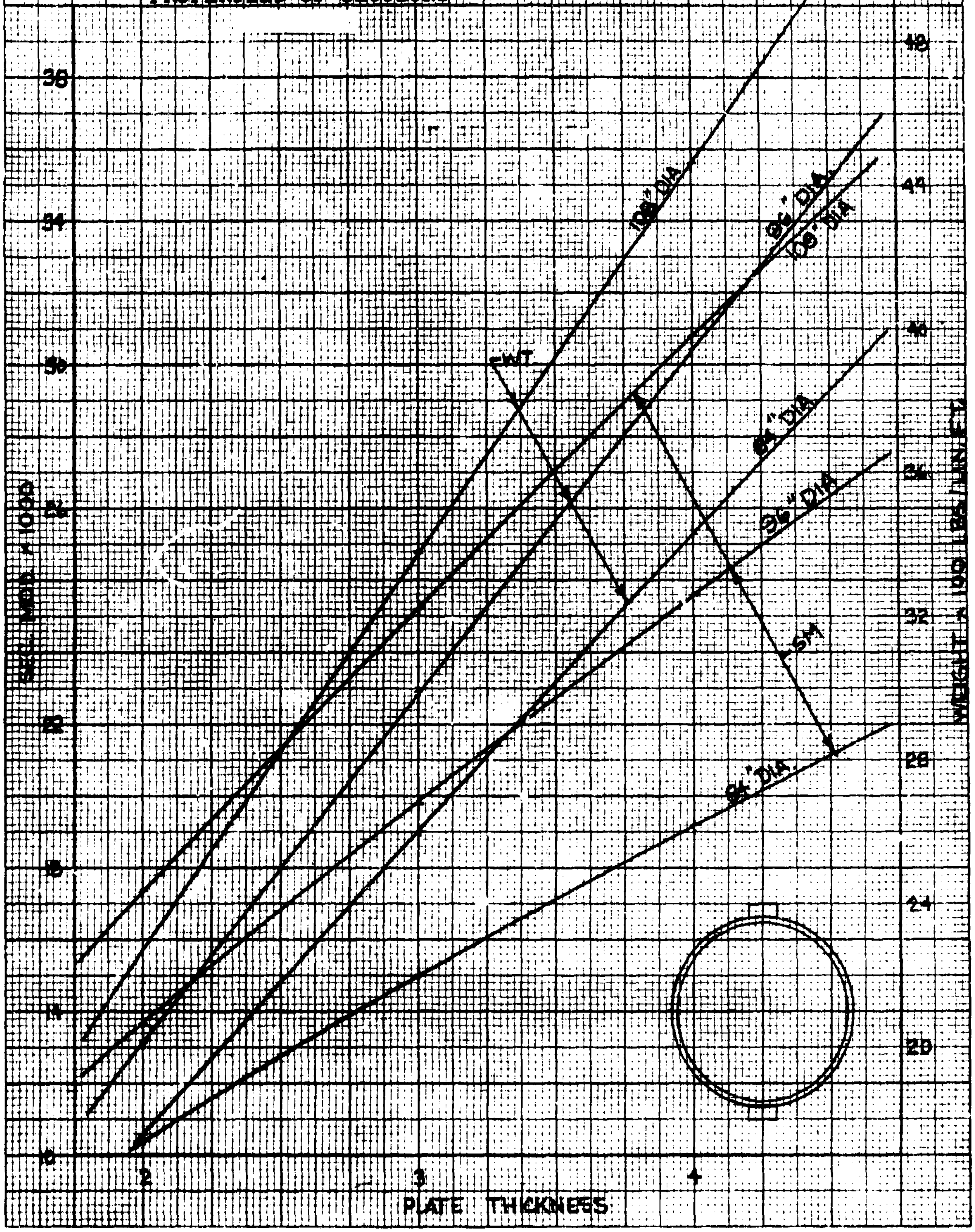
108" DIAMETER KINGPOST MOMENT DIAGRAM FOR 60' WAVE LENGTH

APPENDIX H - Cont'd

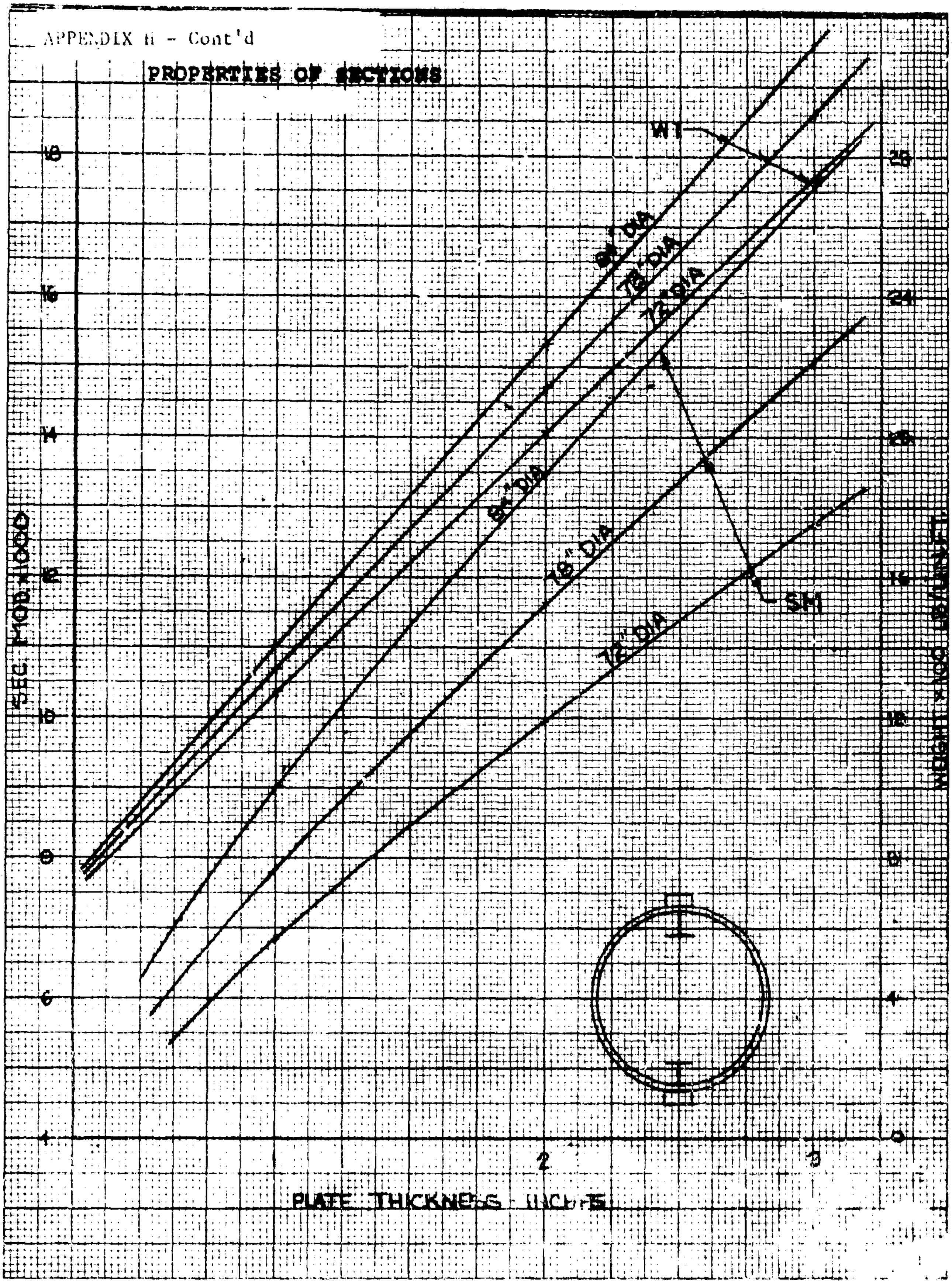
PROPERTIES OF SECTIONS



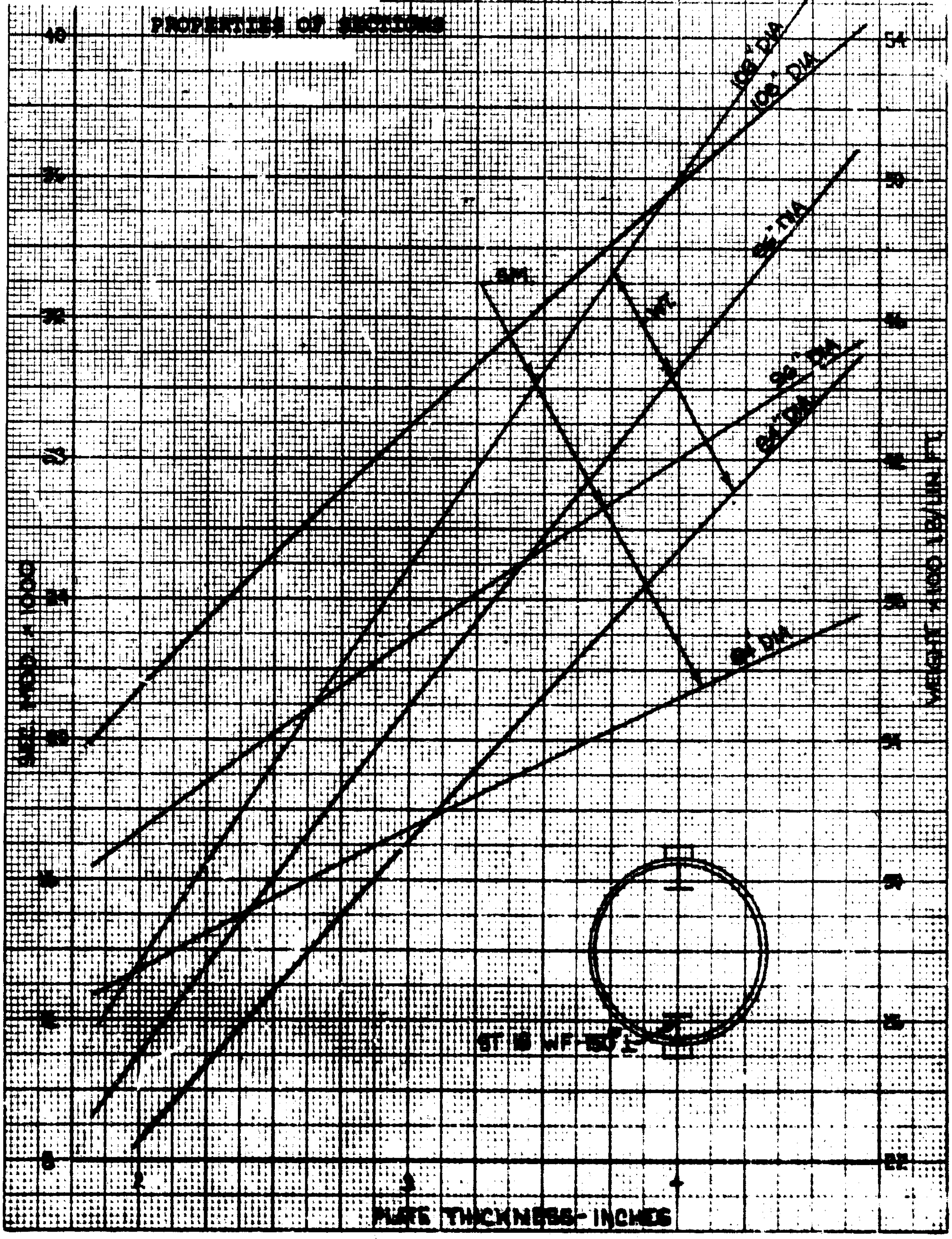
PROPERTIES OF SECTIONS



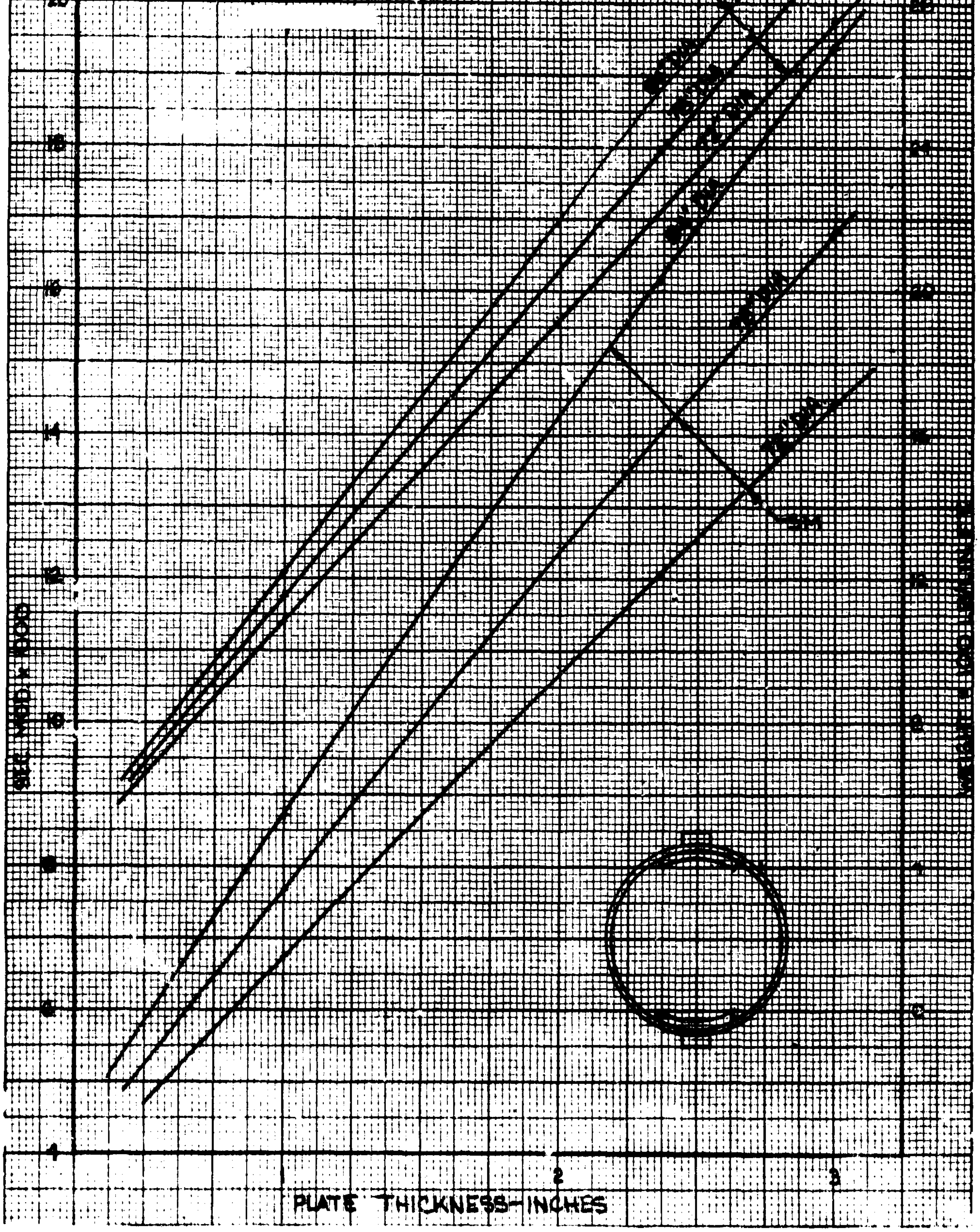
APPENDIX H - Cont'd



APPENDIX H - Cont'd

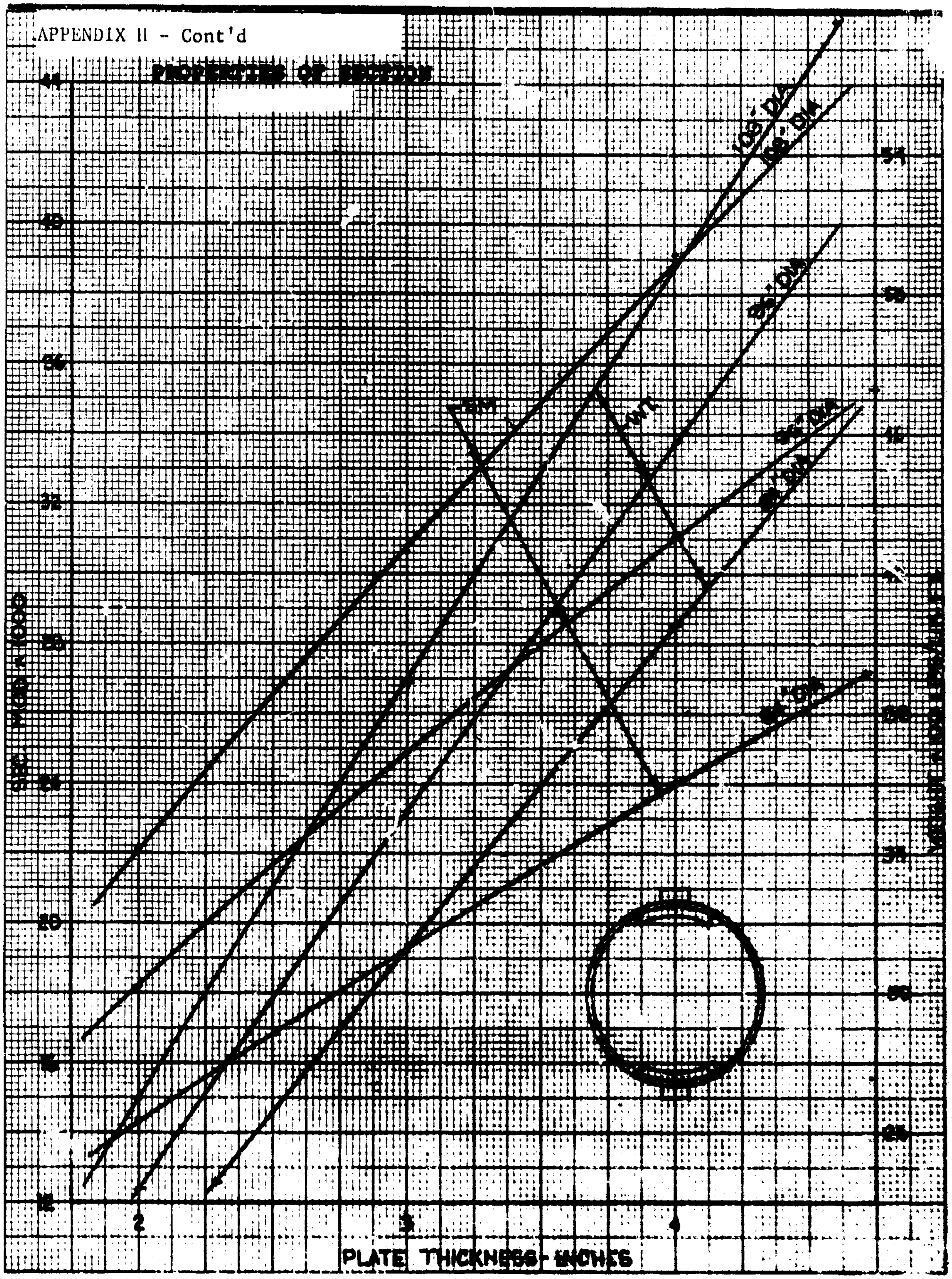


PROPERTIES OF SECTION



APPENDIX II - Cont'd

PROPERTIES OF SECTION



APPENDIX J

ANALYSIS OF SEPARATION TRUSS

Table IV indicates the pipe sizes for the chord members for different wave lengths; the sizes of the struts and diagonals vary to suit the loading.

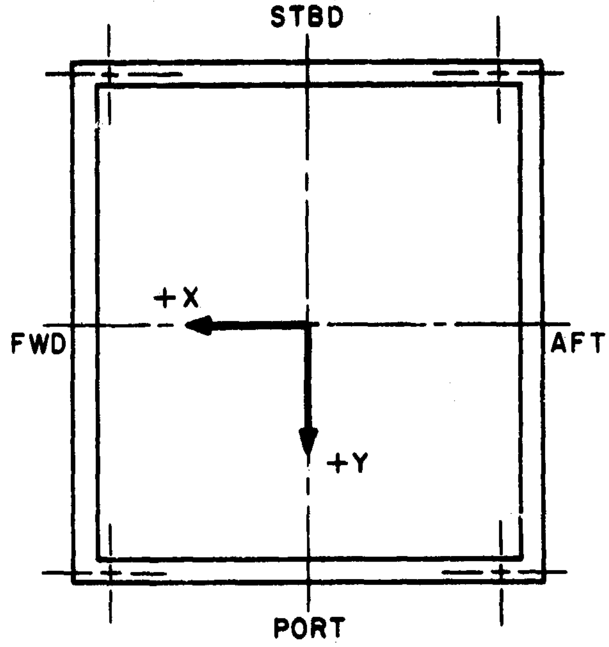
Table IV. Size of Truss Chord Members

Wave Length (Ft.)	Chord Diameter (In.)
5.5	3.5
25	6
40	8
50	8
60	12
80	14
90	14
100	14
110	16
120	16
140	16

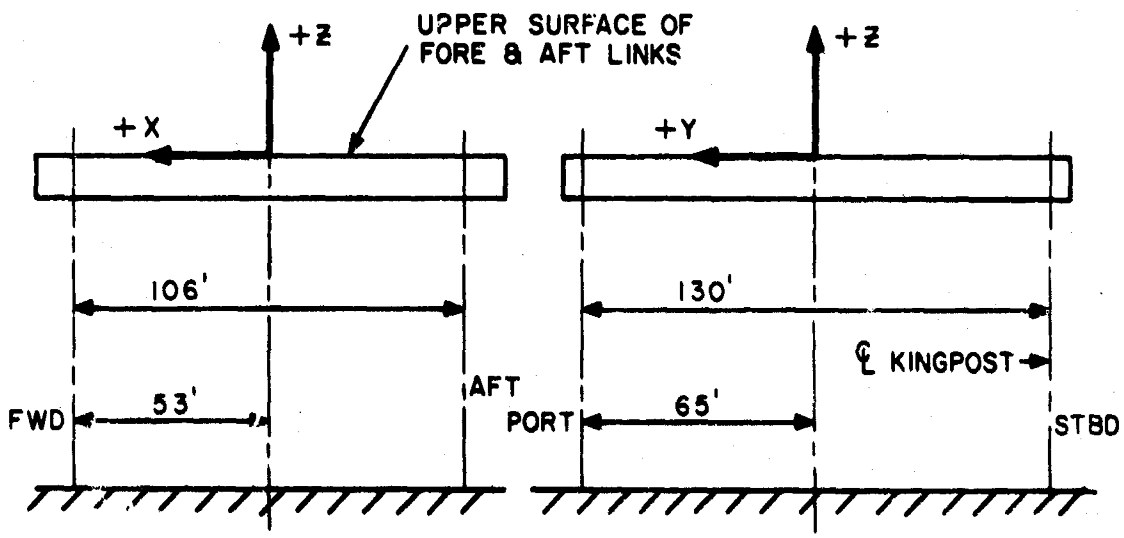
APPENDIX K

DETERMINATION OF CRANE C.G.

In order to determine the c.g. of the crane assembly, let us set up the following coordinate system:



Note: The crane c.g. will be determined below for the condition wherein the trolley, booms and transverse beam are assumed to be geometrically centered with respect to the X and Y coordinate axis.



APPENDIX K - DETERMINATION OF CRANE C.G. (con't)

Item No.	Weight Item	Weight W Kips	X Ft.	Y Ft.	Z Ft.	WX Kip-Ft.	WY Kip-Ft.	WZ Kip-Ft.
1	Trolley Machinery	28.0	0	0	-20.1	0	0	-562.8
2	Trolley Structure	12.0	0	0	-18.6	0	0	-223.2
3	Port Boom	115.8	0	0	-11.6	0	0	-1343.3
4	STBD Boom	113.8	0	0	-11.6	0	0	-1320.1
5	Boom Extend. Mach.	15.3	0	0	-9.3	0	0	-142.3
6	Boom Support Wheels	21.6	0	0	-10.6	0	0	-229.0
7	Transverse Beam	111.2	0	0	-9.3	0	0	-1034.2
8	Articulated Support Units	13.0	0	0	-4.8	0	0	-62.4
9	Hull Separation Mach	20.4	-64.5	0	-3.0	-1315.8	0	-61.2
10	Port Fore & Aft Link	82.3	0	65.0	-3.0	0	5349.5	-246.9
11	STBD Fore & Aft Link	82.3	0	-65.0	-3.0	0	-5349.5	-246.9
12	Fwd Transverse Link	50.7	64.5	0	-3.0	3270.2	0	-152.1
13	Aft Transverse Link	50.7	-64.5	0	-3.0	-3270.2	0	-152.1
	Σ =	717.1				-1315.8	0	-5776.5
	10' Wave Length	C.G.=	-1.8	0	-8.1			

10' Wave Length (items 9–13)

APPENDIX K - DETERMINATION OF CRANE C.G. (con't)

Wave Length	Item No.	Weight Item	Weight W Kips	X Ft.	Y Ft.	Z Ft.	WX Kip-Ft.	WY Kip-Ft.	WZ Kip-Ft.
25' Wave Length	10	Port Fore & Aft Link	99.6	0	65.0	-3.0	0	6474.0	-298.8
	11	STBD Fore & Aft Link	99.6	0	-65.0	-3.0	0	-6474.0	-298.8
	12	Fwd Transverse Link	104.5	64.5	0	-3.0	6740.2	0	-313.5
	13	Aft Transverse Link	104.5	-64.5	0	-3.0	-6740.2	0	-313.5
		Σ =	859.3				-1315.8	0	-6203.1
		25' Wave Length	C.G.=	-1.5	0	-7.2			
40' Wave Length	10	Port Fore & Aft Link	133.1	0	65.0	-3.0	0	8651.5	-399.3
	11	STBD Fore & Aft Link	133.1	0	-65.0	-3.0	0	-8651.5	-399.3
	12	Fwd Transverse Link	209.0	64.5	0	-3.0	13480.5	0	-627.0
	13	Aft Transverse Link	209.0	-64.5	0	-3.0	-13480.5	0	-627.0
		Σ =	1135.3				-1315.8	0	-7031.1
		40' Wave Length	C.G.=	-1.2	0	-6.2			

APPENDIX K - DETERMINATION OF CRANE C.G. (con't)

60' Wave Length

Item No.	Weight Item	Weight W	X	Y	Z	WX	WY	WZ
		Kips	Ft.	Ft.	Ft.	Kip-Ft.	Kip-Ft.	Kip-Ft.
10	Port Fore & Aft Link	213.4	0	65.0	-3.0	0	13871.0	-640.2
11	STBD Fore & Aft Link	213.4	0	-65.0	-3.0	0	-13871.0	-640.2
12	Fwd Transverse Link	459.0	64.5	0	-3.0	29605.5	0	-1377.0
13	Aft Transverse Link	459.0	-64.5	0	-3.0	-29605.5	0	-1377.0
	Σ =	1795.9				-1315.8	0	-9012.9
	60'Wave Length	C.G.=	-0.7	0	-5.0			

APPENDIX L

DETERMINATION OF CLEARANCE BETWEEN LIGHTER AND SEPARATION TRUSS

Definition of Pertinent Factors

L = wave length, ft.

h = wave height, crest to trough, ft.

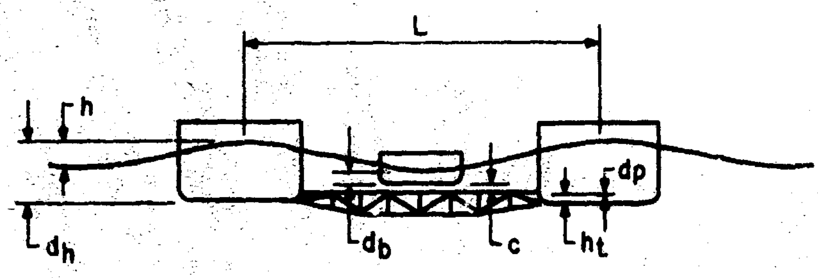
d_h = crane hull draft, ft.

d_b = lighter draft, ft.

h_t = truss height, ℄ to ℄ at hull attachment, ft.

d_p = truss pipe diameter at hull attachment, ft.

c = clearance between lighter and truss, ft.



Referring to the sketch above, the clearance is given by the equation

$$c = d_h - (h + d_b + h_t + 0.5d_p)$$

Now, d_h = 14 ft. and h_t = 2.5 ft.

$$c = 14 - (h + d_b + 2.5 + 0.5d_p) = 11.5 - s$$

where $s = h + d_b + 0.5d_p$

Considering the lighters LCM-8 and LARC-LX, the drafts are as follows:

LCM8-8: d_b = 5'-2" = 5.167'

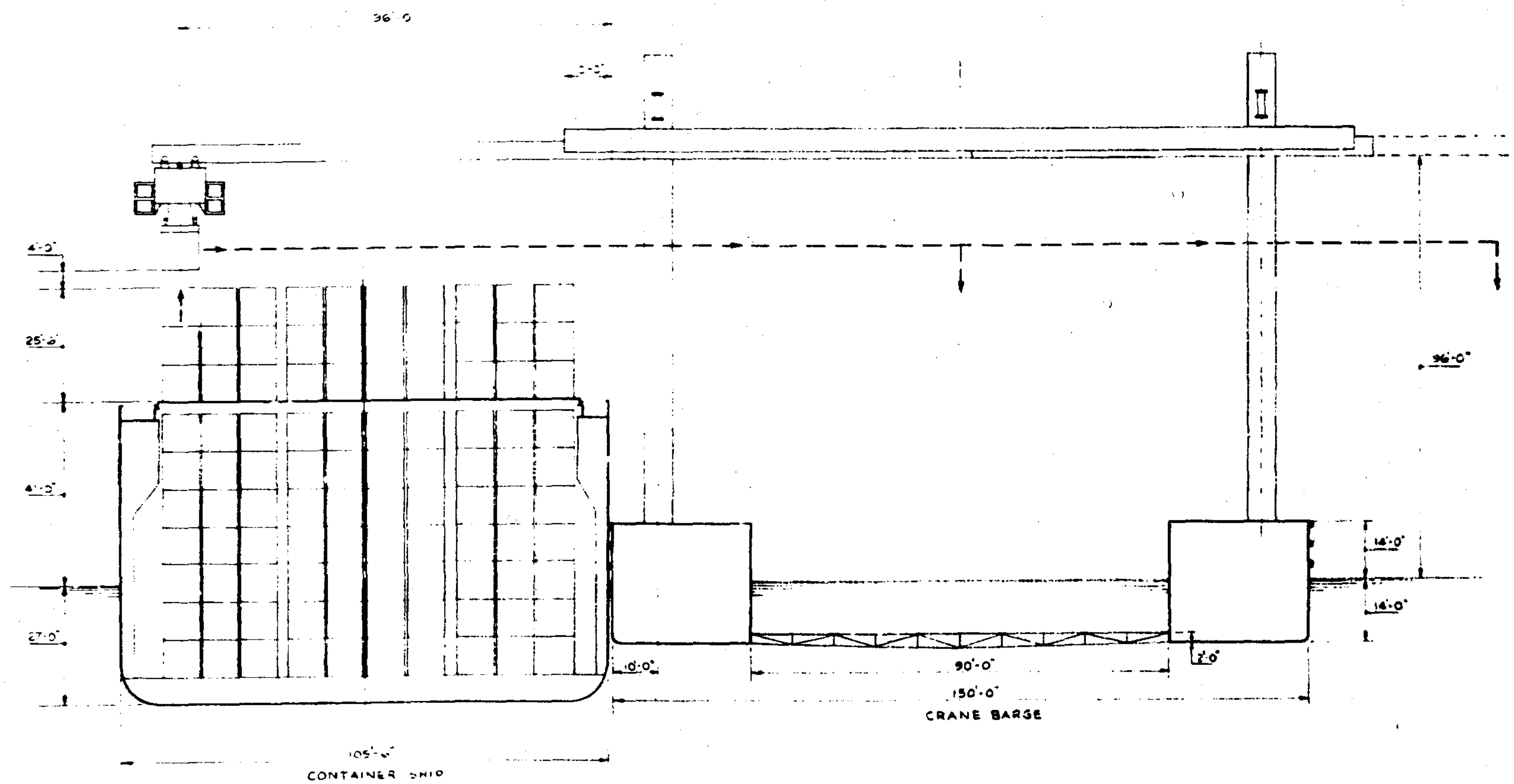
LARC-LX: d_b = 6'-7" = 6.583'

A tabular calculation of clearance for various wave lengths is presented on the following page.

APPENDIX L - DETERMINATION OF CLEARANCE BETWEEN LIGHTER AND SEPARATION TRUSS (con't)

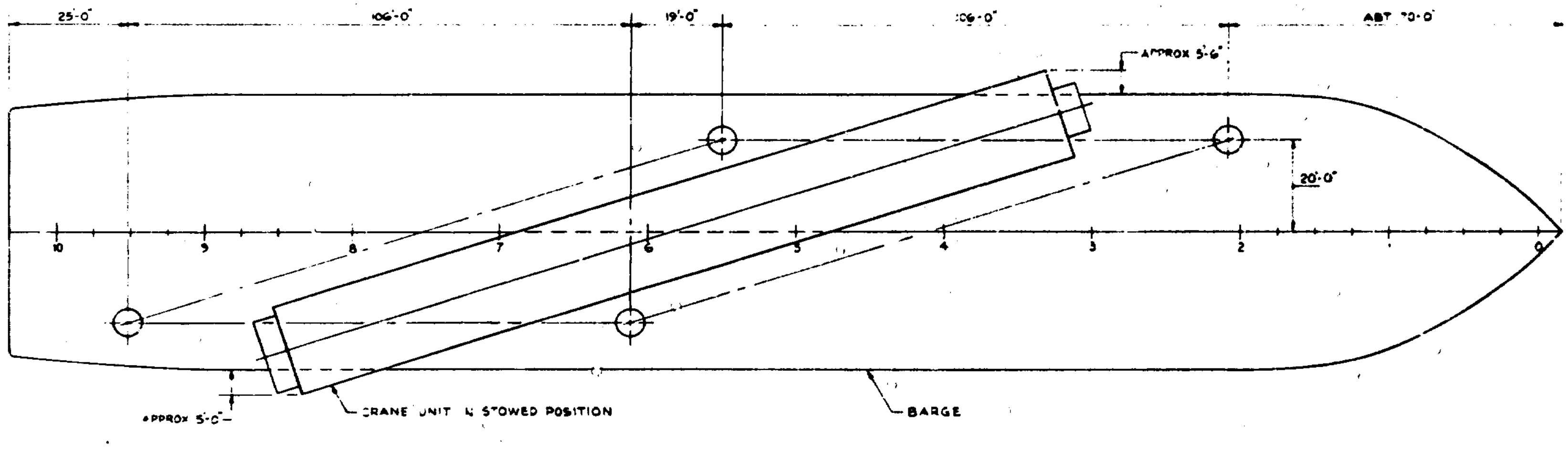
Wave Length	Wave Height	Pipe Radius	LCM-8			LARC-LX		
			Draft	Sum	Clearance	Draft	Sum	Clearance
L	h	$0.5d_p$	d_b	S	C	d_b	S	C
5.5	0.2	.167	5.167	5.534	5.966	6.583	6.950	4.550
25	1.3	.277		6.744	4.756		8.160	3.340
40	2.25	.360		7.777	3.723		9.193	2.307
50	2.9	.360		8.427	3.073		9.843	1.657
60	3.65	.533		9.350	2.150		10.766	.734
80	5.25	.584		11.001	.499		12.417	-.917
90	6.05	.584		11.801	-.301		13.217	-1.717
100	7.0	.584		12.751	-1.251		14.167	-2.667
110	7.85	.667		13.684	-2.184		15.100	-3.600
120	8.75	.667		14.584	-3.084		16.000	-4.500
140	10.65	.667		16.484	-4.984		17.900	-6.400

The positive values of clearance calculated above are plotted against wave height for the LCM-8 and the LARC-LX; negative values of clearance indicate interference between lighter and truss.



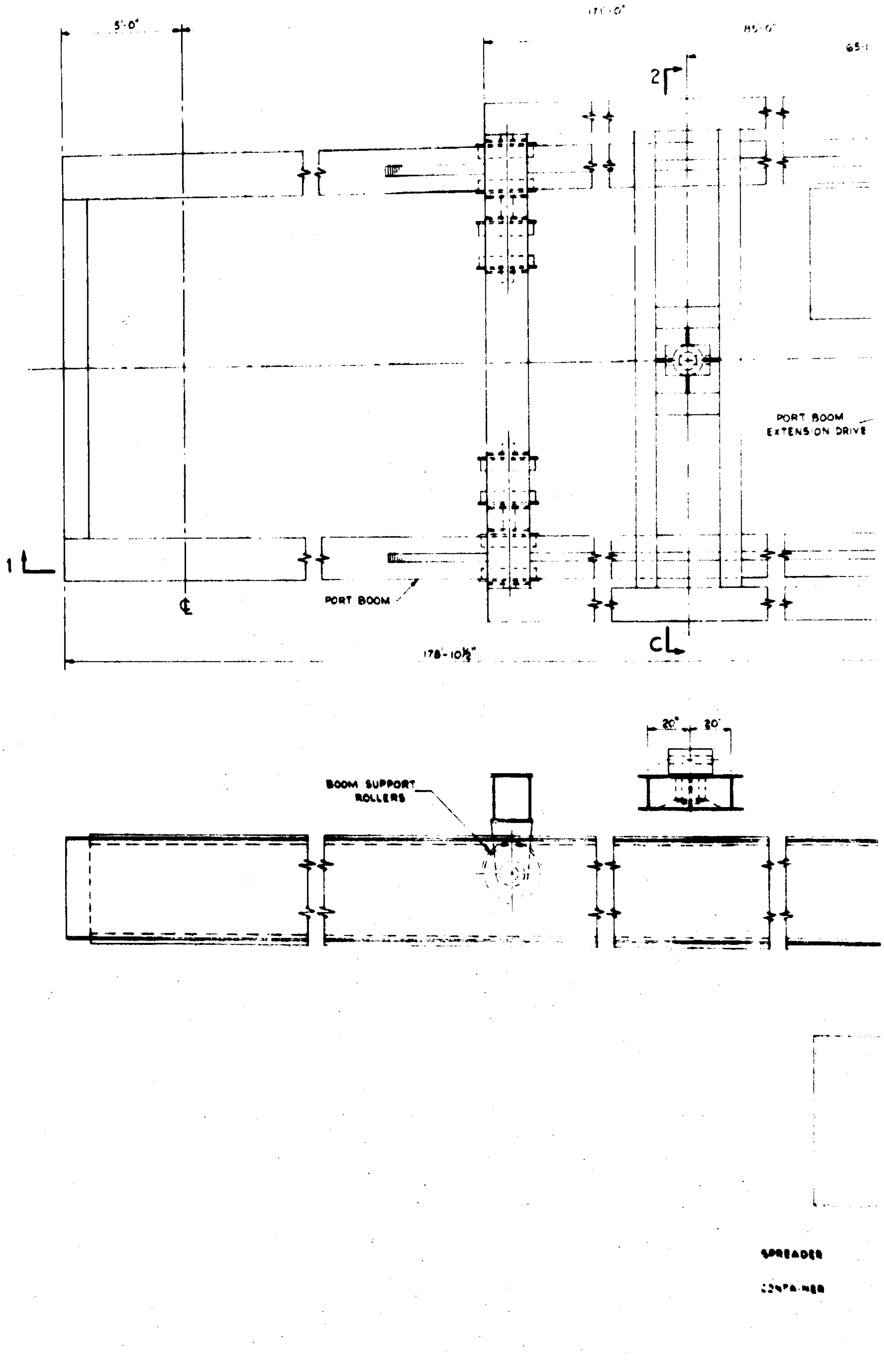
CONTAINER HANDLING PATH

FIGURE 71031-1. BARGE CRANE CONCEPT
CONTAINER HANDLING PATH; DESIGN NO. 1 AND 2
(SCALE 1/8" = 1'0"; DWG. NO. 71031-1)



PRELIMINARY ARRANGEMENT - BARGE CRANE

FIGURE 71031-2. BARGE CRANE CONCEPT
PRELIMINARY ARRANGEMENT, BARGE CRANE; DESIGN NO. 1 AND 2
(SCALE 1/16" = 1'0"; DWG. NO. 71031-2)


5'-0"
171'-0"
85'-0"
2
PORT BOOM
EXTENSION DRIVE
1
PORT BOOM
C
178'-10½"
20"
20"
BOOM SUPPORT
ROLLERS
SPREADER
CONTAINER

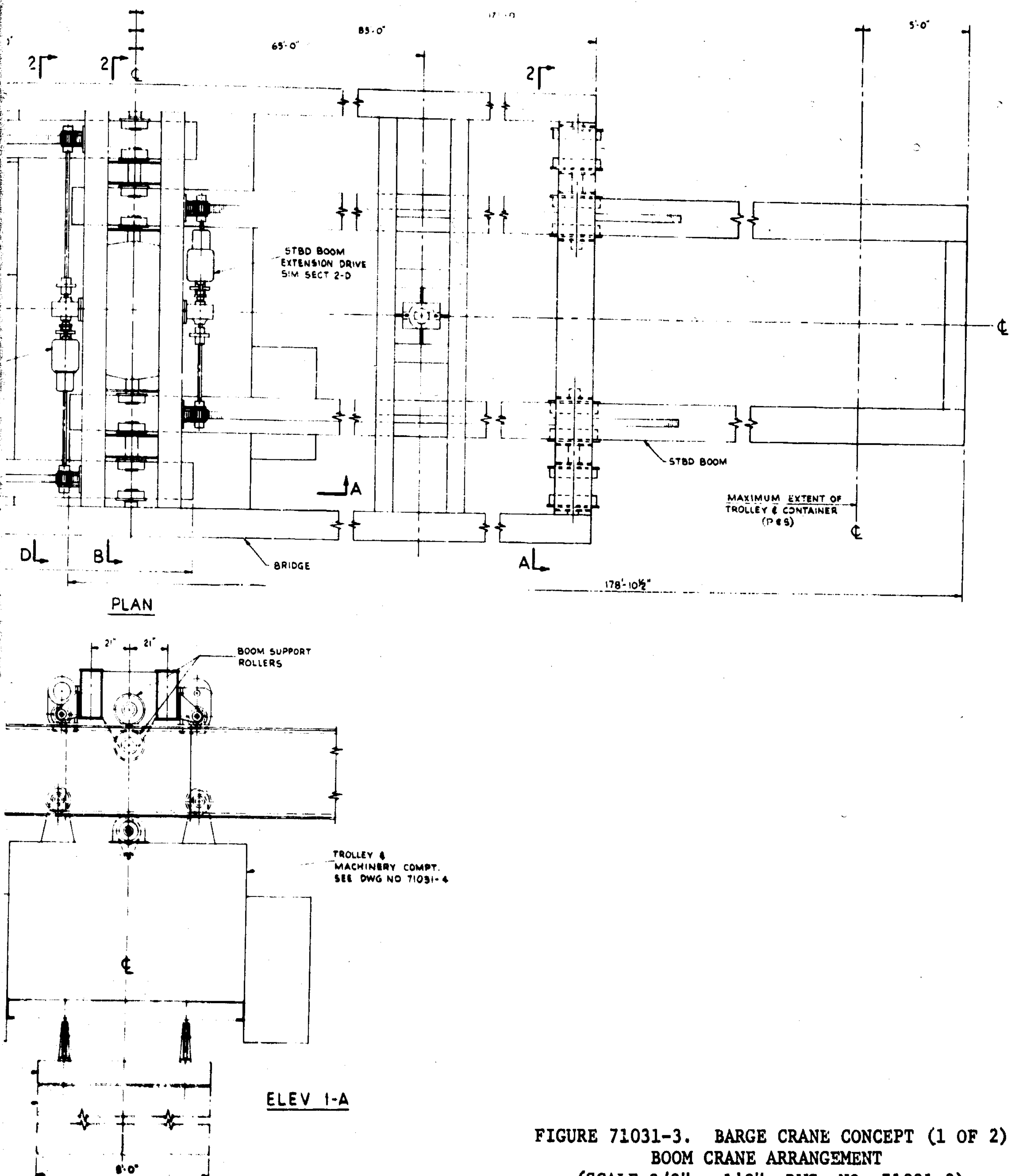
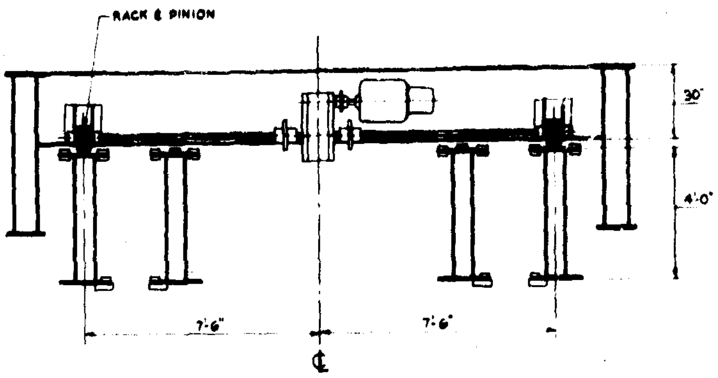
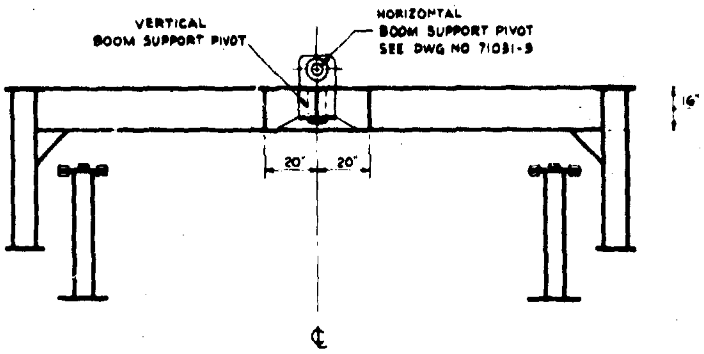


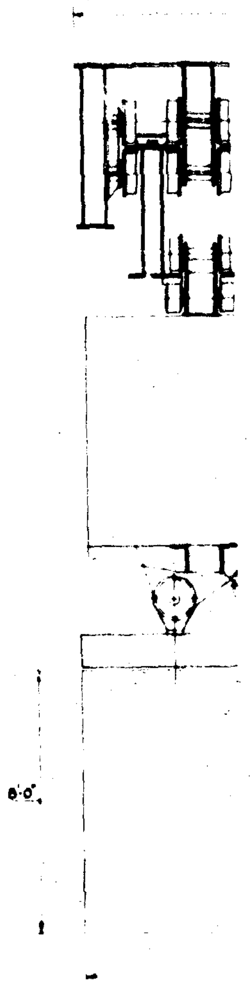
FIGURE 71031-3. BARGE CRANE CONCEPT (1 OF 2)
BOOM CRANE ARRANGEMENT
(SCALE 3/8" = 1'0"; DWG. NO. 71031-3)

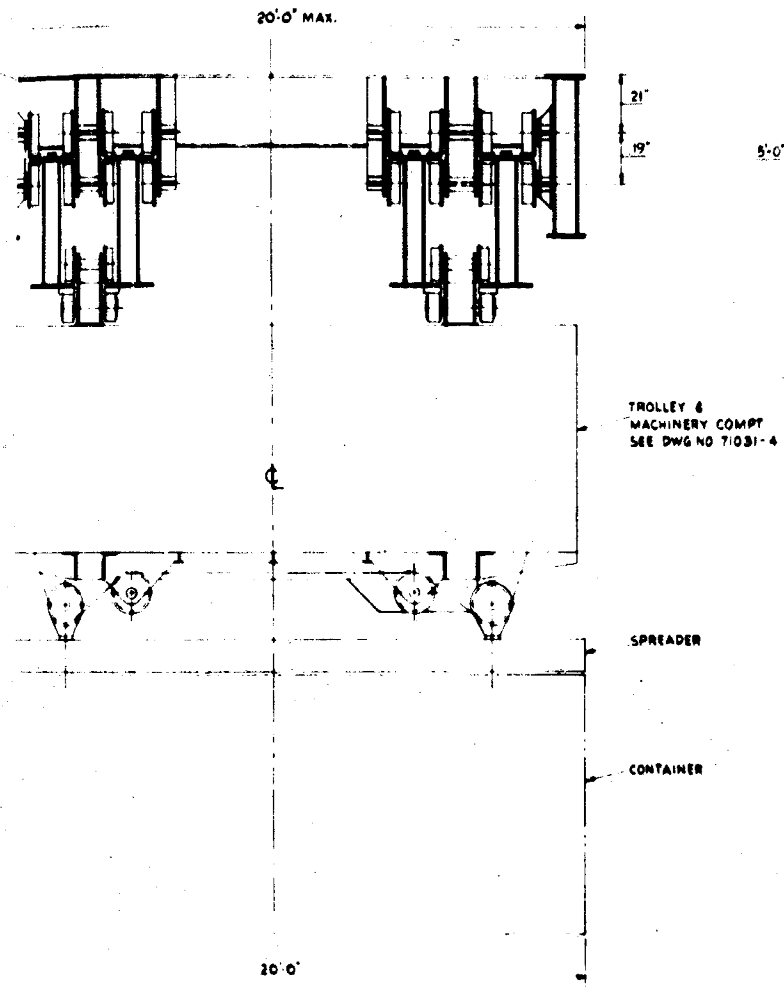


SECT 2-D

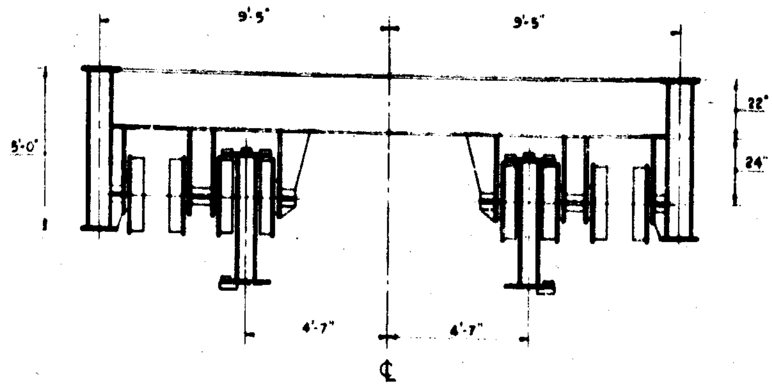


SECT 2-C





SECT 2-B



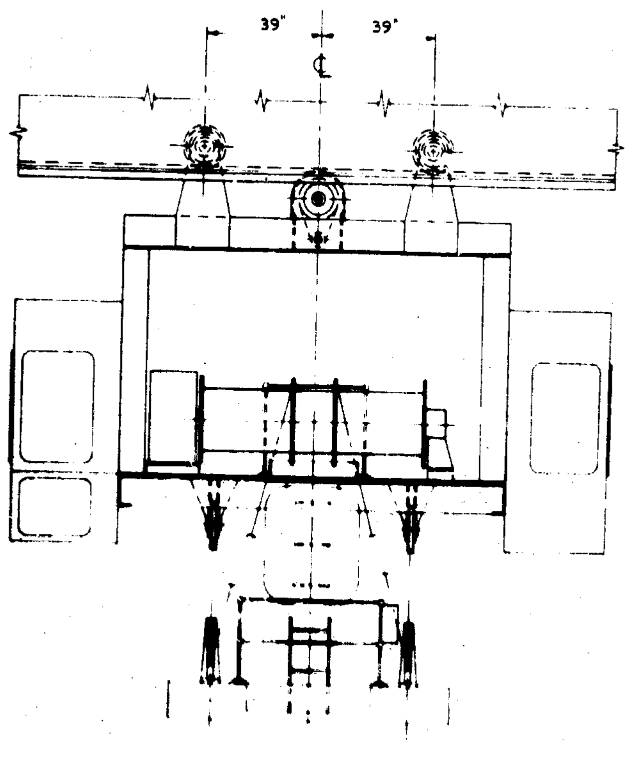
SECT 2-A

FIGURE 71031-3. BARGE CRANE CONCEPT (2 OF 2)
BOOM CRANE ARRANGEMENT
(SCALE 3/8" = 1'0"; DWG. NO. 71031-3)

GEAR REDUCER

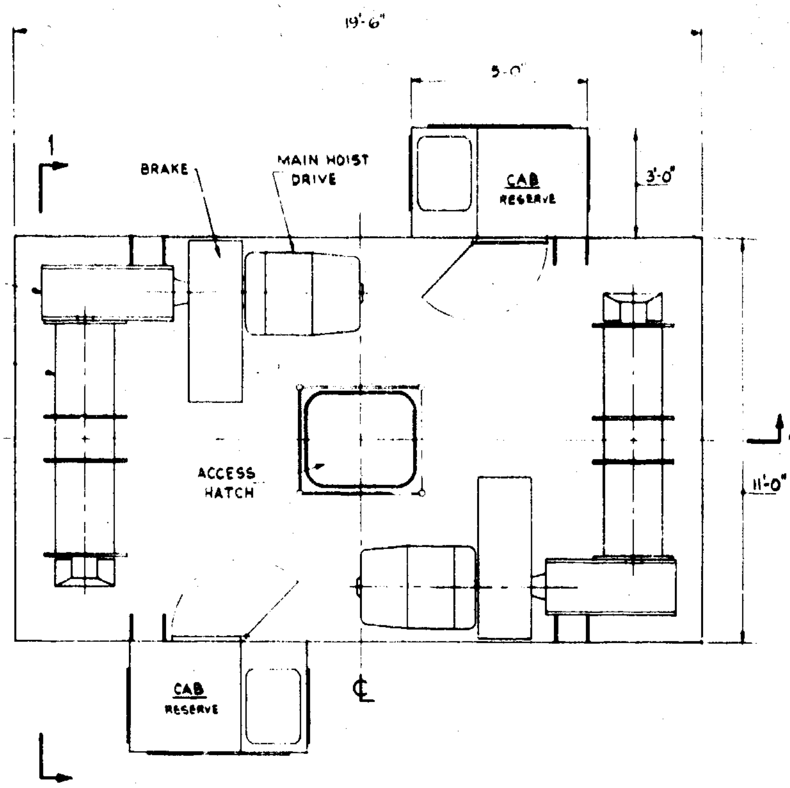
DRUM

1 ℄

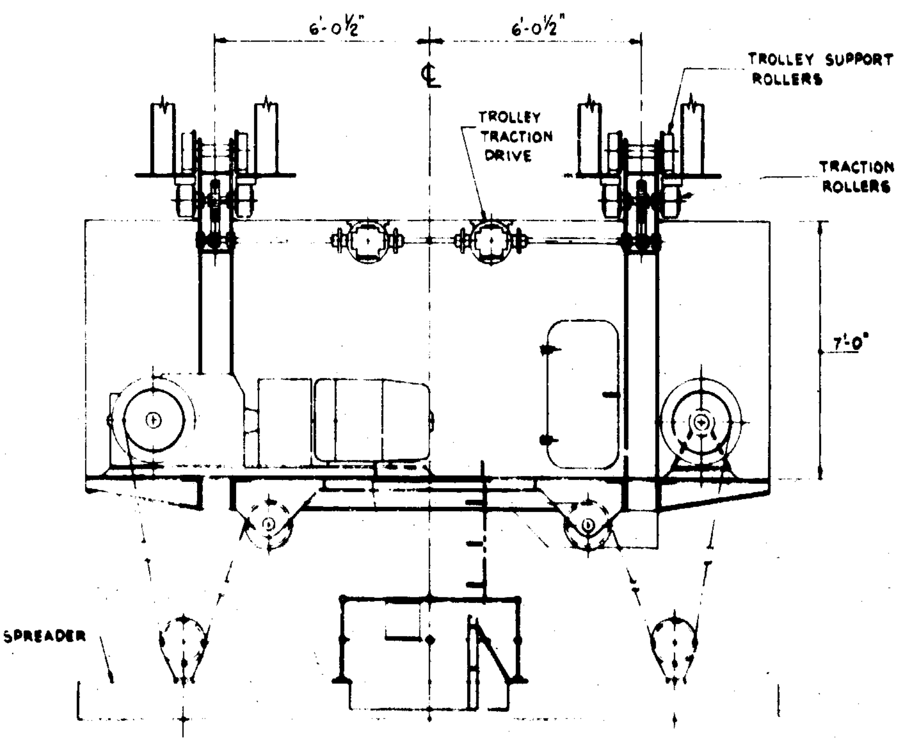


SECT 1-B

SF

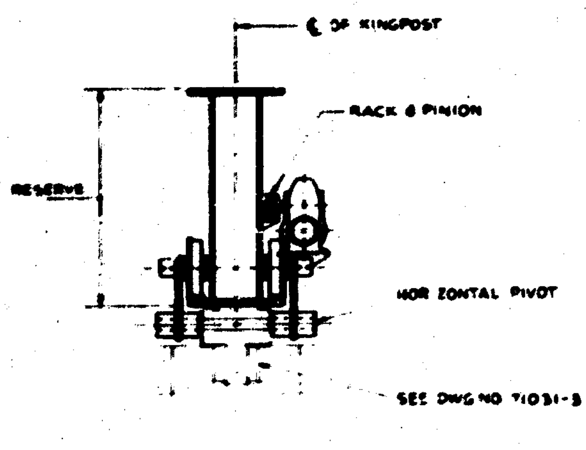


PLAN - MACH. COMPT.

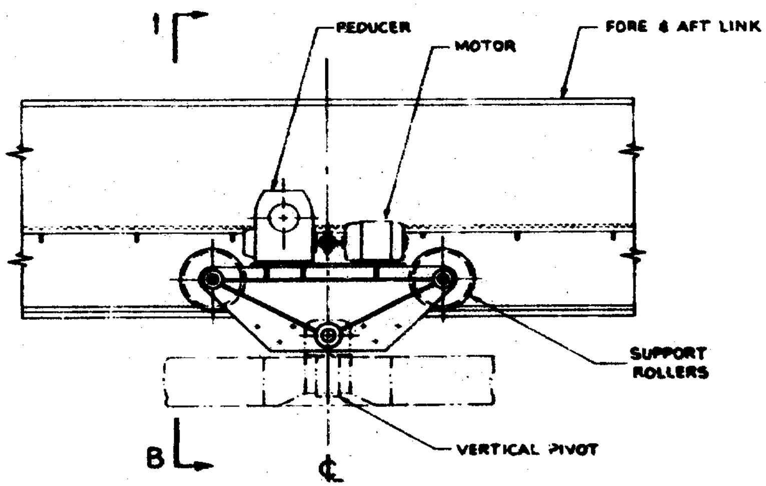


SECT 1-A

FIGURE 71031-4. BARGE CRANE CONCEPT BOOM CRANE ARRANGEMENT (SCALE 3/8" = 1'0"; DWG. NO. 7103-4)

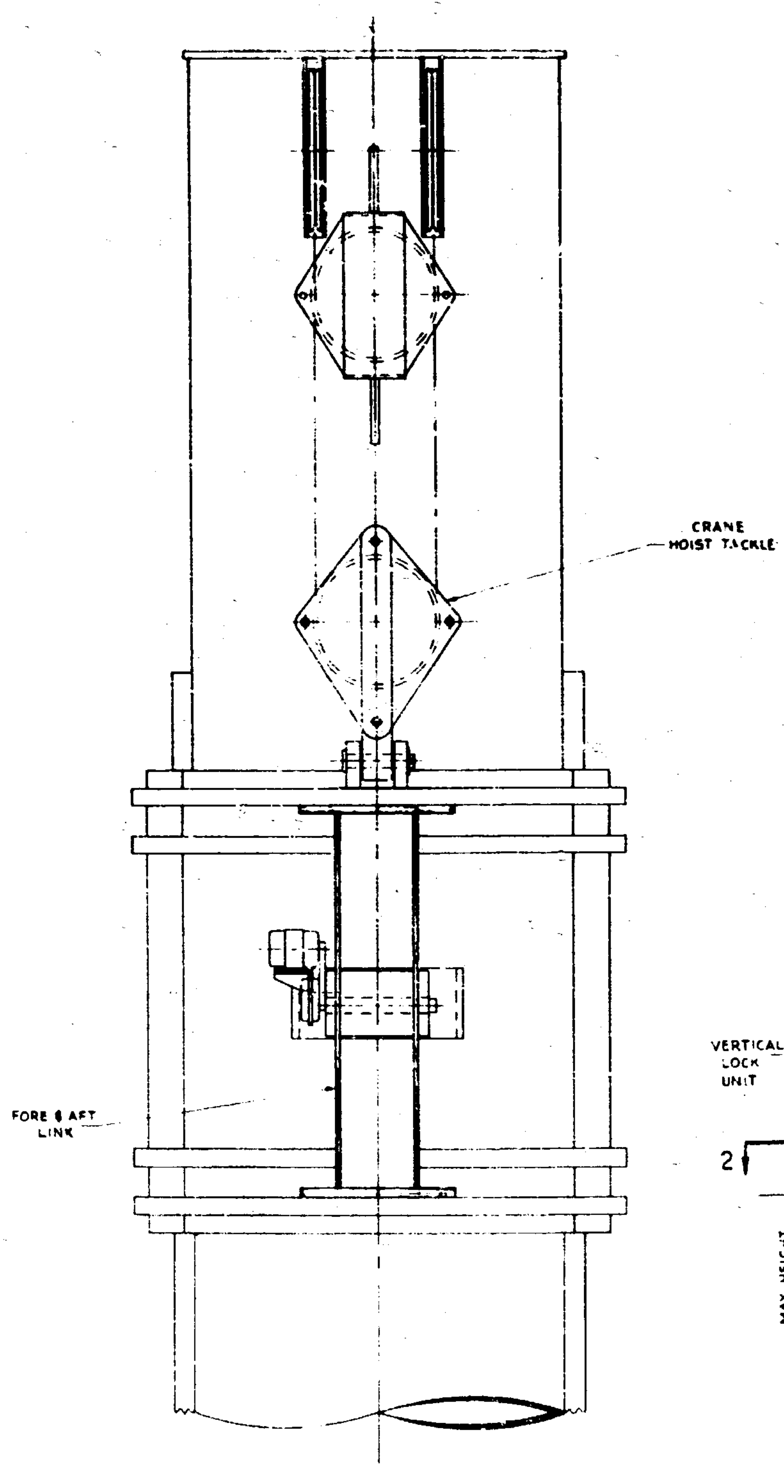


SECT 1-B

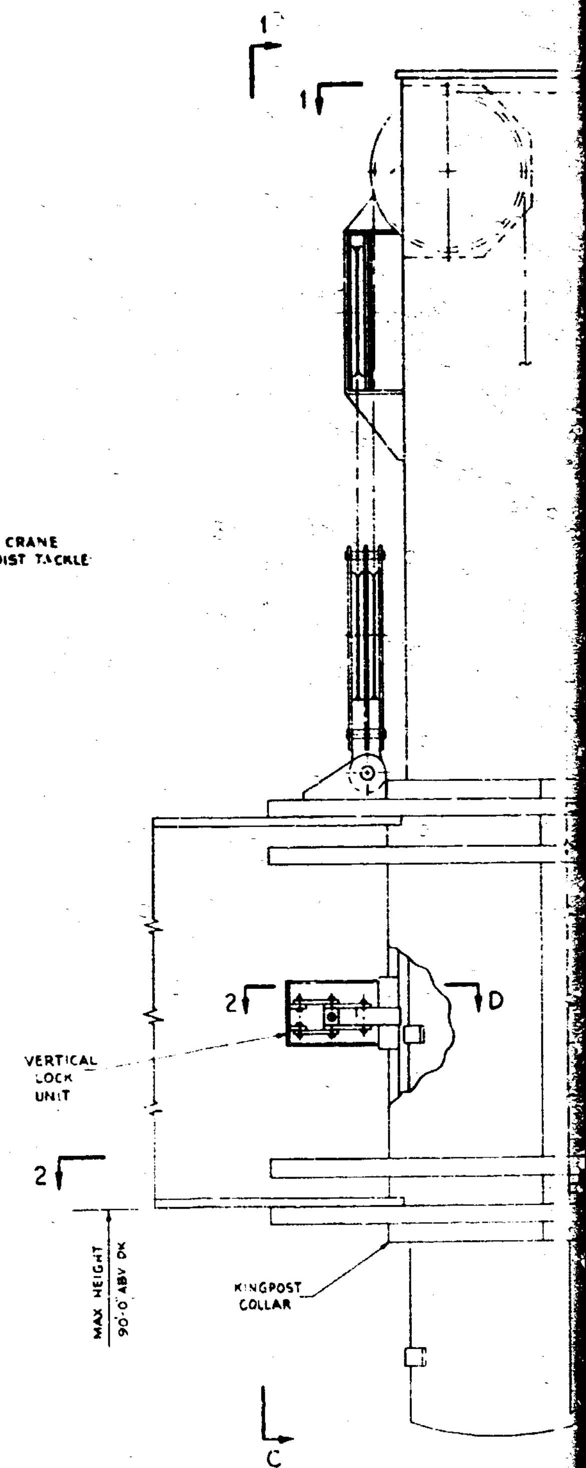


ELEV 1-A

FIGURE 71031-5. BARGE CRANE CONCEPT
BOOM CRANE ARRANGEMENT
(SCALE 3/8" = 1'0"; DWG. NO. 71031-5)



ELEV 1-C



EL

TOP

NOTE STBD

PORT

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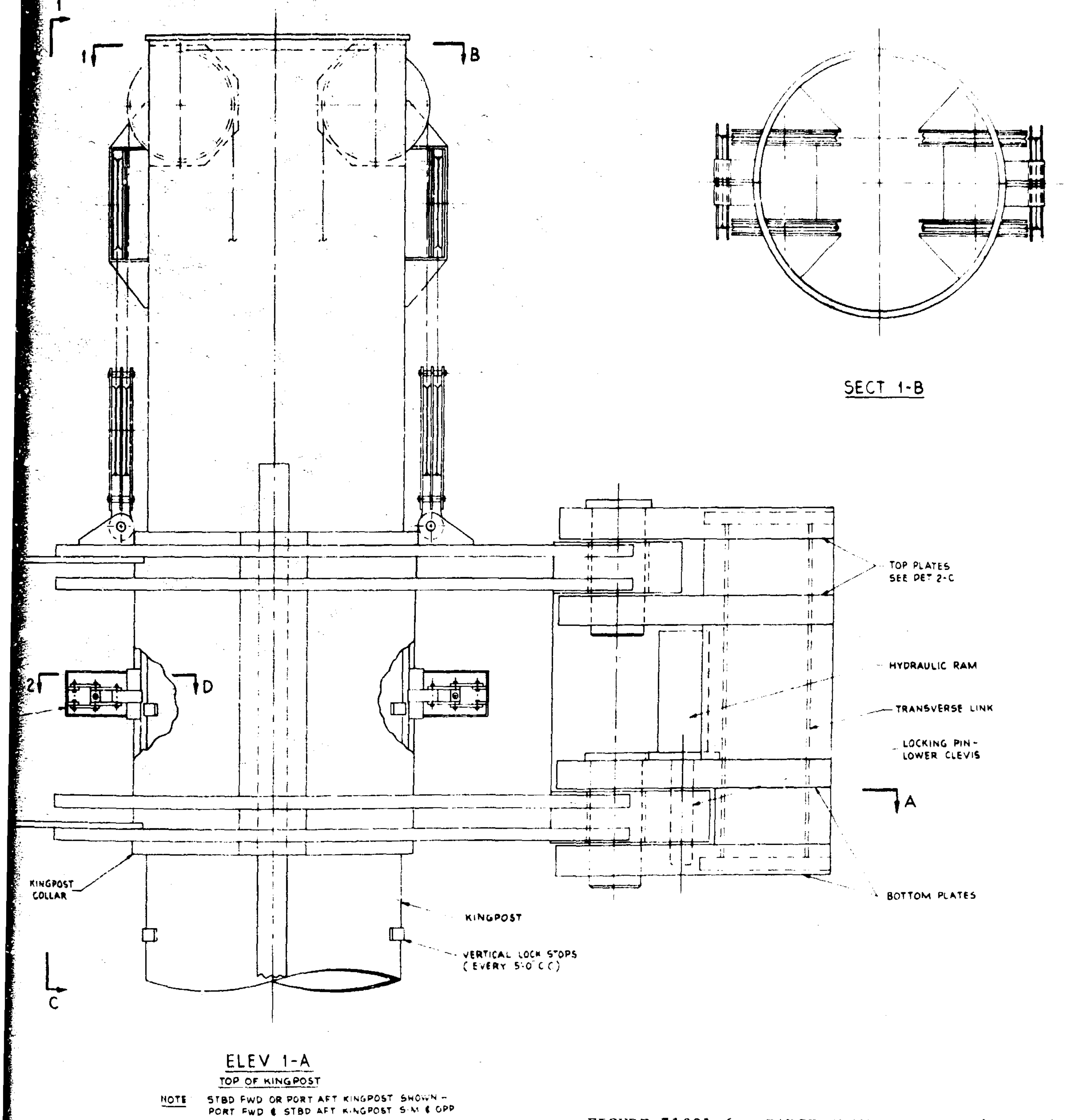
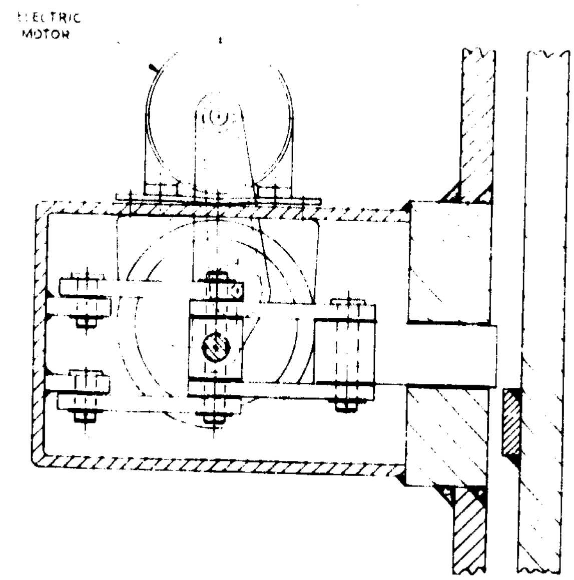
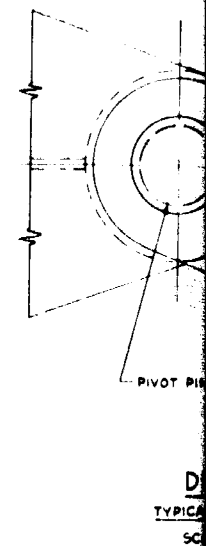


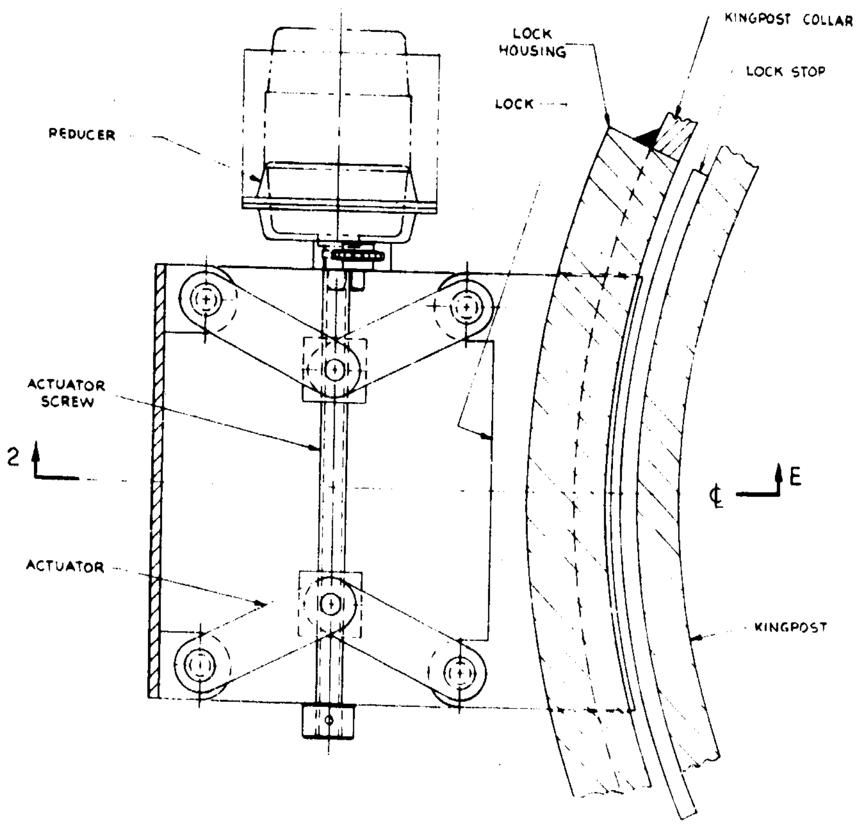
FIGURE 71031-6. BARGE CRANE CONCEPT (1 OF 2)
BOOM CRANE ARRANGEMENT
(SCALE 3/4" = 1'0"; DWG. NO. 71031-6)



SECT 2-E

SCALE- 3"=1'-0"

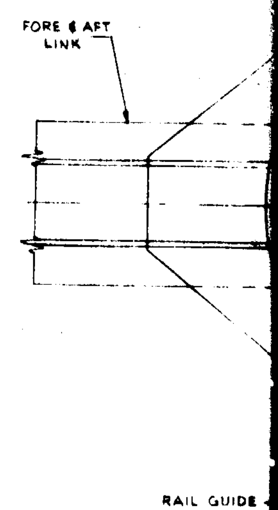




SECT 2-D

VERTICAL LOCK ACTUATORS

SCALE- 3"=1'-0"



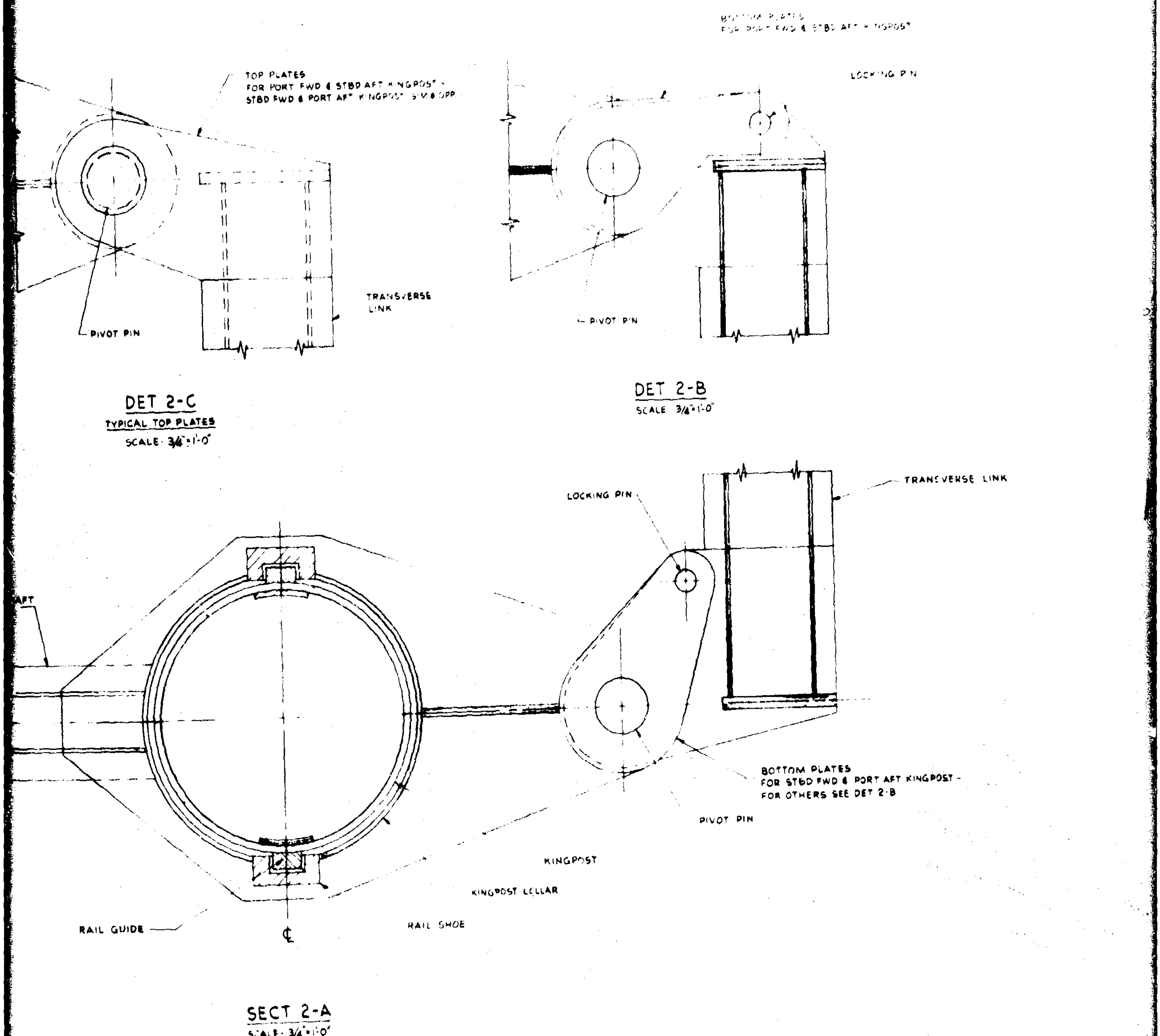
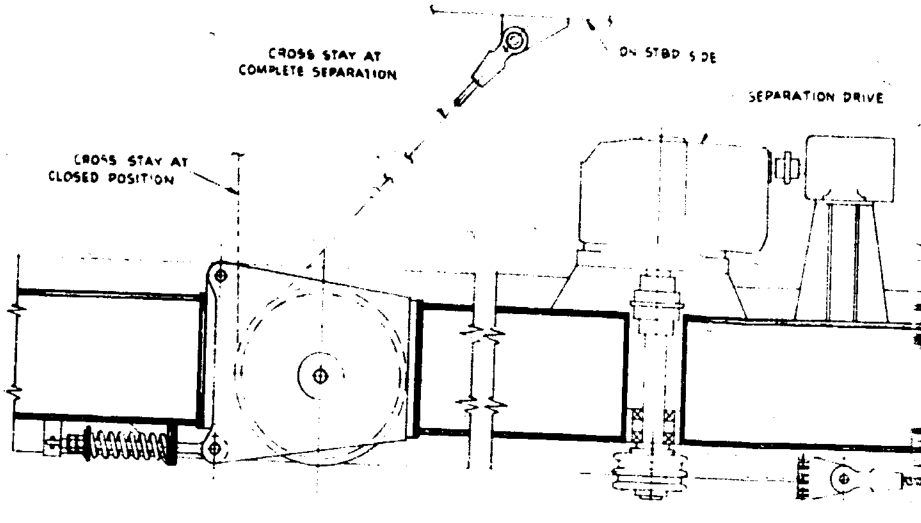
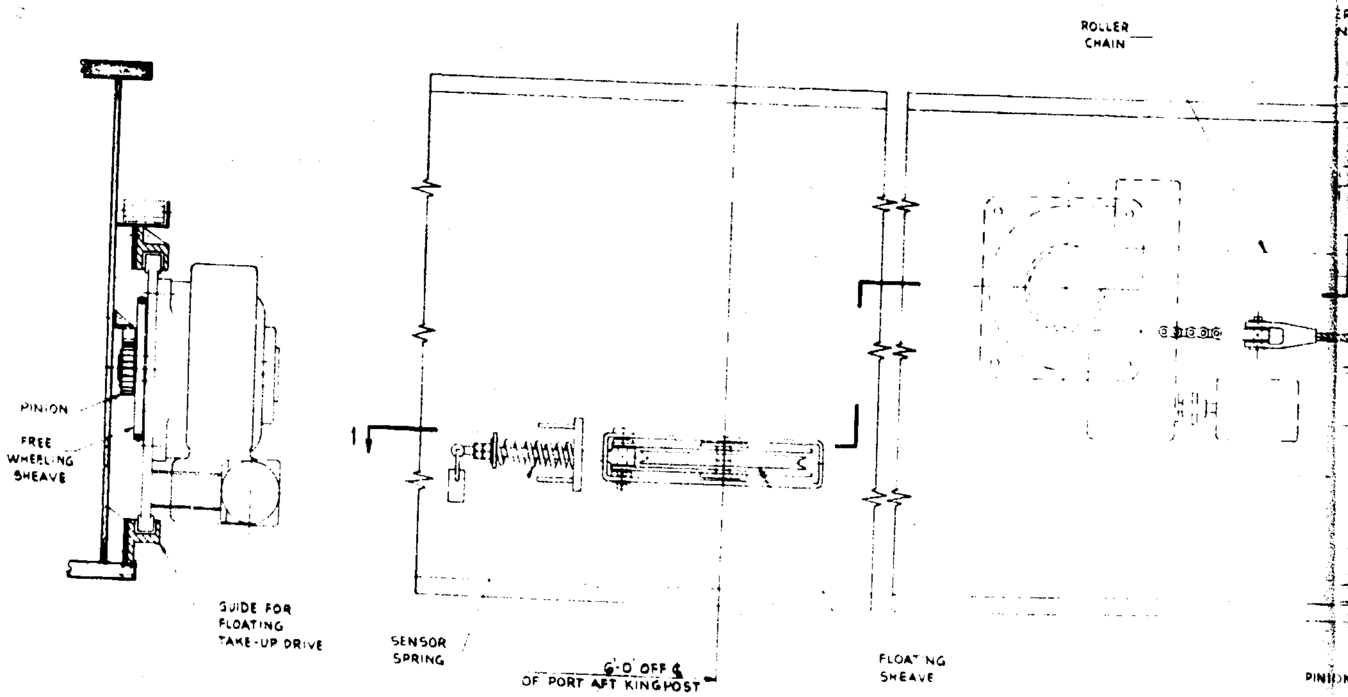
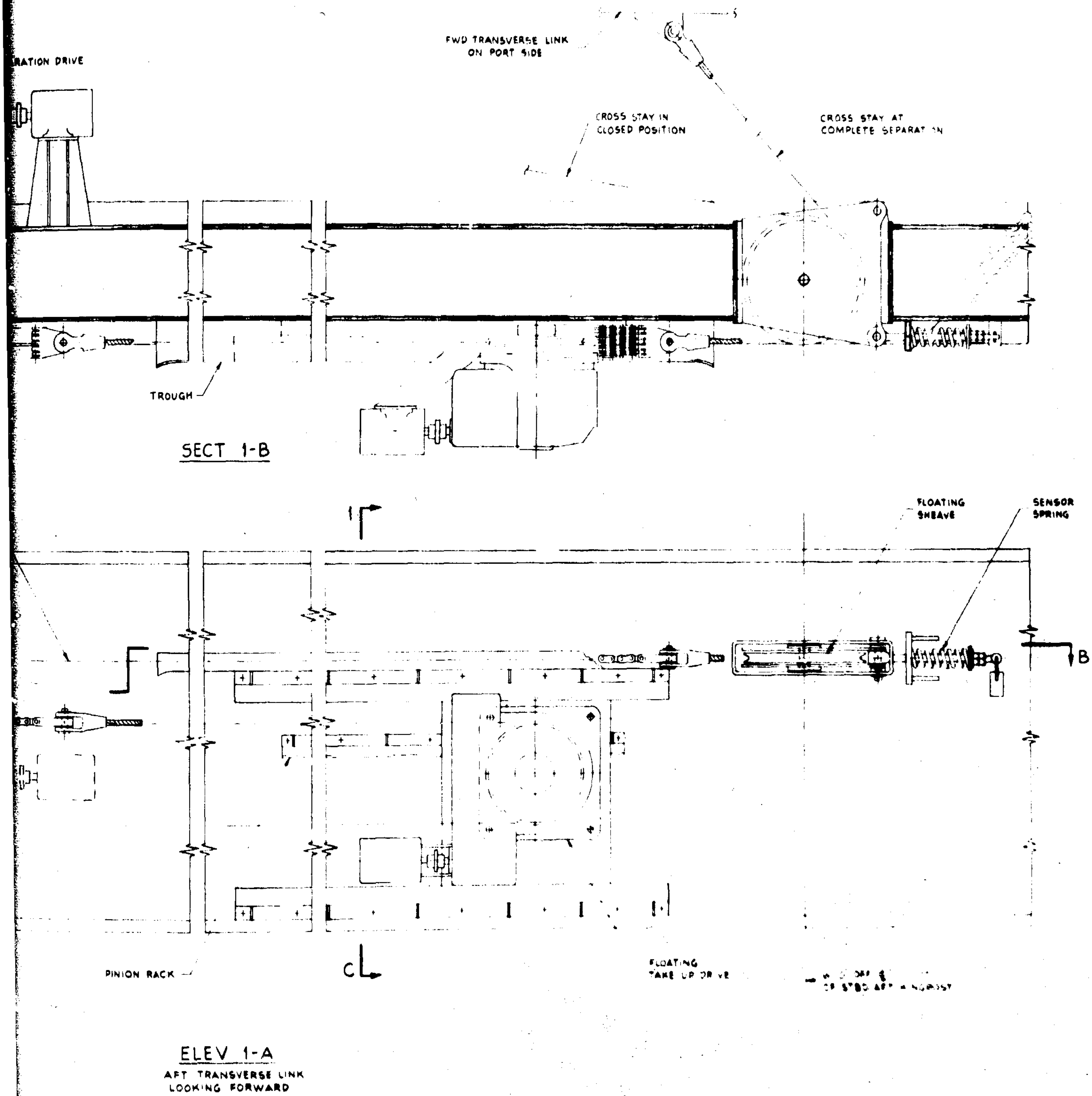


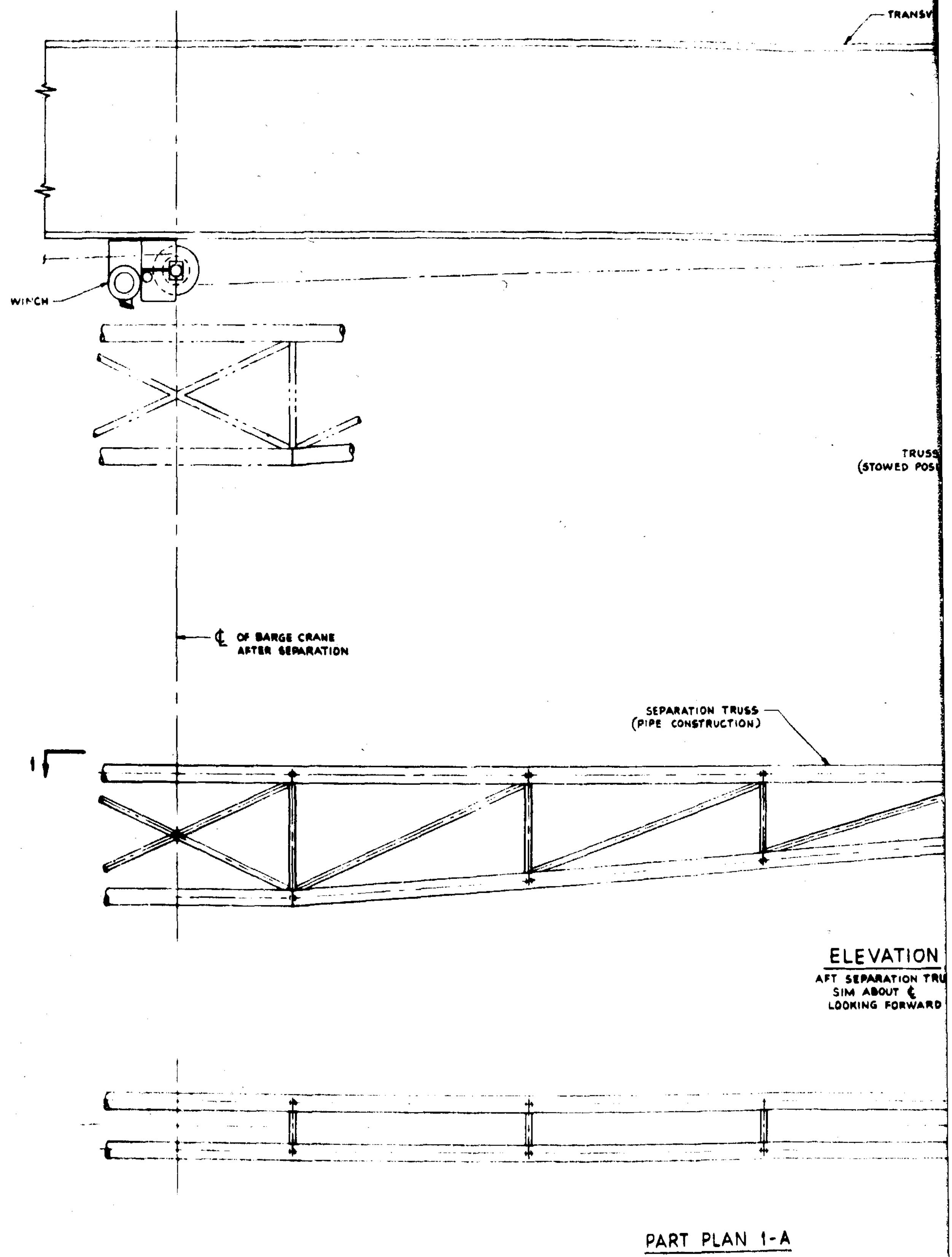
FIGURE 71031-6. BARGE CRANE DESIGN (2 OF 2)
BOOM CRANE ARRANGEMENT
(SCALE AS NOTED; DWG. NO. 71031-6)


CROSS STAY AT
COMPLETE SEPARATION
ON STBD SIDE
SEPARATION DRIVE
CROSS STAY AT
CLOSED POSITION


ROLLER
CHAIN
PINION
FREE
WHEELING
SHEAVE
GUIDE FOR
FLOATING
TAKE-UP DRIVE
SENSOR
SPRING
6'-0" OFF ℄
OF PORT AFT KINGPOST
FLOATING
SHEAVE

SECT 1-C


FWD TRANSVERSE LINK
ON PORT SIDE
CROSS STAY IN
CLOSED POSITION
CROSS STAY AT
TROUGH
SECT 1-B
FLOATING
SHEAVE
SENSOR
SPRING
B
PINION RACK
C
ELEV 1-A
AFT TRANSVERSE LINK
LOOKING FORWARD


TRANSV
WINCH
TRUSS
(STOWED POS
℄ OF BARGE CRANE AFTER SEPARATION
SEPARATION TRUSS (PIPE CONSTRUCTION)
1
ELEVATION
AFT SEPARATION TRU
SIM ABOUT ℄
LOOKING FORWARD
PART PLAN 1-A

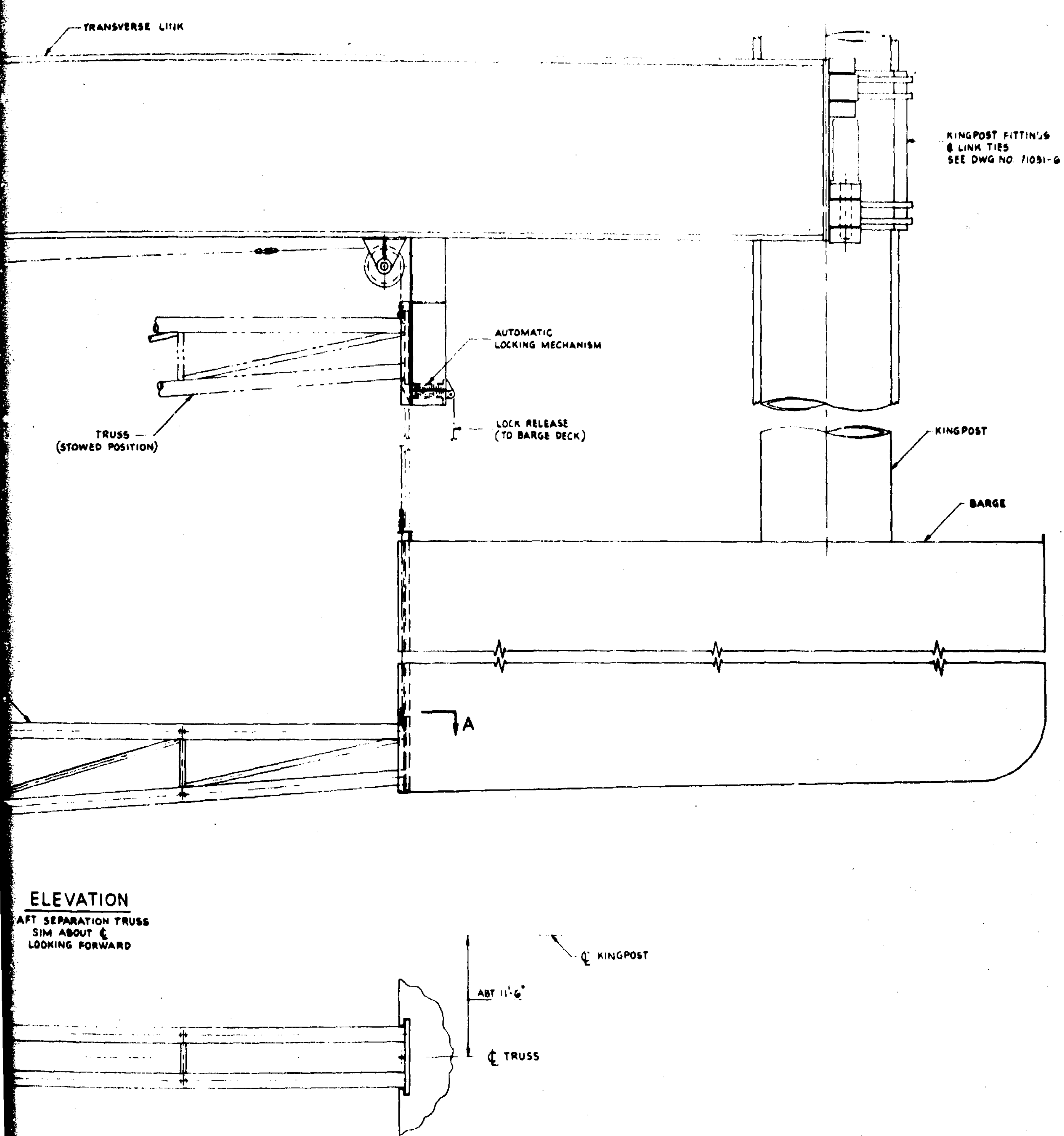
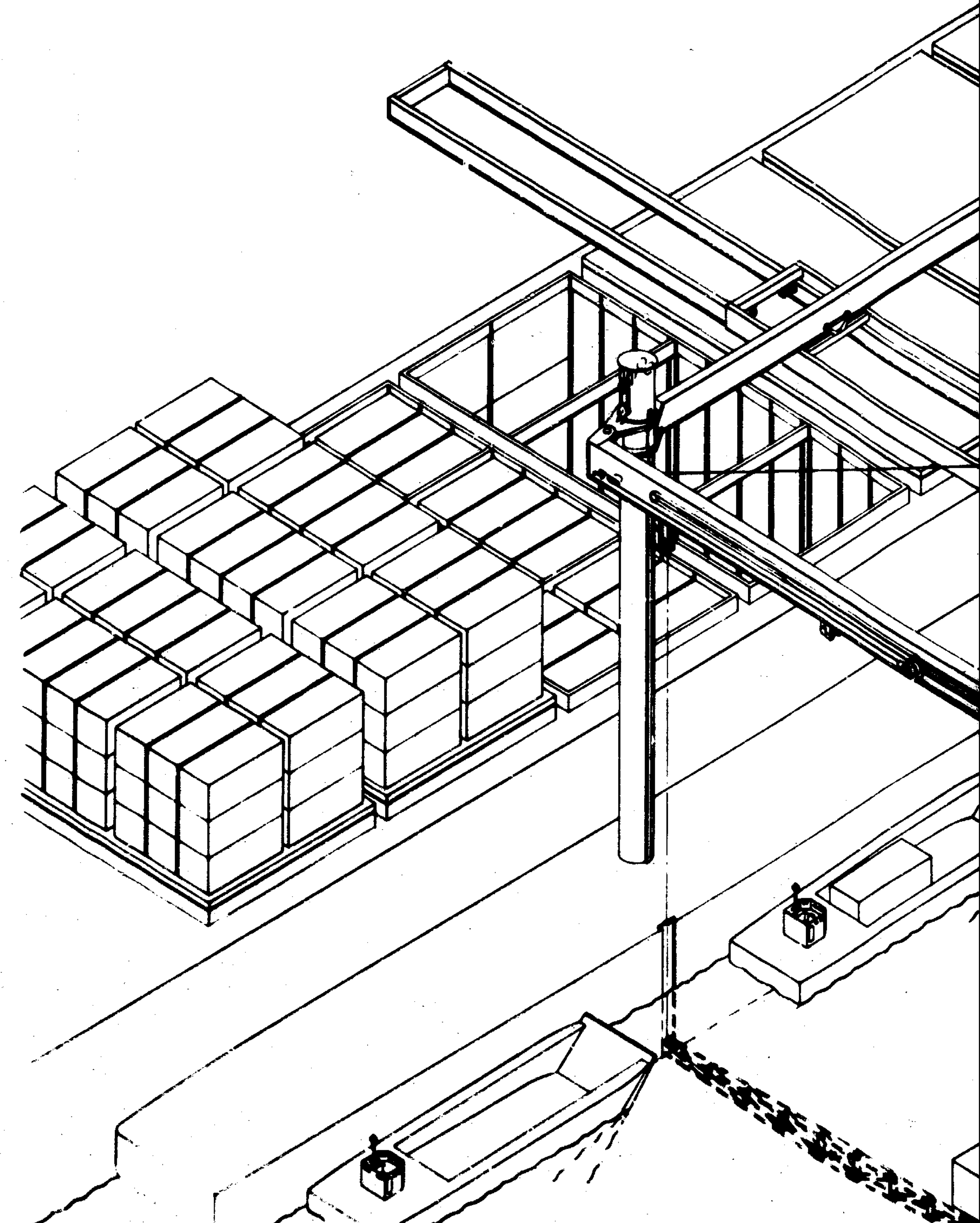


FIGURE 71031-8. BARGE CRANE DESIGN
BOOM CRANE ARRANGEMENT
(SCALE 3/8" = 1'0"; DWG. NO. 71031-8)



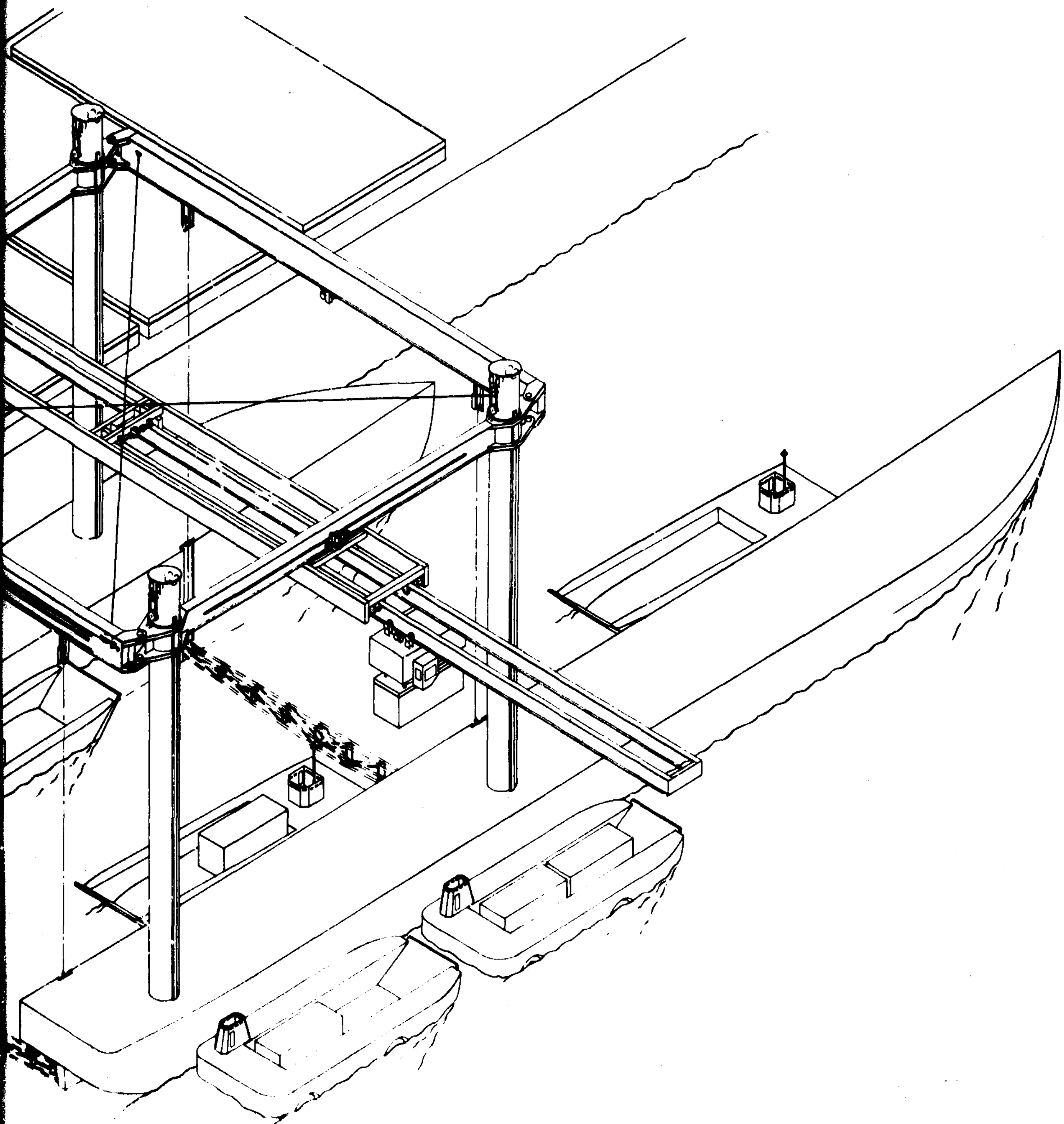
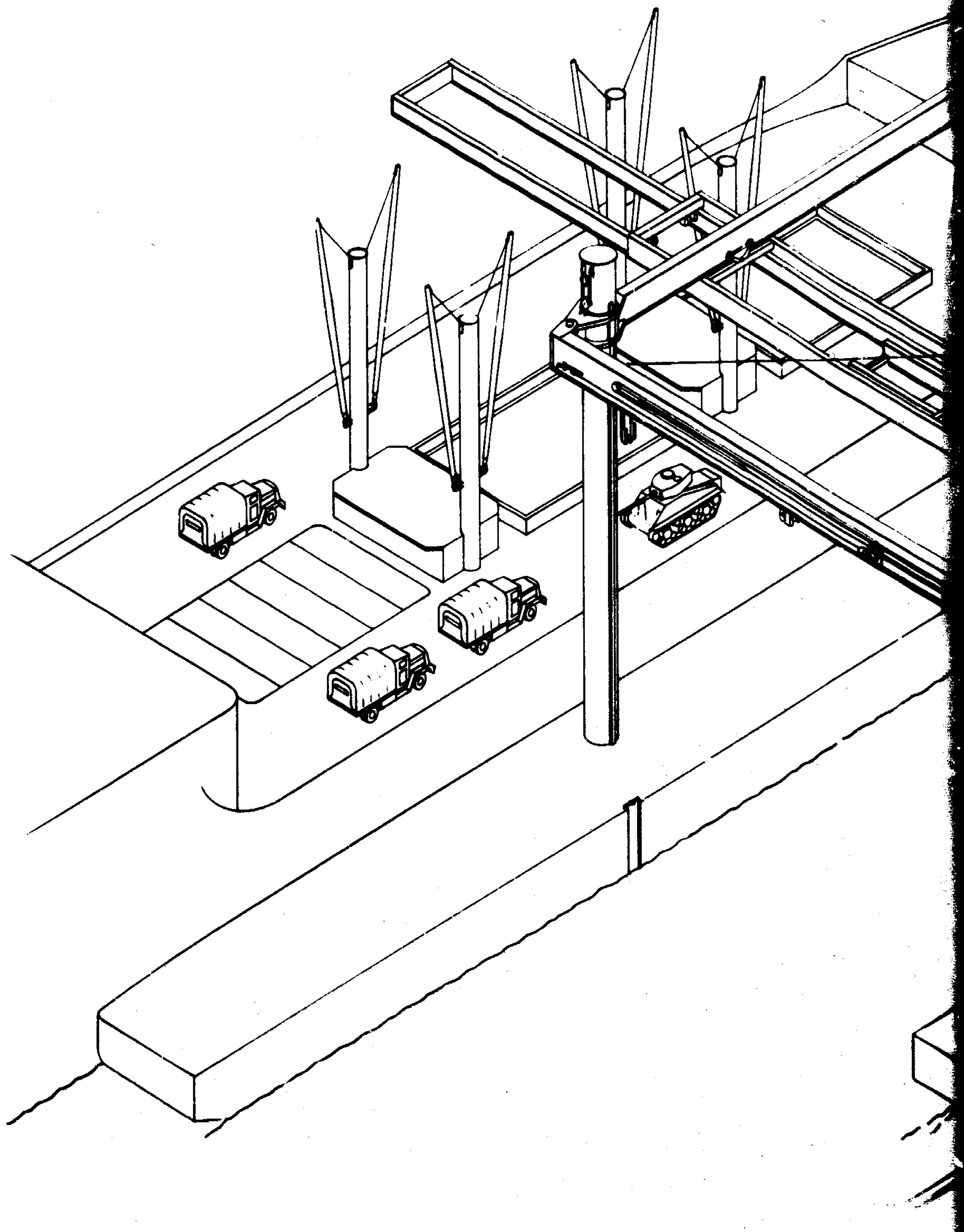


FIGURE 71031-9. BARGE CRANE DESIGN
BOOM CRANE ARRANGEMENT
(SCALE 1/16" = 1'0"; DWG. NO. 71031-9)



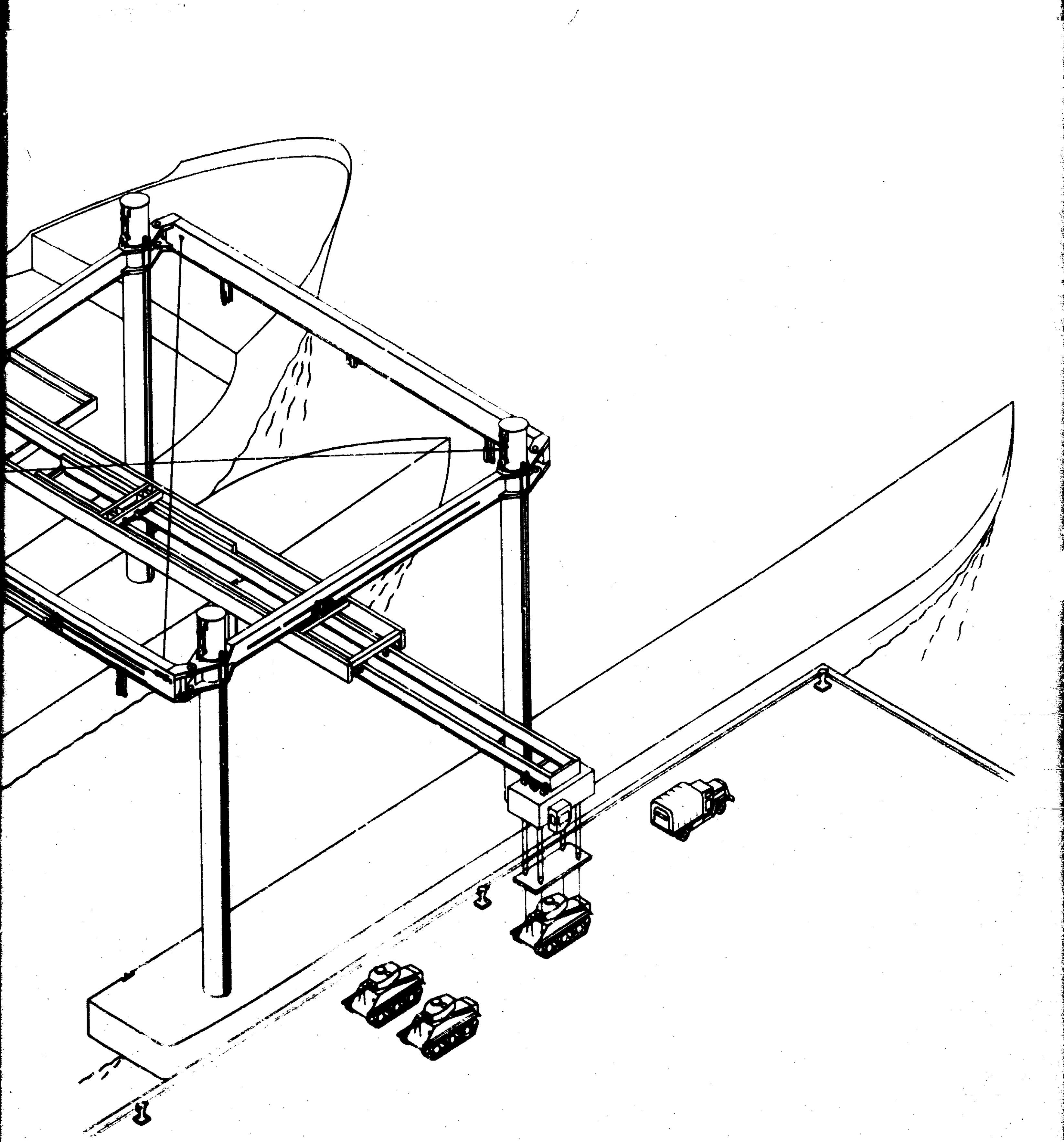
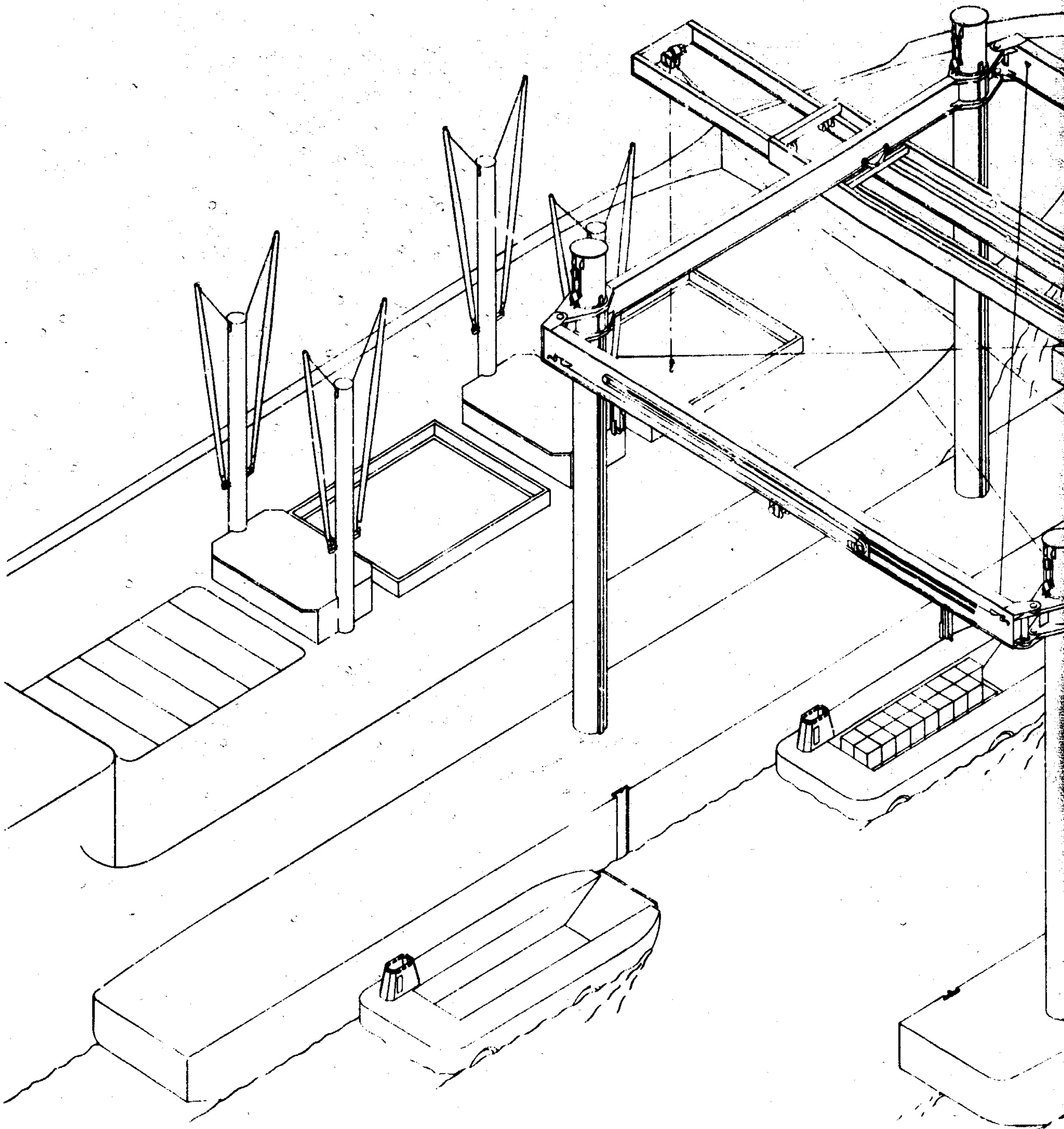


FIGURE 71031-10. BARGE CRANE DESIGN
BOOM CRANE ARRANGEMENT
(SCALE 1/16" = 1'0"; DWG. NO. 71031-10)



A

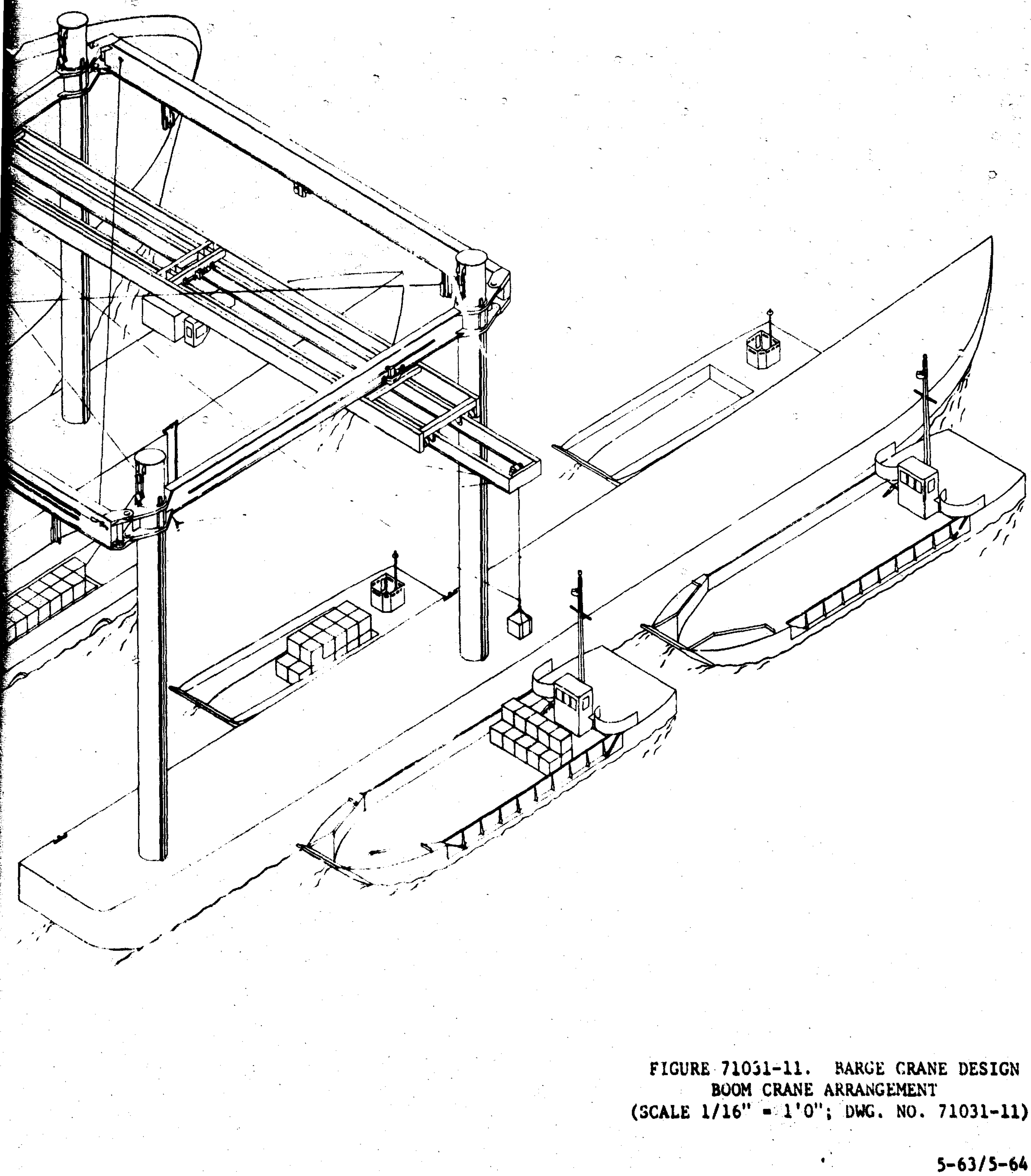
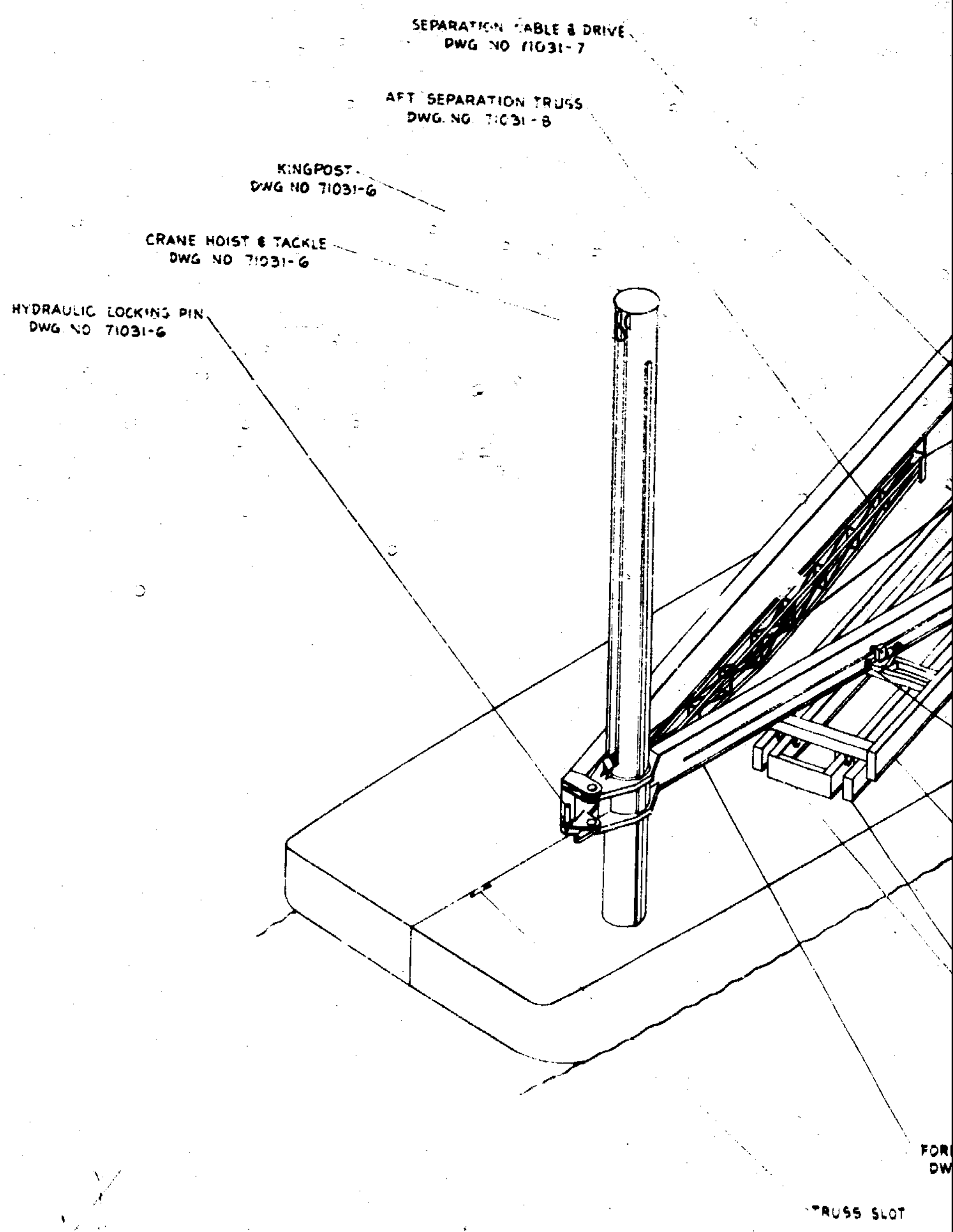


FIGURE 71031-11. BARGE CRANE DESIGN
BOOM CRANE ARRANGEMENT
(SCALE 1/16" = 1'0"; DWG. NO. 71031-11)


SEPARATION CABLE & DRIVE
DWG NO 71031-7
AFT SEPARATION TRUSS
DWG. NO 71031-8
KINGPOST
DWG NO 71031-6
CRANE HOIST & TACKLE
DWG NO 71031-6
HYDRAULIC LOCKING PIN
DWG NO 71031-6
TRUSS SLOT

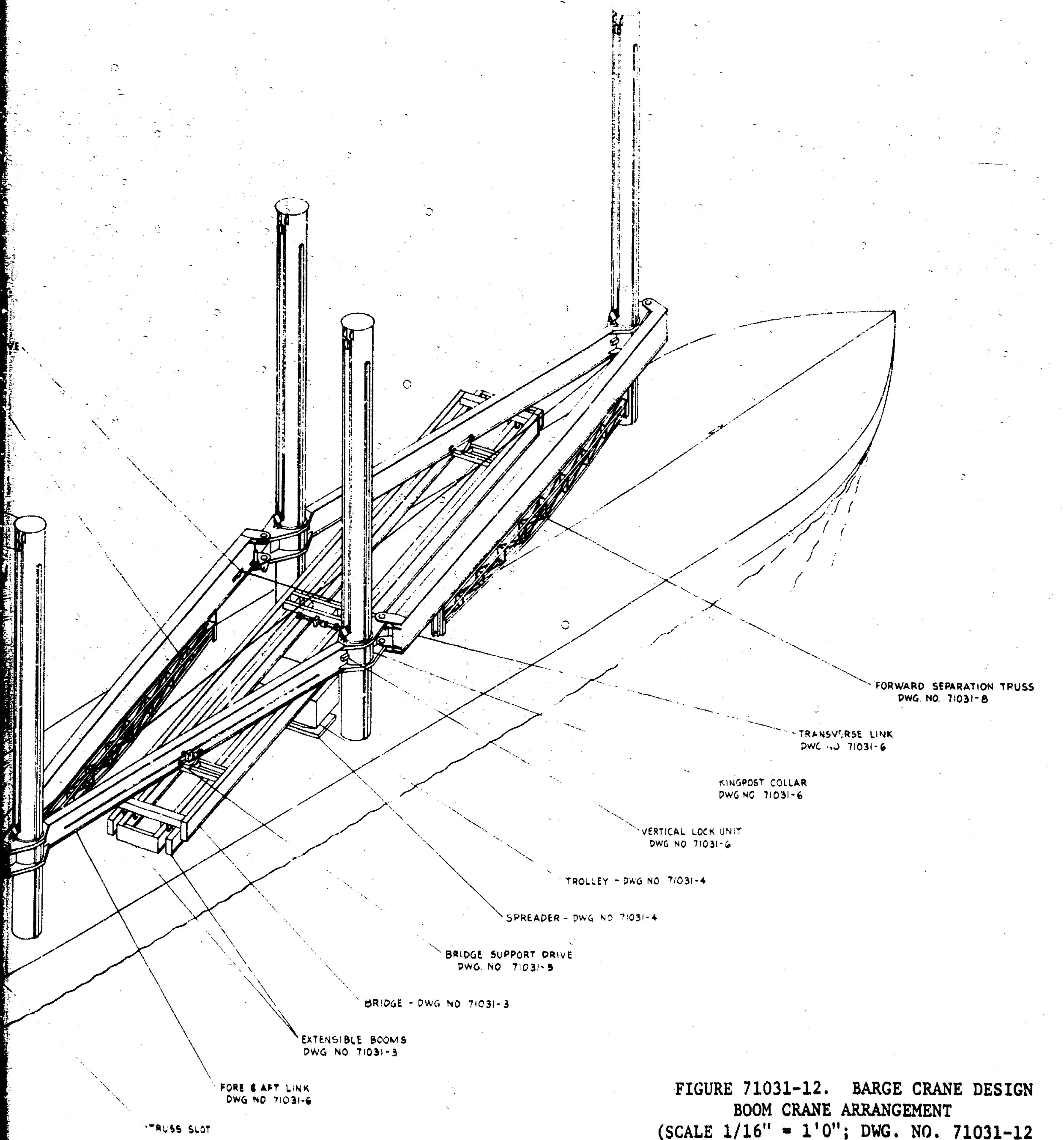
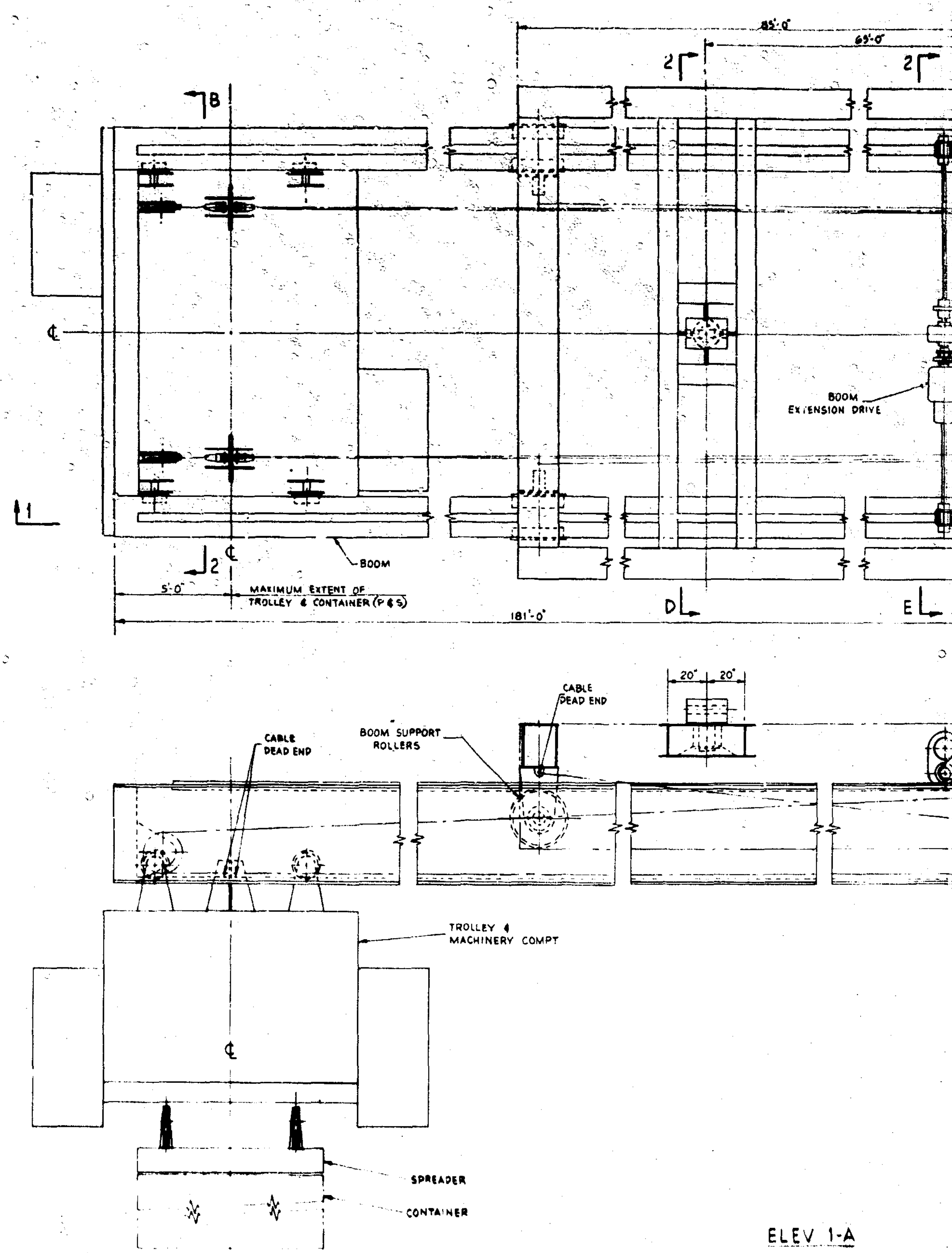


FIGURE 71031-12. BARGE CRANE DESIGN
BOOM CRANE ARRANGEMENT
(SCALE 1/16" = 1'0"; DWG. NO. 71031-12


85'-0"
65'-0"
2
2
B
℄
1
2
BOOM
BOOM EXTENSION DRIVE
5'-0"
MAXIMUM EXTENT OF TROLLEY & CONTAINER (P&S)
181'-0"
D
E
20"
20"
CABLE DEAD END
CABLE DEAD END
BOOM SUPPORT ROLLERS
TROLLEY & MACHINERY COMPT
SPREADER
CONTAINER
ELEV 1-A

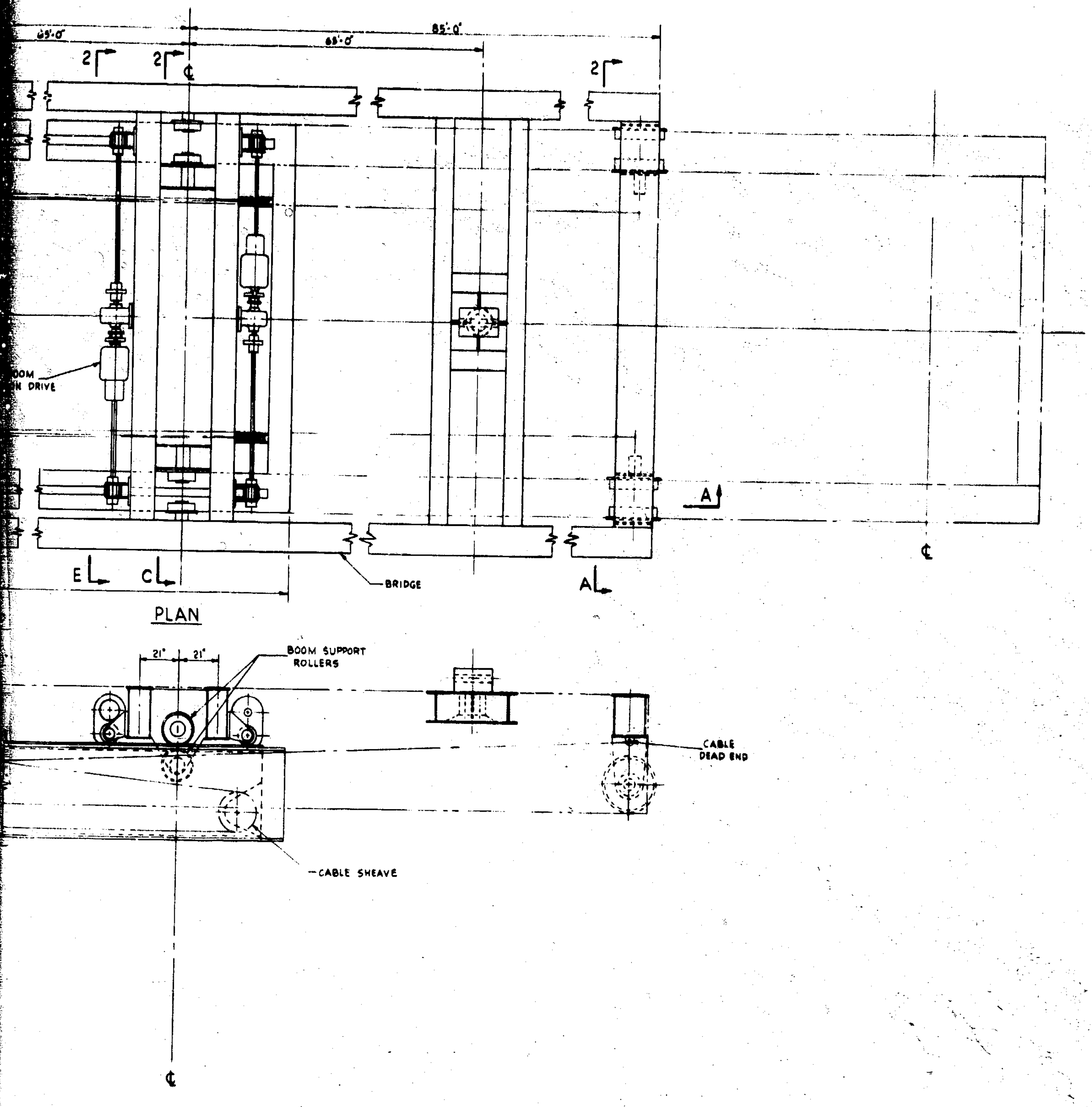
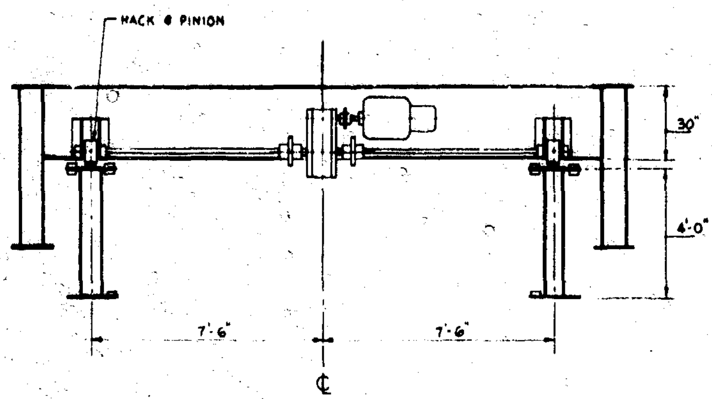
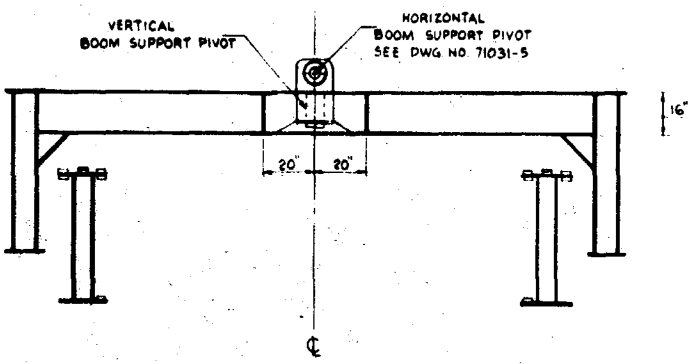


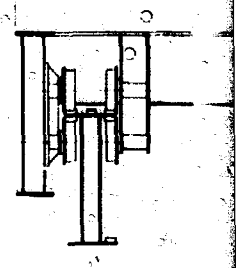
FIGURE 71031-13. BARGE CRANE DESIGN (1 OF 2)
BOOM CRANE ARRANGEMENT
(SCALE 1/16" = 1'0"; DWG. NO. 71031-13)

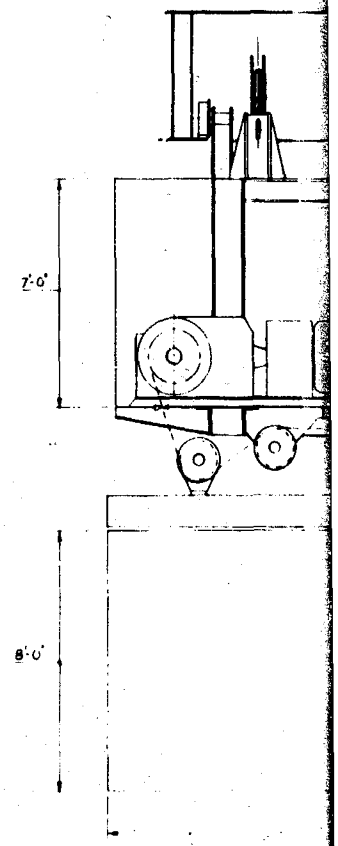


SECT 2-E



SECT 2-D





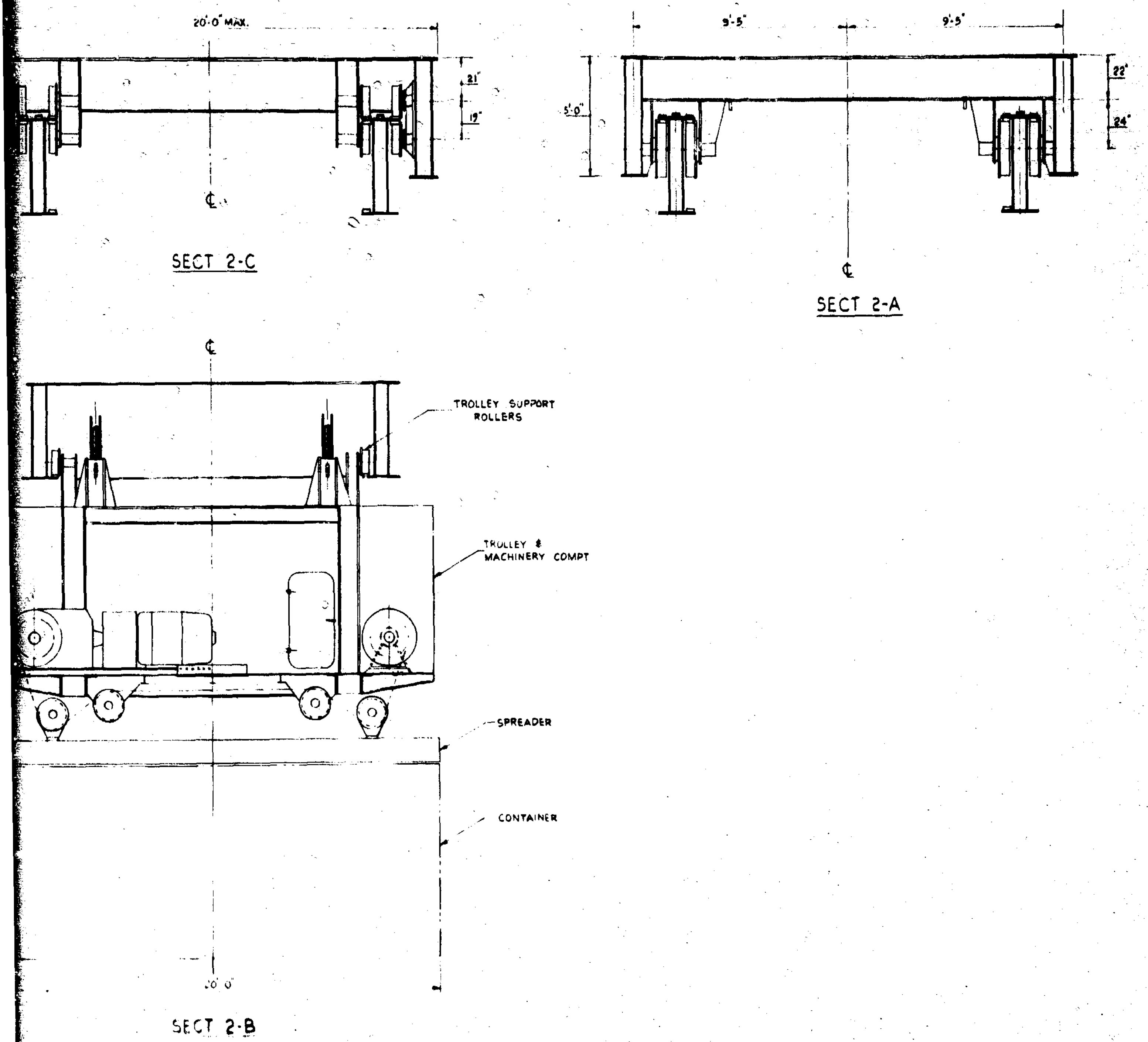


FIGURE 71031-13. BARGE CRANE DESIGN (2 of 2)
BOOM CRANE ARRANGEMENT
(SCALE 1/16" = 1'0"; DWG. NO. 71031-13)