SSC-205

STRUCTURAL DESIGN REVIEW OF LONG, CYLINDRICAL, LIQUID-FILLED INDEPENDENT CARGO TANK BARGES

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1970

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ADDRESS CORRESPONDENCE TO:

SECRETARY
SHIP STRUCTURE COMMITTEE
U.S. COAST GUARD HEADQUARTERS
WASHINGTON, D.C. 20591

1970

Dear Sir:

The possibility of transporting liquid chemical cargoes in large tank barges on the open sea has necessitated an assessment of the state of the art in barge-tank design, to determine what further theoretical and experimental development is required.

Herewith is a final project report containing the review and recommendations of the study.

Sincerely,

J. B. McCarty, Jr.
Capt., U.S. Coast Guard
Acting Chairman, Ship Structure Committee

SSC-205

Final Report

on

Project SR-184, "Chemical Tank-Barge Design"

to the

Ship Structure Committee

STRUCTURAL DESIGN REVIEW OF LONG, CYLINDRICAL, LIQUID-FILLED INDEPENDENT CARGO TANK-BARGES

by

C. W. Bascom
General Dynamics
Groton, Connecticut

under

Department of the Navy
NAVSEC Contract N00024-68-C-5419

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U.S. Coast Guard Headquarters
Washington, D. C.

1970

ABSTRACT

This report describes a program of analytical research to determine the availability of reliable methods for the design of long, large diameter, cylindrical tanks and their supports for transportation of liquids and low-pressure liquified gases in barges for service on rivers or at sea. Loading conditions, existing design/analysis methods, material considerations, and a computer method for predicting stresses are presented.

The major conclusion of the work performed is that design procedures for river barge tanks up to 20 feet in diameter are well established and that no failures due to inadequate design practice have been reported since refrigerated tanks went into service about ten years ago. The present method for designing river barge tanks is a logical starting point for determining the structural configuration of large tanks for oceanic service, but more detailed analysis of loads and resulting stresses should be performed for this application.

Several areas in which theoretical or experimental effort is needed are identified: (1) investigation of tank-saddle-barge interaction, (2) investigation of fatigue criteria for cyclic loading, (3) investigation of buckling criteria, (4) analytical and experimental investigation of slamming, and (5) experimental verification of stresses in a full-scale tank.

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Section 1

INTRODUCTION

The trend in the distribution of large volumes of industrial gases has been toward the refrigerated mode of transport and storage. Independent tank barges have proven to be both practical and economical and this mode of transport is being considered for coastal and oceanic service.

Dimensions of river barges are limited to approximately 10 feet in draft, 53 feet in width, and 300 feet in length; the draft dimension is controlled by river depth, and length and beam by river lock size. These dimensions limit the maximum capacity of the barge to approximately 3, 000 tons. This in turn limits the diameter of the cargo tanks to approximately 20 feet and the length to about 250 feet. Two tanks are usually mounted side by side on the barge, and supported on from 7 to 13 saddles. Stiffeners are installed at the saddles to accommodate the high reaction loads at these points. Typical tanks are fabricated of 1/2-inch carbon manganese steel. Design pressures are as low as 4 to 10 psi and design temperatures are approximately -30°F. The tanks are covered with approximately three inches of insulation. Redundant refrigeration plants and safety valves are provided to ensure against overpressurizing the tanks due to vaporization of the fluid.

From the structural point of view, these large river barge tanks have relatively small thickness-to-diameter ratios, i.e., they are quite thin walled. They operate essentially at atmospheric pressures, and reaction forces rather than pressure stresses govern the design. The empirical procedure for designing these tanks for reaction forces is based on experimental work with stationary tanks having just two supports and relatively heavier walls than today's large river barge tanks.

Independent tank barges are envisioned in the near future for coastal and transoceanic service. Very large barges in the 20, 000-ton range are economically attractive. Since ocean or coastal barges will not be subjected to the dimensional limitations of river barges, tanks as large as 40 feet wide and 400 feet long are envisioned.

Two major questions arise concerning the design of large ocean service tank barges.

1. What loading conditions are applicable to the design of large cylindrical tanks for ocean service?
2. Is the empirical procedure developed for smaller, heavier walled stationary tanks applicable to the larger sizes? If not, what is the most reliable procedure, or what further work is needed to derive an adequate procedure?

Interpretive answers to these questions have been the prime objective of this three-month study. This objective is stated more fully in the schedule of the contract as follows:

"Analytical research shall be undertaken to determine availability of reliable methods for the design of long, cylindrical tanks, and their supports, for the transportation of liquids and low-pressure liquified gases in barges on rivers or at sea. The work shall involve:

1. Description of the loads and loading conditions which must be considered in tank design, for sizes up to 40 feet in diameter and 400 feet in length.

2. Determination of the analytic methods presently available for use in the design of tanks and their supports, when installed in barges.

3. Determination of the most reliable method or combination of methods presently available to extend such design, from the standpoint of safety, economy, and efficient design, to the larger tanks.

4. Determination of those areas in which theoretical or experimental work is needed."

Section 2

APPROACH

Six basic tasks were performed in order to accomplish the four objectives stated in the Introduction.

TASK A - Investigation of Chemical Tank/Barge Operating Conditions.

TASK B - Investigation of Tank/Barge Loadings.

TASK C - Investigation of Tank/Barge Design Characteristics.

TASK D - Evaluation of Stresses in Existing and Projected Designs.

TASK E - Engineering Investigation of Materials Problems Associated with Large Tanks.

TASK F - Preparation of an Interpretive Report Including Recommendations for Research in Major Problem Areas.

Background data for the above tasks was obtained by reviewing the literature and by contacting personnel in the barge industry. The literature review is reflected in the list of references. Much helpful background material was obtained by contacting regulating bodies, surveyors, designers, builders and operators of tank barges. (Some of the many helpful contacts made in the course of the study are listed in the Acknowledgements.)

In order to perform Tasks C and D, it was expedient to work with specific tank/barge configurations. Since existing designs are of a proprietary nature, two hypothetical designs - one river type and one offshore type - were selected for examination. The following procedure was used to determine tank and barge characteristics.

Configurations, i.e., wall thicknesses and number and spacing of stiffeners, were determined for various tank lengths and diameters. These were accomplished by determining reactions due to tank deadweight from elementary structural theory. Dynamic forces due to pitch, roll, and heave were accounted for by applying a dynamic load factor to the static forces. Density of the fluid in the tanks was assumed to be 42 pounds per cubic foot, which is representative of several liquified gases now being transported. Tanks having from 2 to 11 saddles were considered. Three families of tanks were investigated: 20-foot diameter tanks, 200 feet long; 30-foot diameter tanks, 300 feet long; and 40-foot diameter tanks, 400 feet long. The wall thickness for each diameter, length and support (number of saddles) configuration was determined by assuming that the governing criterion was the buckling of short cylindrical columns as defined by Zick[1]. For selected lengths and diameters, curves of critical stress vs. thickness were plotted for configurations having from 2 to 11 supports. From the curves, representative tank wall thicknesses were selected for a river barge and an offshore barge, based on lower limits of tank wall thickness considered practical for fabrication.

Rough, structural designs were also prepared of a river barge and an offshore barge which would accommodate the previously selected tanks. The purpose of this effort was to obtain the weight, stiffness and buoyancy characteristics of the barge for use in evaluating barge/tank interaction due to wave action in the case of an ocean-going tank barge and grounding in the case of a river tank barge.

The two tank/barge configurations were then analyzed for the loading conditions established in Task B. Of these loading conditions, the most severe is grounding for a river barge and sagging/hogging for an offshore barge. Reaction forces were determined for these severe conditions using the iterative procedure outlined by the Coast Guard.[2] This method is based on the assumptions that reaction forces are primarily dependent on bending stiffness and the effects of shear stiffness are negligible.

Stresses in the area of the saddles were then evaluated by two methods, the method of Zick[1] which is now common design practice and the method of Kalnins[3], a more sophisticated computer approach. The calculated stresses in each case were compared with allowable stress. This analysis demonstrated procedures of the two methods and compared results, rather than evaluated the hypothetical designs.

Section 3

CONCLUSIONS AND RECOMMENDATIONS

This section summarizes the most significant conclusions of the investigation and gives recommendations for further theoretical and experimental work. Conclusions regarding specific loads and design/analysis procedures are contained in sections 4 through 7.

Design/analysis procedures for low-pressure, refrigerated tanks for service on rivers are well established. A survey of designers, regulatory bodies, builders, surveyors, and operators indicates that no major failures due to design inadequacy have ever been reported since this type of barge came into service about 10 years ago. In view of the excellent operating history and record of river tank barges, the design procedures for river barge tanks of up to 20 feet in diameter for river barge application are considered adequate. In many cases operators specify structural strength in excess of regulatory body requirements.

Design procedures for river barges are generally applicable for determining the basic configuration of larger tanks contemplated for ocean service. This conclusion is based on the good agreement between midsurface stresses calculated by the established empirical procedure and a more sophisticated computer analysis. However, the empirical procedure does not give stresses at enough points to fully satisfy inputs for analysis of cyclic loads on tanks for ocean service and a more detailed stress analysis will be required for this case. Furthermore, theoretical predictions of stress in large, thin-walled multisupported tanks should be verified experimentally. Theoretical predictions have been verified only on smaller, heavy-walled tanks supported on just two saddles.

Long tanks for ocean service will be subjected to cyclic loads and relatively large deflections as the barge sags and hogs due to ocean wave forces. Cyclic loads are not as significant in river barges and therefore, criteria for evaluation of these loads have not been established. Criteria for cyclic loading and fatigue evaluation including factors for effects of surface imperfections should be established for ocean tank barges. The barge tank/deflections will cause interaction between the barge structure, the tank structure, the saddle structure, and the saddle insulation. The spring constant of the saddle insulation material is nonlinear which greatly complicates exact prediction of the interaction.

During the initial three months work, several specific problem areas needing further analytical and experimental research were identified. These problems are generally not applicable to river tank barges now in service, but apply to the larger ocean barges envisioned for the future.

3.1 EXPERIMENTAL AND ANALYTICAL ANALYSES OF AN AS-BUILT TANK

The foremost problem confronting the designer is the question of adequacy of design/analysis techniques available to him. Our investigation to date has shown good agreement between the simplified approach now used for river barges and a more sophisticated numerical analysis procedure of points on the tank where the simplified method applies. Neither method, however, has experimental data to verify results in the larger sizes envisioned for ocean barges.

A tank barge in the building stage should be instrumented with strain gages for the purpose of checking analytical results. The gages could remain on the tank after it is put into service for a specified time and recordings made of stresses under various loading conditions. Further discussion of an experimental program is contained in Appendix C.

3.2 FATIGUE ANALYSIS

In the course of the study, it became evident that specific experience in designing large thin-walled tanks subject to cyclic loads encountered in ocean service is very limited. It was also evident that the simplified stress analysis procedures approximate membrane or mid-fibre stress at selected points only. A more comprehensive examination of stresses, both inside and outside the tank wall, is necessary for a fatigue analysis. Also, allowable stress limits for fatigue analysis of tank materials have not been determined. Data may exist, and, if so, it must be collected and related to the tank/barge application. If data does not exist, then experimental work will be necessary.

3.3 BUCKLING ANALYSIS

There is wide divergence in the critical compressive buckling stresses determined from methods contained in the literature. For example, the critical buckling stress as determined by the method of Timoshinko is greater by a factor of 2 than the value determined by the method of Zick. This area certainly needs further investigation. A more extensive review of the literature and an investigation of buckling criteria developed for other applications are proposed. A model test program may be necessary if no applicable data is available.

3.4 SLAMMING INVESTIGATION

Slamming is a major area of concern in the design of all hulls for ocean service. Ocean tank barges are no exception where slamming loads appear to affect the tank as well as the barge hull itself. In the area of the forward rake bulkhead, slamming may cause large deformation of the hull which is transmitted up into the forward tank saddle. The tank saddle is separated from the hull by a layer of insulation which may cushion slamming loads, but to what extent this occurs has not been determined. Experimental work with specific model barge hulls should be undertaken to determine pressure distributions.

The next step would be to apply these pressures to the hull tank structure with proper boundary conditions to determine the hull/tank interaction. The problem appears to be quite complex but not impossible to solve utilizing today's computer technology. A further discussion of a model test program for investigating tank barge slamming is contained in Appendix D.

3.5 TANK/BARGE AND SADDLE INTERACTION

The effect of saddle flexibility on sagging/hogging and dynamic loads should be determined. This problem could be approached by utilizing a matrix structural analysis procedure. The tank and barge would each be represented by $(n + 1)$

stiffness matrices, where n is the number of saddles. Each saddle would consist of two stiffness matrices: one for the insulation material and one for the saddle structure. Stiffness of the insulation material would be determined from the manufacturers' data or from testing. Several analyses would be performed to evaluate the effect of hard and soft saddles. Uniform loads due to weight and variable buoyancy loads would be represented by at least three concentrated loads between each support.

Section 4

TANK BARGE LOADINGS

Refrigerated cargo barges are operated at atmospheric pressure, with the dominant load being caused by reactions at the supports due to cargo weight rather than by pressure. Furthermore, the very large tanks envisioned for ocean-going barges will be subjected to dynamic forces, in addition to the weight of the cargo, as the barge pitches, rolls, and heaves in a seaway, and to sagging and hogging forces as the barge hull deflects while waves pass under it. Sagging and hogging are cyclic loads that cause the fatigue strength of the tank to be an important consideration. Vibration of the tank caused by wave motion may occur if natural frequencies of the tank are close to the wave encounter frequencies of ocean-going barges.

River barges are not subjected to the large, dynamic reaction loads of their ocean-going counterpart. The most severe load on a river barge is caused by support reactions in the grounded condition. Reference 4 discusses some design techniques and regulations for river barges transporting hazardous cargoes.

It is common practice in the design of tanks to assume that the saddle reaction forces and longitudinal bending moments may be obtained by an iterative process utilizing a model of an elastic beam (the tank) mounted on another elastic beam (the barge). Inherent in this procedure are the assumptions that the saddle and its foundation are infinitely rigid and that the moment in the tank is always a certain percentage of the overall barge/tank bending moment. Actually, the tanks are mounted on thermal insulation (20# urethane foam) which also cushions the tank and helps to distribute peak saddle loads to adjacent supports. Thus, the assumption of rigid supports is considered to be conservative. The assumption that the tank carries a certain percentage of the overall bending moment is considered reasonable if the tank is held down on the saddles, if the tank stiffness is not less than about one third of barge stiffness, and if the neutral axes of the barge and tank are separated by less than about one half of the tank radius. These conditions are satisfied in typical independent tank barge designs.

The probability of a severe grounding on a pinnacle at the forward rake bulkhead is extremely small, and this fact is acknowledged in the Code of Federal Regulation by the allowance of a stress equal to two thirds of the ultimate tensile stress. In view of the severity of the specified grounding load and the small likelihood that it will occur, a more sophisticated approach for determining tank-barge-saddle interaction in the design of river barges does not appear to be necessary.

If, on the other hand, tanks are designed as structural members of ocean barges subjected to millions of cycles of sagging and hogging, then effects such as saddle flexibility may have more significance. Analysis of the tank mounted on the barge, including the flexibility of the supports, is possible utilizing a stiffness matrix approach. However, a problem arises when determining the flexibility of the foam insulation material which separates the tank from the saddle. Data which adequately describes the elastic and/or plastic characteristics of the insulation material apparently does not exist in the literature.

Due to the uncertainty of the elastic/plastic properties of the saddles, and the time and expense involved in formulating a computer model, the effects of saddle interaction were identified as a problem area for further investigation rather than pursued further in this study. The tank loads used for evaluation of stress analysis procedures were determined in this investigation by the iterative process described in section 4.3.

4.1 CARGO DEADWEIGHT REACTION LOADS (STILL WATER)

This load is common to both river and offshore barges and is quite easy to obtain. Reaction loads may be determined using elementary structural theory, or they may be approximated in symmetrical designs by dividing the total weight of cargo and tank by the number of supports. Table 4-I gives reaction loads for various size tank configurations with hemispherical head and 42 lb/ft^3 fluid, using a moment distribution method.[5]

The American Bureau of Shipping[6] uses a convenient means of approximating weight per foot of cargo tank:

$$W = R^2 \left(256 \frac{t}{R} + 196 \text{ Sp. Gr.}\right)$$

where: W = weight per unit length (lb/ft)

t = tank thickness (in.)

R = tank radius (ft)

Sp. Gr. = specific gravity of fluid (dimensionless)

For multiple-supported tanks on evenly spaced saddles, and when the length of overhangs approaches one half the length of each span, the saddle reaction load in pounds is equal to the product of the weight per foot and the saddle spacing.

With the saddle reaction and weight per foot known, shear and moment diagrams may be plotted for use in calculating tank stresses.

4.2 DYNAMIC LOADS

The most significant dynamic load is caused by acceleration of the mass of the cargo tank and its contents. This load is maximum if the tanks are assumed to be full. If the tanks are assumed to be only partially full, sloshing loads will be present. Dynamic loads under each condition are discussed in the following paragraphs.

4.2.1 FULLY LOADED CONDITION — The Code of Federal Regulations, Title 46 Chapter 1, subparagraph 38.05-2, specifies the following:

> "Cargo tanks in vessels in ocean, Great Lakes, lakes, bays, and sounds, or in coastwise service shall be designed to withstand the following dynamic loadings:
>
> 1. Rolling 30° each side (120°) in 10 seconds.
> 2. Pitching 6° half amplitude (24°) in 7 seconds.
> 3. Heaving L/80 half amplitude in 8 seconds."

Table 4-I. Moments and Shears for Various Tank Configurations (Weight: 42 lb/ft^3)

DIAMETER/ LENGTH (ft)	NUMBER SUPPORTS	OVERHANG (ft)	SUPPORT SPACING (ft)	MOMENT (lb-ft) (x 10^{-6})	SH (x
20/200	2	43.70	107.07	14.1	
	3	28.20	69.09	5.88	
	4	20.81	50.98	3.20	
	5	16.49	40.40	2.01	
	6	13.65	33.44	1.38	
	7	11.65	28.54	1.01	
	8	10.16	25.99	.762	
	9	9.01	22.07	.601	
	10	8.09	19.82	.484	
	11	7.34	17.98	.399	
30/300	2	65.4	160	71.73	2,
	3	42.2	103.5	29.33	1,
	4	31.2	76.3	16.31	1,
	5	24.6	60.5	10.14	1,
	6	20.5	50.0	7.04	
	7	17.4	43.0	5.09	
	8	15.2	37.3	3.87	
	9	13.4	33.1	3.01	
	10	11.0	29.9	2.03	
	11	10.1	27.1	1.71	
40/400	2	87.3	213.8	230.5	6,
	3	56.3	137.9	95.8	4,
	4	41.5	101.8	52.1	3,
	5	32.9	80.6	32.7	2,
	6	27.3	66.8	22.5	2,
	7	23.3	57.0	16.4	1,
	8	20.3	49.7	12.5	1,
	9	18.0	44.1	9.80	1,
	10	16.2	39.6	7.94	1,
	11	14.7	35.9	6.53	1,

These conditions were investigated in Task A and found to be reasonable. Appendix A gives the results of this investigation.

Using the above three conditions for pitch, roll, and heave, together with the characteristics of the barge, a dynamic vertical load factor may be approximated from elementary equations of harmonic motion, resulting in the following maximum values:

$$G_v = \frac{p + r \pm h}{g} \tag{4-1}$$

where: G_v = vertical dynamic load factor (dimensionless)

p = pitch acceleration (ft/sec^2)

r = roll acceleration (ft/sec^2)

h = heave acceleration (ft/sec^2)

g = gravity = 32.2 ft/sec^2

and

$$p = \left(\frac{2\pi}{7}\right)^2 \ell \sin 6°$$

$$r = \left(\frac{2\pi}{10}\right)^2 d \sin 30°$$

$$h = \left(\frac{2\pi}{8}\right)^2 \frac{L}{80}$$

with: ℓ = variable distance from longitudinal center of gravity of barge to the saddle in question (ft)

d = distance from the vertical center of gravity of the barge to center of gravity of the tank.

L = length of barge (ft)

The dynamic load factor, G, may be applied directly to each of the still water saddle reactions to approximate the design load. The dynamic load factor should also be applied to the hydrostatic pressure in the tank.

4.2.2 **PARTIALLY LOADED TANKS — Sloshing loads will be prevalent in partially filled, unbaffled tanks. When they are filled to or near capacity, fluid will act almost as a solid mass, and dynamic loads due to pitch, roll and heave will be transmitted to the supports, as described previously. When tanks are almost empty, the force on the supports will be greatly reduced due to the negligible amount of mass of the fluid. The prediction of loads due to sloshing is difficult in the range of fluid capacity from 90 to 10 percent. Sloshing in liquid fuel tanks of missiles has been treated quite extensively by the aerospace industry.[7] However, the methods developed for missiles do not appear applicable to tank barges for several reasons: (1) the motions of the tank barge vary more than those of the missile; (2) the fluid mass is variable in the missile tank, whereas mass is constant in chemical tank barges; and (3) the orientation of the tanks is vertical with missiles but horizontal in the case of chemical tank barges.**

A preliminary investigation of fluid sloshing was performed in Appendix A (section A-3) with the conclusion that an adequate method of predicting sloshing loads in unbaffled tanks does not appear to exist. This situation can be overcome through a program of experimental and analytical research; however, justification for such a program is questionable when the practical aspects are considered. Most oceanic barge operations will consist of one-way trips with tanks full, and return trips with tanks empty. If partial loads are being considered, then sloshing may be greatly reduced by the installing of baffling of the tank truck type in the tanks.

4.3 GROUNDING LOADS

The condition for grounding is specified in Title 46, Chapter 1, Paragraph 98.03-25 of the Code of Federal Regulations. Grounding loads on the tank will depend on the relative stiffness between the tank and barge and whether or not the tank is held down on the saddles. The tank supports may be designed so as to contribute to the strength of the barge. If this is the case, then the support loads may be determined by considering the tank as a beam on an elastic foundation — the barge. The Coast Guard[8] has formulated this analysis which is essentially as follows: The barge is assumed to be grounded at the forward rake bulkhead. A loading curve, shear curve, and moment curve, such as those shown in section 6, may be obtained using the following procedure which is quoted from reference 8.

> "Starting from the forward (grounded) end, the total barge moment is computed at each saddle. This is done by summing the moments due to barge hull weight, tank loading, grounding force and buoyancy.
>
> A positive moment is one that places the deck of the barge in compression, while forces are positive downward. The buoyancy curve is assumed to vary linearly from zero at the grounding point to a maximum at the after rake tangency point. The moment in each tank abreast is then computed on the basis of the product of the ratio
>
> $$\frac{I_{tank}}{I_{barge} + I_{tank} \text{ x number of tanks abreast}}$$
>
> and the total barge moment at each saddle, except that the moment at the end saddle is computed as though the overhanging section were a cantilever. The tank weight is then divided by the tank length, and the resulting weight per foot is assumed to be evenly distributed. Since the moment is known at each saddle, along with the distributed load between saddles, the shear to the left and right of each saddle is computed and combined to give the reaction at each saddle. These reactions are multiplied by the number of tanks abreast to get the total tank reaction at each saddle location. A check will show that the sum of the reactions equals the weight of the tanks. The cycle is then repeated until the solution converges, the only variation being that the saddle reactions are used to compute the total barge moment in lieu of the uniformly distributed tank loading used in the first cycle."

4.4 SAGGING AND HOGGING LOADS

Sagging and hogging loads are determined in a manner similar to grounding. The buoyant force, however is obtained by balancing the barge on a trochoidal wave with a wave length equal to the barge length. The peaks of the wave are placed on the ends of the barge to create sagging and the midpoint of the barge to create hogging. (This procedure is explained in reference 9.) Load, shear and moment curves for typical sagging and hogging situations are shown in section 6.

Sagging and hogging loads in tanks may be practically eliminated by using two or more tanks end to end rather than one long continuous tank. This configuration may be adopted if large negative forces are predicted in the sagging/hogging analysis. If this approach is used, the barge structure must be designed to carry the entire bending moment in grounding, sagging, and hogging.

4.5 FATIGUE

Continuous tanks represent a significant portion (1/3 to 1/2) of the overall structure of an independent tank barge. Furthermore, the tank structure is located in a good position to contribute to the bending strength of the barge. Thus, it is economically attractive to design the tanks to carry a portion of the overall bending moment.

To accomplish this, the saddles must be designed to transmit loads between the tank and barge in such a manner that the two structures act as an integrated structure. When the structure is integrated, both the tank and the barge must resist the cyclic loads of sagging and hogging. Reference 10 presents an engineering approach to low-cycle fatigue of ship structures. The conclusion of this report is that most of the bending cycles experienced by a ship structure induce low nominal stresses. Therefore fatigue of the main structural girders, per se, is not of prime concern. However, low stress intensities are magnified by unavoidable discontinuities in local areas where the yield strength may be reached or exceeded. Thus, low-cycle fatigue is a real problem in certain localized areas of the ship structure.

From a preliminary examination of typical tank structures, it appears that the area of the structure in the vicinity of the saddles may be a trouble spot, particularly if corrosive fluids are carried in the tank. In this area, heavy stiffeners are joined to the relatively thin tank wall, creating geometrical discontinuities. Furthermore, residual stresses and discontinuities will occur around the welds required to join the stiffener to the tank.

It appears that there are no guidelines available to the tank designer which will assist him in accounting for geometrical discontinuities, weld treatment, and stress corrosion, and thus designing a tank which will resist cyclic loading. Solutions to these problems are necessary before the tank can be utilized as a structural member of an ocean-going barge.

4.6 FORCED VIBRATION LOADS

The forced vibration loads on a tank may be significant if the natural frequency of the tank/barge is close to the frequency of wave encounters. The frequency of wave encounters may, of course, be changed operationally by reducing speed or by changing the course of the barge. This loading condition should be checked especially when very long tanks and barges are being considered for high-speed operation. Figure 4-1 shows the trend in barge frequency versus wave encounter frequency. As barges approach 600 to 700 feet in length, natural frequencies approach the wave encounter frequency. Reference 11 presents a simplified method of calculating the first five frequencies of a tank/barge on an elastic foundation, as follows:

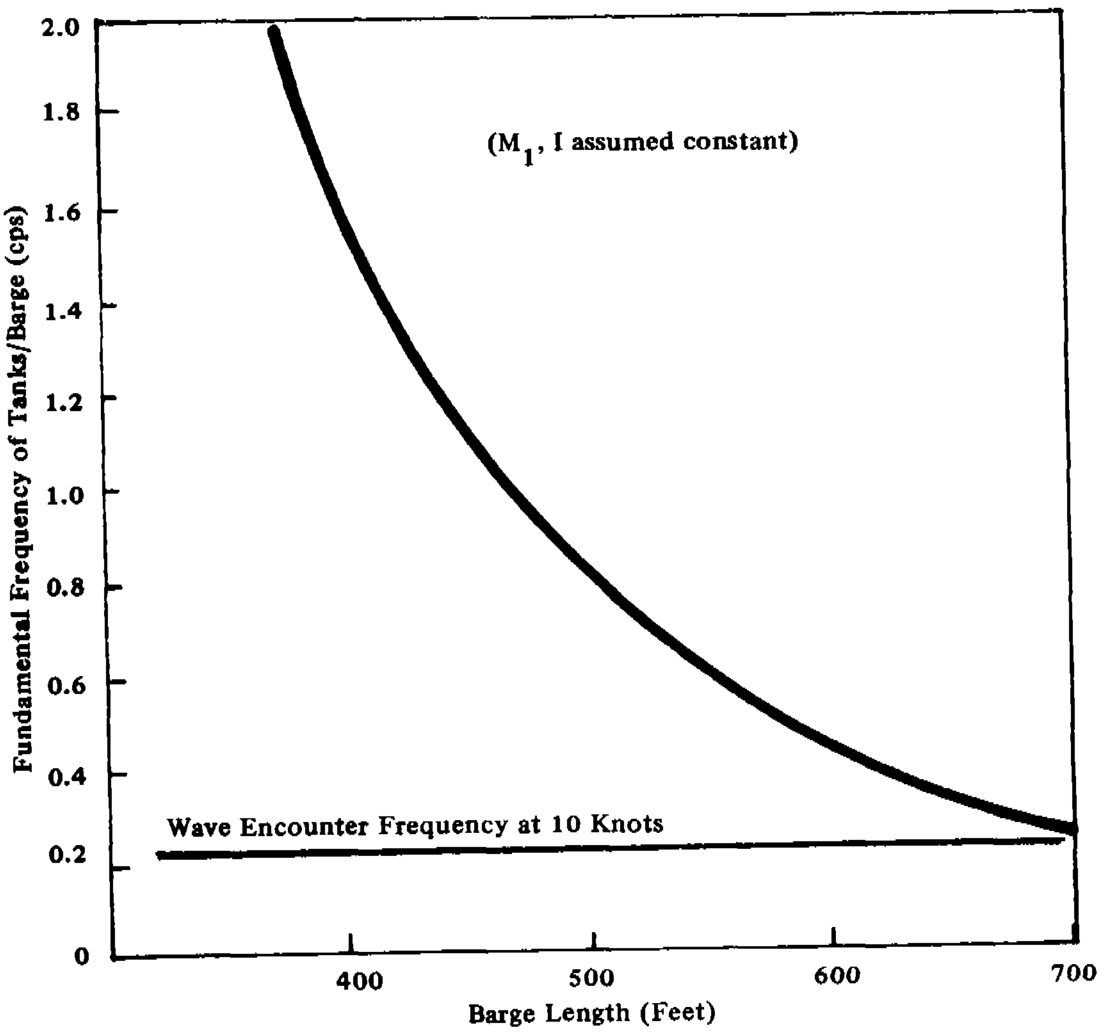


Figure 4-1. Trend in Fundamental Frequencies vs. Barge Length

$$f_n = \frac{A_n}{2\pi}\sqrt{\frac{EI}{M_1 \ell^4}} \tag{4-2}$$

where: f_n = frequency (cycles per second)

A_n = coefficient for mode of frequency (table 4-II)

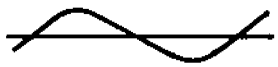
E = modulus of elasticity (psf)

I = moment of inertia of barge and tank (ft^4)

M_1 = mass per unit length $\left(\frac{lb/sec^2}{ft^2}\right)$

ℓ = tank/barge length (ft)

Table 4-II. Values of A_n for Calculating Frequency

MODE	SHAPE	VALUE
1		22.0
2		61.7
3		121.0
4		200.0
5		298.2

4.7 PRESSURE LOADS

Pressure loads on refrigerated cargo tanks generally will not be of major significance in tank design. However, pressure will cause stress components which must be accounted for in the overall evaluation.

The Code of Federal Regulations, Title 46, Chapter 1, Support 38.05-3(g), states that "Cargo tanks in which the temperature is maintained below the normal atmospheric temperature by refrigeration or other acceptable means, shall be designed for a pressure of not less than 110 percent of the vapor pressure of the liquid at which the system is maintained." This is the general rule for most fluids; however, in the case of ammonia, which is essentially at atmospheric pressure during refrigerated transport, the Code in Subpart 98.25 10(d) states that 25 psig must be added to the transport pressure.

In addition to the vapor pressure, the hydrostatic pressure should also be considered in the design of large tanks. In a 40-foot tank carrying 42 lb/ft^3 fluid, for example, hydrostatic pressure on the bottom of the tank is about 11.6 psig.

4.8 TEMPERATURE AND THERMAL LOADS

Temperature and thermal loads are important considerations in the design of refrigerated tanks. Thermal loads will be of major significance if the tanks carry very low temperature (less than -150°F) gases such as liquified natural gases, oxygen, and nitrogen. Highly specialized designs are required for very low temperature applications, and extensive heat transfer analyses are required to predict thermal loadings. Thermal loads on low temperature (above -150°F) applications are not generally a problem if the tanks are properly insulated and if they are gradually cooled during loading operations.

The design temperature for low temperature applications is more important as a basis for material selection than for prediction of thermal stresses. The design approach to thermal stresses should be to minimize them through proper insulation and by installation of spray nozzles or other devices to cool the tank gradually. "Low temperature" steels are suited to tanks with ambient temperatures down to -150°F. Liquified gases having temperatures below -150° will require special designs for insulation and use of cryogenic steels or aluminum. More information on material selection for low temperatures is given in section 7.2.

The design or service temperature used in the selection of tank material may be determined by the method specified in the Code of Federal Regulations, Title 46, Chapter 1, Subpart 38.5-2(b):

> "(b) The service temperature is the minimum temperature at which the cargo is loaded and/or transported in the cargo tank. However, the service temperature shall in no case be taken higher than given by the following formula:
>
> $t_s = t_w - 0.25\ (t_w - t_B)$
>
> where: t_s = service temperature
>
> t_w = boiling temperature of gas at normal working pressure of tank but not higher than +32°F
>
> t_B = boiling temperature of a gas at atmospheric pressure."
>
> "(d) Heat transmission studies, where required, shall assume the minimum ambient temperatures of 0°F still air and 32°F still water, and maximum ambient temperatures of 115°F still air and 90°F still water."

4.9 COLLISION LOADS

The Code of Federal Regulations' requirement for collision shock loads of 1.5g appears to be reasonable. Figure 4-2 shows the stopping distance, barge velocity, and stopping times for 1.5g, assuming constant deceleration. These relationships were obtained from the elementary theory of dynamics. The stopping distances and times appear to be conservative in light of the large momentum of loaded barges and the amount of deformation commonly experienced in barge collisions or groundings.

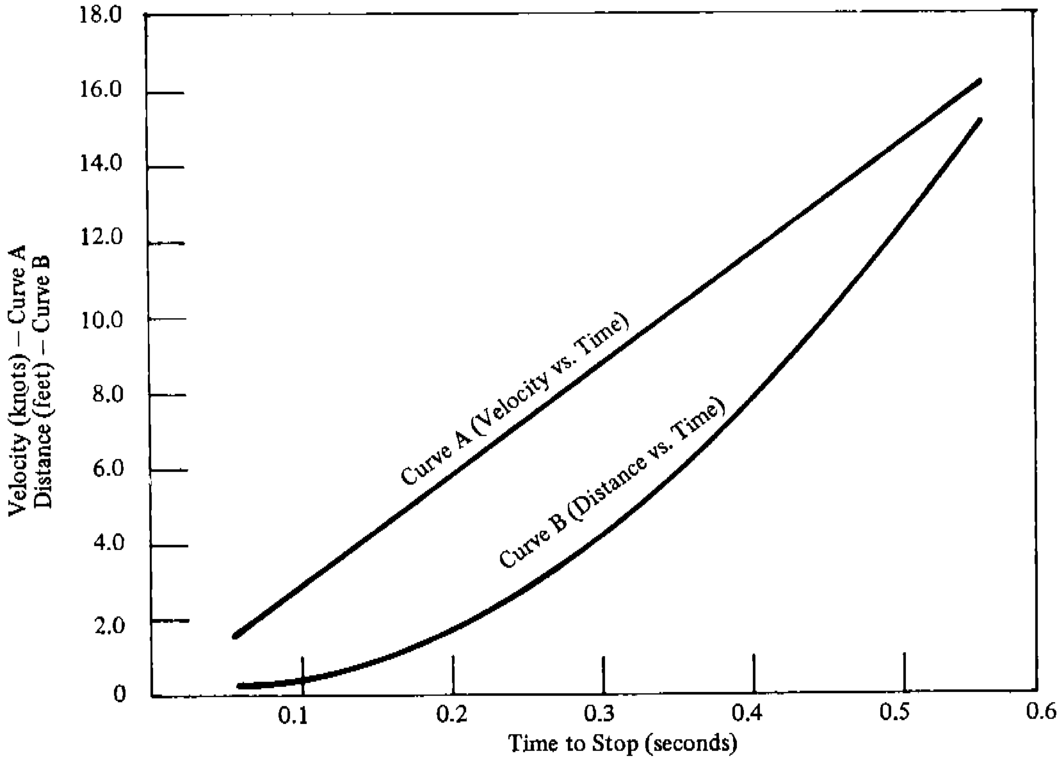


Figure 4-2. Barge Velocity and Distance vs. Time for Constant Deceleration of 1.5g

4.10 LOADS ON TANKS/BARGES DUE TO SLAMMING

Ship slamming, in general, is a problem area where much investigation is being performed. The purpose of including this brief summary of the problem is only to indicate the nature of the problem to readers unfamiliar with the problem. Slamming in the area of the bow may cause large areas of deformation. If the deformation is near a saddle, damage to the tank may occur. Evidence is available which indicates that slamming of barges can cause significant structural damage. However, the problem of surface ship slamming has yet to be completely solved. Certain results, useful in the case of flat bottom barges, are available.

The following paragraphs present several of the latest theoretical treatments of the slamming problem. From the results presented in figures 4-3 and 4-4, it is clear that theory does not predict very exact values. Figure 4-4 should be indicative of the magnitudes of slamming loads that will actually be encountered.

In a recent paper[12] Verhagen presented the following expression for maximum impact pressure:

$$P_{max} = C \rho \ V C_a$$

where: ρ = density of water $\left(\frac{\text{lb-sec}^2}{\text{ft}^4}\right)$

V = relative velocity of craft with respect to water (ft/sec)

C_a = speed of sound in air (ft/sec)

C = an undefined constant

Taking the value of C as 1, the resulting pressures, based on relative velocities from the ship motions program (table 4-III), are presented in figure 4-3.

Assuming a pocket of air between the boat and water, Verhagen's exact expression for maximum pressure is:

$$\frac{1}{\gamma-1}\,\frac{P_1}{P_o}\left\{1-\left(\frac{P_{max}}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}+1-\left(\frac{P_{max}}{P_1}\right)^{-1/\gamma}$$

$$=\frac{-\pi}{8}\,\frac{\rho}{\rho_a}\,\frac{\gamma B}{2h_1}\,\frac{M}{\left[\frac{\pi}{2}\left(\frac{B}{2}\right)^2\rho+M\right]}\left(\frac{W_1-V_1}{C_a}\right)^2$$

where:

γ = gas constant

P_o = atmospheric pressure

C_a = speed of sound in air

B = beam of boat

M = mass of boat

ρ = density of water

ρ_a = density of air

W_1 = water velocity at t_o

P_1 = pressure at t_o

h_1 = height of pocket at t_o

t_o = time when air pocket is sealed

This yields values similar to those of figure 4-4.

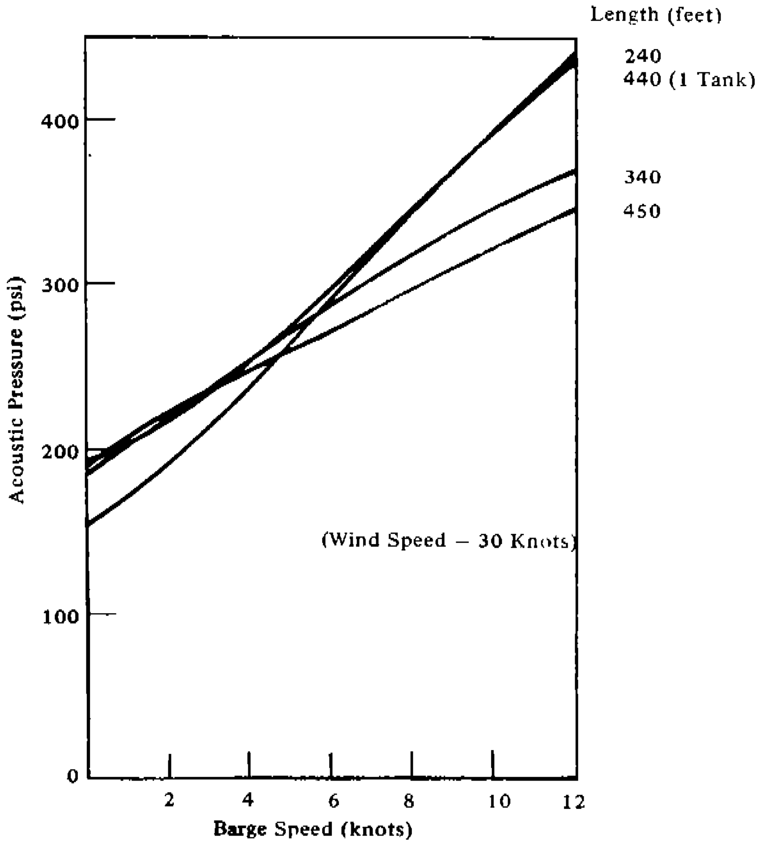


Figure 4-3. Acoustic Pressure vs. Speed for Various Barges

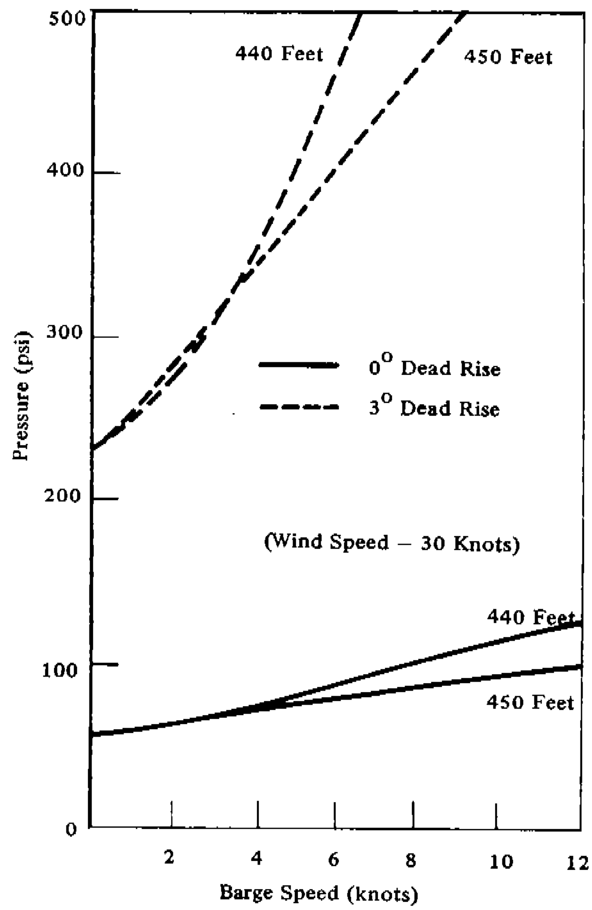


Figure 4-4 Average 1/10 Highest Slamming Peak Pressure

Table 4-III. Average 1/10 Highest Relative Velocity at Slamming Station (30-Knot Wind)

	BARGE LENGTH (FEET)			
SPEED (knots)	440 (ft/sec)	450 (ft/sec)	340 (ft/sec)	240 (ft/sec)
0	12.416	12.307	11.970	9.881
3	14.942	15.002	15.293	13.799
6	19.176	17.563	18.572	18.652
9	23.954	19.962	21.486	23.742
12	28.128	22.082	23.945	'28.301

ACOUSTIC PRESSURE

$$(P_{max} = \rho_{water} \times C_{air} \times V_{fps})$$

	BARGE LENGTH (FEET)			
SPEED (knots)	440 (psi)	450 (psi)	340 (psi)	240 (psi)
0	192.27	190.59	185.37	153.02
3	231.39	232.32	236.83	213.69
6	296.96	271.98	287.61	288.85
9	370.95	309.13	332.73	367.67
12	435.59	341.96	370.81	438.27

Chuang[13] presents, for the impact pressure of "a 20-inch x 26.5-inch rigid flat bottom body:"

$$P(t) = 0.72 \ V^2 \ e^{-1.4t/\tau} \sin \frac{\pi t}{\tau}$$

where: τ is impact duration time

Since Chuang was unable to scale this to larger bodies, and since Verhagen feels a dependence on V^2 is not good for all weights, this expression is not considered further.

Chuang also calculated the impact duration time.

$$\frac{B}{2C_a} \approx 0.045 \text{ sec for a barge (beam = 100 feet)}$$

In reference 13 Chuang presents a series of equations for the maximum pressure due to slamming.

$$P_{max} = 4.5\ V \quad \text{for } 0° \text{ dead rise hull}$$

$$P_{max} = 4.11\ V^{1.6} \text{ for } 3° \text{ dead rise hull}$$

These are plotted in figure 4-6.

Note that if the constant C in Verhagen's acoustic pressure were chosen as 0.29 rather than 1, Verhagen's expression would be identical to the above expression for 0° dead rise.

Slamming studies done on a destroyer indicate that the actual slamming pressures at the bow of a barge (where there is probably some dead rise angle) probably fall in the range between the solid and dotted curves of figure 4-6.

Since test data in this area is quite sparse, further work on barge slamming should be undertaken. An outline for an experimental program for this effort is in Appendix D.

Section 5

STRUCTURAL DESIGN/ANALYSIS OF TANK BARGES

The structural design process for large tanks may be broken down into four straightforward steps once the capacity and tank loadings have been established. This process is essentially that which is presently used for smaller river barge tanks and it may be summarized as follows:

1. From previously determined tank diameter, tank length, pressure, weight and dynamic loadings, determine the basic tank configuration, i.e., an economical combination of tank wall thickness, number of supports, and support spacing.
2. Determine area and moment of inertia requirements for stiffeners and thickness of wear plates due to weight and dynamic reactions.
3. Determine support reaction for the grounded condition in the case of river barges and/or sagging and hogging in the case of ocean barges.
4. Check local stresses in the area of saddles due to grounding and/or sagging/hogging reactions.

If, at any point in the design, the stresses exceed allowable limits, scantlings may be increased and the design continued from that point.

In the case of ocean barges, the designer should also determine if the alternating stress intensity is below the endurance limit for the predicted number of cycles of hogging and sagging.

In determining the basic size of ocean tank/barges, the fundamental bending frequency of the tank/barge structure should be calculated according to the procedure given in section 4.6. The fundamental frequency of the combined barge/tank structure should be greater than the forcing frequency of the waves. Forcing frequency may be estimated using the method given in Appendix A.

5.1 EXISTING DESIGN/ANALYSIS PROCEDURES

The U.S. Coast Guard[2] and the American Bureau of Shipping[6] both furnish guidance on the design/analysis of tank barges. These guidelines use, as a basis for determining stresses at saddles, the method of Zick.[1] Both sources also refer to the method of Brownell and Young (reference 14) which is essentially the same as Zick's method. Tank barge designers, almost without exception, use the Coast Guard procedure to analyze stresses in tanks. The author, in reviewing the literature, did not uncover any other directly applicable simple method of analyzing tank stresses in the area of saddle supports. Røren[15] discusses the problem of ring stiffeners in more detail than Zick or Brownell and Young, and the discrepancies which occur appear to be minor.

The Coast Guard and several design agents utilize a computer program[8] to calculate saddle reactions of independent tank barges in the grounded condition. Using an iterative process, the program determines the total (tank plus barge) moment and the tank moment at each saddle, together with the corresponding vertical reactions. (The iterative process may also be performed with the aid of a desk calculator.) With the tank moments and vertical reaction forces known, the stresses due to longitudinal bending, circumferential bending, direct compression, and tangential shear may be determined using the method of Zick. The stresses may then be compared to design allowable stresses as recommended by Zick or specified in the Code of Federal Regulations.

A numerical method for determining stresses in shells of revolution subjected to axisymmetric or non-symmetric pressure, band loads, ring forces, and ring moments has been published by Kalnins.[3] This method has been programed for computation on the UNIVAC 1107 computer (reference 16). During the study this program was utilized to obtain a comprehensive stress distribution in the area of the saddle. Stresses were computed on the inside, outside, and mid-surface of the tank wall. A discussion of the analysis and a comparison with results due to the Zick method are given in section 6.

Design/analysis of ocean-going barges subjected to sagging and hogging may be approached in a manner similar to the procedure for grounding calculations. Specific criteria for comparing alternating sagging and hogging stresses to design allowables, based on fatigue theory, do not appear to exist, nor does a precedent exist for including dynamic loads in a fatigue evaluation. An approach to this problem is discussed in sections 4.5 and 6.4.

Experience in design/analysis of ocean barges appears to be quite limited. In a recent survey of the industry (reference 17), only one design for independent, cylindrical ocean-going tank/barges was identified. This design was for a barge of approximately 20,000 tons, with two tanks, each about 30 feet in diameter and 300 feet long. The two tanks had originally been designed to be continuous with multiple supports. However, this design resulted in negative (lift-off) forces in the sagging and hogging analysis and the design was modified by increasing the number of tanks to four, each supported on only two supports.

A second ocean-going tank/barge consisting of three intersecting cylinders was reported to be in the construction stage but design details were not available.

5.2 RATIONALE FOR DETERMINING TANK WALL THICKNESS AND NUMBER AND SPACING OF SUPPORTS

In section 4 the tank and saddle loads due to tank and cargo weight are determined for tanks 200, 300 and 400 feet long and with configurations containing from 2 to 11 saddle supports. Uniform circular cross-section, uniformly spaced supports, hemispherical end enclosures, and an overall length/diameter ratio of 10 were assumed in the calculation of these loads. A method which is analogous to that of Zick was employed to determine the required tank thickness. Large tanks should be designed so that the tank is reinforced by circular stiffening rings placed either directly over or adjacent to the supports.

5.2.1 CALCULATION OF STRESSES — Because of the low internal pressure associated with refrigerated cargoes, the circumferential tensile stress in the tank is not necessarily the basis for determining the required tank thickness. This is a departure from the problem which Zick investigated. The maximum circumferential tensile stress occurs at the bottom of the tank and is caused by uniform internal pressure plus hydrostatic pressure due to the weight of liquid enclosed. From

Classical Membrane Shell Theory, the maximum circumferential (hoop) tensile stress is given by

$$(\sigma_h)_{max} = \frac{p^*r}{t} \tag{5-1}$$

where p* is the maximum internal pressure.

It is also necessary to examine other primary tank stresses so that the critical stress condition can be determined. Based on Classical Beam Theory, the longitudinal bending stress distribution in the tank (at $\theta = 0,\ 180^o$ as defined in figure 6-1) is given by

$$\sigma_b(x) = \pm \frac{M(x)}{\pi r^2 t} \tag{5-2}$$

Substitution of the maximum bending moment, M*, which occurs at the supports, into eq. 5-2 yields

$$(\sigma_b)_{max} = \pm \frac{M^*}{\pi r^2 t} \tag{5-3}$$

The transverse shearing stress distribution in the tank (at $\theta = 90^o$) is given by

$$\sigma_s(x) = \frac{V(x)}{\pi r t} \tag{5-4}$$

Substitution of the maximum shear force, V*, which again occurs at the supports, into eq. 5-4 yields

$$(\sigma_s)_{max} = \frac{V^*}{\pi r t} \tag{5-5}$$

5.2.2 ALLOWABLE STRESS LIMITS — Zick places the following limits on primary tank stresses:

a. On circumferential stress, the allowable working stress for the material.

b. On longitudinal tensile stress, the allowable working stress.

c. On longitudinal compressive stress, the smaller of one-half yield stress or the value given by

$$\sigma \leq \frac{E}{29}\left(\frac{t}{r}\right)\left[2 - \frac{200}{3}\left(\frac{t}{r}\right)\right] \tag{5-6}$$

which according to Zick is "based upon the accepted formula for buckling of short steel cylindrical columns."

d. On shear stress, 80 percent of the allowable working stress.

5.2.3 DETERMINATION OF CONTROLLING STRESS MAGNITUDE — Based on the limits on primary tank stresses (section 5.2.2), it was found that the longitudinal compressive stress is critical when determining tank thickness. Figures 5-1, 5-2, and 5-3 show the variation of maximum longitudinal bending stress with thickness for each of the ten different support conditions for a 200-, 300-, and 400-foot tank respectively. Superimposed on each graph is the variation of allowable longitudinal compressive stress with thickness (according to eq. 5-6).

With the help of these parametric curves, typical designs for a 200-foot, 300-foot, and 400-foot tank may be determined.

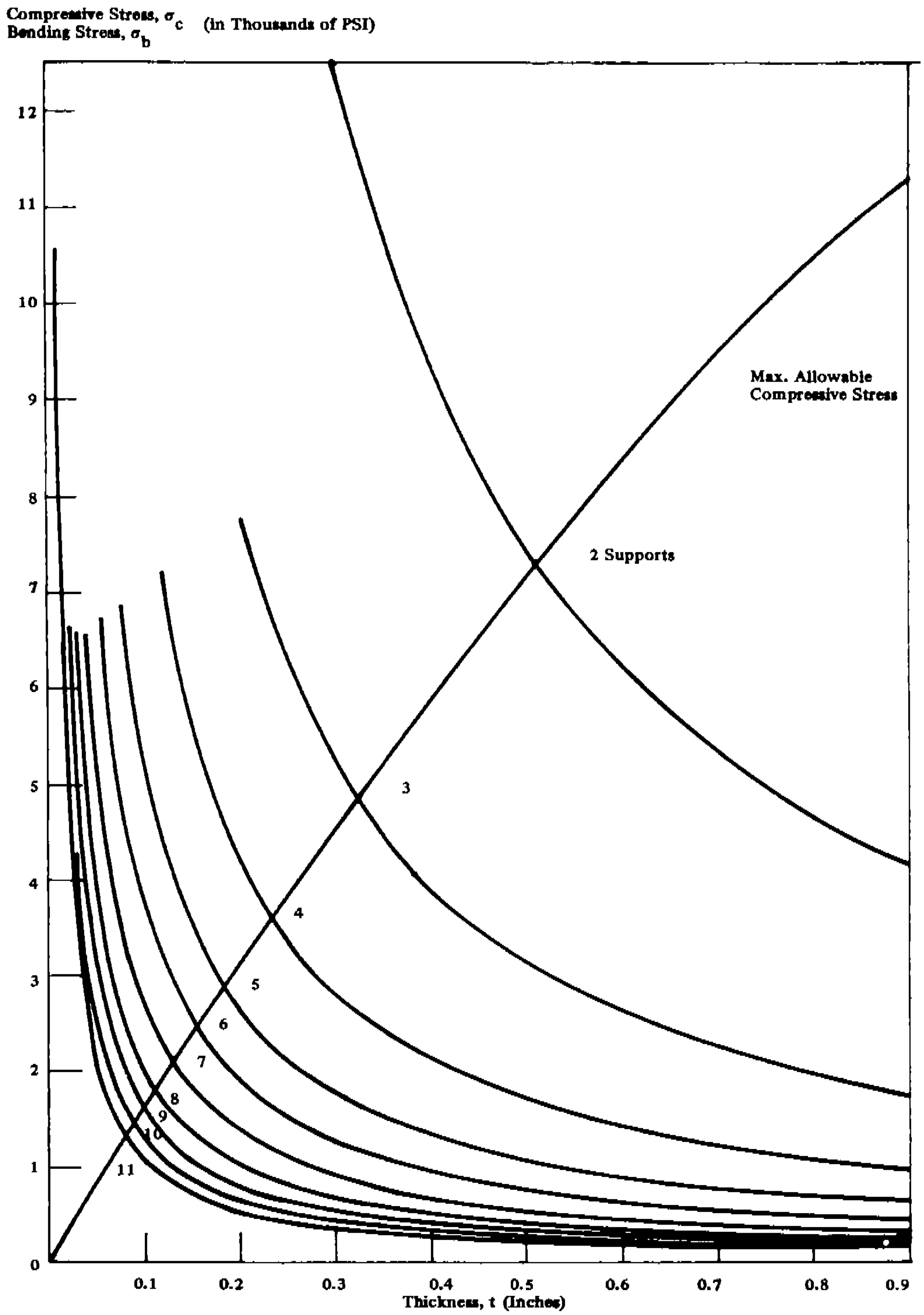


Figure 5-1. **Bending Stress vs. Thickness for Various 200-Foot Tank Configurations**

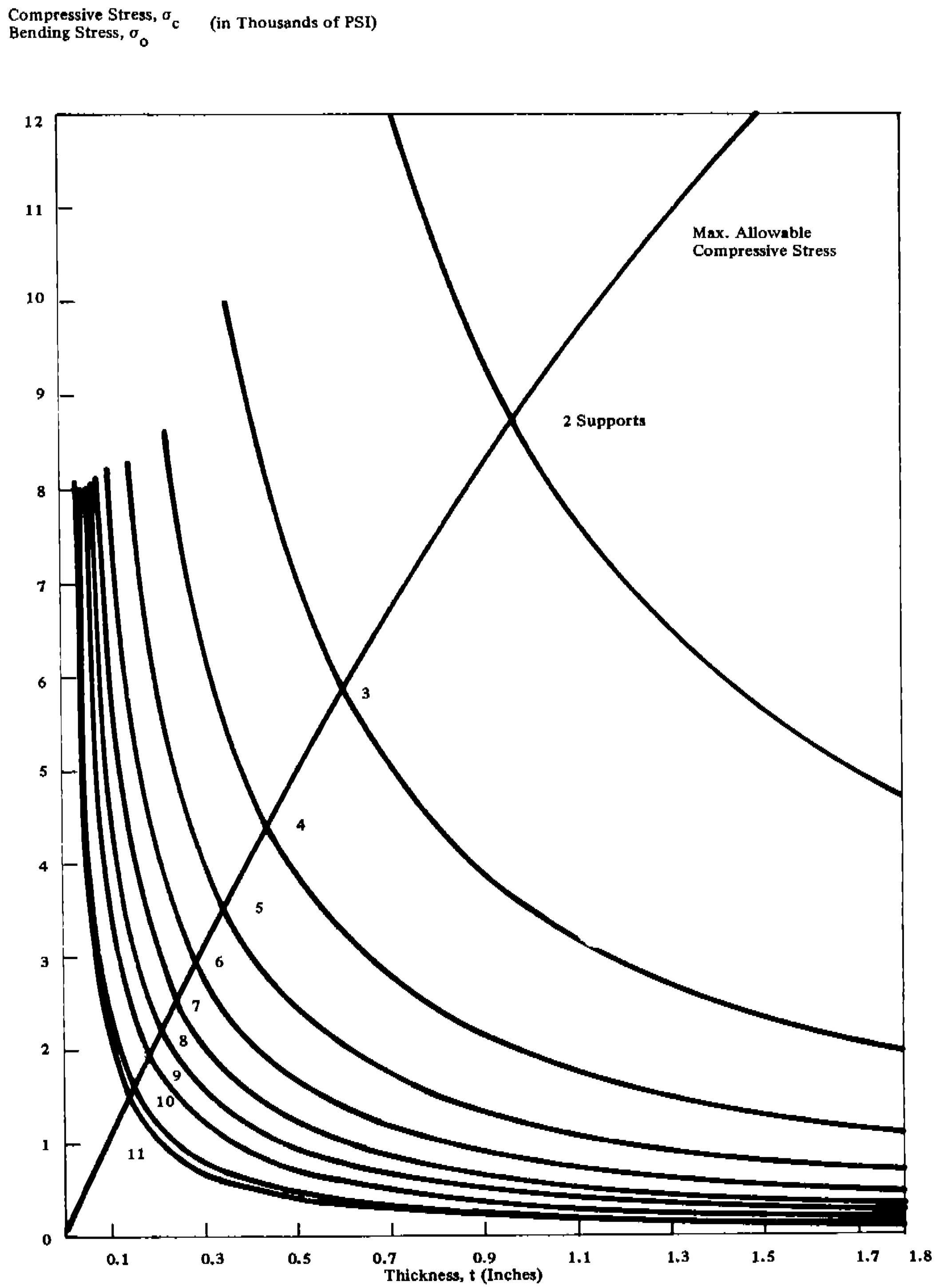


Figure 5-2. Bending Stress vs. Thickness for Various 300-Foot Tank Configurations

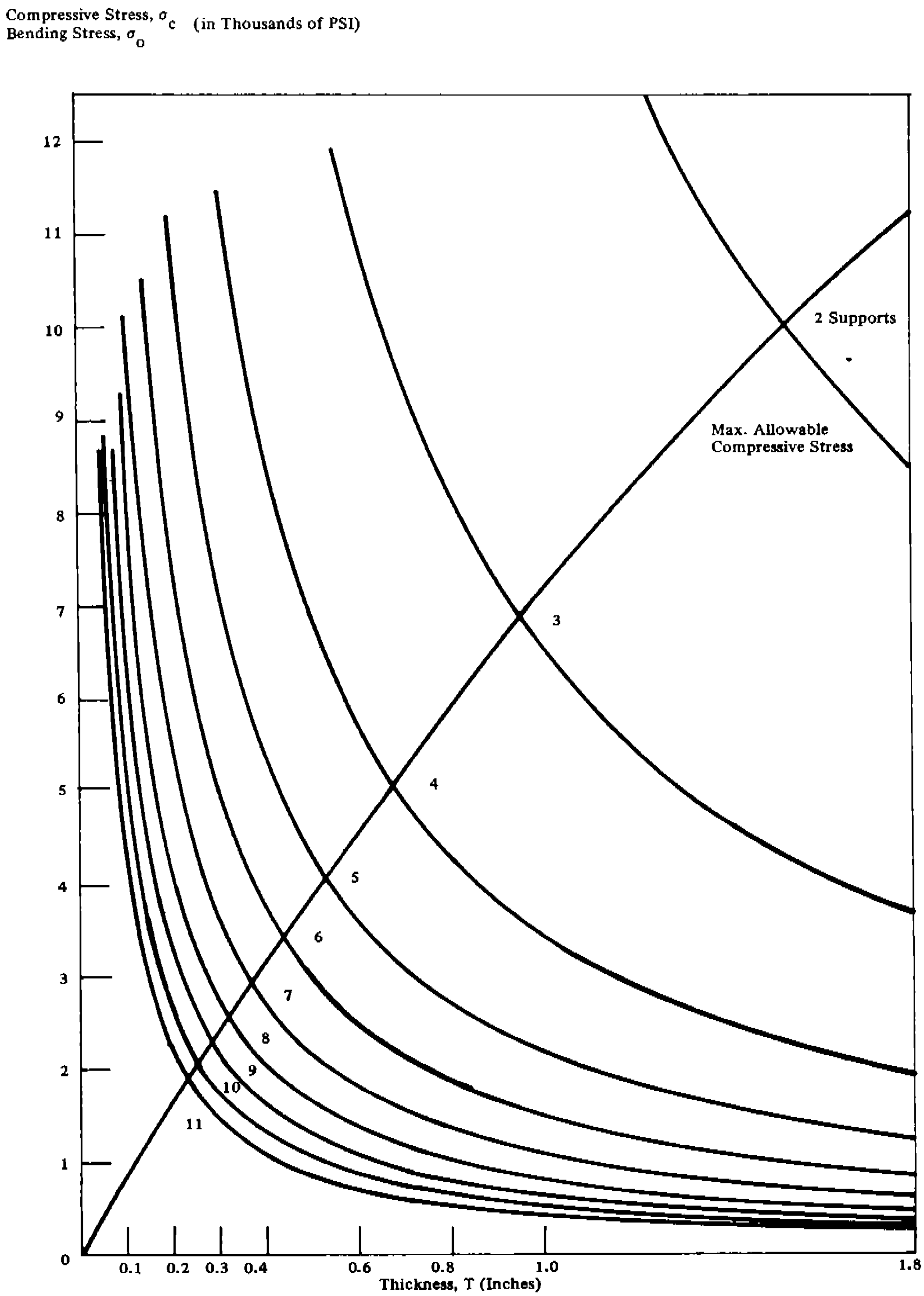


Figure 5-3. Bending Stress vs. Thickness for Various 400-Foot Tank Configurations

5.2.4 DYNAMIC LOADS — Dynamic load factors due to ship motions (heaving, pitching, etc.) are given in section 4. These factors reflect the most severe combination of ship motions. To account for these dynamic loads, the stresses in the tank due to cargo and tank weight must be multiplied by one plus the dynamic load factor.

The uniform internal pressure for which the tanks must be designed is discussed in section 4.7. This pressure contributes to uniform longitudinal tensile stress in the tanks. The value of this tensile stress is a function of the tank radius. To calculate the maximum longitudinal compressive stress level it is necessary to subtract this uniform tensile stress from the bending stress.

5.2.5 CHOICE OF SUPPORT CONFIGURATIONS — A five-support configuration was arbitrarily selected as representative of thin-walled tanks with minimum supports. This selection conforms with the American Bureau of Shipping regulations which state that distances between supports should not exceed twice the tank diameter.

The required tank thickness for a 200-foot, 300-foot, and 400-foot tank was then determined for a five-support configuration by adjusting the bending stress curves and locating their intersection with the allowable longitudinal compressive stress curve; the intersection specifies the minimum thickness required to meet the stress limit.

5.2.6 REQUIRED THICKNESS — The following tank thicknesses were determined for a five-support configuration:

	Configuration	Thickness
a.	200-foot tank	0.2 inches
b.	300-foot tank	0.4 inches
c.	400-foot tank	0.65 inches

Standard fabrication practice for 20-foot diameter tanks calls for a wall thickness of 5/16 inch or greater. It was therefore arbitrarily decided that for the 200-foot tank the minimum wall thickness would be increased to 5/16 inch, although theoretically the thickness could have been 0.2 inches. Utilizing this design, four supports would provide sufficient strength.

5.2.7 DETERMINATION OF STIFFENER SIZE — Design of the ring stiffeners was based on the analysis of 180° arbitrary saddle supports (Appendix B). The circumferential stresses in the stiffener are given by

$$(\sigma_\theta)_{\text{compress.}} = \frac{-0.48\,\overline{Q}}{A} - \frac{0.043\,\overline{Q}\,r}{I/c} \qquad (5\text{-}7)$$

$$(\sigma_\theta)_{\text{tensile}} = \frac{+0.158\,\overline{Q}}{A} + \frac{0.043\,\overline{Q}\,r}{I/c}$$

Zick sets the following limits on the circumferential stiffener stress:

a. on compressive stress, one-half yield stress; and

b. on tensile stress, allowable working stress.

At this point, it was necessary to designate a typical material in order to determine the required stiffener strength (based on the previous stress limits). For this purpose, carbon manganese silicon steel, A516 Gr65, was chosen. Its material strength properties are 65,000 psi ultimate stress, 35,000 psi yield stress, and 16,250 psi allowable working stress. Knowing the limits on circumferential stress in the stiffener, the required cross-sectional area and section modulus are determined from eq. 5-7.

A summary of the tank geometry and design stress versus allowable stresses is presented in table 5-I for each of the three tank sizes studied. Figure 5-4 shows the three representative tank designs drawn to the same scale. The tank configurations are representative of minimum requirements for thickness and number of supports. Flexible tanks are considered to be desirable to prevent "lift off" in grounding and sagging/hogging conditions.

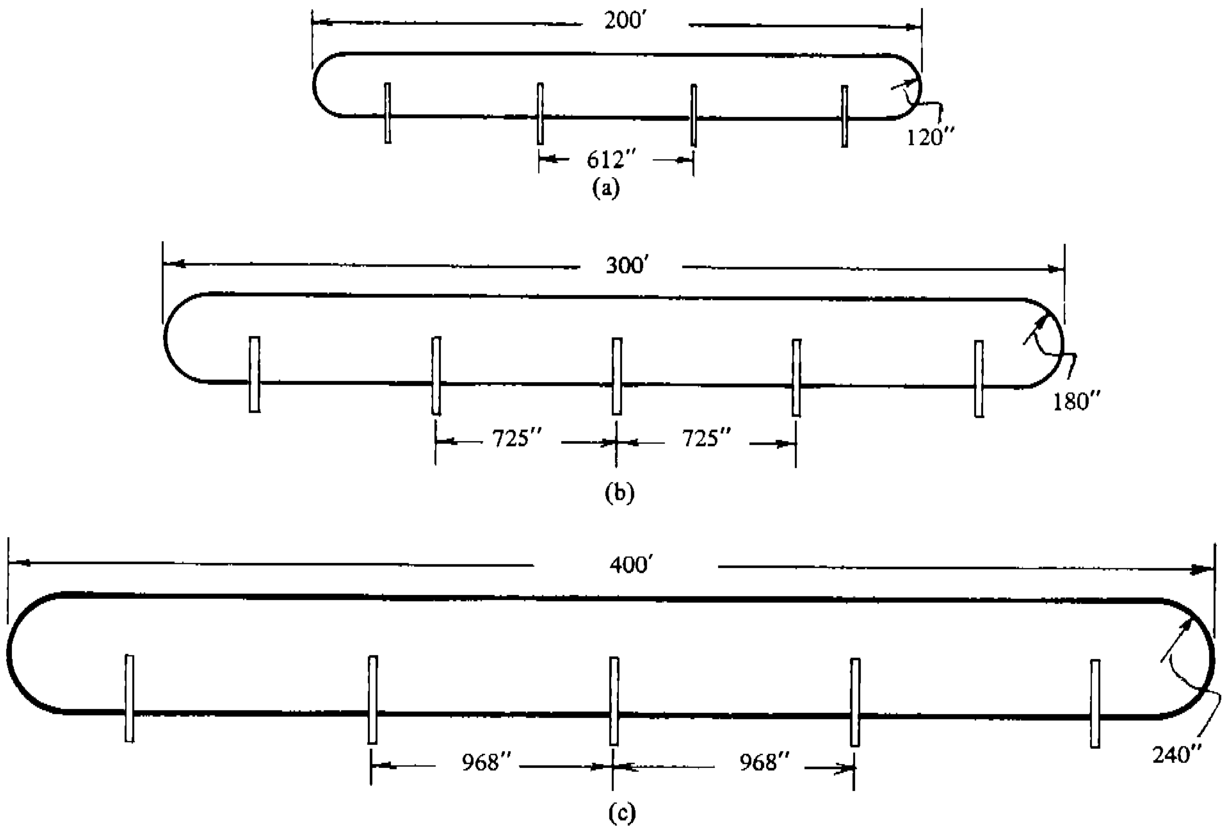


Figure 5-4. Three Representative Tank Designs

Table 5-I. Summary of Tank Configurations Selected for Analysis of Grounding and Sagging/Hogging

	OVERALL TANK LENGTH		
	200 FT	300 FT	400 FT
Tank Radius	120 in.	180 in.	240 in.
No. of Supports	4	5	5
Support Spacing	612 in.	725 in.	968 in.
Width of Supports	12 in.	18 in.	25 in.
Tank Thickness	5/16 in.	.4 in.	.65 in.
Stiffener Sect. Mod.	390 $in.^3$	1875 $in.^3$	6670 $in.^3$
Stiffener Cross-Sect. Area	170 $in.^2$	450 $in.^2$	1000 $in.^2$
Circum. Tank Stress			
Design	6,900 psi	10,600 psi	11,100 psi
Allowable	16,250	16,250	16,250
Long. Tens. Tank Stress			
Design	5,600	6,870	7,600
Allowable	16,250	16,250	16,250
Long. Comp. Tank Stress			
Design	1,760	2,370	3,910
Allowable	4,730	4,130	4,920
Transverse Shear Tank Stress			
Design	4,350	6,890	8,550
Allowable	13,000	13,000	13,000
Circ. Tens. Stiff. Stress			
Design	13,100	13,300	13,300
Allowable	16,250	16,250	16,250
Circ. Comp. Stiff. Stress			
Design	16,900	17,300	18,500*
Allowable	17,500	17,500	17,500

*Exceeds allowable in actual design stiffener size would be increased.

5.3 DESIGN FOR BUCKLING

In between stiffeners, the cylindrical shell is subjected to large compressive bending stress due to the nature of the loading. Consideration must be given, therefore, to the possibility of the shell buckling. The critical compressive stress, σ_{cr}, has been commonly accepted as being 1.3 times the compressive buckling stress (due to uniform axial compression). Such a value was obtained by Flügge.[18] Flügge's result was for a particular shell and buckle geometry and is not generally true, as is shown in reference 19. The results of this study showed that the critical axial compressive stress due to bending is not more than 10 percent greater than the critical stress for a long shell under uniform axial compression, unless the shell is extremely short ($L/r < 0.15$). For relatively large length-to-radius ratios (L/r) and radii-to-thickness ratios (r/t), reference 19 shows that:

$$\sigma_{cr} = \frac{E}{\sqrt{3\,(1-\nu^2)}}\;\frac{t}{r} \tag{5-8}$$

These results show that linear buckling of a circular cylindrical shell due to asymmetric (non-uniform) axial compressive stress distribution will always occur at a load level where the maximum local axial compressive stress equals the uniform axial compressive stress for buckling.

Since the present study is directed to very thin cylindrical shells, initial deviation from the ideal cylindrical surface should be considered. (These may cause buckling at a stress level lower than the theoretical elastic buckling stress.) Timoshenko[20] presents an empirical formula for calculating the ultimate strength of cylindrical shells under axial compression which considers the effect of initial imperfections. This formula is given as:

$$\sigma_{ult} = E\left[\frac{0.6\,\frac{t}{r} - 10^{-7}\,\frac{r}{t}}{1 + 0.004\,\frac{E}{\sigma_{yp}}}\right] \tag{5-9}$$

where σ_{yp} is the yield strength of the material.

Consider a 400-foot tank with a 20-foot radius, an 80-foot span between supports, and a shell thickness of 0.65 inch. Applying eq. 5-8 and eq. 5-9 yields the following:

Assume $E = 30 \times 10^6$ psi, $\nu = 0.3$, and $\sigma_{yp} = 35,000$ psi

Theoretical elastic buckling is:

$$\sigma_{cr} = \frac{30 \times 10^6}{\left[3(1-.09)\right]^{1/2}}\;\frac{0.65}{240} \approx 49,200 \text{ psi}$$

From Timoshenko (reference 20):

$$\sigma_{ult} = 30 \times 10^6 \left[\frac{.6\left(\frac{0.65}{240}\right) - 10^{-7}\left(\frac{240}{0.65}\right)}{1 + .004\left(\frac{30 \times 10^6}{35 \times 10^3}\right)} \right]$$

$$\sigma_{ult} = 10{,}700 \text{ psi}$$

Other test results, given in reference 21, have yielded results similar to those obtained by Timoshenko. Figure 5-5, obtained from reference 21, shows a nondimensional plot of the theoretical elastic buckling curve and an empirical curve based on test data. As can be seen in figure 5-5, cylinders with a slenderness parameter, $\frac{\sigma_{yp}}{E}\frac{r}{t}$, of 0.064 or less can be stressed to their yield stress without buckling whereas cylinders with larger slenderness parameters will buckle at lower stresses.

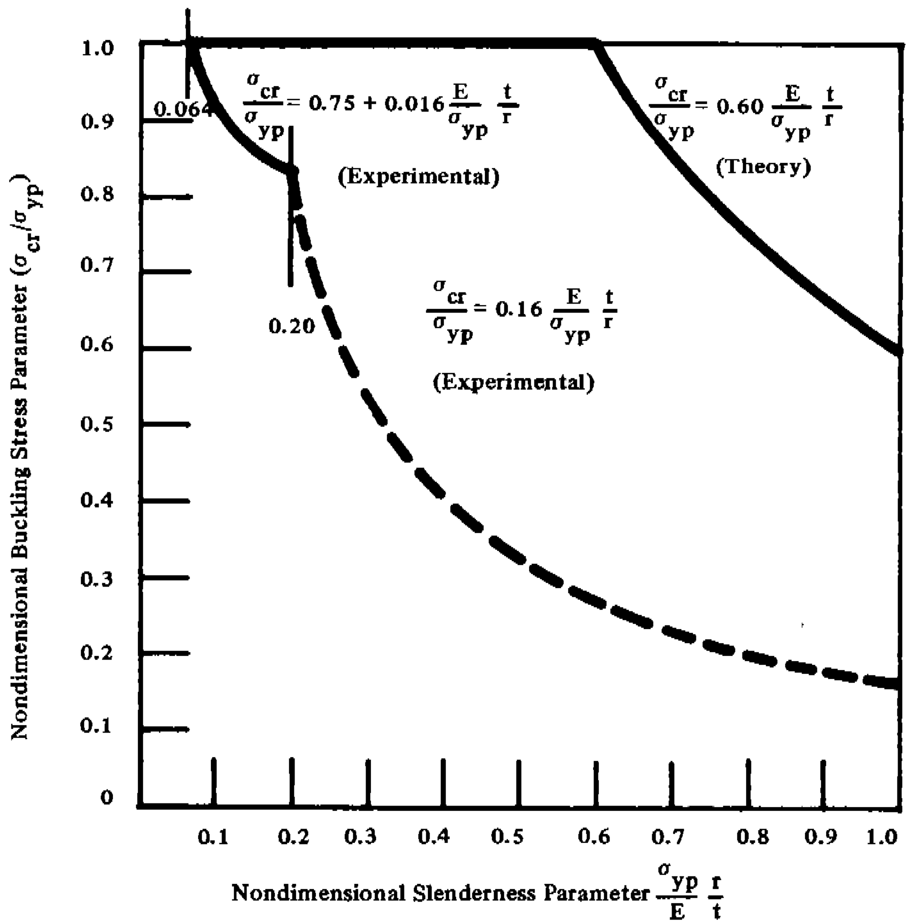


Figure 5-5. Nondimensional Buckling Curve for Circular Tubes in Compression

Applying this curve to our design case, we obtain:

$$\frac{\sigma_{yp}}{E} \frac{r}{t} = \frac{35,000 \text{ psi}}{30 \times 10^6 \text{ psi}} \quad \frac{240 \text{ in.}}{0.65 \text{ in.}} = 0.43$$

And since

$$\frac{\sigma_{cr}}{\sigma_{yp}} = .16 \frac{E}{\sigma_{yp}} \quad \frac{t}{r}$$

then

$$\sigma_{cr} = 13,000 \text{ psi}$$

Timoshenko's results appear to give a lower value of critical buckling stress for this case and, therefore, his is the recommended procedure to follow. On the other hand, applying Zick's critical buckling stress formula to our design case yields a critical compressive stress of 4,920psi. This number appears to be half of the value given by Timoshenko's empirical equation.

5.4 REACTION LOADS DUE TO GROUNDING, SAGGING AND HOGGING

In section 5.2 the rationale for selecting wall thickness and the number and spacing of supports for hypothetical tanks 200, 300, and 400 feet long was presented. In this section, the hypothetical design/analysis procedure will be continued.

It was concluded in section 4.3 that grounding is the most severe condition for river barges and that sagging and hogging loads are most critical in ocean barges (section 4.4). However, in order to analyze these conditions, the tank/barge structure must first be analyzed as a whole. It was necessary therefore to prepare preliminary designs of barges to the point where weight, buoyancy and overall bending strength may be determined. For expediency, the 200-foot tank and a corresponding barge were selected for the grounding calculations, and the 400-foot tank and corresponding barge were selected for the sagging/hogging analysis. Both conditions were analyzed using the Coast Guard procedure aided by a desk calculator.

5.4.1 CONFIGURATION AND CHARACTERISTICS OF BARGES — The general configuration and characteristics of river and ocean barges are shown in figures 5-6 and 5-7. Typical sectional views for the purpose of determining bending strength are shown in figures 5-8 and 5-9. The tank dimensions are shown in figure 5-10.

5.4.2 RESULTS OF GROUNDING CALCULATION — The 200-foot tank configuration, loading diagram, shear curve, and moment curves are shown in figure 5-11. A summary of forces and moments acting on the saddles is given in table 5-II.

5.4.3 RESULTS OF SAGGING/HOGGING CALCULATION — Figures 5-12 and 5-13 give loading, shear, and moment curves for sagging and hogging of the 400-foot tank. Tables 5-III and 5-IV summarize the forces and moments acting at the saddles.

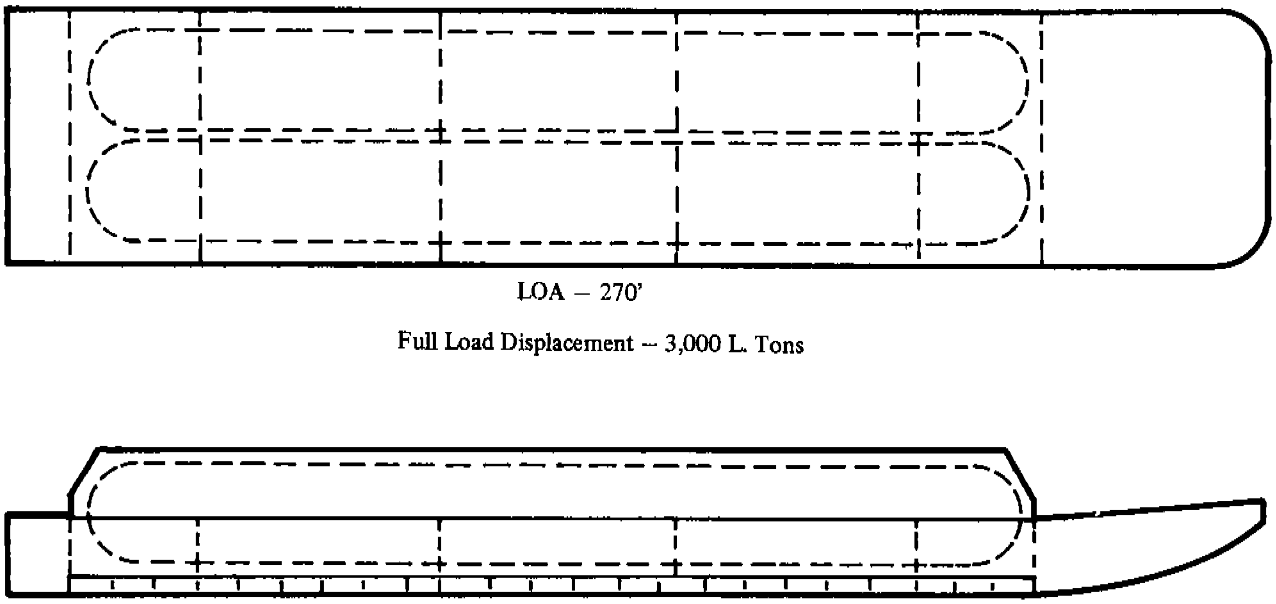


Figure 5-6. Configuration of a Typical River Barge

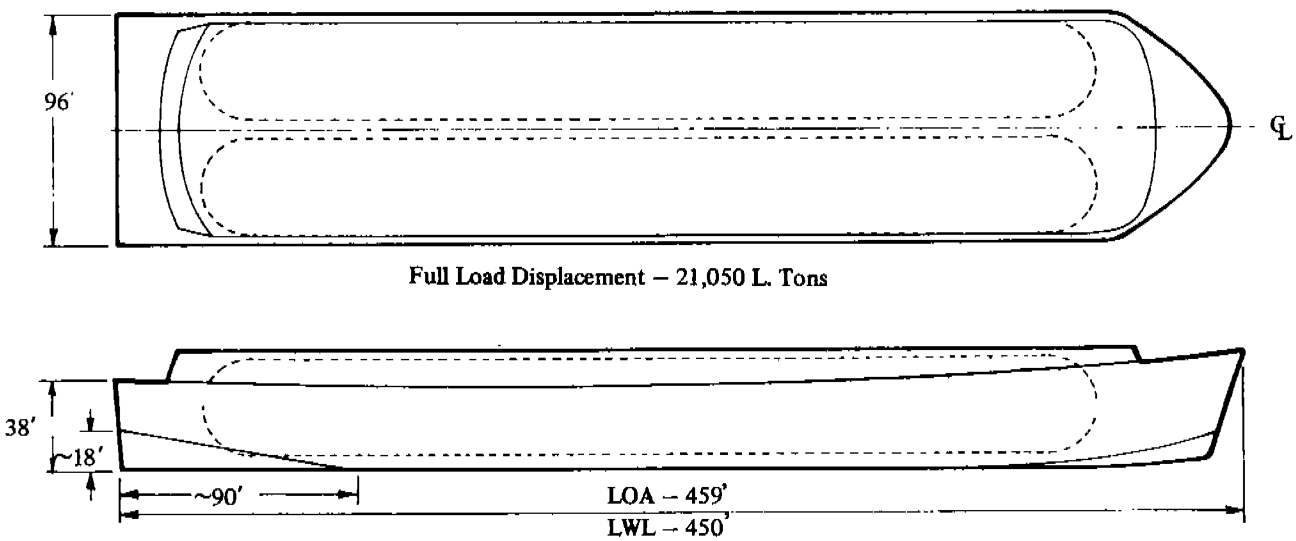
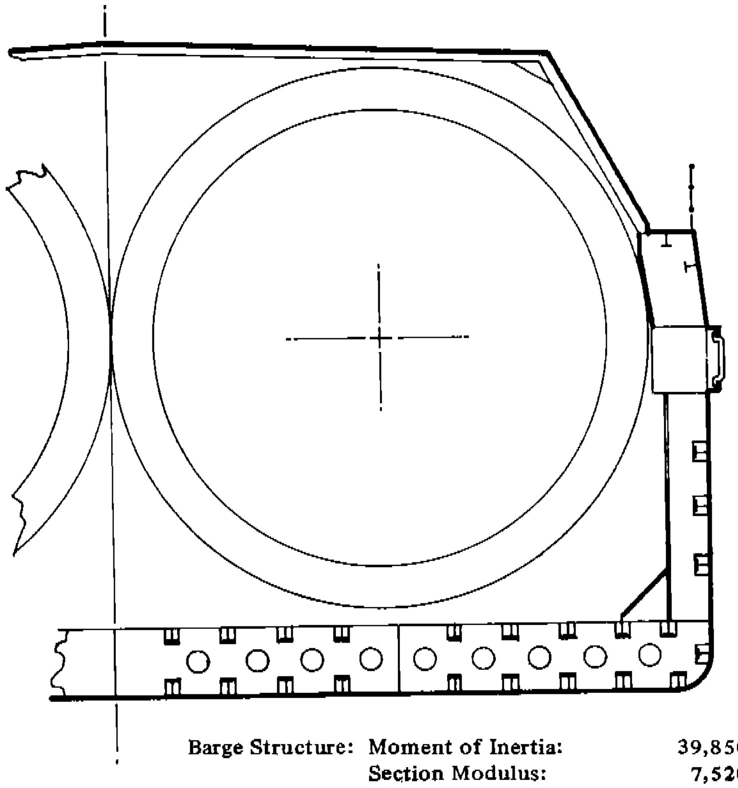
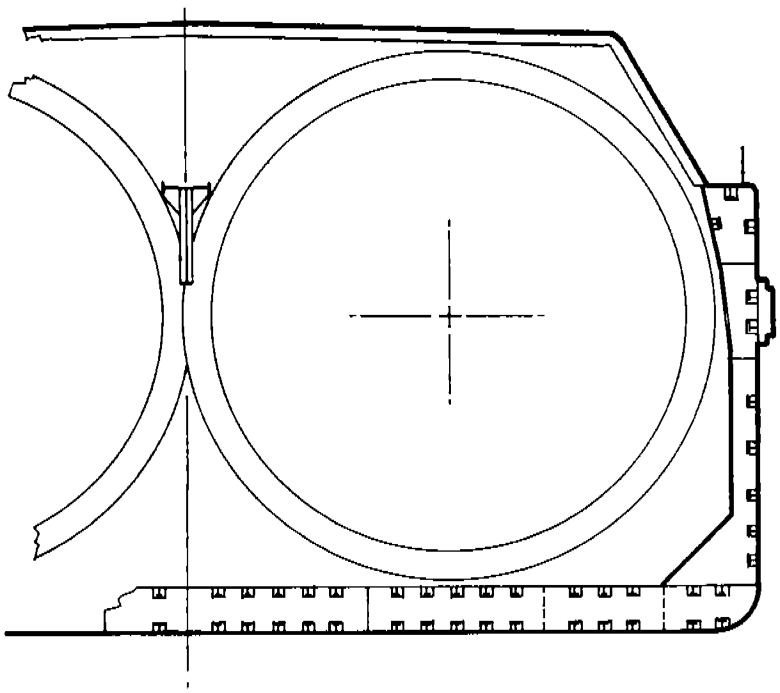


Figure 5-7. Configuration of a Typical Ocean-Going Tank Barge



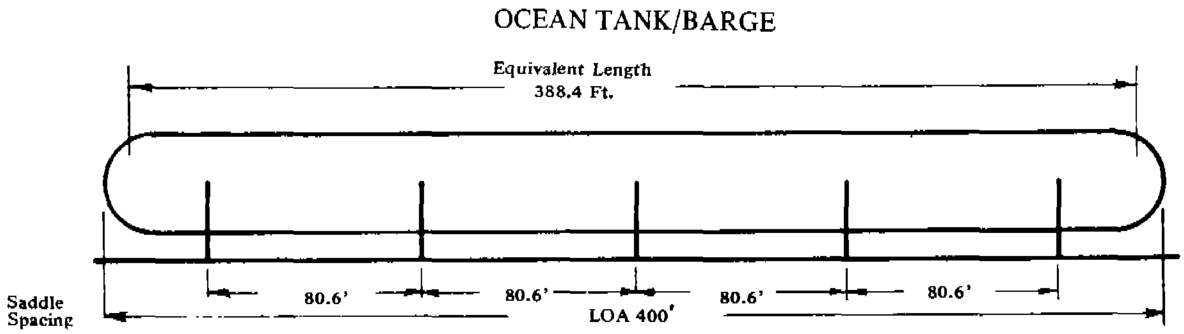
Barge Structure: Moment of Inertia: 39,850 $in.^2 \times ft^2$
Section Modulus: 7,520 $in.^2$ ft

Figure 5-8. Typical River Barge Sectional View

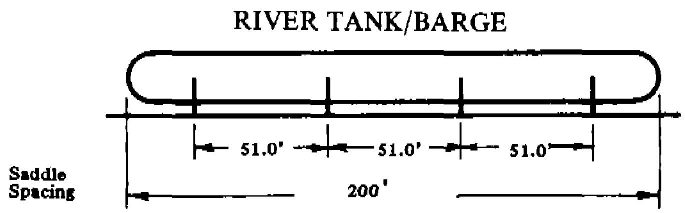


Barge Structure: Moment of Inertia: 732,813 $in.^2 \times ft^2$
Section Modulus: 30,933 $in.^2 \times ft$

Figure 5-9. Typical Ocean-Going Barge Sectional View



Moment of Inertia for One Tank: $I = 196{,}036 \text{ in.}^2 \times \text{ft}^2$
Weight Two Tanks: 1260 L. Tons



Moment of Inertia for One Tank: $I = 11{,}808 \text{ in.}^2 \times \text{ft}^2$
Weight Two Tanks: 260 L. Tons

Figure 5-10. Typical Tank Characteristics and Saddle Supports

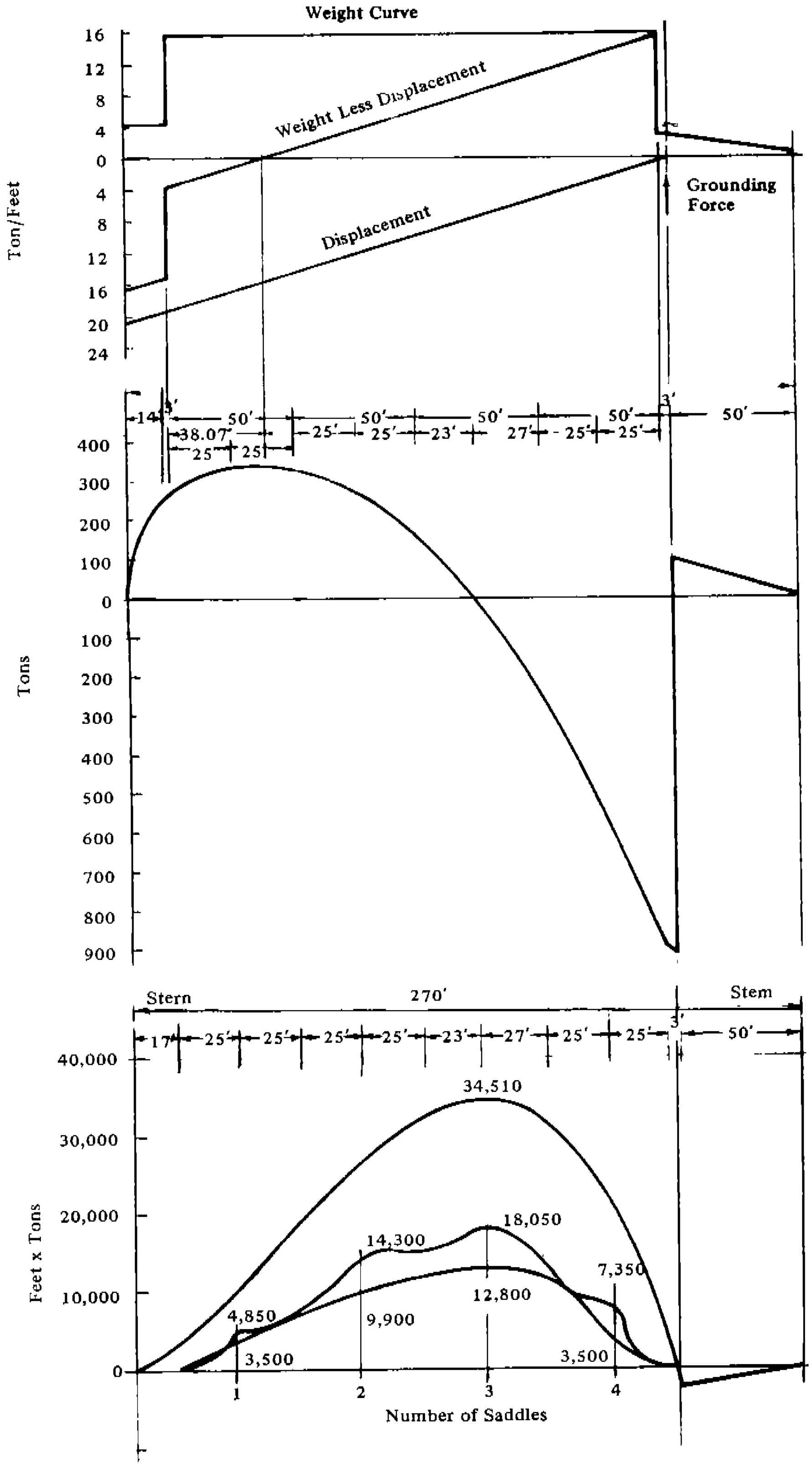


Figure 5-11. Typical Loading, Shear, and Moment Diagrams for a Grounded River Barge

Table 5-II. Support Load Summary and Moment Distribution

Barge Grounded at Fwd Rake Bhd. Two Tanks. S. Water 35 Ft^3/Ton. Saddle Supports 1 thru 4 All Tons @ 2240 lbs (L. Tons.)		TANKS & DW AS. UNIFORM DISTRIBUT. LOAD	CALCULATIONS FOR S. WATER LOAD CONDITIONS 1st	2nd	3rd	4th
			Distrib. of Tanks & DW as Saddle Loads			
Forces At Saddle Support (L. Tons)	No.					
			298	298	298	298
			198	141	121	112
These are the reaction forces at the saddles for full load in two 20 ft. dia. tanks; nominal length 200 feet.	AFT 1		496	439	419	410
			450	507	527	536
			267	256	251	251
	2		717	763	778	787
			381	392	397	397
			506	575	600	609
	3		887	967	997	1006
			142	73	48	39
			298	298	298	298
	FWD 4		440	371	346	337
Moments At Saddle Support (Ft x Tons)	No.					
Moments shown are 37% of the total bending moment. (ft x tons)	AFT 1	3,500	3,500	3,500	3,500	
	2	9,900	12,830	13,900	14,300	
Moments at saddles 1 & 4 are the same and constant due to constant cantilever design of tank ends.	3	12,800	16,300	17,600	18,050	
	FWD 4	3,500	3,500	3,500	3,500	

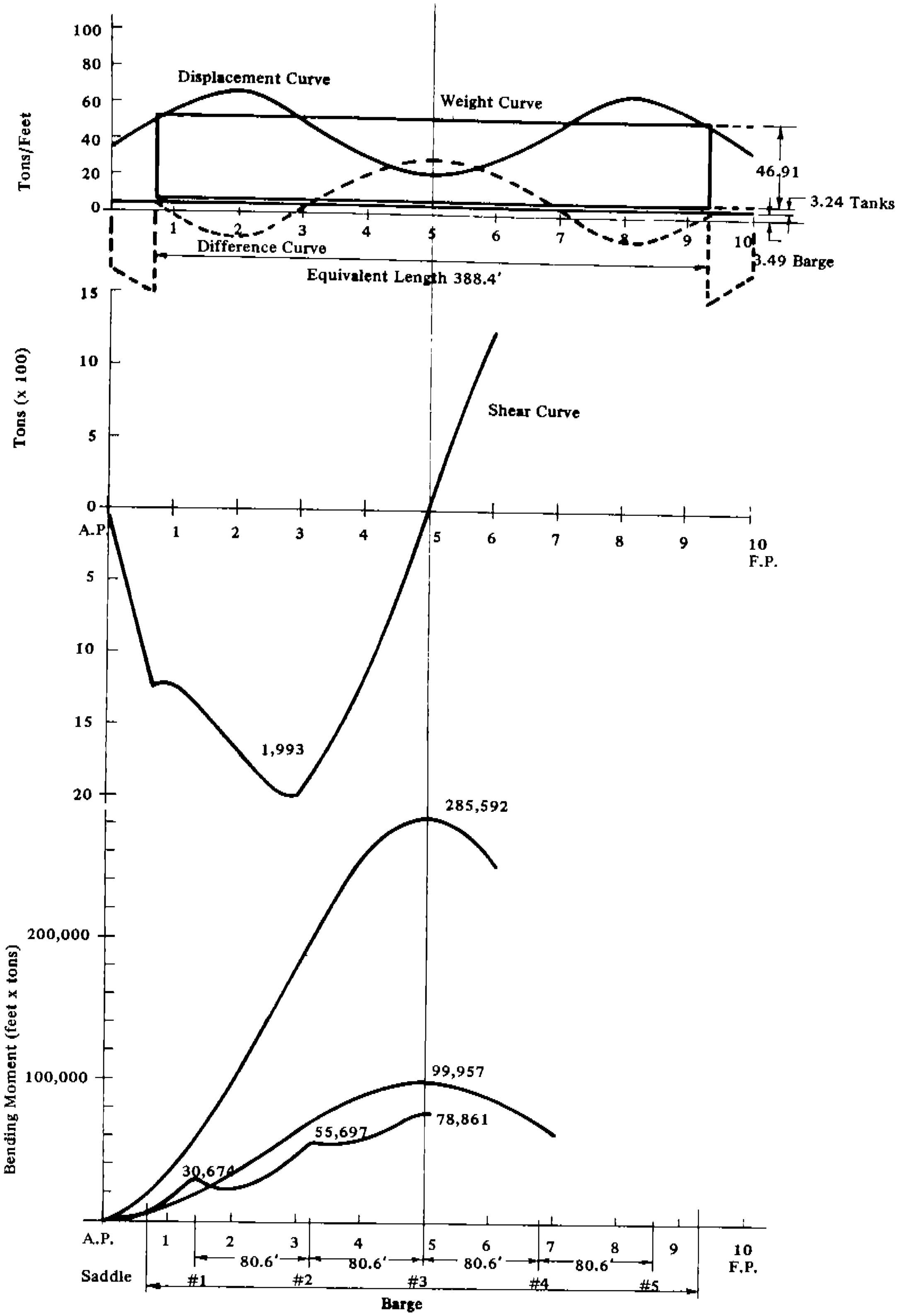


Figure 5-12. **Typical Loading,** Shear, and Moment Diagrams for an Ocean-Going Tank Barge (Sagging Condition)

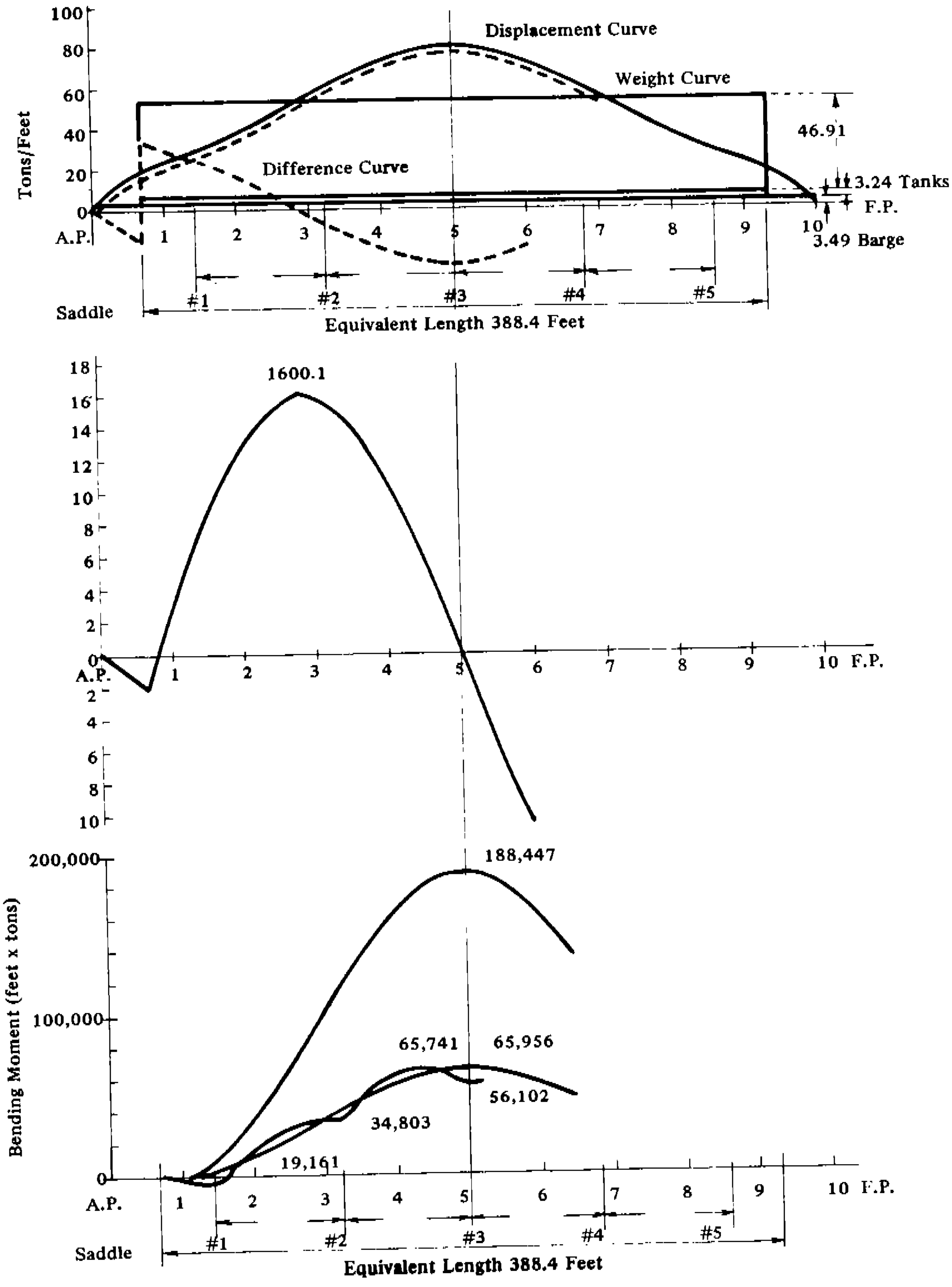


Figure 5-13. Typical Loading, Shear, and Moment Diagrams for an Ocean-Going Tank Barge (Hogging Condition)

Table 5-III. Support Load and Moment Summary — Sagging Condition

Two Tanks S. Water 35 Ft^3/Ton. Saddle Supports: 1 thru 5. All Tons @ 2240 Lbs (L. Tons)		TANKS & DW AS. UNIFORM LOAD	CALCULATIONS FOR S. WATER LOAD CONDITIONS Distrib. of Tanks & DW as Saddle Loads 1st	2nd	3rd	4th
Forces At Saddle Support (L. Tons)	No.					
			1655	1655	1655	
			3198	2887	3004	
These are the reaction forces at the saddles for full load in two 40-ft D tanks, nominal length 400 ft.	AFT 1		4853	4542	4659	
			844	1155	1038	
			2385	2264	2308	
	2		3229	3419	3346	
			1657	1778	1734	
			1657	1778	1734	
	3		3314	3556	3468	
			2385	2264	2308	
			844	1155	1038	
	4		3229	3419	3346	
			3198	2887	3004	
			1655	1655	1655	
	FWD 5		4853	4542	4659	
Moments At Saddle Support (Ft x Tons)	No.					
Moments shown are 35% of the total bending moments (ft x tons)	AFT 1	23, 501	23, 501	23, 501		
	2	71, 400	46, 329	55, 697		
Moments for supports 1 & 5 are constant due to cantilever design at ends of tanks.	3	100, 000	65, 934	78, 861		
	4	71, 400	46, 329	55, 697		
	AFT 5	23, 501	23, 501	23, 501		

Table 5-IV. Support Load and Moment Summary — Hogging Condition

Two Tanks S. Water 35 Ft^3/Ton. Saddle Supports: 1 thru 5. All Tons @ 2240 Lbs (L. Tons)		TANKS & DW AS. UNIFORM LOAD	LOAD CONDITIONS 1st	 2nd	 3rd	 4th
			Distrib. of Tanks & DW as Saddle Forces			
Forces At Saddle Support (L. Tons)	No.					
These are the reaction forces at the saddles for full load in two 40-ft. D. tanks, nominal length 400 ft.	AFT 1		1655 1816 3471	1655 2109 3764	1655 1886 3541	
	2		2226 1699 3925	1933 1809 3742	2156 1752 3908	
	3		2343 2243 4686	2233 2233 4466	2290 2290 4580	
	4		1699 2226 3925	1809 1933 3742	1752 2156 3908	
	FWD 5		1816 1655 3471	2109 1655 3764	1886 1655 3541	
Moments At Saddle Support	No.					
Moments shown are 35% of the total bending moments.	AFT 1	23,501	23,501	23,501		
	2	40,000	26,405	34,382		
Moments for supports 1 & 5 are constant due to cantilever design at ends of tanks.	3	65,956	43,456	56,102		
	4	40,000	26,405	34,382		
	FWD 5	23,501	23,501	23,501		

Section 6

EVALUATION OF STRESSES IN EXISTING AND PROJECTED DESIGNS

In section 5 the rationale for design of typical independent tanks for river and ocean barge applications was discussed. Two hypothetical designs, based on a simplified design/analysis approach, were selected for more detailed investigation of stresses in the tanks.

The objectives of work described in this section are to evaluate the simplified design/analysis techniques with respect to the accuracy of predicting stresses due to critical loads and to examine the validity of an individual stress criteria approach. To accomplish these goals, the representative designs were subjected to computer analysis using linear, thin shell theory applicable to non-symmetrically loaded shells of revolution. A computer program was used for this analysis. A typical structure model used for the computer analysis is shown in figure 6-1. The model represents a theoretical configuration and is not intended as a practical configuration for purposes

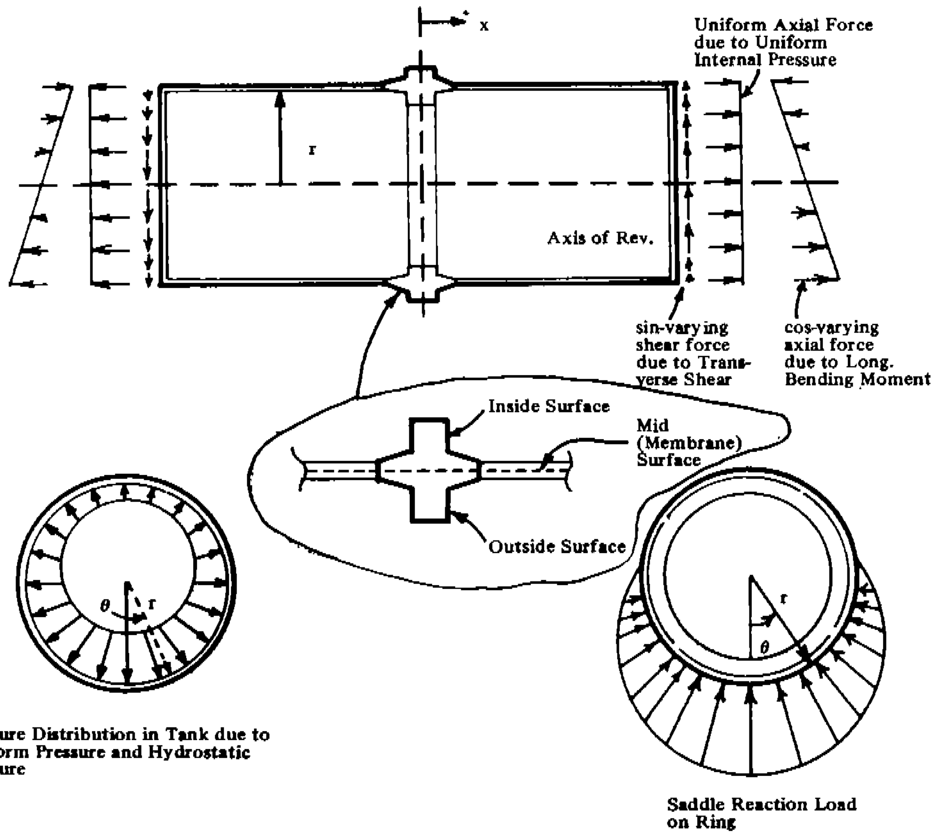
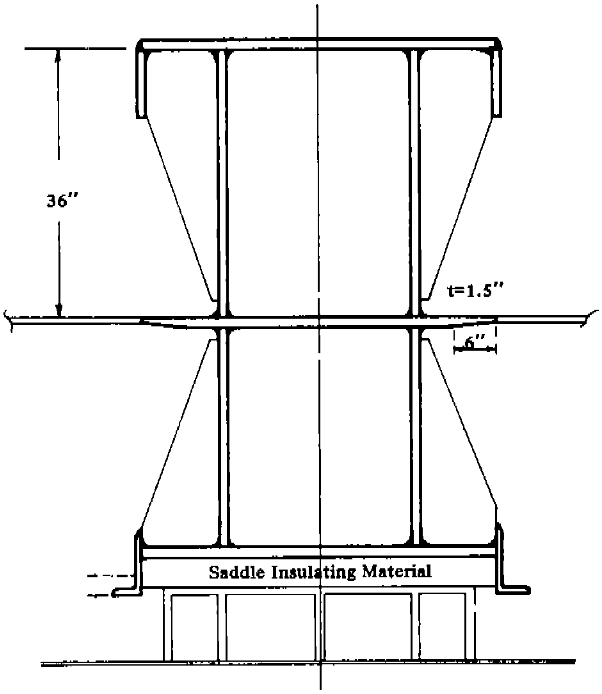


Figure 6-1. Typical Structural Model for Computer Analysis

of construction. The model does represent the proper area and moment of inertia for purposes of analysis. A typical configuration is shown in figure 6-2 which is more adaptable to fabrication but which has the same general properties as the theoretical model.



Section Modulus = 6,940 in.^{3}

Figure 6-2. Typical Tank Reinforcement Adaptable to Fabrication

6.1 STRESS ANALYSIS OF 200-, 300-, and 400-FOOT TANK CONFIGURATIONS

6.1.1 ANALYSIS OF 200-FOOT TANK CONFIGURATION — A 200-foot tank is representative of the size presently being used on inland waterways. The most critical loading condition encountered is grounding. However, grounding occurs infrequently compared to the length of time in normal operation. Therefore the maximum stresses encountered in the grounded condition need not be limited to the allowable working stress (which is approximately one fourth of the ultimate stress for the range of materials presently used in tank construction).

Thus, the 200-foot tank was analyzed for the load conditions of normal operation (including a dynamic load factor), for which it was designed, and for the load conditions of grounding. This second analysis was performed to indicate whether or not the use of conservative stress allowables in the design procedure can compensate for the increased stress levels which typically occur in the grounded condition.

The computer model consists of a section of the tank equal to half the length between supports on either side of a stiffening ring. The 180° saddle support is replaced by the assumed reaction load distribution (given in Appendix A) acting on the stiffening ring.

The shell is loaded by uniform internal pressure and hydrostatic pressure due to the weight of contained liquid. The boundaries, which are at mid span, are not affected by the local bending stress at the stiffening rings (supports). Therefore, a membrane state of stress was assumed in the tank at these locations. It has been shown,[18] however, that this membrane state of stress can be conservatively determined by a beam analogy, i.e., analysis of the cylindrical tank and liquid as a beam supporting a distributed load. The longitudinal normal stress distribution varies about the circumference as the cos θ. The net result of this stress is the longitudinal bending moment. The in-plane shear stress distribution varies about the circumference as the sin θ. The net result of this stress is the transverse shear force. Through the beam analogy, the appropriate stress boundary conditions at mid span for the computer model are obtained.

The most critical stress region (and also the region most crudely analyzed in previous work) is at the stiffening rings over the supports.

High local bending stresses should be expected in this region because of the high concentration of load applied to the tank by the supports and the large change in stiffness from the tank to the stiffening ring. To reduce this local stress, the tank wall was reinforced on both sides of the stiffener; at the stiffener, the thickness of the tank was tripled. The thickness was then tapered over a six-inch length to the normal tank thickness. The size and shape of the reinforcement are based on a small parametric study performed on the 400-foot tank model which is presented under the analysis of the 400-foot tank.

The two load conditions analyzed are shown in table 6-I. For the grounding condition, the load distribution is not symmetric, therefore, only the most severely loaded support was analyzed. For normal operation, each support is loaded approximately the same amount and the assumption of load symmetry about each support was made in the determination of loads. Therefore, the analysis applies to all of the supports for the condition of normal operation.

The longitudinal and circumferential stress distributions on the inner and outer shell surfaces at $\theta = 0$, in the region of the stiffening ring (support), are presented in figures 6-3 through 6-6 for the normal and grounded conditions. High local bending stresses at the shell-stiffener intersection are present in both the longitudinal and circumferential directions. Table 6-II lists the variation of longitudinal normal, circumferential normal, and in-plane shear stress at selected points around the circumference in the shell and in the stiffening ring for both load conditions. The comparison of results of the simplified design/analysis technique and the more sophisticated computer approach is given in section 6-2.

6.1.2 ANALYSIS OF 300-FOOT TANK CONFIGURATION — The 300-foot tank also was analyzed for the condition of normal operation (plus dynamic load factor). The analytic procedure is the same as that employed for the 200-foot tank. The stress distribution and locations of critical areas are similar to those obtained for the 200-foot tank. Because no additional conclusions can be drawn from the analysis, the details are omitted.

Table 6-I. Applied Loads For Normal Operating Condition (Plus Dynamic Load Factor)

200 FT TANK

Load Condition	Normal (Plus Dyn. Load Factor)	Grounded	
Pressure Dist. In Tank			
a. Uniform	10 psi	10 psi	
b. Hydrostatic	3.96 (1 + cos θ) psi	3.23 (1 + cos θ) psi	
Saddle Reaction Load			
a. $-90^{o} \leq \theta \leq 90^{o}$	$-427.6 \cos \theta$ psi	$456.8 \cos \theta$ psi	
b. $90^{o} < \theta < 270^{o}$	0	0	
B.C.s. At Mid Span			
Left / Right	Symmetry of Loading	No Symmetry of Loading	
Axial Force Resultant due to uniform pressure acting on end closures	600 lb/in.	600 lb/in.	600 lb/in.
Axial Force Resultant due to Bending Moment across section	$749 \cos \theta$ lb/in.	$-1131 \cos \theta$ lb/in.	$-1598 \cos \theta$ lb/in.
In-Plane Shear Force Resultant due to Liquid Weight	0	$200 \sin \theta$ lb/in.	$-794 \sin \theta$ lb/in.

400 FT TANK

Pressure Distribution in Tank	
a. Uniform	10 psi
b. Hydrostatic	10.03 (1 + cos θ) psi
Saddle Reaction Load	
a. $-90^{o} \leq \theta \leq 90^{o}$	$-966 \cos \theta$ psi
b. $90^{o} < \theta < 270^{o}$	0

Boundary Conditions at Mid-Span

Left End / Right End	Symmetry of Loading
Axial Force Resultant due to uniform pressure acting on end closures	1200 lb/in
Axial Force Resultant due to Bending Moment across section	$2840 \cos \theta$ lb/in.
In-Plane Shear Force Resultant due to Liquid Weight	0

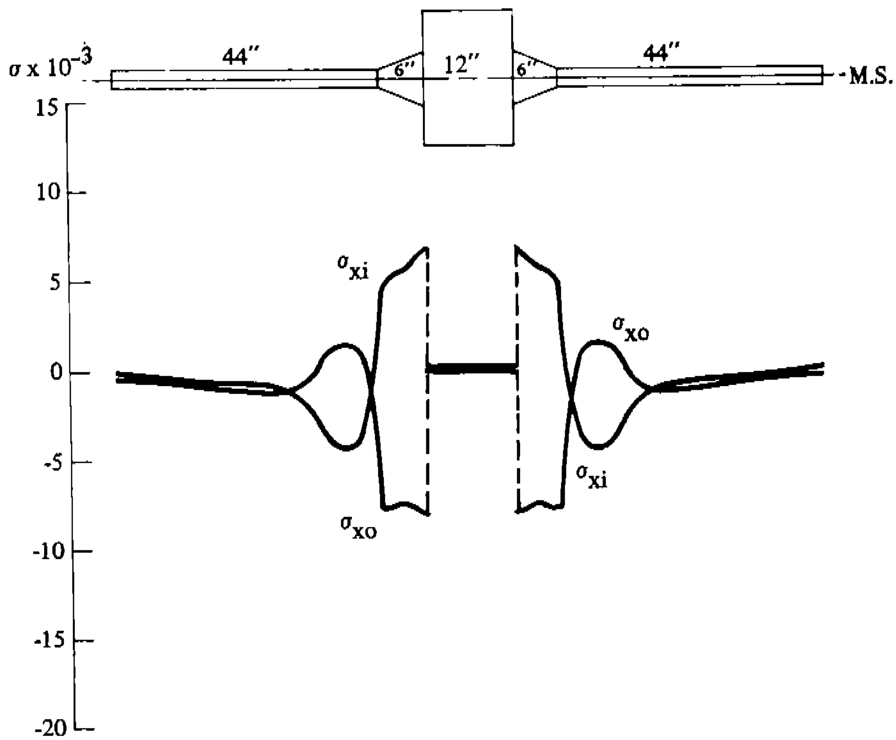


Figure 6-3. 200-Foot Tank — Normal Operation — Longitudinal Stress ($\theta = 0$)

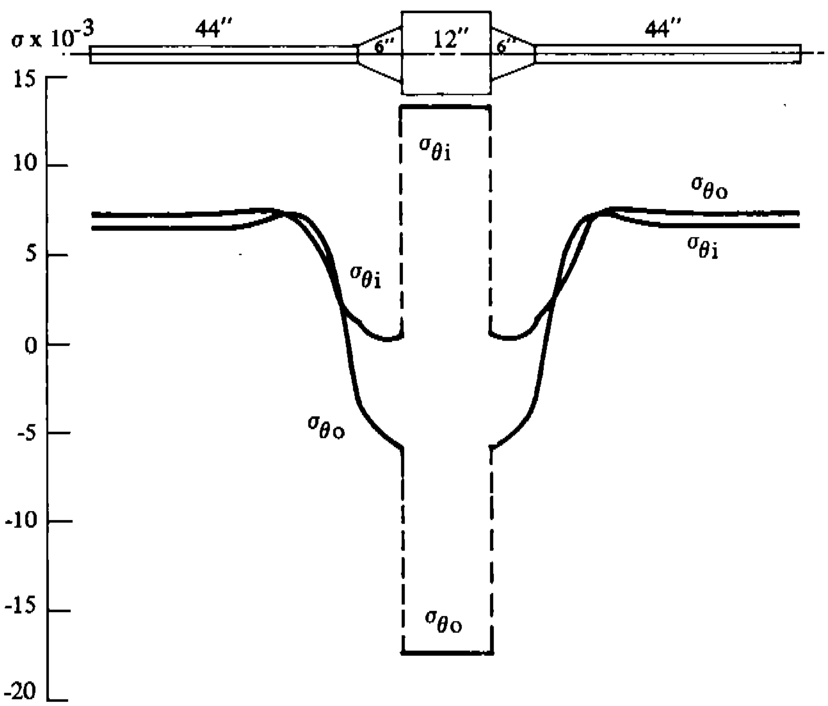


Figure 6-4. 200-Foot Tank — Normal Operation — Hoop Stress ($\theta = 0$)

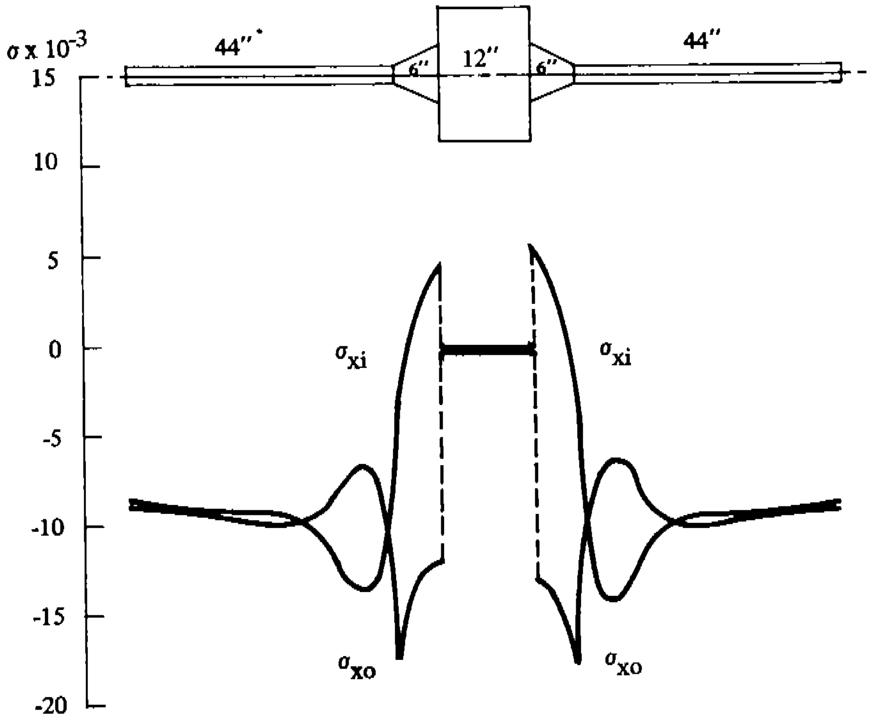


Figure 6-5. 200-Foot Tank — Grounding — Longitudinal Stress ($\theta = 0$)

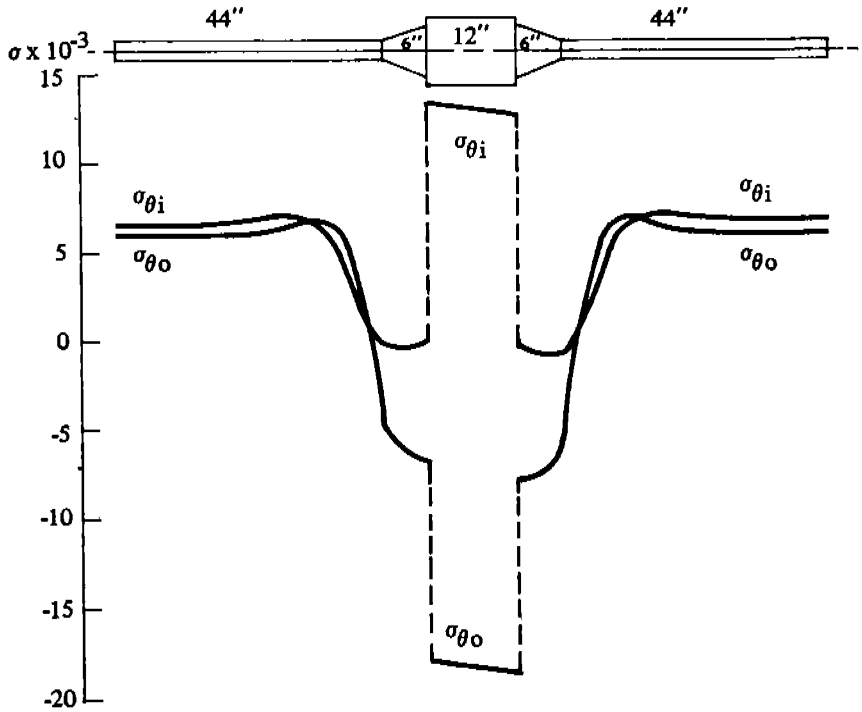


Figure 6-6. 200-Foot Tank — Grounding — Hoop Stress ($\theta = 0$)

Table 6-II. **Maximum Stresses in 200-Foot Tank Based on Computer Analysis**

	*	NORMAL OPERATION				GROUNDED			
		$\theta = 0$	$\theta = 90^{\circ}$	$\theta = 180^{\circ}$	$\theta = 270^{\circ}$	$\theta = 0$	$\theta = 90^{\circ}$	$\theta = 180^{\circ}$	$\theta = 270^{\circ}$
Max. Long. Stresses in Shell (at intersection with stiffening ring)	i	6,700	+4,520	6,190	4,520	5,500	5,183	14,120	5,183
	m	-1,720	1,925	5,550	1,925	-10,500	1,920	14,400	1,920
	o	-8,050	-3,180	4,910	-3,180	-18,300	-3,100	+15,600	-3,100
Max. Circumferential Stresses in Shell (away from shell-ring inter-section negligible bending)	m	7,220	5,550	3,840	5,550	6,750	5,270	3,780	5,270
Max. Shear Stress in Shell (at shell-ring intersection negligible twisting)	m	0	4,400	0	4,400	0	4,750	0	4,750
Max. Circumferential Stress in Stiffening Ring (other stresses not critical)	i	11,870	-16,630	15,270	-16,630	12,800	-17,600	16,300	-17,600
	m	-2,500	-1,300	1,150	-1,300	-2,900	-1,400	1,350	-1,400
	o	-16,950	14,050	-13,000	14,050	-18,300	14,900	-13,700	14,900

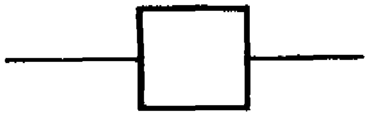
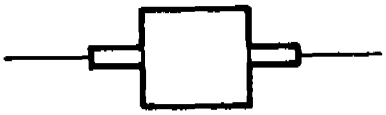
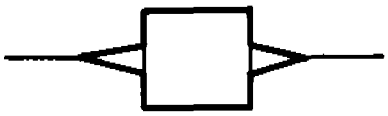
i - inside surface
m middle surface
o - outside surface

6.1.3 **ANALYSIS OF 400-FOOT TANK CONFIGURATION — Chemical tanks as large as 400 feet in length and 40 feet in diameter have been envisioned as efficient carriers for the long-haul transportation of chemicals. Because of the increase in size and the effect of the open-sea environment, the techniques presently employed in design of such tanks are subject to much scrutiny.**

The 400-foot tank was analyzed for the condition of normal operation (plus dynamic load factor) following the approach used for the 200-foot tank. The first computer model of the 400-foot tank had no reinforcement at the shell-ring intersection. Local longitudinal bending stress was nearly equal to the yield stress; successive reinforcement geometries lowered this stress to 14,000 psi. The local longitudinal bending stress for various reinforcement geometries is presented in table 6-III. The applied loads on the 400-foot tank under normal operating conditions, including the dynamic load factor, are presented in table 6-I. The longitudinal and circumferential stress distributions on the inner and outer shell surfaces at $\theta = 0$ in the shell-ring intersection region are presented in figures 6-7 and 6-8. Again, high local bending stresses in the longitudinal and circumferential directions are present at the shell-ring intersection. The variations in longitudinal, circumferential, and in-plane shear stresses at selected points on the circumference in the shell and in the stiffening ring are presented in table 6-IV.

The effect of open-sea environment is a very important factor in determining safe stress levels in the tank because the tank-barge system is subjected to cyclic loading due to the nature of wave motion. The alternating conditions of sagging and hogging result in wide fluctuations in deformation at each point in the tank. Because of this

Table 6-III. Local Longitudinal Bending Stress at Shell-Stiffener Intersection, 400-Foot Tank (Normal Loads)

REINFORCEMENT	PEAK INSIDE STRESS	PEAK OUTSIDE STRESS
None 	+27, 000	-34, 000
Constant (10" Long; 1.3" Thick) 	+19, 000	-22, 500
Taper (10" Long; 2.0" to .65" Thick) 	+11, 000	-18, 500
Taper (12.5" Long; 1.95" to .65" Thick) 	+10, 000	-13, 500

cyclic loading, it is necessary to consider the fatigue properties of tank materials and to determine, by analysis, the stress fluctuations during one load cycle in order to design for the desired tank life. A fatigue analysis of the 400-foot tank, based on stress fluctuations due to sagging and hogging cycles, is illustrated in section 6.4.

6.2 COMPARISON OF DESIGN TECHNIQUES TO COMPUTER ANALYSIS IN THE PREDICTION OF STRESS LEVELS IN THE TANK

6.2.1 200-FOOT TANK CONFIGURATION — The 200-foot tank stress levels predicted by the simplified approach, according to the simplified design analysis procedure of Zick, are given in table 5-I. Stress levels were also obtained by computer analysis for the 200-foot tank. Results of the computer analysis are presented in table 6-II. The simplified design/analysis technique used to determine stress levels in the tank is based on Classical Beam Theory and Classical Membrane Shell Theory. The stress variation, through the tank thickness, is not accounted for in this approach. Only the stress level at the mid-surface of the shell is predicted. Stress distribution through the tank thickness is assumed constant. The computer analysis is based on Classical Thin Shell Bending Theory which permits stress variation through the shell

Table 6-IV. Maximum Stresses in 400-Foot Tank Based on Computer Analysis

NORMAL OPERATING CONDITION

	*	$\theta = 0$	$\theta = 90^{\circ}$	$\theta = 180^{\circ}$	$\theta = 270^{\circ}$
Max. Longitudinal Stress in Shell (at shell-ring intersection)	i	9,900	6,300	6,950	6,300
	m	-3,950	1,850	7,550	1,850
	o	-13,800	-4,600	8,200	-4,600
Max. Circumferential Stress in Shell (away from shell-ring intersection; negligible bending)	m	11,700	7,700	3,700	7,700
Max. Shear Stress in Shell (at shell-ring intersection; negligible twist)	m	0	8,800	0	8,800
Max. Circumferential Stress in Stiffening Ring (most critical stress)	i	10,700	-17,300	15,500	-17,300
	m	-3,750	-1,550	1,550	-1,100
	o	-18,400	13,500	-12,500	13,500

* i = inside surface
m = middle surface
o = outside surface

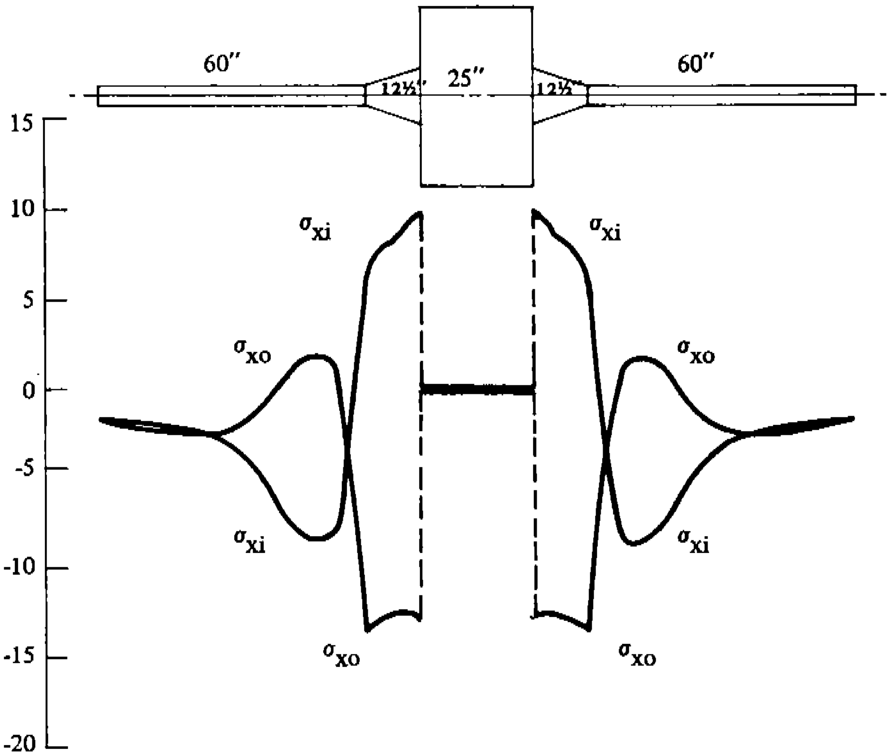


Figure 6-7. 400-Foot Tank — Normal Operation — Longitudinal Stress (θ = 0)

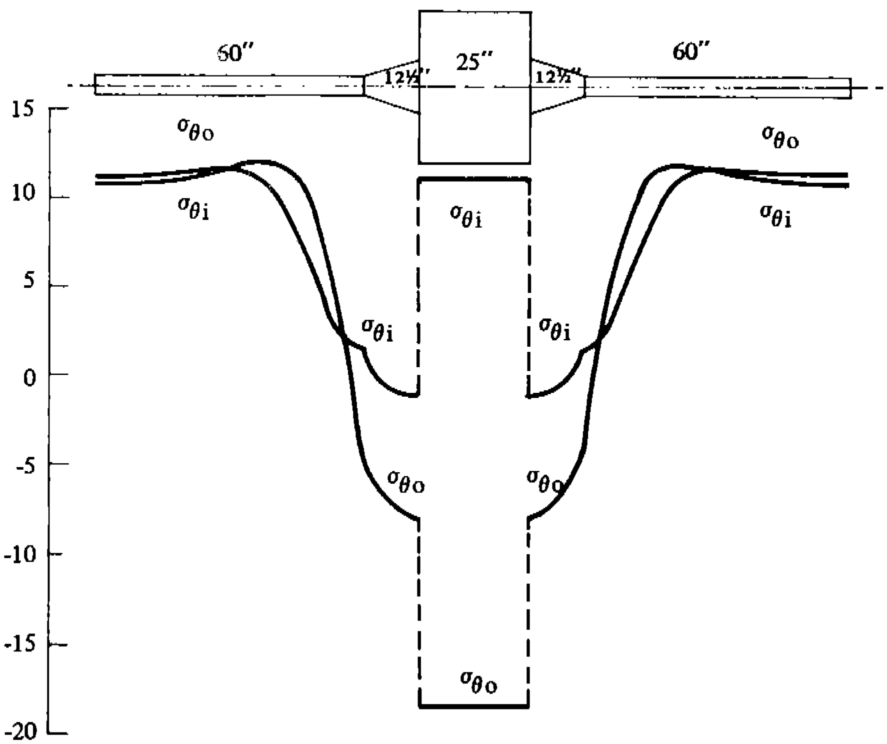


Figure 6-8. 400-Foot Tank — Normal Operation — Hoop Stress (θ = 0)

thickness. Thus, local shell bending effects (as opposed to overall beam type bending) in the shell-ring intersection region can be determined. Examination of table 6-II illustrates this fact.

Table 6-V shows that there is very good agreement for the mid-surface stresses. Therefore, it is concluded that the simplified design techniques are adequate to determine the mid-surface stress levels in the tank. Since the effect of local shell bending is present only in the shell-ring intersection (support) region, the simplified design techniques can accurately predict the stress levels throughout the tank except in the support regions. From figures 6-3 and 6-4, we see that the appreciable effects of local shell bending extend only about 30 inches on either side of the support region (from the middle of the stiffening ring). Therefore, approximately one tenth of the 612-inch span from mid-support to mid-support experiences local shell bending.

In the computer model, the tank wall was reinforced at the intersection with the stiffening ring. This modification was made because of the extremely high local stresses on the inside and outside surfaces of the tank wall. Nowhere in the simplified design/analysis procedure was this stress effect accounted for. Local longitudinal bending of the shell is due to the mismatch in stiffness between the stiffening ring and shell and also due to the high concentration of load acting on the stiffening ring from the saddle support.

Table 6-V. Comparison of Maximum Mid-Surface Stresses

TYPE OF STRESS	DESIGN PREDICTION	COMPUTER ANALYSIS
Circum. Stress	6,900 psi	7,220 psi
Long. Tensile Stress	5,600 psi	5,550 psi
Long. Compress. Stress	-1,750 psi	-1,720 psi
Shear Stress	4,350 psi	4,400 psi

The analysis presented in Appendix B for determining the size of the stiffening ring required to carry the support load is based on Classical Thin Shell Bending Theory, simplified to the case of a ring. This analysis neglects any stiffness contribution due to the attached shell and should therefore be conservative. It does account for stress variation through the ring thickness. Zick's equation for the stress in ring stiffeners is exact in form, but differs in the numerical coefficients; Zick gives coefficients for the cases of 120^{o} and 150^{o} saddles. In the computer model, in order to minimize the effect of circumferential bending of the stiffening ring on the shell, the stiffening ring was symmetrically placed about the shell mid-surface. However, in doing this, the stiffness contribution of the shell in resisting the support load is also minimized. Therefore, the ring alone must resist the support load; this is also the assumption made in Appendix B. The design values of circumferential stress in the stiffening ring were compared with those obtained by computer analysis and the results confirmed this effect (table 6-VI).

Table 6-VI. Maximum Circumferential Stress in Stiffening Ring

	DESIGN VALUE	COMPUTER VALUE
Compressive Stress	-16,900 psi	-16,950 psi
Tensile Stress	15,200 psi	15,270 psi

Again, for circumferential stress in the ring, excellent agreement is shown between the simplified design/analysis procedure and the more sophisticated computer analysis.

6.2.2 300-FOOT AND 400-FOOT TANK CONFIGURATIONS — A comparison of design stress values with those obtained from computer analysis for the 300-foot and 400-foot tanks substantiates the conclusions reached in comparisons for the 200-foot tank. For the 400-foot configuration, a comparison of table 5-I with table 6-IV shows good agreement in stress values at points where stress can be obtained by the simplified approach.

To demonstrate the advantage of positioning the stiffening rings symmetrically with respect to the shell mid-surface, a computer analysis of the 400-foot tank configuration with internal stiffening rings was performed. This imbalance of stiffness about the shell mid-surface induced high local longitudinal and circumferential bending stresses in the shell. The maximum absolute stress increased from 14,000 psi to 23,000 psi. In addition, the expected decrease in stress in the stiffening rings due to the assistance of the shell in resisting circumferential bending was minor - approximately 5 percent.

6.3 ALLOWABLE STRESS CRITERIA

For the purpose of designing typical chemical tanks, the individual stress limits given by Zick were employed. The computer analysis which was performed presents a more comprehensive look at the various stress levels at all points in the tank model.

Based on the results of this analysis, it appears unnecessary to transform the stress state at each point in the shell to principal stresses along principal directions. Examination of tables 6-II and 6-IV indicates that at $\theta = 0, 180^\circ$, the longitudinal stress is maximum while the shear stress is zero. The maximum circumferential stress occurs at $\theta = 0$; as just indicated, the shear stress there is zero. On the other hand, shear stress is maximum at $\theta = 90^\circ, 270^\circ$. At these locations the longitudinal stress is minimum (longitudinal stress due to overall beam bending is zero in the 90°-270° plane). It is concluded that the maximum normal stresses along the coordinate directions and the corresponding maximum shear stress can be compared to individual stress limits without the likelihood of overstress along a principal direction.

According to the Code of Federal Regulations, the allowable stress limits for a tank in the grounded condition are two thirds of the ultimate stress. For the representative material used in this analysis, the allowable stress would be on the order of 40,000 psi. As shown in table 6-II, the maximum stress levels in the 200-foot tank in the grounded condition are well below this allowable stress. It should be noted that the most severe rise in stress in the grounded condition occurred in the longitudinal stress.

This is the result of an increased overall bending moment acting on the most severely loaded support.

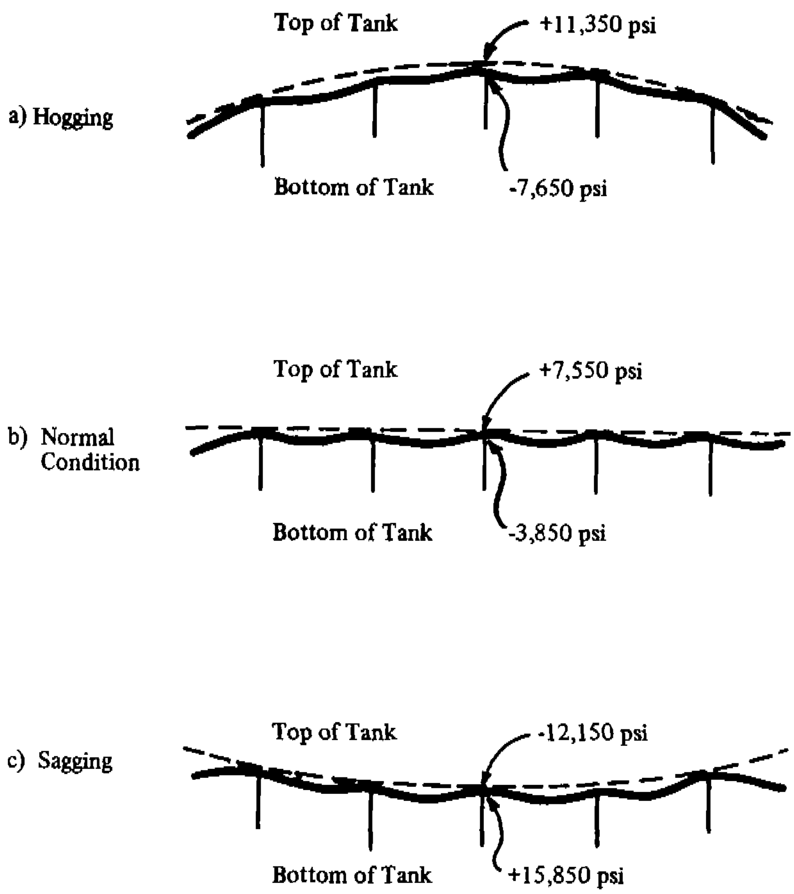
In the grounding condition, the maximum longitudinal compressive stress is increased from -1, 720 psi to -10, 500 psi. Zick's longitudinal compressive stress criterion is based on buckling, and valued at -4, 700 psi. The phenomenon of buckling precludes that of failure by yielding, and therefore it would seem that the tank in the grounded condition would be in danger of buckling. However, as described earlier, the ultimate stress which will produce buckling of such a cylindrical shell has been determined to be -10, 700 psi, based on an analysis by Timoshenko which accounts for initial shell imperfections. This discrepancy certainly deserves additional investigation.

No attempt has been made to judge the magnitudes of the stress limits which are based on the allowable working stress of the tank material.

6.4 DISCUSSION OF FLUCTUATING STRESSES DUE TO SAGGING/HOGGING IN LARGE TANKS

As discussed previously, the use of 400-foot tanks for the transportation of chemicals in coastal waters or open sea introduces the additional effect of cyclic loading due to wave motion. A maximum of about 7. 2 million cycles of hogging and sagging may be expected during a ten-year life period. Of the 7. 2 million wave encounters only a very small number of waves which induce maximum bending will occur. During cycles of hogging and sagging, the longitudinal stress due to bending fluctuates about the stress condition of normal operation.

To illustrate the magnitude of stress fluctuations under severe conditions in a 400-foot tank, the load conditions for the cases of hogging and sagging (determined in section 5) were utilized. Noting that the maximum variation in bending moments occurs at the middle of the tank, a computer analysis was performed for both hogging and sagging for the section of tank half way between supports on either side of the center support. The analysis is analogous to that performed for the normal operating condition of load. Having determined the stress distribution for both extreme cases, the stress intensity (range) and maximum stress at certain critical locations were calculated and are presented in figure 6-9 and table 6-VII. The stress levels shown represent extreme conditions and will occur or be exceeded only a small percentage of the time.



Longitudinal Membrane Stress Cycle in the Shell at the Intersection with the Stiffening Ring (dynamic load factor used in all cases)

Figure 6-9. Investigation of Cyclic Loading for Fatigue Analysis of 400-Foot Tank

Table 6-VII. Local Peak Stress Cycle in the Shell-Ring Intersection

	INSIDE SURFACE				OUTSIDE SURFACE			
	$\theta = 0$	$\theta = 90^{\circ}$	$\theta = 180^{\circ}$	$\theta = 270^{\circ}$	$\theta = 0$	$\theta = 90^{\circ}$	$\theta = 180^{\circ}$	$\theta = 270^{\circ}$
Hogging								
Longitudinal Stress	3,081	6,190	10,131	6,190	-18,342	-2,493	+12,508	-2,493
Normal								
Longitudinal Stress	6,090	6,090	6,930	6,090	-13,820	-2,400	+8,180	-2,400
Sagging								
Longitudinal Stress	21,800	6,036	-9,100	6,036	9,540	-2,340	-14,860	-2,340

Section 7

DISCUSSION OF MATERIALS AND CONSTRUCTION OF PRESSURE VESSELS FOR BULK TRANSPORT OF LIQUID CARGOES ON BARGES

The transport of liquid cargoes in tank barges is regulated by federal law for interstate and overseas shipment. The design, construction and inspection of tanks is governed by Title 46 of the Code of Federal Regulations.

Certain cargoes possessing dangerous or lethal properties are limited by the Code as to the maximum volume or weight which can be carried in a single tank. This is because a tank of such a volume or weight is the maximum that could reasonably be recovered without excessive danger to persons and property should the tank barge suffer a casualty. Such cargoes, and their tanks, are of concern in this study only to the extent that safe handling or equipment for recovery following a barge casualty might be improved in the future.

This discussion is, therefore, directed primarily towards requirements for tanks which contain other than lethal liquids and where no regulations relative to the fluid properties govern the size of the tank. It is, furthermore, primarily concerned with liquids at subatmospheric temperatures and atmospheric pressure where pressure vessel design must be employed and temperature effects on materials must be considered.

This section is not an abstract of requirements pertaining to pressure vessels per se, but an appraisal of those requirements which could affect the construction of very large cargo tanks of pressure vessel design.

7.1 DESIGN/MATERIAL CONSIDERATIONS

Large cylindrical pressure vessels for cargoes being transported at somewhat above atmospheric pressure, theoretically, can have very thin walls. However, at zero positive pressure, the static pressure head of the cargo on the lower portion of a horizontal unstiffened tank can be sufficient to effect considerable deformation. For instance, the pressure exerted by the weight of propane gas on the bottom of a horizontal tank 40 feet in diameter will be on the order of 145 lbs per square foot. The cylinder also can deform from its own weight if unsupported. Therefore, to design a tank which will remain essentially cylindrical and be of minimum weight, a balance must be established between the number and size of stiffeners (internal or external frames), shell thickness, and material. This is, in essence, the purpose of this study. For large tanks, the section modulus, including depth and size of stiffeners and shell thickness, determines the rigidity of the structure and the maximum stresses at the supporting saddles. Design seeks to keep stresses low, hence, material strength properties are secondary to section size. For this reason, high-strength quenched and tempered steels may not be economically justifiable. However, since they have excellent notch toughness and retain their properties of ductility and toughness across welded joints without stress relief, the higher material cost may be offset by fabrication economies.

Minimum shell thickness of 5/16 inch is specified in the Code of Federal Regulations for certain hazardous and dangerous cargoes (in paragraphs 38, 39 and 40 of Subchapter D). For very large tanks, design indicates a greater thickness is necessary for reasonable stress levels, thus this limitation is of no consequence. Since the highest stresses on the tank are found at the saddle supports by which the tank is attached to the barge, local thickening of the shell in these areas will reduce reaction stresses with minimum overall increase in tank weight.

Except where corrosion-resistant materials are used which are not affected by the contained fluids, or the cargo is noncorrosive, most tanks require a corrosion allowance added to the design shell thickness. Paragraph 52.05-12 of Subchapter F specifies the addition of 1/6 the design thickness, or 1/16 inch whichever is less. This addition is generally of small consequence with respect to the cargo capacity of the tank.

7.2 MATERIALS FOR LOW-TEMPERATURE APPLICATIONS

The Code of Federal Regulations, Title 46, Chapter 1, Subchapter F, "Marine Engineering," 1968 edition, designates allowable ferritic materials for low-temperature service in Table 51.24.1. The table refers to ASTM specifications A300-58, A333-63T, A334-63T, A350-61T and A352-60T. While these specifications are all current at this date (1968), each has been updated, and in case of A300, revised to a considerable extent. Therefore, when designing a pressure tank, it must be determined whether the CFR is to be followed to the letter or the latest revision of the applicable specification is to be used.

The U. S. Coast Guard Navigation and Vessel Inspection Circular, No. 7-67, dated 9 November 1967, is a complete (current) guide for the use of all steels (ferritic and austenitic) in all forms — plate, shapes, castings, fastenings, and so forth — for low-temperature service from ambient to below $-320^{\circ}F$. It appears especially valuable in that, to a great extent, it does not tie materials to specifications, but notes chemistries and heat treatments required to provide the strength and toughness for the service temperature. Thus, the designer is free to choose the material best suited to his application with only prudent and reasonable restrictions invoked. Nonferrous materials are not included in Circular 7-67, but may be used at any low temperature upon approval of the application by the Coast Guard Commandant. This is specifically noted in Circular 7-67 and in Paragraph 51.01-85, "Alternative Materials" of Subchapter F, Title 46-Chapter 1 of the Code of Federal Regulations.

With respect to toughness properties of ferritic steels for low-temperature applications, the requirement that fine grain melting practice be employed in making the steel is universally prescribed in specifications. Also, Circular 7-67 emphatically states that where Charpy impact testing is used to evaluate notch toughness, the V-notch specimen only is acceptable. It is stated therein that correlation has been established between the nil-ductility drop-weight test and the Charpy V-notch test and that either of these two methods may be used.

Certain of the ASTM specifications, including A300-63T, specify the Charpy keyhole impact test. Where an ASTM specification is used to designate ferritic steels for barge or ship tanks, and Charpy V or keyhole impact specimens are called for by the specification, the designer must indicate that the requirements of Circular 7-67 for Charpy V-notch tests take precedence.

Those materials which do not undergo a ductile-to-brittle transition with decreasing temperature, such as aluminum and austenitic stainless steel, are, in general, exempt from impact testing. The curve of impact energy versus temperature is a nearly horizontal line to $-320^{\circ}F$, therefore impact tests provide no useful data.

Circular 7-67, Paragraph 4C, notes that evaluation of metal toughness is a field undergoing continued development. This is a reference to the present emphasis on fracture mechanics which seeks to establish quantitative measurements of metal resistance to brittle failure and mathematical analyses of this phenomena. New testing techniques are being evolved which promise to be more significant than the present impact tests, and these will be recognized by the Coast Guard as well as other Code bodies as they are refined and standardized.

The chemical characteristics of certain cargoes transportable in steel tanks prohibit the use of some of the low-alloy, high tensile steels. Ethlene oxide and propylene oxide, for instance, can be safely carried in carbon steel or austenitic stainless steel tanks, but cannot be carried in tanks of copper-bearing low alloy steel, such as Lukens' LT-75 or U. S. Steel's T-1, both excellent low-temperature steels, because of reactivity with the copper. This particular prohibition is noted in Section 40 of Subchapter D, Title 46 of the Code of Federal Regulations, and illustrates that the nature of the cargo must be carefully considered when selecting tank material, in addition to requirements for low-temperature mechanical properties.

Table 7-I is a compilation of ferritic steels, stainless steels and aluminum alloy plate specifications from the American Society for Testing and Materials, together with pertinent mechanical/physical properties, which can be used for sub-ambient temperature cargo tanks. These materials cover the temperature range from ambient to below $-400^{\circ}F$.

7.3 FORMING REQUIREMENTS

The method of forming parts for pressure vessels, in general, is not restricted by Coast Guard or ABS regulations. However, in order to take advantage of allowed mechanical stress relief procedures for completed tanks, parts can be cold formed to only 4 percent plastic strain[22]. If individual parts, such as heads, are strained more than 4 percent through cold work when being fabricated, they must be thermally stress relieved before assembly into the tank in order that the tank may be mechanically stress relieved. It is not clear if this limit on plastic strain applies to cold forming of nonferrous metals; but where a metal is known to strain harden, it may be assumed that thermal stress relief of severely cold worked parts is required.

Tolerances are applied to the formed sections as directed by Paragraphs 56.01-50 and 56.01-75 of Subchapter F of Chapter 1, Title 46 CFR. Out of roundness is limited to 1 percent of the mean diameter. In a 40-foot diameter tank, this amounts to $\pm$ 4.8 inches from the true diameter. Generally speaking, this is a generous tolerance at this diameter. Submarine hulls of this order of magnitude in diameter are held to out-of-roundness tolerances of less than 1 inch.

Mismatch of abutting edges shall not exceed 1/4 of the plate thickness, or 1/8 inch for longitudinal joints and 1/4 inch for circumferential joints, whichever is less. This tolerance requires care in the fabrication of the cylinders, since, in the rolling of plates, it is possible to form cylinders which vary by more than 1/2 inch in di-

Table 7-I Material Properties

DESIGNATION	ALLOY	HEAT TREAT CONDITION	WORKING STRESS -20 TO 650 (PSI)	MODULUS OF ELASTICITY X 10^{o} (PSI)	CHARPY V-NOTCH	COEFF. OF THERMAL EXPANSION (IN./IN./°F)	THERMAL CONDUCTIVITY (FT^2/HR) (FT·°F)	TENSILE STRENGTH (PSI)	YIELD STRENGTH (PSI)	SPEC. HEAT @ 70°F BTU (LB/°F)	DENSITY LBS (CU IN.)
03 B A203 B	2 1/2-3 1/2% Ni Steel	Normalized	17,500	29-30	15 ft lbs @ -75°F	.0000064	24.2	70,000-85,000	40,000	0.11-0.12	.283
A203 E	2 1/2-3 1/2% Nickel Steel	Normalized	17,500	29-30	15 ft lbs @ -150°F	.0000064	24.2	70,000-85,000	40,000	0.11-0.12	.283
A353	9% Nickel Steel	Double Normalized & tempered	23,750	29-30	25 ft lbs @ -320°F	.00000527 (-58 to 32°F)	15.2	100,000-120,000	75,000	0.11	.283
A410	Chrome-Copper-Nickel-Aluminum Alloy	Normalized	15,000	29-30	15 ft lbs @ -150°F	.0000064	30	60,000	30,000	0.11	.283
A516 Gr 55	Carbon-Manganese Silicon Steel	Normalized	13,750	30	15 ft lbs @ -50°F	.0000064	30	55,000-65,000	30,000	0.11	.283
A516 Gr 60	Carbon-Manganese Silicon Steel	Normalized	15,000	30	15 ft lbs @ -50°F	.0000064	30	60,000-72,000	32,000	0.11	.283
A516 Gr 65	Carbon-Manganese Silicon Steel	Normalized	16,250	30	15 ft lbs @ -50°F	.0000064	30	65,000-77,000	35,000	0.11	.283

Table 7-I (Cont.)

DESIGNATION	ALLOY	HEAT TREAT CONDITION	WORKING STRESS -20 TO 650 (PSI)	MODULUS OF ELASTICITY X 10^{6} (PSI)	CHARPY V-NOTCH	COEFF. OF THERMAL EXPANSION (IN./IN./°F)	THERMAL CONDUCTIVITY (FT^2/HR) (FT/°F)	TENSILE STRENGTH (PSI)	YIELD STRENGTH (PSI)	SPEC. HEAT @ 70°F BTU (LB/°F)	DENSITY LBS (CU IN.)
A516 Gr 70	Carbon-Manganese Silicon Steel	Normalized	17,500	30	15 ft lbs @ -50°F	.0000064	30	70,000	38,000	0.11	.283
A517, all grades	Chrome, Nickel and molybdenum low alloys	Q & T	28,750 (150°F & lower)	30	15 ft lbs @ -50°F	.0000064	30	115,000-135,000	100,000	0.11	0.283 0.289
A537 Gr A	Carbon-Manganese-Silicon Steel	Normalized	17,500	30	15 ft lbs @ -75°F	.0000064	30	70,000-90,000	50,000	0.11	.284
A537 Gr B	Carbon-Manganese-Silicon Steel	Q & T	20,000	30	15 ft lbs @ -75°F	.0000064	30	80,000-100,000	60,000	0.11	.284
A538 Gr A	18 Nickel Maraging Steel	Precipitation hardened	52,500	26.5-27.5	By agreement between mill and purchaser	.00000597	11.3	210,000	200,000-235,000	—	0.29
A538 Gr B	18 Nickel Maraging Steel	Precipitation hardened	60,000	26.5-27.5	By agreement between mill and purchaser	.00000597	11.3	240,000	230,000-260,000	—	0.29

Table 7-1 (Cont.)

ALLOY	HEAT TREAT CONDITION	WORKING STRESS -20 TO 650 (PSI)	MODULUS OF ELASTICITY X 10^{0} (PSI)	CHARPY V-NOTCH	COEFF. OF THERMAL EXPANSION (IN./IN./°F)	THERMAL CONDUCTIVITY (FT^2/HR) (FT/°F)	TENSILE STRENGTH (PSI)	YIELD STRENGTH (PSI)	SPEC. HEAT @ 70°F BTU (LB/°F)	DENSITY LBS (CU IN.)
18 Nickel Maraging Steel	Precipitation hardened	70,000	26.5-27.5	By agreement between mill and purchaser	.00000597		280,000	275,000-305,000	—	0.29
Chrome Moly Alloy	Q & T	26,250	30	By agreement between mill and purchaser	.0000064	16.6	105,000	85,000	.115	.283
Chrome Moly Alloy	Q & T	28,750	30	By agreement between mill and purchaser	.0000064	16.6	115,000	100,000	.115	.283
Nickel-Chrome-Molybdenum Alloy Steel	Q & T	22,500	29	By agreement between mill and purchaser	.0000064	25	105,000-125,000	85,000	.11	.284
Nickel-Chrome-Molybdenum Alloy Steel	Q & T	28,750	29	By agreement between mill and purchaser	.0000064	25	115,000-135,000	100,000	.11	.284
9% Ni Alloy Steel	Q & T	25,000	29	25 ft lbs @ -150°F	.0000064	15.2	100,000-120,000	85,000	.11	.283

Table 7-1 (Cont.)

DESIGNATION	ALLOY	HEAT TREAT CONDITION	WORKING STRESS -20 TO 650 (PSI)	MODULUS OF ELASTICITY X 10^0 (PSI)	CHARPY V-NOTCH	COEFF. OF THERMAL EXPANSION (IN./IN./$^\circ$F)	THERMAL CONDUCTIVITY (FT^2/HR) (FT/$^\circ$F)	TENSILE STRENGTH (PSI)	YIELD STRENGTH (PSI)	SPEC. HEAT @ 70°F BTU (LB/$^\circ$F)	DENSITY LBS (CU IN.)
A240 Type 302 304 316 321 347	18-8 Cr-Ni Stainless Steel	Annealed	18,750	28-30	Ductile to cryogenic temps.	.0000092	9.5	75,000	30,000	0.12	.287-.292
A240 Type 304L 316L	18-8 Low Carbon Stainless Steel	Annealed	17,500	28-30	Ductile to cryogenic temps.	.0000092	9.5	70,000	25,000	0.12	.287-.292
A240 Type 410 430	Chrome Stainless	Annealed	16,250	29	Ductile to cryogenic temps.	.0000055	14	65,000	30,000	0.11	.28
B209 5052	Aluminum Alloy	O-Temper	6,250	10.1	Ductile to cryogenic temps.	.000012	80	25,000	9,500	—	.097
B209 5083	Aluminum Alloy	O-Temper	10,000	10.3	Ductile to cryogenic temps.	.000012	68	40,000	18,000	—	.096
B209 5086	Aluminum Alloy	O-Temper	8,700	10.3	Ductile to cryogenic temps.	.000012	73	35,000	14,000	—	.096
B209 5456	Aluminum Alloy	O-Temper	10,500	10.3	Ductile to cryogenic temps.	.000012	68	42,000	19,000	—	.096
B209 6061	T6 Temper, Welded	O-Temper	6,000	10.0	Ductile to cryogenic temps.	.000012	99	24,000 (Welded)		—	.098

ameter. For instance, a 40-foot diameter cylinder may actually be more than 1/2 inch too small or too large. Careful measurement of each rolled plate and weld joint setup, with allowances for or restraint of contraction, will circumvent problems in this respect.

7.4 WELDING CONSIDERATIONS

Welding procedures for all materials used in tanks which will operate at low temperatures must include consideration of notch toughness. In addition to the tensile and bend test specimens used to qualify a welding procedure, specimens are required for toughness testing by either the Charpy V-notch impact test or the nil-ductility drop weight test. Similarly, production weld testing performed in accordance with Section 56 of Title 46, Chapter 1, Code of Federal Regulations, must include one of these toughness tests.

Welding filler metal is restricted to those compositions capable of passing impact or drop-weight tests. ASTM specification A233 (Mild Steel Covered Arc Welding Electrodes) shows that the E xx 12, 13, 14, 20 and 24 classification electrodes cannot be used since no toughness requirement is imposed on metal deposited by these electrodes. Similarly, for low-alloy steel, flux-covered, arc welding electrodes, only the E xx 15, 16 and 18 classifications (low hydrogen and low hydrogen iron powder coatings), except all E 70 xx classes, are required to pass an impact test. Thus, only these would be acceptable.

Bare wire for submerged arc welding and inert gas shielded welding (ASTM A558 and ASTM 599, respectively) is also classified to include or exclude notch toughness tests. Only those grades subject to impact requirements compatible with the plate steel for the design service temperature can be used for low-temperature tanks.

For radiographic quality welding, as is required for Class I and Class II tanks, a certain amount of welding electrode control is required, especially with regard to moisture content in the covering of flux-coated electrodes and surface cleanliness of spooled automatic welding wire. This is especially true for the higher tensile strength electrodes, where absorption of atmospheric humidity can lead to entrapment of hydrogen in the welds with subsequent underbead cracking and hydrogen flakes, or "fish eyes." Prevention of hydrogen entrapment requires baking of the electrodes and holding them in a heated oven until they are to be used. In some cases, especially in shipyards, electrodes may remain out of the oven in the welder's possession for a limited time only — four to six hours — before being rebaked to drive out absorbed moisture.

For aluminum tanks, it must be remembered that the strength of the weld governs the strength of the structure. The magnesium-alloyed, corrosion resistant, 5000 series aluminums have the best weld strength-to-plate strength ratio. The alloy 6061, and other heat-treatable aluminums, while capable of high strength, are ineffective since the welds of large tanks cannot be suitably heat treated. In the as-

7.5 STRESS-RELIEVING CONSIDERATIONS

Paragraph 56.01-70 in Subchapter F on Marine Engineering in the Code of Federal Regulations states that all Class I pressure vessels (unless specifically exempted by other sections of the subchapter) shall be stress relieved. The following paragraphs indicate that only thermal stress relieving is allowed. This would appear to effectively limit the size of tanks for Class I service since thermal stress relieving of very large tanks can be extremely expensive. If a furnace which would accommodate the diameter of the tank does not exist, one must be built and its cost added to the tank cost. Furthermore, since large tanks are assembled on the barge hull because they are too unwieldy to handle when assembled, circumferential welds joining stress-relieved cylindrical sections must be locally stress relieved. This can present formidable problems of uniform heat application and containment to attain stress-relieving temperatures.

Class II vessels, which are the primary concern of this study, must also be stress relieved in many instances. Exceptions to the stress-relieving requirements exist however, which make construction of jumbo tanks practical. Stress relieving is required of mild steel (carbon-manganese-silicon type) only if the shell thickness exceeds 1.25 inches (assuming large tanks over 20 feet in diameter). Alloy steel under 0.58 inch in thickness is exempt from stress relief. Mechanical stress relief, effected by hydrostatic pressurization, is permitted by Merchant Marine Technical Note 7-64 of 3 December 1964 for Class II and Class III vessels. Since mechanical stress relief is the only practical method to employ on very large tanks, the limits prescribed for its use by Technical Note 7-64 should be considered.

The first limitation noted is that the yield strength of the material must be less than 80 percent of the tensile strength. This limitation might apply to many quenched and tempered steels such as ASTM A517, A542 and A543.

As pointed out earlier under Design Considerations, the use of high-strength steel may not be attractive where section modulus, to minimize deflection and stress, is required rather than tensile strength to permit high stress. If, on the other hand, unwanted weight can be eliminated by higher allowable stress, the use of quenched and tempered steels might be very desirable. In such a situation, and in the case of a very large tank, the prohibition against mechanical stress relief of these materials and the present requirement to thermally stress relieve alloy steel over 0.58 inch might be circumvented on the basis of experience and testing which indicates these steels will perform satisfactorily without any stress relief.

In 1966, MPR Associates published a report entitled "Technical Justification for Use of Ni-Cr-Mo Quenched and Tempered Steel in Class B Nuclear Vessels." This report was aimed at the use of Q and T steel for nuclear reactor containment vessels which, because of their size, should be made as light as possible. The function of the containment vessel is to provide maximum safety in the event of malfunction of a nuclear reactor by containing a sudden rise in pressure. These vessels cannot be benefited by thermal stress relief. Toughness is lowered and a tendency toward heat-affected zone cracking develops. The fact that many large structures such as submarines, bridges, penstocks and storage tanks have been constructed is evidence that the stress relief requirement can be waived.

Stress relief is, in large measure, desirable to enhance fatigue resistance which may be an important factor in a tank on an ocean-going barge subject to wave action. In this respect, it may be noted that in the presence of flaws, high-strength steels of 80, 000 to 100, 000 psi yield strength have an endurance limit of about 25, 000 psi, about the same as carbon steels. However, a carbon steel structure designed to a working stress of 15, 000 psi will be over designed with respect to its endurance limit. A quenched and tempered steel designed to 25, 000 psi working stress will still have an infinite fatigue life, assuming adequate welding control and nondestructive testing is performed to assure freedom from gross flaws. Quenched and tempered steels of 100, 000 to 125, 000 tensile strength are not prone to stress corrosion problems or to brittle fracture. Hence, if designed, constructed and tested to minimize fatigue resistance, they could be expected to perform satisfactorily without stress relief. It may be noted that the normalized 9 percent nickel ASTM A353 steel has been approved under ASME Code Case 1308 for fabrication to 1-1/4-inch thickness without stress relief.

The second limitation of mechanical stress relief is that the design temperature shall not exceed 115°F. This study is concerned with low-temperature cargoes; therefore, the implications of this restriction are not germaine to the present case. Similarly, this study is concerned with cargoes transported at pressures less than 100 psi, and mechanical stress relief is generally not required.

A further restriction on mechanical stress relieving is that the cargoes carried shall have a specific gravity of 1.05 or less. Very few cargoes transportable in Class II tanks would fall in this category, thus the effect of this restriction is not considered significant.

Certain details of construction, especially in reinforced openings, cannot be satisfactorily stress relieved by mechanical means because of inherent notches (partial penetration nozzle welds and single bevel welds with non-removed backing strips). Design must take note to avoid such details which are specifically noted in Memorandum 7-64.

Vessels must be designed to eliminate stress concentrations which might lead to excessive plastic deformation or, possibly, to failure under stress-relieving hydraulic pressure.

Lastly, unless an extensive stress determination is performed with strain indicators during the stress-relieving operation, operating pressure is limited to 40 percent of the maximum design pressure. Generally speaking, since the operating pressure thermal stress relief does. Therefore, the use of materials susceptible to stress corrosion cracking in the presence of certain fluids must be carefully analyzed before waiving thermal stress relief in favor of mechanical stress relief.

7.6 NONDESTRUCTIVE TESTING CONSIDERATIONS

Construction costs can be affected by the amount of nondestructive testing specified for a Class II tank. Credit is given in weld efficiency for radiographic inspection up to 100 percent for flush ground, radiographed, and thermally stress-relieved welds in Class I vessels. Spot radiography of one area per 50 feet of weld, used in Class II vessels, reduces design joint efficiency to 90 percent where weld reinforcement is removed, and lowers the allowable working strength of the plate, requiring additional plate thickness. It therefore becomes necessary to compare the cost of additional plate weight (including effect of such additional weight on cargo capacity) and additional welding with the cost of complete radiography. Vessel weight is usually a very small percentage of cargo weight, and radiography is relatively expensive.

An additional efficiency of only 5 percent can be realized for full radiographic coverage for Class II vessels, and, therefore, where spot radiography is acceptable, it is generally preferred to 100 percent coverage. In order to take advantage of provisions for mechanical stress relieving of tanks operating at $-20^{o}F$ or lower, spot radiography must be extended to include junctions between longitudinal and circumferential welds and for 20 times the plate thickness in each direction of weld from the junction. It should also be noted that credit in weld efficiency is not given for mechanical stress relieving as it is for thermal stress relieving.

7.7 CARGO CHARACTERISTICS

To assess the design problems associated with large tank barges, it is necessary to know the physical characteristics of those materials which are currently being carried in tanks or might be used in sufficient quantities to warrant such transportation.

Table 7-II, entitled "Physical Properties of Gasses," is a compilation of some of the more significant physical characteristics of those gasses which are now transported in the liquid state or which might reasonably be so carried in the future. The Coast Guard toxic rating and classification are also included in the table for those gasses which have been so rated and classified. This data was compiled from both Coast Guard and commercial publications (references 4, 7, and 22 through 27).

The boiling point of the cargo must be considered in tank design since it will influence the choice of tank material, degree of stress relief, and welding practice. The specific gravity of the cargo in the liquid state will, in part, determine the structural loading on the barge. The flammability of the cargo is a measure of the hazard due to leakage from the tank.

The Coast Guard toxic rating is an index of the hazard to personnel from contact

Table 7-II. Physical Properties of Gases

MATERIAL	PRINCIPAL USAGE	B/P @ 1.0 ATM (°F)	SP. GR. - LIQ.	FLAMMABILITY LIMITS % IN AIR BY VOLUME LOWER	UPPER	TOXIC RATING	TYPE OF HAZARD	C.G. CLASSIFICATION	TANK MATERIAL	TANK DES. PRESSURE	VAPOR PRESS. (Psia @ 100°F)	VAPOR DENSITY (lbs/ft³ @ 70°F and 1 atm)	CRITICAL TEMP. AND PRESSURE
1) Methane - CH_4	Fuel Gas	-259	0.415	5	15	1	Explosion Asphyxiation	LNG	Sufficient Notch Toughness Class I or Class II U.P.V.	≥110% of Vapor Pressure of Liquid @ Transport Temp.	-	0.0416	-116.5°F / 673 psia
2) Ethane - C_2H_6	Fuel Gas	-128	0.581	3.22	12.45	1	Explosion	LIG	"	"	-	0.0783	90°F / 708 psia
3) Propane - C_3H_8	Fuel Gas	-44	0.585	2.37	9.50	1	Explosion	LIG	"	"	190	0.1164	206°F / 617 psia
4) N-Butane - C_4H_{10}	Fuel Gas	31	0.600	1.86	8.41	1	Explosion	LIG	"	"	51.6	-	306°F
5) Isobutane - C_4H_{10}	Fuel Gas	11	0.603	1.80	8.44	-	Explosion	LIG	"	"	72.2	-	275°F
6) Ethylene - C_2H_4	Fuel Gas	-155	0.566	3.05 2.7	28.6 34.0	1	Explosion	LIG	"	"	400	0.0729	50°F / 742 psia
7) Propylene - C_3H_6	Fuel Gas	-162 -54	0.610	2.0	11.1	1	Explosion	LIG	"	"	226.4	0.1070	197°F / 667 psia
8) Butylene - C_4H_8		21	0.668	-	-	-					63.05	-	296°F
9) Isobutane - C_4H_8		39	0.635	-	-	-					48.84	-	311°F
10) 1, 2 - Butadiene - C_4H_6	Synthetic Rubber	50.5	0.658	2.0	11.5	1	Explosion	LIG	Sufficient Notch Toughness Class I or Class II U.P.V.	≥110% of Vapor Pressure of Liquid @ Transport Temp.	20	-	339°F / Must be Inhibited
11) 1, 3 - Butadiene - C_4H_6	Synthetic Rubber	24	0.650	2.0	11.5	1	Explosion	LIG	Sufficient Notch Toughness Class I or Class II U.P.V.	110% of Vapor Pressure of Liquid @ Transport Temp.	60	-	306°F / Must be Inhibited
12) Middle East Flare Gas	Fuel Gas	-253	-	~ Methane (72.7%)		-	Explosion	-			-	-	-
13) Venezuelan Dry Gas	Fuel Gas	-256	-	~ Methane (90.9%)		-	Explosion	-			-	-	-
14) Commercial Propane	Fuel Gas	-42	-	~ Propane (96%)		-	Explosion	-			-	-	-
15) Commercial Butane	Fuel Gas	22	-	~ Butane (85%)		-	Explosion	-			-	-	-
16) Acetylene - C_2H_2	Fuel Gas	-119.2*	0.620**	2.5	80.0	-	Explosion Asphyxiation	-			-	0.068	97.4°F / 906 psia
17) Anhydrous Ammonia - NH_3	Fertilizers	-28.1	0.680	16.0	25.0	4	Irritant Explosion	LCG	Steel, Class I or Class II U.P.V.	Vapor Pressure +10psi (Refrig.) 250psi Min. (Non-Refrig.)	212	0.0445	270.32 / 1639 psia Reacts With Cu, Zn and Other Materials
18) Argon - A	Welding	-302.6	1.36	-	(None) -	-	Asphyxiation	-			-	0.1033	-187.6 / 706 psia
19) Carbon Dioxide - CO_2	Refrigerant	-109.3*	1.51*	-	(None)	-	Asphyxiation	-			-	0.1144	87.98 / 1073 psia
20) Carbon Monoxide - CO	Chemical Industry	-312.6	0.810	12.5	74	-	Explosion Poison (Very Dangerous)	-			-	0.0725	-218.2 / 509 psia
21) Chlorine - Cl_2	Chemical Industry	-29.0	1.56	-	(None) -	4	Lung Irritant (Lethal)	LCG	Steel, Class I U.P.V.	300 psi Min.	154.7	0.1863	293 / 1118 psia
22) Carbonyl Sulfide -COS	-	-58.3	0.831	11.9	28.5	-	Explosion Toxic (Nervous System)	-			-	0.1521	216 / 897 psia
23) Fluorine - F_2	Space Ind-Plastics	-306.2	1.1	(No Data)		-	-	-			-	0.0983	-247 / 808 psia

LEGEND

Toxic Rating

- 0 - Non Toxic
- 1 - Practically Non Toxic
- 2 - Slightly Toxic
- 3 - Moderately Toxic
- 4 - Highly Toxic
- \- - No Rating

C.G. Classification

- LFG - Liquified Flammable Gas
- LCG - Liquified Non-Flammable Gas
- A - Grade A Flammable Liquid
- LNG - Liquified Natural Gas
- U.P.V. - Unfired Pressure Vessel

* - Sublimation

Table 7-II. (Cont'd)

MATERIAL	PRINCIPAL USAGE	B/P @ 1.0 ATM (°F)	SP. GR. - LIQ.	FLAMMABILITY LIMITS % IN AIR BY VOLUME LOWER	UPPER	TOXIC RATING	TYPE OF HAZARD	C. G. CLASSIFICATION	TANK MATERIAL	TANK DES. PRESSURE	VAPOR PRESS. (Psia @ 100°F)	VAPOR DENSITY (lbs/ft³ @ 70°F and 1 atm)	CRITICAL TEMP. AND PRESSURE
24) Freon - 12 CCl_2F_2	Refrigerants	-21.6	-	(None)		-	Asphyxiation	-			-	0.316	233.6 / 600 psia
25) Freon - 13 $CClF_3$	Refrigerants	-114.6	-	(None)		-	Asphyxiation	-			-	0.273	83.9 / 561 psia
26) Freon - 22 $CHClF_2$	Refrigerants	-41.44	-	(None)		1	Asphyxiation	LCG			213.4	0.229	204.8 / 716 psia
27) Helium - He	Welding	-452.1	0.124	(None)		-	-	-			-	0.01031	-450.31 / 33.2 psia
28) Hydrogen - H_2	Space Industry	-423.00	0.071	4.1	74	-	Explosion	-			-	0.00522	-399.93 / 188 psia
29) Hydrogen Sulfide - H_2S	Chemical Raw Material	-76.4	0.96	4.3	45.5	-	Explosion Highly Toxic	-			-	0.0892	212.7 / 1307 psia
30) Methyl Chloride - CH_3Cl	Refrigerant and Chemical Raw Mat'l	-11.6	1.0	7.6 8.2(1)	19.0 18.7(1)	2	Explosion Toxic	LIG	Sufficient Notch Toughness Class I or Class II U.P.V.	≥110% of Vapor Pressure of Liquid @ Transport Temp.	116.7	0.133	289.59 / 967 psia
31) Nitric Oxide - NO	Acids, Dyes, Explosives	-241.0	1.27	(None)		-	Toxic	-			-	0.0777	-137.2 / 945 psia
32) Neon - Ne	Illumination	-410.7	1.20	(None)		-	-	-			-	0.0522	-379.66 / 396 psia
33) Nitrogen - N_2	Fertilizer	-320.5	0.808	(None)		-	Asphyxiation	-				0.0725	-232.76 / 492 psia
34) Nitrous Oxide - N_2O	Acids, Dyes, Explosives	-127.3	1.22	(None)		1	Toxic	LCG				0.1146	97.7 / 1054 psia
35) Oxygen - O_2	Space Industry Steelmaking	-297.3	1.14	(None)		-	Accelerate Combustion	-			-	0.0828	-181.84 / 730 psia
36) Sulphur Dioxide - SO_2	Sulphuric Acid	13.8	1.44	(None)		-	Respiratory Irritant	-			-	-	315 / 1142 psia
37) Petroleum Ether	Fuel	68	0.63	1.4	5.9	2	Explosion	A			14	0.187	-
38) Ozone - O_3	Space Industry	-169.4	1.62	(None)		-	Respiratory Irritant Toxic	-			-	-	53.6 / 808
39) Vinylchloride	Plastics	7	0.908	4.0	21.7	3	Explosion Toxic	LIG	Sufficient Notch Toughness Class I or Class II U.P.V.	≥110% of Vapor Pressure of Liquid @ Transport Temp.	84.9	0.161	-
40) Ethylene Oxide	Plastics & Fumigant	51	0.88	3.0	100.0	3	Explosion Toxic	(46 CFR 40)	Carbon or Stainless Steel Class I U.P.V. 75 psi Pressure		40	0.114	-
41) Acetaldehyde	Chemicals, Dyes Photography	70	0.783	4.0	57.0	3	Explosion Irritant	A			28.7	0.114	-
42) Casing Head Gasoline	Fuel	<0		1.4	6.0	1	Explosion	A			14.0	-	Properties May Vary
43) Di Methylamine	Dyes, Drugs, Chemicals	45	0.680	2.8	14.4	3	Explosion	LIG	Sufficient Notch Toughness Class I or Class II U.P.V.	≥110% of Vapor Pressure of Liquid @ Transport Temp.	48.0	0.116	-
44) Ethyl Chloride	Refrigerant Anesthetic	54	0.921	3.8	15.4	2	Explosion	A			24.8	0.166	-
45) Methyl Bromide	Refrigerant Fumigant	40	1.732	13.5	14.5	4	Highly Toxic	Poison B			48, Minimum Design Pressure of Tank	0.244	Attacks Aluminum To Form $AL(CH_3)_3$ Spontaneously Igniting Material - Vapor Extremely Hazardous -

LEGEND

Toxic Rating

- 0 - Non Toxic
- 1 - Practically Non Toxic
- 2 - Slightly Toxic
- 3 - Moderately Toxic
- 4 - Highly Toxic
- \- - No Rating

C. G. Classification

- LIG - Liquified Flammable Gas
- LCG - Liquified Non-Flammable Gas
- A - Grade A Flammable Liquid
- LNG - Liquified Natural Gas
- U.P.V. - Unfired Pressure Vessel

* - Sublimation

The critical temperature and pressure is a useful index of stability. If the cargo achieves this state and the temperature is exceeded, there will be a change of state from liquid to gas regardless of pressure increases. This data, combined with liquid and vapor phase densities, defines the safety blow-off requirements which would be needed to preclude tank rupture, should critical conditions be exceeded.

Table 7-II shows that there are a varying number of physical characteristics associated with materials which are or could reasonably be carried in refrigerated tanks. Accordingly, a basic decision must be made initially to design all such refrigerated tanks for the most severe service, design tanks for one or two similar cargoes, or define ranges of characteristics and design for the most severe service within that range.

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Mr. Harry Townsend, U.S. Salvage Corp.

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Appendix A

INVESTIGATION OF THE ENVIRONMENT OF LARGE OCEAN-GOING BARGES

A-1. INVESTIGATION OF DYNAMIC LOAD FACTORS FOR OCEAN BARGES

The Code of Federal Regulations, Title 46, Subpart 38.05-2, [A-1] states that ocean barges shall be built to withstand the following "Dynamic Loading:"

1. Rolling 30° each side - 120° in 10 seconds
2. Pitching 6° half amplitude - 24° in 7 seconds
3. Heaving L/80 half amplitude - L/20 in 8 seconds (L = length of barge)

To investigate these requirements, the following expressions, which define the maximum values of roll, pitch, and heave from the limits given above (assuming sinusoidal motion), may be written as:

Roll $\phi(t) = \frac{\pi}{6} \sin\left(\frac{\pi}{5} t + \phi_o\right)$ Radians

Pitch $\theta(t) = \frac{\pi}{30} \sin\left(\frac{2\pi}{7} t + \theta_o\right)$ Radians

Heave $Z(t) = \frac{L}{80} \sin\left(\frac{\pi}{4} t + Z_o\right)$ Feet

To compute acceleration at a point, the distance from the center of gravity is multiplied by the second derivative of roll and/or pitch, then heave accelerations are added. For example, combined pitch and heave, in phase, yield bow accelerations of

$$ACC_{BOW} = \left[\left(\frac{2\pi}{7}\right)^2 \frac{\pi}{30}\right] \frac{L}{2} + \left(\frac{2\pi}{8}\right)^2 \frac{L}{80}$$

$$ACC_{BOW} = 0.04952\ L \quad (\text{in ft/sec}^2)$$

By employing standard ship motions equations, it is possible to predict the motions of various barge designs in realistic seaways. These motions are used to evaluate the Coast Guard Structural Requirements.

The methods for determining pitch, heave, and roll and associated results for four barge designs (table A-I), are described in the following paragraphs. From figures A-1, A-2, A-3, and A-4, it is evident that the Coast Guard structural requirements are adequate for all normal weather conditions. Barges designed for constant heavy weather operation (wind speed greater than 30 knots) may require more rigorous specifications however.

In the case of the 240-foot barge, it is evident that the low deadweight-to-displacement ratio results in excessive motions. Figures A-1 and A-2 also indicate that in heavy weather a smaller barge will be in greater trouble. This should be expected.

Note in table A-I that the 440-foot barge carries only one tank.

Table A-I. Barge Characteristics

	NUMBER OF TANKS			
	1	2	2	2
Diameter (ft) (tank)	40	40	30	20
Length (ft) (tank)	400	400	300	200
Barges				
LOA (ft)	442	454	344	244
LWL (ft)	440	450	340	240
Beam (ft)	50	96	76	53
Height (ft)	35	40	30	25
Draft (ft)	26	23.6	17.3	12.2
$\overline{GM}$ (ft)	0.8	24.3	17.4	11.2
Displacement (tons)	11,900	23,940	10,990	3,660

In order to predict the pitching and heaving motions of the barge, use was made of the division's Surface Ship Motions Computer Program. A complete description of this program may be found in reference A-2. The following is a general outline of the way in which the program was employed.

The barge was divided into a number of discrete sections. Through the use of standard ship motion equations, the forces and resulting motions on the entire ship were calculated by summing the effects of the sections. In order to better define the motion of each barge in an ocean environment, a random sea (Neumann) was mathematically formulated. This formulation allows a study of the response of each craft to various input waves. Through the use of the established theory of linear superposition, the responses in each component wave can be summed to give a response spectrum for the entire range of waves which might be encountered in an ocean.

Since the acceleration due to coupled pitch and heave motion is often greatest at the bow, the average value of the 1/10 highest bow acceleration was determined. Also calculated was the average value of the 1/10 highest pitch amplitudes. These have been presented in figures A-1 and A-2.

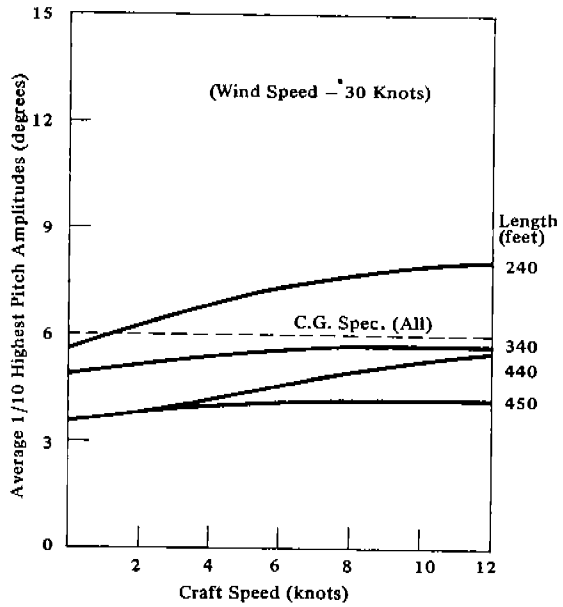

Figure A-1. Pitch vs. Speed (Neumann Sea)

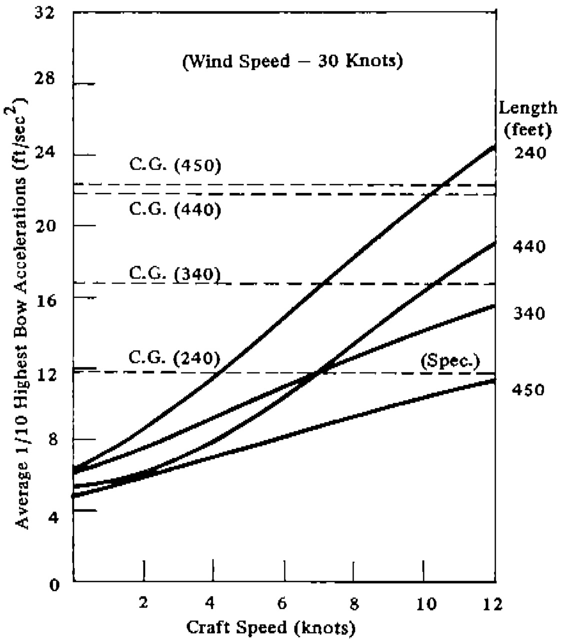

Figure A-2. Bow Acceleration vs. Speed (Neumann Sea)

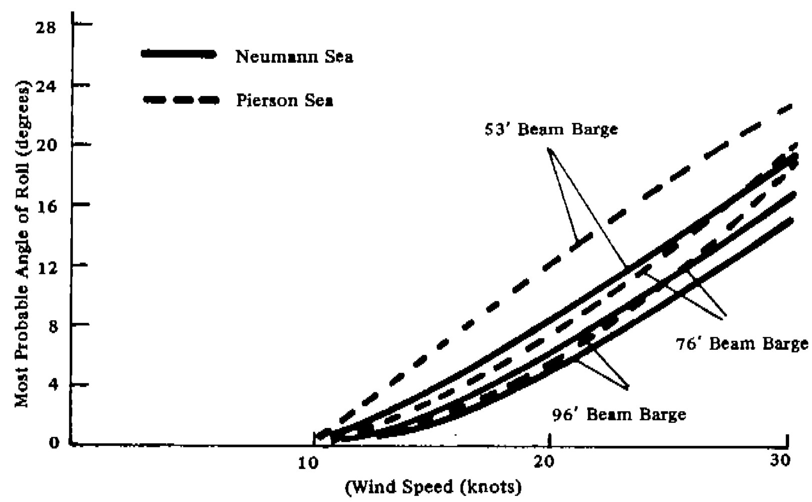

Figure A-3. Roll in Random Beam Seas (Most Probable Roll Angle)

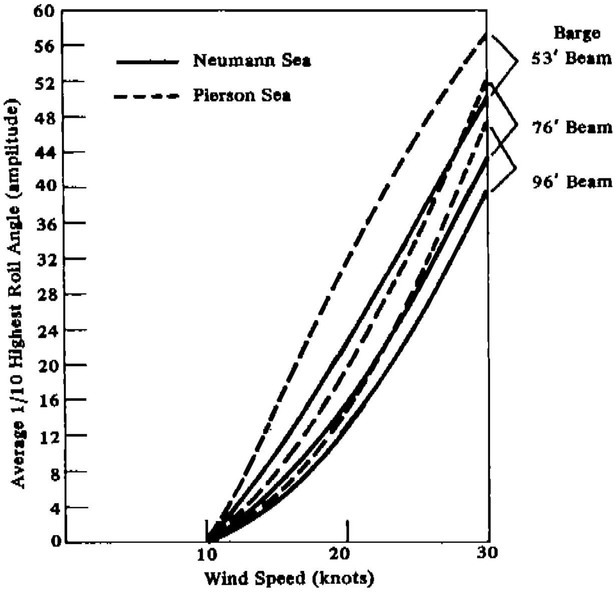

Figure A-4. Roll in Random Beam Seas (Average 1/10 Highest Angle)

Roll motions for each barge were calculated in the following manner. The basic roll equation (reference A-3) may be written as

$$C_1 \ddot{\phi} + C_2 \dot{\phi} + C_3 \phi = C_4 \cos \omega t$$

where the constants C_1, C_2, C_3, and C_4 may be functions of frequency of encounter and various craft parameters. Expressions for these may be found in reference 2.

Define the following:

Undamped Natural Frequency: $\omega_n \equiv \sqrt{\frac{C_3}{C_1}}$

Dimensionless Damping Coefficient: $\nu \equiv \frac{C_2}{\sqrt{C_1 \times C_3}}$

Tuning Factor: $\Lambda \equiv \frac{\omega}{\omega_n}$

where ω = frequency of wave encounter

Magnification Factor: $\mu = \left[(1 - \Lambda^2)^2 + \nu^2 \Lambda^2 \right]^{-1/2}$

The roll equation can be solved to yield (reference A-4)

$$\frac{\phi}{A} = \frac{\omega^2}{g} \mu e^{\frac{\omega^2}{g} \bar{Z}}$$

where:

ϕ/A = the ratio of roll amplitude to wave amplitude

$\bar{Z}$ = distance of center of buoyancy below waterline

Using the expressions for C_1, C_2, C_3 (reference A-3), the magnification factor can be written as follows:

$$\mu = \frac{1}{\sqrt{\left[1 - \frac{\omega^2\left(\frac{B^2 + H^2}{12} + \frac{C_\phi \pi B^3}{256T}\right)}{g\,\overline{GM}}\right]^2 + \left[\frac{BgA_\phi^{\,2}}{4T\overline{GM}}\right]^2}}$$

where:

B = barge beam

H = barge height

T = barge draft

$\overline{GM}$ = barge metacentric height

ω = frequency of encounter

g = gravitational constant

C_ϕ = constant $\approx$ 1.9 for a barge

A_ϕ = function of B and ω such that:

$$A_\phi = .7 \times \left(\frac{\omega^2}{g}\ \frac{B}{2}\right)^2 \quad \text{when } \left(\frac{\omega^2}{g}\ \frac{B}{2}\right) \leq 0.4$$

$$A_\phi = 0.25 \times \left(\frac{\omega^2}{g}\ \frac{B}{2}\right) \quad \text{when } \left(\frac{\omega^2}{g}\ \frac{B}{2}\right) > 0.4$$

In order to use the principle of superposition and achieve a sea spectrum response, the following product is formed.

$$R = \left[\frac{\phi}{A}\right]^2 \times \left[\frac{H_o^2}{\Delta\omega}\right]$$

where $\left(\frac{H_o^2}{\Delta\omega}\right)$ is the amplitude sea spectrum.

Integrating R with respect to frequency yields the total response, TR. There are many statistical values that may be calculated from the value of total response. Among these are:

Most Probable Value = $0.707 \times \sqrt{TR}$

Average 1/10 Highest = $1.8 \times \sqrt{TR}$

Both were computed for each barge in the following two mathematically simulated oceans.

Neumann Sea

$$\left(\frac{H_o^2}{\Delta\omega}\right) = \frac{51.5}{\omega^6} e^{-\frac{725}{V_K^2 \omega^2}}$$

Pierson Sea

$$\left(\frac{H_o^2}{\Delta\omega}\right) = \frac{8.38}{\omega^5} e^{-\frac{9.7 \times 10^4}{V_K^4 \omega^4}}$$

V_K = wind speed in knots

A-2. INVESTIGATION OF WAVE CONDITIONS ENCOUNTERED ALONG TYPICAL BARGE ROUTES AND EXPECTED FREQUENCIES OF ENCOUNTER FOR USE IN HOGGING AND SAGGING CALCULATIONS

Given the sea data presented in table A-II (reference A-5), the expected values of frequency of encounter in head seas can be determined. Since the sea data are in terms of wave period, some manipulations must be performed to yield data in terms of wave frequency. Then, by including the effect of boat speed, a value of frequency of encounter is calculated. This is presented in figure A-5.

Figures A-6, A-7, and A-8 present wave data for the barge routes considered thus far.

In order to be able to calculate hogging and sagging of the barges, we require frequency of encounter for wave length = boat length; the following relation supplies this:

$$f = \frac{1}{2\pi}\sqrt{\frac{2\pi g}{\text{Boat Length}}} + \frac{\text{Boat Speed}}{\text{Boat Length}}$$

(use L_{WL} for boat length)

f in cycles/sec is presented in table A-III for various lengths and speeds. To relate this to sea data, find the percent time that wave length equals boat length (figure A-6). This gives percent of the time that the above frequencies will occur.

Table A-II. Wave Data

PACIFIC COAST				GULF COAST				A	
WAVE HEIGHT (ft)	PERCENT OCCURENCE	WAVE PERIOD (sec)	PERCENT OCCURENCE	WAVE HEIGHT (ft)	PERCENT OCCURENCE	WAVE PERIOD (sec)	PERCENT OCCURENCE	WAVE HEIGHT (ft)	PERCEN OCCUREN
1.0	4.79	CALM+	3.46	1.0	15.71	CALM+	6.39	1.0	7.13
1.5	6.01	< 5	28.52	1.5	20.86	< 5	59.19	1.5	13.75
3.0	21.04	6-7	29.19	3.0	31.19	6-7	22.32	3.0	26.35
5.0	21.90	8-9	20.27	5.0	18.96	8-9	7.70	5.0	21.94
6.5	16.71	10-11	10.90	6.5	7.11	10-11	2.09	6.5	11.83
8.0	10.19	12-13	3.97	8.0	3.89	12-13	0.71	8.0	7.54
9.5	7.28	14-15	2.09	9.5	1.23	14-15	0.11	9.5	4.28
11.0	4.79	16-17	0.917	11.0	0.71	16-17	0	11.0	2.52
13.0	3.62	18-19	0.306	13.0	0.149	18-19	0.15	13.0	1.46
14.0	2.09	20-21	0	14.0	0.037	20-21	0.34	14.0	1.69
16.0	0.25	> 21	0.356	16.0	0.037	> 21	0.97	16.0	0.207
17.5	0.31			17.5	0.037			17.5	0.189
19.0	0.61			19.0	0			19.0	0.36
21.0	0.25			21.0	0			21.0	0.31
22.5	0.10			22.5	0			22.5	0.069
24.0	0			24.0	0.074			24.0	0.173
25.5	0.05			25.5	0			25.5	0.10

NOTE: (From Ocean Wave Statistics by Hogben and Lumb). 1967 Edition.

+ - Includes "Unknown."

Data are for average "Year Round Conditions"

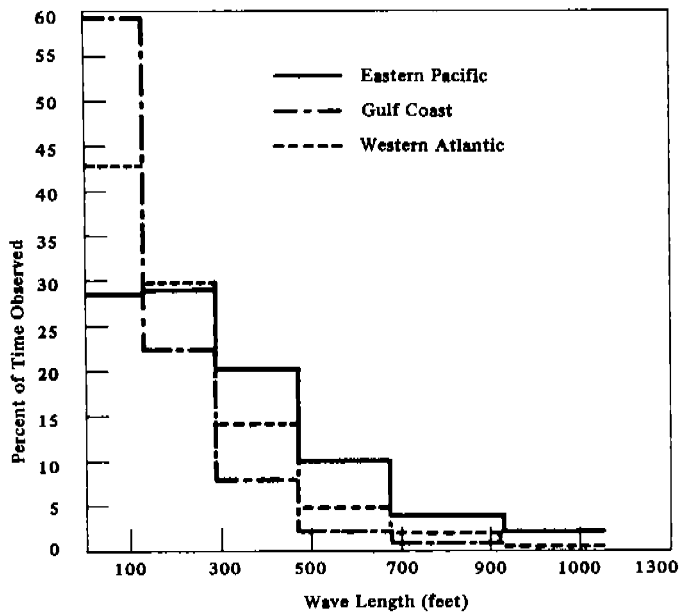

Figure A-5. Wave Lengths Along U. S. Coasts

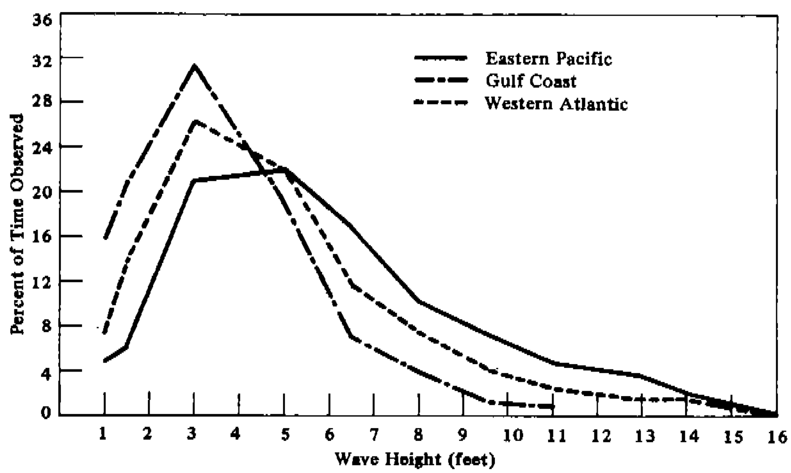

Figure A-6. Wave Heights Along U. S. Coasts

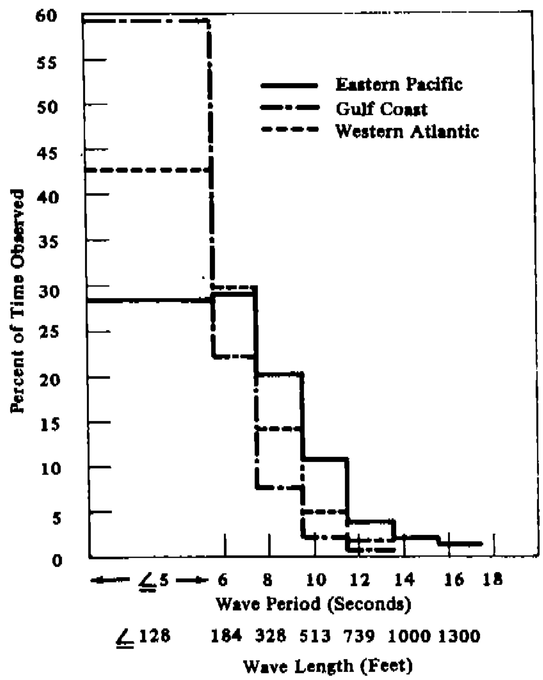

Figure A-7. Wave Periods Along U. S. Coasts

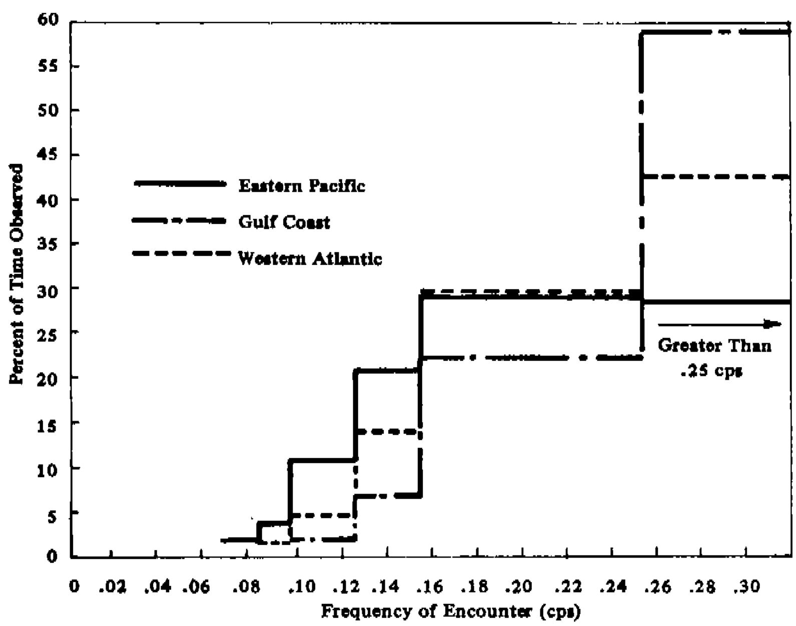

Figure A-8. Frequency of Encounter at 10 Knots

Assuming that a 440-foot barge is operated 300 days per year at 8 knots, the following estimate is made of sagging and hogging cycles in 10 years, using data from figure A-6 and table A-III.

$$\text{Cycles} = 0.2 \times \left[10 \text{ yrs} \times 300 \text{ days/year} \times 86400 \text{ sec/day} \times 0.1386 \right] = 7.2 \times 10^{6}$$

Table A-III. Determining Frequency of Encounter

LENGTH	SPEED (knots)					
(ft)	2	4	6	8	10	12
440	0.1156	0.1233	0.1309	0.1386	0.1463	0.1540
450	0.1142	0.1217	0.1292	0.1367	0.1442	0.1518
340	0.1327	0.1426	0.1526	0.1625	0.1725	0.1824
240	0.1602	0.1743	0.1883	0.2024	0.2165	0.2306

A-3. INVESTIGATION OF EFFECTS OF FLUID SLOSHING

The term "free surface effect" is given to the problem of a ship's reduced stability caused by the sloshing of a liquid cargo. Standard texts such as reference A-6 treat this subject. The simplest solution to this problem is to reduce the amount of free surface. In the case of chemical barges, this is achieved by filling the tanks to capacity. It should be noted that Coast Guard regulations do not require a minimum tank fullness. It may be advisable to require that tanks be filled to capacity or emptied to less than about one-fourth capacity. Operating at half capacity results in the greatest free surface effect.

The effect of fluid motion on barge motion can be divided into two parts:

a. actual forces imparted to the barge by the fluid, and

b. change in center of gravity and moment of inertia of the loaded barge due to motion of the load.

An order of magnitude calculation indicated that, for the expected boat motions, the center of gravity moves only one or two percent of the barge length. From this it was concluded that changes in the location of the center of gravity and moment of inertia may be neglected in calculating barge motions.

The forces imparted to the barge by the fluid are difficult to calculate since the exact fluid motion is not known. It is possible, however, to make some general remarks concerning the effect of fluid motions on barge motions. Clearly, if the fluid had neither mass nor motion, the barge motion would be unaffected. Also, if wave and fluid natural periods were far below that of the barge, there would be little effect. A system forced at a frequency far above its own natural frequency will not respond. Maximum effects due to fluid motion will occur when the natural periods of fluid, barge, and wave are the same. Large effects will be observed when barge and fluid have the same natural period of motion. In table A-IV and figure A-9 the natural periods of barge and fluid motion are compared with the average natural period of the waves found along the barge routes.

The case of half-filled barges at zero speed has been presented. In figure A-10 the natural period of the fluid is presented for various filling depths.[A-7] Although the natural period of barge roll falls generally above the most common average wave period, the natural period of the fluid transverse motion falls among the most common waves.

It can be concluded that fluid motions may not have a large effect on barge roll motions. The effect on barge surging and pitching is not yet clear. The fluid should have very little effect on barge heaving.

Another, less obvious, effect of fluid motions is the possibility of fatigue loading from the cyclic sloshing motions. Although the sloshing loads may be small, the possibility of large fatigue loading on tank walls and supports should be investigated.

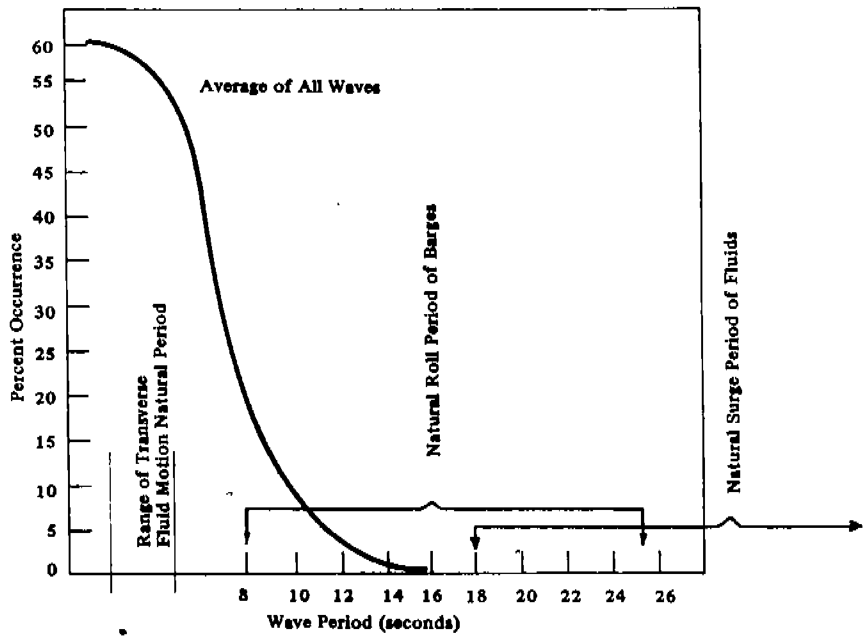

Figure A-9. Comparison of Natural Periods

A great deal of work has been done investigating sloshing in vertical tanks such as rockets. Judging from the lack of available reference material, very little has been done in the field of horizontal tanks. The two problems are somewhat similar but details of mathematical modeling are very different.

The mathematical model (figure A-11), based on references A-8 and A-9, represents a simplified approach. Even so, it yields no numerical results due to the lack of experimental parameters and actual test information. Also needed is a method for applying experimental results to full-scale barges. The mathematical model represents a first approach and is presented as an indication of what must be done to solve the sloshing problem analytically.

Table A-IV. Natural Periods of Motion Barges & 1/2 Full Tanks

BARGE		FLUID PERIOD (SECONDS)					
		LONGITUDINAL MOTION			TRANSVERSE MOTION		
LENGTH (ft)	ROLL PERIOD	1st MODE	2nd MODE	3rd MODE	1st MODE	2nd MODE	3rd MODE
440	25.4	36	12.8	8.4	4.3	2.3	1.75
450	9.47	36	12.8	8.4	4.3	2.3	1.75
340	9.0	21.9	8.25	5.6	3.73	1.99	1.52
240	7.9	17.9	6.74	4.6	3.06	1.62	1.24

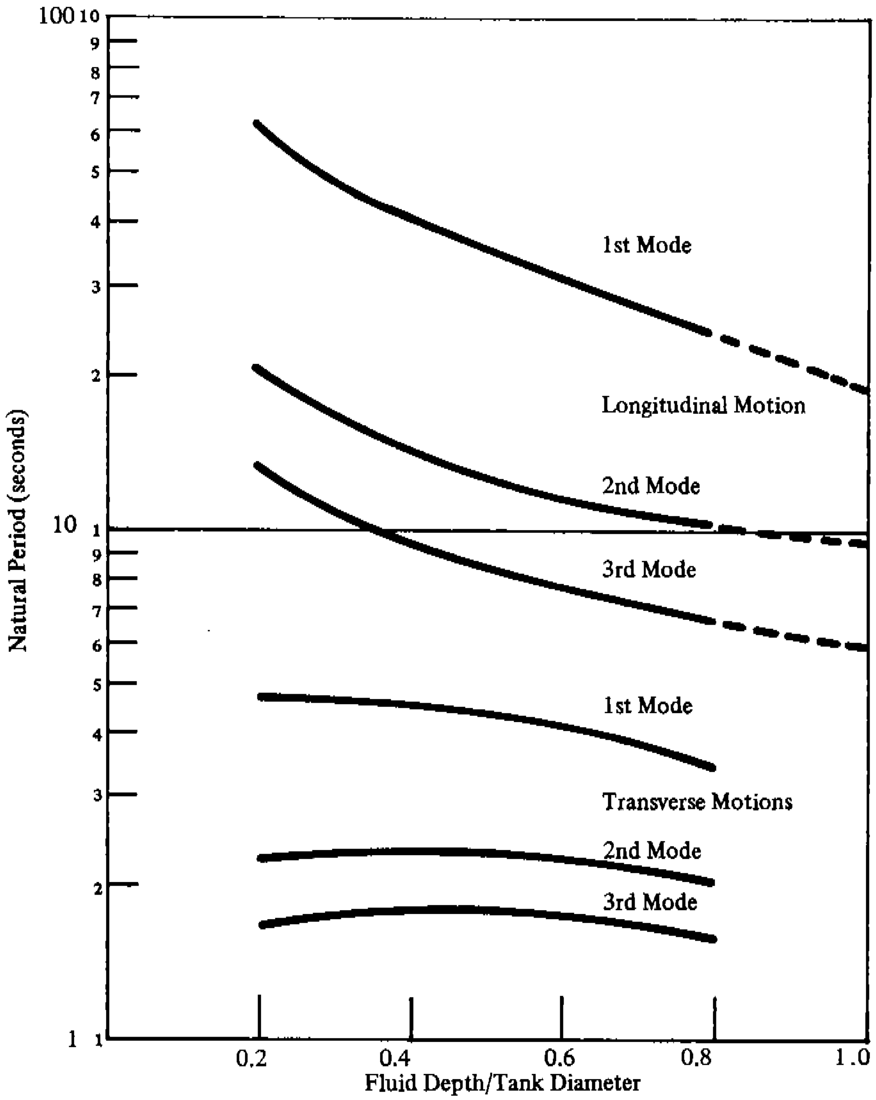

Figure A-10. Natural Periods of Vibration of Fluids Contained in a Circular Cylinder of Radius 20 ft, Length 400 ft

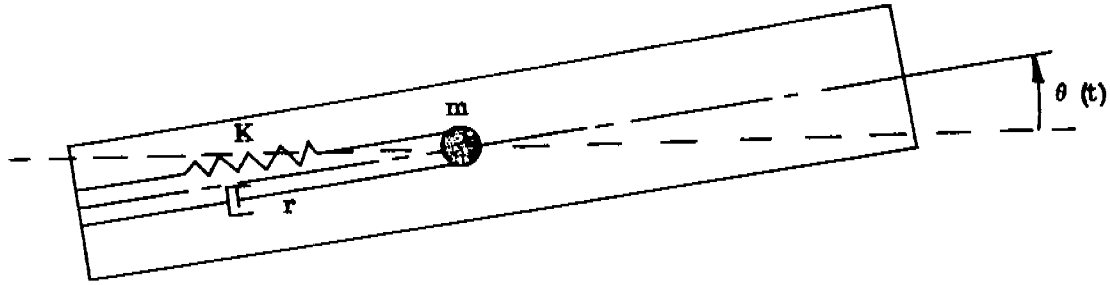

Figure A-11. Simplified Sloshing Mathematical Model

Consider the fluid in a tank pitching at $\theta(t) = \alpha \sin \omega t$ to be a point mass acted on by a spring and damper. The equation of motion is ($x = 0$ is at center of tank):

$$\left[m\ddot{x} + r\dot{x}\cos\theta + kx\cos\theta + mg\sin\theta = 0 \right]$$

For small θ $\sin\theta \cong \theta$ $\cos\theta \cong 1$ and we have a simple equation:

$$m\ddot{x} + r\dot{x} + kx + mg\,\alpha \sin\omega t = 0$$

$$x + \frac{r}{m}\dot{x} + \frac{k}{m}x = -g\,\alpha\sin\omega t$$

Let $x = A\sin\omega t + B\cos\omega t$

$$\dot{x} = \omega A\cos\omega t - \omega B\sin\omega t$$

$$\ddot{x} = -\omega^2 A\sin\omega t - \omega^2 B\cos\omega t$$

$$-\omega^2 A\sin\omega t - \omega^2 B\cos\omega t + \frac{r\,\omega}{m}A\cos\omega t - \frac{r\,\omega}{m}B\sin\omega t + \frac{k}{m}A\sin\omega t$$

$$+ \frac{k}{m}B\cos\omega t = -g\alpha\sin\omega t$$

$$\left(\begin{array}{l} -\omega^2 A - \frac{r\omega}{m}B + \frac{k}{m}A = -g\alpha \\ \\ -\omega^2 B + \frac{r\omega}{m}A + \frac{k}{m}B = 0 \end{array} \right) = \left(\begin{array}{l} A\left(\frac{k}{m} - \omega^2\right) = -g\alpha + \frac{r\omega}{m}B \\ \\ A\frac{r\omega}{m} = \left(\omega^2 - \frac{k}{m}\right)B \end{array} \right)$$

$$\left[-\frac{\left(\frac{k}{m} - \omega^2\right)\left(\frac{k}{m} - \omega^2\right)}{\frac{r\omega}{m}} - \frac{r\omega}{m} \right] B = -g\alpha; \quad B = \frac{\left(g\alpha\frac{r\omega}{m}\right)}{\frac{k^2}{m^2} - 2\frac{k}{m}\omega^2 + \omega^4 + \frac{r^2\omega^2}{m^2}}$$

$$A = \frac{(\omega^2 - \frac{k}{m})}{\frac{r\omega}{m}} \quad \frac{g\alpha \frac{r\omega}{m}}{\left[\frac{k^2}{m^2} - 2\frac{k}{m}\omega^2 + \omega^4 + \frac{r^2\omega^2}{m^2}\right]}$$

Thus, our expression for fluid motion has become:

$$x(t) = A \sin \omega t + B \cos \omega t$$

$$A = \frac{(\omega^2 m^2 - km)\, g\alpha}{(k^2 - 2km\omega^2 + \omega^4 m^2 + r^2\omega^2)} \qquad B = \frac{r\omega m g\alpha}{(k^2 - 2km\omega^2 + \omega^4 m^2 + r^2\omega^2)}$$

Forces on the end of the tank will be $F_T = m\ddot{x}$

$$F_T = -m\omega^2 A \sin \omega t - m\omega^2 B \cos \omega t$$

$$\text{or} \quad |F_T| = \sqrt{m^2\omega^4 A^2 + m^2\omega^4 B^2}$$

$$\Rightarrow \quad |F_T| = m\omega^2 \sqrt{A^2 + B^2}$$

If we had appropriate values for r, m, and k, we would now have a solution, since the Coast Guard specifications give values for α and ω. Various methods for evaluating m have been developed in the case of vertical cylinders and may possibly be applicable to this problem.

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A-7. Investigation of the Natural Frequencies of Fluids in Spherical and Cylindrical Tanks, J. L. McCarty and D. G. Stephens, NASA T. N. D-252, May 1960.

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Appendix B

ANALYSIS OF RING STIFFENERS

Analysis of circular rings subject to applied support loads and applied tank and liquid weight loads was performed; 180° saddle supports of width h were assumed. The load distribution on the ring due to the support was assumed to be $p_r = p_1 \cos\theta$ for $|\theta| \leq 90°$ and 0 for $90° < \theta < 270°$. The load distribution due to the tank and liquid weight was assumed to be $p_\theta = p_2 \sin\theta$ (see figure B-1). The total system of applied loads must be self-equilibrating. Assuming that the ring width equals the saddle width, h,

$$p_1 = -\frac{2\bar{Q}}{h\pi^2 r}$$

and

$$p_2 = -\frac{\bar{Q}}{h\pi r}$$

where $\bar{Q}$ is the total load supported by the saddle.

It is necessary to expand the saddle load, p_r, in a Fourier cosine series to have the proper form for analysis. Performing the necessary integrations yields:

$$p_r = -\frac{2\bar{Q}}{h\pi^2 r}\left\{1 + \frac{\pi}{2}\cos\theta + \frac{2}{3}\cos 2\theta - \frac{2}{15}\cos 4\theta + \frac{2}{35}\cos 6\theta - \frac{2}{63}\cos 8\theta + \frac{2}{99}\cos 10\theta - \ldots\right\} \quad \text{(B-1)}$$

The weight loading, p_θ, already has the form of the n = 1 term of a Fourier sine series:

$$p_\theta = -\frac{\bar{Q}}{\pi r h}\sin\theta \quad \text{(B-2)}$$

From Flugge[B-1] (p 219, Eq. 13 a-c), setting $p_x \equiv 0$, $\frac{\partial}{\partial x} \equiv \frac{\partial^2}{\partial x^2} \equiv 0$ yields the governing differential equations of linear thin shell bending theory for circular rings in terms of v and w, the circumferential and radial displacements respectively:

$$\frac{d^2 v}{d\theta^2} + \frac{dw}{d\theta} + \frac{p_\theta r^2}{D} = 0 \quad \text{(B-3)}$$

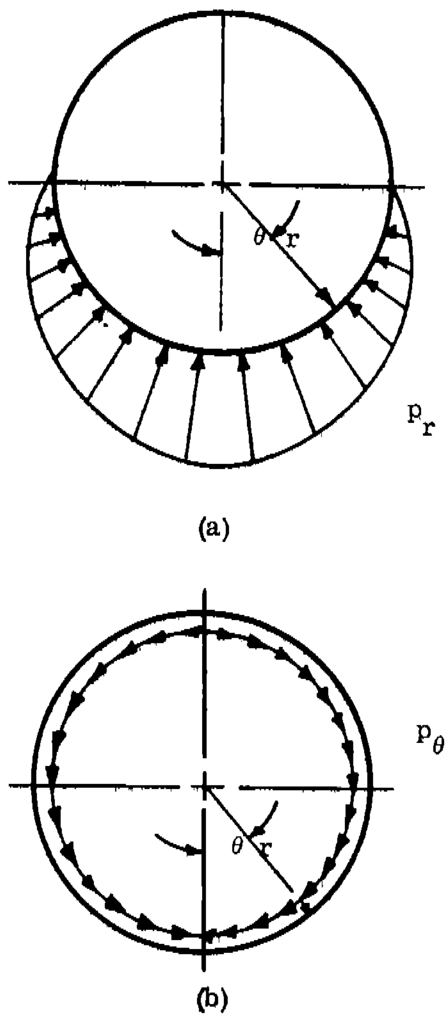

Figure B-1. Loading on Ring Stiffener

$$\frac{dv}{d\theta} + w + k\left[\frac{d^4w}{d\theta^4} + 2\frac{d^2w}{d\theta^2} + w\right] - \frac{p_r r^2}{D} = 0 \tag{B-4}$$

where

$$D = \frac{Et}{(1-\nu^2)} \tag{B-5}$$

and

$$k = \frac{t^2}{12r^2} \tag{B-6}$$

(In Appendix B, t refers to stiffener thickness, i.e., outer radius minus inner radius.)

Differentiation of eq. B-2 with respect to θ and subtraction of eq. B-1 from the result yields:

$$\frac{d^5w}{d\theta^5} + 2\frac{d^3w}{d\theta^3} + \frac{dw}{d\theta} = \frac{r^2}{kD}\left[\frac{d}{d\theta}(p_r) + p_\theta\right] \tag{B-7}$$

Substitution of eqs. B-1 and B-2 for p_r and p_θ gives:

$$\frac{d^5w}{d\theta^5} + 2\frac{d^3w}{d\theta^3} + \frac{dw}{d\theta} = -\frac{r^2}{kD}\left(\frac{2\bar{Q}}{h\pi^2 r}\right)\left[-\frac{4}{3}\sin 2\theta + \frac{8}{15}\sin 4\theta - \frac{12}{35}\sin 6\theta + \frac{16}{63}\sin 8\theta - \frac{20}{99}\sin 10\theta + \ldots\right] \tag{B-8}$$

Let w be a Fourier cosine series:

$$w = \sum_{n=o}^{\infty} w_n \cos n\theta \tag{B-9}$$

Differentation of eq. B-9, substitution into eq. B-8, and equating coefficients of the same trigonometric functions yields the following solution for w. (Note that w_o is determined from eq. B-4.)

$$w = -\frac{r^2}{D(1+k)}\left(\frac{2\bar{Q}}{h\pi^2 r}\right) + \frac{r^2}{kD}\left(\frac{2\bar{Q}}{h\pi^2 r}\right)\left[-\frac{2}{(3)^3}\cos 2\theta + \frac{2}{(15)^3}\cos 4\theta - \frac{2}{(35)^3}\cos 6\theta + \frac{2}{(63)^3}\cos 8\theta - \frac{2}{(99)^3}\cos 10\theta \ldots\right] \tag{B-10}$$

Similarly, assume v is a Fourier sine series:

$$v = \sum_{n=1}^{\infty} v_n \sin n\theta \tag{B-11}$$

Differentiation of eqs. B-10 and B-11, substitution of these results and eq. B-2 into eq. B-3, and equating coefficients of the same trigonometric functions yields the following solution for v:

$$v = -\frac{r^2}{D}\left(\frac{\bar{Q}}{h\pi r}\right)\sin\theta + \frac{r^2}{kD}\left(\frac{2\bar{Q}}{h\pi^2 r}\right)\left[\frac{1}{(3)^3}\sin 2\theta - \frac{1}{(2)(15)^3}\cos 4\theta + \frac{1}{(3)(35)^3}\sin 6\theta - \frac{1}{(4)(63)^3}\cos 8\theta + \frac{1}{(5)(99)^3}\cos 10\theta - \dots\right] \tag{B-12}$$

From Flügge (p 214, eq 9a-h), the circumferential force and moment resultants in terms of v and w displacements are:

$$N\theta = \frac{D}{r}\left(\frac{dv}{d\theta} + w\right) + \frac{K}{r^3}\left[w + \frac{d^2w}{d\theta^2}\right] \tag{B-13}$$

$$M\theta = \frac{K}{r^2}\left[w + \frac{d^2w}{d\theta^2}\right] \tag{B-14}$$

where

$$K = \frac{Et^3}{12(1-\nu^2)} \tag{B-15}$$

Substitution of eqs. B-10 and B-12 for v and w into eqs. B-13 and B-14 yield:

$$N\theta = -\frac{2\bar{Q}}{h\pi^2} - \frac{\bar{Q}}{\pi h}\cos\theta + \frac{2\bar{Q}}{h\pi^2}\left[\frac{2}{(3)^2}\cos 2\theta - \frac{2}{(15)^2}\cos 4\theta + \frac{2}{(35)^2}\cos 6\theta - \frac{2}{(63)^2}\cos 8\theta + \frac{2}{(99)^2}\cos 10\,\theta - \dots\right] \tag{B-16}$$

$$M\theta = -\frac{t^2}{12r^2+t^2}\left(\frac{2\bar{Q}r}{h\pi^2}\right) + \frac{2\bar{Q}r}{h\pi^2}\left[\frac{2}{(3)^2}\cos 2\theta - \frac{2}{(15)^2}\cos 4\theta + \frac{2}{(35)^2}\cos 6\theta - \frac{2}{(63)^2}\cos 8\theta + \frac{2}{(99)^2}\cos 10\theta - \dots\right] \tag{B-17}$$

The circumferential stress in the ring at the inner and outer surfaces is given approximately by

$$\sigma_\theta = \frac{N\theta}{t} \pm \frac{6M\theta}{t^2} \tag{B-18}$$

Substitution of eqs. B-16 and B-17 into eq. B-18 and associating (ht) with the cross-sectional area and ($ht^2/6$) with the section modulus of the stiffening ring yields:

at, $\theta = 0$,

$$\sigma_\theta = -\frac{.48\bar{Q}}{A} \pm .043 \frac{\bar{Q}r}{I/c} \tag{B-19}$$

at $\theta = 180°$,

$$\sigma_\theta = \frac{.158\bar{Q}}{A} \pm .043 \frac{\bar{Q}r}{I/c} \tag{B-20}$$

The maximum compressive stress occurs at $\theta = 0$; the maximum tensile stress occurs at $\theta = 180°$.

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Appendix C

OUTLINE FOR STRAIN GAGE INSTRUMENTATION OF A TANK BARGE

C.1. INTRODUCTION

Generally, strain recording instruments are used to describe strain magnitude, direction, and distribution in areas of complex structures which may not be reducible to mathematical description, or to verify the presence and magnitude of certain strains which had been predicted by theoretical analyses.

The main objective here is to describe, in general terms, presently available instrumentation procedures applicable to an experimental stress analysis of large tanks supported and transported by barge.

C.2. DISCUSSION

As a prerequisite to the installation of any instrumentation, the following information is necessary:

1. Type of fluid which will be carried during the test.
2. Geographic location of the work site and the time of year proposed for installation.
3. Type and availability of electrical power for site work and testing.
4. Number and size of tank penetrations available to permit internal instrumentation installation and the exiting of signal loads.
5. Total number of information channels to be recorded and specific locations at which strain gages and other transducers are required.
6. Duration of the test program.

The following test conditions are representative of the experimental program:

1. Strain data will be recorded during filling and emptying of the tank or tanks.
2. Strain data will be recorded during actual operation of the barge in sea states if the barge selected is for ocean service.
3. Bow-slamming, heaving, twisting, and pitch will be recorded as a function of strain, pressure distribution, accelerations and time, relative to sea state and forward velocity if the barge is for ocean service.
4. Strain gage and transducer data will be recorded using dynamics instrumentation.

C.3 GENERAL AREAS OF INTEREST

General areas of interest have been identified and shown on figure C-1. Strain gages will be applied, both internally and externally, in the area of the support saddle. Measurements of strain decay between saddle supports will be obtained by an array of strain gages on the inner and outer skin of the tank.

Figure C-2 indicates the general areas of interest on the barge hull. Pressure transducers, velocity and acceleration instruments, and strain gages will be used to obtain an understanding of the forces acting on the barge/tank during actual operation.

C.4 PROPOSED INSTRUMENTATION

To simultaneously record dynamic data from a large number of signal sources, multichannel light beam oscillographs or tape recorders may be used. These devices record a processed or conditioned signal which is developed by peripheral equipment. In general, the following equipment is needed to obtain an intelligible signal from a strain gage or other type of transducer:

1. Power supply
2. Amplifier and signal conditioner
3. Recorder
4. Interconnecting cable.

C.5 TEST FACILITIES AND CONSIDERATIONS

Since a test of this type requires considerable time and expense, adequate facilities are required to protect expensive equipment. A suitable structure, centrally located with respect to heavily instrumented areas, can reduce the amount of cable used and result in a significant reduction in installation costs. The possibility of more than one recording station should be considered if heavily instrumented areas are separated by a hundred feet or more. Secondary, unmanned recording stations can be synchronized to the main recording station or operated independently.

An important consideration in the cost evaluation of multichannel dynamic recording systems is the frequency level which the system must respond to and record. The frequency level will be within 0 to 50 Hz. The output signal from a strain gage may be directly recorded by certain light beam oscillographs without amplification, thus eliminating one of the costly components of the recording system. However, a decision to do without amplification must be justified by theoretical analysis.

Finally, it is important to have a clear understanding of the type of data obtainable from strain gages. If a single-element strain gage is used at a point of interest, the strain data does not permit a calculation of the maximum principal stress unless the strain field is a maximum and uniformly uniaxial in the direction of the gage, such as would occur in a controlled tensile specimen test.

A two-element or biaxial strain gage may be used to calculate maximum and minimum principal stresses at the point of interest when the biaxial gage orientation is identical to the biaxial strain field.

A three-element strain gage will allow maximum and minimum principal stresses and directions to be calculated from raw data for the point of interest without the need for specific orientation with respect to the test specimen strain field.

Hence, the three-element strain gage (triaxial strain rosette) is most suitable to determine principal stresses and directions in structures subject to combined loading, such as twisting and bending.

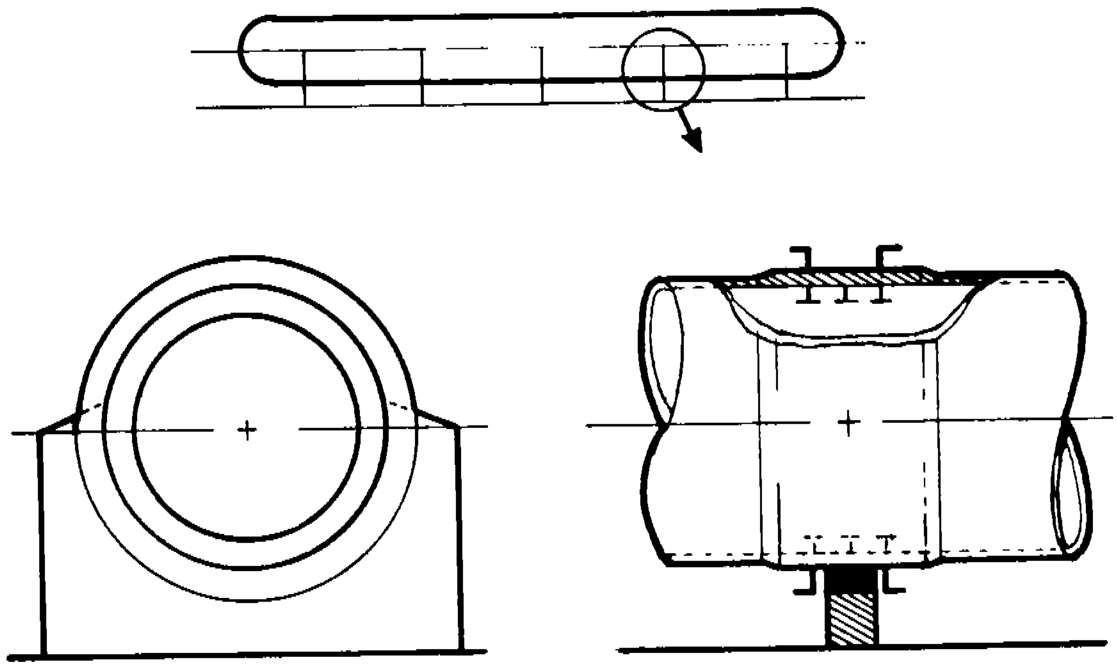

Figure C-1. Typical Location of Extensive Instrumentation

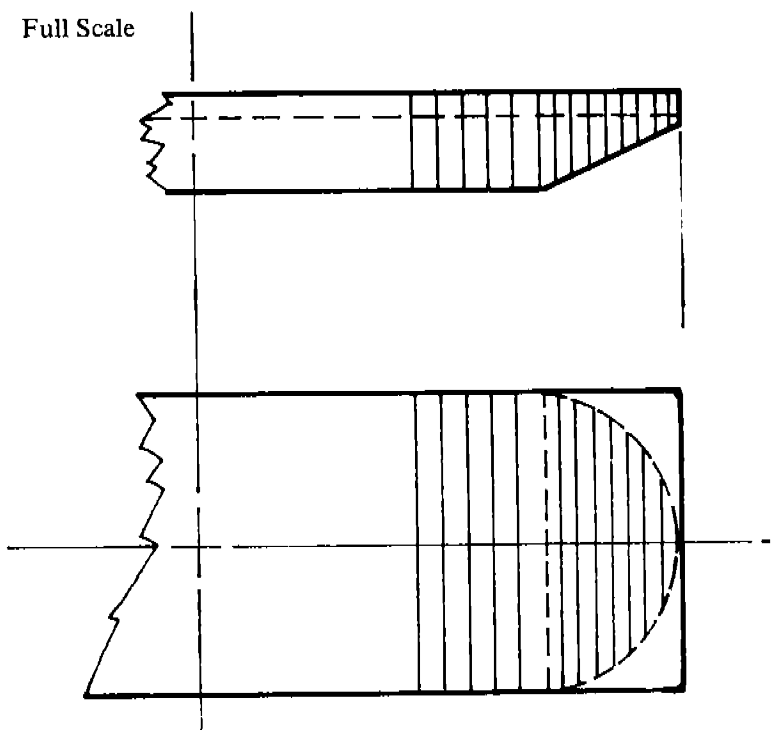

Figure C-2. Typical Installation of Permanent Photoelastic Material on Inside of Hull

Appendix D

DISCUSSION OF APPROACH FOR TANK/BARGE SLAMMING MODEL TESTS

D.1 PROPOSED BOW PRESSURE MEASUREMENTS IN WAVES

This discussion outlines a program of barge model fabrication and towing tests in various regular, sinusoidal wave conditions. The purposes of the tests are to measure the bow impact pressures and normal velocities and to define the pitching and heaving amplitudes of the barge. During the program, the theoretical peak pressures, based on the measured normal velocities, will be calculated and compared with the experimentally determined values. Completion of this proposed program will require the following specific items of work:

D.1.1 WORK STATEMENT

1. Fabricate one wooden middlebody-afterbody barge model to approximately 1/25 scale. The model will provide for the attachment of varying bow configurations and will be fabricated in accordance with lines supplied by the customer.

2. Fabricate three separate wooden bow configurations to the same scale as the middlebody-afterbody model of Item 1 in accordance with lines supplied by the customer.

3. Install towing staff and pitch, heave, and vertical acceleration instrumentation in the middlebody-afterbody section. Install nine crystal-type pressure transducers, and normal velocity instrumentation in the bow section. Normal velocities will be obtained by electrically integrating vertical acceleration measurements.

4. Ballast each model to the predicted displacements (light and heavy), center of gravity positions (longitudinal and vertical), and longitudinal radii of gyration. Calibrate all test instrumentation.

5. Conduct towing tests with each model at two different displacements and at one velocity into waves of four different scale heights. Tests will be conducted in six different wave lengths for each wave height. Approximately 144 tests will be required. Time histories of the water impact pressures at nine stations, normal velocities at three longitudinal stations, vertical accelerations at bow and center of gravity, pitch angles, heave displacements, towing velocity, and wave contours will be recorded simultaneously by a minimum of two oscillographic recorders. The time references of the two recorders will be precisely correlated.

6. Reduce all test data to engineering units and expand to full-scale proportions.

7. Using the normal velocities occurring at each pressure transducer location during the instant of peak pressure, calculate the theoretical maximum impact pressure using methods described in references D-1 and D-2.

8. Prepare graphical presentations of the peak impact pressures (measured and calculated), pitch and heave amplitudes, normal velocities, and vertical

accelerations.

9. Prepare and submit a model test report presenting the test and theoretical results, representative test photographs, and discussions of the testing techniques, instrumentation, and results.

D.2 SLAMMING

The following comments are made concerning the discussion of slamming.

1. It will be useful, while making slamming tests, to investigate the following:
   a. Effect of on-board fluid-containing tanks.
   b. Effect of surge due to towing by a surging tug.
2. The method of static strength design is concerned mainly with seaplane design practice which may prove adequate.

D.3 DISCUSSION

The DTMB Report No. 1994 (reference D-3) has been reviewed, as were references D-1, -2, -4 and -5. We believe that the rise time anomalies found in Table 3 of reference D-3 are a manifestation of the trapped air problem discussed in references D-1 and D-4. Also, their instrumentation system frequency response (1200 Hz) was relatively low compared to the system response (200 KHz) described in reference D-1.

System responses of 200KHz require the use of cathode ray oscilloscopes and streak cameras for data recording. The use of this equipment on our towing tank carriage is not feasible, particularly in view of the fact that we would be interested in nine separate pressure measurements. However, we can achieve 5000 Hz system response utilizing oscillograph recording techniques. It is proposed, therefore, that the bow pressures at nine specific locations be measured by crystal-type pressure transducers with 200 KHz frequency response and recorded by oscillographs incorporating galvanometers having frequency responses of 5000 Hz. Thus, an overall system response of 5000 Hz will result. Crystal-type transducers are desirable because of their small (0.208 inch) diameter pressure face and low sensitivity to acceleration forces.

It is anticipated that maximum full-scale barge normal velocities of 10 to 20 fps may be incurred in waves. Since a barge is relatively lightly loaded, the bow will not cut through the water and may be expected to decelerate significantly under the growing pressure area. In any event, Wagner* suggests that the outboard edge of the wetted width sweeps outward[4] at a speed,

$$C = \frac{V_N}{\mu(c)}$$

where the function μ (c) is given by a power series for non-wedge shapes, and by $\frac{2}{\pi}$ $\tan \beta$ for wedges.

*German mathematician

Thus, for an arbitrary hull dead rise of 2° and a full-scale normal velocity, V_N, of 15 fps, the resulting speed is

$$\dot{C} = \frac{\pi}{2} \times 15 \times \frac{1}{0.0309} \times 12 = 9,150 \text{ in./sec full scale}$$

The point value of the peak pressure would, therefore, remain on a 0.208 inch diameter pressure transducer for 23 microseconds for the full-scale case, and 115 microseconds for a 1/25th scale model with Froude scaling. A 5000 Hz signal has a period of 200 microseconds, which is equivalent to a quarter sine wave rise time of 50 microseconds. Thus, a zero width peak pressure corresponding to the above example could conceivably be measured at model scale, but not full scale. What the pressure pickup "measures" is dependent upon the width of the pressure peak as well as upon the speed with which it traverses the pickup diameter. An example of the variation of this width with wedge dead rise is shown in figure D-1, which is a reproduction of a figure from reference D-2.

In view of the above, it is further proposed that the peak pressures be computed by the methods of references D-1, D-2, and D-5 for comparative purposes. To facilitate these computations, the normal velocities incurred during impact will be measured at the three longitudinal barge stations, corresponding to the three transverse lines of pressure transducers, by electrically integrating the vertical accelerations. The pressure measurements will be recorded simultaneously with the normal velocity measurements on the same oscillograph tape and will, therefore, be correlated by the precision time reference. The normal velocity at the instant of peak pressure at each pressure station can thereby be obtained.

During each model test, measurements of the barge pitching and heaving amplitudes and the vertical accelerations at the bow and center of gravity will be obtained. Also, recordings of the towing velocity and the wave contours (measured outboard of one specific barge station) will be obtained during each test.

It is believed that this combined model test data will fully describe the effects of wave impacts on the various bow configurations.

Once this data has been obtained, the problem becomes one of establishing the relation between static strength design and the highly transient loadings imposed by low dead rise impacts. Our past experience with full-scale seaplanes operating in waves has been that the maximum measured pressures are far in excess of those which the bottom plating could sustain statically over any significant area. It may be of interest that one of the authors of reference D-5 laid the foundation in 1947 for the approach of reference D-6. This is a method which still finds wide use today on hydrofoils as well as seaplanes. It is believed that a similar correlation of structural "successes" and "failures" with theory may be required for barges.

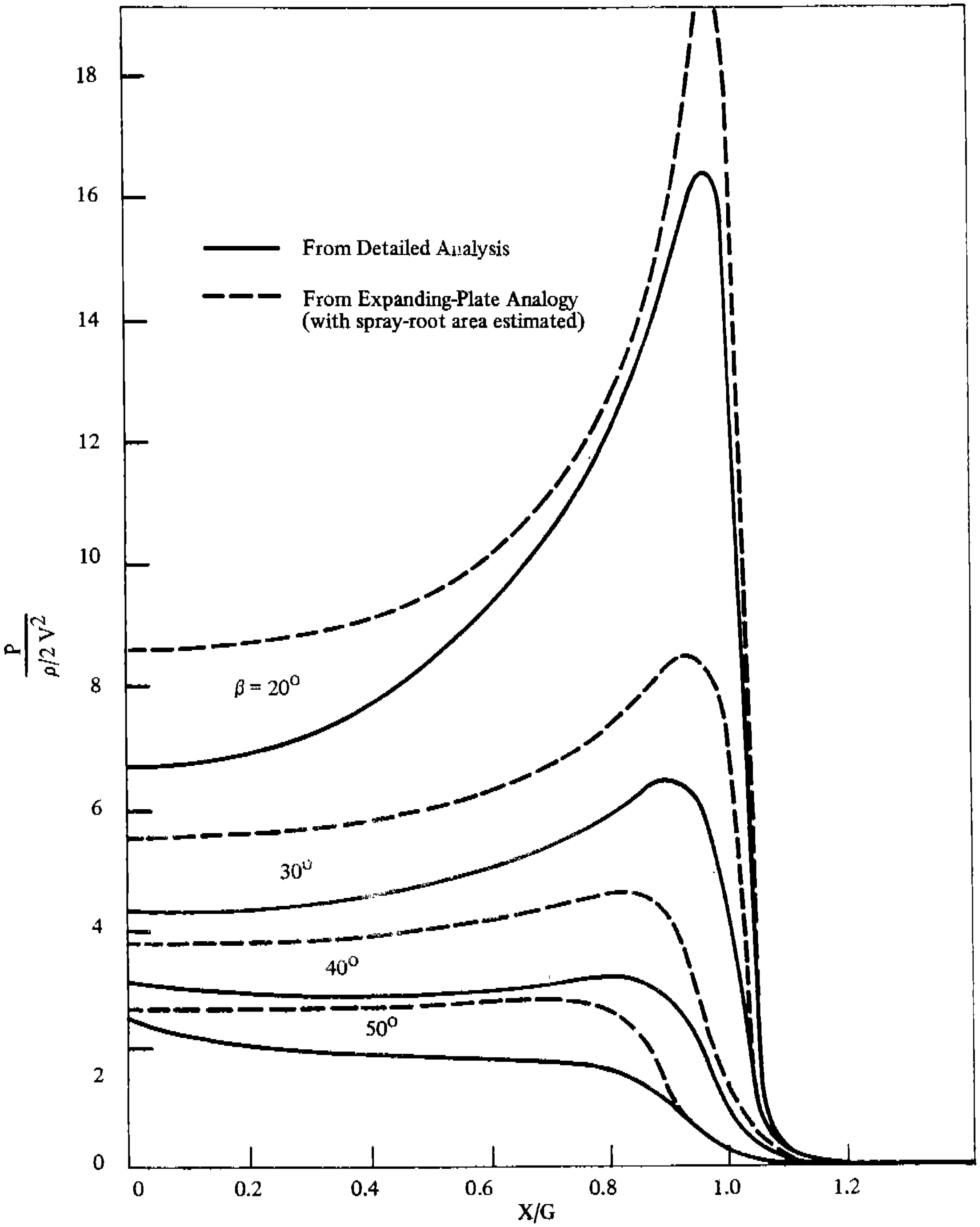

Figure D-1. Pressure Distribution Over the Wedge

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13. ABSTRACT

This report describes a program of analytical research to determine the availability of reliable methods for the design of long, large diameter, cylindrical tanks and their supports for transportation of liquids and low-pressure liquified gases in barges for service on rivers or at sea. Loading conditions, existing design/analysis methods, material considerations, and a computer method for predicting stresses are presented.

The major conclusion of the work performed is that design procedures for river barge tanks up to 20 feet in diameter are well established and that no failures due to inadequate design practice have been reported since refrigerated tanks went into service about ten years ago. The present method for designing river barge tanks is a logical starting point for determining the structural configuration of large tanks for oceanic service, but more detailed analysis of loads and resulting stresses should be performed for this application.

Several areas in which theoretical or experimental effort is needed are identified: (1) investigation of tank-saddle-barge interaction, (2) investigation of fatigue criteria for cyclic loading, (3) investigation of buckling criteria, (4) analytical and experimental investigation of slamming, and (5) experimental verification of stresses in a full-scale tank.

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