SSC-258

A STUDY TO OBTAIN VERIFICATION OF LIQUID NATURAL GAS (LNG) TANK LOADING CRITERIA

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SR-218

28 APR 1976

The Ship Structure Committee developed a serious concern for the design criteria governing liquid cargo tanks, especially with the dramatic increase in the numbers of liquid natural gas (LNG) carriers under construction and in operation. These tanks pose design and fabrication problems that were not previously encountered. The novel and sophisticated containment systems must be evaluated against criteria that have been developed within recent years.

A project was undertaken to survey, evaluate, and develop dynamic load criteria for these tanks. The investigator used analytical results and the data from available model and full scale experiments to compare with the various worldwide criteria that were available as of June 1974.

This report contains the results of that project which conclude that the criteria examined were on the conservative side.

W. M. Benkert
W. M. Benkert
Rear Admiral, U. S. Coast Guard
Chairman, Ship Structure Committee

FINAL TECHNICAL REPORT

on

Project SR-218

"Verification of LNG Tank Loading Criteria"

A STUDY TO OBTAIN VERIFICATION OF LIQUID NATURAL GAS (LNG) TANK LOADING CRITERIA

by

R. L. Bass, J. C. Hokanson, and P. A. Cox

Southwest Research Institute

under

Department of the Navy
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1976

ABSTRACT

A study of LNG tank loading criteria is presented that includes a survey and review of load criteria presently employed in the design of cargo tanks for LNG carriers. Motion and acceleration values as determined from these criteria are compared to ship motion calculations and available full-scale data. A comparison of LNG tank loads, as predicted by current classification society and regulatory agency criteria, is given along with recommended updated criteria in each of seventeen load categories. Model tests and full-scale measurement programs to provide adequate data for verification of load and acceleration criteria are also outlined. The criteria examined were those that were available as of June 1974.

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ACKNOWLEDGMENTS

A large amount of information was needed for this program and could not have been accumulated without the help and cooperation of many individuals. Much of this information was obtained from both personal and written contacts with representatives of the classification societies and regulatory agencies. These individuals include: Messrs. Grove, Faltinsen, Hansen, Olsen, Tobiesen and Vedeler of Det norske Veritas; Messrs. Pless, Östergaard, Payer and Böckenhauer of Germanischer Lloyd; Messrs. Huther, Duval, and Planeix of Bureau Veritas; Mr. Engerrand of IACS and Bureau Veritas; Mr. Jens of IMC Messrs. Robinson, Thompson and Dobson of Lloyds Register of Shippir Dr. Akita of Nippon Kaiji Kyokai; Messrs. Stiansen, Gillis and Chen o the American Bureau of Shipping; and Captain Bell and Lt. Commande Henn and Loosmore of the U. S. Coast Guard. Other individuals who cooperated in providing needed information include: Messrs. Potter ar Schmitt of Methane Tanker Service Company; Dr. Glassfeld of General Dynamics and the Project Advisory Committee; Dr. Chryssostomidis c MIT and Chairman, Project Advisory Committee; and Mr. Coward of Newport News Ship Building and Dry Docking and the Project Advisory Committee. In addition, a special acknowledgement goes to Professor Dan Hoffman of the Webb Institute, who aided us in a consulting capacit and provided considerable input to this report in the areas of ship accelerations and wave-induced loads. All of the above named individuals were exceptionally cooperative and enthusiastic in providing us with valuable data, reports and personal opinions. Without their help the success of this program would have been limited. Also, the comments and guidance of the Project Advisory Committee have been beneficial to this study and their help has been most appreciated.

LIST OF SYMBOLS

a_R	magnitude of collision acceleration vector
a_x, a_y, a_z	longitudinal, transverse and vertical acceleration
B	ship breadth
C_B	block coefficient
D	ship draft
E	parameter of the Rayleigh distribution
E_L	cargo bulk modulus
g	acceleration due to gravity
GM	metacentric height
h	liquid depth
h_e	dynamic external pressure
H	tank height
$H_{1/3}$	significant wave height
K_p	pressure coefficient
K_{yy}	radius of gyration
ℓ	tank length
L_{cg}	location of center of gravity
L_{pp}	ship length between perpendiculars
P	sloshing impact pressure
P_d	dynamic internal pressure
P_o	cargo vapor pressure
P(x)	probability of measuring the given variation in the parameter x
ΔP	tank pressure above cargo vapor pressure
Q	discharge rate of the safety valves
Q	heat input into the tank in case of a fire
T_2	mean zero crossing period
V	service speed
x, y, z	Cartesian distances in the longitudinal, transverse and vertical directions

LIST OF SYMBOLS (Contd.)

γ	cargo specific gravity
ζ	wave height
θ	pitch amplitude
μ	cargo viscosity
ρ	cargo density
σ	stress
ϕ	roll amplitude
ω	wave frequency

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NOTES

I. INTRODUCTION

With the development of large liquefied natural gas (LNG) carriers, the problem of establishing tank load criteria for both design and regulatory purposes has become a critical area of interest. In order to establish these criteria, an estimate of tank accelerations must be made and the resulting loads evaluated. Further analysis is required to determine if these calculated accelerations truly represent the actual response of the vessel and to develop procedures for translating these accelerations into components of static and transient loads for use in designing the cargo tanks and their supporting structure. Therefore, the objectives of this program were to prepare a review of existing LNG tank structural load determination criteria, to evaluate their adequacy, and to plan programs to correct any deficiencies. These objectives were broken down into the following five phases: (1) a survey and review of load criteria presently employed in the design of cargo tanks for LNG carriers; (2) a comparison of motion and acceleration predictions resulting from these criteria to available ship motion calculations, model tests and full-scale data; (3) a survey of methods available for predicting wave-induced loads on LNG carriers; (4) a prediction and evaluation of maximum and cyclic loads on LNG tanks and supporting structures using existing criteria and recommending updated criteria where appropriate; and (5) development of model test and full-scale measurement programs to provide adequate data for verification of the load and acceleration criteria.

This work, in part, was based on the rules and regulations of eight classification societies or regulatory agencies. These agencies include the American Bureau of Shipping, Bureau Veritas, Det norske Veritas, Germanischer Lloyd, Lloyds Register of Shipping, Nippon Kaiji Kyokai, the U. S. Coast Guard, and the International Association of Classification Societies. The rules and regulations of these classification societies and regulatory agencies are specific in stating what loads are to be considered in designing an LNG tank but are generally non-specific in providing formulas or methods for establishing the magnitude of these loads. This results primarily from the fact that LNG ship tank design represents a new technology with unique structural and insulation designs evolving yearly. As a result, the classification societies' rules are somewhat general in order to cover the significant number of current tank designs and for accepting new tank designs. However, most of the societies have their own specific computer programs, and methods for calculating LNG tank loads. In most cases these computer programs are utilized by the ship builder to aid in the design and classification of a particular LNG ship. Since the objectives of this program were to evaluate the agencies' load criteria as they are written and since access to the agencies' computer programs is limited, the results presented in this report represent only an evaluation of the published rules and regulations. Also, it was the intent of this effort to provide a rational review of all the rules and regulations that were available to SwRI as of June 1974. Comparisons of the tanks loads as predicted by the various classification societies were utilized so an evaluation of these criteria could be made. It was not the intent to rate one society's rules over the others but only to conduct a research program into LNG tank load criteria which would be beneficial to all societies in updating their rules and regulations and to provide improved methods for the LNG tank designer.

II. LOAD CRITERIA REVIEW

II.1 Objective

The objective of the load criteria review phase is to provide a survey and review of the load criteria presently employed in the design of cargo tanks for LNG carriers. As part of this review the tank design criteria of the various classification societies and regulatory agencies were listed and summarized in each of 17 load categories. This program phase provided a complete listing and review of all the rules and regulations that were available to SwRI as of June 1974.

II.2 Agency Rules Reviewed

The agencies whose rules were reviewed and the dates of those documents which were available as of June 1974, are listed below:

American Bureau of Shipping (ABS) - Rules for Building and Classifying Steel Vessels, 1973.

Bureau Veritas (BV) - Rules and Regulations for the Construction and Classification of Steel Vessels, 1973.

Det norske Veritas (DnV) - Construction and Classification of Ships for Transport of Liquid Cargos and Liquified Gases, 1973.

Germanischer Lloyd (GL) - Rules for the Classification and Construction of Seagoing Steel Ships, 1973, Volume I.

Lloyds Register of Shipping (LR) - Rules and Regulations for the Construction and Classification of Steel Ships, 1968, and Chapter D, 1973.

Nippon Kaiji Kyokai (NK) - Provisional Rules for LNG Carriers, 1973.

U. S. Coast Guard (USCG) - Tentative Guide for the Review of Flammable Gas Carriers, April 1971.

International Associations of Classification Societies (IACS) - Unified Rules for Gas Tankers (Cargo Containment), March 1974.

II. 3 Tank Load Categories

The tank loads and motions considered for the criteria review are listed in Table II. 1. It is noted that the loads are broken down into 17 different categories. In each category, the criteria from the individual classification societies, were listed and summarized. In many cases, the classification societies had a separate section devoted to each of these specific load categories. In other cases, a load category was not considered as a separate item, and the regulations pertaining to this particular load had to be extracted from another category which combined two or more loads.

For each load category, the listed criteria are given essentially in the words and with the nomenclature of the individual agencies in Appendix A. Each society refers to the different LNG tank configurations in their own nomenclature. However, all societies recognize three basic tank designs: independent, membrane, and integral tanks. As can be seen in Table II. 2, these three broad categories are further subdivided. General characteristics of the common tank configurations are:

(1) Independent Tanks:	self-supporting tanks; generally do not contribute to the structural strength of the ship.
(a) Gravity Tanks:	primarily prismatic in shape; loads are carried by bending stress.
(b) Pressure Tanks:	generally spherical or cylindrical in shape; loads are carried by membrane stress.
(2) Membrane Tanks:	non-self-supporting gravity tanks; loads are carried by the ship's hull through a thin membrane and insulation; designed so that thermal expansion or contraction is compensated for without undue stressing of the membrane.
(3) Integral Tanks:	generally prismatic in shape; tanks form an integral part of the ship's hull and are therefore subjected to the same loads as the adjacent hull structure.

In addition, the agencies use different symbols for the various parameters utilized in determining tank loads. Table II. 3 shows the nomenclature used by the classification societies for the various important parameters.

TABLE II. 1. LOADS AND MOTIONS CONSIDERED IN THE CRITERIA REVIEW

Vapor pressure
Liquid head
Static design external pressure
Weight of tank and contents
Still-water hull deflections
Static inclination
Collision loads
Thermal gradients
Wave-induced loads
Dynamic hull deflections
Accelerations at tank center of gravity
Dynamic external hull pressure
Dynamic internal pressure
Sloshing
Vibrations
Fatigue loads
Fracture loads

TABLE II. 2. NOMENCLATURE USED BY THE CLASSIFICATION SOCIE FOR DIFFERENT TANK CONFIGURATIONS

	ABS	BV	DnV	GL	LL	NKK
Independent						
Gravity type						
Scantlings based on standard practice	Structural Tanks $P_o^* < 10$	Gravity Cargo Tanks $P_o \leq 10$	Type A I	G3A	Structural Tanks $P_o < 10$	Type A
Scantlings based on extensive stress analysis		Self Supporting Cargo Tank	Type A II	G3B		Type B
Pressure Vessel						
High pressure						
Scantlings based on standard practice	Pressure Vessels $P_o > 10$	Pressure Cargo Tanks $P_o \geq 0.70$	Type B $P_o > 10$	P2 $P_o > 42.8$	Structural Tanks $P_o > 10$	Type C $P_o > 10$
Scantlings based on extensive stress analysis						Type B $P_o \geq 10$
Low pressure						
Scantlings based on standard practice				P1A $10 < P_o < 42.8$		Type C $P_o \leq 10$
Scantlings based on extensive stress analysis				P1B $10 < P_o < 42.8$		Type B $P_o \leq 10$
Membrane						
Scantlings based on standard practice	Non-Structural Tanks	Integrated Cargo Tank	Type C	G_2 $P_o \leq 4$		$P_o \leq 10$
Increased scantlings				G_2 $P_o \leq 10$		
Scantlings based on extensive stress analysis						Semimembran $P_o \leq 10$
Integral						
Scantlings based on standard practice	Specially Considered No Specific Regulations	Not Currently Allowed for LNG	Not Currently Allowed for LNG	Not Currently Allowed for LNG	Specially Considered No Specific Regulations	Not Currently Allowed for LNG
Increased scantlings						

*Maximum Allowable Vapor Pressure (psig)

TABLE II. 3. NOMENCLATURE USED BY THE CLASSIFICATION SOCIETIES FOR VARIOUS SHIP PARAMETERS

Parameters	ABS	BV	DnV	GL	LR	NK	USCG	IACS	This Report
Cargo Parameters									
specific weight		δ_o	γ	γ		γ		γ	γ
vapor pressure	P_o	P_o	P_o	P_o	P_o	P_o	P_o	P_o	P_o
Tank Parameters									
tank height		H	h	h_t		h			H
liquid height		d	h			h			h
Ship Parameters									
block coefficient			C_B	δ		C_b		C_B	C_B
length		L	L	L		L	L	L	L
breadth		B	B	B		B		B	B
service speed		V	V	V		V_s		V	V
metacentric height		GM	GM	M_BG		GM		GM	GM
Response Parameters									
longitudinal acceleration			a_x	a_x		a_x		a_x	a_x
transverse acceleration			a_y	a_y		a_y		a_y	a_y
vertical acceleration		γ	a_z	a_z		a_z		a_z	a_z
external static pressure			P_{ed}	P_{as}		P_e			
dynamic liquid head:									
at ship bottom			h_{eb}	h_B		h_{EB}			
at water line			$h_{e\ell}$	h_S		h_{EL}			
at deck			h_{ed}			h_{ED}			
dynamic internal pressure		h_d	P	P_d		h_j		h_{eq}	
sloshing pressure			h_i			h_s			

Reference to Tables II. 2 and II. 3 will allow the various symbols used in Appendix A and the remainder of this report to be understood. Also the paragraph identification numbers from each society's rules are repeated in Appendix A for cross-reference. Therefore, Appendix A represents a compilation of eight classification agencies' rules, just as stated by the societies, for each load criteria. Statements and paragraphs reproduced exactly from the rules are indented from the remainder of the text and appear in quotation marks. Our own comments and paraphrasing of the rules retain the original margins. Since LNG transport is currently limited to independent and membrane tank configurations, with the exception of the USCG, no regulations for integral tanks are presented for agencies other than the USCG. The USCG and IACS accept integral tanks providing the temperature of the hull never falls below $-10^{\circ}C$.

The current differences and similarities among the agencies' rules are summarized at the end of each load category in Appendix A. The detailed evaluation of the different tank load criteria are given in subsequent sections of this report with separate chapters devoted to the acceleration and wave-induced load criteria because of their importance.

III. COMPARISON OF ACCELERATION CRITERIA TO AVAILABLE ACCELERATION DATA AND STATISTICAL CALCULATIONS

III. 1 Introduction

There are only very limited experimental acceleration data for model or full-scale ships in the open literature. This may be due in part to the competitive nature of the shipbuilding industry and the limited history of LNG ship operation. Data from models of LNG ships are available, but these data are for regular waves and primarily for the determination of the ship transfer function. Long-term acceleration predictions from these model data cannot be obtained. For this reason, comparisons of actual acceleration data with the agencies' formulas were made for ships with length/draft, draft/breadth, and length/breadth ratios similar to those of LNG ships. Accelerations obtained with computer programs such as SCORES* [1] were compared with the agencies' formulas for several LNG ships. In order to protect the confidential nature of some reports, only limited identification can be presented here. The following reports on accelerations were used for comparison purposes:

Full-Scale Acceleration Data

- "Acquisition and Analysis of Acceleration Data" [1]
- "Wave Loads on the Fore-Ship of a Large Tanker" [2]
- "Ship Response Results from the First Operational Season Aboard the Container Vessel S. S. Boston" [3]

Acceleration Calculations

- Calculations of Accelerations on Four LNG Ships by the Computer Programs SCORES*

The agencies' formulas predict the maximum acceleration that is expected to be encountered during the service life of the ship. Usually 20 years or 10^8 wave encounters are used for the ship's lifetime. Computer programs in use by the agencies also contain statistical packages which extrapolate the short-term predictions to long-term. Measured data, therefore, must be extrapolated out to 10^8 cycles before valid comparisons with agency formulas can be made.

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[1] The original SCORES package had no provision for acceleration output. A modified version of SCORES, referred to in this report as SCORES*, includes a provision for calculating and printing short-term accelerations. Extrapolation of accelerations from SCORES* to the long-term was accomplished by means of the Webb statistical package.

III.2 Predictions of Long-Term Accelerations from Measured Data

The full-scale measurements presented in this section were collected on three ships during actual service voyages across the North Atlantic. The instrumentation systems and the process of collecting the data were quite similar. Basically, the data acquisition system consisted of an accelerometer, a tape recorder, and a time reference. Recordings of acceleration variations were typically made every eighth hour for a duration of 30 minutes when the ship was in open water. In addition, a provision for continuous monitoring was available during severe weather. The visually estimated weather conditions were recorded in the ship's log. From each of the 30-minute records, the peak-to-peak variations of vertical accelerations were measured and classed according to amplitude and sea condition. The details of the instrumentation system along with the pertinent ship dimensions are found in Table III.1.

Bailey, et al., [1] found that the peak-to-peak acceleration variations obtained from each record could be described satisfactorily by a Rayleigh distribution as given by

$$P(x) = \frac{2x}{E} \exp\left(\frac{-2x^2}{E}\right) \qquad (1)$$

where

$$\sqrt{E} = \left[\frac{\Sigma n_i x_i^2}{N}\right]^{\frac{1}{2}} \qquad (2)$$

x_i = magnitude of a peak-to-peak variation in the parameter x

n_i = the number of observations in the i-th range

N = total number of observations

$P(x)$ = probability of measuring the given variation in the parameter x

$\sqrt{E}$ = the single Rayleigh parameter.

In addition, the $\sqrt{E}$ values (rms acceleration variation in each 30-minute record) obtained at different times and in different sea conditions were also found to be Rayleigh distributed.

Pederson[2] also assumed that the peak-to-peak acceleration variations followed a Rayleigh distribution as given by

$$P(x) = \frac{x}{2E} \exp\left(-\frac{x^2}{4E}\right) \qquad (3)$$

TABLE III. 1. CHARACTERISTICS OF THE THREE SHIPS ON WHICH ACCELERATIONS WERE MEASURED

Parameter	S. S. Wolverine State	S. S. Boston	Large Tanker
Ship type	Dry cargo vessel	Container vessel	Tanker
Length L_{pp}(m)	151.2	152.0	252.
Breadth B(m)	21.8	21.8	39.
Draft D(m)	10.0	9.3	14.
Service Speed V(kn)	17.0	17.0	15.5
Block Coefficient C_B	0.61	0.61	0.7
Length/draft ratio	15.1	16.3	18.
Length/breadth ratio	6.9	7.0	6.5
Breadth/draft ratio	2.2	2.3	2.8
Acceleration measured	Vertical	Vertical, transverse	Vertical
Accelerometer location	Bow	Bow, midship, stern	Bow
(x, z) (m)	(68.8, 22.6)	Bow (66.4, 4.1) Midship(-8.5, -2.8) Stern (-65, 2.1)	(115., 8.2)

By definition, $\sqrt{E}$ used by Pederson is one-half the rms value of the peak-to-peak acceleration as used by Bailey. Pederson obtained the Rayleigh parameter by plotting the cumulative probability distribution as given by Equation (4) on Weibull probability paper.

$$P\ (x \leq x_1) = 1 - \exp\left(-\frac{x_1^2}{4E}\right) \tag{4}$$

By using $P(x) = 0.63$ in Equation (4), and solving $\sqrt{E} = x/2$, Pederson found that this graphical estimation of $\sqrt{E}$ gave, on the average, acceptable values.

Fain, et al., [3] presented no analysis of the accelerations measured on the S. S. Boston. Neither Bailey nor Pederson attempted to extrapolate their acceleration data beyond the short-term. In order to make a valid comparison of these data to the agencies' formulas, we need to calculate the largest acceleration to be expected in 10^8 cycles. The extrapolation to the extreme value will be made in three ways: the exponential method, the Weibull method, and using the combined Rayleigh-Normal distribution which is the method used by Webb Institute.

The input data for the exponential and Weibull methods are the same. The maximum peak-to-peak acceleration variations are first classed according to amplitude. The range of accelerations was broken into several constant width bands and the probability of exceeding a given peak-to-peak variation in acceleration was calculated. Table III. 2 contains a summary of peak-to-peak measurements of accelerations aboard the S. S. Wolverine State. The cumulative probability or the probability of not exceeding a given acceleration was plotted against the midpoint of the acceleration range on log-log paper for the exponential prediction and on Weibull paper for the Weibull prediction. Figure III. 1 presents the exponential prediction obtained from the S. S. Wolverine State data. Figures III.2 through III.7 present the Weibull predictions for all three ships.

The long-term acceleration value for both the Weibull and exponential methods is obtained by fitting a straight line to the data and extending the line to a probability level of 10^{-8}. Figure III. 1 shows 0.88 g's to be exceeded two times out of every 100 variations on the S. S. Wolverine State. Unfortunately, the curve is not linear, and the deviation from a straight line becomes more severe as the probability level approaches 10^{-8}, so no further extrapolation is possible. Similar nonlinear results from the exponential method were obtained for the S. S. Boston, so these plots are not presented in this report.

In contrast, the Weibull prediction graphs, Figures III. 2 - 7 generally exhibit good linearity. By extrapolating the S. S. Wolverine State data (Figure III. 2) to a probability level corresponding to 10^8 cycles, the extreme

TABLE III.2. SUMMARY OF ACCELERATION TEST DATA FOR "S.S. WOLVERINE STATE"

P-P Accel. Range, $\overline{X}$	Number of Events in Sea State											Total Events	Probability of Measuring $\overline{X}$	Cumulat Probabil
	1	2	3	4	5	6	7	8	9	10	11			
0 - .10	28	53	86	55	43	5	1	1	0	0	0	272	.1751	.1751
.11 - .20	10	33	88	72	45	6	2	1	2	0	0	259	.1668	.3419
.21 - .30	2	27	61	84	54	22	3	3	0	0	0	264	.1700	.5119
.31 - .40	1	17	37	47	35	19	11	5	1	0	0	174	.1120	.6239
.41 - .50	0	10	26	33	28	25	7	0	2	1	0	133	.0856	.7095
.51 - .60	0	9	16	19	30	15	6	2	2	3	1	103	.0663	.7758
.61 - .70	1	5	6	23	15	14	15	6	4	3	0	91	.0586	.8344
.71 - .80	1	0	9	16	25	11	21	17	4	2	4	110	.0708	.9052
.81 - .90	0	2	3	14	10	13	11	6	5	0	4	69	.0444	.9496
.91 - 1.0	0	0	3	5	8	8	6	6	2	1	1	40	.0258	.9754
1.01 - 1.1	0	1	0	1	0	5	6	3	1	0	0	17	.0109	.9863
1.11 - 1.2	0	0	0	1	2	3	4	5	0	0	0	15	.0097	.9960
1.21 - 1.3	0	0	0	0	0	1	2	1	0	0	0	4	.0026	.9986
1.31 - 1.4	0	0	0	0	0	0	0	0	0	0	0	0	--	--
1.41 - 1.5	0	0	0	1	0	0	0	0	0	0	0	1	.0006	.9992
1.51 - 1.6	0	0	0	0	0	0	0	0	0	0	0	0	--	--
1.61 - 1.7	0	0	0	0	0	0	0	0	0	0	0	0	--	--
1.71 - 1.8	0	0	0	0	0	0	0	1	0	0	0	1	.0006	.9998
Totals	53	157	335	371	295	147	95	57	23	10	10	1553	.9998	

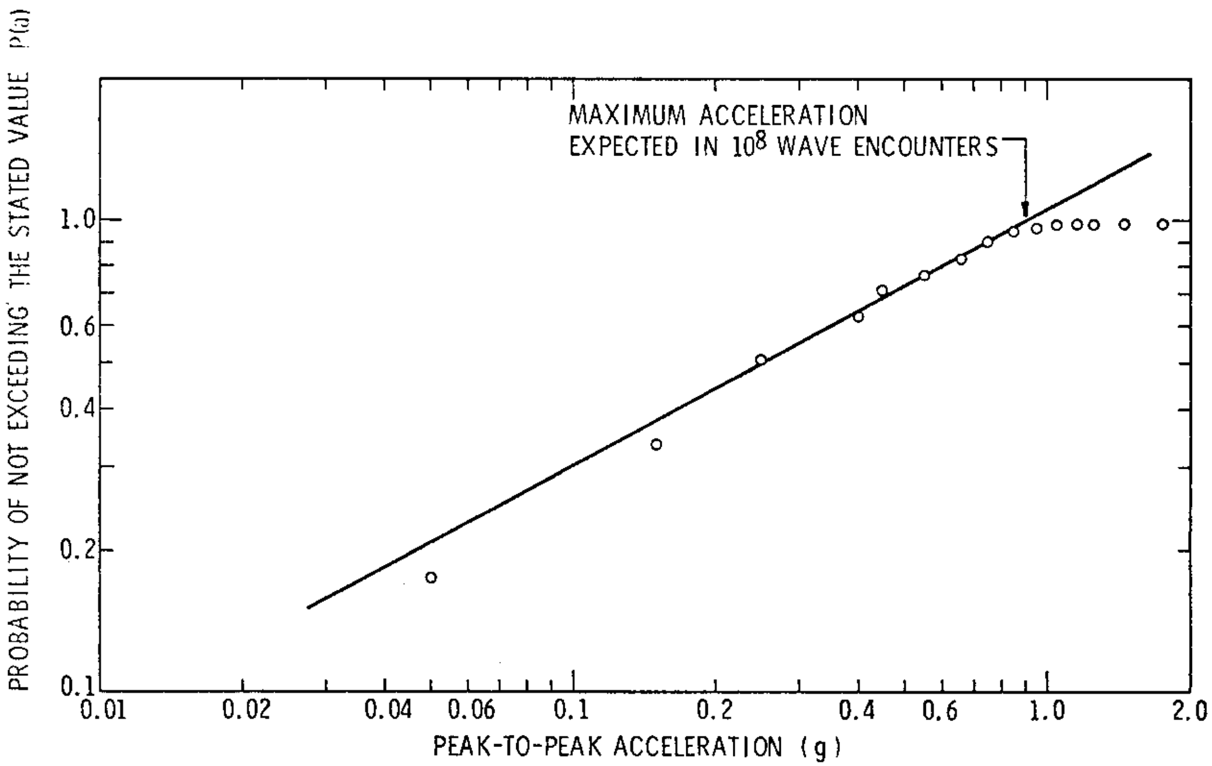


FIGURE III. 1. EXPONENTIAL PREDICTION OF LONG-TERM ACCELERATION FOR THE S. S. WOLVERINE STATE

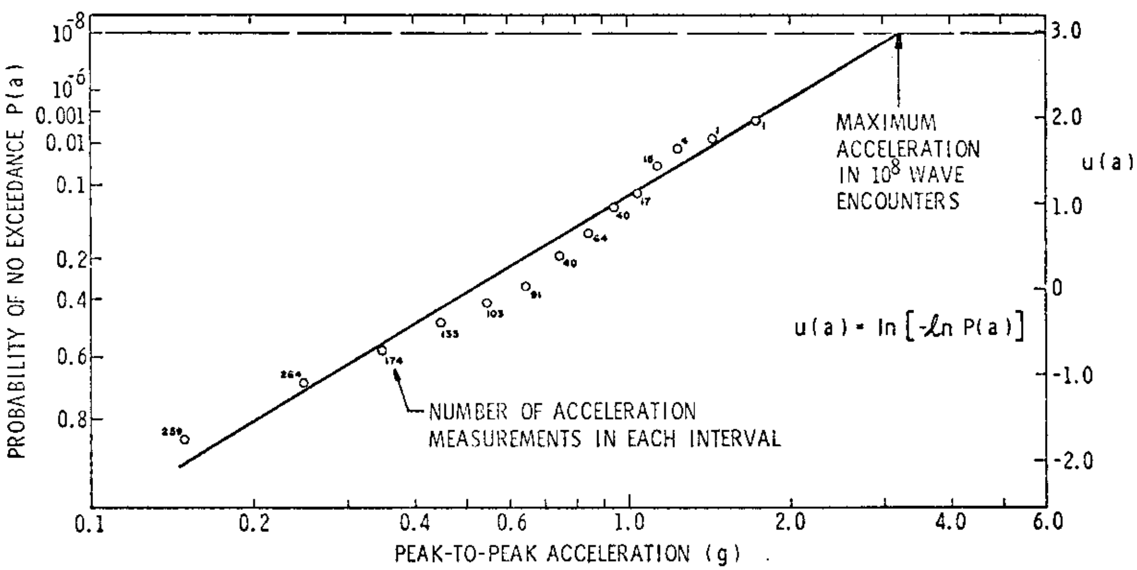


FIGURE III. 2. WEIBULL PREDICTION OF LONG-TERM ACCELERATIONS

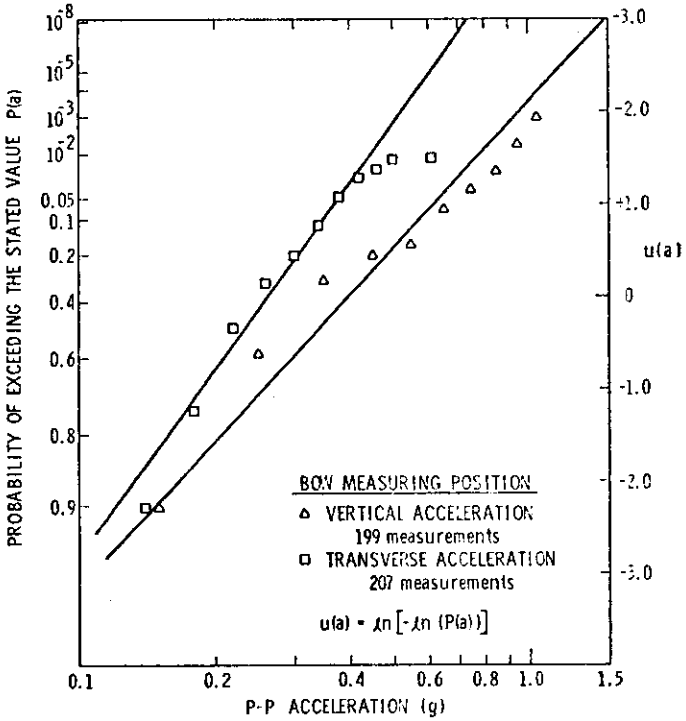
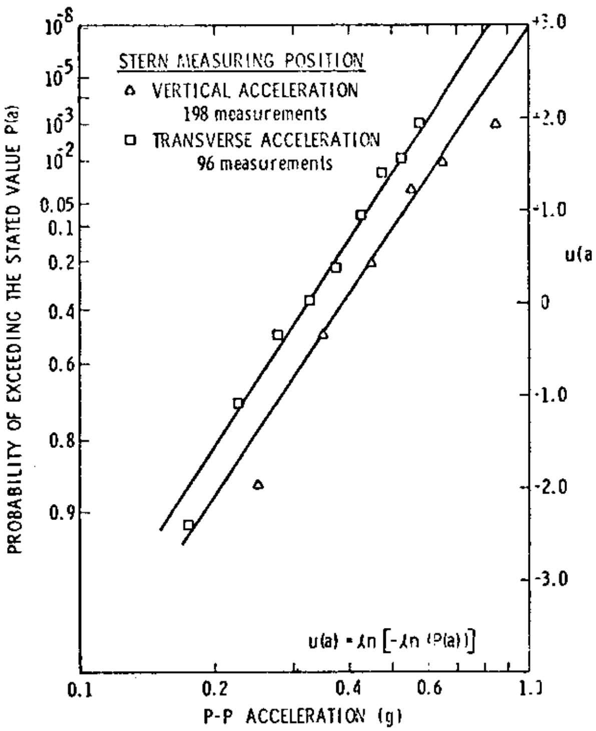


FIGURE III.3. WEIBULL PREDICTION OF LONG-TERM ACCELERATION AT THE BOW OF THE S.S. BOSTON



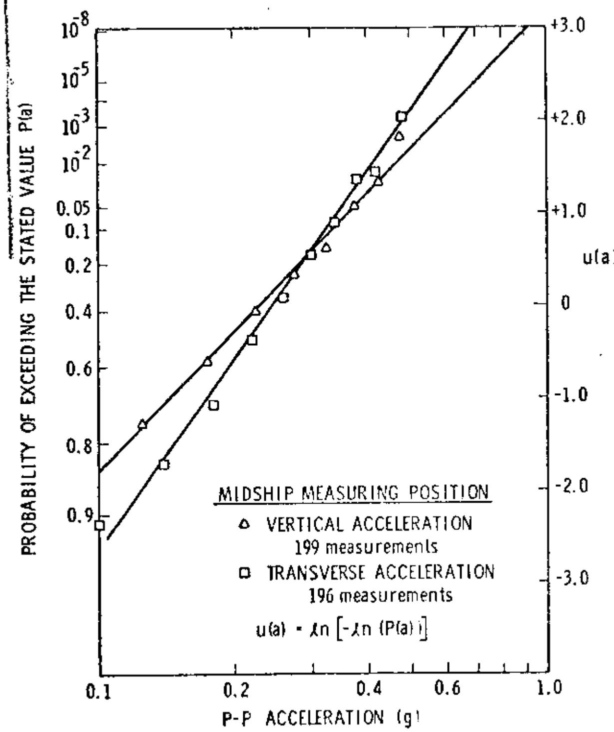


FIGURE III.4. WEIBULL PREDICTION OF LONG-TERM ACCELERATION AT MIDSHIPS ON THE S.S. BOSTON

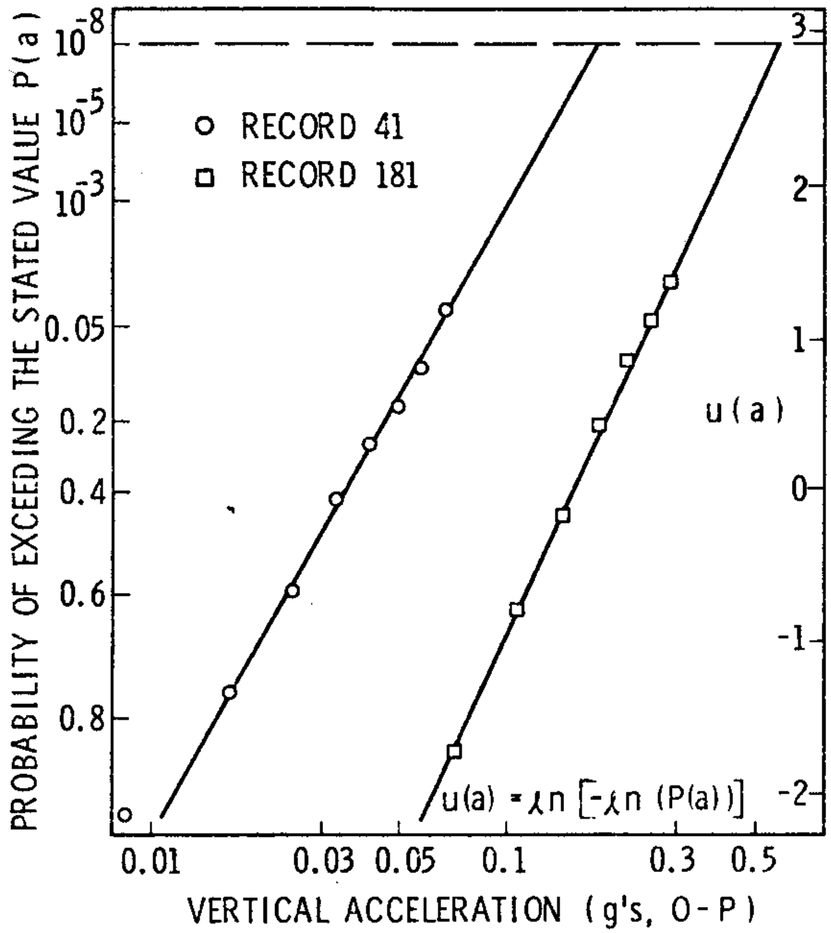


FIGURE III.6
WEIBULL PREDICTION OF LONG-TERM ACCELERATIONS FOR THE LARGE TANKER OBTAINED FROM TWO INDIVIDUAL RECORDS

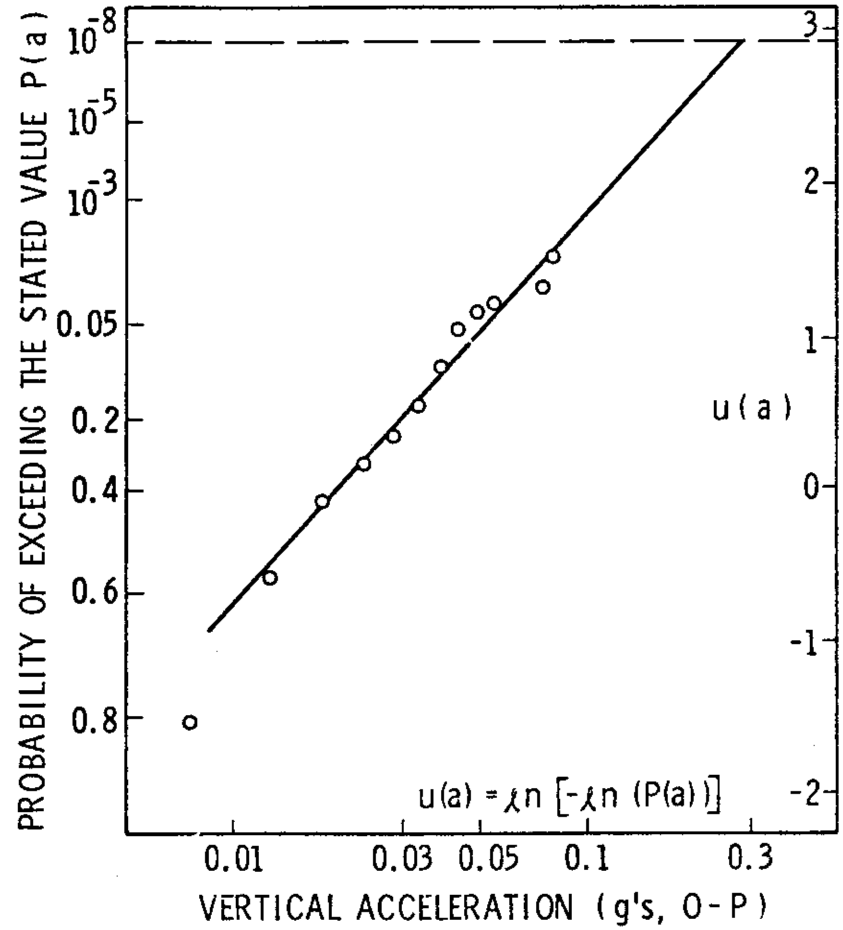


FIGURE III.7
WEIBULL PREDICTION OF LONG-TERM ACCELERATIONS FOR THE LARGE TANKER OBTAINED FROM ALL RECORDS

value is 3.25 g's peak-to-peak. As this acceleration is unreasonably large, we would conclude that the deviation from linearity for the S. S. Wolverine State is too severe to assure that the extrapolation will yield acceptable results.

The number of data points used to compute the probability of exceedance are also shown on Figure III. 2 for each value calculated. It is noted that at the higher probability levels, only one data point was available. If data below approximately 0.6 g and above 1.3 g could be ignored, a good prediction would be obtained from the Weibull plot. Although there is some justification for discarding the larger value because of insufficient data, the lower acceleration values are based on a large number of data points and cannot be ignored.

The Weibull predictions for the S. S. Boston yielded, in general, more reasonable predictions of extreme accelerations. However, only about 200 points were used to prepare Figures III. 3 - 5 for the S. S. Boston, while 1500 points were used in the S. S. Wolverine State predictions. So, until more data have been accumulated on the S. S. Boston, these results should be considered preliminary. Vertical accelerations, measured on the S. S. Boston at a position chosen to correspond with the measuring point on the S. S. Wolverine State, were used to prepare Figure III. 3. The long-term acceleration obtained from this plot is 1.45 g peak-to-peak, which is approximately half the value obtained from the S. S. Wolverine State. Figure III. 8 is a comparison of measured bow vertical accelerations onboard the S. S. Boston and the S. S. Wolverine State. As was noted above, the location of the forward measuring point on the S. S. Boston was chosen to coincide with the location of the accelerometer onboard the S. S. Wolverine State. The figure shows that although the relative amplitude of peak-to-peak accelerations are similar for the two ships, the general trend is different. The acceleration response of the S. S. Boston is nearly flat for sea states 0 - 6, and rises sharply thereafter. The measured accelerations on the S. S. Wolverine State gradually increase from 0.15 g peak-to-peak in sea state 1, to a maximum of 0.7 g peak-to-peak in sea state 8. Part of the difference between the two acceleration trends may be due to the difference in weather distributions encountered by each ship. As shown in Table III. 3, a higher percentage of severe weather was encountered by the S. S. Wolverine State. Based on the gradual rise in acceleration response, one would expect a higher long-term acceleration prediction to result from the accelerations measured on the S. S. Wolverine State, as is the case (Figure III. 3).

The Weibull predictions of extreme accelerations for the large tanker are shown in Figures III. 6 and III. 7. Figure III. 6 presents the data obtained from records 41 and 181, which were both obtained in sea state 5 (ship velocity was approximately 14 knots). Long-term extrapolation of these data resulted in considerably different accelerations even though both recordings were made in the same sea condition and at the same ship location. By extrapolating the straight lines in Figures III. 6 and III. 7 to a

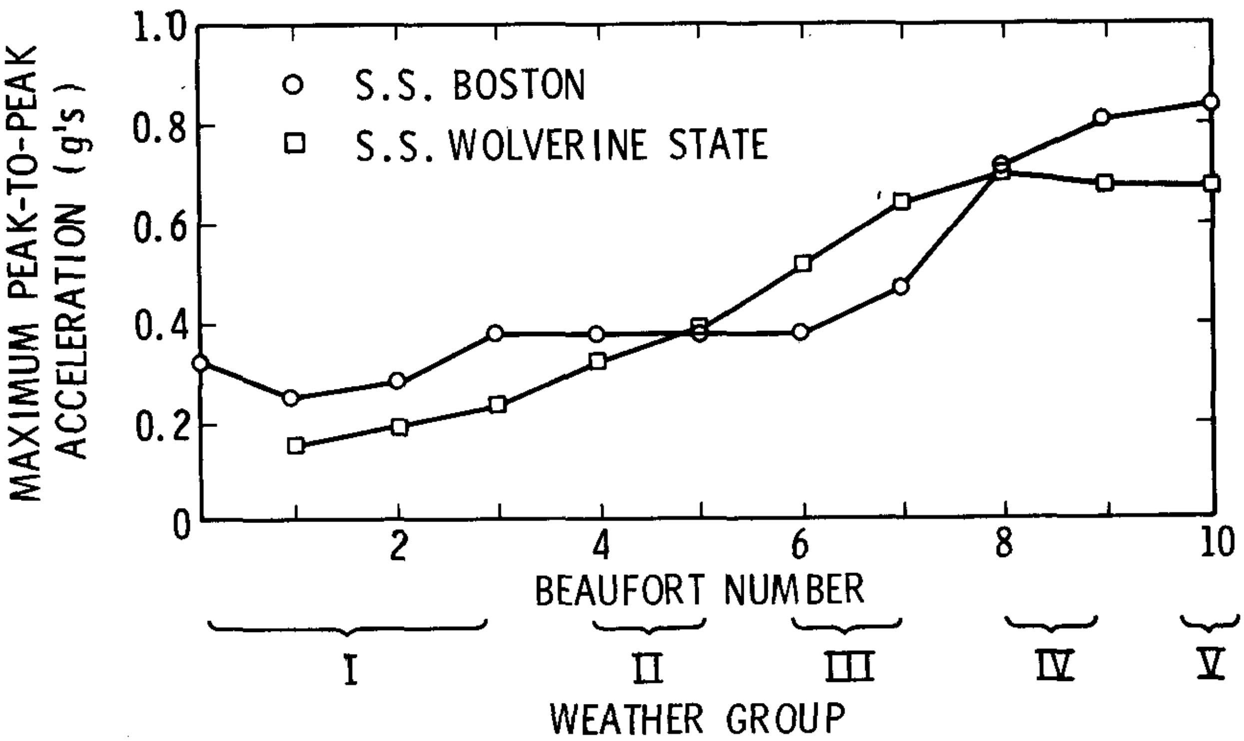


FIGURE III. 8. COMPARISON OF BOW VERTICAL ACCELERATIONS ON THE S. S. BOSTON AND THE S. S. WOLVERINE STATE

TABLE III. 3. COMPARISON OF WEATHER ENCOUNTERED BY THE THREE SHIPS

Weather Group	Beaufort No.	Percentage of Accelerations Measured in Each Weather Group		
		S. S. Wolverine State	S. S. Boston	Large Tanker
I	0-3	26.	45.	70.
II	4-5	43.	39.	26.
III	6-7	22.	7.	3.5
IV	8-9	7.	8.	0.
V	≥ 10	2.	1.	0.
Approx. No. of Events Recorded		1550	200	230

probability level corresponding to 10^8 wave encounters, the largest acceleration to be expected in 10^8 wave encounters is obtained. These predictions assume, of course, that the ship continues to operate in seas identical to those for which the record was made over its entire lifetime. The extreme value determined by this method will be different for each record. In fact, a distribution of $\sqrt{E}$ values can be drawn providing sufficient numbers of records are obtained. Figure III. 7 is such a distribution of $\sqrt{E}$ values. By extending the distribution of $\sqrt{E}$ values to a probability level of 10^{-8}, we obtain the largest expected acceleration in 10^8 wave encounters. The maximum $\sqrt{E}$ acceleration obtained in this manner from Figure III. 7 yielded only 0.3 g's. As noted on Page 10 $\sqrt{E}$ in Pederson's report is one-half the rms value of the peak-to-peak variations. Therefore, we must multiply by two to get the rms value, and then multiply again by some factor to convert the rms value to a peak-to-peak variation. If the acceleration variations were purely sinusoidal, the conversion between rms and peak-to-peak would be $\sqrt{2}$. Of course, the accelerations are not sinusoidal, but a conversion factor of $\sqrt{2}$ may provide a lower bound for the conversion. In order to estimate a reasonable conversion factor, Table III. 4 was prepared. In this table the average peak-to-peak and rms accelerations obtained on the S. S. Wolverine State are listed for each sea condition encountered. The ratio of the peak-to-peak acceleration to the rms acceleration was computed for each sea condition, and the average was obtained. Thus, a conversion factor between peak-to-peak and rms acceleration was found to be approximately 1.9 ($\sigma = 0.338$). Finally, the extreme acceleration predicted by the Weibull method (and taking 2.1, the average of the largest six accelerations, rather than 1.9 for the conversion factor to ensure conservatism) is 1.24 g's p-p (0.85 g's p-p if $\sqrt{2}$ is used rather than 2.1).

The final prediction of extreme values was made using the combined Rayleigh-Normal distribution. The validity of this prediction, as noted by E. G. U. Band[4], hinges on four assumptions:

(1) Each data record is Rayleigh distributed about a rms value $\sqrt{E}$.

(2) The weather conditions experienced by a ship can be represented by five weather groups which are a reclassification of the Beaufort scale.

(3) The value of $\sqrt{E}$ from each record are normally distributed in each weather group, and a mean value, m, and a standard deviation, σ, can be assigned to the distribution.

(4) The contribution of each weather group can be weighted by taking into account the probability of encountering a particular sea condition on a given route.

TABLE III. 4. DETERMINATION OF A CONVERSION FACTOR BETWEEN RMS AND P-P ACCELERATIONS USING THE WOLVERINE STATE MEASUREMENTS

Sea State	No. Measurements	Average Acceleration		Ratio (P-P/rms)
		P-P	rms	
1	53.	0.15	0.11	1.36
2	157.	0.21	0.16	1.31
3	335.	0.24	0.15	1.60
4	371.	0.33	0.19	1.74
5	295.	0.38	0.21	1.81
6	147.	0.52	0.24	2.17
7	95.	0.68	0.29	2.34
8	57.	0.70	0.32	2.19
9	23.	0.68	0.32	2.13
10	10.	0.66	0.32	2.06
11	10.	0.80	0.37	2.16
		Average Ratio		1.897
		Standard Deviation		0.338

The method for computing the largest acceleration in 10^8 cycles, outlined in the following paragraph, was obtained from E. G. U. Band's work. Unfortunately, this method requires as its input rms accelerations as a function of weather conditions. For this reason, predictions using the combined Rayleigh-Normal method can be obtained only for the S. S. Wolverine State.

For each weather group, the mean and standard deviation of all rms acceleration variations were calculated. Then, using Figure III. 9 (reprinted from Reference 5), the acceleration amplitude to be expected in 10^N cycles in each weather group was found graphically. Table III. 5 presents the results of the above calculations. To find the maximum acceleration expected in 10^8 cycles, each maximum expected acceleration for a given weather group was multiplied by the probability of encountering the weather group. Summing over all weather groups yields the desired extreme values. As is indicated in Table III. 5, the maximum anticipated acceleration in 10^8 cycles is 1.94 g's peak-to-peak for the S. S. Wolverine State in the North Atlantic.

The procedure outlined above may be used to obtain the maximum acceleration to be expected in 10^N cycles. Figure III. 10 presents a plot of the maximum expected peak-to-peak acceleration for the various probability levels from 10 to 10^{10}, using the combined Rayleigh-Normal distribution.

Comparisons of acceleration predictions from the three methods described above were made to the accelerations obtained from the agency formulas. The ship dimensions and the location of the point where the acceleration formulas were evaluated are summarized in Table III. 1. The coordinates used in the agency formulas coincide with the location of the accelerometers on each ship, even though the agency formulas are designed to predict accelerations at the center of gravity of the tanks. The acceleration predictions from the agency formulas for all but the midship measuring point on the S. S. Boston correspond to a measuring point that is close to the exposed deck of the respective ships.

ABS, LR, and the USCG do not provide acceleration formulas as the other agencies do. instead, ABS, LR, and USCG provide roll, heave and pitch amplitudes and periods which, when superimposed, are to be used in designing the tank structure. From these amplitudes and periods the accelerations were derived assuming sinusoidal motions. For transverse acceleration, the required motion due to rolling is 60° in 10 seconds. Therefore, the equation of motion is:

$$\phi = \frac{\pi}{6} \sin \left(\frac{2\pi}{10} t\right)$$

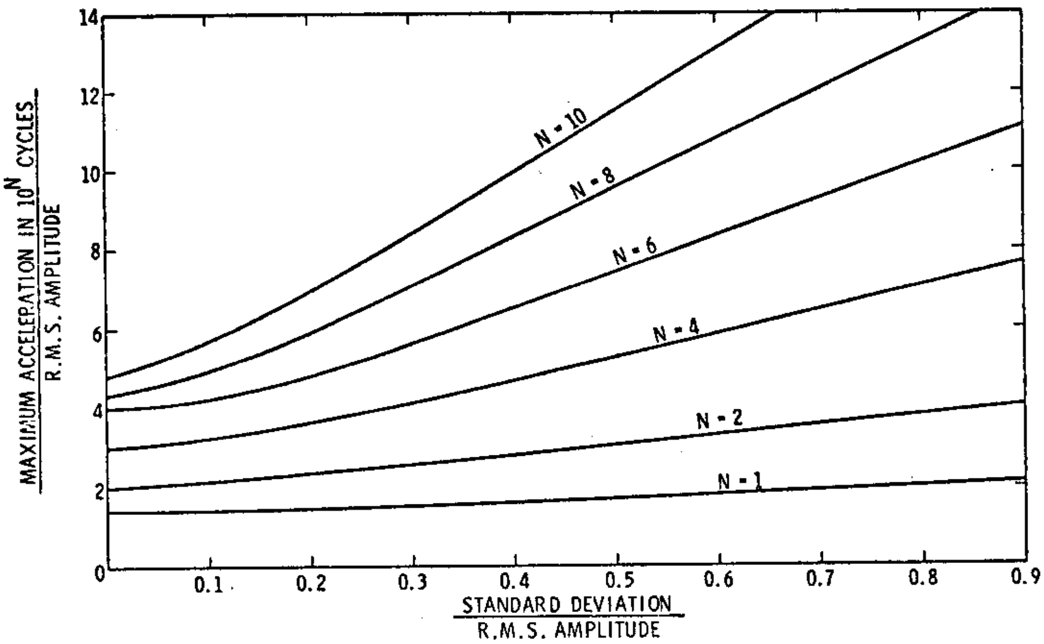


FIGURE III.9 GRAPHICAL SOLUTION USED IN COMBINED RAYLEIGH-NORMAL EXTREME ACCELERATION PREDICTION

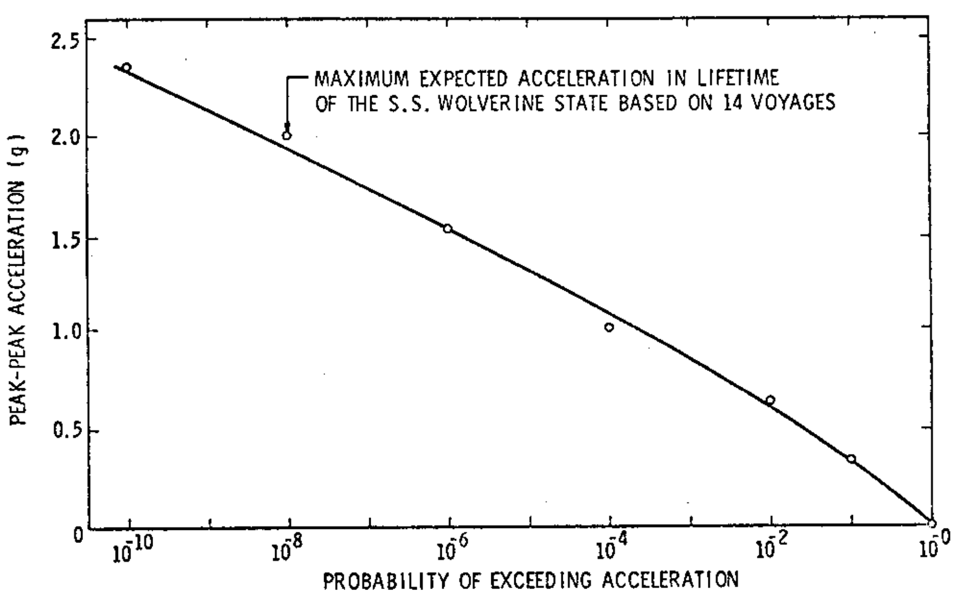


FIGURE III.10. RAYLEIGH-NORMAL PREDICTION OF LONG-TERM ACCELERATIONS FOR THE S.S. WOLVERINE STATE

TABLE III. 5. EXTREME ACCELERATION VALUES FOR "S. S. WOLVERINE STATE" AS PREDICTED BY THE RAYLEIGH-NORMAL METHOD

Weather Group	Beaufort Number	Mean m	Standard Deviation, σ	A_8, Expected Acceleration in 10^8 Cycles in Weather Group	P, Probability of Encountering Each Weather Group*	$P \cdot A_8$
I	1, 2, 3	0.1487	0.0935	1.65	0.42	0.693
II	4, 5	0.1879	0.1079	1.97	0.32	0.6304
III	6, 7	0.2592	0.1202	2.36	0.185	0.4366
IV	8, 9	0.3199	0.1086	2.42	0.065	0.1573
V	10,11,12	0.3474	0.0876	2.26	0.0098	0.0221
						$\Sigma PA_8 = 1.939$

* Probabilities from Bennet for the North Atlantic Route

and the angular acceleration is

$$\ddot{\phi} = \frac{\pi}{6}\left(\frac{2\pi}{10}\right)^2 \sin\left(\frac{2\pi}{10}\,t\right),$$

The transverse acceleration is the product of the amplitude of angular acceleration and the pivot arm z:

$$a_y = \ddot{\phi}z = \frac{4\pi^3}{600.}\,z$$

The longitudinal acceleration is derived in a similar manner from the centrifugal force produced by pitching. The pitch amplitude is 12°, and the period is 7 seconds. The resulting longitudinal acceleration is given by:

$$a_x = \dot{\theta}x = \frac{\pi^2}{105.}\,x$$

The vertical acceleration is derived from the combined effects of heave and pitch.[2] The heave amplitude is $L/40$ and the period is 8 seconds; The pitch amplitude and period are as above. The vertical acceleration is then:

$$a_z = \ddot{\theta} + \ddot{z} = \frac{\pi^3}{367.5}\,x + \frac{\pi^2}{1280.}\,L_{pp}$$

where

θ	=	instantaneous pitch orientation
ϕ	=	instantaneous roll amplitude
x, z	=	position where accelerations are to be calculated.

The acceleration predictions obtained from the full-scale measurements and the agency formulas are summarized in Table III.6. Since the <u>ABS</u>, <u>LR</u>, and <u>USCG</u> formulas, and the <u>GL</u>, <u>NK</u>, and <u>IACS</u> formulas are equivalent, the acceleration values obtained from these agencies are grouped on one line in this and subsequent tables. The <u>ABS</u>, <u>LR</u>, and <u>USCG</u> predictions consistently are smaller than the corresponding entries for the other agencies. The agency formulas are otherwise generally conservative relative to the Weibull and combined Rayleigh-Normal predictions. The excepcompared to the agencies' ±1.26 g's. The nonlinearity of the Weibull plot in Figure III.2 may perhaps explain the high Weibull prediction. Because the combined Rayleigh-Normal method takes into account the probability of encountering various weather conditions on different routes, we feel that its prediction will naturally be more consistent and realistic than the Weibull prediction which ignores weather variations. Indeed, the Rayleigh-Normal procedure produced ±0.97 g's, which compared favorably with the agency calculations. Since measurements were made on the S. S. Boston at three points along the length of the ship, we can compare the trend of the agency formulas with length to the full-scale extrapolations. The trend of the agencies' predictions is to have large vertical and horizontal accelerations at the bow and stern measuring points relative to the midship accelerations. The Weibull predictions for horizontal and vertical accelerations, by contrast, reflect a relatively shallow variation in acceleration as a function of measuring position. We assumed that the BV acceleration formulas provided acceleration values corresponding to the 10^{-8} probability level, although no specific probability level was given in the rules.

III.3 Predictions of Long-Term Accelerations from Statistical Calculations

The calculations presented in this section were performed in the course of classifying four LNG ships according to ABS regulations. The details of the four ships are found in Table III.7; however, only limited information is given in order to protect the confidential nature of the data. The ABS version of SCORES* used to perform the calculations consisted basically of the original SCORES transfer function, Lewis-form description of the ship's hull, real Atlantic wave spectra and the statistical package developed by Webb Institute. A more complete description of the statistical calculation is found in Chapter IV of this report. The output of the SCORES* program according to ABS is peak-to-mean, long-term accelerations.

For simplicity, comparisons for the first three ships will be presented separate from comparisons for Ship No. 4. For each ship the program SCORES* was used to calculate the extreme acceleration for several loading conditions and a variety of locations within the ship. For brevity, the maximum acceleration predicted by SCORES* for each loading condition is summarized in Table III.8. For each ship, the location of maximum acceleration as predicted by SCORES*, regardless of loading condition, was used to calculate extreme accelerations from the agency formulas. Table III.9 presents these comparisons. As is noted in the table, ABS, LR, and USCG regulations are not applicable to these three ships, as they are all longer than 183 meters. DnV, GL, NK and IACS formulas all predict vertical accelerations that are lower than those predicted by the SCORES* program. However, the same agencies predict lateral accelerations that are greater than the SCORES* program. The reason for the low agency predictions relative to the SCORES* vertical accelerations may be

TABLE III. 6. COMPARISON OF ACCELERATION PREDICTIONS OF
FROM FULL-SCALE MEASUREMENTS AND FROM AGENCY FOR

Prediction Method		10^8 Accelerations in g's, Single Amplitude					
	S.S. Wolverine State	S. S. Boston					
	Vertical Accel.	Vertical Accel.			Lateral Accel		
	Bow	Bow	Md. Sp.	Stern	Bow	Md. Sp.	St
Exponential	>± 0.44	NA	NA	NA	NA	NA	
Weibull	± 1.63	± 0.83	± 0.40	± 0.49	± 0.35	± 0.35	±
Combined Rayleigh-Normal	± 0.97	NP	NP	NP	NP	NP	
Agency Formula							
ABS, LR, USCG	± 0.71	± 0.68	± 0.19	± 0.67	± 0.09	± 0.06	±
BV	± 1.00	NP	NP	NP	NP	NP	
DnV	± 1.27	± 1.24	± 0.48	± 1.01	± 0.81	± 0.50	±
GL, NK, IACS	± 1.26	± 1.22	± 0.48	± 1.00	± 0.74	± 0.57	±

NA - This procedure yielded inconsistent results for the S. S. Wolverine State an not used to make predictions for the other ships.

NP - Insufficient data were available; no predictions can be made.

* - See text. Conversion to extreme value yields ± 0.43 g's, or ± 0.63 g's, as acceleration variations.

† - ABS, LR, USCG formulas are not valid for this ship, as Lpp > 183 mete

TABLE III.7. CHARACTERISTICS OF THE FOUR LNG SHIPS ANALYZED BY THE ABS VERSION OF SCORES*

Parameter	Ship #1	Ship #2	Ship #3	Ship #4
Tank configuration	Membrane	Spherical	Membrane	Membrane
Capacity (m^3)	125,000	126,000	125,000	125,000
Length/draft ratio	24.9	24.2	24.9	25.2
Length/breadth ratio	6.7	6.1	6.5	6.7
Breadth/draft ratio	3.7	4.0	3.9	3.8
Service speed V (kn)	15.	15., 19.	16.	20.
Block coefficient C_B	0.763	0.720	0.738	0.765
Metacentric height GM(m) (full-load condition)	2.97	1.74	3.28	3.8

TABLE III.8. LONG-TERM ACCELERATION PREDICTIONS FOR THREE LNG SHIPS BY THE PROGRAM SCORES*

Speed (kn)	Loading Condition	X/L_{pp} [1]	Y/B [2]	Z/D [3]	A_z (g)	A_y (g)
	Ship 1					
15.0	Fl. Ld. Arvl.	0.20	0.31	0.25	--	0.24
15.0	Fl. Ld. Arvl.	0.20	0.21	2.64	0.53	--
15.0	Balst. Depr.	0.20	0.41	2.37	0.57	--
15.0	Cargo Balst.	0.20	0.21	2.64	0.56	--
	Ship 2					
15.0	Fl. Ld. Depr.	0.18	0.0	1.81	0.64	0.24
15.0	Balst. Depr.	0.18	0.0	1.81	0.65	0.24
19.25	Fl. Ld. Depr.	0.18	0.0	1.81	0.72	0.20
19.25	Balst. Depr.	0.18	0.0	1.81	0.73	0.24
	Ship 3					
16.0	Fl. Ld. Depr.	0.23	0.30	2.34	--	0.25
16.0	Fl. Ld. Depr.	0.23	0.39	1.99	0.61	--

[1] Relative distance aft of forward perpendicular.
[2] Distance from center line relative to the breadth.
[3] Vertical distance from base line relative to the draft.

TABLE III.9. COMPARISON OF EXTREME ACCELERATION PREDICTIONS BY AGENCY FORMULAS WITH PREDICTIONS FROM PROGRAM SCORES*

Agency Formulas	Vertical Acceleration			Lateral Acceleration		
	Ship 1	Ship 2	Ship 3	Ship 1	Ship 2	Ship 3
ABS, LR, USCG[1]	±0.93	±0.97	±0.85	±0.17	±0.19	±0.22
BV	±0.28	±0.30	±0.27	±0.13	±0.14	±0.17
DnV	±0.54	±0.64	±0.50	±0.38	±0.53	±0.46
GL. NK. IACS	±0.53	±0.69	±0.36	±0.38	±0.52	±0.47
Largest acceleration from each ship predicted by SCORES*	±0.57	±0.73	±0.61	±0.24	±0.24	±0.25

[1]Not applicable since all ships are longer than 183 m, but the values are given for completeness.

that the z-coordinate of the evaluated point is relatively far from the water line (the formulas of DnV, GL, NK, and IACS are to be evaluated at the tank center of gravity, which usually is not far from the water line). In addition, the considered point for Ships 1 and 3 is off the line of symmetry of the ship ($y \neq 0$). Since the agency formulas for a_y and a_z are independent of y, one would expect discrepancies between the agency and SCORES* predictions. Long-term accelerations obtained from the BV formulas are, for all cases considered, low relative to predictions by SCORES*. As none of the comments cited above for DnV, GL, NK, and IACS are appropriate for the BV formulation, the cause of the low BV predictions is not known.

For ship number 4, extreme lateral and vertical accelerations were calculated by the program SCORES* at ten discrete points on the forward and aft ends of Tanks 2 and 3 for two sea conditions. The orientation of the various points is shown schematically in Figure III. 11. Since all agency formulas except BV are independent of y, all comparisons are limited to points 1 and 6 as shown. Table III. 10 presents the SCORES* predictions of accelerations for points 1 and 6 along with the locations and the sea condition. Sea Condition 1 is essentially a head sea condition, while Sea Condition 2 represents the ship sailing in beam seas. Finally, Table III. 11 presents the agency prediction along with the corresponding values obtained from the SCORES* program (Points 1 and 6 are averaged in the table to reduce the volume of data. Since the x-coordinate of the leading and trailing

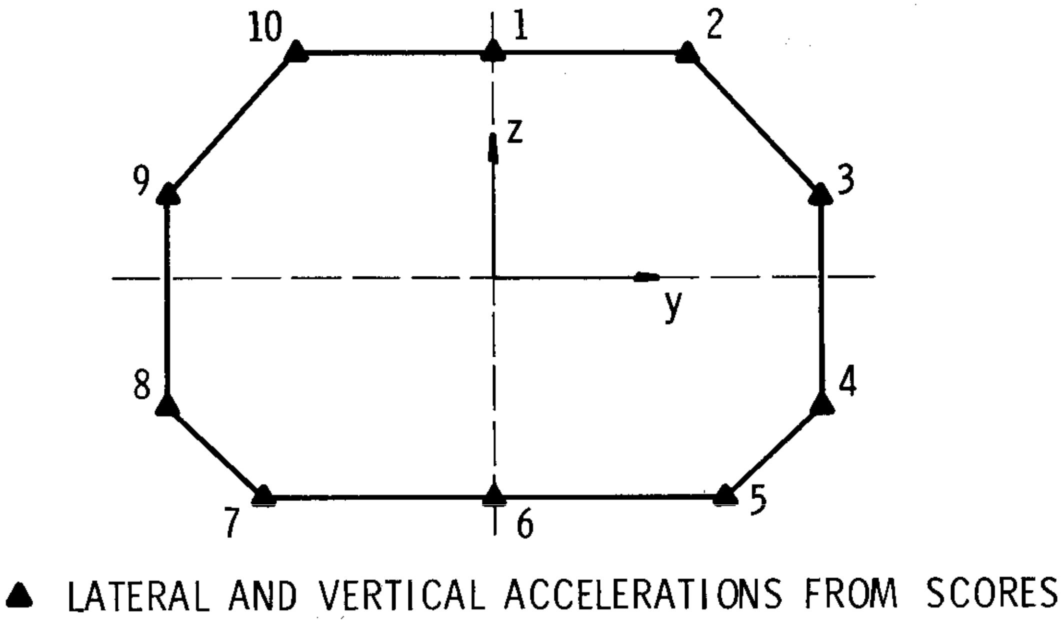


FIGURE III. 11. LOCATIONS OF POINTS WHERE ACCELERATIONS WERE CALCULATED BY SCORES* ON LNG SHIP 4

edge and the z-coordinate of the center of gravity of the respective tanks were used in the agency formulas, the averaging of points 1 and 6 is identical to a linear interpolation to the tank c.g.). Ship 4 is also longer than 183 meters, so the formulas of ABS, LR, and USCG are not valid. DnV, GL, NK, and IACS consistently predict vertical accelerations that are slightly greater than the SCORES* long-term predictions for head seas. Lateral accelerations predicted by these same agencies are approximately five times as great as the SCORES* extrapolated values for beam seas. BV's formulas, while conservative, yield values that are much closer to the SCORES* long-term calculations.

TABLE III. 10. SUMMARY OF EXTREME ACCELERATIONS FOR SHIP #4 OBTAINED FROM PROGRAM SCORES

Vertical Accelerations		Sea Condition 1		Sea Condition 2		$\frac{X}{L}$
Location	Tank	Point 1	Point 6	Point 1	Point 6	
Forward edge	2	$\pm$0. 54	$\pm$0. 54	$\pm$0. 05	$\pm$0. 05	0. 28
Forward edge	3	$\pm$0. 37	$\pm$0. 37	$\pm$0. 06	$\pm$0. 06	0. 16
Trailing edge	2	--	--	$\pm$0. 06	$\pm$0. 06	0. 17
Trailing edge	3	--	--	$\pm$0. 07	$\pm$0. 07	0. 03
Lateral Accelerations						
Forward edge	2	0.	0.	$\pm$0. 02	$\pm$0. 19	0. 28
Forward edge	3	0.	0.	$\pm$0. 05	$\pm$0. 16	0. 16
Trailing edge	2	--	--	$\pm$0. 05	$\pm$0. 16	0. 17
Trailing edge	3	--	--	$\pm$0. 08	$\pm$0. 13	0. 03
Z/D		1. 62	-0. 79	1. 62	-0. 79	
Sea Condition						
Roll		0°		30. 24°		
Pitch		6. 22°		0. 91°		
Yaw		0°		3. 26°		
Heading		180° (head seas)		60° (quartering seas)		

TABLE III. 11. COMPARISONS OF EXTREME ACCELERATIONS FOR SHIP 4 AS GIVEN BY THE AGENCIES AND BY PROGRAM SCORES

Agency Formula	Vertical Accelerations				Lateral Accelerations			
	Tank 2		Tank 3		Tank 2		Tank 3	
	Fore	Aft	Fore	Aft	Fore	Aft	Fore	Aft
ABS,LR,USCG*	±0.89	±0.61	±0.59	±0.29	±0.10	±0.10	±0.09	±0.09
BV	±0.28	±0.20	±0.20	±0.30	±0.14	±0.14	±0.14	±0.14
DnV	±0.58	±0.46	±0.45	±0.37	±0.50	±0.45	±0.45	±0.42
GL,NK,IACS	±0.62	±0.48	±0.48	±0.38	±0.50	±0.48	±0.48	±0.47
Sea Condition 1	0.54	--	0.37	--	0.0	--	0.0	--
Sea Condition 2	0.05	0.06	0.06	0.07	0.11	0.11	0.11	0.11

*Not applicable since this ship is longer than 183 meters; the values are given for completeness.

III. 4 Summary of Comparisons of Agency Formulas with Predictions from Full-Scale Measurements and from Calculations by Program SCORES*

The comparisons presented in this chapter have shown the agency formulas to be generally conservative relative to predictions from full-scale measurements. However, the experimental data were for non-LNG ships which, although similar geometrically in many respects to LNG ships, generally had lower length-to-draft and breadth-to-draft ratios. Note also that the comparisons with experimental data were made at the forward perpendicular, not at the tank center of gravity for which the agencies' equations were formulated. Comparisons with accelerations predicted by the SCORES* program gave mixed results, with lateral accelerations predicted by the formulas being generally conservative relative to SCORES* and vertical accelerations predicted by the formulas being unconservative relative to SCORES*. It is speculated that the relatively high SCORES* predictions for Ships 1, 2 and 3 are due to the point where the accelerations were calculated (high z-value). The SCORES* acceleration predictions for Ship 4 (low z-values near tank c.g.) are lower than the corresponding agency values, justifying this speculation.

IV. WAVE-INDUCED LOADS[3]

IV.1 Introduction

All of the classification societies require that wave-induced loads be determined before classifying an LNG ship, and each society has its own methods of establishing these loads. In most cases these wave-induced loads are calculated by hydrodynamic computer programs, such as SCORES*, but provisions in the rules are also made for determining these loads through empirical formulas. Tank accelerations, determined by both methods, were compared in Chapter III. This chapter will be devoted to evaluating the procedures, embodied in the computer programs, for calculating the wave-induced loads. Similarities and differences in the computing routines used by the various Classification Societies and other Regulatory Agencies, will be examined to the extent permitted by available data and information.

Table IV.1 summarizes the procedures of eight Classification Societies for predicting long-term wave-induced loads. The evaluation of these procedures is discussed in Appendix B, along with general comparisons and evaluations. More specific evaluations are covered in this chapter. They include numerical examples designed to show the effects of variations in input parameters such as wave data, ship geometry, etc. on the short-term and long-term predictions of ship loads.

Wave Data

The major sources of wave data used by the eight principle Classification Societies, i.e., ABS, BV, DnV, GL, LR, NK, RINa and the Russian Registry of Shipping, are observed wave data. These include Walden's data from the North Atlantic, Hogben and Lumb [7], Atlas of the world oceans, Roll's data from the North Atlantic, and Yamanouchi data from the Pacific. The data is arranged in tabular distribution of heights and periods in most cases, with the exception of DnV who uses the Weibull distribution. Cumulative distribution representing the actual data, are also used. ABS uses spectra obtained from measured data at the ocean weather stations represented by 80 spectra divided into 10 groups covering a wide range of wave heights. In all cases where observed data is used, the mathematical spectral formulation is some form of the 2 parameter spectra. In some cases, the observed data is directly substituted, in others it is modified to represent $H_{1/3}$ and a characteristic period. In each of the above

[3] The original draft of this chapter was prepared by D. Hoffman, Webb Institute of Naval Architecture.

TABLE IV.1. PROCEDURES FOR PREDICTING LONG-TERM WAVE LOADS
(Ref. 6)

	ABS - Webb	R.I.Na.	Lloyds
Sources of Information on Procedure	Lewis-Montreal - 1967	R.I.Na. Report No. 120, by A. Gennaro, Dec. 1971	R. Goodman, Southhampton Symposium, 1970 Murray, Trans. N.E.C.I. 1965
Wave Data			
Source:	Pierson-Moskowitz for North Atlantic	Walden data for North Atlantic (N.V.)	Hogben & Lumb; other data when available.
Format:	Observed spectra plotted in groups of constant sign. ht.	Tabular distribution of periods and heights	Tabular distribution of periods and heights
Wave Spectrum formulation	Use random sampling of actual spectra for North Atlantic	N.V.	N.V.
Short-crestedness	Spreading function $\frac{2}{\pi}\cos^2\mu$	Spreading function $\frac{2}{\pi}\cos^2\mu$	Spreading function $\frac{2}{\pi}\cos^2\mu$ or modification
RAO's			
From:	Model tests or calculations. Calculations preferred.	Strip theory calculations or model tests (generally calculations)	Prefer model tests (systematic or random with regression analysis) Strip theory calculations
All headings?	Yes	(Yes (7+)	Yes, up to 14
Assumed distr. of headings	Equal probability	Equal probability	Variable probability
No. of wave lengths	34	13+	Up to 51
Short-term Predict.			
Results in terms of rms response for:		Standard (Subject of variation):	
Ship speeds	1	1+	4
Ship headings	7	7	14
No. of spectra per wave height	10	8	
Sig. wave heights	5	16	
No. of periods per wave height	10	5	1
Long-term Predict.	Integrate probabilities over ship heading at constant wave height; integrate over wave ht.	Integrate probabilities over wave periods, wave hts., ship headings.	Integrate probabilities over wave periods, wave heights, & ship headings; and fair result graphically.
Probability level (or number of cycles) for design.	N - 10^8 for comparative purposes only. Ultimate value depends on factor of safety, etc.	Generally N = 10^2 and 10^8	Depends upon response under study.

TABLE IV.1. PROCEDURES FOR PREDICTING LONG-TERM WAVE LOADS (Contd.)
(Ref. 6)

	Norske Veritas	G.L.	N.K.
Sources of Information on Procedure	Abrahamson, 'Recent Developments in the Practical Philosophy of Ship Structural Design', SNAME, Spring, 1967; or N.V. Publ. No. 77, 1971	H. Söding, Schiff und Hafen, Oct. 1971, H.-G. Schultz, JSTG, 1969.	Shipbuilding Research Association of Japan, Report No. 69, Sept. 1970.
Wave Data			
Source:	Roll's & Walden's data for North Atlantic	Roll & Walden for North Atlantic	Walden for North Atlantic
Format:	Data expressed in form of Weibull distributions.	Distrb. of sig. hts. & periods for N.A., presented graphically as cumulative distr.	Tabular distribution of periods and heights.
Wave Spectrum formulation	Modified Pierson-Moskowitz	I.S.S.C., with $H_V = H_{1/3}$	I.S.S.C.
Short-crestedness	Spreading function $\frac{2}{\pi}\cos^2\mu$	Spreading function $\frac{2}{\pi}\cos^2\mu$	Spreading function
RAO's			
From:	Strip theory calculations or model tests.	Strip theory calculations	Strip theory calculations
All headings?	Yes	Yes	Yes
Assumed distr. of headings	Normally, equal probab.	Equal probability	Equal probability
No. of wave lengths	17 (normally)	22 (normally)	20
Short-term Predict.			
Results in terms of rms response for:			
Ship speeds	2 (max. 3)	5	7
Ship headings	5 (max. 7)	7	7
No. of spectra per wave height	Any number.	22	8
Sig. wave heights	20 values of	15	1
No. of periods per wave height	$\overline{T}\sqrt{g/L}$	8	5
Long-term Predict.	Integrate probabilities over wave periods, wave hts.; fit to Weibull distr. & integrate over ship headings.	Integrate probabilities over wave periods, wave hts. and ship headings.	Integrate probabilities over wave periods, wave hts. and ship headings.
Probability level (or number of cycles) for design.	$N = 10^8$ for comparative purposes. Ult value depends on factor of safety, etc.	$N = 10^6$	$N = 10^6$ for comparative purposes only.

TABLE IV. 1. PROCEDURES FOR PREDICTING LONG-TERM WAVE LOADS (Contd.)
(Ref. 6)

	Bureau Veritas	Register of Shipping of USSR
Sources of Information on Procedure	J. M. Planeix, M. Huther, R. Dubois, 'Sollcitations externes et internes des navires a la mer,' ATMA, 1972.	A. J. Maximadji, Proceedings of the Research Institute of Sea Fleet. 1971, N 134 and 1972, N 140 V. V. Kozljakow, Sudostroyenlye 1966, N 8.
Wave Data Source:	Roll: Hogben & Lumb - N. A. Hogben & Lumb; Yamanouchi & Ogawa - Pacific	Roll's data for North Atlantic
Format:	Fr. Meteor Office data Tables of periods & heights. Tables of spectral ords.	Integral curve of the probability of exceeding of the wave states as a function of $H_{3\%}$.
Wave Spectrum formulation	$S(\lambda) = \frac{\pi}{g^2}\frac{H_{1/3}^2}{T^2}\lambda \exp\left(-\frac{4\pi}{g^2}\frac{\lambda^2}{T^2}\right)$ $H_{1/3}$ is sign. ht. T it means app. period λ is wave length	I. S. S. C., but based on observed $H_{3\%}$ ($H_{3\%} = 1.33\ H_{1/3}$)
Short-crestedness	Varies with wave frequency	Spreading function $\frac{2}{\pi}\cos^2\mu$
RAO's From:	Generally strip theory calculations	Strip theory calculations
All headings?	Yes (6)	Yes
Assumed distr. of headings	From wave data	Equal probability
No. of wave lengths	12 - 20	
Short-term Predict. Results in terms of rms response for:		
Ship speeds	1 - 6	
Ship headings	6	5
No. of spectra per wave height	Usually 1	8
Sig. wave heights	Usually 6	8
No. of periods per wave height		6
Long-term Predict.	Determine short-term probability for each angular spectral comp., multiply by probs. of spectrum & headings. Integrate (36 terms)	Integrate probabilities over wave periods, wave heights, ship headings.
Probability level (or number of cycles) for design.	About $N = 10^7$	$N = 10^8$

cases, a spreading function is used, usually of the type $2/\pi \cos^2\mu$. Based on the above, it should be concluded that all Classification Societies use practically the same wave data with minor changes in formatting and sources. ABS, however, is gradually switching to actual spectra for most ocean routes.

Two-Dimensional Hydrodynamic Coefficient

As indicated in Appendix B, the method used for calculating the hydrodynamic coefficients is universal and the particular approach selected is usually a matter of choice or preference. Most agencies use the "Lewis" [9] form approximation whereby the two-dimensional section is defined in terms of the sectional area coefficient and the beam/draft ratio. The advantage of this simplified approach is primarily economical, as the computer time required is extremely short; yet for most ship sections the "Lewis" form approach yields satisfactory results.

Several of the agencies have an alternative approach using Frank's close-fit method (DnV) or multi-coefficient mapping routine (ABS). Techniques to handle special ship configurations are available for catamarans and the USCG is capable of calculating the motions of a large variety of buoys representing ship shape, axisymmetrical bodies and catamarans. Ideally, the basic two-dimensional hydrodynamic coefficient program should include all three approaches ("Lewis", multi-mapping, and Frank's close-fit) and based on a preliminary analysis of the section shape, the most appropriate routine should be selected automatically.

Several of the Classification Societies have the capabilities of evaluating the pressure distribution on the two-dimensional section as an input for calculating the three-dimensional pressure distribution on the hull. For that purpose, a "Lewis" form technique is inadequate.

Equations of Motions

The general format of the equations of motions is identical for all Classification Societies. The only differences exist in the exact definition of the coefficients. Some of the coefficients which are considered to be zero by some agencies, i.e., no cross-coupling effects between certain motions, may be estimated by another. There may be today as many as 6 slightly different strip theories each of which could be used by one of the agencies. The major programs are SCORES [8] (ABS, LR), Salvasen, Faltinsen and Tuck [10] (DnV, U.S. Navy), Sodin [11] (GL), Fokuda [12] (NK), Delft [13], MIT [14], and University of California, Berkeley [15].

Comparing some of the specific coefficients on the lefthand side of the equation and the excitation forces or moments on the right, reveals rather large differences between the various methods (see Appendix B, pp 8-10). It is difficult to evaluate which is the most appropriate technique as model tests to determine such coefficients are scarce and the few which were performed did not include the evaluation of the more controversial coefficients. However, the values of concern are not the coefficients of the equation of motion but the resulting transfer functions of motions and consequently accelerations, forces, moments, etc. With regard to the external moments, no consideration is given by any of the theories to moments exerted on the ship as a result of liquid sloshing in slack tanks. For a 125,000 m^3 LNG ship, resonant sloshing in slack tanks can produce moments of $\sim 5 \times 10^6$ kg-meters per tank. If these slosh-induced moments are appreciable relative to the other external moments and if their probability of occurrence is high, then the possibility of coupling between liquid motions and ship response exists. The potential for coupling will increase as resonant sloshing occurs in large tanks at worst-case fill levels. Since the magnitude and probability of the slosh-induced moments can be estimated from previous slosh studies, these moments should be included in the forcing function if they are significant and the effects of coupling evaluated.

Response Amplitude Operators (RAO)

Due to the rather involved nature of the calculations and problems of proprietary progress, no easy comparison between the resulting transfer function is available for public use. It is unusual that a ship classed by one Classification Society will be also classed by another. Several internal studies were performed by DnV, ABS and others to establish trends for their own use, but results are not generally available.

The comparison between calculations and model test results are more easily available and form the only criteria for evaluating a method on its own merit. It should be remembered, however, that comparison of measurements in different model wave tanks does not necessarily yield identical results due to model size, measuring techniques, analysis approach, etc. Furthermore, it has been shown that while some calculation methods are good for certain types of ships, others may be better for other types. Hence, there is no general conclusion as to the merits of one procedure relative to another.

Most agencies prefer the calculation method over model tests, since the former is cheaper, faster and in most cases just as reliable as the latter. Ideally, a combination of both approaches will yield the best results. The calculations are usually performed for a minimum of five headings and often a maximum of fourteen. The number of frequencies usually considered for better definition of the RAO's vary from 13 to 22 with capabilities of up to 51.

As discussed previously, for design purposes, no valid comparison between procedures for calculating RAO's can be made at this intermediate stage of the calculations. One can easily show that for two methods, both having identical RAO's, the long-term prediction at 10^{-8} probability represents a 100% deviation due to wave data and statistical methods [16]. Thus, in order to evaluate the differences more systematically, the short and long-term predictions should be first compared.

Short and Long-Term Predictions

The principle of linear superposition is the basis for all short-term predictions and is applied by all the procedures reviewed in Table IV. 1. The variations between one program and another is primarily in the number of speeds and headings used and the assumptions with regard to the probability distribution of each heading over the ship's lifetime operation. These differences usually represent restrictions imposed by the particular computer used due to core size, etc.

The main differences between the various procedures are due to the wave data formatting and the method of integration of the probabilities over wave period, wave height and ship headings. Since most Classification Societies use tabulated forms for the probabilities of wave height, period and headings, the same data is used to integrate the resulting response spectra to obtain cumulative distributions from which the long-term design values can be read at any level of probability. In some cases, a specific distribution such as Weibull (DnV) or Exponential is fitted to the data after the integration over period and height and before the integration over ship heading is performed. Since each wave height and period combination yield one response spectrum, the philosophy of combining all these responses may vary somewhat. If the integration is first performed for groups of constant wave height over the entire period range, the scatter in the rms value due to any possible period is first established.

The next step involves integration over the height using the probability of occurrence of each constant height group as a weighing factor. Finally, the headings can be considered as equal probability of encounter, or other more specific assumptions for certain routes based on the actual relationship between height and heading, as given from the statistical wave data, can be used. The above procedure does not involve any statistical models and the cumulative curve is simply extrapolated to larger periods of time. An improvement can therefore be achieved if a specific distribution such as Weibull or another is used to fit the data and a more reliable extrapolation can be obtained. In either case, the extrapolation is performed for the entire range of heights or periods and individual weather groups are not considered.

When measured spectra are used for the response calculations, the data are subdivided into groups of equal wave height and the standard deviation within the group is a measure of the scatter due to a variety of periods and basic spectral shapes. For each such weather group, a long-term prediction is performed assuming a normal distribution of the rms values within the group. Subsequently, an integration over the wave height yields the final long-term curves representing the total data. The integration over the heading is performed following the integration over period after the rms and standard deviation were established for each heading.

A limited comparison of long-term bending prediction obtained by some of the available procedures, was summarized in the 1973 ISSC Conference [6]. It is apparent that though the order of magnitude is identical in all cases, differences of considerable magnitude may be exhibited due to the various assumptions made in each case. In order to evaluate the sensitivity of the long-term predictions at the 10^{-8} probability level, due to various assumptions such as probability of wave height, probability of heading, etc., a short study was conducted [17] and a summary of the results is given in the following section.

IV.2 Sensitivity of Design Values

The long-term predicted response at a probability level of 10^{-8} is often used by various Classification Societies as a design value at least for comparative purposes. In order to evaluate the sensitivity of such values to small variations in the ship design parameters or in the assumptions made with regard to the short or long-term prediction, a ship configuration was available and the calculations were repeated for several responses, varying some of the input data as discussed in the following sections. The basic ship available for analysis in all cases was a 600,000 dwt VLCC at full load, travelling at a speed of approximately 20 knots.

Effect of Routing or Wave Height Distribution

The assumptions made with regard to specific routing such as the North Atlantic or North Pacific usually boil down to different wave height distributions. Several different routes covered by ocean zones given by Hogben and Lumb [7] were studied and summary of the expected wave height distribution in a cumulative form is given in Table IV.2. A key to the eight routes is also given. Results obtained from the vertical acceleration at the forward perpendicular and vertical acceleration at the deck edge are given in Table IV.3 for the eight routes studies. Differences on the order of less than 10% between one of the most severe routes and some of the mildest routes were obtained. The results indicate the stability of the calculated responses and their lack of sensitivity to small changes in the route assumptions.

TABLE IV.2

CUMULATIVE WEATHER DISTRIBUTION - WORLD ROUTES

$H_{1/3}$(m)	Route							
	1	2	3	4	5	6	7	8
	% of Waves Exceeding the Stated Value							
.61	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
1.22	91.25	86.27	88.79	77.62	82.26	82.72	74.56	79.6
2.44	67.50	68.17	71.14	35.12	56.62	57.60	29.16	54.3
3.05	36.80	35.41	36.01	13.07	19.03	19.96	9.32	20.5
4.57	16.45	17.51	17.18	2.83	6.46	6.82	3.03	8.8
6.10	9.55	4.88	4.55	1.41	1.07	1.13	.51	2.1
7.62	4.60	2.70	2.56	.80	.42	.48	.18	1.1
9.15	2.00	1.07	.97	.43	.12	.15	.09	.4
10.67	.30	.35	.31	.17	.03	.04	.04	.1
15.24	.05	.02	.01	.02	.01	.01	.01	.0

1. *Most severe North Atlantic*
2. North Atlantic (Northern Europe)
3. North Atlantic (Southern Europe)
4. Europe -- Persian Gulf to Northern Europe
5. North Pacific
6. Europe -- Persian Gulf to Bantry Bay
7. Persian Gulf -- USA
8. Europe -- USA West Coast

TABLE IV.3

EFFECTS OF DIFFERENT ROUTINGS ON VERTICAL ACCELERATION
(g, 0-p)

Response	Route							
	1	2	3	4	5	6	7	8
Accel. at F.P.(C.L.)	0.40	0.38	0.36	0.36	0.38	0.36	0.36	0.36
Stbd Vert. Accel	0.52	0.49	0.48	0.46	0.49	0.46	0.46	0.47

Effect of Different Probabilities of Headings

Four assumed probability distributions of headings were evaluated. The first was that of equal probabilities while the latter three emphasized various degrees of head and following seas with minimum time spent at beam seas. The distributions are given in Table IV. 4 and the resulting accelerations at the bow and the deck edge are given in Table IV. 5. The maximum variations in the acceleration levels were of the order of 10-12%, indicating again the noncritical nature of the assumption made.

Simulated Swell Effects

The effect of pure swells or of combinations of a swell and a storm approaching the ship from two different directions is typical of certain ocean zones and in particular the lower east coast of Africa. Since no wave data is available for such conditions and the mathematical formulations fail to cover these conditions analytically, an estimate of the possible effects was studied[17] by shifting the whole spectral family bodily to lower frequencies, thus simulating longer average periods.

The spectra were shifted a total of 0.1ω in increments of 0.02. In all cases, the shift was a bodily shift of the entire spectrum to lower frequencies. Although the resulting spectra may not truly represent a sea in which swells are superimposed upon the resulting sea state, it should give some indication of the effect of the higher energy input at lower frequencies associated with swells. The effects on the accelerations at the bow and the deck at side is shown in Table IV. 6. Variations were again of the same order as in the other cases, i.e., roughly 10%. It should be emphasized that in this case, as well as in the previously discussed cases, the results are characteristics of the specific hull and its condition of operation. It will be shown in the following section that large variations in acceleration may occur due to change in loading conditions and forward speeds. However, the stability of the results seem to be a more general feature and the probability of the long-term prediction doubling in value is very small.

Effects of Design Parameters

The effect of three design parameters, the metacentric height (GM), the transverse radius of gyration (K_{yy}), and the viscous damping parameter were also investigated[17] to determine effect of small variations in the design values. While the three previous cases dealt with effects of assumptions made when calculating the short and long-term predictions, the above three are used in calculating the transfer function. Furthermore, these three parameters affect primarily the roll response and related lateral responses, and have a minimal effect on other responses.

TABLE IV. 4

DIFFERENT COMBINATIONS OF HEADING PROBABILITY

Heading	Run			
	1	2	3	4
0	14.29	40	30	22
30	14.29	4	8	22
60	14.29	4	8	4
90	14.29	4	8	4
120	14.29	4	8	4
150	14.29	4	8	22
180	14.29	40	30	22

TABLE IV. 5

EFFECT OF VARYING PROBABILITIES OF HEADING
ON VERTICAL ACCELERATIONS (g, 0-p)

Heading	Run			
	1	2	3	4
Accel. at F. P. (C. L.)	0.40	0.45	0.43	0.44
Accel. at C. G. (Max. P/S)	0.29	0.28	0.29	0.27

The metacentric height (GM) can be controlled to a certain extent during the design stages and operation of the ship. It is therefore important to know how sensitive will the acceleration response be to small changes. Table IV. 7 indicates an approximately linear relationship between the GM and the roll angle. An increase of 1.25 m (26%) in GM causes a 5.24° (32%) change in the maximum roll angle.

The transverse radius of gyration (K_{yy}) represents the transverse weight distribution across the ship. Though it is not subjected to large variations, some changes are possible with loading condition. Table IV. 7 illustrates the effect of such changes on the roll angle. In determining K_{yy} the liquid cargo or ballast is considered to act as a rigid mass. In actual service, however, the liquid does not move as a rigid mass with the tank motions. Normally, the liquid will move on a moment arm of a length equal to the distance between the liquid center of gravity (c.g.) and the ship center of roll. There is little movement of the liquid about its own center of gravity. If the mass moment of inertia of the liquid about its c.g. is ignored (i.e., the liquid is not assumed a rigid mass) in the K_{yy} calculation, the resulting value of K_{yy} will be changed. To estimate the effect of the rigid mass assumption on K_{yy}, an example calculation was performed for an existing 125,000 m^3 membrane tank LNG ship. Ignoring the mass moment of inertia of the liquid about its c.g., for the case of all tanks full of LNG, resulted in an ~25% reduction in the calculated K_{yy}. For the ballast condition with the liquid mass assumed non-rigid and located at the ship sides yields an ~25% increase in K_{yy}. In actual practice the liquid cargo or ballast will react somewhere between the rigid and non-rigid mass assumptions. The effects of the change in K_{yy}, as reflected by this condition, on either short or long-term predictions, remains to be established. However, it is generally accepted that small changes (<25%) in the K_{yy} will not have a significant affect on long-term predictions of vertical acceleration.

Finally, the influence of an empirical roll damping factor to account for viscous damping and roll damping characteristics, not accounted for by potential flow theories, is considered. Changes in the roll damping characteristics can be made through bilge keels, anti-rolling from passive tanks, etc. Damping values which include these effects can be approximated by analyzing roll decay curves generated in a model tank. In many cases, however, such information is not available and an approximation must be made. In such cases, the roll damping is based on empirical data and is expressed as some fraction of the critical roll damping at the roll natural frequency. The values chosen are normally about 0.08 to 0.10.

Table IV. 8 illustrates the effect of varying the roll damping factor between 0.025 and 0.1. The maximum roll angle and the vertical acceleration at the deck side are given. While the roll angle is quite sensitive to

TABLE IV. 6
EFFECT OF SHIFTING SPECTRA ON
VERTICAL ACCELERATIONS
(g, 0-p)

	Accel. F. P. (C. L.)	Accel. L. C. G. Max. P&S
1 $\Delta\omega = 0$ ($\omega =$ .2, .25, .30. . . .)	0.40	0.29
2 $\Delta\omega = -.02$ ($\omega =$ .18, .23, .28. . .)	0.43	0.31
3 $\Delta\omega = -.04$ ($\omega =$ .16, .21, .26. . .)	0.45	0.32
4 $\Delta\omega = -.06$ ($\omega =$ .14, .19, .24. . .)	0.46	0.32
5 $\Delta\omega = -.08$ ($\omega =$ .12, .17, .22. . .)	0.46	0.32
6 $\Delta\omega = -.10$ ($\omega =$ .10, .15, .20. . .)	0.45	0.31

TABLE IV. 7
EFFECT OF GM AND K_{yy} ON ROLL ANGLE

Metacentric Height (GM) meters	Roll Angle (deg.)
4.86	16.40
6.11	21.64
Transverse Gyradius (K_{yy}) meters	**Roll Angle (deg.)**
21.9	23.76
23.7	21.64
26.5	16.59

TABLE IV. 8
EFFECT OF DAMPING ON ROLL

Gyradius = 23.66 meters   GM = 6.42 meters

Roll Damping	Roll Angle	Vertical Acc. at C.G. (P/S)
.025	30.50	0.31
.050	19.31	0.30
.075	14.73	0.30
.100	12.13	0.30

these changes, the vertical acceleration is hardly affected. The effect on lateral acceleration at the deck would be more pronounced, of course, and probably would vary in about the same proportions as the roll angle.

IV. 3 Summary

In all the above cases, the effects of slight variation in six different parameters on response design values were illustrated. No attempt was made to cover all possible controlling parameters and the cases shown represent a sample only. It is, however, indicative of the nature of possible variations in the long-term acceleration and roll angles and gives some sort of a feel for the factors of safety involved in such calculations.

IV. 4 Typical Numerical Examples

Both vertical and lateral accelerations in most ships are effected by forward speed variations. The exact behavior cannot be generally predicted and must be calculated using one of the procedures that have already been discussed. This was done for the LNG ship identified as Ship #2 in Table III. 7. Calculations were performed using the SCORES* program for two loading conditions and two forward speeds. The results, which represent the long-term accelerations corresponding approximately to a 10^{-8} probability level, are given in Figures IV. 1 and IV. 2. As can be seen in the figures, quite large variations in accelerations result from the changes in loading conditions and smaller variations result from changes in the forward speed.

Though, in general, the zero speed represents the smallest acceleration response and the top speed represents the highest response, it is not necessarily the case under all conditions. Due to the effects of forward speed on the encounter frequency and hence on the resonance frequency, the magnitude of the change will vary. Figures IV. 3 and IV. 4 represent the variations in vertical and lateral acceleration with speed at a point whose coordinates from the center of gravity are given. The vessel in question is a 274 meter container ship. Figures IV. 5 and IV. 6 represent the same response for the same forward speed but at half loaded condition. Though little changes are noticeable for the vertical acceleration case (Figures IV. 3 and IV. 5), the lateral acceleration exhibits some different characteristics in the half load case. The maximum acceleration occurs at 20 knots rather than 30 and at wave heights higher than 7.62 meters. Furthermore, at 15.2 meters significant wave height, reduction in speed will lead to increase in acceleration and at 10 knots the level of acceleration is identical to that at 30 knots.

It is apparent that loading has a substantial effect on the behavior of the ship in waves. In reality, loading can sometimes represent a different

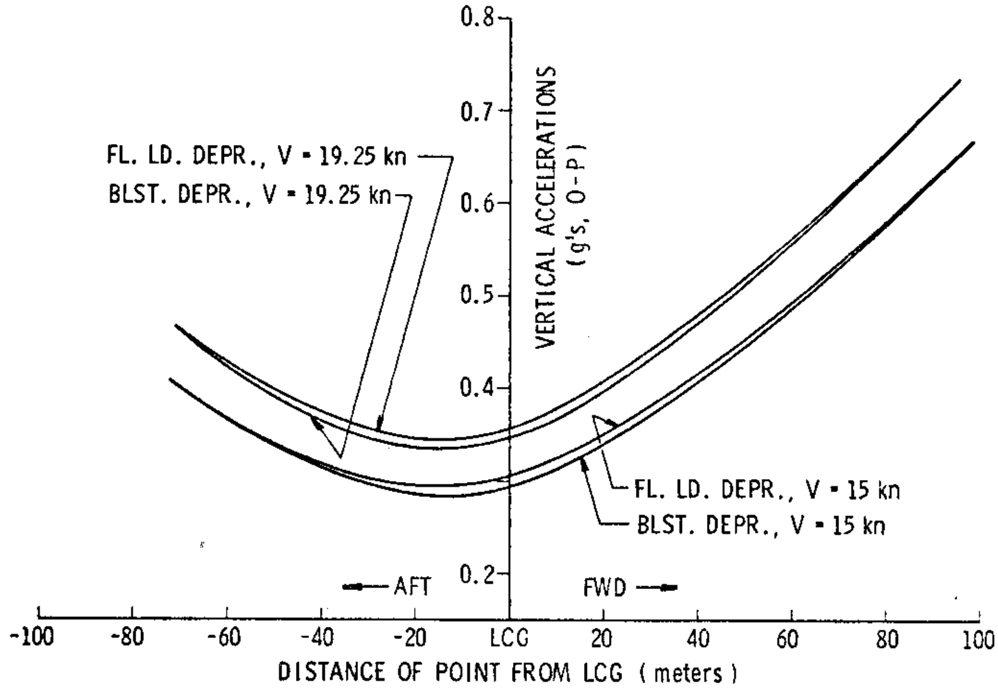


FIGURE IV.1. LONG-TERM VARIATION OF VERTICAL ACCELERATION WITH LOADING CONDITION AND SERVICE SPEED FOR LNG SHIP 2

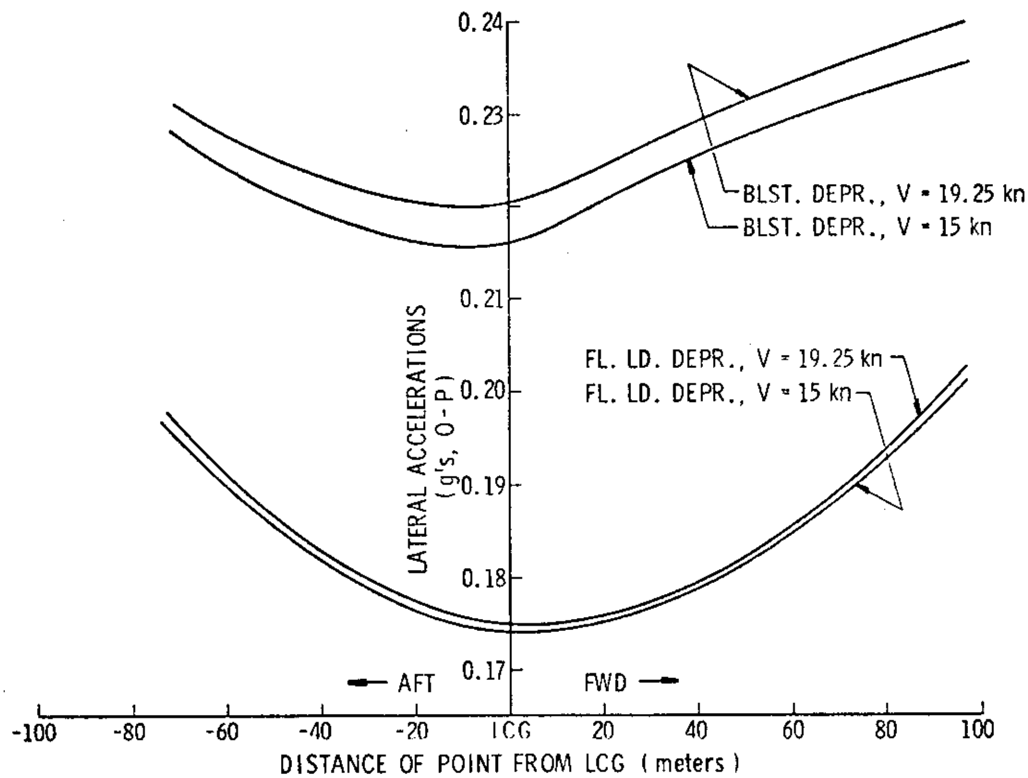


FIGURE IV.2. LONG-TERM VARIATION OF LATERAL ACCELERATION WITH LOADING CONDITION AND SERVICE SPEED FOR LNG SHIP 2

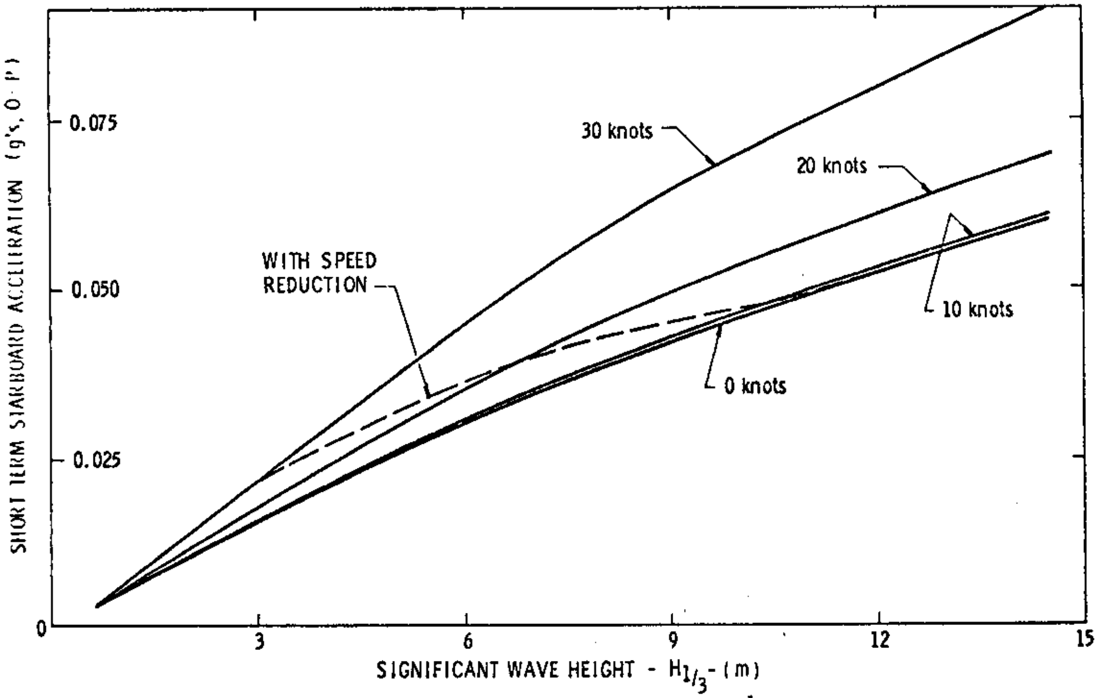


FIGURE IV. 3. SHORT TERM VERTICAL ACCELERATION AS A FUNCTION OF SIGNIFICANT WAVE HEIGHT FOR A FULLY LOADED CONTAINER SHIP

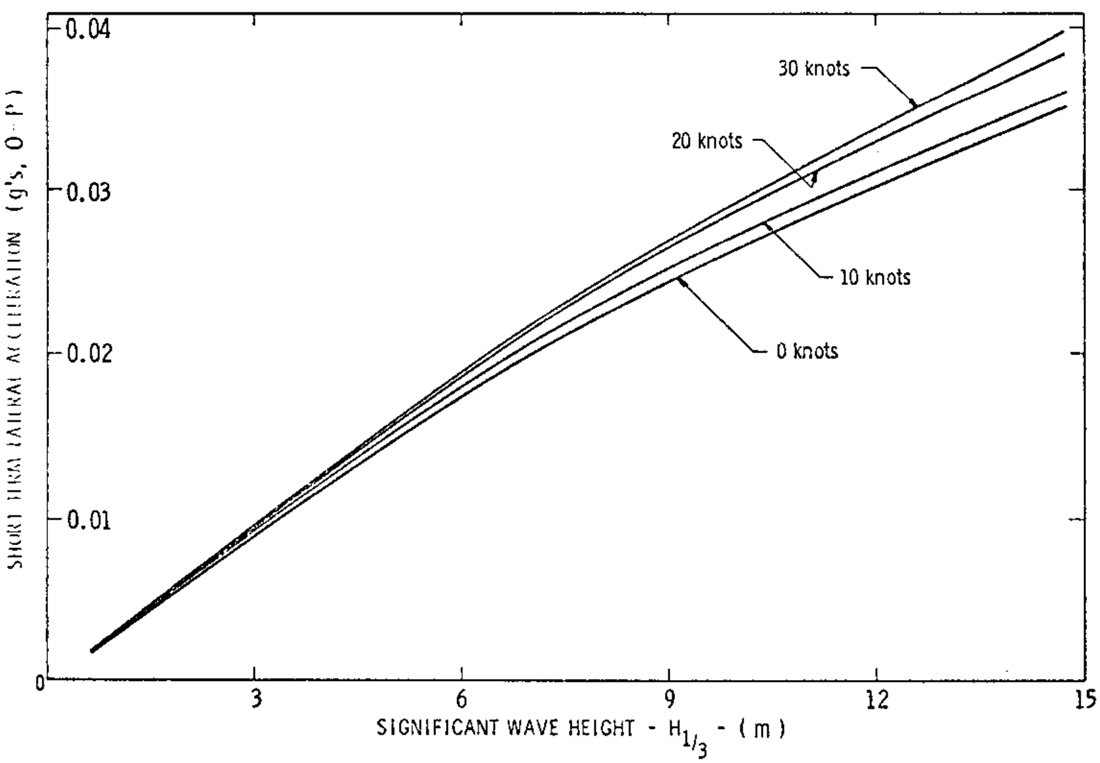


FIGURE IV. 4. SHORT TERM LATERAL ACCELERATION AS A FUNCTION OF SIGNIFICANT WAVE HEIGHT FOR A FULLY LOADED CONTAINER SHIP

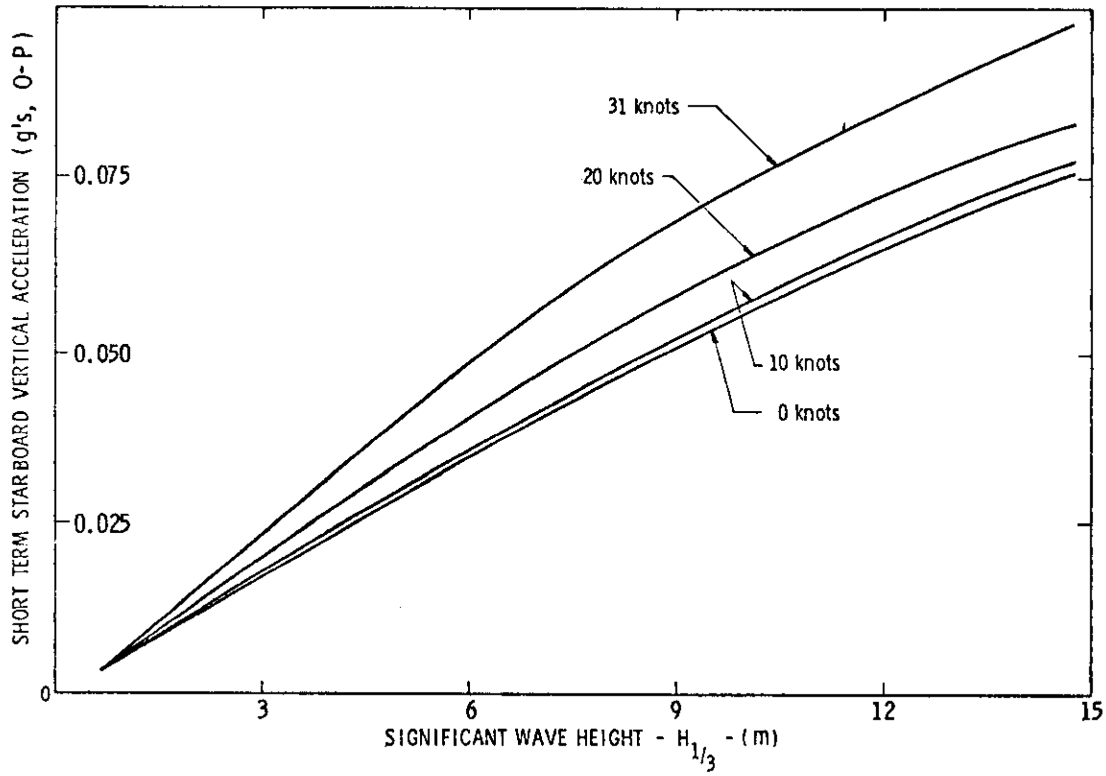


FIGURE IV.5. SHORT TERM VERTICAL ACCELERATION AS A FUNCTION OF SIGNIFICANT WAVE HEIGHT FOR A PARTLY LOADED CONTAINER SHIP

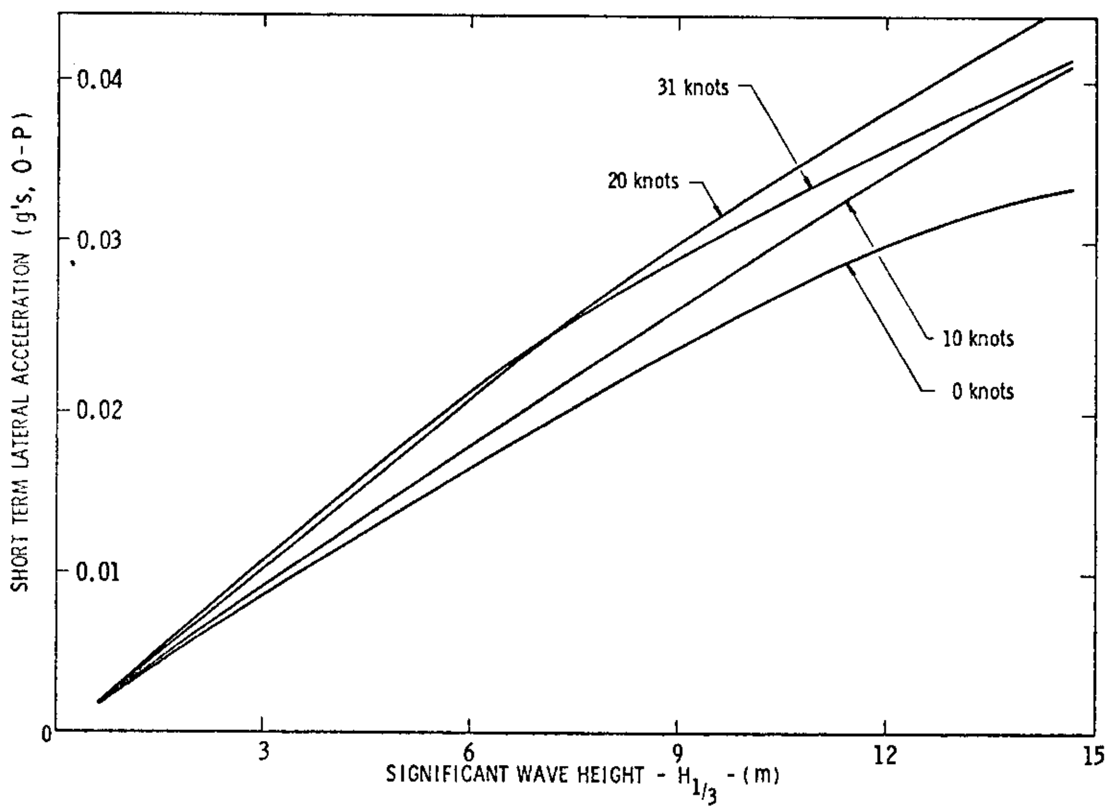


FIGURE IV.6. SHORT TERM LATERAL ACCELERATION AS A FUNCTION OF SIGNIFICANT WAVE HEIGHT FOR A PARTLY LOADED CONTAINER SHIP

ship altogether. In the case shown, the GM was doubled from the full to half load, and the roll frequency moved upward to a higher frequency.

The effect of forward speed adjustment due to heavy weather on the long-term prediction is usually taken into consideration by assuming a speed reduction as the significant wave height increases. Such reduction is often necessary due to added resistance considerations and sometimes due to voluntary reduction of speed to avoid damage through excessive motions. Figure IV. 3 (dashed curve) illustrates the assumed short-term vertical acceleration as accounted from speed reduction. The long-term calculations are based on the mean curves and its standard deviation and results in a substantial reduction in the maximum expected value as compared to the full speed case.

Finally, the effect of heading angles on the acceleration at the bow is illustrated in Figure IV. 7. The results shown are for a 152.4 meter container ship in the loaded condition and the short-term trends for 7 headings (180° = head seas) are illustrated. Also shown are the trends from forward speed conditions at a heading angle of 120°. The purpose of the latter is to enable a comparison in the magnitude of changes in the acceleration due to the effects of heading and speed.

IV. 5 Transient Loadings

The results presented in the preceeding sections are derived on the assumption of wave-induced oscillating forces. The effects of transient accelerations due to slamming, shipping of water, bow flare, etc. are not considered. Very little theoretical work has been so far performed on evaluation of such transient accelerations and the little available information is based on some limited full-scale measurements. The probability of occurrence of such loads is a function of the specific type of ship, its length, bow shape, structural rigidity, draft forward, etc. It is therefore difficult to apply the results obtained for one ship to another without a careful detailed analysis.

In general, the acceleration due to transient loading can assume values as high as the maximum expected from wave-induced loads over the lifetime of the ship. However, the occurrence of peak transient loads, simultaneously with the peak oscillatory wave loads is uncommon, and for the few cases where it might occur, results indicate that the transient load, i.e., that which occurs at the instant of slam, is superimposed on the wave at other than the maximum value. It is therefore possible that the transient accelerations are not additive, and the highest expected value obtained from the long-term statistics is adequate in covering the transient loads which might be encountered in the lifetime of the ship, particularly if speed reduction in heavy seas is not considered.

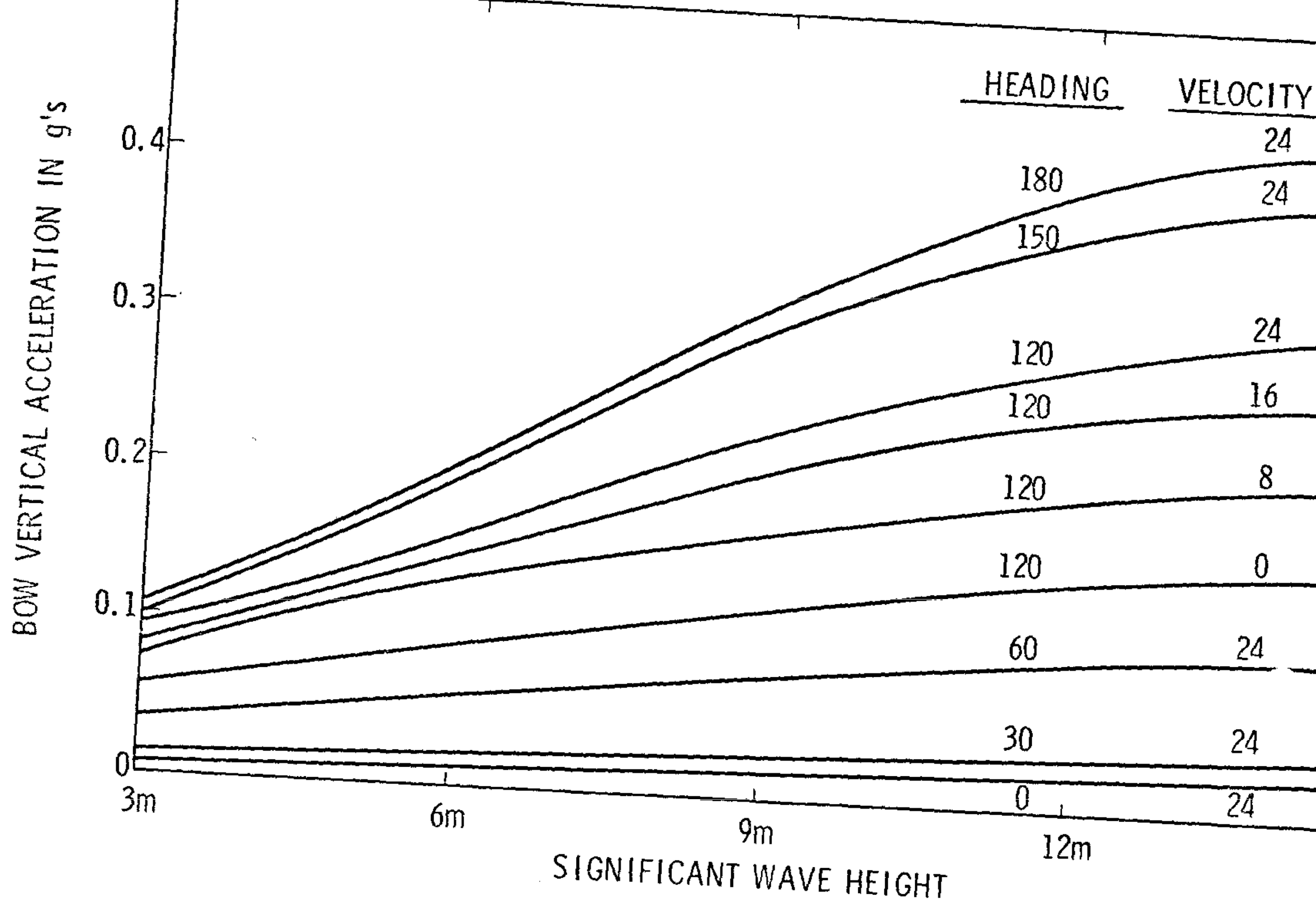


FIGURE IV.7. EFFECTS OF HEADINGS AND SPEEDS ON SHORT-TERM ACCELERATIONS FOR A 152 m CONTAINER SHIP

V. LOAD CRITERIA EVALUATION

V.1 Introduction

Each of the 17 tank loads is evaluated in this chapter. This evaluation included (1) comparing loads predicted by various agency formulas for a specific LNG ship, (2) comparing the agency predicted loads to available model or experimental data or to relevant theoretical methods, and (3) indicating where updated requirements are needed in each load category. Criteria for combining of the 17 loads for establishing final tank design requirements are also presented. In most cases insufficient data or analyses were available to provide a thorough evaluation. However, based on current knowledge, several improvements to present criteria are suggested.

Each of the evaluated tank loads are important to the design of an LNG ship cargo tank. However, some of the tank loads are more important than others, and this relative importance varies with the particular tank type as well as with ship operating restrictions. For example, vapor pressure loads are more important to the design of pressure vessel tanks than for membrane tanks. Also, sloshing loads are of no concern if partial filling is prohibited, but are extremely important when partial filling is allowed. Table V.1 shows a subjective rating of the different tank loads. The highest rated loads are those having the greatest importance to tank design, the poorest methods for prediction, and the requirement of additional research or experimentation before criteria can be developed and validated. It is noted that the tank loads receiving the highest rating have been given the greatest emphasis in this report.

V.2 Criteria Evaluations

V.2.1 Vapor Pressure

The agencies require that LNG cargo tanks be designed for an effective vapor pressure at least equal to the setting pressure of the safety valves. Some agencies also require consideration of the tank vapor pressure at the maximum service temperature and the pressure of the inert gas for tanks unloaded by means of inert gas. Table V.2 presents a comparison of the vapor pressure requirements for the various regulatory agencies and tank configurations.

The procedure for designing a pressure vessel is sufficiently established to ensure that the design of an LNG tank to account for vapor pressure is a straightforward procedure once an accurate value of the maximum expected vapor pressure is determined. The maximum design vapor pressure should include all possible factors that affect the tank vapor pressure. For

TABLE V.1

SUBJECTIVE RATING OF TANK LOADS

Load Category	Relative Importance to Tank Design			Reliability of Prediction Method	Additional Research or Experimental Verification Needed	Overall Rating [1]
	Independent		Membrane			
	Grav.	Pres.				
Vapor Pressure	Low	High	Low	Good[2]	Yes	5
Liquid Head	Mod	Low	Mod	Good	No	7
Static External Pressure	Mod	Mod	Mod	Good	No	7
Weight of Tank & Contents	Mod[3]	Mod[3]	Low	Good	No	7
Still-Water Hull Deflection	Low	Low	Mod	Good	No	7
Static Inclination	Low[3]	Low[3]	Low	Good	No	8
Collision Loads	Low[4]	Low[4]	Low[4]	Poor	Yes	4
Thermal Gradients	Mod	Mod	Mod	Fair	Yes	4
Wave-Induced Loads	High	High	High	Fair[5]	Yes	1
Acceleration of Tank CG	High	High	High	Fair	Yes	1
Dynamic External Hull Pressure	Low	Low	Low	Poor	Yes	4
Dynamic Internal Pressure	Mod	Mod	Mod	Good[6]	Yes[6]	5
Sloshing[7]	High	High	High	Poor	Yes	2
Vibrations	Low	Low	Low	Fair	Yes	5
Fatigue Loads	Low	Low	Low	Poor	Yes	4
Fracture Loads	Low	Low	Low	Poor	Yes	4
Combination of All Loads	High	High	High	Poor	Yes	1

[1] Highest rating is 1 (i.e., most important load).
[2] Exclusive of rollover.
[3] Primarily for tank support design
[4] From the standpoint of acceleration only; not including penetration.
[5] Prediction methods good if accurate representation of sea state is available except for slamming and bow flare immersion impulsive type loads.
[6] Good, providing long-term accelerations are known and method of combining accelerations is validated.
[7] Load important only when partial filling is allowed.

TABLE V.2 COMPARISON OF VAPOR PRESSURE CRITERIA

Agency	Maximum Vapor Pressure (kg/cm^2) for Each Tank Configuration			
	Independent		Membrane	Integral
	Gravity	Pressure		
ABS	0.70	0.70	No regulations	No regulations
BV	0.70	0.70	No regulations	No regulations
DnV	P_1	P_1	P_1	P_1
GL	0.70	$2.0+0.3\ \gamma h$	0.25, 0.70*	0.25, 0.70*
LR	No Regs.	No Regs.	No regulations	No regulations
NK	0.70	$2.0+0.3\ \gamma h$	0.25, 0.70*	No regulations
USCG	0.282, 0.703*	No Regs.	0.282, 0.703*	0.282, 0.703*
IACS	0.70	$2.0 + Ac\gamma^{3/2}$	0.25, 0.70*	No regulations

* With increased scantlings

P_1 - The highest setting of 1) the pressure of the safety valves, 2) the vapor pressure at reference temperature, 3) the vapor pressure at 45°C for tanks without cooling, and 4) the pressure of inert gas for inerting operations.

example, the tank design vapor pressure should include the highest safety valve setting, the vapor pressure at the maximum service temperature, and the maximum pressure for inerting to off-load the tank.

Since the LNG in a tank is continually boiling-off, increasing the pressure inside the tank (this is especially important for ships not equipped to burn boil-off vapors as fuel), the functioning and sizing of the safety relief valves are very important. Also, in-port venting of boil-off vapors may not be allowed by the cognizant regulatory agency, as direct venting of boil-off vapors to the atmosphere may constitute a fire danger to the surrounding community. Generally, docking procedures require approximately five to six hours, in which time only 1/16 of 1% of the tank's capacity will boil off.[18] However, an accident may strand the ship long enough to allow the tank pressure to build up to the relief valve setting. The rate of steady pressure increase in the tanks with no venting has been calculated to be 1.61×10^{-3} kg/cm^2-hr for the LNG carrier Descartes, which has relief valves set to cycle at 0.224 kg/cm^2 on each tank.[19] Assuming that the relief valves do not cycle, either by failure or in the event of a grounding accident, approximately 96 hours would elapse before the pressure in the tank would reach the setting pressure of the safety relief valves.

Narter and Swenson [20] have discussed the sizing of relief valves. Generally, designs approved by the ABS for tanks with a design pressure not much more than 0.281 kg/cm^2 above atmospheric pressure have had their relief valves sized by the following equation with adequately conservative results.

$$Q = 6.33 \times 10^5 \, \frac{FA^{0.82}}{LC} \sqrt{\frac{ZT}{M}} \quad \text{(scfm)} \qquad (5)$$

where

Q = Minimum required rate of discharge in terms of cubic feet of air per minute at standard conditions.

F = Fire-exposure factor which ranges from 1.0 to 0.1 as determined by the performance of the thermal insulation under fire conditions and the degree of metal screening of the insulation from the fire.

A = The area of the tank exposed to fire and to be taken as the total area of a prismatic tank minus the bottom-surface area.

L = Latent heat of the product being vaporized in BTU per pound.

C	=	Constant based on the relation of the specific heats of the product.
M	=	Molecular weight of the product.
T	=	Temperature in degrees Rankine.
Z	=	Compressibility factor of the gas at relieving conditions (if not known use Z = 1.0).

This formula was adopted from the ASME Code for Unfired Pressure Vessel which gives a method for sizing pressure relief valves derived from the Compressed Gas Association. BV and USCG provide formulas that are essentially the same as the ABS formula. DnV and NK's formulas are to be used to compute the heat input into the tank in case of a fire. The valves are then to be capable of discharging the quantity of gas generated by this heat input. No means are provided by these two agencies to compute the

$$\overline{Q} = 12,200\ A^{0.82} \quad \frac{\text{kcal}}{\text{hr}} \tag{6}$$

discharge in their regulations.

LNG rollover may present an additional phenomenon that should be considered in vapor pressure criteria. Sarsten[21] reported an incident involving LNG stratification and rollover at the LNG Terminal in La Spezia, Italy, in 1971. In this case relatively warm, high-density LNG was introduced into a partially filled storage tank through a nozzle below the lighter and cooler LNG already in the tank. The static pressure of the initial tank heel kept the warmer-heavier LNG from vaporizing. Figure V.1 shows the storage tank pressure history following the introduction of the heavier LNG from the Esso Breger. Approximately 24 hours after the completion of the off-loading operation the vaporization rate in the storage tank increased from a nominal 1 m-kg/hr to much greater than seven times the normal release rate (the internal pressure in the tank reached a maximum of 710 mm H_2O, or 40% greater than the design pressure of 500 mm H_2O).

While stratification and subsequent rollover of LNG will not be a problem as long as the cargo is agitated (as is the case whenever the ship is in the seaway), rollover may be a problem when partially filled tanks (or tank) are maintained on return voyages for cool-down purposes. During the return voyage, lighter components of the LNG may vaporize so that when the ship reaches the loading terminal, the cargo will be significantly heavier than before. Then during filling operations, lighter and possibly cooler LNG will be introduced on top of the "aged" LNG already in the tank. If the ship is prevented from leaving port immediately after the filling

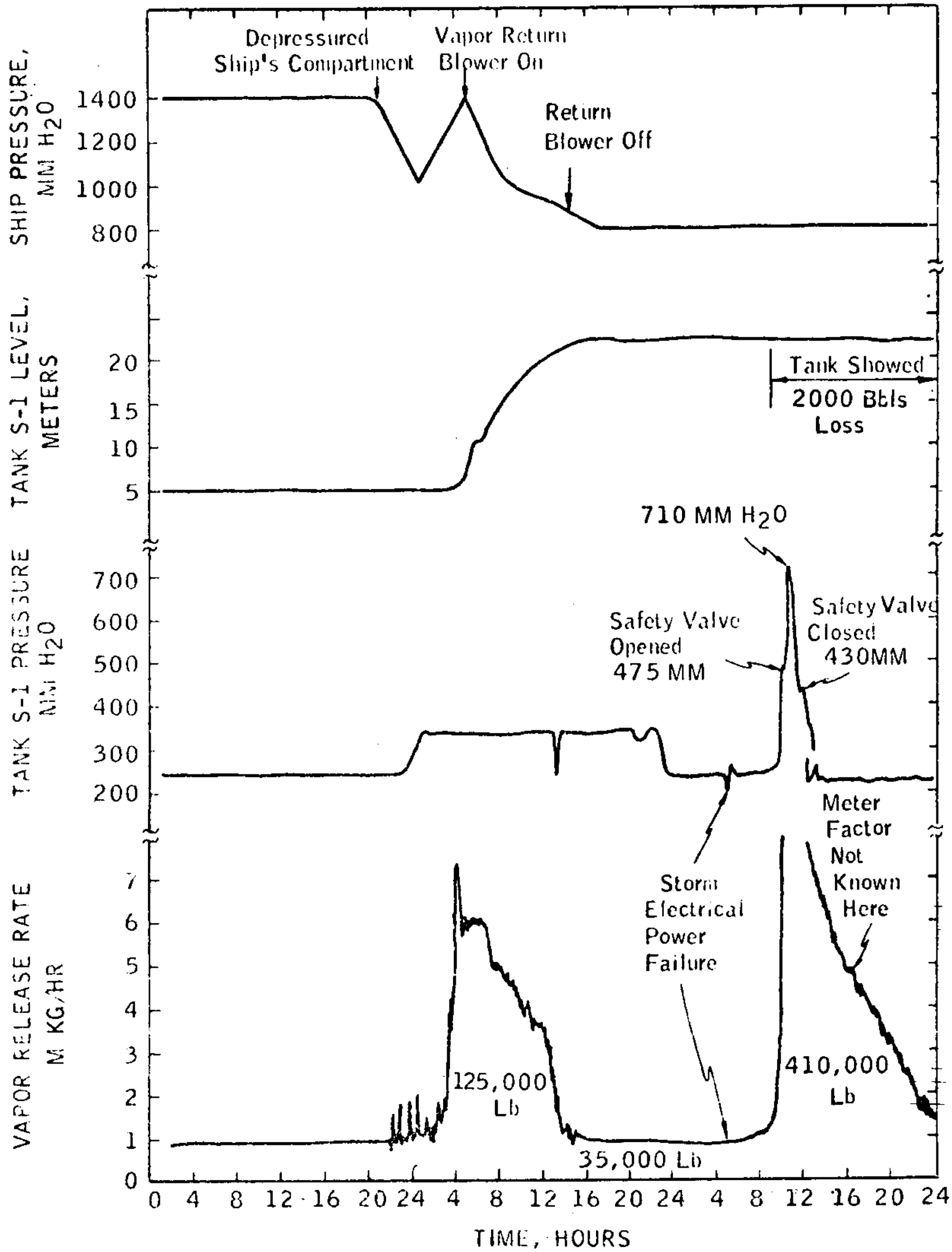


FIGURE V. 1. LNG STORAGE TANK S-1 LOADING HISTORY operation is complete, agitation of the stratified layers will not occur, and rollover may be a problem.

Studies of LNG rollover should be performed to determine whether effects of the high vaporization rates should be included in the design of LNG tanks and associated safety vapor relief systems.

Updated Vapor Pressure Criteria

Procedures for determining the actual in-service vapor pressure are not given by the agencies. Therefore, a more complete vapor pressure design criterion should include:

- a definition of vapor pressure requirements for pressure and non-pressure vessel tanks

- a means of determining the maximum in-service vapor pressure to include:

  (a) the highest safety valve setting

  (b) the vapor pressure at the maximum service temperature

  (c) the highest inerting pressure for off-loading

- the design of the tanks to pressure or non-pressure vessel regulations (such as the ASME codes) dependent on the largest of a, b or c.

- a method to determine the flow capacity of the relief valve system. Perhaps the effect of the high vaporization rate associated with LNG rollover should be included in the sizing of pressure relief systems.

V.2.2 Static Liquid Head

All agencies require static hydraulic testing of completed tanks. The test medium for the hydraulic tests is water for all agencies except the USCG, which requires the test medium have the density of the actual cargo. In addition to hydraulic tests, the agencies require that static and dynamic heads be combined in computing the final design internal pressure head. Table V.3 compares the various agency test requirements concerning static liquid head assuming that the considered tank has the same general dimensions as the number 6 tank of a 125,000 m^3 membrane tank LNG ship,

TABLE V.3. COMPARISON OF STATIC LIQUID HEAD CRITERIA FOR TANK 6 OF A 125,000 m^3 MEMBRANE TANK LNG SHIP

Agency	Required Test Head (m of water) for Each Tank Configuration			
	Independent		Membrane	Integral
	Gravity	Pressure		
ABS	26.4	26.4	No requirements	26.4
BV	26.4	15.0	26.4	26.4
DnV	29.1	29.1	29.1	No requirements
GL	25.0	No Reqs.	No requirements	24.6
LR	26.4	26.4	26.4	26.4
NK	26.5	26.5(27.5)*	26.5	26.5
USCG**	> 12.6	> 12.6	> 12.6	> 12.6
IACS***	MARVS	MARVS	MARVS	MARVS

* For Type C Pressure Vessel Only.

** Test head may be no less than the stated value; test liquid is to have a density equal to the density of LNG.

*** MARVS = Maximum Allowable Relief Valve Setting.

and a design vapor pressure[4] of 0.7 kg/cm^2 and a cargo density of 0.5 T/m^3. This tank is 24 m in depth and for a cargo density of 0.5 T/m^3, the actual static head would be 12 m H_2O. It is noted from Table V. 2 that the required test heads for all agencies exceed the actual static head. Also, in most cases, a safety factor of approximately 2.0 is apparent. This safety factor is not as large when considering that the static tests are also designed to account for other static tank loadings such as vapor pressure. Since the determination of actual static head is straightforward, once the tank depth and cargo density are defined, the static head criteria should essentially ensure that the tank design withstand a test pressure reasonably greater than the actual static head. This is the case for all the agencies; therefore, no updated criterion is needed.

V. 2. 3 Static Design External Pressure

In order to prevent implosion of tanks, all agencies require consideration be given to the differential pressure to which any portion of a tank may be exposed. Table V. 4 shows the various agencies' requirements concerning static design external pressure. As can be seen from the table, only DnV, GL and NK require that the tank be designed to withstand specific pressure loadings. These loadings are defined in terms of opening pressures of relief valves, the liquid head which may result from shipping of green seas on tanks which protrude through exposed decks, and compression forces due to tank shell weight and contraction of insulation. All agencies except DnV and GL require that tank securing arrangements be adequate to prevent flotation of empty tanks which could occur in the event the hold space was flooded to the design draft. The USCG requires that membrane tanks be evacuated to the negative pressure setting of the vacuum relief valves plus the pressure setting of the secondary barrier pressure relief valve.

Depending on the tank design, different measures to limit the differential pressure on a tank boundary should be required. Differential pressure may be limited in independent and integral (where they are allowed) tank designs by providing pressure relief valves on the hold spaces, and vacuum relief valves on the tanks. In addition, compressive forces on the tank wall due to the weight and contraction of the adjacent insulation must be considered. An additional contribution to the differential pressure on independent and integral tank configurations is due to the flotation of empty tanks that may occur when the hold spaces are flooded up to the design draft.

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[4] The agencies generally use the symbols kg/cm^2, kp/cm^2, or kgf/cm^2 to indicate that the units of vapor pressure are kg-force/cm^2. Likewise, the symbols T/m^3 indicates that the units of density are metric tons/m^3.

TABLE V.4. COMPARISON OF STATIC DESIGN EXTERNAL PRESSURE CRITERIA

Load	Agency							
	ABS	BV	DnV	GL	LR	NK	USCG	IACS
Opening pressure of vacuum relief valves, generally ≥ 0.25 kg/cm^2			I M	*		I M It		
Set pressure of relief valves for completely enclosed spaces, otherwise zero			I M	*		I M It		
External water head for tanks protruding through exposed decks, otherwise zero			I M			I M It		
Compressive forces in shell due to weight & contraction of insulation & weight of tank shell			I M			I M It		
Provide some means to prevent flotation of empty tanks that may occur when the hold spaced are flooed up to the design draft	I	I			I M It	I M It		I M It
Test actual tank by evacuating tank to negative pressure of vacuum relief valve plus the setting of the secondary barrier relief valve							M	

I - Consider loading for independent tank designs.

M - Consider loading for membrane tank designs.

It - Consider loading for integral tank designs when this design is allowed.

* - Independent gravity tank only.

For membrane designs, consideration must be given to the problem of tank wall buckling caused by the collection of vapor between the membrane and the insulation in the event a leak develops. Some pathway for the vapor to escape through the insulation must be provided for, and the required size or the discharge capacity of the pathway should be determined from the anticipated rate of vapor collection from a given crack size.

An additional contribution to the differential pressure must be considered for all tanks which protrude through exposed decks is the liquid head which may appear above the top of the tank tops during shipping of green seas. Ochi [22] measured pressures on the exposed deck of a 13-ft Mariner model and found that although the probability of shipping green seas (deck wetness) was relatively low, pressures on the order of 0.70 kg/cm^2 were typical for full draft, sea state 7. Presently only DnV and NK require consideration of shipping of green seas.

Updated Static External Pressure Criteria

Improved static external pressure criteria should include the following points:

- On all tanks, appropriate pressure and vacuum relief valves are to be placed on the tanks and hold spaces. Formulas for sizing the relief valves should be provided.
- A pathway for vapor to escape from between the tank wall, and insulation should be provided on membrane tanks.
- For tanks protruding through the exposed deck, the tank cover should be designed to withstand the pressure associated with shipping of water forward. Formulas for this purpose should be provided.

V.2.4 Weight of the Tank and Its Contents

All the agencies except ABS and LR require that some consideration be given to the weight of the tank and its contents. Although the structural details of the supporting arrangements are quite different for independent and membrane configurations, the agency requirements concerning this load are primarily independent of tank design. For instance, BV, GL, NK and IACS require that the static weight of independent and membrane tanks and their contents augmented by the dynamic accelerations be used to calculate the load on the ship's hull, the tank walls, and the support and securing arrangements. DnV requires that the static load due to 99% filling by volume be used for designing independent and membrane tanks.

ABS and LR provide no regulations concerning tank weight or the weight of the cargo. BV requires that no deformations in the membrane wall liable to induce excessive bending stress occur.

Updated Weight of Tank and Contents Criterion

A complete criterion for this load should include the following elements:

- The cargo mass should be calculated assuming the tank is full.
- The weight of the cargo and the tank should be calculated from the mass of the cargo and the tank multiplied by the acceleration due to gravity and augmented by the maximum vertical acceleration that is expected to be encountered by the ship.
- The weight calculated above should be used to design the structural details of the supporting arrangements. For Independent tanks the supports should transmit the entire load to the ship's hull while preventing excessive stress concentrations in the tank wall, ship hull and the supports themselves. For membrane tanks the supporting insulation should be designed to transmit the entire load to the hull without introducing excessive stress concentrations on the tank wall, insulation or the hull of the ship. Maximum limits for stress concentrations should be explicitly given, for each material commonly used in LNG ships.

V.2.5 Still-Water Hull Deflections

The effect of still-water hull deflections on the tank supporting and securing arrangements and on the tank itself should be considered. Deflections of the tank wall and stress concentrations in the hull, tank wall and the support structures should be avoided. The agencies provide almost no regulations concerning still-water hull deflections; instead, the regulations are generally concerned with bending moments. However, the agency requirements, whether concerned with hull deflections or bending moments are not specific. For instance, DnV requires that static forces (due to hull deflections) imposed on the tank be considered. BV and GL require that the design of tanks and their supports take into account proper combinations of various loads including ship hull deflections and the weight of the full tank. BV, in addition, requires that the tanks not take part in bending of the ship and are not subject to abnormal stress due to ship hull deformations. ABS (horizontal pressure tanks) and LR (pressure tanks), $P_o > 0.70\ kg/cm^2$, require that the tank supports provide uniform support to the pressure vessel without introducing moments in the tank due to hull deflections in the seaway. IACS requires that cargo tanks be restrained from bodily movement under static and dynamic loads, while allowing expansion and contraction of the tank under temperature variations and hull deflections without undue stressing of the tank and hull. NK presents no requirements concerning hull deflections; however, requirements for hull bending moment are given. Loads acting on the tanks through tank supports due to bending and torsional moments are to be calculated taking into account deflections of the double bottom and tank bottom where the tank and hull bottoms are coupled by a supporting structure. The USCG requires that fatigue tests be conducted on model membrane tanks. The model is to be prestressed in tension to the maximum amount caused by cargo cooling, static head, cargo pressure, and the still-water hull deflection. The hull deflections in the seaway determine the cycling amplitude.

Updated Still-Water Hull Deflection Criteria

A complete still-water hull deflection criteria should include the following requirements:

- For all tank designs, deformations in the hull should not lead to excessive stress concentrations in the tank shell or in the supporting arrangements.

- For membrane designs the deformations in the hull should not lead to excessive strains in the membrane.

- Maximum allowable stresses and strains should be explicitly provided for materials commonly used in LNG ships.

V.2.6 Static Inclination

The tank and associated securing arrangements should be designed to withstand a specified static inclination without exceeding the design stress. The secondary barrier, for those ships so equipped, should be designed to contain the cargo at the specified static inclination. All the agencies except ABS and BV have requirements concerning static inclination. DnV, GL, LR, NK and IACS all require the tanks be designed to withstand a static inclination of 30°. DnV and IACS require that the supports and the tank be designed so that the stress at 30° static inclination be less than the design stress. LR[5] and IACS require that the secondary barrier be of sufficient

[5] Assuming one tank has failed.

extent to contain the cargo at the 30° inclination. The USCG allows a maximum heel angle of 15° during the final condition of flooding; this may be increased to 17° if no part of the deck is immersed.

Updated Static Inclination Criteria

A more complete static inclination criteria should include the following points:

- The tank and supporting arrangements should be designed to withstand independently a maximum list and a maximum trim of specified magnitudes.
- At the maximum list or trim the stress in the tanks and supports should not exceed the design stress.
- The maximum allowable stresses should be given for each material commonly used in LNG ships.
- At either maximum inclination, no part of the deck plating should be immersed.
- Assuming that one tank has failed, the secondary barrier should be capable of preventing any LNG from coming into contact with the hull structure (or any material not designed to withstand sudden cooling to the temperature of LNG) when the ship is at the maximum list or trim angle.

V.2.7 Collision Loads

Because longitudinal accelerations produced by the wave-induced loads are low, there is a need to specify a longitudinal acceleration which would guarantee structural integrity of tank supports in the event of collision. There is a remarkable consistency in the agency requirements as summarized in Table V.5. All agencies, except BV and USCG, require a longitudinal acceleration of 0.5 g for collision. BV requires 0.3 g, and the USCG sets requirements in terms of survivable damage. Where the agencies specify only an 0.5 g longitudinal acceleration, it was assumed that this acceleration could come from fore or aft. GL, NK and IACS give reduced acceleration from aft.

Although accident reports were not reviewed to substantiate the acceleration levels specified by the agencies, a recent study of tanker collisions for the U. S. Coast Guard [23] indicates that the acceleration levels associated with ship-to-ship collisions are low. This study included a survey of the literature, a study of accident reports plus an analytical

TABLE V.5. SUMMARY OF RULES REGARDING COLLISION LOADS

Agency	Acceleration Acting on Full Tank (g)					
	Independent Tanks		Membrane Tanks		Integral Tanks	
	Fwd	Aft	Fwd	Aft	Fwd	Aft
ABS	0.5(1)	0.5(2)			SC(3)	SC
BV	0.3(4)	0.3(2)				
DnV	0.5	0.5(2)	0.5	0.5(2)		
GL	0.5	0.25	0.5	0.25		
LR	0.5	0.5(2)	SC	SC		
NK	0.5	0.25	0.5	0.25		
USCG	SD(5)					
IACS	0.5	0.25	0.5	0.25		

(1) For tank with vapor pressure, $P_o > 0.7\ kg/cm^2$. No requirements for tanks with lower vapor pressure.

(2) Stated for longitudinal direction only. Both fore and aft directions were assumed.

(3) SC indicates that these tanks (in general) are specially considered.

(4) Where pitching keys are provided they are to be determined from a longitudinal force equal to 0.4 g while a longitudinal force equal to 0.8 g is considered for the lower keys.

(5) USCG specifies guidelines in terms of survivable damage (SD). No acceleration associated with collision are specified; however, certain tanks are to be designed to ABS or equivalent standards (see Section A.2.7).

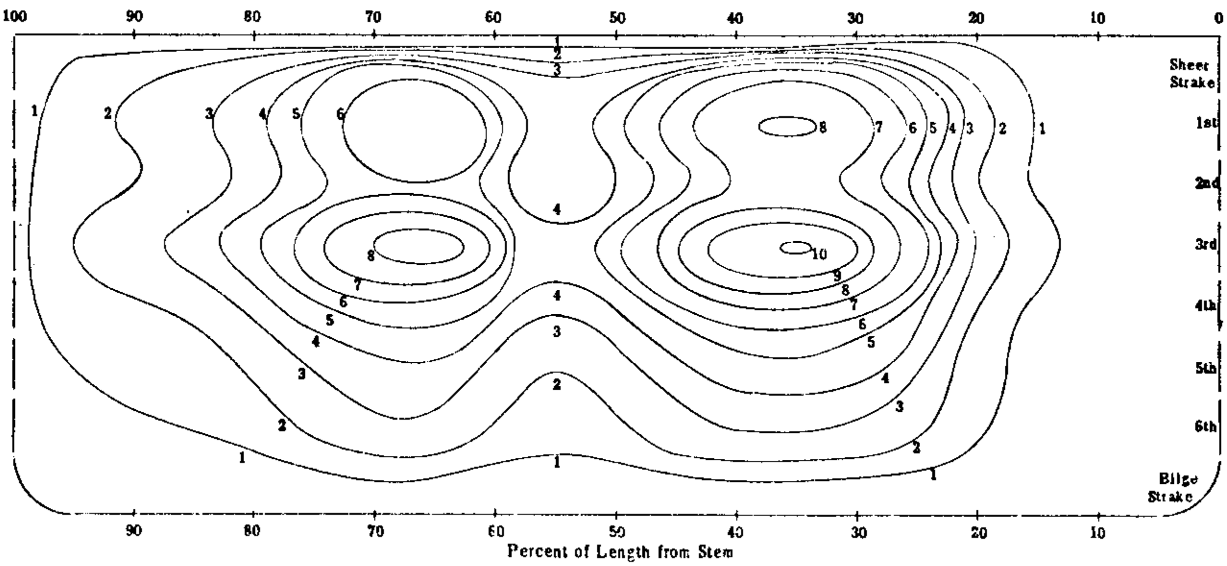
investigation of the penetration phenomena. It is reported that model tests of ship-to-ship collisions by Spinelli [24, 25] suggest acceleration levels of about 0.1 g. Also, calculations in Reference 23 tend to support these results. Acceleration levels were not reported directly, but the total energy of deformation and the depth of penetration were calculated for several collision conditions. If a linear force-deformation relationship[6] is assumed, then the peak force during the collision can be estimated. This force, acting on the mass of the lightest ship will give the peak acceleration. Using this approach, accelerations for the seven collision cases investigated in Reference 23 were found to range from about 0.05 g to 0.22 g, well below the longitudinal acceleration specified by the agencies. However, collisions with fixed objects could easily result in higher accelerations.

As a minimum, the rules of each agency should state explicitly the longitudinal acceleration for collision coming from both fore and aft directions. Also, a better approach would be to establish an acceleration vector which reflects the most probable direction in the event of collision. Hawkins, et al., [26] show that damage from collisions with fixed obstacles occurs most often within 20 percent of midship and that damage from collisions with vessels alongside occurs most often within 30 percent of the midship. Contours of equal probability (based on an arbitrary scale from 1 to 10) are given in Figure V.2 for these two conditions. Because the agencies do not require that the collision loads be combined with any other load, the tank supports are checked for a longitudinal acceleration vector only. It appears from accident studies, reflected in Figure V.2, that collision vectors most probably lie between a_x and a_y. Thus, a more realistic collision criteria would result from a simultaneous application of longitudinal and lateral accelerations. The magnitude of these accelerations could be established from model test, full-scale test, or perhaps from an extension review of accident reports.

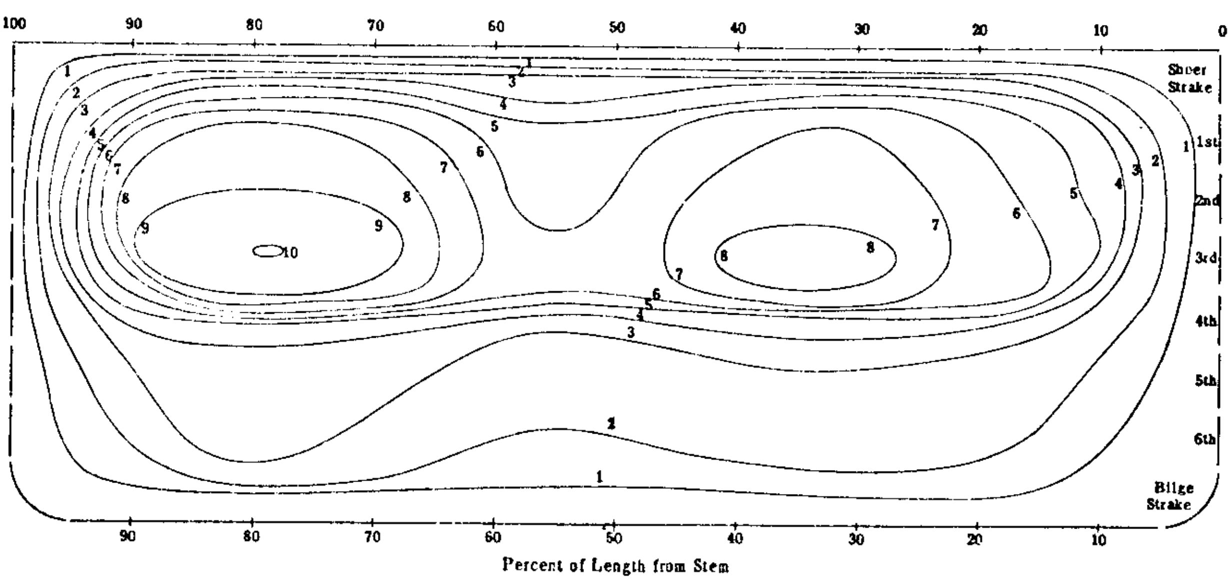
Updated Collision Criteria

The tank and supporting structures should be designed to withstand a collision acceleration of specified magnitude and direction when the tank is filled to its maximum capacity with the heaviest cargo. For convenience, the acceleration vectors could be specified in the longitudinal and transverse directions. Then the resultant acceleration would be:

[6] This assumption will generally overestimate the peak force.



(a) Collisions with Piers, Quays, and Locks



(b) Collisions with Vessels Alongside

FIGURE V.2. DAMAGE LOCATION PROBABILITY CONTOURS

$$a_R = \sqrt{a_x^2 + a_y^2} \tag{7}$$

and the relative direction would be determined from

$$\tan\theta = \frac{a_y}{a_x} \tag{8}$$

Although not part of the load criteria, the extent of damage which the ship is designed to withstand, without rupture of the cargo tank, should be specified. This requirement would have the same aim as the acceleration criteria, that is, to prevent loss of cargo and thus reduce the hazard associated with cargo spillage in the event of a collision.

V.2.8 Thermal Loads

The temperature differences between the hull girder and the cargo tanks on LNG ships is quite high (up to 210°C). Because some means of attachment between the hull and the tank are required for tank support, the potential exists for high thermal gradients and high thermal stresses. Furthermore, if the hull were to cool locally to temperatures below the allowable service temperatures for the hull material, brittle fractures could result. Thus, thermal loads are particularly important for LNG ships both from the standpoint of tank design and ship hull structural integrity. Temperature loads as discussed here will refer to stationary and transient temperature gradients in the tanks or loads introduced in the tanks due to temperature gradients in the ship's hull. These loads are limited primarily to those produced by the cargo and not those caused by environmental conditions, i.e., solar heating of the deck.

Agency rules for thermal loads are not specific, as indicated by the summary in Table V.6. BV only requires that hull temperatures be prevented from dropping below minimum values, whereas ABS and LR require that the design of tank supports minimize load transfer between the tank and the hull due to thermal contractions. The other agencies specify that thermal stresses due to both temperature transients and stationary temperature gradients be considered. DnV further requires expansion provisions for LNG piping. Only two agencies specifically mention temperatures for which the calculations are to be performed. GL sets a minimum cargo temperature for full tanks and the USCG specifies a minimum thermal gradient from the top to the bottom of empty tanks.

TABLE V.6. SUMMARY OF RULES REGARDING THERMAL LOADS[1]

AGENCY	Require stress control through support design.	Require consider-ation of thermal transients.	Require consider-ation of stationary thermal gradients.	Require expan-sion provisions for piping.	Specify control of minimum hull temperature.	Specify minimum cargo temperature for calculations.	Require design evaluation by testing.
ABS	O						O
BV					O[2]		
DnV		O	O	O			
GL		O	O			O	
LR	O						
NK		O	O[3]				
USCG		O	O[4]				
IACS		O	O				

(1) Rules apply only to independent and membrane tanks.

(2) BV specifies the minimum steel temperature for normal service and with leakage.

(3) NKK specifically requires consideration of stationary gradients through the thickness; thermal gradients and stationary gradients for partially filled tanks.

(4) USCG specifically requires consideration of thermal gradients for only one independent tank (IIT of Table II.2). Thermal stresses in full as well as empty tanks are to be considered with the thermal gradient for the empty tank specified as 83.3°C. Membrane tanks are to be tested including thermal shocks and thermal gradients.

The agencies should provide guidelines on the extremes in temperature to be considered for computing thermal stresses due to stationary thermal gradients. For example, this would consist of a minimum cargo temperature (perhaps considering some degree of super-cooling) and a maximum ambient or hull temperature. As for thermal transients, they depend almost entirely upon the methods by which the tanks are precooled prior to filling, and it would be inappropriate to establish a restrictive criterion which might penalize effective cooldown systems. This is not the case for stationary thermal gradients, in that all tanks are subjected to them.

Becker and Colao [27] addressed the problem of thermal stresses in LNG tank structures. The authors indicate that a conservative estimate of thermal shock stress is given by

$$\sigma = \alpha E \Delta T \qquad (9)$$

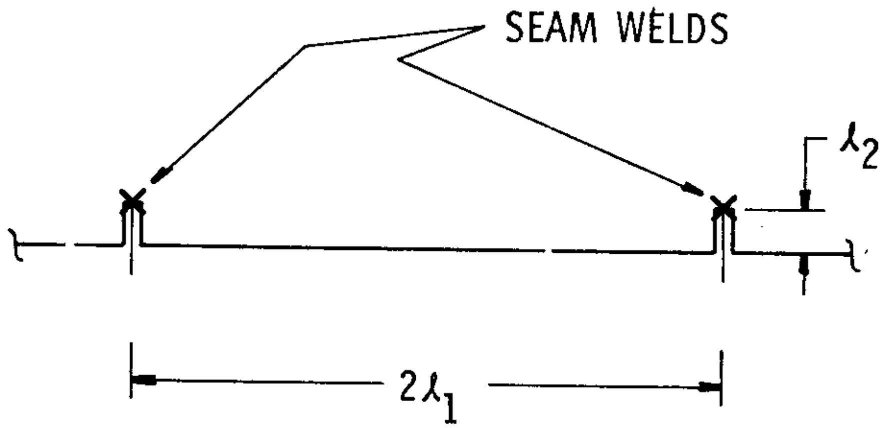
where α is the coefficient of thermal expansion of the material, E is the elastic modulus, and ΔT is the total temperature change. This is the stress that would be introduced in a uniform member fully restrained against contraction. To investigate thermal stresses, the following coefficients of thermal expansion for typical tank materials were used:

INVAR:	$\alpha = 2.34 \times 10^{-6}$ cm/cm/°C
5083 AL:	$\alpha = 18.9 \times 10^{-6}$ cm/cm/°C
9% NICKEL STL:	$\alpha = 9.55 \times 10^{-6}$ cm/cm/°C

Considering a total temperature change of 183°C (plus 21°C to -162°C) the stresses computed by Eq. (10) are

INVAR:	$\sigma \approx 645$ kgf/cm^2
5083 AL:	$\sigma \approx 2{,}780$ kgf/cm^2
9% NICKEL STL:	$\sigma \approx 3{,}680$ kgf/cm^2

It is apparent that these stress levels are unacceptable for the 9% nickel steel and 5083 aluminum, but that the stress is quite low in the INVAR. However, contrary to conclusions by Becker and Colao,[27] Eq. (9) does not necessarily represent the upper limit of the stress in an INVAR membrane. This can be demonstrated by considering, for example, a segment of INVAR lining as shown in Figure V.3. Depending upon the ratio of ℓ_1 to ℓ_2, the bending stresses introduced in the vertical tabs by contraction in the horizontal segments can be quite large. The load P of Figure V.4, derived for a unit strip of the membrane, is given by the following equation:



FIBURE V. 3. SECTION THROUGH AN INVAR MEMBRANE OF A GAS-TRANSPORT MEMBRANE TANK

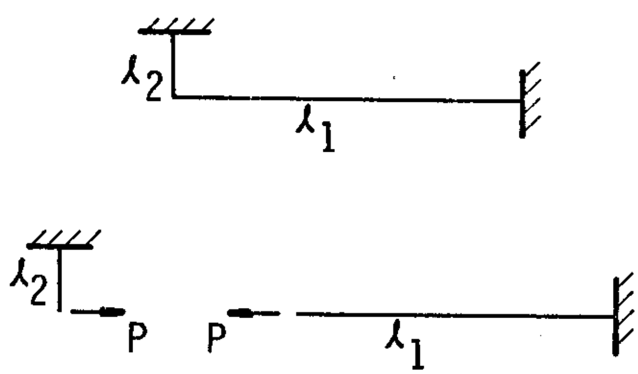


FIGURE V. 4. SCHEMATIC OF MEMBRANE FOR STRESS CALCULATION

$$P = \frac{\alpha E \Delta T \ell_1}{\frac{\ell_2^3}{3I} + \frac{\ell_2}{A}} \quad (10)$$

where

$$I = \frac{h^3}{12}$$

$$A = h$$

and h is the thickness of the membrane. If we choose the following parameters,

$$\ell_1 = 38.1 \text{ cm}$$

$$\ell_2 = 1.27 \text{ cm}$$

$$\Delta T = 183^\circ C$$

$$h = 0.1016 \text{ cm}$$

P, computed from Eq. (10), is 7.36 kgf. Using the standard equation for flexure, the bending stress is found to be approximately 221 kg/cm^2, which is quite high. However, the geometry used in this particular example may not be representative of the dimensions used in the GAZ-Transport membrane tank. The stress level introduced by flexure can be controlled by varying the lengths of ℓ_1 versus ℓ_2. Even so, the illustration does emphasize that stresses significantly higher than those predicted by Eq. (9) can be achieved in typical tank structures due to thermal contractions.

In summary, it appears that thermal stresses introduced by very localized cooling can be quite high, particularly in aluminum and 9% nickel steel. Thermal stresses in INVAR may be low or high depending upon the structural configuration. Superimposed on other static stresses, such as those due to liquid head, vapor pressure and ship hull static deflections (there should be no dynamic loads associated with the initial filling operation), stresses above the yield of the material can be introduced. The only way to limit the transient stresses to reasonable values for all tank designs and materials is to specify a tank cooldown procedure whereby thermal gradients are kept to a minimum. This is current practice in the industry, and it should be reflected in the rules. As already indicated, the temperature extremes should also be specified for computing stresses associated with stationary thermal gradients.

Updated Thermal Load Criteria

Thermal stresses in the tank must be considered for both stationary and transient thermal gradients with due consideration of tank restraint. Stationary thermal gradients are to be calculated for the worst case combination of the minimum expected cargo temperature and the maximum expected hull temperatures. The minimum cargo temperature and the maximum hull temperature must be approved by the societies. In the absence of such approval, the cargo temperature shall be the boiling temperature at atmospheric pressure and the hull temperature shall be 45°C.

V.2.9 Wave-Induced Loads

A review and evaluation of the current methods used by the classification societies to calculate the wave-induced loads are given in Chapter IV and in Appendix B. Table IV.1, compiled by Lewis, [6] shows that, in practice, all agencies calculate the wave-induced loads and that the calculation procedures are similar. However, current practice is not always reflected in the rules of the societies. The summary of the rules given in Section 2.9.9 of Appendix A reveals that two of the six classification societies, ABS and LR, plus the USCG, do not call for calculation of the wave-induced loads. Four of the societies, BV, DnV, GL and NK, plus IACS, require calculation of the wave-induced loads for most types of tanks. A brief summary of the rules for the wave-induced loads is given in Table V.7. Among the agencies which do address these loads, there is considerable variation in the specificity of the rules.

It is not possible to specify in the rules all details of the complicated procedures required for computing the design values of the wave-induced loads. However, the rules should certainly reflect current practice by the societies and state that the calculations must be performed by the society or by specially approved methods. Our review indicated that the methodology for calculating the loads is constantly being refined and, therefore, specific regulations are probably not warranted at this time. Areas in which changes are occurring include: (1) the use of measured rather than visually observed sea state data, (2) the methods of representing the sea spectra, i.e., the use of actual measured spectra versus mathematically derived spectra, (3) better definition of the hydrodynamic roll damping, (4) better treatment of the ends of the ship in computing the RAO's, and (5) calculation of the dynamic components of the wave-induced loads which include slamming, whipping and springing. Differences in the calculation procedures used by the societies do exist and will probably continue to exist in spite of the efforts by associations such as the IACS. Although the differences in the resulting design values (long term predictions) are thought to be small, this cannot be determined for certain until trial calculations for identical ships are performed by all societies.

TABLE V.7. SUMMARY OF RULES PERTAINING TO THE CALCULATION OF WAVE-INDUCED LOADS

AGENCY	Tank Types for which Calculation of the Wave-induced Loads are Required	Ship Speed	Speed Reduction Allowed	Sea Spectrum	Sea State Data	Heading Probability	Probability Level for Maximum Value
ABS							
BV	I, M(1)	V_s (2)	Yes	P-M(3)	DBC(4) or N.A.	DBC	
DnV	I(Type A-II, Type B)(5), M		Yes		N.A.	Equal	10^{-8}
GL	I, M						(6)
LR							
NK	I (Type B Prismatic; Type A, B & C Pres. Vessel), M		Yes	(7)	DBC or N.A.	Equal	(6)
USCG							
IACS	I, M		Yes			(8)	10^{-8} Normally

(1) I - independent tank
M - membrane tank

(2) Ship Service Speed

(3) Pierson-Moskowitz

(4) DBC (determined by course); NA (North Atlantic)

(5) Calculation sometimes required for Type B

(6) Most probable in ship's lifetime

(7) Equation defined in rules

(8) Account may be taken of reduction in dynamic loads due to necessary variation of heading when this consideration has also formed part of the hull strength assessment.

Updated Criteria for the Wave-Induced Loads

The rules of each society should reflect its current practices. Even though it is not possible to specify all parameters which affect the calculations, guidelines should be provided whenever possible. For example, the following guidelines concerning the ocean environment might be included:

- Whenever possible, measured sea spectra for the actual route shall be used for the calculations; otherwise, the modified ISSC or the Pierson-Moskowitz spectra can be used.

- Sea state data shall be determined by route; otherwise, the use of North Atlantic data is required.

- If possible, the heading probabilities shall be determined by route. If not, they are assumed to be equal.

- The probability level for the maximum value prediction shall be established by the ship's expected service history; otherwise, a 10^{-8} value will be assumed.

In addition, the dynamic components of the wave-induced loads associated with slamming, whipping and springing shall be determined with due regard to the phasing of these loads with the slowly varying components of the wave-induced loads. Speed reductions in heavy weather can be considered in the calculation of the wave-induced and "dynamic" loads. If, however, the dynamic loads produced by slamming are not treated separately, no speed reduction is allowed. (It is generally accepted by the societies that the higher speed in severe seas will tend to cover the effect due to slam.)

V.2.10 Dynamic Hull Deflections

Deflections of the ship hull must be considered in the design of all LNG tanks. Even for the TECHNIGAZ self-supporting auto-compensated tanks, Alleaume and Alvarez de Toledo [28] report that approximately 30% of hull strains are transferred to the tanks. Of course, membrane tanks are designed to work with the hull of the ship, and for practical purposes strains in the membrane tanks produced by hull deflections are equal to those of the hull itself. Jackson and Kotcharian [29] give the cyclic strain history used in qualification fatigue testing of the Conch Ocean membrane tank. They report that the strain history shown in Figure V.5 represents the most severe case in terms of ship size and was computed for a ship of about 40,000 cubic meters. Peak strain is 1.1 mm/m at a probability

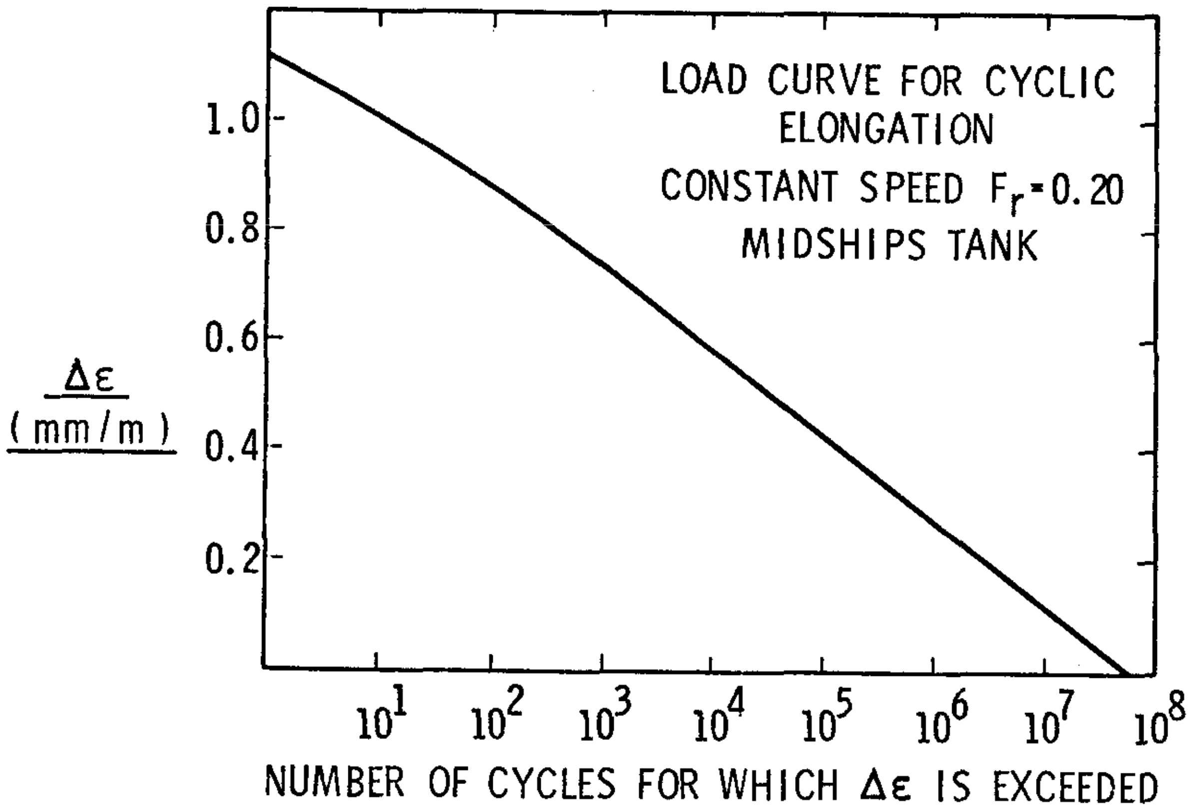


FIGURE V.5. CYCLIC STRAIN HISTORY FOR FATIGUE VERIFICATION OF CONCH OCEAN (now Technigaz) MEMBRANE TANK (Ref. 28) level of 10^{-8} Even though the different tank designs will respond to hull deflections in different ways, these deflections must always be considered in the tank structural analysis.

Not all agencies address the problem of ship hull deflections on LNG tanks. As is apparent from the summary of the rules presented in Table V.8, BV, for example, has no rules or guidelines for consideration of such effects. Those agencies that do address the problem generally do so in one of two ways, either by requiring that supports be designed to minimize tank hull interactions or by requiring that the loads and stresses introduced by the interaction be properly accounted for. IACS requires both that tank hull interactions be minimized by proper support design and that stresses resulting from the inevitable interaction be considered. However, IACS only addresses the problem for independent tanks. No consideration is given to membrane or integral tanks. Also, only one agency requires that the effects of localized hull deflections be considered; DnV specifically mentions deflections of the ship's double bottom.

While it does not seem advisable for the agencies to specify upper limits of strain for which the tanks are designed, as was done by Jackson and Kotcharian [29] for the Conch Ocean membrane tank, the rules should state the importance of hull deflections for all types of tanks and define more specifically the loads which should be considered when computing tank hull deflections. Overall deflections of the hull should include contributions of the wave-induced loads, thermal gradients, whipping and springing. Whipping and springing are part of the wave-induced loads but are not routinely included in the calculation procedures at this time. However, methods for calculating these loads are being developed rapidly, as discussed in Section V.15 (Vibrations). Localized deformations would be those attributable to normal hydrodynamic pressures associated with buoyancy plus those associated with slamming. Peak localized deformations from slamming are most likely to occur in the region 0.1 L to 0.3 L aft of the forward perpendicular, whereas the peak loads associated with overall hull deformation are most likely to occur near midship. Even though the maxima may not occur in the same region of the hull, contributions of overall hull bending deflections and local deflections will affect all tanks.

Updated Dynamic Hull Deflection Criteria

Tank supports should be designed to minimize tank-hull interactions; however, because these interactions usually cannot be avoided entirely, stresses in the tanks produced by the interactions must be considered for all tank types.

TABLE V.8. SUMMARY OF RULES REGARDING DYNAMIC HULL DEFLECTIONS

Agency	Require Consideration of Hull Deflections on Tank Loads and/or Stresses			Require Design of Supports to Minimize Tank-Hull Interactions		
	Ind	Memb.	Int.	Ind.	Memb.	Int.
ABS				O		SC(1)
BV						
DnV	O(2)	O(2)				
GL	O	O	O			
LR				O	SC	
NK	O	O				
USCG		O(3)				
IACS	O			O		

(1) SC indicates that the tank type is specially considered (in general) by the agency

(2) DnV specifically mentions deflections of the ship's double bottom in addition to overall hull longitudinal deflections.

(3) USCG requires testing of membrane tanks which is to include cyclic loads caused by maximum at-sea hull deflections.

In addition to the interaction caused by thermal contractions which are treated under thermal loads, interactions due to overall hull beam bending and local hull deformations, such as deflections in the double bottom, must be considered.

Hull deflections produced by the following loads are to be computed:

- Vertical and horizontal hull bending moments and shear as well as torsion associated with the slowly varying wave-induced loads.
- Vertical bending moments associated with slamming, whipping and springing.
- Local hull deformations produced by normal hydrodynamic pressures and slamming.

V.2.11 Accelerations

The design of the ship structure, tank walls, and tank supports is affected by the anticipated magnitudes of inertia forces due to longitudinal, transverse and vertical accelerations to which the various structures may be subjected. The basic approach used by all agencies is to determine the maximum anticipated accelerations in 10^8 wave encounters, 20 years or the life of the vessel. Because of the complexity of the problem, the agencies have resorted to statistical models such as those described in Chapter III (extrapolation of full-scale data or calculations of accelerations from observed sea conditions and the transfer function for ship response) to obtain the long term design acceleration. In addition, the agencies have developed formulas which they hope predict the behavior of the ship in a seaway in a reasonably realistic, but convenient, manner. The formulas or guidelines as indicated in Section A.2.11.9, can be broken into two categories. The first group consists of the agencies which provide formulas that may be used to calculate the long-term accelerations. These formulas were developed in building-block fashion by studying how the long-term acceleration changes when variations in ship length, breadth, speed, block coefficient or loading condition are introduced. In this manner, the formulas of DnV, GL, NK, and IACS were developed. [30,31] The second group, ABS, BV, LR and USCG require that the tanks be capable of withstanding simultaneous rolling, pitching and heaving motions of specified amplitudes and periods. This is as far as ABS, LR and USCG go; acceleration formulas have to be derived before the guidelines can be used for design purposes. (Expressions for a_x, a_y, and a_z are presented in Chapter III under the assumption that the motions could be considered sinusoidal.) BV provides a calculation procedure that is quite different from the other agencies' approach. BV's formulas provide acceleration as a function of roll, heave and pitch motions. The amplitudes and periods are given in terms of ship dimensions, so that acceleration can be calculated at any point in the ship.

Comparisons of the agency formulas with extrapolations from full-scale data and statistical calculations from program SCORES* were made in Chapter III. The results showed that the agency formulas for a_y and a_z were generally conservative relative to other prediction procedures. (No comparison of longitudinal accelerations has been made, as no data were available.) It was noted that BV formulas were generally less conservative than those of other agencies, and that the guidelines of ABS, LR, and USCG, when applicable, were the most conservative. In the case of the three LNG ships, the DnV, GL, NK and IACS formulas for lateral accelerations were about five times the SCORES* predictions, while at the same position the formulas predict vertical accelerations that were smaller than the SCORES* calculated values. However, as noted in Chapter III, the points considered may have been outside the valid region for these particular formulas.

Updated Acceleration Criteria

- Since comparisons of agency formulas with full-scale data were generally not possible for actual LNG ships, the conclusions stated above should be considered tentative. Full-scale data should be obtained on LNG ships, at a variety of measuring points, and over long periods of time. These data should be extrapolated to the long term using some method that is sensitive to the probability of encountering the various sea conditions on different routes and different headings, such as the combined Rayleigh-Normal method. Using these data, it is further recommended that the degree of conservatism of the agency formulas be estimated. If the conservatism is substantial, it may be desirable to revise the formulas in order to make the accelerations more realistic. Substantial economic savings may be realized if this is the case.

- For those agencies which limit the application to ships shorter than 183 meters, new guidelines should be developed, as the new generation of LNG ships is on the order of 300 meters long.

V.2.12 Dynamic Internal Pressure

Dynamic internal pressure is defined as all dynamic pressures acting on the interior of the tank with the single exception of sloshing pressures, which are handled separately. Therefore, dynamic internal pressures are determined for full or nearly full ($h/H > 90\%$) tanks. The basic approach to establishing the dynamic internal pressure is to determine a ship acceleration at the tank CG, and, using this acceleration in combination with the tank geometry and cargo specific gravity, to determine the dynamic pressure acting on the tank structure. All agencies use this basic approach in their criteria for establishing dynamic internal pressure. However, the specific formulas for dynamic internal pressure (P_d) and the method of establishing $P_{d_{max}}$ for tank design vary with each classification society.

To evaluate the agencies' P_d criteria, the maximum dynamic internal pressures as predicted from the various agency formulas for the No. 6 tank of a 125,000 m^3 membrane tank LNG ship were utilized as a basis of comparison. The results are shown in Table V.9 for the five classification societies which provide specific formulas. The location of the resulting maximum dynamic internal pressure for each of the societies is also indicated. In some cases, the agencies include a static head or a vapor pressure in the formulas for the dynamic internal pressure criteria. In such cases, the nondynamic pressure terms were eliminated such that the comparisons in Table V.9 are made on a uniform basis. Also, in some cases, the formulas were not specifically for a membrane tank; however, the dynamic pressure values were generated for the tank and ship dimensions of the 125,000 m^3 membrane tank ship since the tank structural design will not alter the dynamic pressures.

It is noted that the dynamic pressures range from about 6 to 11 meters of water. These differences in $P_{d_{max}}$ result from (1) each agency having different acceleration criteria resulting in differing values of accelerations used in the P_d formulas, (2) basic differences in the formulas for determining P_d , and (3) differing methods for establishing $P_{d_{max}}$. The acceleration values utilized in the agency formulas are presented in Table V.9(c). The accelerations are essentially the same for GL, NK, and IACS, with the only differences occurring in the value for the longitudinal acceleration, a_x. BV uses only the vertical acceleration, a_z, in their P_d formulas. In addition, no values of acceleration are required when using the graphical method of DnV. It is our understanding that these graphical results were generated by utilizing long-term a_x, a_y, and a_z accelerations for the tank and ship dimensions which are inputs to the

TABLE V.9. DYNAMIC INTERNAL PRESSURES FOR TANK 6 OF A 125,000 m^3 MEMBRANE TANK LNG SHIP

(a) Tank 6 Dimensions

Tank Length	ℓ	=	34 m
Tank Breadth	b	=	37 m
Tank Height	H	=	24 m
Tank CG from ⊈	X	=	56 m aft of midships
Cargo Specific Gravity	γ	=	.447

(b) Maximum Dynamic Pressures per Agencies' Methods

Agency	$P_{d_{max}}$ (m H_2O)	Method of Determination	Location of $P_{d_{max}}$	Tank Application
BV	8.87	Formulas which include a_z, pitch and roll amplitudes and ship and tank geometry	Corners of Aft Wall	Integral and Independent
DnV	5.92/ 6.97	Graphical Method based on elliptical combinations of a_x, a_y, a_z, and g. Graphs include parameters representing tank and ship geometry but not accelerations	5.92 on entire top, 6.97 on entire bottom. Linear variation in between	Independent
GL	8.22	Maximum value of: $\gamma a_x \ell = 2.52$; $\gamma a_y b = \underline{8.22}$; $\gamma a_z H = 5.90$	Entire longitudinal walls	All Tank Types
NK	10.74	$\gamma\sqrt{(a_x\ell)^2+(a_y b)^2+(a_z H)^2}$	At the tank corners	All Tank Types
IACS	5.90	Acceleration ellipse which combines a_x, a_y, a_z, and g elliptically. $P_{d_{max}} = \gamma(z\, a_\beta)_{max}$; $a_\beta = g + (a_x + a_y + a_z)$; z = largest liquid height in the a_β direction.	At the tank corners (not necessarily the largest P_d) only a_z was considered.	All Tank Types

(c) Tank 6 Accelerations Used to Determine $P_{d_{max}}$

Agency	a_x	a_y	a_z
GL	$\pm$.166	$\pm$.497	$\pm$.55
NKK	$\pm$.240	$\pm$.497	$\pm$.55
IACS	$\pm$.217	$\pm$.497	$\pm$.55
BV	----	----	$\pm$.255

graphical method. Also, the graphs were generated by combining the accelerations a_x, a_y, a_z, and g elliptically, as is done in the IACS rules. The DnV acceleration values are similar to those indicated in Table V.9(c) for the other agencies. Therefore, the differences in the maximum dynamic pressures exhibited in Table V.9(b) result from the methods for establishing the location and the magnitude of maximum dynamic pressure rather than from differences in the agency acceleration values. For example, BV's formulas are quite different from the other agencies and include both pitch and roll amplitude which the other agencies exclude. The maximum dynamic pressure from the BV formulas (8.87 m H_2O) results from considering the accelerations and tank dimensions at the corners of the aft tank wall. The graphical DnV method provides two pressure values for the maximum dynamic pressure: 5.92 and 6.97 m H_2O on the entire top and bottom, respectively. A linear variation between top and bottom is assumed for the side and end walls. GL uses the maximum value of the individual coordinate dynamic accelerations times their appropriate tank lengths as their criteria for $P_{d_{max}}$. For Tank 6 of the example ship, this is 8.22 m H_2O acting on the entire longitudinal walls. On the other hand, NK's maximum dynamic pressure results from combining all three coordinate P_d's. The resulting value of 10.74 m H_2O is the highest of any of the agencies and represents the greatest degree of conservatism. IACS combines a_x, a_y, a_z, and g elliptically. The details of this procedure are presented in Appendix A.2.12. The maximum (dynamic + static) IACS pressure is determined as follows:

$$P_{d_{max}} = \gamma \, (z \, a_\beta)_{max} \tag{11}$$

where z is the largest liquid height above the point considered in the a_β direction. Therefore, for each tank design, a trial and error procedure is required to determine the maximum value of the product ($z\,a_\beta$). The value of 5.90 meters of water indicated in Table V.9 for IACS is the largest dynamic pressure when assuming that only a_z accelerations were present on the tank. If the a_y and a_x accelerations were included, it is possible that the resulting $P_{d_{max}}$ would be larger.

Since g is included in the definition of the acceleration a_β, the pressures calculated by IACS include both dynamic and static contributions to the internal tank loading. For the points under consideration, the hydrostatic pressure has to be subtracted from the IACS $P_{d_{max}}$ to determine the dynamic contributions. GL and NK treat the three coordinate accelerations as acting independently of the gravitational acceleration and, therefore, the dynamic internal pressures given by GL and NK are determined without using g. If the gravitational acceleration is eliminated from the definition of a_β in the IACS rules, the resulting dynamic pressure for any point in the tank would be different than that obtained by including g and then subtracting out the hydrostatic head. Since the ship coordinate accelerations occur simultaneously with g, the IACS method is realistic. Also, the combining of a_x, a_y, a_z and g elliptically, is more realistic since the probability of the maximum values of the long-term accelerations a_x, a_y, and a_z occurring simultaneously, as is assumed in the NK rules, is unlikely. It should be noted that dynamic pressures determined by the IACS method exhibit only small differences when compared to those obtained from NK or GL. The variations shown on Table V.9(b) result from the methods of determining $P_{d_{max}}$. The major difficulty associated with using the IACS criteria is that it requires a trial and error procedure to determine the maximum internal dynamic pressure, whereas the other formulas are much simpler and straightforward.

The determination of a reasonable design dynamic internal pressure requires an accurate knowledge of the appropriate tank accelerations and a realistic procedure for applying these accelerations to determine $P_{d_{max}}$. Therefore, the validity of the dynamic internal pressure criteria can only be established after the validity of the long-term acceleration values, used in calculating the dynamic internal pressure, has been substantiated. However, since the results in Chapter III indicate that the societies' acceleration formulas are usually conservative, the current dynamic pressures should also be conservative. The results on Table V.9 also reveal that regardless of the technique utilized to determine the maximum dynamic pressure, the same order of magnitude for these pressure results.

Updated Dynamic Internal Pressure Criteria

New P_d criteria should include substantiated acceleration values in formulas which give the resulting internal dynamic pressure distribution on the tank walls. The validity of these criteria and the degree of conservatism associated with their use will require comparisons with both full-scale dynamic pressure and acceleration measurements on LNG ship tanks. The following steps should be included for internal dynamic pressure criteria.

- Establish accurate long-term acceleration values a_x, a_y, and a_z.
- Combine these acceleration values elliptically and combine with the gravitational acceleration to determine the resulting (dynamic + static) pressures on the tank walls using some rational approach such as the IACS method.
- Utilize the maximum value of dynamic pressure for establishing tank structural requirements.

Since the utilization of the IACS method requires a trial and error technique which must be programmed for a specific tank or ship. It is likely that this procedure could be simplified by developing mathematical expressions, tables, or graphs which would result in equivalent values of dynamic pressure. As part of an update of dynamic internal pressure criteria, the IACS method should be utilized for a wide range of typical accelerations and ship tank geometries. These data should then be summarized in chart or simple equation form to provide a more easily used method for determining dynamic internal pressures. This can be accomplished only after the accelerations a_x, a_y, and a_z are substantiated.

V.2.13 Dynamic External Pressures on Hull

Dynamic external pressures on a ship's hull result from several sources. These include: (1) time-varying hydrostatic pressure due to ship motion, (2) slamming pressures due to bow emergence, (3) wave slap near the ship flare, and (4) deck pressures due to shipping of green water. The prediction of variations in hydrostatic pressures created by ship motions is straightforward, and their magnitudes are low when compared to the impulsive pressures which result from slamming and wave slap.

The external hull pressures do not act directly on the tank, so the effect of these pressures on tank design is difficult to establish. This difficulty is compounded for LNG ships because of the wide range of tank designs and tank orientations relative to the ship's hull. The design of an LNG tank to withstand the loads created by external hull dynamic pressures requires a knowledge of (1) the distribution and magnitude of the time-varying pressures on the hull, and (2) how these loads are transferred through the hull structure to the external tank walls and support systems.

The accurate prediction of slamming pressures on any type of ship is difficult, and sufficient full-scale data are not available to substantiate existing analytical techniques. [32] Since no full-scale slamming measurements on LNG ships exist, establishing specific criteria for designing tanks for this type of loading is not presently possible. The classification societies, for the most part, do not provide specific formulas to account for dynamic external pressures on the hull in the design of LNG tanks. DnV, NK, and GL provide formulas which predict a pressure distribution on the hull but provide no specific methods for utilizing these pressures for tank design.

All of the classification societies' rules reflect the need for considering dynamic external pressures as part of the tank design.. Both BV and ABS give no specific formulas but do indicate that ship slamming loads are to be included in designing LNG cargo tanks. In addition, DnV indicates that heading angles should be changed and speed reduced in heavy weather to reduce the effects of slamming. Also, DnV states that tanks be given special design considerations if supported in such a way that the deflection of the hull transfers significant stresses to the tank. GL specifies that external loads for the ship bottom and the side shall be determined by their computer codes. DnV, GL and NKK all require that the loads due to external hull pressures be calculated from long-term distribution of ship motions in a seaway. ABS, LR, USCG, and IACS have no specific requirements concerning dynamic external hull pressures.

Comparison of Agencies' Formulas

In order to evaluate the dynamic external pressures as predicted by the formulas given by NK, GL, and DnV, a current 125,000 m^3 membrane tank ship was used. The agency formulas (as given in Appendix A) provide a longitudinal pressure variation as a function of ship dimensions, ship speed, block coefficient, and ship draught. Pressure distributions can be determined at any longitudinal position from ship bottom to ship deck. Figure V.6 shows the results of utilizing the three agencies' formulas for the 125,000 m^3 LNG ship. The resulting dynamic external hull pressures are plotted versus longitudinal position on the ship. Also shown are the locations of the fore point of the No. 1 tank and the aft point of the No. 6 tank. The results reveal a significant variation in pressures when comparing the three agencies' results. In all cases, the highest pressures occur at the fore perpendicular. Predicted pressures at this point range between 8 and 40 meters of water. It is interesting to note that both DnV and NK predict the highest pressures at the water line and the lowest pressures on the ship deck, while the GL formulas predict the highest pressures on the ship bottom. In general, the NK values are the highest. The values presented in Figure V.6 are of relatively low magnitude when compared with slamming pressures.

The static design values shown on Figure V.6 may be sufficient to provide a hull design that will not transfer impulsive slamming and other types of external hull pressures to the tank support structure in sufficient magnitude to cause damage. An evaluation of this would require an accurate knowledge of the external hull pressure-time and spatial histories and a method of determining the tank wall/support/hull system responses to these loads. It should be noted that peak slamming pressures can reach as high as 175.0 m H_2O on the ship's bow, [22,32] and the ship's flare can experience pressures of ~15.0 m H_2O from wave slapping. Deck pressures that result from shipping green water are on the order of 7.0 m H_2O.

Many factors affect the magnitude, duration and the probability of slamming pressures on an LNG ship. Even though an accurate prediction of these values is difficult, previous ship slamming studies provide valuable

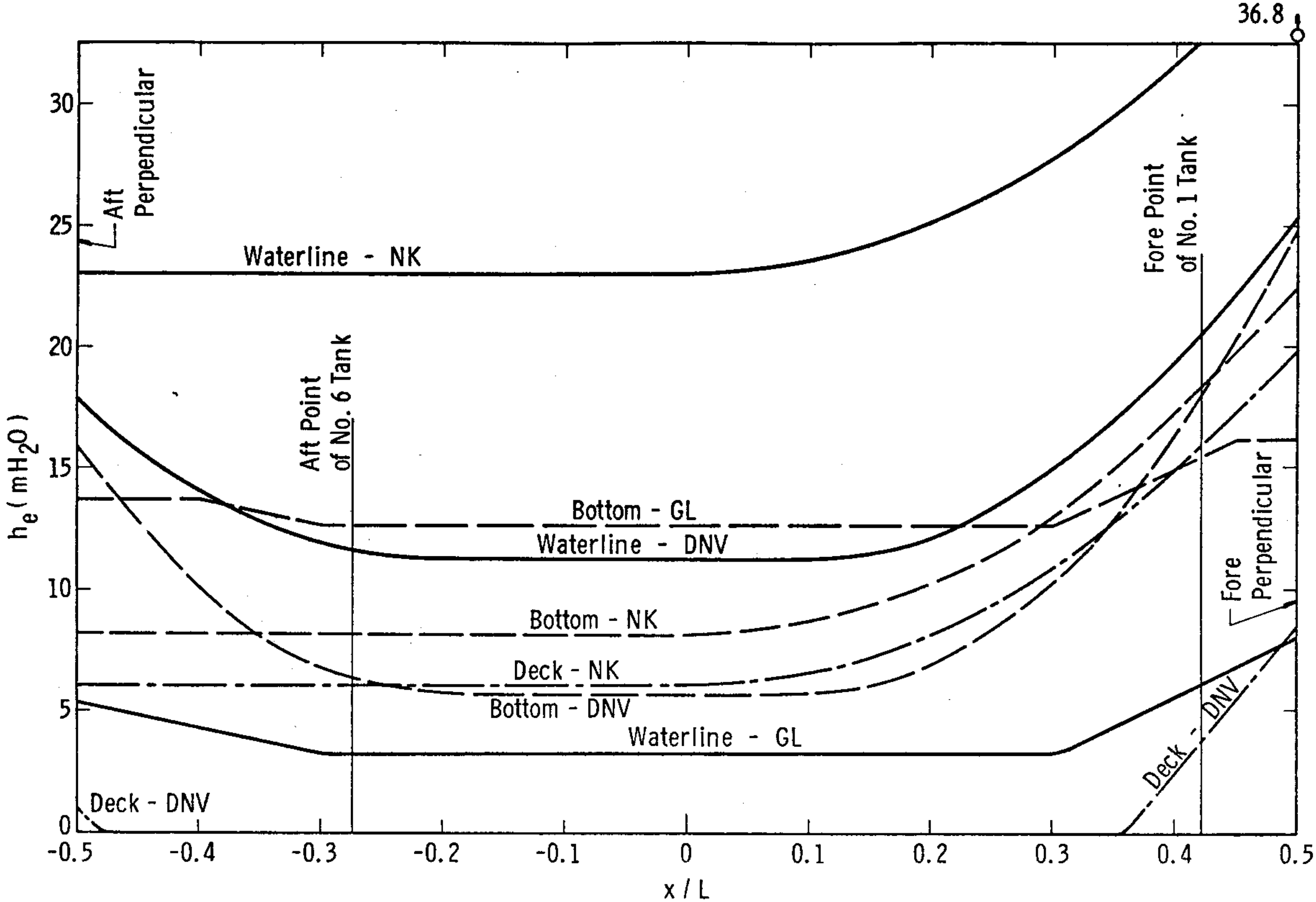


FIGURE V.6. DYNAMIC EXTERNAL HULL PRESSURES FOR information that can be utilized to establish guidelines for designing LNG cargo tanks. [22,32] These include:

- The probability of occurrence of slamming increases significantly with increasing severity of the sea.

- The probability of occurrence of slamming decreases significantly with increasing ship draught and course angle.

- The probability of occurrence of impact at the ship's forward portion is much higher than at aft locations, with the greatest impacts occurring 0.2 L aft of the forward perpendicular.

- The probability of impact at all locations increases with increasing ship speed.

- Slamming is always accompanied by an impact pressure on the ship's bottom. The impact pressure is of an impulsive type, and its duration is extremely short, on the order of 0.1 second.

- Intermediate wave lengths relative to the ship's length and large wave heights are the most conducive to ship slamming.

Since slamming and shipping of green water pressures occur at the fore point of the ship, the likelihood of tank damage is greater for the No. 1 and No. 2 tanks. While the magnitude of slamming pressures is high, their duration is relatively short. Therefore, an equivalent static design pressure will be much lower.

Updated External Hull Pressure Criteria

Updated slamming pressure criteria will require:

(1) A determination of time and spatial variations of dynamic external hull pressures and their probability of occurrence for each LNG ship design, considering ship geometry, speed, course and loading conditions.

(2) Utilizing the predicted maximum external hull pressures to determine the hull/tank support response and establish scantlings to ensure acceptable stress levels in the tank walls and support elements.

Since both requirements 1 and 2 will necessitate a unique set of calculations for each LNG ship and tank design, specific formulas are difficult to establish for this tank load. However, calculations of external hull pressures for typical ranges of LNG ship geometries, speeds, courses and loading conditions would establish the range of magnitudes and probabilities of these pressures. Full-scale or model external hull pressure data could then be used to substantiate the calculations. Design graphs or formulas could be produced to provide equivalent external hull pressures for various LNG ships and operating conditions. The design of the tanks and support structures would utilize the equivalent external hull pressures from these design graphs or formulas. In the event that future tank/support/hull structures reduce to a few basic designs, then additional design graphs and formulas for determining the tank loads that result from the equivalent external hull pressures could be produced. At present, the complexity of establishing reasonable tank design loads, in this category, for the large number of LNG tank designs, implies that each ship/tank design should be considered on an individual basis with requirements 1 and 2 stipulated. Most of the classification societies generally follow this approach. Future experience with LNG ships in combination with analysis and model and full-scale test results would then allow the appropriate design graphs, formulas and charts to be produced.

V.2.14 Sloshing Pressures

In general, criteria for slosh-induced tank loads are not specific, and only NK and DnV provide formulas for determining these loads. The other classification societies usually state that partial filling is to be avoided. In the event partial filling must be utilized, the rules indicate that special measures are to be taken to avoid the risk of resonance and to ensure that the tank withstands the slosh-induced dynamic pressures. With the exception of NK and DnV, no specific methods are given.

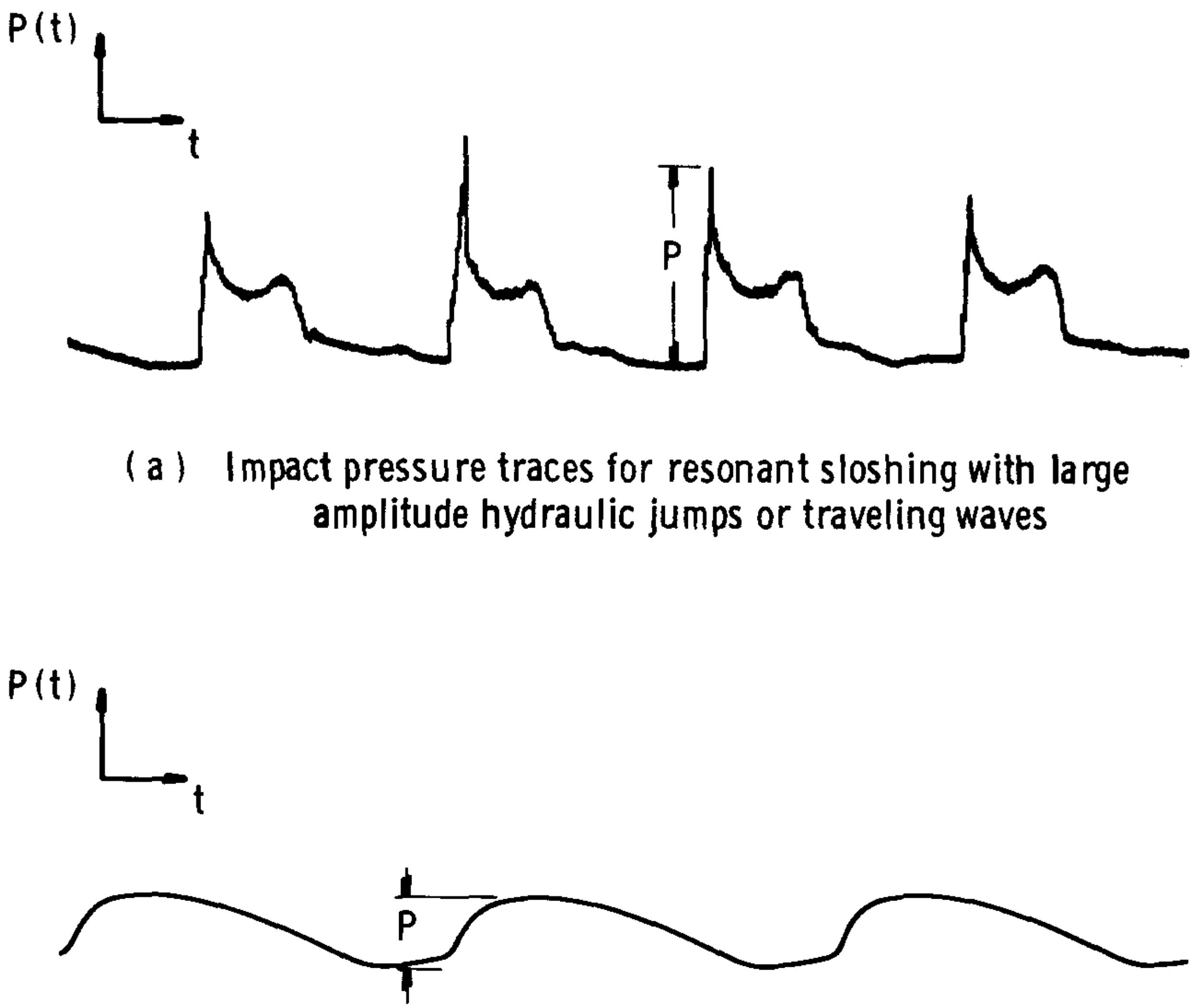
Sloshing Phenomena

The determination of dynamic loads which result from sloshing of liquids in partially filled tanks has been studied extensively in recent years for the space program. [33] The results of these studies are not directly applicable to sloshing problems associated with ship cargo tanks since emphasis was placed on frequencies and total forces as they related to control system requirements for space applications. In addition, the slosh-induced loads in rocket fuel tanks result from low-amplitude excitations which are small when compared to typical ship motions. The sloshing phenomena in ship cargo tanks result from large amplitude, nonlinear sloshing behavior which has not heretofore been studied extensively and which is not amenable to theoretical analysis.

With the advent of super tankers, the concerns about the consequences of liquid sloshing have increased because the probability of resonant sloshing is higher with the larger ships. As a result, the transport of liquid cargos in partially filled tanks is prohibited for many of these ships. However, in the case of LNG ships, partially filled conditions are needed because (1) chilled-down liquid is required to maintain cold tanks on return trips, (2) higher specific gravity liquids than LNG are transported in tanks designed for LNG, (3) partial unloading is desirable when multi-port stops are made, and (4) loading or unloading at sea creates significant time periods at undesirable fill depths. Conditions (3) and (4) apply to all types of liquid cargo ships. Therefore, the ship tank designer must be able to accurately predict the resulting slosh loads to ensure an adequate structural design. In the case of LNG carriers with membrane tanks, special considerations must be given to sloshing loads, as these tanks are more susceptible to local damage from such loadings than are conventional tank structures.

In general, sloshing is affected by liquid fill depth, tank geometry, and tank motion (amplitude and frequency). The liquid motion inside a ship tank has an infinite number of natural periods, but the lowest mode is the most likely to be excited by the motions of the ship. The sloshing phenomena in cargo tanks that are basically rectangular in shape can usually be described by considering only two-dimensional fluid flow, while in spherical or cylindrical tanks, three-dimensional flow effects are present. The sloshing phenomena in basically rectangular tanks are divided into two classes: low and high liquid fill depths. The low fill depth case is represented by $h/\ell < 0.2$, and is characterized by the formation of hydraulic jumps and traveling waves for excitation periods around resonance. At higher fill depths, large standing waves are usually formed in the resonant frequency range. When hydraulic jumps or traveling waves are present, extremely high impact pressures (typical of those present in ship slamming) can occur on the tank walls. Figure V.7(a) shows typical pressure traces from model tests[34] recorded under this sloshing condition. It is noted that the pressure pulses are neither harmonic nor periodic since the magnitude and duration of the pressure peaks vary from cycle to cycle even though these traces were obtained with harmonic oscillation. Figure V.7(b) is representative of typical pressure traces that result when standing waves are present at higher fill depths or when non-resonant, small-amplitude sloshing occurs at any fill depth. Impact pressures typical of those shown in Figure V.7(a) can also occur on the tank top when the tanks are filled at the higher fill depths. These types of pressures can cause local structural damage and should be considered the most important.

Three-dimensional flow occurs in spherical or cylindrical tanks, and the types of pressures exhibited in Fig. V.7 can also occur on these tanks structures. The most important loads on these tanks are the total forces



(a) Impact pressure traces for resonant sloshing with large amplitude hydraulic jumps or traveling waves

(b) Pressure trace for non-resonant or low amplitude standing wave sloshing

FIGURE V.7. TYPICAL PRESSURE WAVEFORMS ON TANK WALLS WITH SLOSHING LIQUIDS or moments on the tank walls which determine tank support structure requirements. Spherical or cylindrical tank walls are usually thick enough that local impact pressures are not a problem. This is not the case for membrane tanks. For either two- or three-dimensional resonant sloshing, the prediction of forces or pressures with large excitation amplitudes is extremely difficult, and experimental data obtained with scale model tanks are usually used to establish these loads.

Model Test Data

A large number of model tests have been conducted for investigating sloshing in liquid cargo tanks.[34-40] Nearly all model tests have considered the six degrees of ship motion individually and investigated sloshing by varying tank amplitude and frequency harmonically, usually in heave, surge, pitch, or roll. A considerable number of tests have been conducted on scale models of LNG cargo tanks to obtain design information as well as to ascertain the factors responsible for recorded tank damages. In most studies, the scaling of impact load data to full-scale has considered only Froude scaling and thus eliminated any possible scaling effects of fluid properties such as viscosity, compressibility, or vapor pressure (cavitation). Under these assumptions, pressures scale by:

$$\left(\frac{P}{\rho g \ell}\right)_p = \left(\frac{P}{\rho g \ell}\right)_m \tag{12}$$

where the subscripts m and p are for the model and prototype, respectively. The frequencies (ω) between prototype and model are given by

$$\left(\frac{\ell \omega^2}{g}\right)_p = \left(\frac{\ell \omega^2}{g}\right)_m \tag{13}$$

In scaling pressure data, a pressure coefficient is defined as follows:

$$K_P = \frac{P}{\rho g \ell \phi} \tag{14}$$

where ϕ is the pitch, roll or yaw angle. For translation, $\ell\phi$ is usually replaced by the translational amplitude. Typical values for this pressure coefficient at resonance are shown in Figure V.8 versus fill level for a 1/52-scale model of a prismatic LNG cargo tank.[41] The results indicate that the pressure coefficient at resonance reaches a maximum at a low fill depth of $0.2 < h/H < 0.3$. In this fill depth range, the slosh phenomenon changes from a hydraulic jump to a longitudinal traveling wave, and the recorded pressures are typical of the impulsive type impact pressures shown in

Figure V. 7(a). Subsequent model tests [34] have shown that K_P can approach 20 in this region. Similar pressure levels are also recorded when liquid impacts on the tank top at higher fill depths. It should be noted that these pressure coefficients are based on a statistical evaluation of many cycles of test data. On certain individual cycles the pressures can be two to five times larger than those indicated in Figure V. 8. Additional efforts have been undertaken to establish worst-case pressure magnitudes over long periods of operation. [35]

The slosh studies performed with model tanks of LNG ships [34-40] include both prismatic and spherical tanks, and the results indicate that the measured forces and loads on the tank structure are greatly affected by tank fill depths, geometry, and excitation amplitude and frequency. In the case of the spherical tank studies, [36] the loads on the internal components, such as towers, were also investigated.

Some preliminary studies have been conducted to determine the effects of fluid properties on scaling model slosh loads data to full scale. [34,35,41] The results in Reference 41 indicate that liquid viscosity, compressibility, and cavitation, in combination with Froude scaling, may be important in scaling slosh forces and pressures. Including these fluid properties with the additional stipulation of geometric similitude between model and prototype yields

$$\frac{P}{\rho g \ell \phi} = F \left[\frac{\rho g \ell}{\mu} , \frac{\rho g \ell}{E_L} , \frac{\Delta P}{\rho g \ell} \right] \tag{15}$$

where the first, second and third dimensionless groups in the bracket include viscous, compressibility and cavitation effects combined with Froude scaling, respectively. All previous model studies have taken the function F to be a constant (i. e., K_P), and no allowance for fluid effects was considered. Depending on the liquid cargo, some of these fluid properties could be important. For example, LNG is transported with a tank pressure slightly above its vapor pressure, implying that cavitation effects could be important. Also, LNG has an extremely low viscosity, and less damping would be present in an LNG tank than in other types of liquid cargo carriers. Experimental programs are currently under way [34,35] to establish the effects of these fluid properties on the scaling of model data. At present, these studies are not complete, and, therefore, the appropriate scaling considerations are not established. However, the test results to date indicate that fluid properties will probably have a minor effect on scaling impact pressures when large amplitude sloshing, typical of a ship cargo tank, is present. The variations in peak impact pressure magnitudes on each successive slosh cycle are much larger than the magnification of pressures

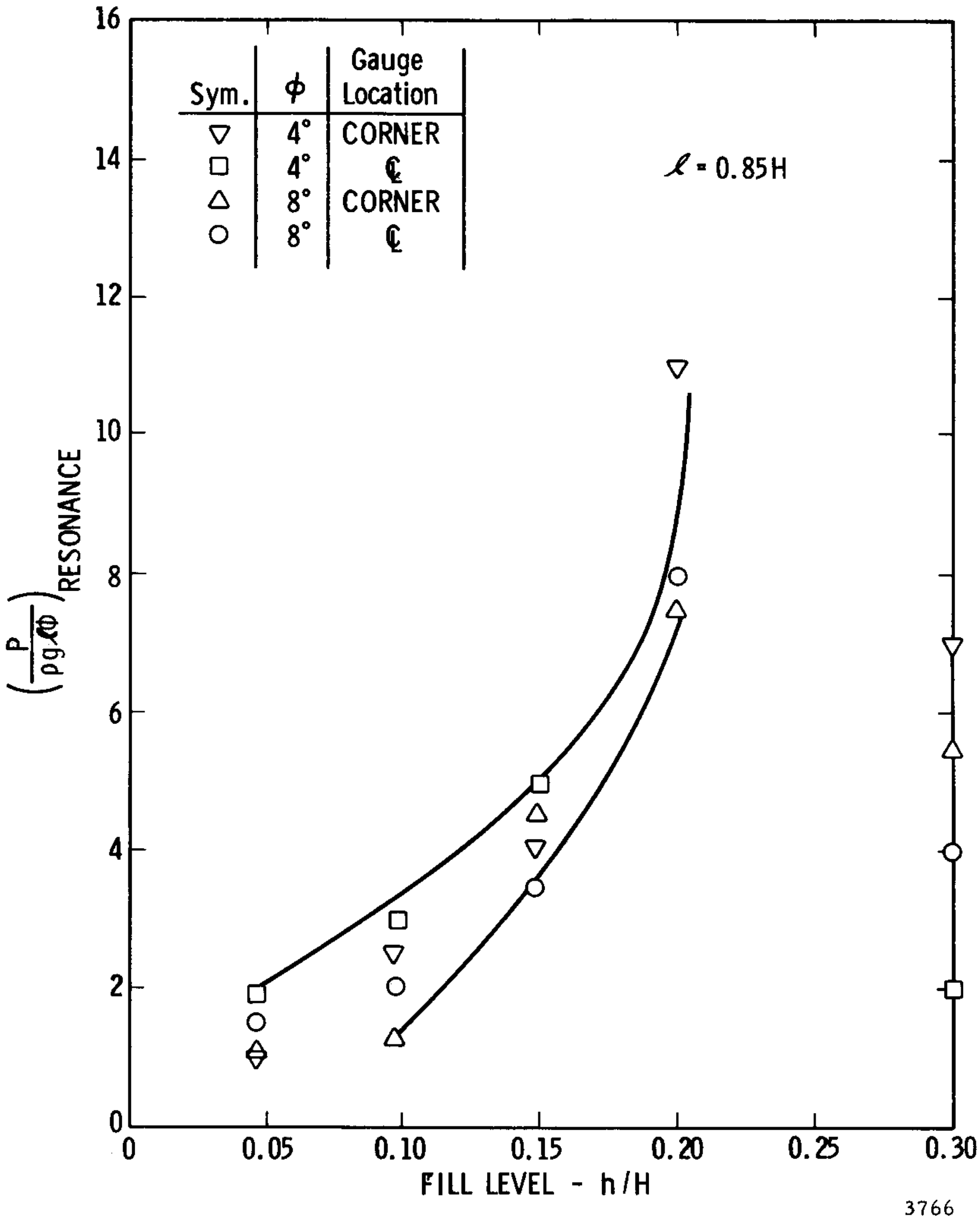


FIGURE V.8. PRESSURE COEFFICIENT K_P AT RESONANCE VS. FILL LEVEL MODEL TEST IN PITCH (Ref. 41) that would result when including scaling effects for liquid properties. Therefore, the present use of only Froude scaling (Eq. (12)) should provide reasonable estimates of full-scale loads. The results of the current scaling effects programs will provide even more accurate scaling expressions.

Full-Scale Data

There are no full-scale slosh data available for LNG cargo ships. However, slosh pressure measurements on an OBO carrier tank were presented in Reference 35. In this program, a number of pressure gages were installed at different locations in an OBO tank carrying water ballast, and simultaneous recordings of pressures and ship roll motions were taken with different filling heights during a voyage from Japan to the Persian Gulf. As part of a subsequent model test program, recorded roll motions of the ship were imposed on a model tank, scale 1/30, and the pressures at corresponding locations measured. Extremely high impact pressures were recorded at the underside of the top wing tank. Examples of model and test data are given in Table V.10, which shows that the magnitudes and pressure peaks are quite similarly distributed. Tabulation is the percentage of all peaks in a sample that lie within different pressure intervals. The pressures obtained in the model were scaled to full scale using Froude scaling,

$$\frac{P}{\rho g \ell \phi} = K_P \tag{16}$$

TABLE V.10. COMPARISON OF MODEL TO PROTOTYPE OBO IMPACT PRESSURES (Ref. 35)

PRESSURE kgf/cm^2	Percent of Peaks in Pressure Range		TEST CONDITION
	MODEL	PROTOTYPE	
0 - 6	83.3	96.0	h/b = 0.215
6 - 12	13.6	10.0	Random rolling
12 - 18	3.1	2.5	(Max. roll angle 7.4 deg.)
18 - 25		1.5	RMS of roll angle 2.9 deg.) $T_R/T_o \approx 1.0$

It should be noted that both the model and the prototype liquids were water. and, therefore, the question of modeling full-scale liquids, such as LNG with water, and then accounting for the effects of fluid properties on scaling criteria, has not been established.

Slosh Loads Using Agencies' Formulas

In order to evaluate the sloshing criteria as given by DnV and NK, sloshing loads in Tank 6 of a 125,000 m^3 membrane tank ship were determined for comparison. In addition, worst case impact pressure resulting from resonant sloshing were predicted for this tank utilizing model test results taken from previous studies with a similar tank design. [41] The results are shown in Figure V.9. It is noted that the design pressures as determined from NK and DnV cover only a certain range of fill depths. DnV specifically states that their rules apply only for fill depths between 20 and 90 percent. They have no specific rules below the 20 percent fill depth, and above 90 percent the criteria for internal dynamic pressures apply. In addition, DnV calculates the slosh-induced loads at 70 percent fill depth and assumes that these loads will not be exceeded between 20 and 70 percent fill depth. In addition, DnV calculations are to be used only when $T_p/T_x \geq 1.24$ and $T_r/T_y \geq 1.4$, where T_p and T_r are resonant ship pitch and roll periods, respectively, and T_x and T_y are resonant slosh periods in the longitudinal and transverse directions, respectively, and determined at $h/H = 0.7$. The DnV formulas give different slosh pressures at the top and bottom of the tank which act at the mid-wall location (h_{ix}, h_{iy}). The sloshing pressure at the corners (h_i) is the vectorial sum of the mid-wall pressures, and linear variations are assumed elsewhere. The highest DnV predicted slosh-induced pressures occur at the top corners of the tank. Model test studies have shown that the highest slosh-induced loads occur near the static liquid fill level.

NK stipulates that their sloshing formulas are valid only below a fill depth of 70 percent. Their formulas combine the side and end-wall tank loads to determine a tank corner load identical to the procedure of DnV. Their formulas also provide for a calculation of the longitudinal and transverse resonant slosh periods as a function of fill depth. The resulting dynamic slosh pressures are therefore a function of fill depth and act at the static liquid level. NK stipulates that if the longitudinal resonant sloshing period equals the ship roll period, then the formulas are no longer valid. For the No. 6 tank of the example ship, this occurs at a fill depth of 23 percent as shown in Figure V.9.

The magnitude of the dynamic slosh pressures as determined from NK and DnV are in good agreement, as both range between 5 and 13 m H_2O. The stipulation by both agencies that the formulas are not to be utilized at or near resonant sloshing is the primary reason why the magnitudes of the

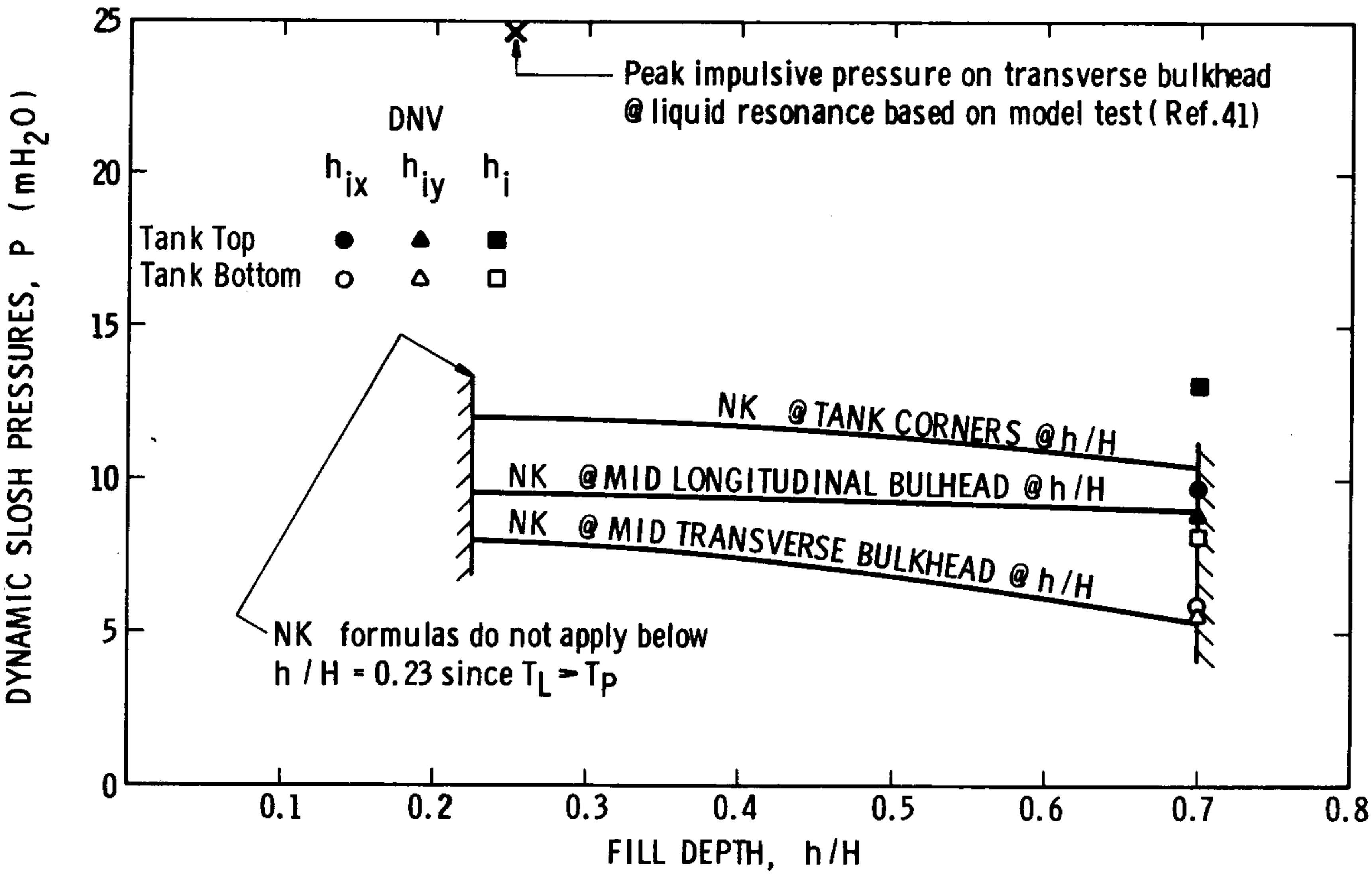


FIGURE V. 9. COMPARISON OF DYNAMIC SLOSH PRESSURES FOR TANK NUMBER 6 OF A 125,000 m^3 LNG MEMBRANE TANK SHIP pressures are much lower than the 25 meters of water predicted from scale model tests with resonant sloshing in pitch at h/H = 0.25. This value was obtained assuming K_p = 10. As previously indicated, on some individual cycles the pressure coefficient can be much larger than 10. At resonance, the agencies' formulas appear non-conservative with respect to predicting peak dynamic slosh pressures. The peak slosh pressures are of an impulsive nature with the pressure peak occurring over a short duration when compared to the slosh period. Therefore, the static design values as predicted by either NK or DnV may be adequate to withstand the peak impulsive pressures. An evaluation of this, of course, will depend on a complete knowledge of the tank structural specifications and the expected pressure-time histories of the impulsive slosh pressures.

Based on test results to date, no substantiated criteria to handle slosh loads can be established. Substantiated criteria await full-scale slosh load data. However, knowledge of the basic sloshing phenomena and the factors which affect liquid sloshing, as well as the results of model tests, does provide information on which to develop new tank design criteria to account for slosh-induced loads.

Recommended Sloshing Pressure Criteria

In general, the designer must consider the effects of tank acceleration, liquid fill depth, and tank geometry when determining the expected sloshing loads.

Tank Acceleration and Fill Depth

Resonant sloshing creates the largest impact loads in liquid cargo tanks. Studies to date have indicated that model test results as well as calculations can be successfully used to estimate the lowest frequency resonant condition, where resonant sloshing is likely to occur. The question arises, can this mode be excited by the ship's motion? The tank designer must evaluate the probability of the ship's motion exciting the tank to resonance at some time in the ship's life. This consideration should take into account that the resonant frequencies of sloshing will vary with fill depth and tank geometry. In view of the possibility of a tank being excited into resonant sloshing, the designer must consider that the mode of ship motion is a time-dependent function of the sea state as well as the ship's loading conditions, speed, and course. Considering that each of the modes of ship motion is a random variable represented by a spectrum composed of contributions over a wide range of frequencies, the possibility exists for ex-

The designer has to accept that the tank will be subjected to the ship's motions and accelerations, and the design problem cannot be limited to a determination of tank geometry and fill depths where resonance will not occur. Therefore, the determination of loads should follow a probabilistic procedure involving the response at different conditions and their probability of occurrence. However, a good estimate of the possibility of resonant sloshing can be achieved through consideration of the natural slosh frequency and the pertinent motion spectra. The designer must understand the types of sloshing phenomena and loads that can exist at different fill depths for the basic tank geometries utilized in ships. In estimating the most severe loads which result from resonant sloshing, the designer will probably have to turn to model test data for setting prototype structural requirements. Therefore, the designer must also understand the scaling implications when using model test data. The designer will need to know local impact pressures on tank walls when considering the design of local scantlings while the total sloshing forces will be needed to design the tank supporting structures.

Tank Geometry

The overall geometry of a tank will be selected primarily on criteria other than the prevention of liquid slosh. However, it may be advantageous to adjust tank shape and dimensions to improve tank characteristics with respect to slosh prevention, at least in some range of fill depths. This will normally mean deciding what fill depths are to be carried in the tanks and then adjusting tank width-to-length ratios such that the tanks are compatible with ship geometry while reducing the probability of resonant sloshing.

Structural Response to Impact

The fact that extremely high impulsive type impact pressures can occur during resonant sloshing will, in general, require the designer to investigate the structural response to such loads. This will require taking into account the duration of the impact pressures and the correlation between magnitude and duration of these pressures. The natural frequencies of the structure subjected to impact will be of great importance in these calculations as well as any significant damping parameters. Since the general trend is that natural frequencies of many parts of ship structures will decrease with increasing ship sizes, this aspect should require increased consideration in future tank designs.

Updated Slosh Load Criteria

In general, an updated sloshing load criteria should include the following requirements:

- For the basic tank geometry, establish resonant slosh periods versus fill depth using either theoretical analysis or model test results.

- Determine ship motions (amplitudes and frequencies) and compare to resonant slosh periods.

- Adjust tank dimensions to reduce probability of resonant slosh at desired fill levels.

- For worst-case sloshing conditions, establish pressures and forces exerted on the tank wall utilizing either model tests with appropriate scaling criteria or available theoretical analysis.

- For the anticipated pressures and forces, investigate structural response to dynamic impulsive pressures and set final tank wall requirements. Determine tank support requirements based on total forces, and establish the magnitude and phase of the slosh loads relative to the other dynamic tank loads.

- Based on above, set final tank design requirements.

A flow chart showing how these steps interrelate is given as Figure V. 10.

V. 2. 15 Vibrations

The following statement in the report of Committee 7 of the 5th International Ship Structures Congress[6] points out the significance of vibration in ship design.

> "In recent years some ships have shown damages which are difficult to explain from a structural point of view. The damages have occurred in transverse frames and horizontal girders in cargo tanks of large tankers and swash plate in aft peak tanks, in the form of cracks from cut-outs for longitudinals in web-plates, from the end of sniped-stiffners, near the toe of tripping brackets, etc. In several cases calculations have shown that natural frequencies of the damaged structural components have coincided,, or have been very close to, main excitation frequencies from the engine or propeller. This results in large vibration amplitudes, which are likely to cause cracks of the type mentioned. Measurements aboard several ships have verified this."

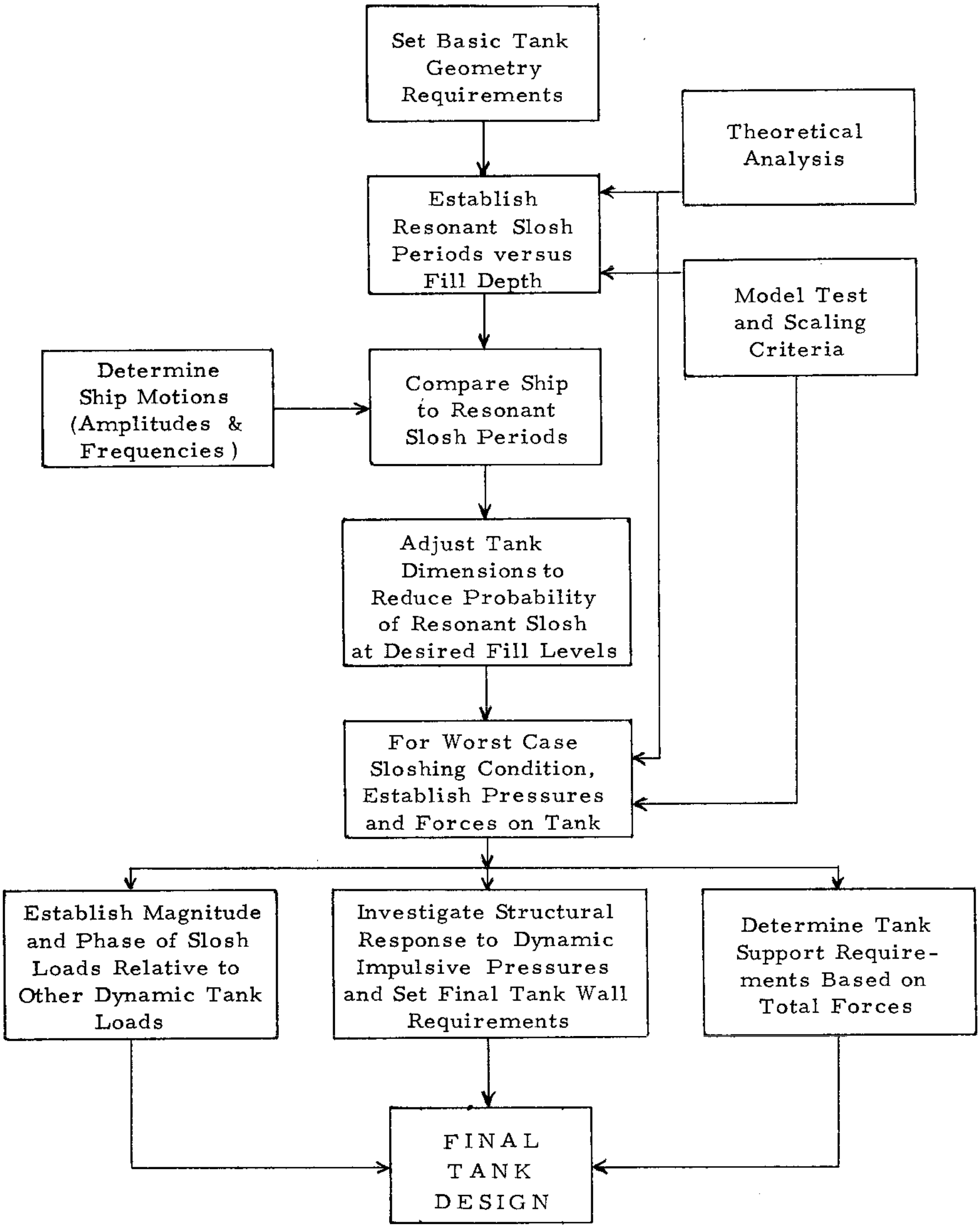


FIGURE V.10. LIQUID CARGO DESIGN GUIDELINES FOR CONSIDERING SLOSH-INDUCED TANK LOADS

This is true of all ship structures, including portions of LNG tanks. The committee also notes

> "Model tests of submerged, clamped uniform plates have shown that the damping is only 5-10% of the critical value in the range of 5-15 Hz. Resonance amplification factors greater than 10 are often found for submerged panel structures in practice."

In addition to local resonances, fundamental vibrations of the total tank structure can occur. Vinje [42] has shown that the fundamental mode of vibration for a completely filled Moss-Rosenberg aluminum spherical tank is in the excitation range of a four-bladed propeller rotating at 1.6 Hz, i.e., in the range of excitation of the propulsion equipment. Hence, for LNG tanks, whose primary objective is to maintain a liquid tight seal for containment of the LNG cargo, vibrations which give rise to fatigue cracking might cause serious problems.

The agencies' rules for tank vibrations are summarized in Table V.11. The regulations apply primarily to membrane-type tanks, implying that hull resonances are the main concern. Only DnV and NK require consideration of independent tanks, and only NK specifically mentions local vibrations of the tank structure. Neither GL, LR or IACS have any guidelines or regulations regarding tank vibrations. Also, there is no mention in the rules of any agency of the particular problems of hull springing and whipping which are associated with the fundamental resonance of hull vibration. While the frequency of this vibration (approximately 0.5 to 2 Hz) may be too low to excite local tank resonances, it can have a significant effect on membrane type tanks or other tanks rigidly attached to the hull in that cyclic strains will be introduced.

Whipping is the transient two node vibration of the hull which follows slamming, bow flare immersion and sometimes the shipping of green seas. Hoffman and Van Hooff, [43] Kumai, [44] and Van Gunsteren [45] have shown that springing is introduced by waves or wave harmonics that give resonance with this vibration period. Committee #2 of the Fifth International Ship Structures Congress [6] summarized the differences between springing and whipping as follows:

- "Springing is continuous, while whipping is transient."
- "Springing is mainly evidenced in relatively calm seas when the energy of the wave spectrum is contained in short waves and the resulting ship motions are almost zero. Whipping is observed in rough seas when the

TABLE V. 11. SUMMARY OF AGENCY RULES
FOR TANK VIBRATIONS

AGENCY	Require Consideration of Hull Vibrations on Tanks	Require Consideration of Local Vibrations on Tanks	Require Consideration of Propeller and Propulsion Equipment Excitation Forces	Require Model Testing	Require Resonance Testing	Require Prototype Testing
ABS	M(1)			M		
BV	M			M		
DnV			I, M			I(2), M(2)
GL						
LR						
NK		I	I, M, IT			
USCG				M(2)	M	M
IACS						

(1) M = Membrane tank
I = Independent tank
IT = Integral tank

(2) May be required.

ship motions include large relative bow motions and the momentum due to added mass has large variations."

Thus, stresses in the ship hull induced by springing most often occur in fairly calm seas, and the magnitudes of these stresses are generally regarded as being small relative to the wave-induced loads. Whipping, on the other hand, occurs in rough seas and measurements [46] have shown whipping stresses equal in magnitude to the wave-induced stresses for bow flare slamming on an aircraft carrier. Although we are aware of no reported measurements of whipping or springing stresses on LNG tankers, Lewis [47] shows that by geometrically increasing ship size, hull flexibility and hull stresses due to springing and whipping are likely to increase. Also, aside from geometric considerations, the energy in the sea spectra which excites the ship in its fundamental resonance mode increase with increasing ship length. These factors indicate that springing and whipping will probably be important considerations for LNG tankers.

Analytical techniques for treating springing and whipping in ships are rapidly developing. Goodman [48] has applied strip theory to springing in short waves and has shown good agreement with measured values on one ocean going ship. However, a major drawback to the application of strip theory is an inadequate description of the hydrodynamic damping. Hoffman and Van Hooff [43] have observed an unexplained increase in damping with increasing speed in still water. Once the hydrodynamic damping is properly defined, springing can perhaps be calculated routinely as for the slowing varying wave-induced loads. Kaplan and Sargent [49] have studied the whipping of ships by computer simulation. Because of nonlinearities associated with slamming and bow flare immersion, a time domain resolution was required. Excitation forces due to slamming were computed using linear relative motion characteristics determined from SCORES*. Calculations performed for the USS ESSEX aircraft carrier (whipping due to bow flare immersion) showed whipping stresses of greater magnitude than the slowly varying wave-induced stress in rough seas.

Updated Vibration Criteria

Rules for vibrations should address two main problem areas:

- The possibility of exciting overall and local resonances of the tank structure by all likely sources of excitation,
- The effects of cyclic strains which are induced in tanks due to local or overall hull vibrations.

The survey of the literature indicated that both overall tank resonances, as well as resonances of local substructure, are possible and that these vibrations can cause cracking which would not be predicted by normal stress analysis procedures. Also, as ship size and power continue to increase, as will apparently be the case for LNG ships, sources of excitation become greater and the problems may magnify. Also, it appears that springing and whipping may be significant in large LNG ships and should be considered from the standpoint of the cyclic strains which will be introduced, particularly in membrane, semimembrane and integral tanks.

V.2.16 Fatigue Loads

Because a liquid-tight barrier is essential for LNG tanks, some knowledge of the propensity of the primary and secondary barriers to develop fatigue cracks is essential. Whether the analysis approach is a classical fatigue life calculation, which includes the time for crack initiation and propagation, or is based only on the time for subcritical crack growth (fracture mechanics), an accurate definition of the time history of the loading is required. The approach now followed by many of the agencies is to require that the tank be designed for no through-cracking (safe life) but with consideration also given to crack propagation (fail safe). Loads, reflecting the operating history of the ship over its lifetime, must be defined for the fatigue life calculation.

The agencies' rules concerning fatigue loads are summarized in Table V.12. Two agencies, ABS and LR, have no regulations, and BV stipulates only that fatigue life calculations may be required if reduction of the secondary barrier is considered. USCG rules are limited to membrane tanks and provide no guidance for loads other than to require testing of tank models which includes cyclic loading. The other agencies, DnV, GL and NK, are more specific. Their rules suggest the use of a semi-log-linear spectrum for the load history. The linear distribution extends from one cycle of load at the maximum value (generally computed at a probability level of 10^{-8}) to zero load at 10^8 cycles. GL does not give a probability level for which the maximum loads are to be computed. Rather, they state that the cycles of load and, thus, the probability level should be based on expected service. IACS rules are similar to those of DnV and NK except that the cumulative load distribution is to be based on the calculations for the wave-induced loads rather than the semi-log-linear assumption. DnV, GL, NK and IACS also consider loading and unloading cycles (which we interpret as changes in the still-water loads) but provide no rules or guidelines for the combination of these loads with the wave-induced loads. In addition, no mention is made of the potential contribution to fatigue of the high frequency loads associated with springing, whipping or local vibrations.

Proper combination of the many types of random loading which occur during the ship's lifetime is a difficult problem and has not been solved satisfactorily. For the slowly varying wave-induced loads, the still-water loads

TABLE V.12 SUMMARY OF AGENCY RULES REGARDING FATIGUE LOADS

AGENCY	Tank Types For Which Fatigue Analysis Is Required	Wave-Induced Loads Required For Analysis	Assumption of Semi-Log-Linear Load Spectrum Is Allowed	Probability Level For Maximum Wave-Induced Load	Consideration Given To Loading And Unloading Cycles	Model Testing Required
ABS						
BV	I(1)					
DnV	I(2), M	O	O	10^{-8}	O	
GL	I, M, IT	O	O	O(3)	O	
LR						
NK	I(4), M(4), IT	O	O	10^{-8}	O	
USCG	M			10^{-8}		O
IACS	I(4), M	O		10^{-8}	O	O(5)

(1) I - Independent tank
M - Membrane tank
IT - Integral tank

(2) Required for Type A2 and Type B. Not required for Type A1.

(3) Probability level determined by ship service.

(4) Type B tanks only.

(5) Model testing may be required.

represent a shift in the mean and this superposition is straightforward. However, for the higher frequency dynamic loads associated with springing or whipping, the still-water plus the wave-induced loads comprise the mean. Even higher frequency loads, such as those associated with tank vibrations, are super-imposed upon a mean which is the sum of the dynamic, wave-induced and still-water loads.

Considering first only the still-water and wave-induced loads, superposition is easily accomplished. Nominal in-bound and out-bound loading conditions are probably sufficient for defining the mean loads. The cumulative distribution of the wave-induced loads is then subdivided in discrete loads using the procedures specified by the agencies (see DnV Figure 1 in Appendix A, Page A-105). For each of these discrete loads, one-half of the cycles should be applied to the in-bound loading condition and one-half to the out-bound loading condition. Using this approach, the effects of loading and unloading only represent shifts in the means and not independent load cycles as now considered by the agencies. Loading and unloading should be treated as separate load cycles only if there are stresses associated with transients such as might be produced by chilldown. Stationary thermal gradients should be treated as a componant of the mean loads.

High frequency loads such as those produced by whipping and springing are more difficult to treat. The most straightforward approach would be to use experimental strain data from similar ships to adjust (increase) the magnitude of the wave-induced loads to account for dynamic and vibratory load effects. Although straightforward, this procedure requires the accumulation of data from similar ships over a substantial time period. The strain data for both in-bound and out-bound voyages would be keyed to sea states or weather groups. Proper combinations of the records would then depend upon the estimated service history of the ship in question. To make the adjustment, the measured strains from operating ships would be filtered to eliminate all components except those produced by the slowly varying wave loads. Fatigue damage corresponding to the original composite records and to the filtered records would then be evaluated using fatigue gages [50] or analytical procedures such as the method of exceedances. [51] From these calculations, a scaling factor could be established which would result in equivalent fatigue damage for the wave-induced and composite loads. A single conservative factor could be established for a range of similar ships and routes or a separate factor could be established for each ship depending upon the expected service and, therefore, the record "mix"[7] used. Because other factors such as slight changes in hull form,

[7] Combination of records to reflect service time in each sea state or weather group.

ship size, etc., would probably be of equal or greater importance than the routing, a single factor, based on several LNG ships and routes, seems to be the most practical approach.

An analytical approach for combining the loads would be attractive and one can perhaps be developed based on a statistical combination of the still-water, wave-induced dynamic loads. A basis for the combination is the Central Limit Theorem [52] in probability, which states that "Sums of independent random variables tend to Gaussian distribution." SwRI has investigated this approach for application to fatigue testing of helicopter components. Multiple sine waves, whose frequencies were non-commensurate and whose phases bore no particular relationship, have been replaced by a single component of suitable amplitude and frequency. The amplitude of the single component was set equal to the rms value of the composite signal. Its frequency was determined from the condition that the variances of the signals were independent so that the sum of the variances was equal to the variances of the sum. Results thus computed were compared with fatigue damage computed by the method of exceedances, and good correlation was obtained. The approach was also demonstrated for the superposition of narrow band random signals. Because the wave-induced loads and the dynamic loads can probably be represented by superposition of a series of sine waves, such a procedure is perhaps viable. However, this process has not been verified experimentally and would require considerable development. Nevertheless, it poses an attractive alternative to an experimental approach if the loads associated with whipping and springing can be defined with the accuracy of the wave-induced loads.

Francis, et al., [53] suggest a procedure, closely allied to those discussed above, for determining a constant amplitude sinusoidal fatigue loading which is equivalent (in terms of cumulative fatigue damage) to a given random loading environment. The procedure was proposed for application to experimental data but could be used equally well with analytically derived loads if an effective frequency and rms value of the random composite[8] signal can be established.

The procedure rests on the following assumptions:

(1) The stress response of primary and secondary ship structure is approximately a narrow-band, stationary Gaussian process.

(2) The "S-N" curve is log-log linear and described by the equation $N^{\ b} = c$.

[8] Super-imposed wave-induced loads and dynamic loads.

A stepwise description of the procedure is repeated from Reference 53.

STEP 1 Determine the mean or constant component of the random load signal, and use this value as the mean level of the equivalent sinusoidal signal.

STEP 2 Determine the effective frequency, w_o, of the equivalent sinusoidal signal by using the average number of zero crossings with positive slope per second.

STEP 3 Determine the RMS stress level, σ_{rms}, of the random signal.

STEP 4 Determine the slope parameter b of the S-N curve from constant amplitude sinusoidal fatigue test data on the material system of interest. That is, plot log σ on the vertical axis (where σ is the peak stress per cycle) versus log N (cycles to failure); the slope of this line is $-1/b$. Note that the slope may be considered to be independent of the value of the stress ratio. If the required data are not available, use $b = 3.5$ for welded structure.

STEP 5 Calculate the stress equivalence factor k by the formula $k = \sqrt{2}\,[\Gamma\,(1 + b/2]^{1/b}$, where Γ is the gamma function. This equation is plotted for convenience in Figure V. 11.

STEP 6 Calculate the peak amplitude of the equivalent sinusoidal signal by the relation $\sigma_{max} = k\sigma_{rms}$

Though consideration of the mean and varying loads is necessary to represent truly the ship's load history, Gurney and Maddox [54] and Francis, et al., [43] have concluded that for typical welded ship structures the shift in the mean can be neglected. The reason is that the current knowledge of residual stresses in as-welded structures is inadequate and that these stresses locally can equal or exceed the stresses associated with the static or mean loads. Thus, while the mean loads must be accounted for to accurately represent the load history, their significance in fatigue calculation in a welded structure is uncertain. This uncertainty must be weighed by the agencies in drafting their rules. Only further testing, better understanding of residual stresses and perhaps long-term operating results with LNG ships can finally resolve this issue.

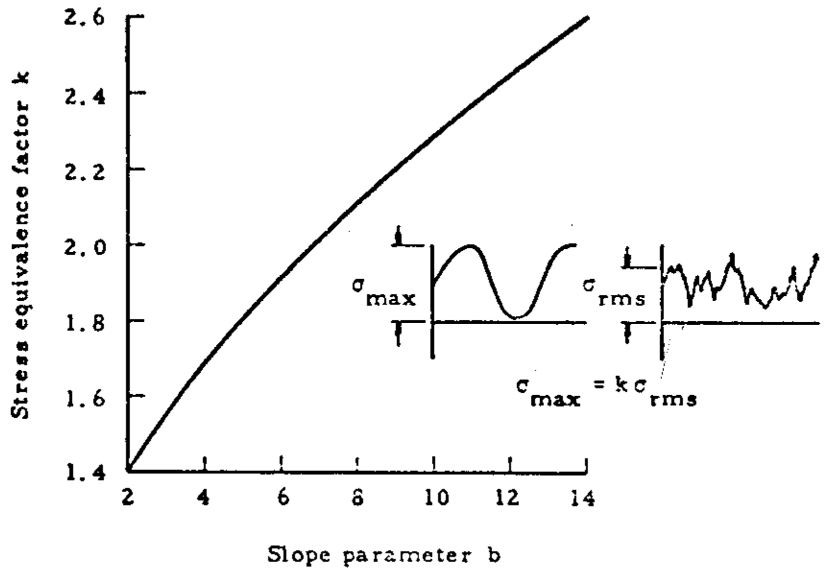


FIGURE V.11. STRESS EQUIVALENCE FACTOR

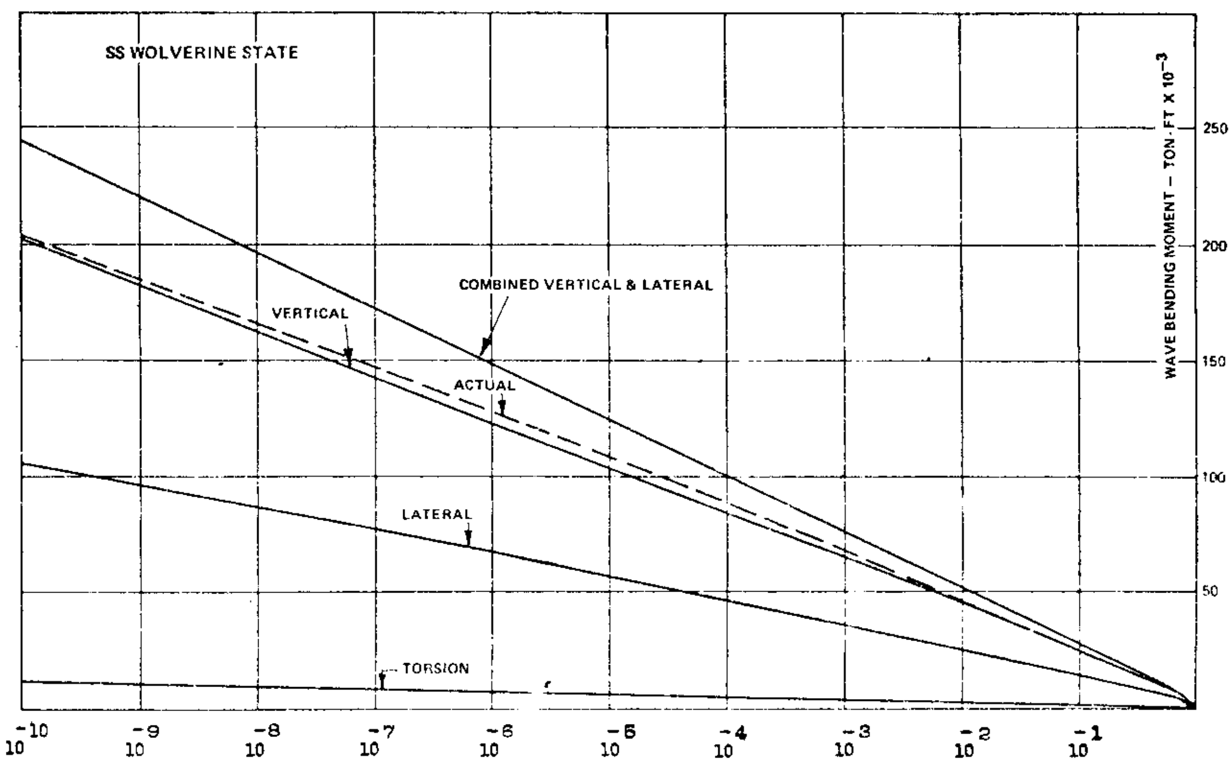


FIGURE V.12. LONG-TERM DISTRIBUTION OF BENDING MOMENT FOR THE S. S. WOLVERINE STATE, LIGHT-LOAD CONDITION (Ref. 51)

Three of the agencies, DnV, GL and NK allow a semi-log-linear representation of the cumulative distribution of the fatigue loads. A fair question is, "Does a semi-log-linear approximation accurately represent the cumulative distribution?" Figure V. 12 shows the cumulative distribution of the midship moments computed by Lewis, et al., [55] for the SS Wolverine State. It is obvious that a straight line fairly represents the curve shape but underestimates the loads in the mid range and at the low end of the spectrum. Note, however, that the "actual predicted load" is overestimated for all probability levels except for the very lowest if a straight line of approximation is drawn from the calculated value at 10^{-8} probability level. Similar results are obtained from the cumulative probability distribution of accelerations. Figure V. 13 (borrowed from Section III of the report) shows the extreme value prediction of acceleration for the SS Wolverine State based on 14 voyages. Starting with the extrapolated value at 10^{-8} and drawing a straight line approximation to zero acceleration at a probability level of one, it is apparent that the line underestimates the loads in the mid-range where maximum fatigue damage occurs. If, however, a straight line approximation is used based on the maximum acceleration predicted by the agency formulas, then a conservative estimate is obtained except for the very lowest probability levels. The conclusion to be reached is that a straight line approximation for calculated loads represents a conservative approximation to the actual loads (at least this has shown to be true for the SS Wolverine State); however, if we assume that the calculated value at 10^{-8} probability level is correct, then a straight line approximation underestimates the loads in the mid-range. Thus, we feel that the straight line approximation should be used only for values which are computed based on agency formulas and that, if wave-induced loads are computed using the agency's hydrodynamic computer programs, then the distribution determined from these calculations should be used. The use of the computed distribution is the approach taken by IACS.

Updated Fatigue Load Criteria

The rules should reflect the potential contribution of the still-water, wave-induced, dynamic and vibratory loads to fatigue damage of the tanks, and should also address the proper combination of the loads. To be representative of the ship operating conditions, superposition of the wave-induced loads on the still-water loads associated with the in-bound and out-bound voyages is required. For this approach, the wave-induced loads should be increased by a factor (to be determined experimentally) so that fatigue damage, equivalent to that produced by the total loads, would be obtained. Alternately, a new method can perhaps be developed based on a statistical combination of all loads in a manner which would yield equivalent fatigue damage. Finally, a straight line approximation to the wave-induced loads is appropriate only if the extreme values of the load are conservative estimates based on agency

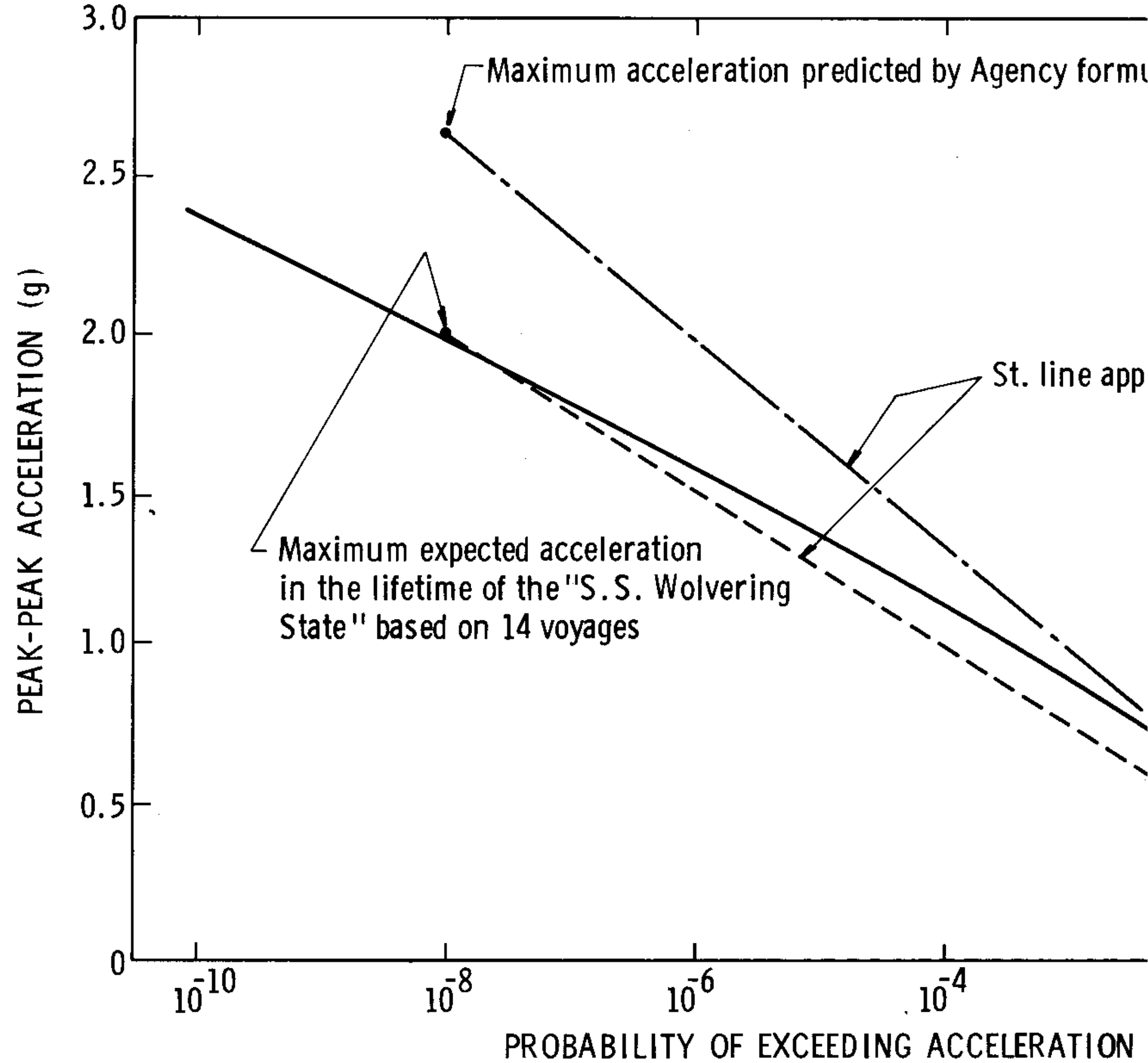


FIGURE V. 13. LONG-TERM DISTRIBUTION OF ACCELERATION
THE S. S. WOLVERINE STATE

formulas. Where calculations are available based on the agencies' computer programs, then the distribution determined from these calculations should be used.

V.2.17 Fracture Loads

A fracture mechanics analysis provides the basis upon which the classification societies allow a reduction in the secondary barrier. The analysis is performed to predict the maximum extent of crack growth over a period of time generally defined as two weeks or 15 days. This time period approximates the anticipated vessel running time between cargo loading and discharge points. The initial crack size is usually fixed by the sensitivity of the LNG gas leak detection system. Based upon the initial crack size and the calculated crack growth over the two-week period, estimates are obtained for the final crack size relative to the critical crack size and for the size of the secondary barrier required to contain leakage from the primary barrier. Favorable results from the fracture mechanics analysis gives the societies confidence to reduce the extent of the secondary barrier.

Loads for the fracture analysis, as specified by the agency rules, are summarized in Table V.13. Two agencies, ABS and LR, have no rules regarding a fracture mechanics analysis. BV specifies only that a fracture mechanics analysis must be conducted if reduction of the secondary barrier is to be considered. A fracture mechanics analysis is required for two types of tanks by the USCG, but no guidelines are provided to define the loads. The other agencies, DnV, GL, NK and IACS, require that such an analysis be performed and also provide guidelines regarding the load spectrum. Generally, these agencies define a semi-log-linear load spectrum which extends from zero load to the maximum load in 2×10^5 cycles (approximately two weeks). DnV and IACS further specify that these loads (the dynamic or varying loads) are to be super-imposed upon the static (mean) loads.

The comments which were made regarding the Fatigue Loads (Section V.2.16) apply to the Fracture Loads as well. Most important is the proper consideration of all "dynamic" loads, used here to represent the slowly varying wave-induced loads, springing, whipping and vibration[9] loads. Only IACS suggests that all "dynamic" and static loads are to be considered and none of the agencies provide guidelines for combining the various "dynamic" loads or for combining the dynamic with the static loads. Whipping may be particularly important for the fracture analysis because the load spectrum

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[9] "Vibration loads" is intended to represent tank stresses due to local tank vibrations as well as tank loading associated with hull vibrations.

TABLE V.13. SUMMARY OF AGENCY RULES REGARDING LOADS FOR FRACTURE MECHANICS ANALYSIS

AGENCY	Tank Types For Which Fracture Mechanics Analysis Is Required	Calculation of Wave-Induced Loads Required for Analysis	Assumption of Semi-Log-Linear Load Spectrum Is Allowed	Probability Level For Maximum Wave-Induced Load	Load Duration (Days)	Static Loads Are To Be Taken Into Account?
ABS						
BV	I(1), (2)					
DnV	I, M	O(3)	O	10^{-8}	15	O
GL	I, M, IT	O(3)			14	
LR						
NK	I, M, IT	O(3)	O	10^{-8}	15(4)	
USCG	I(5)				14(6)	
IACS	I, M(7)	O(8)	O		15	O

(1) I - Independent tank
M - Membrane tank
IT - Integral tank
(2) BV specifies that a fracture mechanics analysis is required if reduction in the secondary barrier is to be considered.
(3) Implied but not specifically stated.
(4) 2×10^5 total cycles (not less than 15 days)
(5) IST and SPT tanks only (refer to Table II.2).
(6) The greatest of (a) 14 days, (b) the time to off-load the cargo in an emergency, (c) the anticipated average vessel running time between cargo loading and discharge points.
(7) Semi-membrane.
(8) Normally to be carried out.

represents very severe sea states. Ways of combining the various loads were discussed in Section V.2.16. As for the Fatigue Loads, the merit of including the mean or static loads in the analysis is unknown because of the lack of knowledge about residual stresses in all-welded structures.

The validity of representing the load spectrum by a semi-log-linear distribution over a two-week interval should be questioned for the Fracture Loads just as it was for the Fatigue Loads. We were unable to examine full-scale data of individual voyages to test the accuracy of the straight-line representation. However, data are available and several voyages during which extreme weather was encountered should be examined to determine the cumulative load distribution. Until this is done, the semi-log-linear spectrum is a suitable approximation that may, in fact, be conservative for the two-week period.

Updated Fracture Load Criteria

The rules should state more explicitly the importance of springing, whipping and tank vibrations, as well as the slowly varying wave-induced loads, to the fracture mechanics analysis. This will require that guidelines also be provided for the proper combination of the wave-induced loads with the other varying loads and static loads. In addition, full-scale data from individual voyages in which extreme weather was encountered should be examined to either verify the semi-log-linear spectrum now used or determine the proper representation of the load spectrum over a two-week period in which extreme ship loads are experienced.

V.2.18 Combination of Loads

The proper combination of all loads acting on a ship and on the substructure within the ship is a difficult problem. Much of the problem arises because phasing of the dynamic loads is lost by extrapolation to the extreme values. Thus, while the maximum loads can be estimated by statistics, the means of properly combining them is often lost. As a result, common practice has been to combine the various dynamic loads by the square root method. This method is based upon the principle that for statistically independent variables, the sum of the variances is equal to the variance of the sum. Thus, the method works well for loads which are completely independent but will produce errors if there is any correlation.

Another problem related to phasing is the position of the ship when certain maximum loads occur. For example, it is necessary to know the pitch and roll angles of the ship when the maximum normal or transverse accelerations occur. This relationship is necessary in order to combine the static (gravitational) acceleration with the dynamic accelerations which are determined with respect to axes fixed to the ship. The problem of phasing and position related to the extreme loads has not been solved satisfactorily.

Proper combination of the static and dynamic loads also presents some difficulties. If the so-called "static loads" were truly static, they would never change and could always be added directly to the dynamic loads. However, the static loads also vary but at a much slower rate then the dynamic loads. Static hull deflections, for example, are produced by the ship's ballast and loading condition, which varies from out-bound to in-bound voyage. Generally, two sets of static loads are considered, those associated with the out-bound voyage and those associated with the in-bound voyage. In Chapter IV it was shown that much higher accelerations are likely to occur in the in-bound (light load) condition than in the out-bound, heavily loaded condition; however, the reverse may be true for bending moments, torsion and vertical shear. Thus, the problem becomes one of selecting the proper static load for the proper dynamic load.

It is also apparent that different loads are used to design different parts of the tank. The static external pressure may govern the design of the top of a spherical tank because of local buckling. This is a condition of minimum vapor pressure. However, the design of the bottom of the same tank will almost certainly be governed by the maximum internal pressure which is associated with the maximum vapor pressure as well as the maximum liquid head. Thus, the maximum load will be different for different parts of the tank. It is apparent, therefore, that several different load conditions, both static and dynamic and in combination, must be investigated to find the "critical" combination for each part of the tank.

Figure V. 14 has been prepared to illustrate graphically the combination of loads which must be considered in the design of LNG tanks. Although this section addresses the proper combination of the loads, it is generally the stresses which are combined. This is the approach followed in the figure. While Figure V. 14 does give general relationships between the static and dynamic loads, it does not address the problem of the phasing of the various dynamic loads. This problem has already been discussed to some extent in Chapter IV. Ideally, the ship should be in dynamic equilibrium in the sea under the action of the design loads. Unfortunately, the ship response occurs in irregular seas, and the exact sea which produces the maximum values may not be known. Also, each maximum load would most likely occur in a different sea condition so that several conditions would have to be examined. Even then, the peak stress in the tank might not occur at the conditions which produce the individual maximum loads.

Perhaps the only exact approach for solving the problem is to compute the time history of the ship's response in irregular seas. This approach would yield a continuous record of all loads acting on the ship at

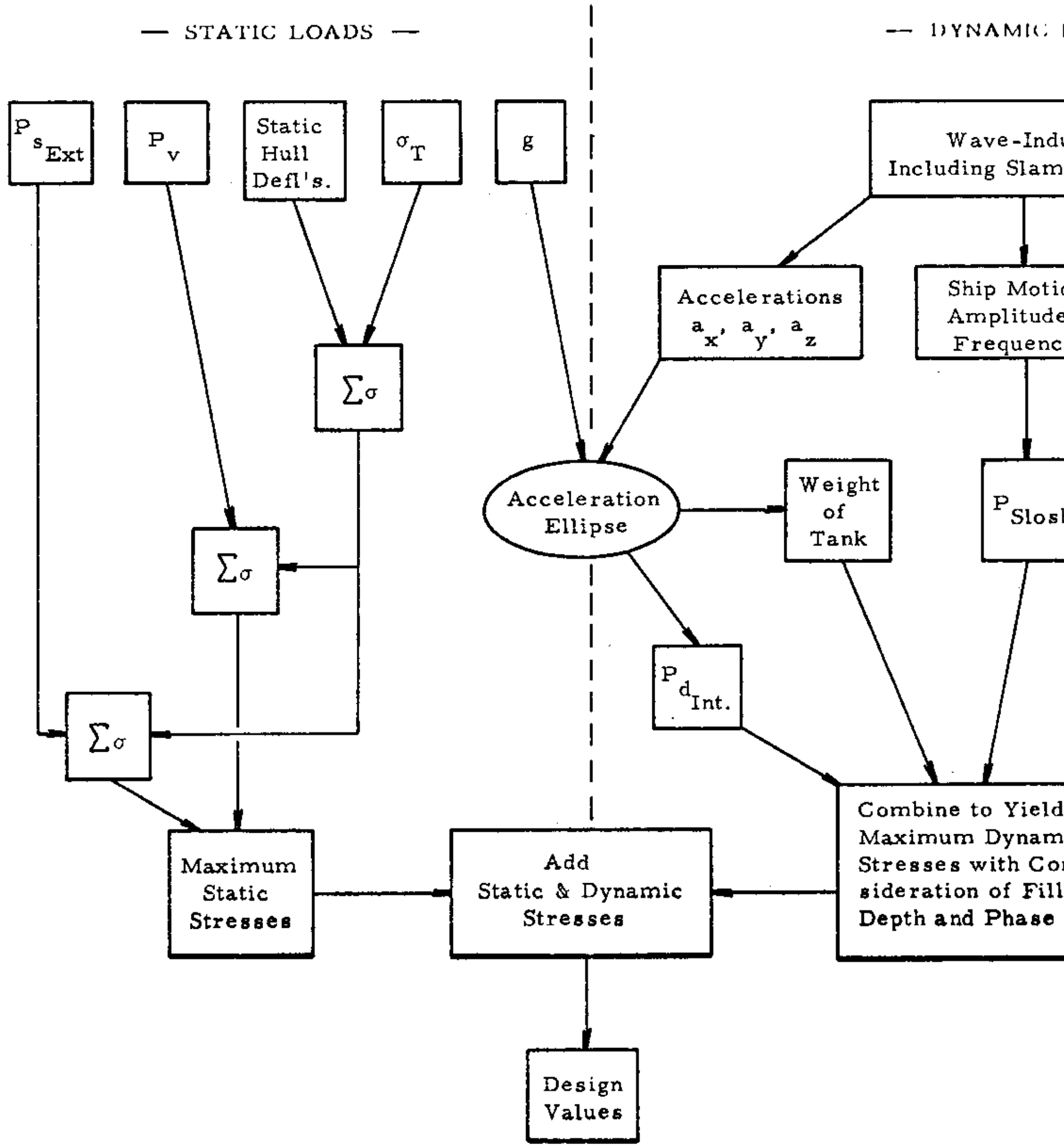

FIGURE V. 14. COMBINATION OF TANK STATIC AND DYN each instant of time. The disadvantage is the rather long time history that would be required to establish the extreme values for all loads.

In the absence of the time history of the ship response, the most suitable approach appears to be the square root method if the loads can be shown to be completely independent. Otherwise, a correlation factor for the loads must be established from full-scale data in order to properly combine those loads which are not completely independent. A similar approach has been taken by some of the classification societies in developing the ellipse for the combination of the accelerations.

VI. RECOMMENDED DATA ACQUISITION PROGRAMS

VI. 1 Introduction

In the previous chapters comparisons of agency formulas with full-scale data, results of model tests, and analytical predictions were made whenever possible. For the most part, the comparisons for LNG ships were limited to model tests and analytical predictions based on the SCORES* computer program. Because full-scale data are not available for LNG ships, comparisons of the agencies' formulas with full-scale data were based on data taken from other types of liquid and dry cargo ships which possess geometry similar to that of LNG tankers. Thus, the greatest need now is for full-scale experimental data from LNG ships in service to allow verification and/or improvement of the tank loads. Since a better understanding of the 17 tank loads may result in substantial economic savings due to reduced design values, an outline for a comprehensive data acquisition program is presented here. For those load categories which require verification or further development, a general discussion of the recommended programs (full-scale, model, long or short-term, analytical), parameters which should be measured and instrumentation required is given. Finally, recommendations regarding the selection of a test bed for acquisition of full-scale data on an LNG ship are presented.

VI. 2 Recommended Programs for Each Load Category

VI. 2. 1 Vapor Pressure

Most agencies already require that pressure gages be provided for monitoring the tank vapor pressure. These gages could be used to periodically monitor the vapor pressure during in-service voyages. This information would then be available for determining the contribution of the vapor pressure to the total load and the cyclic load history on the tank structure.

As indicated in Chapter V, the phenomena of LNG rollover may be a problem for some LNG carriers. This phenomena can readily and economically be investigated in laboratory bench tests. An analytical program should be conducted prior to the experiments to determine (1) whether rollover is likely to occur in LNG ships, (2) at what densities, temperatures, and vapor pressures rollover is most like to occur, (3) the minimum amount of LNG needed before rollover occurs, (4) the maximum pressure expected to be generated in the event rollover occurs. The results of the analytical program will fix the level of effort for the experimental program, so the experimental details cannot be completely outlined at this time. However, it is clear that, as a minimum, LNG temperature, tank internal pressure, LNG density, tank fill level, and LNG vaporization rate will have to be accurately measured. Thermocouples or other temperature-sensitive devices should be placed at several locations within the tank and on the interior tank walls. A wide-range pressure gage will be needed to measure the transient pressure during rollover, while a narrow-range pressure gage would be convenient for recording internal pressure prior to rollover. Compatibility of the pressure and temperature transducers with cryogenic temperatures will probably be the most severe instrumentation problem.

Sizing of pressure relief valves is extremely important for LNG tankers. An analytical study is recommended to determine whether the current method of sizing relief valves is consistent with the pressures generated in typical operating conditions. This may be especially important if the probability of LNG rollover is found to be high.

VI. 2. 2 Static Liquid Head

Knowledge of the static liquid head is important in determining the magnitude of the load on the tank structure. Since liquid level can be calculated if the boil-off rate is known, no additional programs are recommended beyond those for vapor pressure.

VI. 2. 3 Still-Water and Dynamic Hull Deflections

In the past, still-water and wave-induced hull bending moments have received considerable attention via full-scale measurements, [56,57] model tests, [58,59,60] and analytical studies. In some cases, short-term hull bending moments have been extrapolated to the long-term in order to provide hull design values. [4,61,62] Unfortunately, very few studies were on LNG ships (reference 63 was an analytical study of deflections on an LNG ship) and only some were concerned with hull deflections. An analytical study is needed to (1) determine which loads, i.e., bending moments, shears, and torsion, are important to the relative motions between the hull and the tank, (2) determine, by finite element procedures, the effect of hull deformations on all tank types, i.e., what stresses are introduced into the tanks, and (3) evaluate procedures for extrapolating the short-term bending moment or stress to the long-term for design purposes[10]. The results of this study should be compared with the proposed stress survey outlined in the next paragraph.

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[10] As seen in Figure VI. 1 which is reprinted from Reference 57, large differences in the long-term values result from different extrapolation procedures.

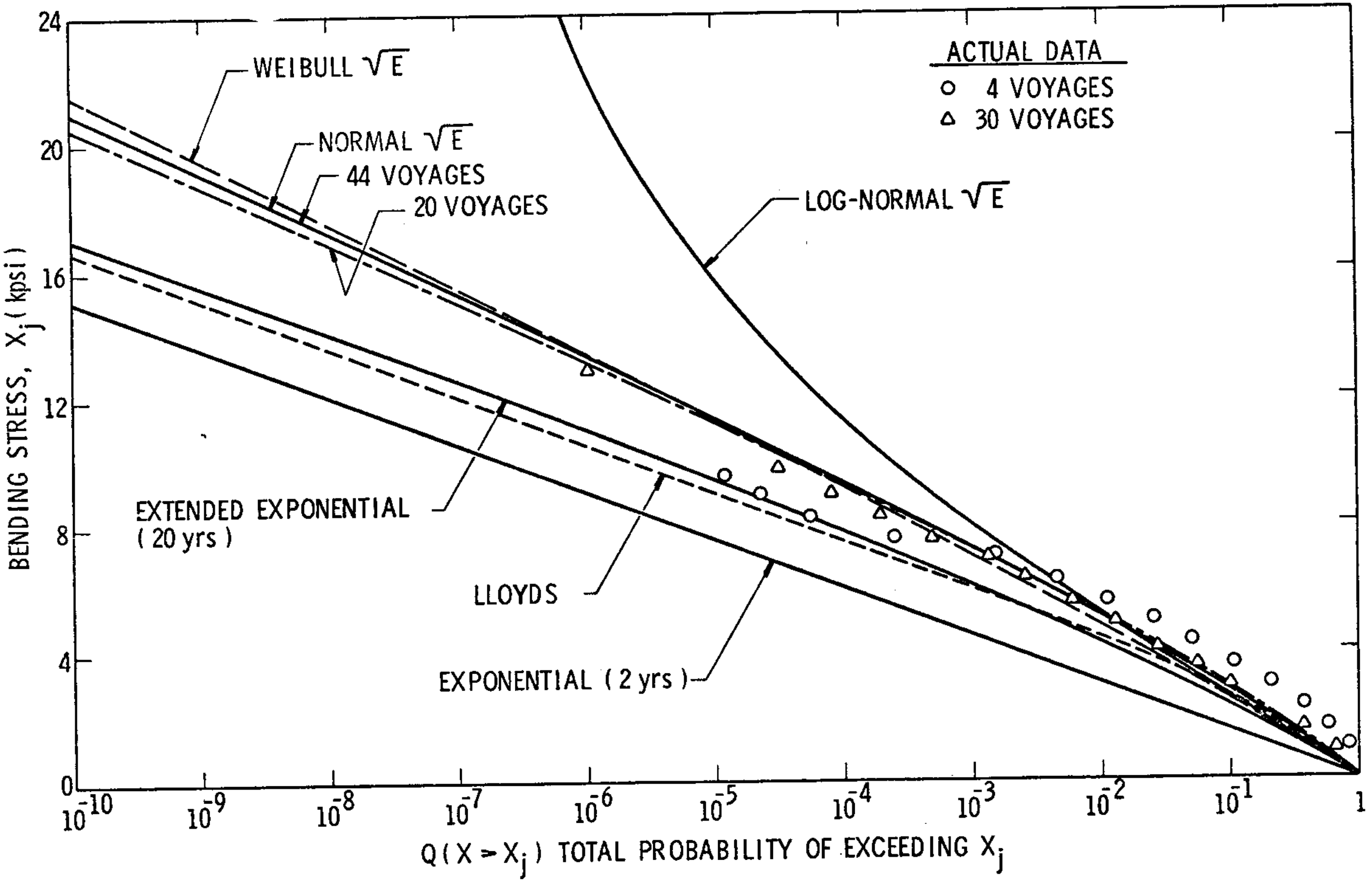

FIGURE VI-1. COMPARISON OF LONG-TERM PREDICTION PROCEDURES USING S. S. WOLVERINE STATE DATA

Hull girder deflections are determined from the same types of measurement systems used to record the ship's bending moments, torsion, and shear. What is needed, however, is enough information to determine the total ship deformation between two points on the ship; that is, between two points of attachment of the LNG tank to the hull. Once these deformations are known, the stresses introduced into the tank by these deflections can be determined. With strain gages, this will require a strain survey along the length of the hull over one or two tank bays in order to properly calculate the deflections. Alternately, deflection gages can be employed to measure the total deflection between two points, i.e., perhaps lasers can be used to optically measure the total deflection.

The instrumentation needed for long-term, full-scale measurements have already been developed. Strain gages have been used for some time to measure hull bending moments, and lasers have been recently used to measure deflections.[64] Automatic digital equipment is now available to reduce the data in real time and simultaneously update the long-term predictions. It is important that sea conditions, accelerations, and inclinations be recorded simultaneously with the hull bending moments or deflections. For this reason, an instrumentation system capable of recording as many as thirty channels, is necessary. Further comments on the test program will be found in the discussion on the test bed.

VI.2.4 Collision Loads

Enough data for full-scale collisions and model experiments probably exist to adequately define the direction and magnitude of peak accelerations to be expected in ship-to-ship collisions, grounding, etc. Thus, a comprehensive analytical program is recommended to review and analyze existing accident reports and pertinent material from the literature for the purpose of establishing design values for accelerations resulting from collisions. This survey should be complemented by a program to study the collision process from the viewpoint of analyzing in detail the dynamics of the collision. The major obstacle is the structural response of the ship, but reasonable force deformation characteristics can be established approximately. Such a study would have as its aim, the determination of total penetration, ship motions, accelerations, etc., as well as sufficient structural details to show the effects of structural design on the results of the collision. The limits of tolerable damage and the severity of a survivable collision could be established for such a model for inclusion in the rules.

VI.2.5 Thermal Gradients

The chill-down procedure for many types of LNG tank configurations involves spraying LNG onto the interior walls of each tank for as long as

36 hours prior to loading the cargo. As indicated by Poth, [65] the current guidelines do not allow sprayed LNG droplets to come into contact with tank walls, which may cause over-stressing. The spray systems currently in use have the spray nozzles mounted on a central column in each tank. During chill-down the tower cools faster than the rest of the tank. This may result in large thermal stresses. A joint analytical and experimental program is needed to optimize the spray systems or spray procedure to effect a faster more uniform chill-down. This could obviously result in substantial economic savings.

Tests should be conducted on models so that the chill-down time and the required volume of LNG would be drastically reduced. In addition, the optimized spray system resulting from the program could be installed on the model. Tests could then be conducted to determine the feasibility of the modified chill-down procedure.

Tests required should be conducted on models so that the chill-down time and the required volume of LNG would be drastically reduced. In addition, the optimized spray system resulting from the program could be installed on the model. Tests could then be conducted to determine the feasibility of the modified chill-down procedure.

Stresses induced by the chill-down are, of course, important and, in fact, the purpose of an elaborate chill-down system is to minimize these stresses. However, the stresses are so dependent on tank design and material that each design must be considered separately; however, for the test tank, the stresses should be monitored at selected tank locations in order to judge the effectiveness of the chill-down system.

VI.2.6 Accelerations

Since accelerations influence the magnitude of several other loads (i.e., dynamic internal pressure and sloshing), it is important that additional full-scale data be generated for LNG ships. Accelerometers should be positioned at several points along the ship at convenient places around the tanks. Three accelerometers, oriented so that they are sensitive to transverse, longitudinal, and vertical motions, should be placed at each measurement position. Simultaneous recording of weather conditions, as a minimum, should also be made. Real time digital analysis equipment is currently available so extrapolation of short-term acceleration data to the long-term should be handled automatically. Further comments on this experimental program will be found in the section on the LNG test bed.

In addition to the full-scale measurements of acceleration, an analytical program should be conducted to determine the best method for extrapolation of short-term data to the long-term. Further comments on this analytical program will be presented in the section on wave-induced loads.

VI. 2. 7 Dynamic External Hull Pressure

In the past, several studies have been made to determine the probability of ship slamming or shipping green seas in severe seather. [32, 49,66] However, few measurements of external hull pressures have been made during slamming or shipping of green seas. For this reason full-scale measurements of dynamic external hull pressures should be made on a full-scale LNG ship. Pressure gages should be located at various positions on the ship bottom, on the ship bow, and on top of the forward deck. Strain gages located in the same positions should be used to monitor local stresses introduced by slamming or shipping of green seas. This information should be recorded simultaneously with the acceleration data and hull deflections as previously discussed. In addition, simultaneous recordings of stresses in the tank support structure or in the tank walls in the case of membrane tanks would be helpful in determining the effect of slamming or shipping of green seas on the tank structure. However, since the incidence of slamming or shipping of green seas is likely only during severe sea conditions, this full-scale measurement program may be relatively expensive. For this reason, scale model tests of LNG ships in severe sea conditions may be preferred.

VI. 2. 8 Dynamic Internal Pressure

Since dynamic internal pressure is influenced mainly by accelerations, measurements of dynamic internal pressure should be accompanied by simultaneous measurements of accelerations. Pressure gages should be located at various positions on the interior of the tank walls to monitor pressures in full LNG tanks. These gages can also be used for the determination of sloshing pressures. The effect of dynamic internal pressure on the tank structure can be determined by strain-gaging the tank at critical locations.

VI. 2. 9 Sloshing

In addition to obtaining full-scale sloshing load data, several other types of programs would be extremely beneficial for establishing LNG sloshing loads. As stated in Chapter V, numerous test programs have been conducted using scale models of LNG tanks to investigate slosh loadings. None of these studies covered the complete range of LNG ship excitation amplitudes, frequencies, fill depths, and tank geometries while obtaining both pressures and forces. However, a composite of all previous model studies cover quite a broad range of parameters which affect LNG slosh loads.

This model data has been presented in numerous reports from various agencies throughout the world. The data is presented in different forms, and therefore no consistent method of presenting impact pressures and total tank forces exist among the various experimental studies.

Therefore, one important task that needs to be accomplished is a compilation of all currently available model data on a uniform basis. This review and presentation of currently-available model data would indicate where additional data needs to be obtained to provide a complete picture of tank pressures and forces for various excitation amplitudes, fill depths, and tank geometries.

After the model data review has been completed, additional experimental tests need to be performed to fill in conditions where current data are unavailable. These tests should be aimed at supplementing previous studies to provide a complete picture of slosh loads (pressures and forces) versus fill depths for various excitation amplitudes and frequencies, for both prismatic and spherical tanks. This data should then be presented in simplified graphical form so the designer can estimate slosh loads in typical LNG tanks. As part of this update, the scaling considerations for predicting full-scale loads with model data should be included and methods for predicting long-term, worst case pressure levels established. In addition to peak impact pressure magnitudes, the scaled pressure time histories between model and full-scale should be indicated since the design of the tank structure will be based on analyzing structural response to an impulsive loading rather than designing the tank to withstand a peak pressure level. Therefore, the design curves should indicate not only long-term, worst case peak pressure magnitudes, but also the time over which they act.

With the completed summary of all model scale loads, data obtained in full-scale slosh measurements could be utilized to update the design loads. It is anticipated that the full-scale results will indicate that model tests provide a conservative estimate of LNG tank sloshing loads since typical ship motions will not provide as severe a condition as resonant sloshing with harmonic excitation.

In addition to utilizing full-scale data to compare with model test results, sloshing data obtained with scale models of LNG ships tested in ship model basins could also be used. For example, 1/70 and 1/36 scale models of 125,000 m^3 membrane tank LNG ships have previously been tested at the Netherlands Ship Model Basin. These scale models could be fitted with scale replicas of the LNG tanks utilized in these ships. Sloshing pressures could then be measured with the ship models for both regular and irregular waves. Model motions and accelerations would be monitored as well as the wall impact pressures. Corresponding tests duplicating the motions of the tanks in pitch or roll or surge could be conducted in laboratories that have previously performed scale model slosh studies. Correlations would show the validity of conducting laboratory tests with harmonic pitching motions only and perhaps indicate adjustments or correlations that must be made in order to achieve better agreement with sloshing introduced by actual ship motions. This type of testing would represent a comparison without scale effects since the tank size and geometry with the harmonic studies would be identical to the tank fitted in the LNG ship models. Subsequent pressure measurements made on full-scale ships for which the models were scaled would indicate if the prediction methods developed under slosh study programs are valid. The results of both the model basin and full-scale measurements would be used to provide final design charts.

VI.2.10 Fatigue and Fracture Loads

To properly evaluate the fatigue loads on an LNG tank requires an extensive analytical and experimental program which involves the history of all loads acting on the tank. The essential ingredients of such a program should include:

(1) A study to establish the proper combination of loads for fatigue life prediction (refer to Section V.16).

(2) Analytical prediction of the time history of the tank loads for independent and membrane type tanks.

(3) A thorough finite element stress analysis to predict locations of maximum stress in the tank.

(4) Analytical prediction of the tank fatigue life.

(5) A full-scale experimental program to measure all pertinent tank loads as recommended in this data acquisition program.

(6) Strain measurement at critical locations as determined in (4).

(7) Extrapolate short-term strain measurements to ship lifetime predictions and predict tank fatigue life using methods of exceedance or actual records played through fatigue gages.

(8) Compare extrapolated experimental results to predicted analytical values.

(9) Evaluate contributions of wave-induced loads, dynamic loads and vibrations to the tank fatigue life.

This program would basically bring together all of the other load measurements and develop a rationale for the proper superposition of the loads. It involves comparisons of the time history of all measured and predicted loads and stresses. Thus, it will be quite extensive and must be performed on a fully-instrumented ship in conjunction with all other experiments.

Results of this program would also apply to the fracture analysis required for the LNG tank.

VI. 2. 11 Wave-Induced Loads

As indicated in Chapter IV of this report, much emphasis has been placed on the calculation of wave-induced loads in previous programs. Since Korvin-Kroukovsky introduced the strip-theory approach to calculate ship motions, much work has been done in refining the original theory. However, these refinements have resulted in only a slight improvement of the original theory. For this reason, the emphasis should now be placed on improving the input to the strip-theory calculations and the proper extrapolation and combination of the results, rather than on improving the calculation of the transfer function. The most important input to the strip-theory calculation is the description of the sea. Special emphasis should be placed on finding new ways to describe sea conditions, especially non-fully developed seas. For instance, there is no current theory available to describe non-directional seas.

In addition, analytical studies should be conducted to determine the best method for extrapolating short-term data to the long-term. As indicated in Chapter III and in Chapter V of this report, significant differences in long-term values result from different extrapolation procedures. At this time, the Weibull extrapolation procedure used by DnV and the combined Rayleigh-Normal procedure used by the Webb Institute of Naval Architecture, seem particularly attractive.

VI. 3 The LNG Ship Test Bed

It is obvious that the above set of measurements would require the monitoring of a great many parameters and a large data acquisition system. However, it should be noted that much of the information required in the various load areas overlap and therefore, reducing the number of measurements to a more reasonable number would still provide extremely beneficial information. Since there are numerous LNG ship designs, the ships chosen for a test bed should provide measurements which will be applicable to the greatest number of LNG ships. Ideally, at least two types of LNG ships would be appropriate: a ship with membrane tanks, and another one with independent tanks. These ships should represent the new generation of LNG carriers which have a total load capacity on the order of $125,000 m^3$. The required measurements fall into the following six categories:

- Pressure
- Strain
- Temperature
- Motion
- Fatigue
- Environment

With the exception of thermal measurements, these types of parameters have all been measured on full-scale ships in the past, and prior experience can be utilized in setting up instrumentation systems on LNG ships. Strain gages, pressure transducers, and thermocouples mounted on the tank walls will have to be compatible with cryogenic temperatures. Special calibration procedures may have to be devised for these particular transducers. Pressure instrumentation for both dynamic external hull pressure and for sloshing pressure measurements should be able to record accurately the impulsive type pressures that will occur. Model studies with sloshing can be utilized to establish the magnitude and duration of the pressure pulses that are expected. Similarly, previous slamming measurements will help establish requirements of the external hull pressure measuring system. The gages used for sloshing pressures can also be utilized to determine the dynamic internal pressures in full tanks.

The most important part of the data acquisition will be the data processor. The data processor should be capable of handling inputs of all parameters described above. For convenience, the system should be capable of continually updating the long-term prediction of the design variables. Similar data acquisition systems have been recording strain, acceleration, and other motion data automatically during adverse operating conditions. For example, measurements currently being made on a Japanese bulk carrier[64] are being recorded on digital type tape recorders with magnetic tape capable of recording 60 channels at frequencies from 126 to 8,000 Hz. Past experience with developing data acquisition systems for obtaining full-scale measurements can be utilized to set up this data acquisition system.

VII. CONCLUSIONS

Conclusions which can be drawn from this study are given below in each of the subject areas covered in Chapters II through VI. Recommendations were covered adequately in Chapters V and VI, and are not repeated here.

Criteria Review

There are many similarities and differences among the rules of the various agencies. In terms of their similarities, the rules can be grouped by agency as follows:

- ABS, LR, USCG
- Dnv, GL, NK
- BV
- IACS

The IACS rules are most like those of DnV, GL, and NK, yet they are a composite of the rules of the various agencies and thus incorporate some of each. IACS recommendations are the most complete in their scope of coverage, but they are less specific than the rules of some of the other agencies. BV rules are different in format from those of the other agencies so that any similarities which may exist are not apparent. The rules of DnV, GL and NK are the most current and appear to reflect current practice of the societies; whereas, the rules of ABS, LR and USCG are outdated in that they do not require calculation of the wave-induced loads. These calculations are performed routinely by all of the agencies.

In general, the rules are not very specific. Where guidelines for the loads are provided, additional development plus verification by comparison with full-scale measurements from in-service LNG tankers is needed.

Acceleration Comparisons

Very few full-scale measurements have been reported for LNG ships. Comparisons of accelerations predicted by agency formulas with full-scale data from other ships of similar geometry showed the agency formulas to generally give conservative estimates. However, comparisons of the formula accelerations with results for LNG ships predicted by the program SCORES*, revealed both areas of conservatism and unconservatism. Further comparisons with accelerations measured on full-scale LNG ships are needed.

Wave-Induced Loads

Direct comparisons of the results of the Classification Societies' computer program for determining the wave-induced loads could not be made. Comparisons of the calculation procedures revealed that the methodology is very similar but that small differences exist throughout. For example, methods for computing the RAO's should yield almost identical results, although minor differences exist among the agencies. The greatest differences are in the method in which the wave data is used and in the long-term prediction from short-term results. Dynamic effects, such as slamming, whipping and springing are not accounted for.

Additional analytical development is required for improved descriptions of roll damping, surge motions, and the sea environment. Further development is also needed in the method of extrapolating short-term results to long-term predictions. Comparisons with short and long-term full-scale data from LNG ships is necessary to substantiate the extrapolation procedures.

Criteria Evaluation

Very little full-scale data from LNG ships was available for use in evaluating the criteria. Thus, most of the evaluation was by agency-to-agency comparison of the formulas plus comparisons with model tests and limited full-scale data from other types of cargo ships. Detail evaluation and comparison of the formulas revealed similarities and differences, just as were found from the criteria review. These comparisons indicated the need for further development of some of the formulas, such as those for sloshing pressures and external hull pressures, and for full-scale data from LNG ships with which to improve and/or verify all of the formulas. The evaluations also revealed the need for a comprehensive procedure for combining all of the different loads to produce design values as well as load-time histories for calculating fatigue and fracture. For the numerous loads where no formulas are given, the specific requirements and methodology for establishing the loads need to be more thoroughly listed.

Recommended Programs

Because of the lack of full-scale in-service data on LNG ships with which the current criteria can be compared and improved or verified, comprehensive analytical and experimental programs are recommended to provide the required data. The programs consist of analytical studies, model tests and full-scale measurements which will enhance the state of knowledge and generate the data required to improve and verify the load criteria. Because many of the programs are inter-related (a complete definition of the fatigue loads require knowledge of all of the loads), two fully instrumented test ships are suggested, one ship with membrane tanks and one with independent tanks. Proper data correlation is essential and therefore data gathering and reduction should be automated to the greatest extent possible.

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APPENDIX A

SUMMARY OF AGENCY REQUIREMENTS FOR EACH LOAD CATEGORY

A.1 Introduction

This appendix presents load criteria from the eight classification societies or regulatory agencies as stated by their rules in each of the seventeen load categories utilized in this report. At the end of each load category a summary indicating similarities and differences among the societies' rules is given.

A.2 Tank Loading Criteria

A.2.1 VAPOR PRESSURE

A.2.1.1 American Bureau of Shipping

Suitable means are to be provided to relieve the vapor pressure in the hold spaces should leakage of cargo occur. Tanks with a maximum allowable working pressure given by MAWP > 0.703 kg/cm^2 are designed to pressure vessel regulations; no regulations or guidelines for MAWP < 0.703 kg/cm^2.

A.2.1.2 Bureau Veritas

Generally, cargo tanks are to be designed for an effective vapor pressure equal to the setting pressure of the safety valves. The safety valves on all tank configurations are subject to the following requirements. Other requirements for tank design on specific tank configurations are presented separately.

"22-51-11 Every cargo tank is to be fitted with at least two pressure safety valves. When the cargo tank volume is less than 20 m^3, however, only one valve may be provided."

"22-51-12 The safety valves of every cargo tank are to be so designed as to be capable of discharging, without the effective pressure in the tank raising by more than 20% above the maximum service pressure, the total flow Q given by the following formula:

$$Q = 10,500 \frac{f\ S^{0.82}}{L\ C} \sqrt{\frac{T}{M}}$$

where:

Q : is expressed in cubic metres/hour of air at atmospheric pressure and at a temperature of 0°C.

f : is a coefficient having the value given in the table below

	f
Pressure cargo tanks above weather deck:	
(a) non-insulated or having an insulation not in accordance with the conditions stated in (b)	1
(b) covered with an insulation having fire resistance characteristics approved by the Administration and a total heat exchange coefficient not exceeding 0.5 kcal/hm^2 °C	0.5
Pressure cargo tanks below weather deck:	
(a) where the setting pressure of safety valves exceeds 0.7 kgf/cm^2.	0.5
(b) where the setting pressure of safety valves does not exceed 0.7 kgf/cm^2:	
1) tanks located in constantly inerted spaces	0.1
2) tanks located in non-inerted spaces	0.2
Gravity cargo tanks	0.1

Note: Where the tanks are located partly above the weather deck and partly below, the flow will be the sum of the flows calculated separately for both parts providing for the corresponding values of coefficient f.

S : is the total external area, in m^2, of the tank, excluding any appendix such as domes. For flat-bottomed tanks, however, the bottom area may be deduced.

T : is the absolute temperature, in K degrees, on the pressure side of safety valves at accumulation conditions.

C : is a constant given by the following formula in function of the ratio $\gamma = \frac{C_p}{C_v}$ of specific heats

$$C = \left[\gamma \left(\frac{2}{\gamma + 1} \right)^{(\gamma+1)/(\gamma-1)} \right]^{\frac{1}{2}}$$

L : is the latent heat of vaporization of the liquid, in kcal/kg, at accumulation conditions

M : is the molecular weight of the discharged gas."

"22-51-13 The safety valves of each cargo tank are to be capable, without the pressure rising by more than 20% above the maximum service pressure, of discharging the vapors generated by the heat flow through the walls, assuming an ambient temperature of 45°C, and those displaced during loading at maximum rate."

"22-51-14 The safety valves are to be set to commence discharging at a pressure not exceeding by more than 5% the maximum service pressure."

Independent Tanks

Pressure cargo tanks are to be designed for a pressure at least equal to the maximum service pressure at the maximum service or loading temperature, subject to a minimum of 0.7 kg/cm^2. No specific regulations for self-supporting gravity tanks are given.

Membrane Tanks

No specific regulations for integrated tanks are given as they are specially considered.

A.2.1.3 Det norske Veritas

Regarding the sizing of safety relief valves, the rules state:

"B508. The total capacity of the pressure relief valves is to be sufficient to ensure that the pressure in the tank will not rise more than 20% above the design pressure in either of the conditions mentioned below. The pressure relief valves are to discharge that volume of gas which is displaced when the amount of liquid admitted to the tank per unit of time is 1.5 times the maximum capacity of the delivery pipes. The pressure relief valves are to discharge that volume of gas which is produced by the evaporation of cargo when heat at a flow rate of Q kcal/h is conveyed to the tank, and this heat is used exclusively to evaporate the cargo.

$$Q = F\ S^{0.8}\ \text{kcal/h}$$

F = coefficient given in the table.
S = the surface area of the tank in m^2.

The maximum back pressure which may occur in the pipe systems for escape gas immediately outside the pressure valves, is to be taken into account when determining the capacity of the pressure relief valves."

Table for coefficient F.

Item	Tank arrangement	F
1	Tanks without insulation, on deck	5.0×10^4
2	Tanks with insulation, on deck	2.5×10^4
3	Tanks in completely closed holds, not covered by item 4 below	2.5×10^4
4	Tanks, with insulation, in closed holds, or tanks in closed, insulated holds where the insulation will not be damaged by fire on deck or on the sea near the ship	1.2×10^4

"B509. For tanks designed for a gauge pressure of 0.5 kp/cm^2 or less, which are placed in closed spaces always filled with inert gas, the parts of the tanks situated below the ship's water line in light ship condition are not to be included when determining S. In order that a tank may be considered as insulated, the insulation is to be non-destructible by fire.

The heat transfer coefficient is not to exceed 0.5 kcal/ $m^2h\ ^\circ C$. Tanks in holes to which, in case of fire, an inflow of fresh air cannot effectively be prevented, are to be regarded as tanks on deck. If more than 50% of the tank surface lies above deck level, the tank is to be taken as situated on deck. If 50% or less of the tank surface lies above deck level, the F-value may be determined by linear interpolation based on the tank surface."

"B510. Tanks which cannot, with a reasonable factor of safety, withstand an internal vacuum equal to 0.25 kp/cm^2, are to be equipped with vacuum relief valves.
Tanks equipped with refrigerating systems are in all cases to be equipped with vacuum relief valves or some other devices in order to prevent development of unacceptable vacuum in the tanks."

All tanks must be designed to the following requirements concerning design vapor pressure:

"B201. The maximum allowable vapor pressure, p_o, at the top of the tank is not to be taken less than:

* the highest set pressure of the safety valves

* the vapor pressure at the reference temperature

* the vapor pressure at a temperature of 45 °C for tanks without cooling or insulation

* the pressure of the inert gas for tanks unloaded by means of inert gas."

Independent Tanks

Type A independent tanks may be subjected to pressure tests on a case-by-case basis. Type B tanks are normally hydraulically tested to 1.5 times the design pressure.

Membrane Tanks

Membrane tanks are subjected to the same requirements as independent tanks.

A.2.1.4 Germanischer Lloyd

The vapor pressure for all tank configurations is not to be less than the highest setting pressure of the safety valves on the cargo tanks. For pressure tanks in which the pressure of the cargo is only dictated by the ambient temperature, the design vapor pressure is not to be less than the vapor pressure at the maximum expected temperature during most unfavorable transport conditions. Generally, for pressure tanks the reference temperature is 45°C, but this may be altered depending on the service area of the ship.

Independent Tanks

For independent gravity tanks G3A, G3B, the design pressure $P_o \leq 0.7$ kg/cm². No other guidelines are given.

Pressure tanks, P1, have a design vapor pressure given by:

$$0.7\ [\mathrm{kg/cm^2}] \leq P_o < 2 + 0.1\ h_t \gamma \qquad [\mathrm{kg/cm^2}]$$

but $$P_o < 3 \qquad [\mathrm{kg/cm^2}]$$

where

h_t is the tank height in meters not including the dome, if any

γ is the maximum specific weight of cargo $[\mathrm{t/m^3}]$

Pressure tanks, type P2, have a design pressure given by:

$$P_o \geq 2 = 0.1\ h_t \gamma \qquad [\mathrm{kg/cm^2}]$$

but $$P_o \geq 3 \qquad [\mathrm{kg/cm^2}]$$

Membrane Tanks

The design vapor pressure for membrane tanks is the same as for integral tanks.

A.2.1.5 Lloyd's Register of Shipping

No regulations are given.

A.2.1.6 Nippon Kaiji Kyokai

Safety valves for all tank designs are subject to the following requirements. Design requirements for specific tank configurations are presented separately.

"5.5.4-1. Two or more safety valves against overpressure are to to be provided on each tank at the uppermost point of gas part. In case of pilot-type safety valves, a separate pressure detecting terminal is to be provided."

"5.5.4-2. The total capacity of safety valves against overpressure is to be such that it is capable of discharging the amount specified in the following (1) or (2), whichever is the greater, at a pressure not exceeding 1.2 times the MARVS of the safety valve:

(1) The total amount of gas at ambient temperature of 45°C by adding the amount of gas generated due to heat input into the tank to the gas quantity discharged during loading at a full capacity.

(2) The quantity of gas to be generated by heat input into the tank in case of fire, represented by the amount of gas generated from the heat quantity obtained from the following formula. However, where specially approved by the Society in consideration of the hull structures and tank structures, the coefficient of 12,200 in the following formula may be reduced to the value not less than 6,100. In case where application of the formula is not practicable due to shape, structure and arrangement of the tanks, the calculation formula will be given in each case.

$$Q_h = 12,200\ A^{0.82}$$

where:

Q_h = Heat input, (kcal/h).
A = Total surface area of the tank, excluding the surface area below the minimum design draught in the service condition of the ship, (m^2)."

"5.5.4-3. Overpressure safety valves attached to pressure tanks for temperature below ambient are to comply with the requirements in 5.5.5."

"5.5.5-1. Two or more safety valves are to be provided in each tank, and are to be set at a discharge pressure not exceeding 1.05 times the design vapor pressure of the tank."

"5.5.5-2. The total capacity of discharge of safety valves in each tank is to be sufficient enough to relieve the volume obtained from the following formula at a pressure not exceeding 1.2 times the MARVS. However, for tanks lagged with insulating materials, the required capacity of discharge of safety valves may be reduced to the value not less than 1/2 W_r depending on degree of heat insulation effectiveness, where approved by the Society.

$$W_r = 1.56 \times 10^5 \times \frac{A^{0.82}}{L_h}$$

where:

W_r = Required discharge quantity, (kg/h).
A = The following value depending on shape and dimensions of each tank:

$D_t \times (U + 0.3\, D_t)$... for tanks of cylindrical form having dished or semi-elliptical heads
$D_t \times U$ .... for tanks of cylindrical form having hemispherical heads
D_t^2............ for spherical tanks

D_t = Outside diameter of tanks, (m).
U = Overall external length of tanks, (m).
L_h = Latent heat for vaporization of cargo at 1.2 times the approved working pressure of tanks, (kcal/kg)."

"5.5.5-3. Safety valves are to be attached to tanks near the highest part of vapour space so as to be able to discharge vapor gas during operation. No shut-off valves is to be fitted between tanks and safety valves, except a set of interlocking-type shut-off valves which are so arranged that when some of them are closed the others are to be automatically opened. In this case, total capacity of two or more safety valves opened are at all times to satisfy the requirements in preceding 2."

"5.5.5-4. One or more safety valves are to be fitted on each pressure container for liquid. Capacity and attachment of safety valves are generally to comply with preceding 1, 2, and 3, respectively."

Independent Tanks

The design vapor pressure shall not exceed 0.7 kg/cm^2 on Type A independent prismatic tanks, except where specially approved by the Society. The scantlings of the strength members of Type B independent prismatic tanks are not to be less than those of Type A independent prismatic tanks, unless specially approved by the Society.

"4.7. Type B independent pressure vessel configuration tank is to be designed for $P_o > 0.7$ kg/cm^2. Where specially approved, Type A independent pressure vessel tank may be designed for $P_o \leq 0.7$ kg/cm^2, in which case scantlings may be reduced at the Society's discretion."

"4.8. Type C independent pressure vessel configuration tank is designed and constructed in accordance with pressure vessel standards, taking the design vapor pressure P_o, so as to make the ratio of stress corresponding to P_o to the total design stress in the tank sufficiently large."

The design vapor pressure P_o is not to be less than the value in either of the following paragraphs:

"(1) Where an exact stress analysis of the tank is carried out the design vapor pressure is to be determined so that a stress component induced by the design vapor pressure is sufficiently larger than the maximum dynamic stress component induced by the dynamic loads."

"(2) The design vapor pressure is obtained from the following formula:

$$P_o > 2.0 + 0.3\ \gamma h \qquad [\text{kg/cm}^2]$$

where

h = height of tank, excluding dome [m]

γ = design specific gravity of cargo [t/m^3] "

Membrane Tank

"4.9.1. A membrane tank is to be designed for a vapor pressure $P_o \leq 0.25$ kg/cm^2. If the scantlings are increased accordingly, and approved by the Society, P_o may be increased to a higher value, but less than 0.7 kg/cm^2."

A.2.1.7 United States Coast Guard

Regarding the sizing of safety relief valves, the rules state:

"VIII. A. 2. a The rate of discharge to be provided for tanks and piping systems in consideration of heat input due to fire shall not be less than that determined from the following formula:

$$Q = FGA^{0.82}$$

$$G = \frac{633,000}{LC}\sqrt{\frac{zT}{m}}$$

Q = Minimum required rate of discharge in cubic feet per minute of air at standard conditions (60°F and 14.7 psia).

G = Gas factor

F = Fire exposure factor - may be interpolated when tank falls under two or more F categories

F = 1.0 for pressure vessel type tanks above deck

F = 0.5 for pressure vessel type tanks above deck insulated in a manner satisfactory to the Commandant as discussed in Section X.

F = 0.5 for pressure vessel type tanks installed in a completely enclosed space below deck.

F = .25 for pressure vessel type tanks installed in inerted holds

F = 0.2 for non-pressure vessel type tanks in holds

F = 0.1 for non-pressure vessel type tanks in inerted holds

F = 0.1 for membrane type tanks

A = area in square feet as follows:

$A = \pi D(U + .3D)$ for cylindrical tanks with spherically dished or semi-ellipsoidal heads.

$A = \pi DU$ for cylindrical tanks with hemispherical heads

$A = \pi D^2$ for spherical tanks

A = external area less the bottom surface area for non-pressure vessel type tanks

M = molecular weight of the cargo

T = temperature in degrees R (460 + degrees F) at the relieving conditions (120 percent pressure at which the safety relief valve is set).

D = outside diameter of the tank in feet

U = external overall length of the tank in feet

C = constant based on relation between specific heats of gas; if not known, use C = 315

L = latent heat of vaporization for the material, in BTU per pound

Z = compressibility factor of the gas at relieving conditions; if not known, use Z = 1.0"

Independent Tanks

No specific guidelines for the IPT and SPT pressure vessel tanks, which are designed in accordance with Marine Engineering Regulations.

The design pressure, P_o, for an IIT gravity tank conforming to ABS scantlings is to be not greater than 0.282 kg/cm^2. If the scantlings are appropriately increased, $P_o \leq 0.703$ kg/cm^2.

The design pressures for IST gravity tanks are determined according to pressure vessel standards.

Membrane Tanks

The design pressure for the IMT membrane tank shall be no greater than 0.282 kg/cm^2 when the hull structure conforms to ABS scantlings or equivalent. If the ABS scantlings are appropriately increased and specifically approved by the USCG, the design pressure may be increased to 0.703 kg/cm^2.

Integral Tanks

The design pressure P_o for an IGT tank conforming to ABS scantlings is to be not greater than 0.282 kg/cm^2. If the scantlings are appropriately increased, $P_o \leq 0.703$ kg/cm^2.

A.2.1.8 International Association of Classification Societies

Unless the entire cargo system is designed to withstand the full vapor pressure of the cargo, maintenance of cargo tank pressure below the maximum allowable relief valve setting should be provided. Usually the reference temperature is 45°C.

Independent Tanks

Design vapor pressure for Type A, B tanks is to be less than 0.7 kg/cm^2.

"1.4.c. Type C tanks meeting pressure vessel criteria will have a vapor pressure P_o given by

$$P_o \geq 2 + A c \gamma^{3/2} \quad [kg/cm^2]$$

where

A is a material constant having the following values:

0.3 for carbon manganese steels and 9% Ni-Steel

0.16 for aluminum alloy (for other materials, the value of A will be determined by the Classification Society).

c is a characteristic tank dimension, taken to be the greatest of:

h, 0.75 B, 0.6 ℓ

where

h = height of tank in meters (ship's vertical dimension)

b = width of tank in meters (ship's transverse dimension)

ℓ = length of tank in meters (ship's longitudinal dimension)

and γ is the relative density of cargo (γ = 1 for water)."

However, the Classification Society may allocate a tank complying with the above criterion to Type A or Type B dependent on the configuration of this tank and the arrangement of its supports and attachments.

Membrane Tanks

The design vapor pressure, P_o, should not normally exceed 0.25 kg/cm^2; however, if the hull scantlings are increased accordingly, P_o may be increased to a higher value but less than 0.7 kg/cm^2.

A.2.1.9 Summary of Vapor Pressure Criteria

While all agencies probably have standard procedures for the sizing of safety valves, only BV, DnV, NK and the USCG provide specific formulas in the agency rules cited in Chapter II. The BV and USCG formulas are to be used to calculate total amount of heat input into the tank caused by fire (m^3/hr). The safety valve is to be capable of safely discharging this amount of gas. The DnV and NK rules require that the pressure relief valves discharge the volume of gas produced by the evaporation of cargo when a heat input to the tank is introduced. A formula is provided to calculate the flow rate in Kcal/hr. In addition, BV, DnV, and NK require that the safety valves are to capable of discharging the boil-off flow generated by heat transfer through the tank walls, when the ambient temperature is 45°C, without exceeding an internal pressure of 1.2 times the maximum service pressure.

Independent Tanks

The agencies divide independent tanks into two categories. Tanks with a MAWP[11] < 0.7 kg/cm^2 fall into the broad category of gravity type tanks. Those tanks with a MAWP > 0.7 kg/cm^2 are generally referred to as pressure vessel tanks. Usually pressure vessel designs are required to be subject to more detailed stress analysis than gravity type tanks. GL, NK, and IACS define pressure vessel tanks as those with a vapor pressure greater than 0.7 kg/cm^2 but less than a pressure which is to be calculated from the specific gravity of the cargo and a characteristic dimension of a tank.

Concerning vapor pressure design requirements, the following regulations apply. ABS tanks with a MAWP > 0.7 kg/cm^2, and USCG IPT and SPT configurations are to be designed according to pressure vessel regulations. BV, DnV, GL, and IACS require that the design vapor pressure be greater than or equal to the highest setting of the safety valves, but not less than the pressure at a temperature equal to the maximum service temperature. For tanks with no cooling or insulation, DnV requires the tanks be designed for a pressure not less than the vapor pressure at a temperature of 45°C. For tanks to be unloaded by an inert gas, DnV required that the tanks be designed for a pressure not less than the pressure of the inert gas. DnV may also require that Type A independent tanks be subjected to a pressure test. Type B tanks will normally be hydraulically tested at 1.5 times the design pressure. LR has no requirements concerning vapor pressure.

[11] Maximum Allowable Working Pressure

Membrane Tanks

The requirements for DnV membrane tanks are the same as for independent tanks. GL, NK, USCG, and IACS all require the design pressure be less than or equal to 0.25 kg/cm^2. However, if the scantlings are appropriately increased, the design pressure can be increased to a maximum of 0.70 kg/cm^2. ABS and BV specially consider membrane tanks; no requirements are given.

Integral Tanks

USCG requirements for integral tanks are the same as for membrane tanks.

A.2.2 STATIC LIQUID HEAD

A.2.2.1 American Bureau of Shipping

Independent Tanks

In addition to the following rules, also see Section I.2.11, regardint the combination of the static and dynamic loads.

"24.37.1. Structural primary containers and supporting arrangements are to be designed to withstand:

a. Static Head Effect: A test head of 2.44 meters of water above the top of the tank, or .610 m above the top of the hatch, whichever may be the greater.

b. Combined Static and Dynamic Effects: Provision is to be made for the combined effect of static pressure, internal vapor pressure (if any), and simultaneous rolling, pitching, and heaving."

Membrane Tanks

No specific regulations.

A.2.2.2 Bureau Veritas

Independent Tanks

No specific regulations concerning pressure cargo tanks; however, the rules state that after their completion, the tanks are to be tested according to pressure vessel regulations.

"Boilers and pressure vessels are to be submitted on completion to a hydraulic test under a pressure P_e as defined hereafter as a function of design pressure P:

$$P_e = 1.5\ P \qquad \text{where } P \leq 40 \quad kg/cm^2$$

$$P_e = 1.4\ P + 4 \qquad \text{where } P > 40 \quad kg/cm^2 \text{ "}$$

"29.91.21. After completion, self-supporting gravity cargo tanks are to undergo a test under hydraulic pressure corresponding to a water head of 2.4 meters above the tank top (or 0.60 meters above the dome top if this value is greater). Particular test conditions are to be adopted, with the adminstration's agreement, when the setting pressure of safety valves exceeds 0.25 kg/cm^2, also for gravity cargo tanks intended to carry high density products."

Membrane Tanks

Integrated Cargo Tanks are to be tested according to the requirements of the Administrations Materials and Fabrication regulations, and also be subjected to suitable leak detection test.

"3.32 - All the compartments adjoining a cargo tank are to be water tested under the load height relating to the higher of the following levels:

- 2.40 m above the tank top,
- 0.60 m above the dome."

"3.33 - Where a water test is required for a cargo tank, it is to be carried out before the insulation is laid, by water filling to a height above the tank top, in meters, equal to:

$$h_a = H\sqrt{\delta_o}$$

complemented by a compressed air filling at the setting pressure of the safety valves, without exceeding 0.24 kgf/cm^2.

H is, in meters, the depth of the tank,

δ_o is the cargo specific gravity, determined in taking account of the variety of possible supply areas; more specially, for natural hydrocarbons, the value adopted is never to be less than the specific gravity of the pure product increased by 20%.

If the results of the tests are not satisfactory for the tank concerned, the remainder of the cargo tanks may be required to be tested."

A.2.2.3 Det norske Veritas

Static loads are considered to be due to 99% filling by volume of the tank with a cargo of design density.

Independent Tanks

"c. 102 - For tanks designed with a maximum vapor pressure of $0.25 \ kp/cm^2$ gauge and a cargo density $\gamma \leq 1.0$, the pressure head h is to be corrected as follows:

$$h' = (h - 2.5)\frac{1+\gamma}{2} + 2.5$$

h = corrected pressure head in meters.

For tanks carrying liquids with $\gamma > 1.0$, the pressure head h is given by:

$$h' = (h - 2.5)\ \gamma + 2.5$$"

"c. 103 - For tanks where the maximum vapor pressure is greater than $0.25 \ kp/cm^2$, the pressure head as determined from 102 is to be increased by:

$$h = 10\ (p_o - 0.25) \text{ meters.}$$

p_o = maximum setting of pressure relief valves in kp/cm^2."

Membrane Tanks

Membrane tanks are to be designed as being subject to the requirements for Type A1 Independent tanks.

A.2.2.4 Germanischer Lloyd

Independent Tanks

"26.G.2.1 - Internal static load (internal overpressure) for gravity tanks is given by:

$$p_{is} = n \cdot h + p_o \qquad [t/m^2]$$

where h = distance in [m] from the surface of the liquid to the structural member under consideration (lower edge of plate, midpoint of unsupported span) for a filling-up ratio of 99 percent.

$n = \frac{1+\gamma}{2}$, where $\gamma \leq 1,0$ $[t/m^3]$

$n = \gamma$, where $\gamma > 1,0$ $[t/m^3]$

γ = specific gravity of the heaviest cargo

p_o = design vapor pressure according to E. 3. converted into [m WS] (i. e., head of water in meters)"

In addition, Type G3 gravity tanks are normally to be subjected to a hydraulic or hydropneumatic test.

Membrane Tanks

Membrane tanks Type G2 are normally to be subjected to a hydraulic or hydropneumatic test. These tests may be waived providing the same degree of safety can be achieved through (1) documentation of material properties relating to crack propagation and fatigue damage, (2) extensive stress calculation, (3) complete nondestructive testing, or (4) strict fabrication tolerance.

A. 2. 2. 5 Lloyds Register of Shipping

"3. 7116 The tanks shall be designed to withstand:
A test head of 2. 44 m of water above the top of the tank or 0. 61 m above the top of the hatch, whichever may be greater."

No other requirements with respect to particular tank types are given.

A. 2. 2. 6 Nippon Kaiji Kyokai

Static liquid head is combined with dynamic liquid head in the NK rules; therefore, see Section 2. 13 of this report. In addition, tanks are to be subjected to the following pressure tests described below:

Independent Tanks

"14.2.3-2 Type A and B independent tanks are to be subjected to pressure test and leak test of the following:

(1) Where a tank is hydrostatically tested, the tank is to be tested to the water head pressure up to the top plate of the tank (excluding the dome, which will be excluded hereinafter) in addition to either the pneumatic or hydrostatic pressure corresponding to a water head pressure of 2.45 m above the tank top plate or 0.6 m above the top of hatch opening from the tank top plate, or a pressure equal to the design vapor pressure of the tank, whichever is the greatest. Confirmation is to be made that there is no leakage and/or no harmful deformation under such pressure.

(2) Where a tank is not hydrostatically tested according to the requirements of preceding 1, the tank is to be hydrostatically-pneumatically tested and confirmation is to be made that there is no leakage and/or no harmful deformation under such pressure. This hydrostatic-pneumatic test is to be carried out to the water head pressure corresponding to the design internal pressure specified in 4.3.3 (2) at the tank bottom in addition to the pneumatic pressure specified in preceding (1)."

"14.2.3-3 Type C independent pressure vessel configuration type tanks are to be tested to the pressure of 1.5 times the tank design vapour pressure in addition to water head up to the tank top plate. Confirmation is to be made that there is no leakage and/or no harmful deformation under such pressure."

Membrane Tanks

"14.2.3-4 Tests for a membrane type tank are to be as follows:

(1) Tank hold boundaries containing a membrane type tank, are to be tested in accordance with the requirements of Sub-Para. 1, Art. 5, Chap. 2, Part 1 of the NK Rules. And where deemed necessary by the Society, the tank hold boundaries are to be also subjected to a leak tightness test such as pneumatic pressure test, etc.

(2) Membrane type tanks are to be subjected to the test which has been developed at the design stage specified in 4.9 and accepted by the Society."

"14.2.3-5 Tests for a semi-membrane type tank are to be generally in accordance with the requirements of preceding 2 or 4, according to the type of structure of the cargo containments."

A.2.2.7 United States Coast Guard

Independent Tanks

Pressure vessel tanks IPT, SPT are to be designed according to Marine Engineering Regulations or equivalent.

IIT and IST gravity tanks are to be designed to ABS scantlings or equivalent. The tanks must be static tested with a head of cargo at least equal to the highest level the liquid may attain plus the maximum venting pressure. In no case shall the head of cargo be taken to be less than 1.22 meters above the cargo hatch or expansion tank.

Membrane Tanks

The IMT tank is subject to the same load as the IST gravity tank.

Integral Tanks

The IGT tank is subject to the same load as the IST gravity tank.

A.2.2.8 International Association of Classification Societies

All tank designs are to be hydrodynamically or hydropneumatically tested according to the rules of the appropriate classification agency. The tests in general are to be designed to approximate, as far as possible, the design stress, and so that the pressure at the top of the tank is at least equal to the MARVS (Maximum Allowable Relief Valve Setting).

A.2.2.9 Summary of Static Liquid Head Criteria

All classification societies and the USCG require static hydraulic testing of the completed tanks. ABS, BV, and LR specify a minimum static head of water which is the greater of 2.4 m above the tank top plate or 0.6 m above the cargo hatch. Requirements in the NK rules are the same except that a head of water up to the tank top plate plus the maximum vapor pressure is used if it produces greater maximum pressures. The USCG defines the static test head in terms of the actual liquid cargo cargo (rather than water) and sets the minimum as 1.22 m above the cargo hatch. Static head tests are specially considered by DnV. IACS states that the static head tests should be according to the Classification Society rules but, in general, should be performed so that the stresses approximate, so far as possible, the design stresses.

In addition to the static head testing, most societies specifically require that a static liquid head be combined with the dynamic head in computing the tank internal design pressure head. ABS stipulates that the static test head be combined directly with the dynamic loads. Special equations are given by DnV and GL for the static pressure head which is to be combined with an "additional internal pressure head" due to ship accelerations. These equations account for the head due to vapor pressure and, for cargo less dense than water, the head is based on a density which is an average between that of the cargo and water. NK and IACS account for the static head by combining the gravitational and dynamic accelerations and using the actual cargo head when computing the tank dynamic internal pressures (see Section II. 2. 13).

Membrane Tanks

IACS, DnV, and USCG regulations for membrane tanks are the same as for independent gravity tanks. BV requires a leak detection test for the compartments adjoining the tanks. In this test, the compartments are filled to 2.4 m above the tank top. In addition, the tanks are also subjected to a static water head test corresponding to a liquid level h_a, above this tank top given in terms of the height of the tank and cargo density.

NK requires that the tank boundaries be subjected to a leak tightness test. In addition, semi-membrane tanks are to be hydro-pneumatically tested with an equivalent static head corresponding to a fill level of 2.45 m above the tank top. Confirmation is to be made that no harmful leakage or deformation occurs under this pressure.

Integral Tanks

USCG requirements for integral tanks are the same as for independent tanks.

A.2.3 STATIC DESIGN EXTERNAL PRESSURE

A.2.3.1 American Bureau of Shipping

Suitable means are to be provided to relieve the vapor pressure in the hold spaces should leakage of cargo occur.

Independent Tanks

"24.31.1 Independent, all-welded cargo tanks, suitably supported and securely anchored in position, are to be constructed in accordance with the requirements of 24.37. The arrangements for supporting and anchoring the tanks are to be adequate for the static and dynamic loads and are to include means to prevent flotation of empty tanks if the hold spaces be flooded. Independent tanks are to be designed to withstand, when empty, the external flooding which could occur with the ship at its designed load draft."

Membrane Tanks

No requirements.

A.2.3.2 Bureau Veritas

All tank types are to meet the following regulation. Additional regulations are specific to the various tanks.

"22-23-31 - Safety valves or equivalent devices are to be provided in order to avoid an overpressure in the space between cargo tank and secondary barrier in case of leakage of the tank."

Independent Tanks

For pressure cargo tanks, thickness is to be increased or stiffness is to be provided, if necessary, when the tanks are likely to be subjected to vacuum.

For self-supporting gravity tanks, in which there is no double hull nor double bottom, the attachments of tanks are to be such as to avoid the lifting of tanks assumed empty in case of flooding of the hold containing the tanks.

Membrane Tanks

"22-36-13 - Fastening of the integrated cargo tanks to the hull structure is to permit a sufficient resistance to a possible vacuum in the cargo tanks or to a possible overpressure in the insulation space taking into account the safety devices provided to limit the value of the vacuum or of the overpressure."

"22-37-16 - Where there is no double hull nor double bottom, the insulation is to be designed and built so as to keep its properties in case of flooding of the compartments containing the tanks."

A.2.3.3 Det norske Veritas

Independent Tanks

"6.B.203 - The design external pressure, p_{ed}, is to be based on the difference between the minimum internal pressure (maximum vacuum) and the maximum external pressure to which the tank may be subjected simultaneously. The design external pressure is to be based on the following formula:

$$p_{ed} = p_1 + p_2 + p_3 + p_4$$

p_1 = opening pressure of the vacuum relief valves. For tanks not fitted with vacuum relief valves, p_1 is to be specially considered, but is in general not to be taken less than 0,25 kp/cm^2.

p_2 = for tanks or part of tanks in completely closed spaced: the set pressure of the pressure relief valves for these spaces. Elsewhere $p_2 = 0$.

p_3 = external head of water for tanks or part of tanks on exposed decks. Elsewhere $p_3 = 0$.
$p_3 = a\ (bc - y)$ metres for tanks, type A.
$p_3 = 1.0\ (bc - y)$ for tanks, type B.
a, b, c and y are given in Chapter II, Sec. 15 C 100.

p_4 = compressive forces in the shell due to weight and contraction of insulation and weight of the shell, including corrosion allowance."

Membrane Tanks

Membrane tanks are subject to the same loads as independent tanks.

A.2.3.4 Germanischer Lloyds

Independent Tanks

No requirement given for pressure vessel tanks. For gravity tanks, the rules state:

"26.G.2.2 External static load (external overpressure) for gravity tanks

$$p_{as} = p_1 + p_2 \qquad [t/m^2]$$

p_1 = setting value of the vacuum relief valves [m WS]. For tanks not fitted with vacuum relief valves, p_1 is to be specially considered but is, in general, not to be taken less than 2.5 [m WS].

p_2 = for tanks or parts of tanks in completely closed spaces: the set pressure of the pressure relief valves for these spaces [m WS]. Elsewhere, $p_2 = 0$."

Membrane Tanks

No requirements.

A.2.3.5 Lloyds Register of Shipping

Independent and Membrane Tanks

" D.7111 Arrangements are to be provided to prevent excessive pressure coming on to the containment spaces either during service or in the event of leakage from the cargo tanks."

A.2.3.6 Nippon Kaiji Kyokai

Independent and Membrane Tanks

"3.3.3 The suitable devices to prevent tanks from floating in the hold spaces are to be filled or the hull structures are to be adequately strengthened, for the purpose of preventing the hull structures from a catastrophic failure by floating the tank when the tank hold is flooded.

"4.3.3(3) Design external pressure P_e due to the difference between minimum internal pressure and maximum external pressure of the tank is generally given by the following formula;

$$P_e = P_1 + P_2 + P_3 + P_4 + P_5 \ (kg/cm^2)$$

where:

P_1 = Set pressure of vacuum relief valve. For tanks not fitted with vacuum relief valve, it is to be at the discretion of the Society.

P_2 = Opening pressure of the devices for preventing overpressure in the spaces surrounding cargo tank, for tanks or part of tanks in completely closed spaces.

P_3 = External pressure due to head of water for tanks or part of tanks on exposed decks. Elsewhere $P_3 = 0$.

P_4 = Compressive forces in the shell due to weight of insulation and shell including its attachments.

P_5 = Compressive forces in the shell due to contraction of insulation, where deemed necessary according to insulation types."

A.2.3.7 United States Coast Guard

Independent Tanks

No specific regulations. IPT, SPT, IIT, and IST tanks are designed according to ABS scantlings or equivalent.

Membrane Tanks

The IMT tanks are to be designed to ABS scantlings or equivalent. In addition, prototype tank tests are required in which conditions representing the actual service life are thoroughly tested. These tests are to include internal and external pressure loadings. The tank shall be evacuated to the negative pressure setting of the service tank vacuum relief valve plus the pressure setting of the secondary barrier pressure relief valve.

Integral Tanks

No specific requirements. IGT tanks are designed according to ABS scantlings or equivalent.

A.2.3.8 International Association of Classification Societies

External design pressure loads are to be based on the difference between the minimum internal pressure (maximum vacuum) and the maximum external pressure to which any portion of the tank may be subjected simultaneously.

Independent Tanks

"5.7 Antiflotation chocks are to be provided for independent tanks. The antiflotation chocks are to be suitable to withstand an upward force caused by an empty tank in a hold space flooded to the load draught of the ship, without plastic deformation likely to endanger the hull structure."

Membrane Tanks

No additional requirements.

A.2.3.9 Summary of Static Design External Pressure Criteria

Independent Tanks

All agencies, except ABS and LR, provide guidelines for computing the external design pressure. These guidelines are based on the differential pressure to which any part of the tank may be exposed. For gravity tanks, GL bases the computation of design external overpressure (differential pressure) on the setting pressure of the pressure relief valves in the containment space and the setting pressure of the vacuum relief valves on the tanks. DnV and NK require the consideration of additional loads due to (1) external pressure caused by a head of water for tanks or tank portions on exposed decks, and (2) compression forces in the shell due to the weight of insulation and shell including its attachments.

In some cases, NK also requires consideration of compression forces in the shell due to contraction of insulation. IACS, although not specific, requires the pressure differential be calculated from the difference between minimum internal pressure and the maximum external pressure to which any part of the tank may be subjected simultaneously.

ABS and LR, while providing no specific guidelines for calculation of the net external pressure, do require that some means be provided to prevent excessive pressure from coming on to the containment space during service or in event of leakage of cargo from the tanks. And ABS specifically requires that independent tanks be designed to withstand the external pressure due to hold flooding at the design draft. USCG requires only that independent tanks be designed according to ABS scantlings, or equivalent.

Membrane Tanks

DnV, LR, NK, and IACS requirements for membrane tanks are the same as presented above for independent tanks. BV requires that the insulation be designed and built so as to keep its properties in case of flooding of the compartments containing the tanks. In addition, fastening of membrane tanks to the hull structure is to prevent a sufficient resistance to possible vacuum in the cargo tanks or a possible overpressure in the insulation space taking into consideration the safety devices used to limit the value of the vacuum or the overpressure. The USCG requires prototype tests of IMT membrane tanks which adequately model the anticipated service conditions a full-scale ship will encounter. These tests shall include internal and external pressure loadings. The tank will also be evacuated to the pressure corresponding to the tank vacuum relief setting plus the secondary barrier pressure relief setting. The ABS and GL rules give no requirements relating to static external pressure.

Integral Tanks

No specific requirements for integral tanks concerning static external pressure are given by the agencies.

A.2.4 WEIGHT OF TANK AND CONTENTS

A.2.4.1 American Bureau of Shipping

No requirements are given.

A.2.4.2 Bureau Veritas

Independent Tanks

"22-32-31 Pressure cargo tanks are to be supported by seatings, the details of which are to be submitted to the Administration. Supports are mainly to be designed to transmit to the hull the loads corresponding to the weight of the full tanks supplemented by the dynamic effects due to the motions of the ship, while avoiding that the tanks take part in bending of the ship and are subject to abnormal stresses due to deformation of the ship."

Self-supporting gravity tank supports and similar devices are to be designed and built to transmit the loads corresponding to the weight of the full tanks supplemented by the dynamic effects due to the motions of the ship to the hull structure (while limiting stress concentrations in the hull and tank structure).

In addition, rolling keys for self-supporting gravity tanks are to be designed with consideration of the transverse force F_{ℓ}, calculated from the maximum tank weight and simultaneous rolling, pitching, heaving accelerations.

Membrane Tanks

"22-37-14 For integrated tanks, the insulation is to be capable of transmitting to the structure of the ship the loads due to the cargo without deformations liable to induce, in the membrane, bending stresses for which it has not been designed."

A.2.4.3 Det norske Veritas

"B200 The static load due to 99% filling by volume of the tank with a cargo of design density is to be considered."

A.2.4.4 Germanischer Lloyd

Tanks together with their supports and other fixtures are to be designed taking into account proper combinations of the various loads including tank and cargo weight with corresponding reactions in way of supports.

A.2.4.5 Lloyds Register of Shipping

No requirements are given.

A.2.4.6 Nippon Kaiji Kyokai

Loads acting on tanks through tank supports (interaction forces between hull and tank) are to be obtained from internal pressure in consideration of the accelerations and the components of static weight due to ship motion. See also Section 2.13 for related information.

A. 2. 4. 7 United States Coast Guard

Rolling, pitching and heaving loads are to be calculated as described in Section 2.11 from the weight of the tank and its contents. Tanks and supports are to be capable of withstanding these loads.

A. 2. 4. 7 International Association of Classification Societies

"2.11 Tanks together with their supports and other fixtures are to be designed taking into account proper combinations of various loads including tank and cargo weight with the corresponding reactions in way of supports."

A. 2. 4. 9 Summary of Weight of Tank and Contents Criteria

Independent Tanks

BV requires that supports be designed to transmit to the hull the loads corresponding to the weight of the full tanks supplemented by the dynamic effects due to motions of the ship while limiting stress concentrations in the hull and tank structures. In addition, the transverse force, F_{ℓ}, on gravity tanks is to be calculated from the maximum tank weight and simultaneous rolling, pitching, and heaving accelerations. NK requires that the calculation of internal pressure include consideration of the components of accelerations and static weight of a fully loaded tank due to ship motions and inclinations. GL and IACS require that tanks and supports be designed to withstand the proper combination of static loads, loads due to elastic ship deflections, dynamic loads due to ship motions, sloshing loads, thermal loads, and loads due to weight of tank and contents. The USCG requires that the dynamic roll, pitch, and heave forces be calculated from the weight of the tank and its contents and the roll, pitch, and heave motions. DnV, although not specific, requires consideration of the static load due to 99% filling (by tank volume) with a cargo of design density. ABS and LR present no requirements.

Membrane Tanks

In addition to the requirements for independent tanks, BV requires that the insulation on membrane tanks be capable of transmitting to the ship hull the loads due to the cargo without deformations liable to induce excessive bending stress in the membrane. DnV, NK, USCG, and IACS requirements for membrane tanks are the same as for independent tanks. ABS and LR give no requirements concerning weight of tank and contents.

Integral Tanks

No specific requirements concerning weight of tank and contents for integral tanks are given by the agencies.

A.2.5 STILL-WATER HULL DEFLECTIONS

A.2.5.1 American Bureau of Shipping

"24.7 Still-water bending-moment calculations for the governing loaded and ballasted conditions are to be submitted."

Independent Tanks

"24.49.4 - Foundations for horizontal tanks ($P_o > 0.703$ kg/cm^2) are to be fitted at only two points in order to minimize throwing any local loads into the tank from the working of the vessel, or the supports are to be designed to absorb safely the normal deflections of the hull and tank. In addition to the foundation, chocks are to be provided to prevent shifting, but these chocks need not be in contact with the tank shell."

Membrane Tanks

No regulations.

A.2.5.2 Bureau Veritas

Generally the arrangement of ballast tanks is to be such that the still-water bending moment of the loaded or ballasted ship is as small as possible.

Independent Tanks

"22-32-31 - Pressure cargo tanks are to be supported by seatings, the details of which are to be submitted to the Administration. Supports are mainly to be designed to transmit to the hull the loads corresponding to the weight of the full tanks supplemented by the dynamic effects due to the motións of the ship, while avoiding that tanks take part in bending of the ship and are subject to abnormal stresses due to deformation of the ship."

Membrane Tanks

"22-37-14 - For integrated tanks, the insulation is to be capable of transmitting to the structure of the ship the loads due to the cargo without deformations liable to induce in the membrane bending stress for which it has not been designed."

A.2.5.3 Det norske Veritas

Independent Tanks

"B-204 Static forces imposed on the tank from deflection of the hull have to be considered."

Membrane Tanks

Membrane tanks are subject to the same requirements as Independent Tanks.

A.2.5.4 Germanischer Lloyd

"G.11 Tanks together with their supports and other fixtures are to be designed taking into account proper combinations of various loads including loads corresponding to the elastic ship deflections."

A.2.5.5 Lloyds Register of Shipping

Independent Tanks

"7205 Pressure vessel tanks in which the vapor pressure, $P_o > 0.70$ kg/cm^2, will have seatings designed to ensure uniform support to the pressure vessel having due regard to the deflections of the hull structure in a seaway."

Membrane Tanks

No specific requirements.

A.2.5.6 Nippoń Kaiji Kyokai

With regard to hull deflections for all tank configurations, the rules state:

"4.3.3-(4) Loads acting on tanks through tank supports (interaction forces between hull and tank) are to be obtained from the following.

(a) Hull deformations due to vertical bending moment, horizontal bending moment and torsional moment in waves, and vertical still-water bending moment.

(b) Internal pressure in consideration of accelerations and the components of static weight due to ship motions.

(c) Water pressure distribution on the hull where deemed necessary."

Independent Tanks

Concerning Type A independent prismatic tanks, the rules state:

"4.5.2-3(1) Structural analysis of horizontal, longitudinal and transverse girders is to be carried out by a frame work analysis, a finite element method or equivalent methods accepted by the Society, at least taking into account the effects of bending moment, shearing force, axial force and torsional moment. The equivalent stress of the combined primary stress as a result of such an analysis is not to exceed the value specified in 4.4.2(2).

(2) For preceding (1), the interaction force between the hull and tank specified in 4.3.3(4) is to be calculated taking into account the reaction due to the deflection of the double bottom and tank bottom, where the tank bottom and the double bottom is coupled by the supporting structure. The loads specified in 4.3.3(4) (a) to (c) may be obtained from:

(a) For vertical wave bending moment, the approximate formula given in 4.3.2-4.

(b) For local vertical bending moment of a tank hold due to the dynamic external water pressure distributions around the hull, the approximate formula given in 4.3.2-3.

(c) For internal pressure distribution of the tank, the formula for internal pressure given in 4. 3. 3(2) (a) in which the approximate acceleration specified in 4. 3. 2-2 is used."

With respect to Type B independent prismatic tanks, the rules state:

"4.6.2-1. For the type B independent prismatic type tank specified in this Section, it is pre-supposed that the scantlings of the tank's strength members are based on an exact stress and deformation analysis of the tank."

"4.6.2-2. For the evaluation of the overall structural response of the tank, a three-dimensional analysis is to be carried out by a frame work analysis and/or a finite element analysis or equivalent methods, taking into account the effect of the hull deformations due to vertical and horizontal bending moments and torsional moment and the local deformation. The model for the analysis is to include the tank with its supporting structures as well as a reasonable part of the hull."

Concerning Type B independent pressure vessel configuration tanks, the rules state:

"4.7.2-1. Type B independent pressure vessel configuration tank specified in this Section are to be based on the exact analysis of the stresses and deflections at any place in the tanks as well as its supporting structure."

"4.7.2-2. For the evaluation of the overall structural response of the tank, the tank including its supporting structure is to be analyzed by a finite element analysis and/or a shell theory or equivalent method, taking into account the effect of the hull deformation due to vertical and horizontal bending moments and torsional moment."

Membrane Tanks

"4.9.2-1. A membrane tank is to be so designed as to withstand sufficiently all the static, dynamic and thermal stresses through the total life of a ship, and not to result in excessive plastic deformation and fatigue failure."

A.2.5.7 United States Coast Guard

Independent Tanks

The IPT and SPT tanks are to be designed in accordance with Marine Engineering Regulations.

"4.a.(3) The IIT and IST tanks on board vessels of U. S. registry must be designed to the minimum appropriate standards of ABS or equivalent."

"4.a.(4) For foreign vessels, the IIT and IST tanks must be designed in accordance with the requirements of a cognizant classification society and must have the specific approval of the Coast Guard."

Membrane Tanks

Moderate scale fatigue testing is required per Section 3.c.(2)(a) of the USCG rules. Among other restraints, it is required that the structure be statically prestressed in tension to the maximum amount caused by cargo cooling, static head, pressure of cargo and still-water hull deflections. In addition, the structure is to be cycled above and below the static level an amount equivalent to that caused by maximum at-sea hull deflection plus the maximum caused by the dynamic loading criteria.

Integral Tanks

The IGT tanks are to be designed to the same requirements as the IIT and IST independent tank designs.

A.2.5.8 International Association of Classification Societies

With respect to tank supports for all tank designs, the rules state:

"5.1 Cargo tanks are to be supported by the hull in a manner which will prevent bodily movement of the tank under static and dynamic load while allowing contraction and expansion of the tank under temperature variations and hull deflections without undue stressing of the tank and of the hull."

Independent Tanks

"5.7 Antiflotation chocks are to be provided for independent tanks. The antiflotation chocks are to be suitable to withstand an upward force caused by an empty tank in a hold space flooded to the load draught of the ship, without plastic deformation likely to endanger the hull structure."

A.2.5.9 Summary of Still-Water Hull Deflection Criteria

Independent Tanks

DnV requires that static forces (due to hull deflections) imposed on the tank be considered. BV and GL require that the design of tanks and their supports take into account proper combinations of various loads including ship hull deflections and the weight of the full tank. BV, in addition, requires that the tanks do not take part in bending of the ship and are not subject to abnormal stress due to ship hull deformations. ABS (horizontal pressure tanks) and LR (pressure tanks), $P_o > 0.70$ kg/cm^2, require that the tank supports provide uniform support to the pressure vessel without introducing moments in the tank due to hull deflections in the seaway. The USCG requires that independent tanks be designed according to the Marine Engineering Regulations or equivalent. IACS requires that cargo tanks be restrained from bodily movement under static and dynamic loads, while allowing expansion and contraction of the tank under temperature variations and hull deflections without undue stressing of the tank and hull. NK presents no requirements concerning hull deflections; however, requirements for hull bending moment are given. Loads acting on the tanks through tank supports due to bending and torsional moments are to be calculated taking into account deflections of the double bottom and tank bottom where the tank and hull bottoms are coupled by a supporting structure.

Membrane Tanks

BV requires that the insulation transmit to the ship structure the loads due to the cargo without introducing deformations in the membrane which would cause excessive bending stress. NK requires that the membrane be designed to withstand sufficiently all static, dynamic and thermal stresses for the entire ship lifetime without resulting in plastic deformation or fatigue failure. The USCG requires that fatigue tests be performed on model LNG tanks prestressed in tension to the maximum amount caused by cargo cooling, static head, cargo pressure and the still-water hull deflection, with the hull deflections in the sea-way determining the cycling amplitude. IACS and DnV requirements for membrane tanks are the same as for independent tanks. ABS, GL, and LR present no requirements for membrane tanks.

Integral Tanks

USCG requires that the IGT tank configuration be designed according to the same requirements as the IIT and IST independent tank configurations. No requirements concerning still-water bending moments are given by the other agencies.

A.2.6 STATIC INCLINATION

A.2.6.1 American Bureau of Shipping

No regulations specifically for LNG tanks or supports.

A.2.6.2 Bureau Veritas

No specific regulations

A.2.6.3 Det norske Veritas

Independent Tanks

"B401 The tank with supports is to be designed for a static inclination of 30° without exceeding design stresses for for static plus dynamic loads. The tank with supports is to withstand a force corresponding to a longitudinal acceleration of $a_x = 0.5\,g$ without failure. These loads need not be combined with wave induced loads."

Membrane Tanks

Membrane tanks are to be designed for the same load as independent tanks.

A.2.6.4 Germanischer Lloyd

With respect to static inclination, all tank designs are subject to the same load criteria:

"26-G.4.1 Tanks with supports are to be designed for loads corresponding to a static inclination of 30°."

Static inclination and collision acceleration loads need not be combined with the wave-induced loads or with each other.

A.2.6.5 Lloyds Register of Shipping

"D-7106 The extent of the secondary barrier is to be such that the liquefied gas will not come into contact with the hull structure in the event of failure of one cargo tank with the ship heeled to an angle of 30°."

A.2.6.6 Nippon Kaiji Kyokai

"4.3.3(5) Loads acting on tank supports specified in 4.2.6 are to be obtained from the following:

(a) Interacting forces between hull and tank, specified in preceeding (4).

(b) Loads acting on tank supports for heel of 30°.

(c) Collision loads acting on the tank corresponding to 0.5 g from forward and 0.25 g from aft, where g is acceleration of gravity."

Membrane Tanks

Membrane tanks are subject to the same static inclination load as independent tanks.

A.2.6.7 United States Coast Guard

With regard to static inclination, all tank designs are subject to the same load criteria:

"IV F.2.C.(1) The stability in the final condition of flooding may be regarded as sufficient if the righting lever curve has a minimum range of 20° beyond the position of equilibrium in association with a residual righting lever of at least 100 mm (4 inches)."

"IV F.2.C.(2) The angle of heel in the final condition of flooding should not exceed 15°, except that if no part of the deck is immersed, an angle of heel up to 17° may be accepted."

A.2.6.8 International Association of Classification Societies

"5.2 The tanks with supports are to be designed for a static inclination of 30° without exceeding allowable stresses given in 4. The static inclination need not be combined with wave-induced loads or with collision loads."

"6.15 The functions of the secondary barrier are to be ensured assuming a static angle of heel equal to 30°."

A.2.6.9 Summary of Static Inclination Criteria

Independent and Membrane Tanks

IACS, DnV, GL, LR, and NK all require independent tanks to be designed to withstand a static inclination of 30°. LR and IACS further require that the secondary barrier be of sufficient extent to contain the cargo at 30° heel. The USCG allows a maximum heel angle of 15° during final condition of flooding; if no part of the deck is immersed, a heel angle of 17° may be acceptable. ABS and BV present no requirements concerning static inclination for independent or membrane tanks.

Integral Tanks

No specific requirements concerning static inclination are given by the agencies.

A.2.7 COLLISION LOADS

A.2.7.1 American Bureau of Shipping

Independent Tanks

No regulations for a pressure vessel tank with a design vapor pressure $P_o \leq 0.703$ kg/cm^2. With respect to pressure vessel tanks with a design vapor pressure given by $P_o > 0.703$ kg/cm^2, the rules state:

"24.49.2 In addition to the pressure requirements as basis for design as given in Section 32, the cargo pressure-container foundations and securing arrangements are to be designed to withstand the dynamic loadings given in 24.37.1b (accelerations at tank C.G.) and are also to be capable of withstanding the forces resulting when a cargo hold is flooded with the cargo tanks empty. The supports and securing arrangements are also to be capable of withstanding a longitudinal acceleration of 0.5 g with the tank filled with liquified gas."

Membrane Tanks

No requirements.

A.2.7.2 Bureau Veritas

Independent Tanks

With regard to collision loads on self-supporting gravity tanks, the rules state:

"22.34-41 Pitching keys are to be designed to prevent movement of the tanks due to pitching, also to an acceleration corresponding to a collision. Normally, their design is to take into account a longitudinal force G equal to 0.3 times the maximum weight of the full tank."

"22.34-42 Where upper pitching keys are provided, they are to be determined from a longitudinal force equal to 0.4 G while a longitudinal force equal to 0.8 G is considered for the lower keys."

Membrane Tanks

With respect to insulation for membrane tanks, the rules require protection of insulation against moisture penetration and against shocks by a suitable means.

A.2.7.3 Det norske Veritas

Independent Tanks

"B401 The tank with supports is to be designed for a static inclination of 30° without exceeding design stresses for static plus dynamic loads. The tank with supports is to withstand a force corresponding to a longitudinal acceleration of $a_x = 0.5$ g without failure. These loads need not be combined with wave-induced loads."

Membrane Tanks

Membrane tanks are subject to the same collision load regulations as independent tanks.

A.2.7.4 Germanischer Lloyd

With respect to collision loads, all tank designs are subject to the following regulation:

> "G.4.2 The tank supports are to be designed for a collision force corresponding to a longitudinal acceleration of 0.5 g from forward and 0.25 g from aft."

A.2.7.5 Lloyds Register of Shipping

Independent Tanks

No guidelines given for tanks with a design vapor pressure of $P_o \leq 0.70\ kg/cm^2$. For tanks with a design vapor pressure given by $P_o > 0.70\ kg/cm^2$, the following regulation applies:

> "Chapter D-7206 The supports and securing arrangement should also be capable of withstanding a longitudinal acceleration of 0.5 g."

Membrane Tanks

Membrane tanks are specially considered.

Integral Tanks

No specific guidelines for carriage of LNG.

A.2.7.6 Nippon Kaiji Kyokai

Independent Tanks

> "4.3.3(5) Loads acting on tank supports specified in 4.2.6 are to be obtained from the following.
>
> (a) Interaction forces between hull and tank, specified in preceding (4).
>
> (b) Loads acting on tank supports for a static heel of 30°.
>
> (c) Collision loads acting on the tank corresponding to 0.5 g from forward and 0.25 g from aft, where g is acceleration of gravity."

Membrane Tanks

Membrane tanks are subject to the same requirements as independent tanks.

Integral Tanks

No requirements for carriage of LNG.

A.2.7.7 United States Coast Guard

Guidelines for collision protection are in terms of survivable damage. No accelerations associated with collision are specified. However, the following tanks are to be designed to ABS scantlings:

Independent tanks:	IIT (perhaps IST)
Membrane tanks:	IMT
Integral tanks:	IGT

A.2.7.8 International Association of Classification Societies

Suitable supports are to be provided to withstand a collision force corresponding to 0.5 g from forward and 0.25 g from aft without deformation likely to endanger the structure.

A.2.7.9 Collision Load Summary

Independent Tanks

ABS (pressure vessel tanks $P_o > 0.702$ kg/cm^2), DnV and LR require that the tanks and supports be designed to withstand a force corresponding to a longitudinal acceleration of 0.5 g. GL, NK, and IACS require that the tanks and supports be capable of withstanding a longitudinal acceleration of 0.5 g forward and 0.25 g aft. BV requires that pitching keys be designed to prevent movement of tanks during a collision force corresponding to a longitudinal acceleration equal to 0.3 times the maximum weight of the full tank. In addition, BV requires that the design of upper pitching keys include consideration of a longitudinal force of 0.4 g while sizing of lower keys is controlled by a longitudinal force of 0.8 g. USCG requirements are in terms of survivable damage; no accelerations are given.

Membrane Tanks

BV requires only that the insulation on membrane tanks be protected from shocks by a suitable means. DnV, GL, NK, USCG, and

IACS requirements for collision loads on membrane tanks are the same as for independent tanks. ABS and LR have no specific requirements concerning collision loads on membrane tanks.

Integral Tanks

No specific regulations concerning collision loads for integral tanks are given by the agencies.

A.2.8 THERMAL LOADS

A.2.8.1 American Bureau of Shipping

"24.13.5 All cargo carriers are to be supported and held in position in such a manner that neither the tanks nor the hull structure are subjected to excessive stresses as a result of thermal expansion, or the normal motion of the vessel, or both."

In addition, all tank designs are to be tested according to the following regulations:

"24.1.2 All primary containers for low-temperature cargos, the insulation and the cargo-handling equipment are to be tested under service conditions prior to final action in regard to classification. The primary containers are to be filled to the normal capacity level with cargo at the minimum service temperature."

A.2.8.2 Bureau Veritas

In general, the design of supports and attachments are to be so as to avoid the temperature of the ship structure in way of the supports, adjacent hull, and similar devices being lowered below the values allowed for the steel used. With respect to steels, the rules state:

"22-24-21 Steels defined in Chapter 25 for ship construction may be used for the construction of secondary barriers or of other parts liable to reach a low temperature provided that this temperature, determined while assuming an external conventional temperature of +5°C, does not fall below the following values:

	Normal Service	Tank Leakage
Grade A	0°C	-10°C
Grade D	-10°C	-30°C
Grade E	-20°C	-50°C "

"22-24-22 In the case of steels for low temperature defined in Chapter 25, the minimum service temperature is to be at least 5°C above the temperature required for the impact test of the corresponding grade of steel."

A. 2. 8. 3 Det norske Veritas

Regarding thermal gradients, the regulations state:

"B501 Transient thermal loads during cooling-down periods are to be considered for tanks intended for cargos with a boiling point below -50°C. Transient thermal loads are also to be considered when the cargo temperature exceeds 100°C."

"B502 Stationary thermal loads are to be considered for tanks where design, supporting arrangement and operating temperature may give rise to significant thermal stresses."

In addition, piping systems for liquefied gas tankers are subject to the following regulation:

"C-302 All pipes are to be mounted in such a way as to minimize the risk of fatigue failure due to temperature variations of the hull girder in a seaway. If necessary, they are to be equipped with expansion bends. Use of expansion bellows will be specially considered. Slide type expansion joints will not be accepted."

A. 2. 8. 4 Germanischer Lloyd

All tank configurations are subject to the following requirements:

"26.G.5.1 Transient thermal loads during cooling down periods are to be considered for tanks intended for cargos with boiling points below -50°C."

"26.G.5.2 Stationary thermal loads are to be considered for tanks where design, supporting arrangement and operating temperature may give rise to significant thermal stresses."

Regarding the secondary barrier, the rules state:

"26.K.3.2 The full secondary barrier is to be designed in such a way that it will prevent the hull steel temperatures in the event of leakage from falling below the temperature for which the hull steel is suitable under emergency conditions."

The design and reference temperatures are given in the following requirements:

"26.F.1.1 Design temperature is the minimum temperature at which cargo may be loaded and/or transported in the cargo tanks. Moreover, the design temperature is not to be taken higher than:

$$t_o = t_w - 0.25\ (t_w - t_b) \quad [^\circ C]$$

where:

t_w = boiling temperature of the cargo at the normal working pressure of the cargo tank, but not to be taken higher than $0^\circ C$.

t_b = boiling temperature of the cargo at atmospheric pressure $[^\circ C]$"

"26.F.1.2 t_o need not be taken less than t_w if reliable arrangements are provided so that the temperature cannot be lowered below t_w."

"26.F.2.1 Reference temperature is the maximum temperature at which cargo may be transported under the most unfavorable conditions."

"26.F.2.2 For pressure tanks, the reference temperature is the temperature which shall not be exceeded during operation. Generally, the reference temperature is $45^\circ C$. However, lower reference temperatures may be accepted for ships operated on restricted areas or on voyages of limited duration and account may be taken in such cases of a possible insulation of the tanks. On the other hand, higher values of the reference temperature may be required for ships permanently operated in areas of high ambient temperature. Respective remarks will be entered into the Certificate."

A.2.8.5 Lloyds Register of Shipping

"D 7118 The tanks are to be supported on substantial foundations arranged to avoid excessive concentration of load on the ship's structure or on the tank. Provision is to be made for the thermal contraction of the tanks on cooling from ambient to service temperature and arrangements are to be made to control movement of the tanks when the vessel is rolling and pitching."

A.2.8.6 Nippon Kaiji Kyokai

All tank configurations are subject to the following criteria:

"4.3.3(6) For tanks intended for cargos with a boiling point below -50°C specified in the Rules, thermal loads due to temperature differences and/or irregular temperature distributions in tank structures including tank supports are to be considered at the following conditions, where deemed necessary.

(a) A transitional condition where temperature distribution in tanks abruptly changes at precooling, loading, etc.

(b) A stationary condition where temperature distribution in tanks abruptly changes in the direction of tank depth at partly loaded or ballast conditions, etc.

(c) A stationary condition where temperature distribution abruptly changes in the direction of thickness of tank plates at full loaded condition, etc. At a connecting part of two kinds of materials having different thermal expansion coefficients, the thermal loads due to the difference of thermal expansion are to be considered."

A.2.8.7 United States Coast Guard

Independent Tanks

"4.a (6) Design stress for the IIT tank must be computed based on the sum of the following:

(a) Static cargo head to 4 feet above the tank dome top.

(b) Maximum pressure relief valve setting.

(c) Full dynamic loads as discussed in A. i. of this section.

(d) Full tank thermal stresses.

(7) In addition to (6), the sum of stresses due to:

(a) Relief valve set pressure.

(b) Slack tank dynamic loads.

(c) Empty tank thermal stress, especially accounting for the vertical gradient in the tank filled with cold cargo vapor (a 150°F thermal gradient is a reasonably conservative figure in absence of any other data)

may not exceed design stress."

Membrane Tanks

No specific guidelines for thermal gradients are given. However, prototype tanks are to be built and tested. Features of the test should include thermal shock and gradients:

"3. c. (4)(c)((19)) Cold shock it through a sufficient number of cycles (but in no case fewer than three) in order to obtain consistent and reliable thermal gradient data and overstress indications."

In addition, see A. 2. 5. 7 regarding thermal gradient requirements of still-water hull deflections.

Integral Tanks

No specific requirements.

A. 2. 8. 8 International Association of Classification Societies

Tanks together with their supports and other fixtures are to be designed taking into account:

"2.61 Transient thermal loads during cooling down periods are to be considered for tanks intended for cargos with a boiling point below -55°C.

"2.62 Stationary thermal loads are to be considered for tanks where design, supporting arrangement and operating temperature may give rise to significant thermal stresses."

A.2.8.9 Thermal Gradients Summary

Independent Tanks

DnV, GL, NK, and IACS require that tank designs take into account transient thermal loading during cool-down periods (for tanks carrying cargos with a boiling point below -50°C; IACS allows -55°C). In addition, stationary thermal loads are to be considered where tank design, supporting arrangement and operating temperature may give rise to significant thermal gradients (i.e., normal temperature distribution in tanks changes because ship is operating in ballasted or partially filled condition). BV and GL require that the secondary barrier be designed to prevent the hull steel temperatures from falling below the values for which the hull steel is suitable under emergency conditions. The USCG requires that the design stress for IIT tanks be computed from the sum of pressures, dynamic loads and full tank thermal stress. In addition, the sum of stresses due to pressures, slack tank dynamic loads, empty tank thermal stresses, is to be computed. For thermal stresses, a vertical gradient in the tank filled with cold vapor (an 83°C gradient is a reasonably conservative value if no other data is available) must be considered. LR requires only that provision be made for thermal contraction of tanks during cool-down. ABS requires that tanks be supported in such a manner as to prevent excessive stress in the tanks or hull as a result of thermal expansion, and that all tanks be tested under service conditions with cargo at minimum service temperature.

Membrane Tanks

For all regulating agencies, except USCG, requirements for thermal gradients on membrane tanks are the same as for independent tanks. USCG requires that prototype membrane tanks be built and tested. Tests should include enough thermal shock cycles to obtain consistent thermal gradient data and overstress indications.

Integral Tanks

No requirements for carriage of LNG in integral tanks.

A.2.9 WAVE-INDUCED LOADS

A.2.9.1 American Bureau of Shipping

No specific regulations are given, but related regulations are included in 2.11 - Accelerations at Tank Center of Gravity.

A.2.9.2 Bureau Veritas

Independent and Membrane Tanks

The direct determination of a ship's motion at sea, where applicable, is to be made in consideration of ship's speed, a sea representation, and the navigating conditions.

"5-36-11 - The applicable ship's speed in calculations is the contract service speed."

"5-36-12 - Where a ship receives the sea ahead, with angles of incidence ranging between $-45°$ and $+45°$ inclusive, it is generally admitted that the speed is reduced by 40% when the significant heights are greater than 5 meters. No reduction is applicable for other angles of incidence."

"5-36-21 - A sea condition is represented by a Moskowitz-Pierson spectrum defined by:

- the significant height, in meters,
- the mean apparent period, in seconds.

For the purpose of studying the long term behavior of a ship, a discrete family of unidirectional spectra is considered, defined in terms of the course followed by the ship. When no course is specified, a family of spectra is considered derived from the Roll Tables for the North Atlantic together with the Hogben and Lumb compilation."

"5-36-22 - The existence probability of each sea condition is determined in terms of the course followed by the ship."

"5-36-31 - A discrete distribution of angles of encounter of the ship for each sea condition is considered."

"5-36-32 - The probability of encounter along a given direction is determined in terms of the course followed by the ship."

A.2.9.3 Det norske Veritas

Calculation of the wave-induced loads is not required for Type A-1 tanks.

Type A-II independent tanks (and in some cases Type B tanks) are subject to the following wave-induced load criteria:

"6.B.902 The loads for design against plastic deformations and buckling are to be taken as the most probable largest loads in 10^8 wave encounters (probability level $Q = 10^{-8}$) for a ship operating on the North-Atlantic. The wave loads are to be determined according to accepted theories, model tests or full-scale measurements. All heading angles are to be given the same probability of occurrence, and speed reduction in heavy weather may be taken into account. All types of wave-induced loads and motions exerted by hull and cargo on the tank structure are to be considered. Generally, these types of loads are:

- Vertical, transverse and longitudinal acceleration forces.
- Internal liquid pressure in the tank (full and partially full).
- External water pressure on the hull.
- Vertical and horizontal bending of the hull girder.
- Torsion of the hull girder."

Concerning Type B tanks, the rules state:

"6.E.103 In the design of the tank, the dynamical loads due to the ship's motions in a seaway are to be taken into account. For tanks supported in such a way that the deflection of the hull transfer significant stresses to the tank, the wave-induced loads may be required to be calculated as given in B 902. For saddle-supported tanks and other tanks, where a calculation of loads according to B 902 is not required, design accelerations given in B 301 are to be used. For saddle-supported tanks, the supports are also to be calculated for the most severe resulting acceleration. The most probable resulting acceleration in a given direction β may be found as shown in Fig. 6. The half axes in the 'acceleration ellipse' may be found from the formulae given in B 301."

Membrane Tanks

Membrane tanks are subject to the same requirements as Type A-II independent tanks.

A.2.9.4 Germanischer Lloyd

Independent and Membrane Tanks

"26.G.1.2. The calculations of internal and external dynamic loads due to ship's motion in the seaway will usually be carried out by this Society. The total load is the sum of static and dynamic loads."

"26.G.3.1 The determination of dynamic loads is to take account of the long term distribution of ship motions including the effects of surge, sway, heave, roll, pitch, and yaw on irregular seas."

"26.G.3.2 On ships operating in unrestricted service, for the design of tanks and their supports against plastic deformation and buckling, the statistically expected most probable largest sea load in the ship's lifetime shall be taken. All kinds of wave loads are to be taken into account, such as:

a) inertia forces due to vertical, transverse and longitudinal accelerations (see also 3.6).
b) internal dynamic heads in tanks when partly or completely filled with liquids.
c) external dynamic loads on the hull.
d) loads due to vertical and horizontal bending as well as due to torsion."

"26.G.3.3 Ships for restricted service will be given special consideration."

A.2.9.5 Lloyds Register of Shipping

No specific regulations are given, but related regulations are included in 2.11 - Accelerations at Tank Center of Gravity.

A.2.9.6 Nippon Kaiji Kyokai

Independent Tanks

Direct wave-induced load calculations for Type A independent prismatic tanks are not required. The use of approximate formulas is allowed.

For Type B independent prismatic tanks, the following guidelines apply:

"4.3.2-1. Dynamic loads (wave induced loads) are to be taken from a long-term distribution under short-crested irregular waves in an assumed service area for a ship. And from such distribution the maximum expected value is obtained as the most probable largest load which the ship will experience and the load spectrum against fatigue is obtained as the most probable largest load spectrum. The long-term distribution is to be determined according to a direct calculation by computer programmes including accepted theories and statistical estimations or results of model tests or full-scale measurements statistically examined.

(1) Conditions of the direct calculations are as follows:

(a) The total life of ship is about 20 years.
(b) Indication of waves in the service area is to be determined according to long-term observation data for about 10 years. If the data are not available, the long-term observation data on the North Atlantic may be used.
(c) Energy spectrum of wave is to be generally as given in the following formula.

$$[f(\omega)]^2 = 0.11\, H^2 \omega_1^{-1} (\omega/\omega_1)^{-5} \exp[-0.44\, \omega/\omega_1)^{-4}]$$

where

ω_1 = $2\pi/T$.

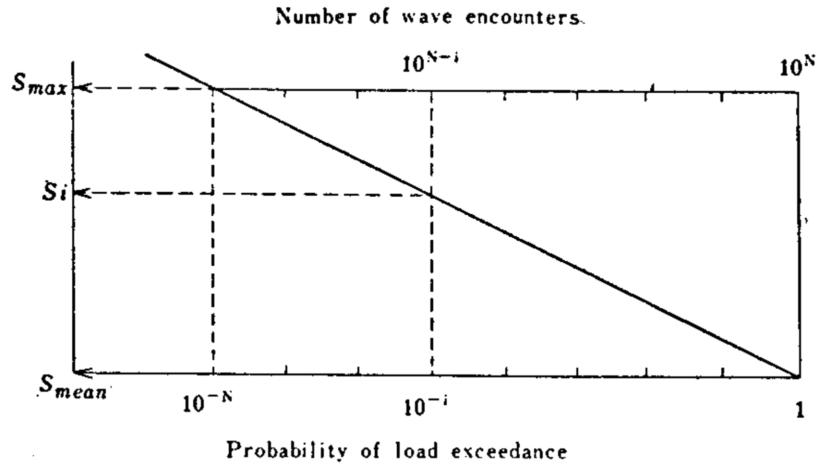
H = Significant wave height, (m).

T = Mean period of wave, (sec).

The wave spectrum is to be assumed as $\cos^2 x$ distribution within the range from $+\pi/2$ to $-\pi/2$ against the mean progressing wave direction.
(d) All heading angles are to be given with the same probability of occurrence.
(e) Speed reduction, etc. in heavy weather may be taken into account at the discretion of the Society.

(2) A load spectrum may be taken as a straight line approximately shown in the Fig. 4.3.2-1 (2), provided that the maximum expected value (S max) at a probability level $Q = 10^{-N}$ is estimated by suitable method subject to the satisfaction of the Society: In this case,,the total number of wave encounters is 10^N for a ship operating. "

The requirements for Type A, B, and C pressure vessel configuration tanks are the same as for Type B independent prismatic tanks.



NK Fig. 4.3.2-1 (2) Long-term Wave-induced Load Spectrum

Membrane Tanks

The requirements for membrane and semi-membrane tanks are the same as for Type B independent prismatic tanks.

A. 2. 9. 7 United States Coast Guard

No specific regulations are given, but related regulations may be found in A. 2. 11 - Accelerations at Tank Center of Gravity.

A. 2. 9. 8 International Association of Classification Societies

Independent and Membrane Tanks

"2.41 The determination of dynamic loads is to take account of the long term distribution of ship motions including the effects of surge, sway, heave, roll, pitch and yaw on irregular seas the ship will experience during her operating life (normally taken to correspond to 10^8 wave encounters). Account may be taken of reduction in dynamic loads due to necessary speed reduction and variation of heading when this consideration has also formed part of the hull strength assessment."

"2.42 For design against plastic deformation and buckling, the dynamic loads are to be taken as the most probable largest loads the ship will encounter during her operating life (normally taken to correspond to a probability level of 10^{-8}).

"2.43 When design against fatigue is to be considered, the dynamic spectrum is determined by long term distribution calculation based on the operating life of the ship (normally taken to correspond to 10^8 wave encounters). If simplified dynamic loading spectra are used for the estimation of the fatigue life, these are to be specially considered by the Classification Society."

A.2.9.9 Wave-Induced Loads Summary

Independent and Membrane Tanks

BV, DnV, GL, NK, and IACS specify that wave-induced dynamic loads[12] are to be calculated from the long-term distribution of ship motions on irregular seas experienced by the ship in its operational lifetime (normally taken to be 10^8 wave encounters). Except for GL, these agencies may allow a reduction in speed or a variation in heading during heavy weather which will reduce the calculated dynamic loads to some extent. BV is most specific here; speed reductions of 45% are allowed in head seas when the angle of incidence is between $\pm 45°$, providing the significant wave height is greater than 5 meters. BV and NK allow the calculation of wave-induced loads be based on the anticipated ocean service region, or if the service region is undesignated or unlimited, BV and NK, like DnV, require that the calculations be based on the properties of the North Atlantic. GL and IACS do not specify the ocean area the calculations are to be based on. NK specifies that the load spectrum can be approximately represented by a straight line providing the maximum expected value S_{max} at a probability level 10^{-N} is estimated by a suitable method (N is the number of wave encounters). The other agencies do not specify the method to obtain the load spectra. ABS, LR, and USCG give no information directly concerning wave-induced loads, but related information can be found in Section 2.11.

Integral Tanks

DnV, GL, and IACS requirements for wave-induced loads on independent tanks also apply for integral tanks. The other agencies present no specific requirements for integral tanks.

[12] Generally, wave-induced loads refer to vertical, longitudinal, and transverse ship motions and accelerations, external hull pressure, vertical and horizontal bending of the hull girder and torsion.

A.2.10 DYNAMIC HULL DEFLECTIONS

A.2.10.1 American Bureau of Shipping

Independent Tanks

No specific regulations are given, but with respect to supports, the rules state:

"24.49.4 Foundations for horizontal tanks are to be fitted at only two points in order to minimize throwing any local loads into the tank from the working of the vessel, or the supports are to be designed to absorb safely the normal deflections of hull and tank. In addition to the foundation, chocks are to be fitted to prevent shifting, but these chocks need not be in contact with the tank shell."

Membrane Tanks

No specific regulations for membrane tanks are given.

A.2.10.2 Bureau Veritas

No specific requirements for the consideration of dynamic hull deflections on LNG tanks are given.

A.2.10.3 Det norske Veritas

Independent Tanks

Type A-1 independent tank bulkhead plating and stiffeners are to be given scantlings according to Chapter II, Section 14. In addition, the following requirements apply:

"6.C.201 For webs, girders and stringers, a structural analysis is to be carried out to ensure that the stresses are acceptable. Calculation methods applied are to take into account the effects of bending, shear, axial and torsional deformations as well as the hull/cargo tank interaction forces, due to the deflection of the double bottom and cargo tank bottom."

"6.C.202 The following loads and stresses are to be taken into consideration:

- Static loads according to B 200.
- Dynamic additional loads due to the ship's movement in a seaway. See B 300 and C203-208.
- Thermal stresses."

Type A-II independent tanks are subject to the following regulation:

"6.B 801 The calculations are to be based on the most severe realistic loading conditions with the ship fully or partly loaded."

"6.B 901 The loads given in 902, 903 and 904 are to be used for tanks, Type A-II, and in special cases for tanks, Type B."

"6.B 902 The loads for design against plastic deformations and buckling are to be taken as the most probable largest loads in 10^8 wave encounters (probability level $Q = 10^{-8}$) for a ship operating on the North-Atlantic. The wave loads are to be determined according to accepted theories, model tests or full-scale measurements. All heading angles are to be given the same probability of occurrence, and speed reduction in heavy weather may be taken into account. All types of wave-induced loads and motions exerted by hull and cargo on the tank structure are to be considered. Generally, these types of loads are:

- Vertical, transverse and longitudinal acceleration forces.
- Internal liquid pressure in the tank (full and partially full).
- External water pressure on the hull.
- Vertical and horizontal bending of the hull girder.
- Torsion of the hull girder."

Pressure vessel tanks, Type B, are subject to the following requirements:

"3.E 103 In the design of the tank, the dynamical loads due to the ship's motions in a seaway are to be taken into account. For tanks supported in such a way that the deflection of the hull transfer significant stresses to the tank, the wave-induced loads may be required to be calculated as given in B 902. For saddle-supported tanks and other tanks, where a calculation of loads according to B 902 is not required, design accelerations given in B 301 are to be used."

Membrane Tanks

Membrane tanks are subject to the same requirements as Type A-II independent tanks.

A.2.10.4 Germanischer Lloyds

All tank designs are subject to the following guidelines:

"26-G.1.1 Tanks together with their supports and other fixtures are to be designed taking into account proper combinations of the various loads listed hereafter:

- Internal static load.
- External static load.
- Load due to insulation
- Loads corresponding to the elastic ship deflection.
- Internal and external dynamic loads due to motion of the ship.
- Sloshing loads.
- Thermal loads.
- Tank and cargo weight with the corresponding reactions in way of supports.
- Loads in way of towers and other attachments."

A.2.10.5 Lloyds Register of Shipping

Independent Tanks

No guidelines are given for tanks with a design vapor pressure, $P_o \leq 0.70 \text{ kg/cm}^2$. With regard to tank supports, the rules state:

"D7118 The tanks are to be supported on substantial foundations arranged to avoid excessive concentration of load on the ship's structure or on the tank. Provision is to be made for the thermal contraction of the tanks on cooling from ambient to service temperature and arrangements are to be made to control the movement of the tanks when the vessel is rolling and pitching."

For tanks with a design vapor pressure $P_o > 0.7 \text{ kg/cm}^2$, no specific guidelines are given. Regarding tank supports, the following rules are given:

"7205 Cargo tank seatings and securing arrangements are to be suitable for dynamic loading to the extent given in D7116 (see tank accelerations) and should also be suitable for the forces arising when a cargo hold is flooded with the cargo tanks empty. Seatings are to be designed to ensure uniform support to the pressure vessel having due regard to deflections of the hull structure in a seaway. When the cargo is to be carried at temperatures below ambient, provision is to be made for expansion and contraction."

Membrane Tanks

Membrane tanks are specially considered.

A.2.10.6 Nippon Kaiji Kyokai

"4.3.3 (4) Loads acting on tanks through tank supports (interaction forces between hull and tank) are to be obtained from the following.

(a) Hull deformations due to vertical bending moment, horizontal bending moment and torsional moment in waves, and vertical still water bending moment.

(b) Internal pressure in consideration of the accelerations and components of static weight due to ship motion.

(c) Water pressure distribution on the hull, where deemed necessary."

Membrane Tanks

Membrane tanks are subject to the same requirements as independent tanks.

A.2.10.7 United States Coast Guard

Independent Tanks

No specific guidelines.

Membrane Tanks

Some moderate scale fatigue testing is required, with the number of cycles based on anticipated conditions of primary hull bending for the life of the vessel (10^8 cycles for an anticipated 20 year life). Cyclic loads shall include that caused by maximum at-sea hull deflection.

Integral Tanks

No specific guidelines.

A. 2. 10. 8 International Association of Classification Societies

Independent and Membrane Tanks

"5.1 Cargo tanks are to be supported by the hull in a manner which will prevent bodily movement of the tank under static and dynamic load, while allowing contraction and expansion of the tank under temperature variations and hull deflections without undue stressing of the tank and hull."

A. 2. 10. 9 Summary of Dynamic Hull Deflections Criteria

Independent Tanks

ABS requires only that the foundations on horizontal tanks safely absorb the normal deflections of the hull and tank. LR and IACS require that the tank supports be designed so as to avoid excessive load concentrations on the hull and tank structures, and so that no bodily movement of the tanks occurs during dynamic loading (for IACS static loading is to be considered in addition to dynamic). Provision is to be made for thermal contraction of the tanks during filling operations.

Calculations of deflections on DnV Type A-1 tanks, are to include effects due to bending, shear, axial and torsional deformations and hull/cargo tank interaction forces. For DnV (Type A-II tanks), GL, IACS and NK, the loads for design against plastic deformations and buckling are to be taken as the most probable largest loads experienced by the ship in its lifetime (usually 10^8 wave encounters or 20 years). These loads will include vertical and horizontal bending and torsion of the hull girder.

BV gives no requirements concerning hull deflections but outlines a procedure for calculating hull bending moments. USCG gives no requirements for hull deflections on independent tanks.

Membrane Tanks

USCG requires some moderate scale fatigue testing of the IMT tank with the number of cycles based on the anticipated conditions of primary hull bending for the life of the vessel (10^8 cycles). Cyclic loads shall include those caused by maximum hull deflection experienced at sea.

BV, GL, NK, and IACS requirements for membrane tanks are the same as for independent tanks. DnV requirements for membrane tanks are the same as for the A-II independent tanks. LR and ABS have no specific requirements for membrane tanks.

Integral Tanks

No specific requirements concerning dynamic hull deflections are given for integral tanks by the agencies.

A.2.11 ACCELERATIONS AT TANK CENTER OF GRAVITY

A.2.11.1 American Bureau of Shipping

Independent Tanks

"24.37.1 --b (Combined Static and Dynamic Effects). Provision is to be made for the combined effect of the static pressure, internal vapor pressure (if any), and simultaneous rolling, pitching and heaving where each is defined as follows:

(1) A complete 30° roll, port and starboard (i.e., through 120°), in a period of 10 seconds

(2) A pitch of 6° half amplitude in a pitch period of seven seconds (i.e., 24° in 7 seconds)

(3) A heave of L/80 amplitude in a period of eight seconds (i.e., through L/20 in 8 seconds) where L is the length of the ship.

With the loading determined in accordance with the above, the stress in any item, unless otherwise specially approved, is not to exceed either three-quarters of the minimum yield strength or three-eights of the minimum ultimate strength of the base material or weld deposit in the final post weld condition, whichever is the least. For vessels over 183 meters (600 feet) in length, the combined static and dynamic effects may be specially considered, upon submission of a detailed analysis."

"24.37.3 The scantling of nonstructural primary containers are subject to special consideration. In each case the primary containers, supporting arrangements, secondary barrier and insulation will be reviewed as a complete unit subject to the satisfactory completion of the tests described in 24.31.2 prior to final approval."

"24.31.2 Primary containers which are not self-supporting may be given consideration, provided all details, arrangements and materials of the primary container, insulation and the supporting structure are suitable for the service conditions. Preliminary tests are to be made to ascertain that the design arrangements will function satisfactorily in all respects. The tests are to simulate the most severe operating conditions, including minimum service temperature, static and dynamic loads, hull vibration and slamming."

A.2.11.2 Bureau Veritas

The following requirements apply to all tank designs; however, the rule values may be replaced by other values if replaced by results of model tests or by a calculation considered to be suitably representative. In the latter case, the method used and the assumed conditions are subject to approval. A calculation guideline is found in 2.9 - Wave-Induced Loads. Motions as well as accelerations are presented here.

The motions are assumed to be periodic, of apparent frequency, f, measured in Hz, and of amplitude, A, measured in meters or in radians. The value of the amplitude of peak to peak motions is 2A. Motions considered in this section include heaving, pitching, longitudinal motion and transverse motion.

Heaving is the oscillation of normal translation. Concerning heave motions, the rules state:

"5-32-21 The maximum value of the apparent frequency is:

$$f = 0.08 \ (\omega_M + \omega_m)$$

with:

$$\omega_M = \frac{12.4}{\sqrt{L}} + 7.8 \ \frac{V}{L}$$

$$\omega_m = 0.41 + 0.0086 \ V \text{ "}$$

"5-32-22 The apparent frequency corresponding to the maximum amplitude is:

$$f = 0.0525 \ \omega_M + 0.105 \ \omega_m$$

with

$$\omega_M = \frac{12.4}{\sqrt{L}} + 4.7 \ \frac{V}{L}$$

$$\omega_m = 0.31 + 0.00294 \ V \text{ "}$$

"5-32-23 The maximum heaving amplitude, in meters, is:

$$A = 3.8 - 0.01 \ (L - 250)$$

without being taken greater than 3.8."

Concerning pitching, oscillation of rotation about a transverse axis, the rules state:

"5.32.31 The maximum value of the apparent frequency is identical with that for heaving."

"5.32.32 The apparent frequency corresponding to maximum amplitude is identical with that for heaving."

"5.32.33 The maximum frequency of pitch, in radians, is:

$$A_T = \frac{19}{L}$$

without being taken greater than 0.17."

"5.32.34 The axis of rotation is to be located on the midship perpendicular."

Concerning rolling, oscillation of rotation about a longitudinal axis, the rules state:

"5.32.41 The roll frequency is:

$$f_R = 1.724\sqrt{\frac{\overline{GM}}{B^2 + 4\overline{KG}^2}}$$

where:

$\overline{GM}$: distance from ship's center of gravity to transverse metacenter, in meters,

$\overline{KG}$: height of ship's center of gravity above the keel, in meters."

"5.32.42 The maximum roll amplitude, in radians, is:

- cargo ships, oil tankers and bulk carriers:

$$A_R = 0.01\,(63.0 - 0.9B)$$

with $0.21 \leqslant A_R \leqslant 0.52$

- for passenger ships, container ships and gas carriers:

$$A_R = 0.01\,(72.5 - 0.9B)$$

with $0.30 \leqslant A_R \leqslant 0.52$ "

"5.32.43 Where the ship is provided with an anti-rolling system, the values derived from 42 above are reduced by 50%."

"5.32.44 The axis of rotation is to be located 0.8 T meters above the keel."

Roll and pitch motions are to be superimposed. With respect to simultaneous rolling and pitching, the rules state:

- at maximum pitch, the roll amplitude is 50% of its maximum value,
- at maximum roll, the pitch amplitude is 60% of its maximum value.

Accelerations are treated first as independent accelerations in the longitudinal and transverse directions and then the accelerations are combined. Regarding longitudinal accelerations, the rules state:

"5.34.11 The applicable values of normal acceleration are:

- at the fore perpendicular:

$$\gamma_{AV} = \frac{1450}{L + 60} \qquad \text{with } \gamma_{AV} \leqslant 12$$

- at the midship perpendicular:

$$\gamma_{M} = \frac{384}{L - 7} \qquad \text{with } \gamma_{M} \leqslant 12$$

- at the aft perpendicular:

$$\gamma_{AR} = \frac{1270}{L + 60} \qquad \text{with } \gamma_{AR} \leqslant 12$$

- forward of the midship perpendicular:

$$\gamma = \gamma_M + 2.83\ (\gamma_{AV} - \gamma_M) \left(\frac{x}{L}\right)^{3/2}$$

- aft of the midship perpendicular:

$$\gamma = \gamma_M + 2.83\ (\gamma_{AR} - \gamma_M \left(\frac{x}{L}\right)^{3/2}$$ "

"5.34.12 The applicable values of longitudinal acceleration are:

$$\gamma = \sqrt{\gamma_o^2 + 0.49}$$

where γ_o is the greater of the two values:

$$\gamma_o = \left(\frac{230}{L + 15}\right)^2 \left(\frac{x}{L}\right) < 4 \left(\frac{x}{L}\right)$$

$$\gamma_o = 9.81\ A_T$$ "

Regarding transverse accelerations, the rules state:

"5.34.21 The applicable value of normal acceleration is the greater of the two values:

$$\gamma = 60\, f_R^2\, A_R\, y$$

$$\gamma = 60\, f_R^2\, A_R^2\, (z - 0.8\, T)$$

"5.34.22 The applicable value of the transverse acceleration is:

$$\gamma = \sqrt{\gamma_o^2 + 0.49}$$

with $\gamma_o = [9.81 + 60\, f_R^2\, (z - 0.8\, T)]\, A_R$ "

The maximum normal acceleration is to be calculated from the normal components of acceleration during transverse and longitudinal motions.

"5.35.21 The maximum value of normal acceleration at a given point is the greater of the following values:

$$\gamma = \sqrt{\gamma_t^2 + 0.36\, \gamma_\ell^2}$$

$$\gamma = \sqrt{\gamma_\ell^2 + 0.25\, \gamma_t^2} \quad "$$

Independent Tanks

"22.32.12 Scantlings of pressure cargo tanks are to be increased to provide for hydrodynamic pressure and dynamic loading due to motions of the ship where justified by dimensions of the tanks or by the relative importance of the hydrostatic pressure in relation to the maximum service pressure."

In addition, supports and attachments are to be designed per

"22.32.31 - to transmit to the hull and loads corresponding to the weight of the full tanks supplemented by the dynamic effects due to motions of the ship, while avoiding that the tanks take part in bending of the ship and are subject to abnormal stresses due to deformation of the ship;

- to permit free expansion or contraction of the tanks from pressure and temperature variations;
- to avoid any movement of the tanks due to motions of the ship."

"22.32.32 Supports or other attachments are to be provided to avoid movement of the tanks under a longitudinal acceleration of 0.3 g."

"22.34.11 Supports and other similar devices on self-supporting gravity cargo tanks are to be designed and built so as:

- to transmit to the hull structure the loads corresponding to the weight supplemented by the dynamic effects due to the motions of the ship while limiting stress concentrations in the hull structure and in the tank structure;
- to permit free contraction of the tanks;
- to avoid that the temperature of the ship structure in way of the supports and similar devices is lowered below the value allowed for the steel used."

Membrane Tanks

Integrated tank designs are to be subjected to model scale tests to determine the behavior of cyclic pressure variations due to ship motions and cyclic deformations of the ship.

A.2.11.3 Det norske Veritas

Independent Tanks

Type AI (Sec. 3-C) tanks are given scantlings according to Chapter II, Sec. 14. Girder systems are designed to the following accelerations:

"6.B.301. For tanks, type AI, and in general for tanks, type B, the following design accelerations are to be used, unless other values are justified by independent calculations.

Vertical acceleration:

$$a_z = \pm a_o \sqrt{1 + 25 \left(\frac{x}{L} + 0.05\right)^2 \left(\frac{0.6}{C_B}\right)^2}$$

Transverse acceleration:

$$a_y = \pm a_o \sqrt{0.25 + 6\left(\frac{x}{L} + 0.05\right)^2 + \chi\left(1 + 0.6\ \chi \frac{z}{B}\right)^2}$$

Longitudinal acceleration:

$$a_x = \pm a_o \sqrt{0.25 + \left(0.7 - \frac{L}{1200} + 5\frac{z}{L}\right)^2 \left(\frac{0.6}{C_B}\right)^2}$$

a_x, a_y and a_z are the maximum dimensionless (i.e., relative to the acceleration of gravity) accelerations in the respective directions and may be assumed to act independently.

a_z does not include the static weight.
a_y includes the component of the static weight in the transverse direction due to rolling.

x = longitudinal distance in meters from amidship to the centre of gravity of the tank with content. x is positive forward of amidships, negative aft of amidships.

z = vertical distance in meters from the ship's actual waterline to the centre of gravity of the tank with content. z is positive above and negative below the waterline.

$$a_o = 0,2 \frac{V}{\sqrt{L}} + \frac{30}{L}$$

Generally $\chi = 1.0$ is to be used. For particular loading conditions and hull forms, determination of χ according to the formula

$$\chi = \frac{13GM}{B}, \quad \chi \geq 1,0$$

may be required."

For Type AII (Sec. 3-D) tanks, the rules state:

"901. The loads given in 902, 903 and 904 are to be used for tanks type AII, and in special cases for tanks, type B.

"902. The loads for design against plastic deformations and buckling are to be taken as the most probable largest loads in 10^8 wave encounters (probability level $Q = 10^{-8}$) for a ship operating on the North-Atlantic. The wave loads are to be determined according to accepted theories, model tests or full-scale measurements. All heading angles are to be given the same probability of occurrence, and speed reduction in heavy weather may be taken into account.

All types of wave-induced loads and motions exerted by hull and cargo on the tank structure are to be considered. Generally, these types of loads are:

- Vertical, transverse and longitudinal acceleration forces;
- Internal liquid pressure in the tank (full and partially full)
- External water pressure on the hull;
- Vertical and horizontal bending of the hull girder;
- Torsion of the hull girder.

For Type B (Sec. 3-E) regulations are the same as for Type AI except as noted for Type AII above.

Membrane Tanks

Same as for independent tank Type AII.

A.2.11.4 Germanischer Lloyds

Independent and Membrane Tanks

"G.3.3.1 The determination of dynamic loads is to take account of the long term distribution of ship motions including the effects of surge, sway, heave, roll, pitch and yaw on irregular seas."

"G.3.3.2 On ships operating in unrestricted service, for the design of tanks and their supports against plastic deformation and buckling, the statistically expected most probable largest sea load in the ship's lifetime shall be taken. All kinds of wave loads are to be taken into account, such as:

a) inertia forces due to vertical, transverse and longitudinal accelerations (see also 3.6).

b) internal dynamic heads in tanks when partly or completely filled with liquids.

c) external dynamic loads on the hull.

d) loads due to vertical and horizontal bending as well as due to torsion."

"G.3.3.3 Ships for restricted service will be given special consideration."

"G.3.3.6 The accelerations acting on tanks are estimated at their center of gravity, and include the following components:

Vertical acceleration (vertical to the base line, i.e., in z-direction) due to heave, pitch, and if applicable, roll; however, static weight components not included.

Transverse acceleration (vertical to the ship's side, i.e., in y-direction) due to roll, pitch, yaw and sway including gravity component of roll.

Longitudinal acceleration (in longitudinal direction, i.e., in x-direction) due to surge and pitch including gravity component of pitch.

For the purpose of approximation, the accelerations may be calculated approximately according to the following formulae:

Vertical acceleration:

$$a_z = \pm\, a_o \sqrt{1 + \left(5.3 - \frac{45}{L}\right)^2 \left(\frac{x}{L} + 0.05\right)^2 \left(\frac{0.6}{\delta}\right)^{3/2}}$$

Transverse acceleration:

$$a_y = \pm\, a_o \sqrt{0.6 + 2.5\left(\frac{x}{L} + 0.05\right)^2 + \chi\left(1 + 0.6 \cdot x\,\frac{z}{B}\right)^2}$$

Longitudinal acceleration:

$$a_x = \pm\, a_o \sqrt{0.15 + \left(0.4 - L/2000 + 3\,\frac{z}{L}\right)^2 \left(\frac{0.6}{\delta}\right)^2} \leq 0.4$$

x = distance in longitudinal direction from amidship (L/2) to the center of gravity of the tanks, with liquid, in (m); positive sign forward of L/2, negative sign aft of L/2.

z = vertical distance from the waterline of the ship to the centre of gravity of the tanks, with liquid, in (m); positive sign above the water line, negative sign below the waterline.

$$a_o = 0.2\,\frac{v}{\sqrt{L}} + \frac{34 - 600/L}{L}$$

v = maximum speed in calm water in (kn)

Generally, $\chi = 1.0$. For particular loading conditions and hull forms, determination of χ according to the formulae below may be necessary.

$$\chi = \frac{13 \cdot \overline{M_B G}}{B} \geq 1.0$$

$\overline{M_B G}$ = metacentric height in (m)

a_x, a_y and a_z are the maximum dimensionless (i.e., relative to the acceleration of gravity) accelerations. They are considered as acting separately for calculation purposes."

A.2.11.5 Lloyds Register of Shipping

Independent Tanks

"D 7116 The tanks for $P_o \leq 0.70$ kg/cm^2 shall be designed to withstand:

(i) A test head of 2.44 m of water above the top of of the tank or 0.61 m above the top of the hatch, whichever may be the greater.

(ii) The combined effect of internal vapor pressure (if any) and rolling, pitching and heaving as follows:

(1) A complete 30° roll port and starboard (i.e., through 120°) in a period of ten seconds.

(2) A pitch of 6° half amplitude in a pitch period of seven seconds (i.e., through 24° in seven seconds).

(3) A heave of 0.0125L half amplitude in a period of eight seconds (i.e., through 0.05L in eight seconds).

With the loading determined in accordance with the above, the stress in any item shall not exceed three-quarters of the yield stress or three-eights of the ultimate stress."

"D 7205 Cargo tank seatings and securing arrangements are to be suitable for dynamic loading to the extent given in D 7116 and should also be suitable for the forces arising when a cargo hold is flooded with the cargo tanks empty. Seatings are to be designed to ensure uniform support to the pressure vessel having due regard to deflections of the hull structure in a seaway. When the cargo is to be carried at temperatures below ambient, provision is to be made for expansion and contraction."

Membrane Tanks

Specially considered - no specific guidelines.

A.2.11.6 Nippon Kaiji Kyokai

Independent Tanks and Membrane Tanks

"4.3.2.2. For design accelerations of each direction specified in this paragraph, vertical acceleration ($a_z g$) is not generally including the static weight, and transverse acceleration ($a_y g$) and longitudinal acceleration ($a_x g$) are generally including the component of the static weight in each direction due to the inclination of the ship. Except in case of predicting the long-term distributions specified in preceding 1 (see wave-induced loads),* the acceleration in each direction acting at the centre of gravity of tank with content may be obtained from the following approximate formulae:

Vertical acceleration: $(1 + a_z)\,g$

$$a_z = \pm\, a_o \sqrt{1 + \left(5.3 - \frac{45}{L}\right)^2 \left(\frac{X}{L} + 0.05\right)^2 \left(\frac{0.6}{C_b}\right)^{3/2}}$$

Transverse acceleration: $a_y\,g$

$$a_y = \pm\, a_o \sqrt{0.6 + 2.5 \left(\frac{X}{L} + 0.05\right)^2 + \chi \left(1.0 + 0.6\, \chi\, \frac{Z}{B}\right)^2}$$

Longitudinal acceleration: $a_x\,g$

$$a_x = \pm\, a_o \sqrt{0.25 + \left(0.7 - \frac{L}{1200} + 5\,\frac{Z}{L}\right)^2 \left(\frac{0.6}{C_b}\right)^2}$$

where:

$$a_o = 0.2\,\frac{V_s}{\sqrt{L}} + \frac{34 - \frac{600}{L}}{L}$$

X = Longitudinal distance from amidship to the center of gravity of tank with content, (m). X is positive forward of amidship, negative aft of amidship.

Z = Vertical distance from the ship's actual water line to the center of gravity of tank with content, (m). Z is positive above and negative below the water line.

g = Acceleration of gravity, (m/sec^2).

V_s = service speed of the ship, (knots).

χ = 1.0 in general. For particular loading conditions and hull forms, χ is to be obtained from the following formula:

$$\chi = \frac{13\,\overline{GM}}{B}$$

$\overline{GM}$ = Metacentric height, (m)."

* (Page 38)

A.2.11.7 United States Coast Guard

Independent Tanks, Membrane Tanks and Integral Tanks

"VI. A. 1. All tanks and supporting structures in vessels in ocean; Great Lakes; lakes, bays, and sounds; or coastwise service must be designed to withstand at least the following dynamic loads (except as noted in 2. below):

a. Rolling 30° on each side of upright with a period of 10 seconds. The dynamic roll force on the tank and supports then becomes 0.00642 Wr (tons); where W is the filled tank weight (tons) and r is the vertical arm (feet) between the vessel's roll axis and the full tank center of gravity. The tank walls must be designed to withstand the resulting dynamic hydraulic loadings.

b. Pitching 6° half amplitude with a period of 7 seconds. The dynamic pitch force then becomes 0.002624 Wl (tons); where W is the filled tank weight (tons) and l is the longitudinal arm (feet) between the vessel's pitch axis and the full tank center of gravity. Tank bottoms must be designed to withstand the resulting dynamic hydraulic loads.

c. Heaving L/80 feet half amplitude with a period of 8 seconds. The dynamic heave force then becomes 0.0002395 WL (tons); where W is the filled tank weight (tons) and L is the vessel's length (feet). Tank bottoms must be designed to withstand the resulting hydraulic loads.

2. Not withstanding the comments of 1. above, when it can be shown to the satisfaction of the Commandant that the imposed acceleration forces are unrealistic for the size of ship involved (generally over 600 feet in length), the requirements may be relaxed somewhat upon submission of substantiating data on a per case basis."

A.2.11.8 International Association of Classification Societies

Independent and Membrane Tanks

"2.46 The accelerations acting on tanks are estimated at their center of gravity and include the following components:

Vertical acceleration: motion accelerations of heave, pitch and possibly roll (normal to the ship base).

Transverse acceleration: motion accelerations of sway, yaw and roll - gravity component of roll.

Longitudinal acceleration: motion accelerations of surge and pitch - gravity component of pitch.

The following formulas are given as guidance for the components of acceleration due to ship motions in the case of ships with $L \geq 50\,m$. These formulas correspond to a probability level of 10^{-8} in the North Atlantic.

Vertical acceleration:

$$a_z = \pm a_o \sqrt{1 + \left(5.3 - \frac{45}{L}\right)^2 \left(\frac{X}{L} + 0.05\right)^2 \left(\frac{0.6}{C_B}\right)^{3/2}}$$

Transverse acceleration:

$$a_y = \pm a_o \sqrt{0.6 + 2.5 \left(\frac{X}{L} + 0.05\right)^2 + \chi \left(1 + 0.6\ \frac{\chi z}{B}\right)^2}$$

Longitudinal acceleration:

$$a_x = \pm a_o \sqrt{0.06 + A^2 - 0.25\,A}$$

where

$$A = \left(0.7 - \frac{L}{1200} + \frac{5z}{L}\right)\left(\frac{0.6}{C_B}\right)$$

and L = length of ship between perpendiculars in meters.

C_B = block coefficient

B = greatest moulded breadth in meters

x = longitudinal distance in meters from amidship to center of gravity of the tank with content. x is measured forward of amidship, negative aft of amidship.

z = vertical distance in meters from ship's actual waterline to the center of gravity of tank with content. z is positive above and negative below the waterline.

V = service speed in knots.

and

$$a_o = 0.2 \frac{V}{\sqrt{L}} + \frac{34 - \frac{600}{L}}{L}$$

Generally, $\chi = 1.0$. For particular loading conditions, determination of χ according to the formula below may be necessary

$$\chi = \frac{13\ GM}{B} \qquad \chi \geq 1.0.$$

a_x, z_y, a_z are the maximum dimensionless (i.e., relative to acceleration of gravity) accelerations in the respective direction and they are considered as acting separately for calculation purposes.

a_z does not include static weight.

a_y includes the component of static weight in the transverse direction due to rolling."

A.2.11.9 Accelerations Summary

Independent Tanks

All agencies provide some guidelines for the calculation of accelerations. Basically, the guidelines can be broken into two categories: 1) the superposition of motions of the vessel, from which accelerations can be calculated[13], and 2) formulas describing the acceleration at any point in the ship a_x, a_y, and a_z[14]. ABS, USCG and LR require that the tanks be capable of withstanding simultaneous rolling, pitching and heaving motions of specified amplitudes and periods without resulting in loads for which the structure has not been designed. ABS, LR, and USCG require that the effects of the liquid pressure head be superimposed on the rolling, pitching, and heaving motions. From the ABS and LR guidelines, it is possible to calculate accelerations in the transverse and vertical directions; however, there is not enough information to calculate the longitudinal accelerations completely (the contribution due to surge is not specified). The USCG guidelines provide formulas which are consistent with the accelerations calculated from the ABS and LR rules. For vessels over 183 meters long, the loads given above may be specially considered.

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[13] See Chapter III for the derivation of the calculation.

[14] a_z = vertical acceleration due to heave, pitch and, if applicable, roll; static weight components not included.
a_y = transverse acceleration due to roll, pitch, yaw and sway including gravity components of roll.
a_x = longitudinal accelerations due to surge and pitch including gravity components of pitch.

BV, DnV, GL, NK, and IACS present approximate formulas for calculation of accelerations a_x, a_y, a_z acting on the tanks as a function of the length of the ship, the contract service speed, the block coefficient, and the point at which the acceleration is to be calculated. Generally, the acceleration loads (DnV, GL, NK, and IACS) for design purposes are to be taken as the most probable largest loads in the anticipated service life of the vessel (usually 10^8 wave encounters).

BV provides a calculation procedure that is quite different from the other agencies' approach. The accelerations are calculated from roll, heave and pitch motions, but the motions are more explicit than for ABS, LR, and USCG. The acceleration formulas allow the calculation of normal (a_z) and longitudinal accelerations in longitudinal motion (pitch, surge) and normal and transverse accelerations in transverse motions (roll, sway). In addition, the agency provides a method for combining normal accelerations to obtain the maximum value of the normal acceleration at any point in the ship.

IACS provides a method for properly combining the accelerations (static and dynamic) using an ellipse. NK specifies the use of the ellipse for determining the reactions on saddle-supported cylindrical tanks produced by the tank accelerations.

Membrane Tanks

ABS and LR specially consider membrane tank designs so no regulations are given. ABS does require preliminary tests simulating the most severe operating conditions including minimum service temperature, static and dynamic loads, hull vibration and slamming. In addition to the requirements for independent tanks, BV requires model scale tests to determine the behavior of cyclic pressure variations due to ship motions and cyclic deformation of the ship's hull. DnV, GL, NK, USCG, and IACS requirements for independent tanks are also to be applied to membrane designs.

Integral Tanks

USCG requirements for integral tanks are the same as for independent tanks.

A.2.12 DYNAMIC EXTERNAL PRESSURES ON HULL

A.2.12.1 American Bureau of Shipping

Independent Tanks

No regulations or guidelines given for tank design. Hull scantlings are to be determined in accordance with Sections 22 and 23, which give no specific formulas for dynamic external pressures.

Membrane Tanks

"24.31.2 Primary containers which are not self-supporting may be given consideration, provided all details, arrangements and materials of the primary container, insulation and the supporting structure are suitable for the service conditions. Preliminary tests are to be made to ascertain that the design arrangements will function satisfactorily in all respects. The tests are to simulate the most severe operating conditions, including minimum service temperature, static and dynamic loads, hull vibration and slamming."

A.2.12.2 Bureau Veritas

Independent and Membrane Tanks

"22.21.24 The form of the hull is to be chosen so as to reduce the loads due to sea motions, in particular slamming."

A.2.12.3 Det norske Veritas

Independent Tanks

Regarding Type A-I, A-II, B (Section 6-B) independent tanks, the rules state:

"B801 The calculations are to be based on the most severe realistic loading conditions with the ship fully or partly loaded."

"B901 The loads given in 902, 903, and 904 are to be used for tanks, type AII, and in special cases for tanks, type B."

"B902 The loads for design against plastic deformations and buckling are to be taken as the most probable largest loads in 10^8 wave encounters (probability level $Q = 10^{-8}$) for a ship operating on the North-Atlantic.

The wave loads are to be determined according to accepted theories, model tests or full-scale measurements. All heading angles are to be given the same probability of occurrence, and speed reduction in heavy weather may be taken into account.

All types of wave-induced loads and motions exerted by hull and cargo on the tank structure are to be considered. Generally, these types of loads are:

- Vertical, transverse and longitudinal acceleration forces.
- Internal liquid pressure in the tank (full and partially full).
- External water pressure on the hull.
- Vertical and horizontal bending of the hull girder.
- Torsion of the hull girder."

Regarding Type A-I (Section 6-C) independent tanks, the rules state:

"C201 For webs, girders and stringers, a structural analysis is to be carried out to ensure that the stresses are acceptable. Calculation methods applied are to take into account the effects of bending, shear, axial and torsional deformations as well as the hull/cargo tank interaction forces due to the deflection of the double bottom and cargo tank bottom."

"C202 The following loads and stresses are to be taken into consideration:

- Static loads according to B200.
- Dynamic additional loads due to the ship's movement in a seaway. See B300 and C203-208.
- Thermal stresses."

"C204 The dynamic additional external water pressure head at the ship's bottom amidship is given by:

$$h_{eb} = 1.025\left(0.3 + \frac{V}{6.1\sqrt{L}}\right)\left(\frac{L}{10}\right) c^{-\frac{L}{300}} \quad \text{if } L \leq 300 \text{ m}$$

$$h_{eb} = 11.3\left(0.3 + \frac{V}{6.1\sqrt{L}}\right) \qquad \text{if } L > 300 \text{ m.}$$

In the fore and afterbody, h_{eb} has to be multiplied by a factor β:

Forebody:

$$x > 0.1\ L : \beta = 1 + 12\left(\frac{\frac{x}{L} - 0.1}{C_B}\right)^2$$

Afterbody:

$$x < -0.2\ L : \beta = 1 + 20\left(\frac{x}{L} + 0.2\right)^2$$

h_{eb} = pressure head in meters of water.

x = distance from amidships in meters. "

"C205 The dynamic additional external water pressure head at the actual water line amidships is given by:

$$h_{ed} = 1.025\left(0.8 + \frac{V}{6.1\sqrt{L}}\right)\left(\frac{L}{10}\right) c^{-\frac{L}{300}} \quad \text{if } L \leq 300 \text{ m.}$$

$$h_{ed} = 11.3\left(0.8 + \frac{V}{6.1\sqrt{L}}\right) \quad \text{if } L > 300 \text{ m.}$$

In the fore and afterbody h_{ed} has to be multiplied by a factor β:

Forebody:

$$x > 0.1\ L : \beta = 1 + \frac{2500}{L + 300}\left(\frac{\frac{x}{L} - 0.1}{C_B}\right)^2$$

Afterbody:

$$x < -0.2\ L : \beta = 1 + \frac{2000}{L + 300}\left(\frac{\frac{x}{L} + 0.2}{C_B}\right)^2$$ "

"C206 The dynamic additional external water pressure head at the deck is given by:

$$h_{eD} = h_{ed} - 1.025\ (D - d) \quad \text{if } (D - d) \leq \frac{h_{ed}}{1.025}$$

$$h_{eD} = 0 \quad \text{if } (D - d) > \frac{h_{ed}}{1.025}$$

D = moulded depth in meters.

d = actual draught in meters.

If $(D - d) > \frac{h_{ed}}{1.025}$ the dynamic additional external pressure is zero at a distance $\frac{h_{ed}}{1.025}$ above the water line. The pressure over a cross section may be considered to vary linearly between deck (or the point of zero pressure), actual water line and bottom and being constant over deck and bottom."

Regarding Type B (Section 6-E) tanks, the rules state:

"E103 In the design of the tank, the dynamical loads due to the ship's motions in a seaway are to be taken into account. For tanks supported in such a way that the deflection of the hull transfers significant stresses to the tank, the wave-induced loads may be required to be calculated as given in B902. For saddle-supported tanks and other tanks, where a calculation of loads according to B902 is not required, design accelerations given in B301 are to be used."

Membrane Tanks

"F101 Membrane tanks are to be designed as being subject to the same loads as for independent tanks, type AII."

A.2.12.4 Germanischer Lloyd

Independent, Membrane, and Integral Tanks

"26-G 3.9 External loads in accordance with 3.2 (see Acceleration tion at Tank C.G.), for bottom and side shell will be determined by means of computer programs of this Society. For the purpose of approximation, the loads stipulated in Section 4, C.1.--2., may be taken."

The external dynamic loads for tanks situated above the weather deck, are given according to Section 16.C.2., which are not listed here since such tanks are a rare exception for LNG transport.

"Sec. 4, C.1 The external load h_s for determining the scantlings of the ship's sides is to be calculated according to the following formulae:

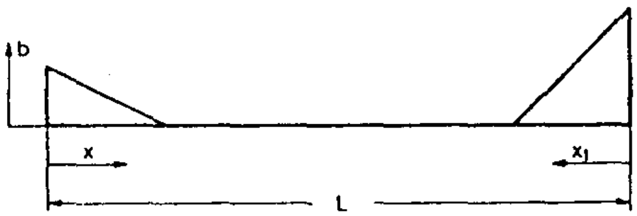
$$h_s = z_2 + c\left(1 - \frac{z_2}{2\,T} + b\right) \; [t/m^2]$$

where the lower edge of plating or the centre of the span ℓ is below the TWL,

$$h_s = c\left(1 + b\ \frac{8}{8 + z_1}\right)\ [t/m^2]$$

where the lower edge of plating or the centre of the span ℓ is above the TWL

c	$= 0.023\ L\ [t/m^2]$	for $L < 100$ m
c	$= 3.7 - 140/L\ [t/m^2]$ ($L_{max} = 300$ m)	for $L \geq 100$ m
z_1, z_2	= vertical distance between TWL and lower edge of plating or center of span ℓ in [m] (TWL = deepest load water line, z_1 above TWL, z_2 below TWL)	
b	$= 0.7 - 3.5\ x/L$	for $0 \leq x/L \leq 0.2$
b	$= 0$	for 0.6 L amidships



b	$= 1.5 - 7.5\ x_1/L$	for $0 \leq x_1/L \leq 0.2$
x, x_1	= distance of the position considered from the aft or from the forward perpendicular in [m] (see sketch)."	

"Sec. 4, C. 2 The external load for determining the scantlings of the bottom structure is to be calculated according to the following formula:

$$h_B = T + c\,(0.5 + b)\quad [t/m^2]$$

b and c see under 1., b need not be taken greater than derived for $x = 0.1\ L$ or $x_1 = 0.05\ L$."

A. 2. 12. 5 Lloyds Register of Shipping

No requirements are given.

A. 2. 12. 6 Nippon Kaiji Kyokai

Independent Tanks and Membrane Tanks

"4. 3. 2. 3 Except in case of predicting the long-term distribution specified in preceding 1 (see wave-induced hull deflections) dynamic additional external water pressure head may be obtained from the following approximate formulae.

(1) Dynamic additional external water pressure head at the actual water line;

$$h_{EL} = \begin{cases} 23\ \beta\ (m) & : L \geq 230\ (m) \\ 0.1\ L\beta\ (m) & : L < 230\ (m) \end{cases}$$

where:

$$\beta = 1 + 2.4 \left(\frac{X}{L}\right)^2$$

β = Correction factor for the forebody. As for the afterbody, it is taken as 1.

X = Distance from the midship, (m).

(2) Dynamic additional external water pressure head at the bottom of the ship;

$$h_{EB} = \begin{cases} 8.02\ \beta\ (m) & : L \geq 230\ (m) \\ 0.035\ L\beta\ (m) & : L < 230\ (m) \end{cases}$$

where:

$$\beta = 1 + 7.2 \left(\frac{X}{L}\right)^2$$

β = Correction factor for the forebody. As for the afterbody, it is taken as 1.

X = Distance from the midship, (m).

(3) Dynamical additional external water pressure head at the deck'side;

$$h_{ED} = \begin{cases} h_{EL} - 1.025\,(D - d) & : (D - d) \leq \dfrac{h_{EL}}{1.025} \\ 0 & : (D - d) > \dfrac{h_{EL}}{1.025} \end{cases}$$

If $(D - d) > \frac{h_{EL}}{1.025}$, h_{ED} is to be taken as zero at $\frac{h_{EL}}{1.025}$ above the water line."

A.2.12.7 United States Coast Guard

Independent Tanks

No specific requirements for dynamic external hull pressures.

Membrane Tanks

IMT (Chapt. IV C.3): Designed to ABS scantlings or equivalent. Also requires prototype testing to prove the adequacy of the entire integrated system. Loads shall include internal and external pressure loadings. (Note: this external pressure can perhaps be interpreted as pressure between primary and secondary barrier, in which case it would not apply to this loading condition).

Integral Tanks

The IGT tanks are to be designed to ABS standards or equivalent.

A.2.12.8 International Society of Classification Societies

No requirements for independent or membrane tank designs.

A.2.12.9 Summary of External Hull Pressure Criteria

Independent Tanks

BV requires that the form of the hull be designed to reduce the loads due to sea motions especially slamming. DnV, GL and NK provide similar approximate formulas for the calculation of the dynamic additional water pressure (in meters of water) due to the movement of the ship in a seaway. These formulas give the equivalent head of water as a function of the ship length, the ship's service speed, the block coefficient, and the point at which the pressure is to be calculated. In addition, DnV requires that the calculation take into account the hull/cargo interaction forces due to the deflections of the double bottom and the cargo tank bottom (see A.2.10). GL specifies that external loads for bottom and side shell will be determined by their computer codes in accordance with their Section 26-G.3.2. (see A.2.11). All three agencies, DnV, GL and NK require that the loads due to external hull pressures be calculated from the long-term distribution of ship motions in the seaway. ABS, LR, USCG and IACS have no requirements concerning dynamic external hull pressures.

Membrane Tanks

ABS and USCG require tests of prototype tanks to insure that the design of the entire system will perform satisfactorily under service conditions. ABS requires that the tests simulate the most severe operating conditions including among other loads, all static and dynamic loads and slamming. USCG requires that the tests simulate internal and external pressure loadings (see A.2.12.7). BV, DnV, GL and NK require that membrane tanks be subject to the same regulations as independent tanks. LR and IACS present no regulations for membrane tanks concerning dynamic external hull pressure.

Integral Tanks

The agencies provide no specific guidelines for integral tanks.

A. 2. 13 DYNAMIC INTERNAL PRESSURE

A. 2. 13. 1 American Bureau of Shipping

No specific requirements are given.

A. 2. 13. 2 Bureau Veritas

Independent Tanks

"22-32-12 Scantlings of pressure cargo tanks are to be increased to provide for hydrodynamic pressure and dynamic loading due to the motions of the ship where justified by the dimensions of the tanks or by the relative importance of the hydrostatic pressure in relation to the maximum service pressure.

"22-34-14 Stiffeners of pressure cargo tanks are to be provided, where necessary, in way of the supports of the tanks. The scantlings of such stiffeners are to take into account the dynamic loading due to the motions of the ship."

"22-33-31 For the calculation of scantlings of stiffeners and plates in self-supporting gravity tanks, the following pressures are to be considered:

$P_1(t/m^2)$: hydrostatic pressure corresponding to a height of 2.40 meters above the top of the tank (or 0.60 meters above the top of the dome if greater). For plates, the height is to be taken from the lower edge of the plate.

If the setting pressure exceeds 0.25 kgf/cm^2 and for the carriage of high density liquefied gases, the value of P_1 will be specially considered.

$P_2(t/m^2)$: pressure equal to the dynamic head h_d determined as per 10-22.22, the height z being measured from the top of the tank and, for plates, down to the lower edge of the plate."

"10.22.22 The applicable dynamic load height for sizing tank plating is:

$$h_d = h_o + \delta(y \sin A_R + z \cos A_R)$$

with

$$h_o = p_o + \delta(0.6\ \ell\ A_T + y_p \sin A_R - z_p \cos A_R)$$

where index p relates to the point on the tank top furthest removed from the ship's centerline.

The apparent specific gravity δ of the cargo is equal to the greatest of the following values:

$$\delta = \delta_o \left(1 + 0.1\sqrt{\gamma_t^2 + 0.36\ \gamma_\ell^2}\right)$$

$$\delta = \delta_o \left(1 + 0.1\sqrt{0.25\ \gamma_t^2 + \gamma_\ell^2}\right)$$

where δ_o is the specific gravity of the cargo, which is to be determined having regard to the various possible sources of supply; specially, for natural hydrocarbons, the applicable value is not to be less than the specific gravity of the pure produce plus 20%.

where

γ_ℓ = maximum value of normal acceleration, in m/s^2, derived from 5-34.11. For a given tank, γ_ℓ is determined for the tank end furthest removed from amidships, without the distance of such tank end to amidships being less than ℓ.

γ_t = maximum value of normal acceleration, in m/s^2, derived from 5-34.21,[15] calculated athwart the considered tank."

Membrane Tanks

"22-36-21 As a rule, each design proposed for integrated tanks is to be submitted to model tests permitting namely to check its behavior under the effect of

- cyclic pressure variations due to ship motions.
- cyclic deformations of the ship.
- vibrations of the ship."

A.2.13.3 Det norske Veritas

Independent Tanks

Regarding Type AI tanks, and Type AII and B when applicable, the rules state:

"C.203 The dynamic additional internal pressure head in a full tank may be found from Fig. 3. For tanks carrying liquids with $\gamma \neq 1{,}0$, the pressure head, h_i, is to be multiplied by γ."

For Type B Independent Tanks, the rules also state:

"E.102 The internal pressure, p, used to determine the thickness of any specific part of the tank is given by the following formula:

$$p = p_o + (1 + a_z)\, h_s$$

p_o is defined in B 201.

a_z is defined in B 301.

h_s = static liquid pressure, defined in B 202."

Membrane Tanks

No specific guidelines are given.

---

[15] See A.2.11.2 on acceleration loads in the longitudinal and transverse directions.

MEAN INTERNAL DYNAMIC LIQUID PRESSURE IN FULL TANKS

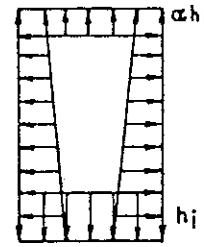
CARGO DENSITY $\gamma = 1{,}0$ tonne/m³

h_i = Dynamic pressure amidships at tank bottom in metres of water.

αh_i = Dynamic pressure amidships at tank top in metres of water.

l_t = Distance between transverse tight bulkheads in metres.

b_t = Distance between longitudinal tight bulkheads in metres.



TRANSVERSE PRESSURE DISTRIBUTION

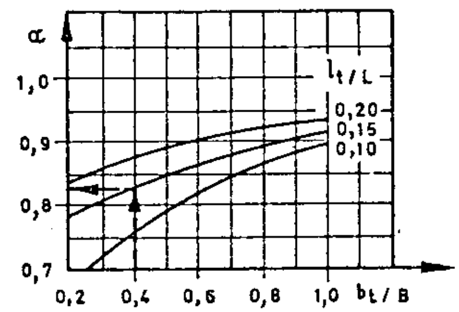
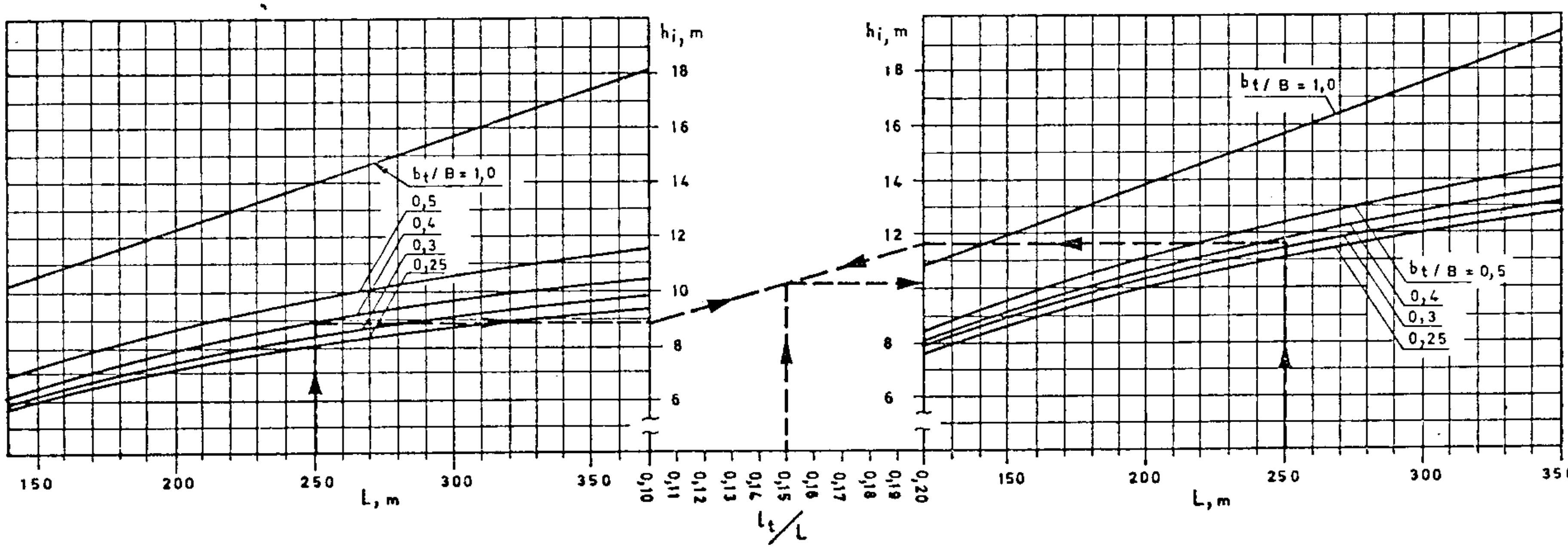


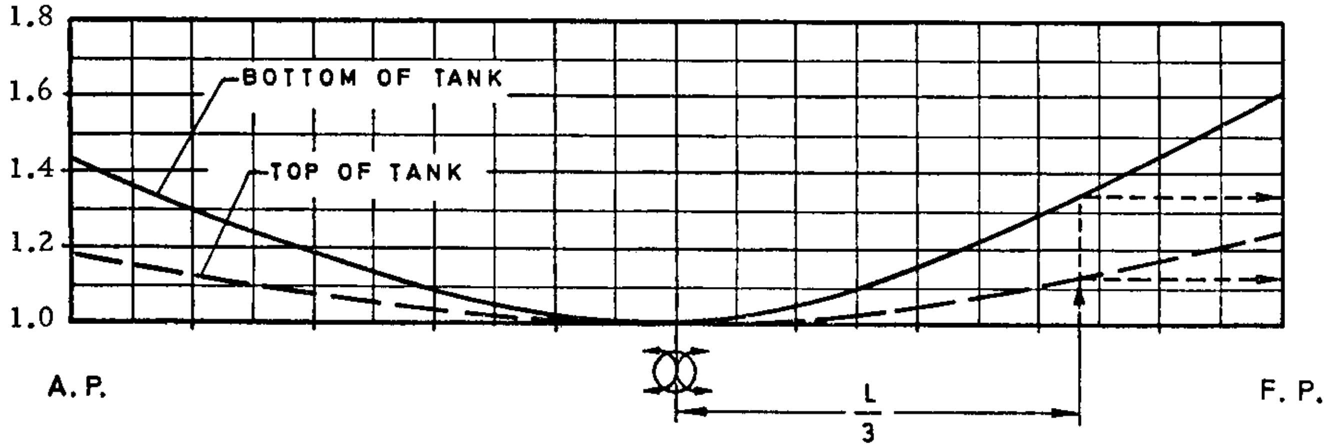
Fig. 3B FACTOR FOR DETERMINING DYNAMIC PRESSURE AMIDSHIPS AT TOP OF TANK



DnV Fig. 3A MEAN DYNAMIC PRESSURE AMIDSHIPS AT BOTTOM OF TANK

DnV Fig. 3 C.

LONGITUDINAL DISTRIBUTION OF INTERNAL, DYNAMIC PRESSURE IN FULL TANKS RELATIVE TO THE PRESSURE AMIDSHIPS. THE DIAGRAM GIVES THE DYNAMIC PRESSURE AT THE TRANSVERSE SECTION THROUGH THE CENTRE OF GRAVITY OF THE TANK.



Example:

Find the internal dynamic liquid pressure in a full tank with centre of gravity L/3 forward of amidships.

$L = 250$ m
$b_t/B = 0{,}4$
$l_t/L = 0{,}15$

From Fig. 3A: $h_l = 10{,}2$ m at bottom amidships.

From Fig. 3B: Factor for pressure at top $\alpha = 0{,}83$, i.e. pressure at top amidships, $\alpha \times h_l = 10{,}2 \times 0{,}83 = 8{,}5$ m.

From Fig. 3C: Dynamic pressure at bottom at L/3 is $10{,}2 \times 1{,}35 = 13{,}9$ m.

Dynamic pressure at top of tank at L/3 is $8{,}5 \times 1{,}13 = 9{,}6$ m.

A.2.13.4 Germanischer Lloyds

All tank designs are subject to the following regulation concerning internal dynamic loads in filled tanks:

"26.3.7 The internal dynamic loads are to be determined under consideration of the accelerations as per 3.2 and 3.6. The largest individual load resulting from the three accelerations is decisive. The individual loads $p_d\,[t/m^2]$ are:

a) in longitudinal direction $\quad p_{dx} = \gamma \cdot x' \cdot a_x$

b) in transverse direction $\quad p_{dy} = \gamma \cdot y' \cdot a_y$

c) in vertical direction $\quad p_{dz} = \gamma \cdot z' \cdot a_z$

x', y', z' = maximum lengths of the heads of liquid in [m] in x- or y- or z-direction above the point under consideration.

For the forward and aft tank boundary:

$$p_{dx} = \gamma \cdot \ell_t \cdot a_x$$

(to be compared with b) and c))

For the tank sides:

$$p_{dy} = \gamma \cdot b_t \cdot a_y$$

(to be compared with a) and c))

For the lower and upper tank boundaries:

$$p_{dz} = \gamma \cdot h_t \cdot a_z$$

(to be compared with a) and b))

ℓ_t = length of tank in [m]

b_t = width of tank in [m]

h_t = height of tank in [m]

γ = specific gravity of the heaviest liquified gas in $[t/m^3]$ "

A.2.13.5 Lloyds Register of Shipping

No specific guidelines are given.

A. 2. 13. 6 Nippon Kaiji Kyokai

All tank designs are subject to the following regulation:

"4. 3. 3(2) Internal pressure is to be taken into account for each directional acceleration and the static weight due to inclination of the ship at the full loaded condition. Internal pressure loads may be given as internal pressure distributions shown in the following.

(a) Prismatic tanks (see Fig. 4. 3. 3(2)(a)).
Water pressure head in meters at any point (j) on the tank wall is given as follows:

$$h_j = h_{j.st} + h_{j.dyn}$$

$$h_{j.st} = 10\ P_o + \gamma Z_j$$

$$h_{j.dyn} = \gamma \sqrt{(x_j a_x)^2 + (y_j a_y)^2 + (z_j a_z)^2}$$

where:

P_o = Design vapor pressure, (kg/cm^2).

γ = Assumed specified gravity of cargo intended to be carried, (t/m^3).

x_j, y_j and z_j = As shown in Fig. 4. 3. 3(2)(a), (m).

a_x, a_y and a_z = As specified in 4. 3. 2-2.

(b) Spherical tanks
Internal pressure $P(\phi, \theta)$, (kg/cm^2) at any point on the wall of spherical tank is to be considered referring to both of the following loads:

(i) $$P(\phi, \theta) = P(\phi, \theta)_{st} + P(\phi, \theta)_{dyn}$$

$$P(\phi, \theta)_{st} = P_o + 0.1\ \gamma R(1 - \cos\phi)$$

$$P(\phi, \theta)_{dyn} = \sqrt{P_1^2 + P_2^2 + P_3^2}$$

$$P_1 = 0.1\ \gamma R\ (\sqrt{1 + a_x^2} - a_x \sin\phi \cos\theta - 1)$$

$$P_2 = 0.1\ \gamma R\ (\sqrt{1 + a_y^2} - a_y \sin\phi \sin\theta - 1)$$

$$P_3 = 0.1\ \gamma R a_z\ (1 - \cos\phi)$$

where:

P_o, γ, a_x, a_y and a_z = As specified in preceding (a).

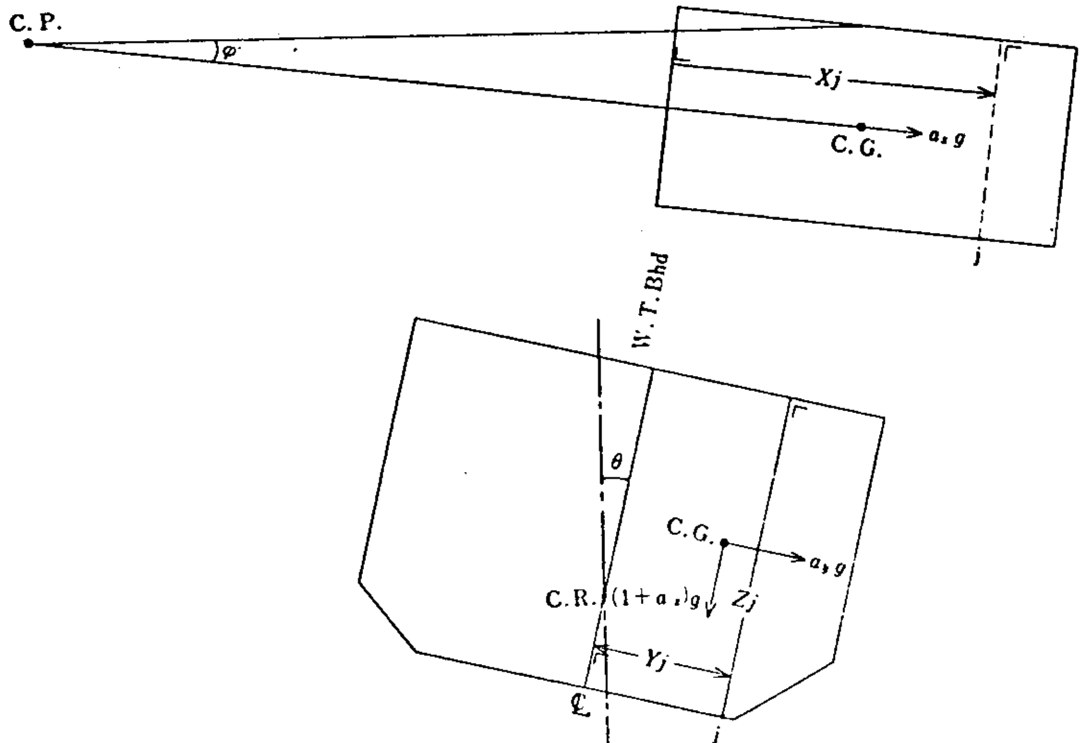
R = Inner radius of sphere, (m).

ϕ and θ = As shown in Fig. 4. 3. 3(2)(b).

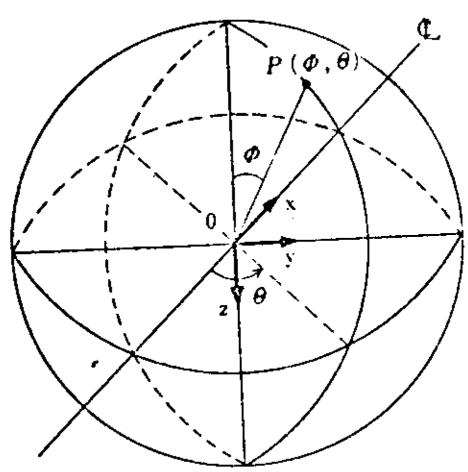
(ii) $P(\phi, \theta)_{min} = P_o + 0.1\ \gamma R\ (1 + a_z)\ (1 - \cos\phi)$

where:

P_o, γ, R and a_z = As specified in preceding (i).



NK Fig. 4.3.3 (2) (a) Prismatic Tank



NK Fig. 4.3.3 (2) (b) Spherical Tank

(c) Horizontal cylindrical tanks

Internal pressure $P(X_i, \phi)$, (kg/cm^2) at any point on t wall of the cylindrical tank installed horizontal along longitudinal direction of the ship is to be considered referring to both of the following loads:

(i) $P(X_i, \phi) = P(X_i, \phi)_{st} + P(X_i, \phi)_{dyn}$

$P(X_i, \phi)_{st} = P_o + 0.1\ \gamma R(1 - \cos\phi)$

$P(X_i, \phi)_{dyn} = P_1^2 + P_2^2 + P_3^2$

$P_1 = 0.1\ \gamma X_i a_x$

$P_2 = 0.1\ \gamma R(\ 1 + a_y^2 - a_y \sin\phi - 1)$

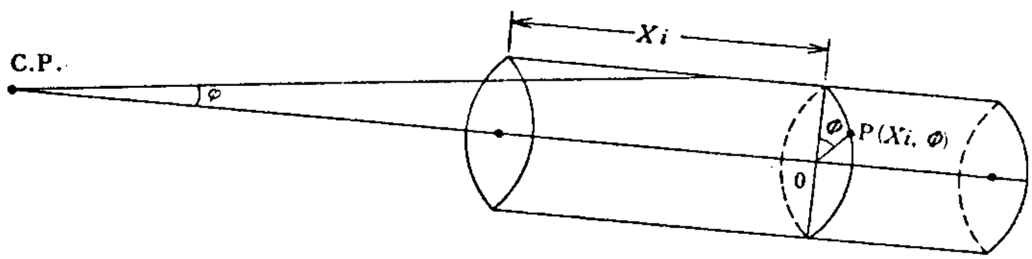
$P_3 = 0.1\ \gamma R a_z (1 - \cos\phi)$

where:

ϕ and X_i = As shown in Fig. 4.3.3(2)(c).

R = Inner radius of cylinder, (m).

P_o, γ, a_x, a_y and a_z = As specified in preceding (b).



NK Fig. 4.3.3 (2) (c). Cylindrical Tank

(ii) $P(X_i, \phi)_{min} = P_o + 0.1\ \gamma R(1 + a_z)(1 - \cos\phi)$

where:

P_o, γ, R and a_z = As specified in preceding (i).

(d) Internal pressure distribution of tanks with other sha may be obtained according to the consideration based the preceding (a) to (c)."

A.2.13.7 United States Coast Guard

No specific guidelines are given, but see A.2.11 - Acceleration at Tank Center of Gravity.

A. 2. 13. 8 International Association of Classification Societies

All tanks are subject to the following regulations concerning dynamic internal pressure:

"2. 21 The following formula gives the value of internal pressure head (or design liquid pressure), in meters of fresh water, resulting from the design vapor pressure P_o and the liquid pressure defined in 2. 22 but not including effects of liquid sloshing

$$h_{eq} = 10\ P_o + (h_{gd})_{max}$$

Equivalent procedures may be applied.

"2. 22 The internal liquid pressures are those created by the resulting acceleration of the center of gravity of the cargo due to the motions of the ship (see A. 2. 4). The following formula gives the value of internal pressure head, in metres of fresh water, resulting from combined effects of gravity and dynamical accelerations:

$$h_{gd} = a_\beta\ Z_\beta\ \gamma$$

where

a_β is the dimensionless (i. e., relative to the acceleration of gravity) acceleration resulting from gravitational and dynamical loads in an arbitrary direction β (see Fig. 1).

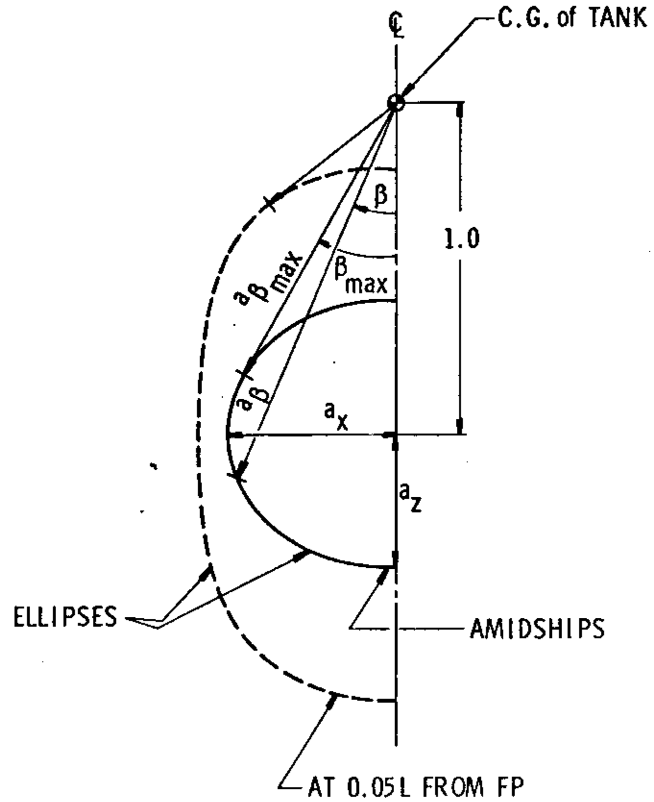
Z_β is the largest liquid height in meters above the point where the pressure is to be determined, in the β direction (see Fig. 2).

γ is the maximum specific weight of the cargo, in t/m^3, at the design temperature.

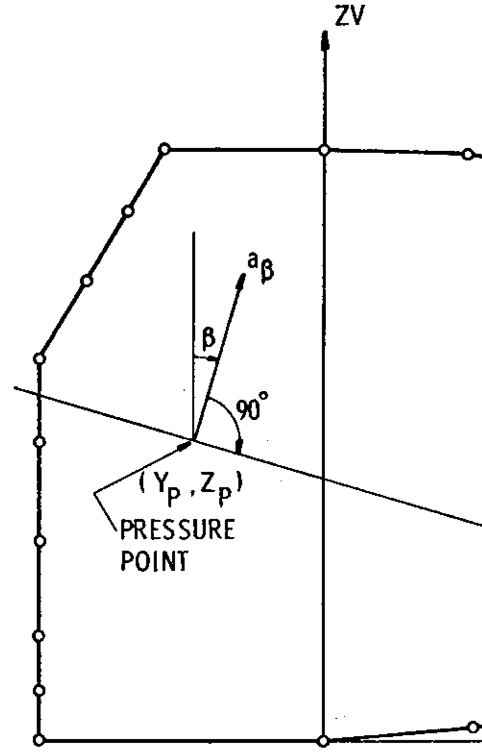
The direction β which gives the maximum value $(h_{gd})_{max}$ of h_{gd} is to be considered.

Where acceleration in three directions needs to be considered an ellipsoid is to be used instead of the ellipse in Fig. 1.

The above formula applies to full tanks."



IACS FIGURE 1. DETERMINATION OF THE LONG TERM ACCELERATION FOR COMPUTING DYNAMIC INTERNAL PRESSURE



IACS FIGURE 2. LOCATION OF LA
COMPUTING DYNAMIC INT

A.2.13.9 Summary of Dynamic Internal Pressure Criteria

Independent Tanks

Dynamic internal pressure is defined for the purpose of this report as all dynamic pressures acting on the interior of the tank with the single exception of sloshing which is described in Section A.2.14. BV, DnV, GL, NK, and IACS all have similar requirements concerning the calculation of dynamic internal pressure. All agencies provide formulas that give the dynamic internal pressure in terms of an effective liquid head (meters of water). Each agency requires the calculation of dynamic internal pressure include consideration of the cargo specific gravity, the location of the point where the pressure is to be calculated, the vapor pressure (except GL), and an acceleration as described below.

GL, NK[16] and IACS require that the effects of accelerations in the x, y and z directions be considered. (GL also gives a formula for determination of pressures at the tank boundaries, the results of which are to be compared with the results of the generalized pressure calculation. For design purposes, the larger of the two loads is to be considered.) BV by contrast, requires that the calculation of dynamic internal pressures include effects of acceleration in the z direction along with the roll and pitch amplitudes. For DnV type B independent tanks, the dynamic internal pressures are to be calculated from the acceleration in the z-direction. For all other DnV independent tanks, the loads due to dynamic internal pressure are to be determined graphically.

Membrane Tanks

BV requires that each tank design be subjected to model tests to determine the behavior of the tank during cyclic pressure variations due to ship motions. GL, NK, and IACS requirements for membrane tanks are the same as for independent tanks. ABS, DnV, LR and USCG provide no specific guidelines concerning dynamic internal pressure on membrane tank designs.

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[16] NK provides separate guidelines for determination of dynamic internal pressure loadings on prismatic, spherical and horizontal cylindrical tank configurations. These different formulas essentially are the same, the differences for the most part reflect differences in the geometry of the tanks, and the choice of different coordinate systems (cartesian, spherical or cylindrical).

Integral Tanks

No requirements are given by the agencies concerning transport of LNG in integral tanks.

A.2.14 SLOSHING

A.2.14.1 American Bureau of Shipping

No specific guidelines are given.

A.2.14.2 Bureau Veritas

"5-43-13 Where holds are expected to be 40% to 90% filled, measures approved by the Administration are to be taken to prevent the risk of resonance.
The applicable values for dynamic pressures will be determined for each specific case."

"5-43-14 Bulkheads may be required to be strengthened if the expected fill conditions are such that:

$$0.1 \leq d/\ell \leq 0.3$$ "

A.2.14.3 Det norske Veritas

Independent Tanks

The Type AI independent tanks are subject to the following sloshing loads:

"C 207 If a tank may be partly filled, sloshing forces are to be taken into consideration. The internal loads for tank filling between 20% and 90% of the tank depth are as follows:

- Static pressure head for filling height 70% of the tank depth.
- Additional pressure corresponding to pressure relief valve setting. See B 201.
- Dynamic additional pressure.

The dynamic additional pressure head at the tank corners is given by:

$$h_i = \gamma \sqrt{h_{ix}^2 + h_{iy}^2} \text{ meters of water head}$$

$$h_{ix} = a_1 \left(\frac{T_x}{T_p}\right)\left(\frac{\ell_t}{L}\right)\sqrt{1 + \left(\frac{a_o\, L}{80}\right)^2}$$

$$h_{iy} = a_2 \left(\frac{T_y}{T_r}\right)\left(\frac{b_t}{B}\right)\sqrt{1 + \left(\frac{a_o\, x_o\, B}{6\, L}\right)^2}$$

Table for a_1 and a_2:

	a_1	a_2
Tank top	165	45
Tank bottom	100	30

T_x = resonant period of liquid in longitudinal direction in sec.

T_y = resonant period of liquid in transverse cirection in sec.

T_p = period of pitch in sec.

T_r = period of roll in sec.

T_x, T_y, T_p and T_r may be found from Figs. 4 and 5.

ℓ_t = free distance between tight or wash bulkheads in meters.

b_t = free distance between longitudinal tight or wash bulkheads in meters.

x_o = distance from amidship to the centre of the tank in meters.

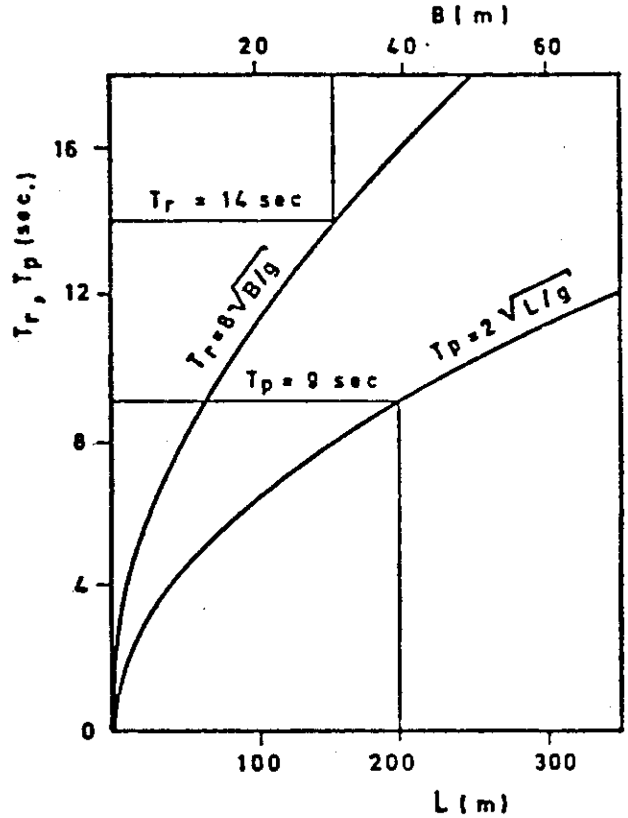
$a_o = 0.2 \dfrac{V}{\sqrt{L}} + \dfrac{30}{L}$

The dynamic additional pressure may be considered to decrease linearly to h_{ix} in the middle of transverse bulkheads (between longitudinal tight or wash bulkheads) and h_{iy} in the middle of longitudinal bulkheads (between transverse tight or wash bulkheads). The pressure variation from tank top to tank bottom may be taken as linear.
Conditions for the use of the formulae for h_{ix} and h_{iy}:

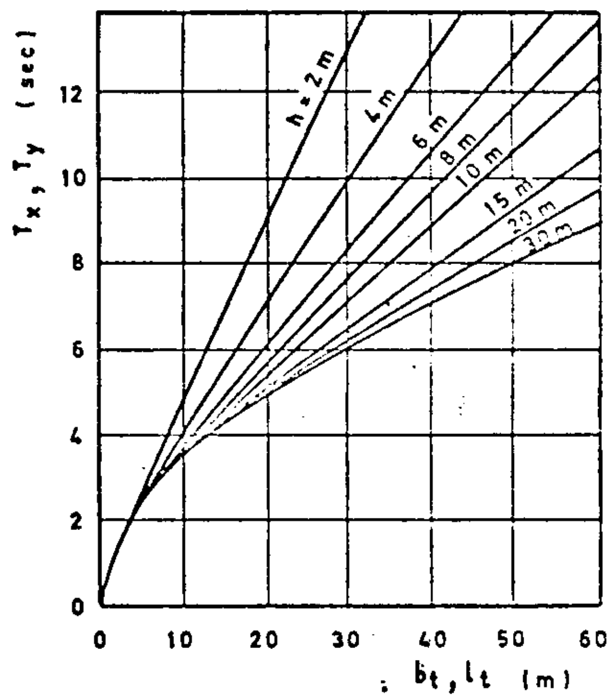
$$T_p/T_x \geq 1,25 \text{ at } 70\% \text{ filling.}$$

$$T_r/T_y \geq 1,40 \text{ at } 70\% \text{ filling.}$$

The filling height h is to be taken equal to 70% of the tank depth.
For determination of T_x use ℓ_t
For determination of T_y use b_t



DnV Fig. 4.
PERIODS OF SHIP MOTIONS.



DnV Fig. 5.
RESONANT PERIODS OF LIQUID IN TANKS.

"D 101 The scantlings of the Type AII tank's strength members are to be based on a complete structural analysis of the tank and are generally not to be less than those for independent tank, AI."

(We assume that the same sloshing loads defined for AI apply.) For Type B, no guidelines are given. Perhaps it is intended that it be the same as for Type AI.

Membrane Tanks

No specific guidelines.

A. 2. 14. 4 Germanischer Lloyds

For all tank designs, the following regulations concerning sloshing loads are given:

"26. F. 3. 8 When use of partial filling of the tanks is contemplated, investigations are to be carried out in order to avoid resonance of the liquid for pitching and rolling."

A. 2. 14. 5 Lloyds Register of Shipping

No guidelines are given.

A. 2. 14. 6 Nippon Kaiji Kyokai

Independent and Membrane Tanks

"4. 1-4. Where it is intended to partly fill cargo in tanks, the tanks are to be arranged and of suitable size to avoid, as far as possible, synchronization of natural period of oscillation of liquid in the tank with the natural periods of rolling and pitching of the ship."

"4.3.3(10) Where a tank may be partly filled below 70% of the depth, sloshing forces are to be taken into consideration acting on the tank upwards near from the liquid level on such loaded condition. The sloshing effects are to be generally studied by model test, etc. Where accepted by the Society, sloshing forces for a prismatic tank may be estimated as the static water head shown in the following formulae.

$$h_{st} = \sqrt{h_{sx}^2 + h_{sy}^2}$$

$$h_{sx} = \gamma \ell_t \, \{\varphi + \eta \, (h/D_t) \, (1 - h/D_t) \, (T_L/T_P)^2\} \quad \text{provided } T_L \leqq T_P$$

$$h_{sy} = \gamma b_t \, \{\theta + \eta \, (h/D_t) \, (1 - h/D_t) \, (T_T/T_R)^2\} \quad \text{provided } T_T \leqq T_R$$

where:

Standard value of η may be taken as 2.4, where provided with no extrusion such as girder, etc. on the inner surface of the tank. Where provided with extrusions such as girder, etc. on the inner surface of the tank, the value may, however, be increased or decreased according to h/D_t, tank forms, size and arrangement of girders, etc. at the discretion of the Society.

h_{st} = Water head at the cross line of transverse and longitudinal tank boundary (at the tank corner), (m).

h_{sx} = Water head at the middle of transverse tank boundary, (m).

h_{sy} = Water head at the middle of longitudinal tank boundary, (m). The water head between the corner and the middle may be considered to decrease linearly from h_{st} to h_{sx} (or h_{sy}) in the middle of transverse tank boundary (or longitudinal boundary).

T_L and
T_T = Resonant period of liquid in longitudinal and transverse directions respectively, (sec). And are given as follows;

$$T_L = 2\pi / \sqrt{\frac{\pi g}{\ell_t} \tanh \frac{\pi h}{\ell_t}}$$

$$T_T = 2\pi / \sqrt{\frac{\pi g}{b_t} \tanh \frac{\pi h}{b_t}}$$

T_P and
T_R = Periods of pitching and rolling, (sec), which are are generally given by the following formulae respectively.

$$T_P = 0.6\sqrt{L}$$

$$T_R = 0.8\ B / \sqrt{\overline{GM} - \frac{\gamma}{12\ W}\ \ell_t b_t^3}$$

φ and
θ = Amplitudes of pitching and rolling, (rad.), which are generally given by the following formulae, respectively, but θ need not exceed 0.611.

$$\varphi = 0.175 - \frac{0.025}{100} L$$

$$\theta = 1.667\ \pi / \sqrt{L} + 0.175$$

ℓ_t and

b_t = Tank length and tank breadth respectively, (m). Where provided with swash bulkheads, near the middle of tank, they may be replaced by the following respectively.

$$(1 + 1.2a)\ \frac{\ell_t}{2}$$

$$(1 + 1.2a)\ \frac{b_t}{2}$$

a = Opening ratio of swash bulkhead.

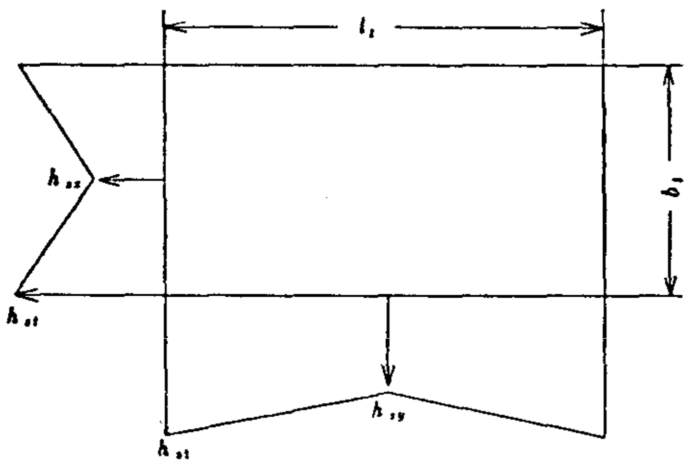
h = Liquid level, (m).

g = Acceleration of gravity, (m/sec^2).

γ = Design specific gravity of cargo, (t/m^3).

W = Displacement of the ship at the partly loaded condition, (t).

$\overline{GM}$= Distance from the center of gravity of the ship to the transverse metacenter, (m)."



NK Fig. 4.3.3 (10) Distribution of Sloshing Forces

A. 2. 14. 7 United States Coast Guard

No specific guidelines are given. Test requirements for membrane tanks make no specific reference to partially filled tanks or fluid resonance.

A.2.14.8 International Association of Classification Societies

"2.51 When partial filling is contemplated, the risk of significant loads due to sloshing induced by any of the ship motions mentioned in A.2.4.6 (vertical, transverse and longitudinal accelerations) is to be considered."

"2.52 When risk of significant sloshing induced loads is found to be present, special tests and calculations will be required."

A.2.14.9 Summary of Sloshing Pressure Criteria

Independent Tanks

BV requires for tanks between 40% and 90% full-tank, some means be provided to prevent resonance of the liquid cargo. The dynamic pressures are to also be determined for each specific case. For tanks between 10% and 30% full, bulkhead stiffeners may be required. For tanks that may be partially filled, GL requires that the resonant frequency of the tank be determined. This frequency should not be near the pitch and roll frequencies in order to avoid liquid resonance. If the risk of significant sloshing-induced loads is found to be significant, IACS requires that special tests and calculations be made. DnV and NK provide similar guidelines for determining the dynamic liquid pressure (slosh pressures) in partially-filled tanks. Both agencies give the dynamic additional pressure as a function of ship and tank dimensions, specific gravity of the cargo, natural period of oscillation of the cargo in the x and y directions, and natural period of oscillation for roll and pitch motions.

DnV provides the resonant periods of liquid motion and periods of ship motions for a 70% tank filling. From this information, the sloshing pressure (in meters of water) at the corners of the tank can be determined. To determine the dynamic pressure due to sloshing for any other point on the tank, DnV specifies a linear interpolation. In addition to the dynamic additional pressure, DnV requires consideration of internal loads due to static head for 70% filling and an additional pressure corresponding to the pressure relief valve setting. The value of the sloshing pressure based on a 70% filling is assumed to be valid at fill depths between 20 and 90%. Below 20% no values are given and above 90% the tank is assumed full.

NK provides formulas for calculating the resonant periods of the cargo and the roll and pitch periods as a function of the ship dimensions, and the tank fill depth. From this information, the slosh pressures can be calculated at the tank corners and the middle of the transverse and longitudinal sides. For any other point in the tank, dynamic pressures can be obtained by a linear interpolation.

ABS, LR, and USCG have no specific regulations for slosh loads on independent tanks.

Membrane Tanks

BV, GL, NK, and IACS provide the same requirements for membrane tanks as for independent tanks. The other agencies provide no guidelines concerning slosh loads on membrane or integral tanks.

Integral Tanks

No requirements for sloshing loads on integral tanks are given by the agencies.

A.2.15 VIBRATIONS

A.2.15.1 American Bureau of Shipping

Independent Tanks

No specific requirements are given.

Membrane Tanks

"24.31.2 Primary containers which are not self-supporting may be given consideration, provided all details, arrangements and materials of the primary container, insulation and the supporting structure are suitable for the service conditions. Preliminary tests are to be made to ascertain that the design arrangements will function satisfactorily in all respects. The tests are to simulate the most severe operating conditions including minimum service temperature, static and dynamic loads, hull vibration and slamming."

A.2.15.2 Bureau Veritas

Independent Tanks

No specific requirements are given.

Membrane Tanks

"22-36-21 As a rule, each design proposed for integrated tanks is to be submitted to model tests designed to check its behavior under the effect of:

- cyclic pressure variations due to ship motions
- cyclic deformations of the ship
- vibrations of the ship."

A.2.15.3 Det norske Veritas

Independent Tanks

"B 601 Design of hull and cargo tanks, choice of machinery and propellers are to be aimed at keeping vibration exciting forces and vibratory stresses low.
Calculations or other appropriate information pertaining to the excitation forces from machinery and propellers, are to be submitted for tanks, type AII, and may be required, in special cases, for tanks, type AI and B. Full-scale measurements of vibratory stresses and/or frequencies may be required."

Membrane Tanks

Membrane tanks are subject to the same loads as independent tanks.

A.2.15.4 Germanischer Lloyd

No specific regulations are given.

A.2.15.5 Lloyds Register of Shipping

No specific regulations are given.

A.2.15.6 Nippon Kaiji Kyokai

The vibration exciting force due to propeller and machinery on tank structures is to be considered.

Independent Tanks

Regarding vibration loads on Type B independent prismatic tanks, the rules state:

"4.6.2-11 The scantlings of stiffened plates and girders are to be designed so that a resonance between the frequencies of those structures and an exciting source causing vibrations does not give any bad effect to the tank. In this case, the natural frequencies of the stiffened plates and the girders may be taken as a minimum value at the immersed condition."

Type B independent pressure vessel configuration tanks are to be subject to the same requirements as Type B independent prismatic tanks.

Membrane Tanks

No specific requirements.

A. 2. 15. 7 United States Coast Guard

Independent Tanks

No specific requirements.

Membrane Tanks

Membrane tank designs are to be designed so that the mechanical integrity of the system is maintained for the life of the vessel. Moderate scale fatigue testing, a resonance search, and a prototype test are required per Section IV-C. 3. c.(4).

Integral Tanks

No specific requirements.

A. 2. 15. 8 International Association of Classification Societies

Vibration loads are considered only in the insulation material. Tests are to be conducted to ensure that the insulation materials have a sufficient resistance to vibrations.

A. 2. 15. 9 Vibration Summary

Independent Tanks

Only DnV and NK present requirements specific to vibrations on independent tanks. DnV requires that stresses due to vibrations from machinery and the propellers be kept low. Calculations or other pertinent information (such as test data) are to be submitted to the agency for review. Full-scale measurements of vibratory stresses may be required. NK requires consideration of vibration effects from machinery or propellers on the tank structure. The design of stiffeners or plating is to be such that resonances of these structures have no deleterious effect on the tank. IACS requires that insulation materials have a sufficient resistance to vibrations.

Membrane Tanks

ABS, BV, and USCG require preliminary testing to ensure that the design is sound with respect to vibratory loading. In addition, the USCG requires that a resonance search be made by either model test or mathematical modeling to verify that the resonant frequencies for the tanks are far removed from those generated by the vessel (machinery, propeller, etc.). DnV and IACS rules for membrane tanks are the same as for independent tanks. GL, LR, and NK present no specific guidelines for membrane tanks concerning vibrations.

Integral Tanks

No specific requirements concerning vibration loads on integral tanks are provided by the various agencies.

A.2.16 FATIGUE LOADS

A.2.16.1 American Bureau of Shipping

No specific requirements are given.

A.2.16.2 Bureau Veritas

Independent Tanks

"22.23.41 For self-supporting gravity cargo tanks, as well as for pressure cargo tanks requiring a secondary barrier, the Administration may consider a reduction of this secondary barrier provided all necessary justifications are supplied and, in particular:

- a complete analysis of the stresses due to the actual static and dynamic loads
- a fatigue analysis
- a fracture mechanics analysis
- a buckling analysis."

Membrane Tanks

No specific requirements are given.

A.2.16.3 Det norske Veritas

Independent Tanks

Type A1, A2 and in some cases type B independent tanks are subject to the following tentative rules for fatigue loads and fatigue analysis:

"6.B.903 The load spectrum for design against fatigue is to be taken as the most probable largest load spectrum the ship will experience during 10^8 wave encounters on the North-Atlantic.

Generally, the load spectrum shown in Fig. 1 may be used. This load spectrum may be replaced by a number of 8 fatigue loads, each of which is represented by a certain number of cycles, n_i, and an alternating load $\pm P_i$. Corresponding values of P_i and n_i are given by:

$$P_i = \frac{17 - 2i}{16} P_o$$

$$n_i = 0.9 \cdot 10^i$$

i = 1, 2, 3, 4, 5, 6, 7, 8.
P_o = load on probability level $Q = 10^{-8}$.

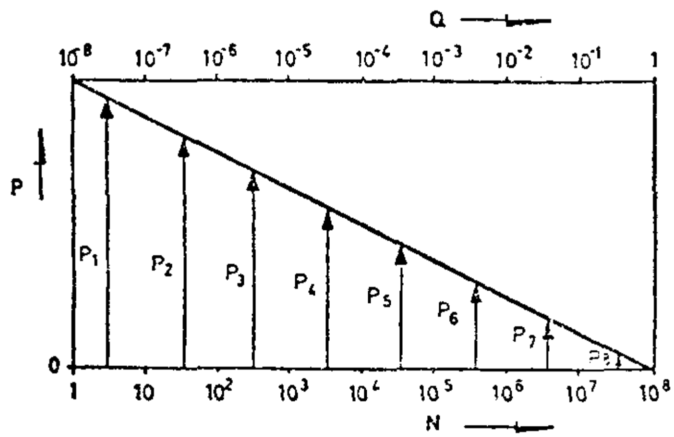


Fig. 1.
DnV LONG TERM WAVE-INDUCED LOAD SPECTRUM.

N = number of wave encounters

Q = probability of load exceedance

P = most probable largest wave-induced load."

"B1101 An analysis according to 1102 and 1103 is to be carried out for tanks, type A II, and may, in special cases, be required for tanks, Type B."

"B1102 A fatigue analysis is to be carried out for parent material and welded connections at areas where high dynamic stresses or large-stress concentrations may be expected.
The fatigue properties are to be well documented for the parent material and weld metal being used in the design.
For less investigated and documented materials, the data on fatigue properties are to be determined experimentally.
Due attention is to be paid to the effect of:

- Specimen size and orientation.
- Stress concentration and notch sensitivity.
- Type of stress.
- Mean stress.
- Type of weld.
- Welding condition.
- Working temperature.

The number of specimens to be tested at each stress level is not to be less than 6.
The fatigue strength of the structure considered is to be illustrated by Wöhler curves."

"B1103 The fatigue analysis is to be based on the fatigue loading given in 903. The number of complete stress cycles due to loading and unloading is in general to be 1000. The cumulative effect of the various fatigue loads is to satisfy the following requirement:

$$0.9 \sum_{i=1}^{i=8} \left(\frac{10^i}{N_i}\right) + \frac{10^3}{N_9} < 0.5$$

N_i = number of cycles to fracture for wave-induced fatigue load number i, according to Wöhler curves.
N_9 = number of cycles to fracture for the fatigue load due to loading/unloading.
The effect of stresses produced by static load as given in 200 is to be taken into account."

Membrane Tanks

Membrane tanks are subject to the same fatigue loads as independent tanks.

A. 2. 16. 4 Germanischer Lloyd

With respect to fatigue loads, the rules state:

"26. G. 3. 4 Where fatigue is to be considered when ascertaining the tank structure scantlings, the fatigue life is to be determined by long term distribution calculation on a range of wave encounters to be expected during the ship's lifetime.

For all tank designs, the following material is to be used as a guideline for calculating fatigue loads:

"26. H. 5. 1 A fatigue analysis is to be carried out for parent material and welded connections."

"26. H. 5. 2 If necessary the data on fatigue properties of parent material and welding metal being used are to be determined experimentally. The fatigue strength of the structure is to be illustrated by Wöhler curves for the "as built" condition."

"26. H. 5. 3 The fatigue analysis is to be based on the fatigue loading according to G. 3. 4.
The cumulative effect of the various fatigue loads σ_i of m steps of the stepped cumulative frequency distribution according to G. 3. 4 for the part of structure considered is to satisfy the following condition:

The sum of the relative number of cycles $\frac{n_i}{N_i}$ corresponding to the fatigue loads σ_i of the steps and the relative number of cycles $\frac{n_{BE}}{N_j}$ must be less than 0. 5, i. e.,

$$\sum_{i=1}^{m} \frac{n_i}{N_i} + \frac{n_{BE}}{N_j} < 0.5$$

m = number of steps of the cumulative frequency distribution of loads above the fatigue life of the Wöhler curve representing the structure considered

n_i = number of the cycles of step i of the cumulative frequency distribution according to G. 3. 4

N_i	=	number of cycles to fracture according to load σ_i in Wöhler experiment
σ_i	=	load of step i of the cumulative frequency distribution
n_{BE}	=	number of cycles due to loading and unloading operations the considered structure is exposed If not known, $n_{BE} = 10^3$.
N_j	=	number of cycles to fracture according to load σ_{BE} in Wöhler experiment
σ_{BE}	=	load due to loading and unloading operations."

A. 2. 16. 5 Lloyds Register of Shipping

No specific requirements are given.

A. 2. 16. 6 Nippon Kaiji Kyokai

All tank designs are subject to the following regulations on fatigue loads:

"4. 3. 3. (8) Fatigue loads having significant effects on the tank are be generally obtained as the load spectrum specified in 4. 3. 2-1(2), which follows. In this case, the total numb cycles are generally to be taken as 10^8. This load spe trum used for the fatigue analysis specified in 4. 2. 4 m be replaced by a number of 8 fatigue loads, each of whi is represented by a certain number of cycles, N_i, and alternating load S_i. Corresponding values of S_i and] are given as follows (see Fig. 4. 3. 2-1(2)).

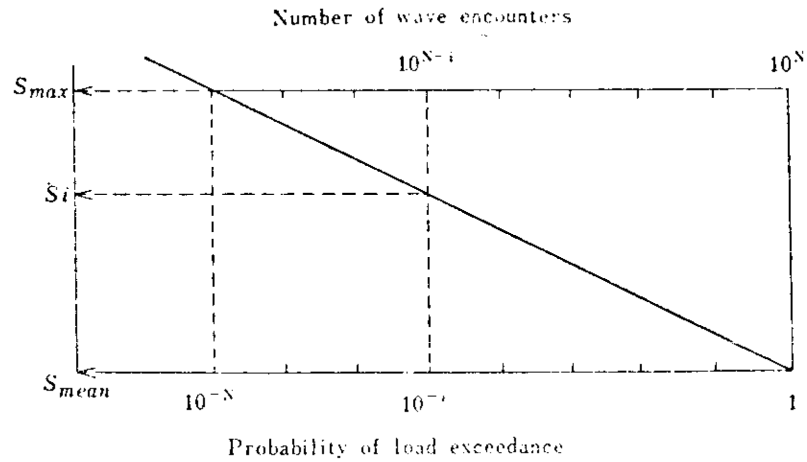
$$S_i = \frac{17 - 2i}{16} S_{max}$$

$$n_i = 0.9 \times 10^i$$

where:

$i = 1, 2, \ldots, 8.$

S_{max} = Maximum expected value on probability $Q = 10^{-8}$. "



NK Fig. 4.3.2-1 (2) Long-term Wave-induced Load Spectrum

"4.3.2-1(2) A load spectrum may be taken as a straight line approximately shown in Fig. 4.3.2-1(2), provided that the maximum expected value (S_{max}) at a probability level $Q = 10^{-N}$ is estimated by a suitable method subject to the satisfaction of the Society. In this case, the total number of fatigue cycles is 10^N for an operating ship."

"4.2.4 Analysis of fatigue strength may be required, where deemed necessary by the Society. In this case, it is assumed that the total life of ship may be 20 years, and the standard cumulative effect of the fatigue loads is as obtained from the following formula:

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \ldots\ldots\ldots + \frac{n_i}{N_i} + \frac{10^3}{N_j} < 0.5$$

where:

$n_1, n_2, \ldots n_i$ = Number of stress cycles at the stress level suitably selected during the total life of the ship.

$N_1, N_2, \ldots N_i$ = Number of cycles to fatigue fracture for the respective stress levels according to S-N curves.

N_j = Number of cycles to fracture for fatigue loa due to loading and unloading."

Independent Tanks

For the type B independent prismatic tank and type B independent pressure vessel tank:

"4.6.2-8 A fatigue analysis is to be carried out for parent materials and welded joints at areas where high dynamic stresses or large stress concentrations may be expected. In this case, S-N curves are to obtained from tests, etc. taking into account the following:

(1) Specimen size and orientation.
(2) Stress concentration and notch sensitivity.
(3) Type of stress.
(4) Mean stress.
(5) Welding condition.
(6) Working temperature.

In addition, the number of specimens to be tested at each stress level is not to be less than 6. In this case, the S-N curves at the suitable confidence level accepted by the Society are to be obtained. The fatigue load is to be according to the requirements of 4.3.3(8), and the cumulative effect of the various fatigue loads is to satisfy the requirements specified in 4.2.4."

Membrane Tanks

Fatigue strength analysis of the Type B semi-membrane tank is to be generally in accordance with the requirements of 4.6.2.8 (same as for Type B independent prismatic tanks).

A.2.16.7 United States Coast Guard

Independent Tanks

No requirements are given.

Membrane Tanks

Moderate scale fatigue testing of the IMT design is required with the following restraints:

"((1)) Number of cycles based on anticipated conditions of primary hull bending for the life of the vessel. 10^8 cycles for an anticipated 20 year life is a reasonable figure."

"((2)) Structure statically prestressed in tension to the maximum amount caused by cargo cooling, static head pressure of cargo, and still water hull deflection. This means that under most circumstances, permission will be granted to run the tests at ambient temperature and pressure."

"((3)) Structure cycled above and below the static level an amount equivalent to that caused by maximum at-sea hull deflection plus the maximum caused by the earlier discussed dynamic loading criteria. A .04% strain is a reasonable figure for estimating purposes."

Integral Tanks

No requirements are given.

A.2.16.8 International Association of Classification Societies

"2.43 When design against fatigue is to be considered, the dynamic spectrum is determined by long-term distribution calculation based on the operating life of the ship (normally taken to correspond to 10^8 wave encounters). If simplified dynamic loading spectra are used for the estimation of the fatigue life, these are to be specially considered by the Classification Society."

Independent Tanks

"3.51 For independent type B tanks, the effects of all dynamic and static loads are to be used to determine the suitability of the structure with respect to:

- plastic deformation
- buckling
- fatigue failure
- crack propagation.

Statistical wave load analysis in accordance with 2.4 (dynamic loads due to ship motions, finite element analyses or similar methods and fracture mechanics analyses or equivalent approach), are normally to be carried out."

"3.55 Where deemed necessary by the Classification Society, model tests may be required to determine stress concentration factors and fatigue life of structural elements."

"3.56 The cumulative effect of the fatigue load is to comply with the following formula:

$$\sum \frac{n_i}{N_i} + \frac{10^3}{N_j} < C_W$$

where

n_i = number of stress cycles at each stress level during the life of the ship

N_i = number of cycles to fracture for the respective stress level according to the Wöhler curve

N_j = number of cycles to fracture for the fatigue loads due to loading and unloading

C_W = is a coefficient to be determined by the Classification Society dependent on the test procedures and data used to establish the Wöhler curve ($C_W \leq 1$)."

"4.19b Stress in type C independent tanks may be limited by fatigue analysis, crack propagation analysis and buckling criteria."

Membrane Tanks

"3.21 For membrane tanks, the effects of all static and dynamic loads are to be considered to determine the suitability of the membrane and of the associated insulation with respect to plastic deformation and fatigue."

"3.22 Before approval is granted, a model of both the primary and secondary barrier, including corners and joints, is normally to be tested to verify that it will withstand the expected combined strains due to static, dynamic, and thermal loads. Test conditions are to represent the most extreme service conditions the tank will see in its life. Material tests are to insure that aging is not liable to prevent the materials from carrying out their intended function."

"3.3 For semi-membrane tanks, structural analysis is to be performed in accordance with the requirements for membrane tanks taking into account the internal pressure given in 2.2" (A.2.13.8 of this report)."

A.2.16.9 Fatigue Load Summary

Independent Tanks

BV regulations specify only,that a reduction in the secondary barrier may be allowed if a fatigue analysis is made. DnV, GL, NK, and IACS by contrast are very specific as to the type and magnitude of fatigue analysis required. All of these agencies require that the load spectrum for fatigue is to be the most probable largest load spectrum the ship will experience in 10^8 wave encounters. All these agencies will allow a straight line load spectra to be used (except GL which does not specify), provided that the maximum expected value is estimated by some means approved by the agency. A fatigue load analysis is required on the following tank designs: DnV: A-II and in some cases B; GL: All tank configurations; NK: B; IACS: B. In addition, IACS may require that model tests be conducted for some tank designs to determine stress concentration factors and fatigue life of structural elements. ABS, LR and USCG provide no regulations for the calculation of fatigue loads on independent tanks.

Membrane Tanks

The USCG requires moderate-scale fatigue testing of the IMT design. The number of fatigue cycles is to be 10^8, and the structure is to be prestressed to the maximum deflection caused by cargo cooling, static liquid head, and still-water hull deflection. The structure is to be cycled above and below the static level by an amount equivalent to that caused by the maximum at-sea hull deflection plus the maximum wave-induced loads. The IACS requires that a model of the primary and secondary barrier be tested to verify that it will withstand the maximum anticipated strains due to static, dynamic and thermal loads. Material tests are also to be conducted to ensure that aging of materials will not affect the integrity of the tank design. IACS semi-membrane tanks are to be subjected to the structural analysis that membrane tanks undergo, while taking into account the dynamic internal pressure loading. DnV, GL and NK have the same requirements for membrane tanks as for independent tanks. ABS, BV, and LR provide no guidelines for membrane tanks.

Integral Tanks

No specific guidelines concerning fatigue loads are given by the various agencies.

A.2.17 FRACTURE LOADS

A.2.17.1 American Bureau of Shipping

No specific requirements are given.

A. 2. 17. 2 Bureau Veritas

Independent Tanks

"22-23-41 For self-supporting gravity cargo tanks, as well as for pressure cargo tanks requiring a secondary barrier, the Administration may consider a reduction of this secondary barrier provided all necessary justifications are supplied and, in particular:

- a complete analysis of the stresses due to the actual static and dynamic loads
- a fatigue analysis
- a fracture mechanics analysis
- a buckling analysis. "

Membrane Tanks

No specific requirements are given.

A. 2. 17. 3 Det norske Veritas

Independent Tanks

Independent tanks are subject to the following rules for fracture loads and fracture analysis:

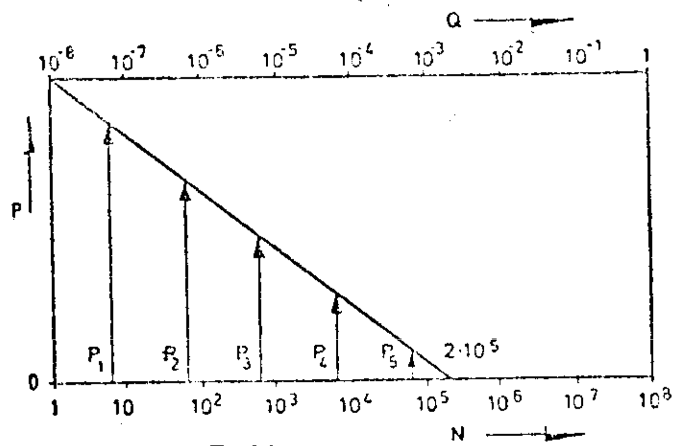
"6. B. 904 The load spectrum for design against crack propagation is to be taken as the load spectrum representing the worst period of 15 days in the most probable largest load spectrum the ship will experience during 10^8 wave encounters on the North-Atlantic. Generally, the load spectrum shown in Fig. 2 may be used. This load spectrum may be replaced by a number of 5 fatigue loads, each of which is represented by a certain number of cycles, n_i, and an alternating load $\pm P_i$. Corresponding values of P_i and n_i are given by:

$$P_i = \frac{5.5 - i}{5.3} P_o$$

$$n_i = 1.8 \cdot 10^i$$

i = 1, 2, 3, 4, 5.

P_o = load on probability level $Q = 10^{-8}$.



DnV Fig. 2.
LOAD SPECTRUM REPRESENTING THE WORST PERIOD OF 15 DAYS IN THE LONG TERM INDUCED LOAD SPECTRUM.

N = number of wave encounters.

Q = probability of load exceedance.

P = most probable largest wave-induced load."

"B1001 An analysis according to 1002 is to be carried out for tanks, type A II, and may be required, in special cases, for tanks, type B."

"B1002 A fatigue crack propagation analysis is to be carried out for areas with high dynamic stresses. The analysis is to consider propagation rates in parent material, weld metal and heat-affected zone.
The fracture mechanical properties are to be well documented for the material, comprising parent material and weld metal, and thicknesses used in the design. For less investigated and documented materials, the data on fracture mechanical properties are to be determined experimentally according to ASTM E399-70T.

Depending on material, fracture mechanical properties determined under dynamic loading may be required. The method used for this determination has to be approved by the Society. Assuming that a through thickness crack of length a_i exists, the length a_f, which this crack will grow to under dynamic loading, is to be determined.
a_f is to be based on a stress spectrum corresponding to the worst period of 15 days in the long term load spectrum as given in 904. The effect of stresses produced by static loads

as given in 200 is to be taken into account. The permissible length of a_f is to be considered by the Society in each separate case. a_f is to be taken equal to the minimum flaw size that can be detected by means of monitoring systems, for instance gas detectors, but is not to be taken less than the plate thickness.
In particular cases, a special evaluation of crack growth, for instance by means of experiments, may be required."

Membrane Tanks

Membrane tanks are subject to the same fracture loads as independent tanks.

A.2.17.4 Germanischer Lloyd

With respect to fracture loads, the rules state:

"26.G.3.5 Where crack propagation is to be considered when ascertaining the tank structure scantlings, the largest load spectrum expected to occur during the most severe 14 days' weather period is to be taken."

For all tank designs, the following material is to be used as a guideline for calculating fracture loads.

"26.H.6.1 Fracture mechanics analysis shall consider propagation rates in parent material, weld metal and heat-affected zone."

"26.H.6.2 The fracture mechanical properties are to be documented for the various thicknesses of parent material and weld metal alike, possibly by experiment according to ASTM E399-70T."

"26.H.6.3 It is to be determined to which length an assumed through thickness crack will grow to under dynamic loading. The calculation is to be based on a stress spectrum as stipulated under G.3.5. The initial length of the existing crack is to be taken equal to the minimum flaw size that can be detected by means of a monitoring system (e.g., gas detectors), however, not less than the plate thickness."

A.2.17.5 Lloyds Register of Shipping

No specific requirements are given.

A. 2. 17. 6 Nippon Kaiji Kyokai

Independent Tanks

All independent tank configurations are subject to the following regulation concerning fracture loads. Fracture loads specific to particular configurations are found after the general statement.

"4. 3. 3(9) Fatigue loads used for the calculations of the fatigue crack propagation specified in 4. 2. 5 (see below) are to be generally obtained as the load spectrum representing the fixed period of time specified in 4.11.1[17] in the most probable largest load spectrum the ship will experience during 10^8 wave encounters. In this case the total number of cycles is generally to be taken as 2×10^5. Where accepted by the Society, this load spectrum may be replaced by a number of 5 fatigue loads, each of which is represented by a certain number of cycles, N_i, and an alternating load S_i. Corresponding values of S_i and N_i are given as follows (see Fig. 4. 3. 2-1(2)).

$$S_i = \frac{5.5 - i}{5.3} S_{max}$$

$$n_i = 1.8 \times 10^i$$

where

$i = 1, 2, \ldots\ldots, 5$

S_{max} = As specified in preceding (8). (See fatigue loads)"

For Type B independent tanks, the rules state:

"4. 6. 2-9 Concerning the design of secondary barriers onboard Type B independent prismatic tanks, the fracture mechanics analysis is to be carried out in accordance with the requirements specified in 4. 2. 5 using the fatigue loads specified in 4. 3. 3(9), and it is to be confirmed that a crack does not propagate up to the permissible crack length in an assumed period."

"4. 2. 5-1 For type B tanks, the fracture mechanical properties, namely the fatigue crack propagation and fracture toughness of the parent materials and welded joints (including heat affected zone) are to be made clear at the lowest working temperature. In special case, where deemed

[17]Not to be less than 15 days.

necessary by the Society, for tanks other than type B it may be required to examine the fracture mechanics analysis."

"4.2.5-2 The method of the fracture mechanics analysis for type B tanks is to be as given in the following Sub-Paragraphs:

(1) Calculate maximum dynamical stress induced in the tank according to the requirements specified in 4.3 (design loads) as exactly as possible, and obtain the stress spectrum specified in 4.3.3(9).

(2) Assume a size of crack which can be detected by means of monitoring systems taking into account of the kind of stress, the structural details of the tank, the tank materials, the detecting means of leakage, etc.

(3) Calculate a crack length to which the through thickness crack specified in preceding (2) propagates under the dynamical stress specified in 4.3.3(9).

(4) Calculate a critical crack length in consideration of the fracture toughness of the materials and the maximum dynamical stress specified in preceding (1).

(5) Confirm that the crack length obtained in preceding (3) is considerably smaller compared with the critical crack length in preceding (4)."

Membrane Tanks

No specific requirements are given for fracture loads on membrane tanks; however, see related regulations on fatigue loads. Fracture mechanics analysis for semi-membrane designs is generally to be in accordance with the requirements of 4.6.2-9 (Type B independent tanks).

A.2.17.7 United States Coast Guard

"IV.C.2.a.(2) Check the design[18] using analytical tools such as three-dimensional finite element or finite difference analysis and photoelastic analysis in order to determine the maximum stresses in the material and the stress field patterns, paying particular attention to support attachments and tolerance limits."

---

[18] For IST and SPT Tanks

"IV.C.2.a.(3) Subject the material to a fracture mechanics analysis in order to determine the critical crack size and the crack propagation rate with a given maximum stress (either at an assumed level or as computed in (2) above). Also determine the fracture mechanical properties (either from tables for well documented materials or determined experimentally for lesser research materials) for the material being used in the design."

"IV.C.2.a.(4) Determine the minimum flaw size that will allow the passage of sufficient gas to be sensed by the gas detectors."

"IV.C.2.a.(5) Using the minimum flaw size of (4) above, determine the length this crack will grow to during the greatest of the following:

(a) Two weeks
(b) The time required to offload the cargo in an emergency, including the time to remove the cargo contained between the primary and secondary barriers.
(c) The anticipated average vessel running time between cargo loading and discharge points."

"IV.C.2.a.(6) Compare the crack length after growth with the critical crack length. If the critical crack length is larger than the crack length after growth (designs approved to date have had ratios of approximately 10 : 1) then the design is acceptable."

Membrane Tanks

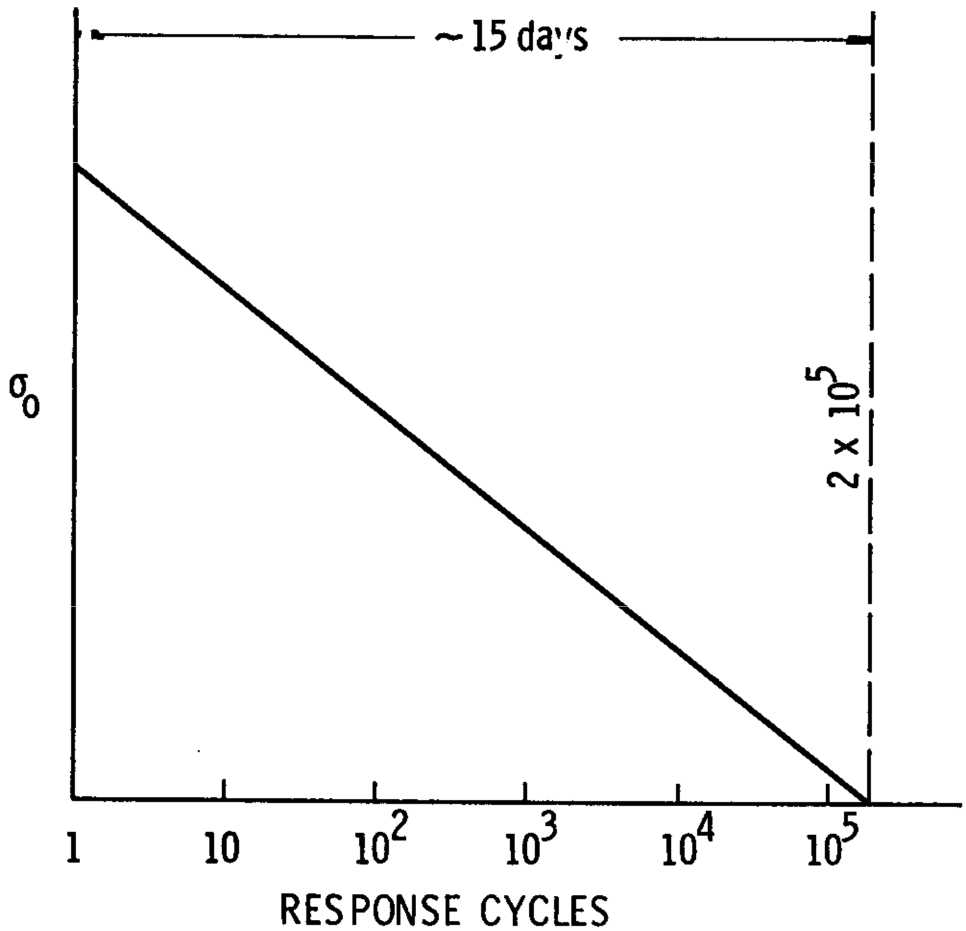
No requirements are given.

Integral Tanks

No requirements are given.

A.2.17.8 International Association of Classification Societies

"2.44 In order to practically apply crack propagation estimates, simplified load distributions over a period of 15 days may be used. Such distributions may be obtained as indicated in Fig. 3."



IACS FIGURE 3. SIMPLIFIED LOAD DISTRIBUTION FOR ESTIMATING CRACK PROPAGATION

Independent Tanks

"3.51 The effects of all dynamic and static loads are to be used to determine the suitability of the structure with respect to:

- plastic deformation
- buckling
- fatigue failure
- crack propagation

Statistical wave load analysis in accordance with 2.4 (dynamic loads due to ship motions), finite element analyses or similar methods and fracture mechanics analyses or equivalent approach, are normally to be carried out."

Membrane Tanks

Structural analysis of semi-membrane tanks in accordance with the requirements for independent tanks, may be required. There are no requirements for a fracture analysis of membrane tanks.

A.2.17.9 Fracture Loads Summary

Independent Tanks

BV specifies only that a reduction of the secondary barrier may be allowed if a fracture analysis is carried out. DnV, GL, NK and IACS require that a load spectra for fracture be calculated for the worst 14 day period over the lifetime of the ship. In order to simplify the crack propagation estimates, DnV, NK and IACS will allow a straight line approximation to the load spectra. DnV will also allow the straight line approximation to be replaced by 5 fatigue loads (5 points on the load spectra curve). GL does not specify the shape of the load spectra curve.

The methodology of the fracture mechanics analysis for DnV, GL, NK and USCG is essentially the same:

1) Using a finite element analysis or other suitable technique, determine the maximum stress for the tank under consideration, with special emphasis on supports and attachments.

2) Assume a minimum crack size which will allow detection of a gas leak by a monitoring system.

3) Calculate the maximum size to which this crack will grow during exposure to the worst two weeks of the ship's lifetime.

4) Calculate a critical crack size using the properties of the material of the tank and the maximum stress specified in (1) above.

5) Confirm that the critical crack size (4) is much greater than the maximum crack length (3) obtained during exposure to 14 days of severe weather.

The USCG is most specific here; a design is acceptable if the ratio of critical crack size (4) to the maximum crack length (3) is better than 10. This kind of fracture analysis is required on DnV type AII tanks, and in some cases, type B tanks, all GL independent tanks, NK type B tanks, and USCG IST and SPT tanks. ABS and LR present no guidelines for the calculation of fracture loads on independent tanks.

Membrane Tanks

The DnV, GL and NK (semi-membrane, Type B) requirements for membrane tanks are the same as for independent tanks. All the other agencies provide no guidelines for fracture loads on membrane tanks.

Integral Tanks

No specific guidelines concerning fracture loads are presented by the various agencies.

NOTES

APPENDIX B

A GENERAL DISCUSSION AND EVALUATION OF THE METHODS FOR PREDICTING WAVE-INDUCED LOADS[11]

B.I INTRODUCTION

General

The problem of wave-induced loads on a ship at sea is that of determining successive conditions of dynamic equilibrium of forces and moments acting in and on an elastic body moving in the irregularly disturbed interface of two different media. This problem can be simplified by considering external loads only, on the underwater part of the ship, which is considered to be a rigid body in an ideal fluid. Motions and other ship responses in waves are regarded as linear functions of wave height, and both the irregular waves and the irregular responses can be considered as the sum of many sinusoidal functions. Hence, the analysis begins with the study of harmonic oscillations of a rigid body, moving at forward speed on the surface of an ideal fluid under the action of regular gravity waves.

Though in principle, the ship motion problem has been solved for three-dimensional cases[67, 68], the analytical solution is limited to forms such as a sphere or an ellipsoid. In view of this, a less rigorous strip theory solution has been developed which is suitable for long, slender bodies, where each cross-section of the ship is considered to be part of an infinitely long cylinder. Hence, a series of individual two-dimensional problems can be solved separately and then combined to give a solution for the ship as a whole. The idea was originally introduced by Korvin-Kroukovsky[69] and has since been endorsed, criticized and improved by many authors[10, 13, 70].

The main drawback of the strip theory is that it neglects the mutual interactions between the various cross-sections, which are of particular importance for certain frequency ranges, depending on the size of the body. Hence, in waves that are either very long or very short, relative to a ship, the theoretical justification of strip theory is somewhat questionable. This statement is particularly applicable to lateral motions, since the hydrostatic restoring force is small or non-existent under these circumstances.

In spite of the above reservations, the basic strip theory has been found to be satisfactory for motions, forces and moments[71], and it is the only suitable method for numerical computation. A major recent contribution to the theory has been the inclusion of all the forward speed terms in the equations of motion in order to satisfy the symmetry relationship proved by Timman and Newman[72]. All the modified strip theories developed in the past five years[11, 73] have practically identical forward speed terms.

[11] The original draft of this chapter was prepared by D. Hoffman, Webb Institute of Naval Architecture.

Since we are concerned with successive conditions of dynamic equilibrium, it should be noted that a complete solution of the problem of wave loads and bending moments cannot be obtained without first determining the motions.

State-of-the-Art Development

In order to evaluate the state of development of ship motion and load calculation in waves, a short analysis of the basic approach to the problem will first be given. The mathematical formulation of the problem, i.e., a ship advancing at constant mean speed with arbitrary heading into regular sinusoidal waves, can be presented in most general form by defining the velocity potential so as to satisfy the Laplace equation, as well as several boundary conditions, within the assumptions of the ideal fluid, linearized theory. At this initial stage, no strip theory assumption is required. The time-dependent part of the potential can be decomposed into three components representing the potentials due to incident wave, defraction and the mode of motion considered, as in the original theory by Korvin-Kroukovsky[69]. However, an additional time-dependent term due to steady forward motion of the ship has been added in more recent theories[10].

Once the formulation of the component potentials is completed, the hydrodynamic forces and moments acting on the hull can be determined. Using the Bernoulli equation, the pressures in the fluid are defined and expanded in a Taylor series about the undisturbed still-water position of the hull. Ignoring steady pressure terms, the linearized time-dependent pressure on the hull can be formulated and integrated over the hull surface. The hydrodynamic forces and moments can be obtained in two superimposable parts: those associated with a wave passing a restrained ship (excitation) and those acting on a body forced to oscillate in calm water.

In order to obtain a numerical solution, the application of strip theory approximations are necessary for the integration of the sectional exciting and motion-related forces over the length of the ship. These section forces involve two-dimensional added mass, damping and displacement terms. The speed-dependent coefficients are expressed in terms of a speed-independent variable, which is evaluated by means of a strip theory, and of a speed-dependent term which is obtained from a line integral along the waterline as given by Stoke's theorem. Hence, the main difference between Korvin-Kroukovsky's original strip theory and the more recent "new" methods is in the formulation of the problem. Previously, strip theory assumptions were applied in the initial formulation, and the forward speed effect was only introduced in certain terms. In the "new" theories the assumptions with regard to strip theory were made after the general terms for the coefficients in the equations of motion were determined, including the forward speed terms.

In addition to the above, Salvesen, et al.[10], include a term in the coefficients associated with the aftermost sections, which are not usually included in the strip theory and are claimed to be important for bluff bodies. These terms are independent of the strip theory assumptions. A comparison of results obtained for a container ship using the principles of the original strip theory and the modification for bluff bodies was presented by Floksta[71].

Using either the old or the new approach, the formulation of hydrodynamic forces and moments permits the equations of motion to be solved and the amplitudes and phase angles of motion determined. Then the longitudinal distribution of all forces--including those that are dependent on the motions and forward speed--can be evaluated and shearing forces and bending moments calculated, usually at midship, for any instant in the motion cycle. In general the solutions for two instants of time suffice to determine the amplitudes and phase angles of these quantities.

The extension of regular wave results to short-crested irregular seas, by means of the superposition principle, was accomplished by St. Denis and Pierson[74], on the assumption that both the irregular waves and the ship short-term processes are stationary stochastic processes.

Though the method of extending the calculations to irregular waves is universal, the techniques vary considerably depending on the wave input data used and the statistical model applied to the data for long-term predictions. In most cases, spectral representation of the wave is used though it can vary between mathematical formulation or actually measured data, single spectrum or a family of spectra, etc. Once the wave spectra is linearly superimposed on the specific response transfer function, at a constant speed and heading, that response is expressed statistically in terms of the root-mean-square of the process and its multiples representing the $1/n^{th}$ highest expected values. However, the extrapolation of the rms to extreme values as expressed by the $1/n^{th}$ highest is usually limited to return periods characterized as steady state conditions of the sea. Such periods are limited in time and cannot be extended beyond four hours or approximately 5000 reversals. A more reliable extrapolation to longer periods of time is therefore required and the use of order statics, or combined cumulative distribution is therefore called for. Such extrapolation can be applied to periods representing a storm, a year of operation or the lifetime of the vessel. A more detailed description of the statistical models is given in the following section.

Due to the undeterministic nature of the above conditions, the selection of a single design value to represent a specific response under operational conditions is not always easy. A typical way of presenting motion and load analysis is by referring to the level of response expected to be exceeded once in the lifetime of the ship or 10^8 reversals. Such a definition requires the use of assumptions with regard to the loading of the ship relative to the waves, the speed expected at each sea state, the specific route selected, etc. The response is also largely dependent on the basic design of the ship and the particular loading condition as expressed in terms of the metacentric height $\overline{GM}$, and the longitudinal and transverse weight distribution. All these variables have been discussed in Chapter IV with emphasis on the response in general and acceleration in particular.

Simplified Approximation of Ship Responses

Due to the lengthy and costly procedure usually associated with the determination of ship responses, several attempts to short-cut the procedure were formulated to enable the designer to estimate a design value during the preliminary design phase. Det norske Veritas (DnV) developed approximation formulas for maximum acceleration[30, 31] at a probability level of 10^{-8}. These were based on the results of strip theory calculations assuming rigid body dynamics and on full-scale measurements. Linear accelerations in longitudinal, transverse and vertical directions at any point on the ship are given in a non-dimensional form as a function of the ship's length, breadth, block coefficients and metacentric height. The results are given in terms of the acceleration due to pure heave motion which is determined as functions of the forward speed and the ship length. Other approximations similar in nature have since been proposed by other Classification Societies and Regulating Bodies. It will be shown in the following sections (also refer to Section IV in the body of the report) that accelerations are a function of much larger number of variables than those used in the approximate formula and therefore the results obtained by such simplified calculations can be close in some cases and out by an order of magnitude in other cases.

Other responses, such as bending moments due to waves have traditionally been approximated by static methods such as by the superposition of a stationary wave of specific height and period on the ship. The ship is balanced on the wave to give the correct trim and heel, and the shear force and bending moment distribution along the hull is obtained from consideration of weight and buoyancy. This particular approach proceeded the dynamic method described previously and is known to yield rather conservative answers.

A more generalized method for approximating many of the responses is by means of interpolating experimental or theoretical data which is classified and stored in the computer to obtain results for a similar ship configuration under similar operating conditions. Such results can be used as first approximation and are usually of the same order of reliability as the acceleration formula discussed above.

Other approximation techniques often used in preliminary stages include the extrapolation of regular wave tank response data to long-term statistical predictions. Several approaches, all representing gross approximation, can be used. These are usually based on the response value at resonance. The recommended approach to determine responses excited due to waves is by means of computers. Data required for such calculations will be discussed in the following section.

Input-Output of Ship Response Calculations

Two basic input requirements are called for in determining the response of a vessel to irregular waves. One is the wave data, which includes the information necessary to define the sea spectra, covering all possible operational conditions and the description of the expected route or routes for which the responses are to be evaluated, and the other is the ship design information, which includes the geometry of the outer hull contour, the longitudinal, transverse and vertical weight distributions and the initial stability data. Specific design parameters, such as the displacement, center of gravity, etc., can be evaluated from the above input data. Additional input includes the specification of the response required and the speeds and headings at which it is to be calculated; a viscous damping correction factor, to account for effects not counted for by potential theory; and the specific frequencies at which the response transfer function is to be evaluated. While the wave data is required for the indeterministic portion of the calculation, i.e., the statistical response, the ship design data represent that required for generating the transfer function.

The input format may vary somewhat from one program to another. The ship geometry may be represented in terms of the ship offsets, mapping coefficients or the beam-to-draft ratio and the sectional area coefficients. The wave data may be given in a form of groups of wave height and period combination and their frequency of occurrence or the spectral ordinate of actually measured wave data representing a wide array of conditions can be used. Figure B.I illustrates the system configuration of ship motion calculation procedure indicating the input and the output alternative. Further breakdown of some of these components will be given in the following sections along with the discussion of the specific techniques which can be used to advance from one block to the next.

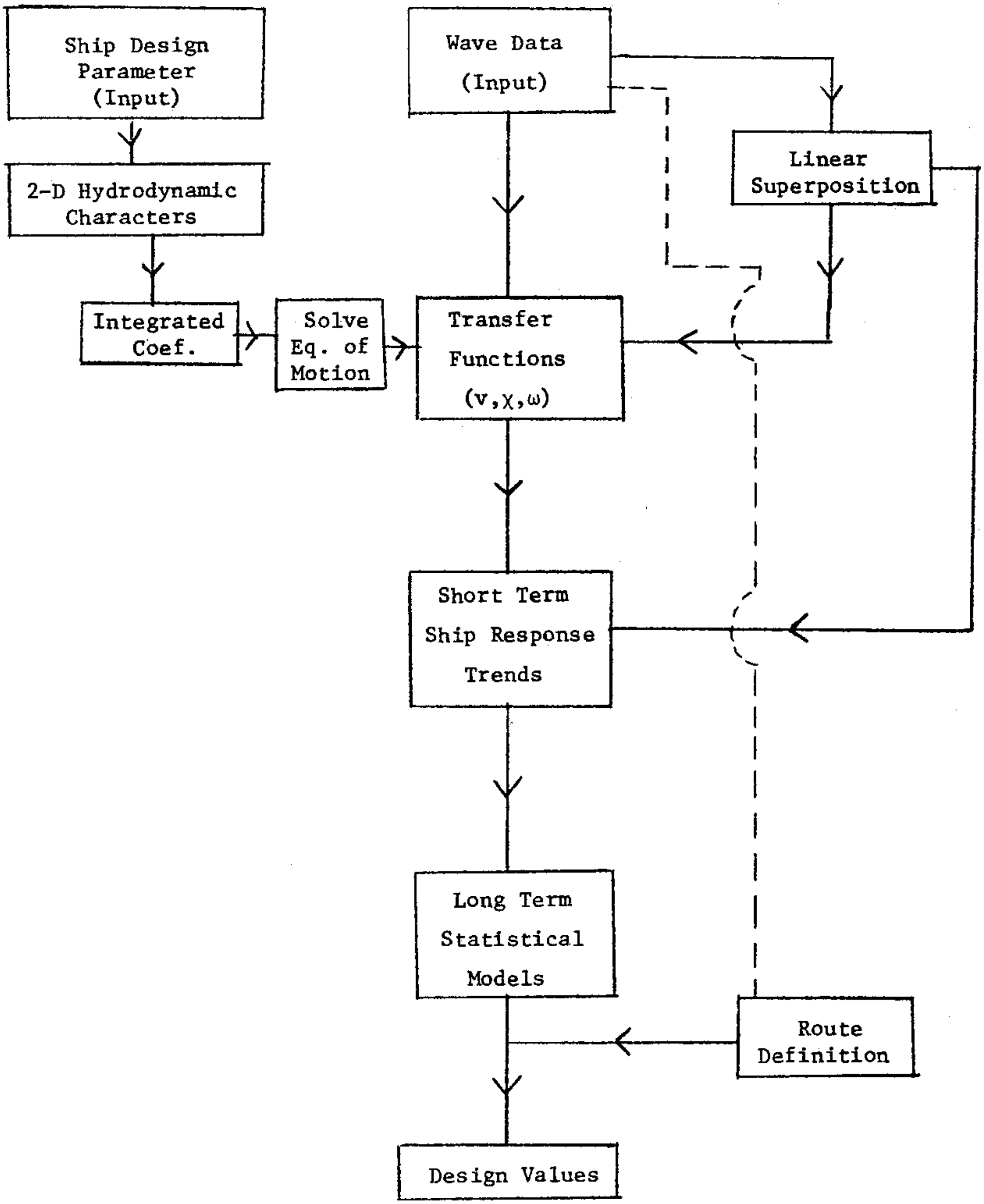


FIGURE B-1. SHIP MOTION CALCULATIONS SYSTEM DIAGRAM

General

The basic functional blocks of the ship motion calculation system were given in Figure B-I. Each of these components will now be discussed in detail emphasizing alternative techniques presently available and their relative merits. Though the following discussion deals in the general subject of ship motions evaluation, the latter are the cause of many other responses such as acceleration stresses, shear forces, etc. Due to the linear assumptions made with regard to the ship motions, each of these responses can also be treated under the same assumptions and if the basic transfer functions of the response are given per unit wave height, and the nature of the excitation is known, the response to the excitation can also be defined. Hence, the mathematical tools such as the linear superposition principle or the statistical models are applicable to any of the linear ship responses.

The purpose of this note is to investigate the methods used to calculate the magnitudes of the maximum acceleration expected over the lifetime of the ship in the longitudinal, transverse and vertical directions and hence, determine the maximum acceleration in any arbitrary direction. The acceleration in a specific direction can be defined if the motion components in this direction and their phase angles are given. Hence, the vertical acceleration is a function of pure heave and the appropriate pitch and roll components, the horizontal acceleration is a function of pure sway and the yaw and roll components and the longitudinal acceleration is a function of pure surge and the pitch and yaw components. In addition to these, the force of gravity will always be acting and the "static" acceleration component should be added in all three cases, i.e., g in the case of vertical, $g \sin \phi$ in the case of horizontal where ϕ is the roll angle and $g \sin \theta$ in the case of the longitudinal acceleration where θ is the pitch angle.

The maximum acceleration in an arbitrary direction can be obtained based on the assumption that a_z, a_y and a_x, the vertical, horizontal and longitudinal accelerations are statistically independent variables. Hence, the maximum acceleration at each plane can be easily determined as $\sqrt{a_x^2 + a_y^2}$ or $\sqrt{a_y^2 + a_z^2}$, etc., where a_z includes the static component corrected for the instantaneous pitch and roll angles. It should be remembered that the acceleration in the longitudinal direction, due to pure surge cannot be directly obtained from the solution of the equations of motion due to the strip theory concept which requires the integration of the hydrodynamic coefficients along the longitudinal axes. The components due to pitch and yaw are available and usually an approximation of the acceleration due to surge in terms of a percentage of the pure heave acceleration is made.

Ship Design Parameters Inputs

The ship design parameters, which include primarily the buoyancy and weight distribution data along the hull comprise the major input for determining the motion transfer functions. Basic flotation principles require the longitudinal centers of gravity and buoyancy to lie along the same vertical line which in turn must be at 90° to the waterline for the ship to be in equilibrium. In order to balance the ship, the buoyancy curve, as defined by the immersed volume and the weight curve as defined for the specific loading condition must be first defined.

The most basic way of describing the hull is in terms of its offsets at convenient transverse locations. The integration of the individual areas will yield the sectional characteristics of each transverse location such as area, center of area, etc., and hence the volume, its center and other geometric characteristics of the ship, can be easily defined. The weight curve is usually defined in terms of blocks of weight described by the forward and aft ordinates and the distance between them. Balancing the ship yields the waterline coordinates fore and aft and hence the draft at each transverse section which is the required input for the calculation of the two-dimensional hydrodynamic characteristics of each oscillating section. Most of the input data can be generated from the buoyancy and weight curves. Ideally, the vertical and lateral weight distribution are also required, however, in reality these are represented by center of gravity and the transverse radius of gyration at each transverse section.

Most programs yield the motion at the center of gravity of the ship and very often acceleration at the bow and stern will automatically be given. If however, accelerations are required at other locations the space coordinates relative to the center of gravity must be given. Other inputs include various options of the program, such as the specific speeds at which the motions are to be evaluated, the relative heading angles, the specific frequencies at which the transfer function is to be defined, etc. The ability of the specific ship to resist roll due to viscous damping is also considered as an input parameter, very often empirically applied.

The above constitute the total input required to determine the response transfer function. The first phase of the calculation, described under the following heading, required only the geometrical definition of each section up to the design waterline.

Two-Dimensional Hydrodynamic Calculations

As indicated previously, due to the complexity of the three-dimensional shape of the ship, the calculations of the hydrodynamic characteristics due to vertical, lateral and angular oscillation about the free surface is not feasible and the characteristics of infinitely long cylinders having a cross-section identical to that of the ship's transverse section is calculated instead. Each cylinder is defined in terms of its cross-sectional area by means of offsets between the keel and the waterline intersection. In order to determine the added mass and damping due to oscillation on the free surface, two techniques are available: conformal mapping and source sink distribution. The one-to-one correspondence between the points on two distinct complex planes expressed by a single analytical function is the basis for conformal mapping. The method involves the mathematical manipulation of the boundary geometry from the plane in which the potential solution is desired to one where it is known, such as a semi-circle. The potential is constructed of a source potential and a sum of multiple potentials both placed in the origin and each satisfying the Laplace equation, the free surface conditions, the finite depth and the radiation boundary condition. The added mass and damping can hence be evaluated as well as the pressure on the hull contour which can be obtained from the linearized form of the Bernoulli equation. An alternative to mapping is the distribution of source singularities, and the method by Frank[75] was found suitable for most sectional shapes. Pulsating source singularities of constant strength are placed on each straight line segment connecting two offset points on the contour. In both cases, the results are identical if the section geometry was properly defined. In the latter case, this depends on the number of offset points used to define the section while in the case of mapping, the number of coefficients is often a factor in setting the accuracy of the procedure. A specific case of conformal mapping, involving two coefficients only, is commonly referred to as the "Lewis" form[76], and it usually represents a fair description of most transverse ship sections. However, the description of more complex bow or stern sections requires the addition of more coefficients and is usually referred to as close-fit[77].

Both multi-mapping coefficients and the Frank source sink methods fail to perform for certain types of sections such as a bulb in the former case or a shallow draft flat bottom section in the latter case. Ideally, it is therefore handy to have all three options, i.e., the simplified two-parameter mapping, the multi-parameter mapping and the source sink distribution available, to be applied to the appropriate sections in accordance with their complexity. The simplified "Lewis" form solution is much faster than the other two alternatives and should be used whenever applicable.

Comparative studies to evaluate the three methods have indicated that for a "Lewis" form all three methods yield practically identical results. Yet for non-"Lewis" form sections, substantial differences in the added mass and damping coefficients can sometimes be demonstrated. It will be noted in the following sections that the overall effect on the ship motion responses is negligible in most cases.

The hydrodynamic coefficients are all expressed in terms of the frequency of oscillation and are stored in an array to avoid repetition of these lengthy calculations. A total of eight coefficients is usually generated including the added mass or mass moment of inertia in pure heave, sway and roll respectively, the damping for the above three cases, and the cross-coupling terms of added mass and damping between sway and roll and roll and sway which are usually considered identical.

Hence, for each station an array of approximately 25 X 8 depending on the number of frequencies is generated and stored for the specific loading conditions. As indicated before, the two-dimensional pressure distribution and hence, the forces acting on the oscillating cylinder can also be defined in terms of their amplitude and phases.

The Equations of Motion

As most ships have lateral symmetry, separation of the generalized six linear coupled differential equations of motions as shown below is justified:

$$\sum_{k=1}^{6} \left| (M_{jk} + A_{jk})\, \ddot{\eta}_k + B_{jk}\, \dot{\eta}_k + C_{jk}\, \eta_k \right| = F_j\, e^{i\omega t} \qquad j = 1, \ldots, 6 \tag{17}$$

where

M_{jk} = generalized mass

A_{jk} and B_{jk} = hydrodynamic added mass and damping coefficients

C_{jk} = hydrostatic coefficients

F_j = amplitude of exciting forces and moments where $j = 1, \ldots, 6$ refer to surge, sway, heave, roll, pitch, yaw respectively.

ω = frequency of encounter

η_k, $\dot{\eta}_k$, $\ddot{\eta}_k$ = displacement, velocity and acceleration.

Integration of the sectional hydrodynamic coefficients along the hull yields the coefficients A_{jk} and B_{jk} and the exciting forces. This is done under the assumption of strip theory assuming long slender hull forms, ignoring interaction between the sections and ignoring surge motion.

Due to symmetry considerations, the remaining five equations can usually be divided into two groups: longitudinal motions, i.e., heave and pitch; and lateral motions, i.e., roll, yaw and sway. This substantially reduces the number of cross-coupling terms in these equations and simplifies the calculations.

Once the hydrodynamic coefficients are inserted in the equations for the specific frequency of encounter, as reflected through the encountered wave length, ship speed and heading, the frequency independent hydrostatic coefficients are calculated and substituted in the equations of motion. The procedure is repeated for several frequencies; in each case the equations are solved to give the motions.

Slight variations in the definition of the coefficients of the equations of motion, as well as in the excitation forces and moments, exists among the several available programs. Though the actual values of each coefficient may vary substantially in some cases, the resulting motion is generally the same.

Transfer Functions

The resulting distribution of the specific response as a function of frequency for a unit wave height excitation is usually referred to as the transfer function. Alternatively, it is often given a non-dimensional form divided by the wave height or slope as the case may be, before squaring the values. These squared non-dimensional responses on the basis of frequency are referred to as Response Amplitude Operators (RAO). The plot of the transfer function is of limited value as it does not reflect magnitude for design purposes; however, the resonance frequency is of great interest and the value of the RAO at that frequency can be used for a very rough approximation of possible magnitudes of motions, assuming a regular wave of a certain height having a frequency identical to or close to the resonant conditions. A sample transfer function for bow vertical accelerations, obtained for a model of a 125,000 m^3 LNG ship,[78]is shown in Figure B-2. As stated above, the transfer function cannot predict the maximum acceleration for design purposes, but can predict the resonant frequency. For this particular ship encounters with waves of frequency $\omega = 0.425$ (period = 14.8 sec) may in all probability result in severe accelerations. This leads to another useful feature of the RAO curve which is in determining the probability of encountering resonance at specific sea zones characterized by a certain mean period. It is evident that if the heave resonance of a system occurs at a period of 19 seconds, the probability of such an occurrence in most world oceans is very small because the wave required to excite the resonance would be unusually long (550 meters). Likewise, a roll resonance at a period of 8.5 seconds may mean some rather large and frequently occurring roll angles in most open water oceans. The transfer function definition is therefore important in general terms of resonant frequency definition and order of magnitude of the responses; the question arises, however, as to what degree of accuracy should be pursued.

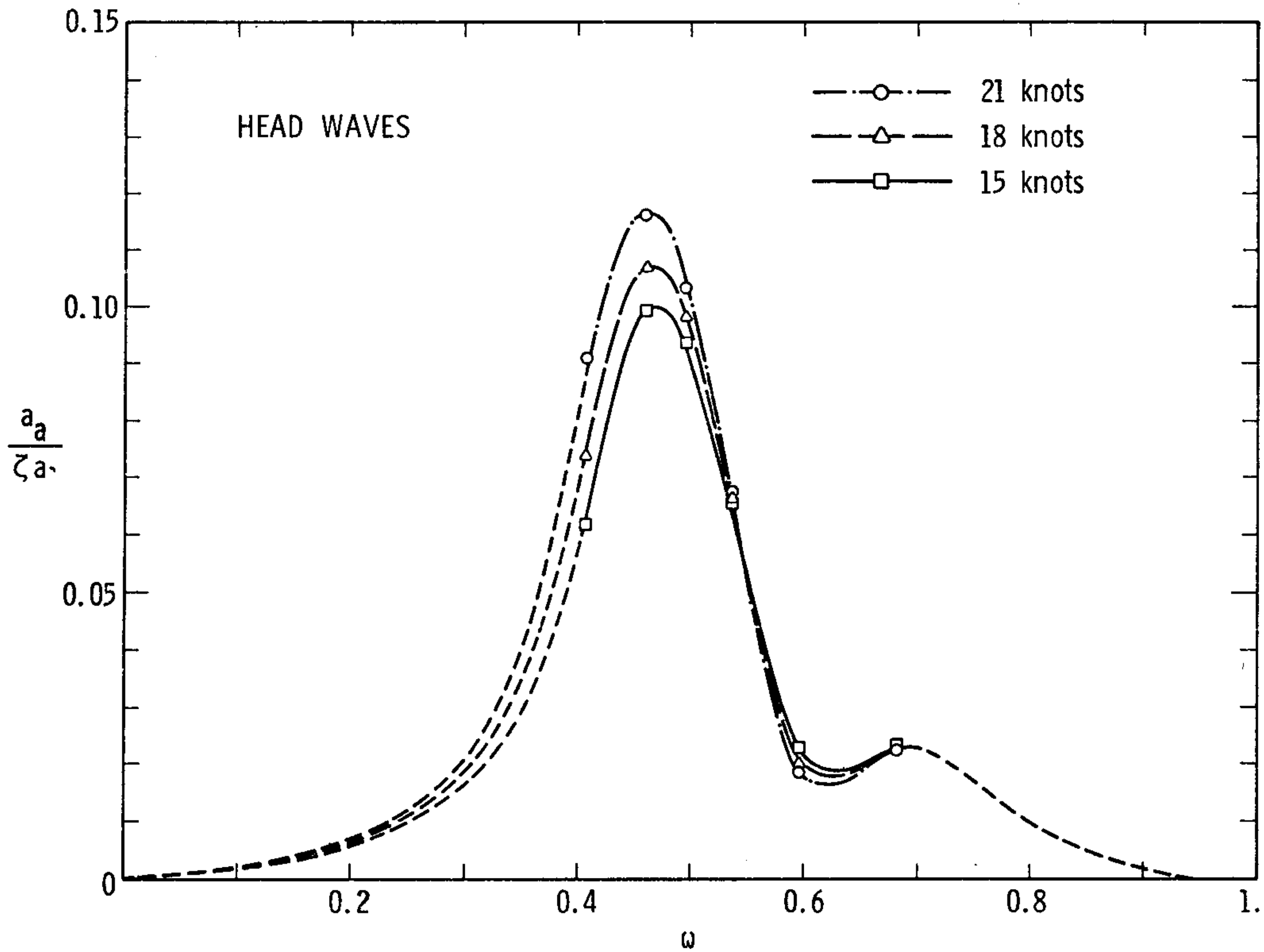


FIGURE B-2. MEASURED ACCELERATIONS a_a AT THE FORWARD PERPENDICULAR OF A MODEL 125,000 m^3 LNG SHIP BEING TOWED IN REGULAR HEAD SEAS OF AMPLITUDE ζ_a (Ref. 78)

For the past fifteen years, large emphasis was placed on improving the accuracy of the transfer function through better definition of the hydrodynamic coefficients, through more rational mathematical approach to solution of the problem, and through refined forward speed effects, blunt ends and bulb effects, close-fit techniques, etc. The main reason for the pursuit of perfection was primarily the fact that the original theory as presented by Korvin-Kroukovsky was not always mathematically rational. However, the various modifications haven't necessarily changed the results significantly and furthermore, the search for design values have shifted the emphasis to the statistical solution for determining the motions under more realistic sea environment. Hoffman[79] recently showed that in many cases, the type of wave input data has a more significant effect on the rms response distribution and on the long-term responses than the variations in the transfer function due to various theories and slight variations in the assumptions. It is also evident that the statistical distributions used for the long-term predictions are much less sensitive to small variations in the transfer function shape and magnitude. In order to illustrate the above, some discussion of wave data inputs and statistical methods of extrapolation to longer return periods will be covered in the next section.

Wave Input Information

Two basic issues are in question when discussing wave data applicable to ship motion calculations: (1) availability of data, and (2) data formatting for practical use. The availability of wave data, suitable for direct or indirect application to ship response calculation, was discussed by Hoffman[80]. This work covered the three major sources, i.e., observation, measurements, and forecasting-hindcasting techniques. The following discussion will address itself primarily to the type of wave data ideally required for load analysis under realistic sea conditions and the alternatives which are available due to limited availability and lack of a generalized description of all seas.

The method formulated by St. Denis-Pierson[74] to obtain the response of a ship to waves utilizes the wave spectrum, which can be expressed mathematically if the basic statistics of the sea are available, or even better, which can be based on measured data reduced to spectral form. The mathematical spectral formulation which has been widely accepted for ship motion analysis is of the Bretschneider[81] type, but unfortunately it only adequately simulates fully developed sea conditions. Though it may represent a more severe sea condition, it does not necessarily excite the most severe response of the system. For a complete analysis, the response of the system to all possible sea conditions is of prime importance and hence extensive measured data in spectral form, or a generalized mathematical spectral formulation, must be available. The influence of various types of wave data formatting on the predicted loads is currently being studied[82] and preliminary results indicate that such effects will vary from one ship to another and most likely will be a function of the type of response in question such as acceleration, bending moment, etc. Though alternatives to spectral formulation are sometimes used, such as an equivalent wave height or a simplified wave system consisting of 3-4 components, it is generally agreed that for ship calculations, regardless of the type and size, the wave formatting required must be in a spectral form. Hence, the major differences between the various techniques can be reduced to different input data used to generate the mathematical spectra or the source of measured data used. Other differences are associated with the steps taken in the process of linear superposition and will be discussed in the following section.

The most general definition of the sea is in terms of the "Sea State". From tables such as given by Hoffman and Marks[83], it is replaced by a mean wind speed which in turn is expressed in terms of a mean significant wave height ($H_{1/3}$). The latter is the single input parameter in a Pierson-Moskowitz type sea spectrum which defines the prescribed sea state. A modification over the above is often used by input of the mean zero crossing period T_2 as well as $H_{1/3}$. The two values are substituted in a more generalized spectral formulation, sometimes referred to as the modified Pierson-Moskowitz spectra. This however, does not substantially change the result due to the fact that most Sea State Charts display a constant relationship between the mean height and period of the following type:

$$T_2 = 1.96 \ H_{1/3}^{\frac{1}{2}} \qquad (18)$$

By substituting the above relationship in the two parameter Pierson-Moskowsitz modified spectrum, the one parameter spectrum results. It is therefore the least practical to use unless a more realistic relationship between the height and period exists. This can be achieved by substituting observed height and period in pairs which are generally available for certain sea areas in a tabulated form. By substituting several pairs of $H_{1/3}$ and T_2 values, several spectra are generated each representing a possible condition. By weighing each spectrum by its frequency of expected occurrence, a mean spectrum can be obtained. If measured values of $H_{1/3}$ and T_2 for the area in question are available, the degree of reliability of the spectra is substantially increased.

Ideally measured spectra, representing a wide range of heights and periods and represented by the spectral ordinate, should be used due to the absence of a satisfactory mathematical formulation capable of representing conditions of cross seas or non-fully developed seas. Files of wave data representing typical ocean areas such as Station "India" in the northeast Atlantic, south of Iceland or Station "Papa" in the northwest Pacific at the entrance to the Gulf of Alaska can be used to represent realistic sea conditions as an interim solution. Such full-scale data are usually arranged in groups of wave height or groups of period covering the entire range. Each group consists of several spectra in an adequate number to represent the possible scatter about the mean. A typical example is given by Hoffman[79] for Station "India".

Wave files of the above-described nature are necessary in order to obtain long-term predictions covering a period of a storm of 20-30 hours duration or covering the lifetime of the ship. For shorter periods characterized by 4-8 hours representing conditions within one weather group, such as defined by a range of wave height or period, a more limited wave file can be used. However, it should always consist of at least eight spectra within a group in order to obtain a realistic mean and standard deviation. The unique definition of the mathematical spectra can be overcome by generating several spectra representing conditions close to the specific case, hence allowing for possible scatter about the mean. This approach, however, is only a partial remedy due to the fact that the mathematical spectrum, when plotted on a non-dimensional basis, collapses into a single line, whereas measured spectra varies substantially about the mean line. Hence, the introduction of scatter about the mean may somewhat improve the response calculation but would still be limited by the nature of the mathematical formulation.

Statistical Extrapolation

The superposition of each spectrum on the transfer function yields a response spectrum usually characterized by its root-mean-square (rms) value. When several spectra, each representing the same basic environmental condition, are superimposed on the transfer function, the response can be described in terms of the mean and the standard deviation about it. If the procedure is repeated to represent a wide range of environmental conditions, such as a range of wave heights from 0-15 meters, the trend of the particular response, as a function of wave height, is obtained. The distribution of the rms and its standard deviation as a function of the sea state is a useful intermediate step in the design procedure. Such trends can be generated as a function of the forward speed indicating the effect of the latter on the responses, or they can be generated as a function of heading, load distribution, etc. It constitutes a very useful operational envelope for the man on the bridge and helps set operational criteria.

Short-term trends, as described above, are all limited to the mean rms or the mean plus or minus the standard deviation. Hence, it does not yield extreme values nor does it indicate the level of response expected over a long period of operation. The extrapolation into long-term requires an additional statistical model. Two basic approaches are usually considered in predicting extreme ship responses to ocean waves. The first is to use a mathematical model covering the ship response to all sea conditions and hence to obtain a cumulative distribution of all responses. The value to be exceeded once in the lifetime of the ship or a fleet of ships can thus be determined. The other approach is to deal only with the extreme values of response which are presumably associated with the most severe wave conditions.

The first approach has the advantage of taking into consideration voluntary slow-down by the Captain under heavy weather conditions and hence can be used to predict the computative highest expected response over all possible operational conditions. It requires, however, large amounts of data over the entire weather range.

The second method does not require the low value data; however, the definition of the extreme response is not always so easily detectable, as the most extreme wave does not necessarily produce the worst response. Furthermore, the extrapolation to longer periods of time, using order statistics to describe the distribution of the extreme, may be ideal for a storm duration but may not be ideal over the lifetime of the ship.

The first method can also vary in the specific statistical distribution chosen for extrapolation. The Normal, Weibull, Log Normal and Exponential distribution have been used by various investigators. It has recently been shown[57] that the first two distributions, as can be seen in Figure VI-1, seem to give the best approximations for full-scale measurements while the latter two overestimate and underestimate, respectively. The Normal distribution method[57] is usually applied to individual weather groups, and the data are then integrated to take account of the expected weighing of each group. The Weibull distribution method[84] is usually applied to the total data regardless of the weather distribution.

In all the above cases, the final product is a cumulative distribution showing the expected increase in response level as a function of the return period or the number of reversals. The long-term curve is usually given for combined effects of all speeds and average heading distribution into the waves. The result is single design value representing the specific response under a certain loading condition. The meaning of these design values will now be discussed.

Design Values

The long-term cumulative distribution is usually reduced to a single design value by reading the expected response level at a probability of 10^{-8}. This probability level is equivalent to 10^8 reversals, which represents a typical lifetime of the ship. Hence, if the maximum expected vertical bending moment over the lifetime of the ship is required, the value can be read directly from a curve which was derived on the basis of certain assumptions with regard to the relative heading between the ship and the wave, ship speed, specific route chosen, etc.

Bending moment response is known to be practically independent of forward speed. Acceleration, however, could vary substantially with forward speed, and if a single design value is required to represent the highest expected over the lifetime of the vessel, some consideration must be given to the fact that, under adverse weather conditions, the ship usually slows down either due to added resistance or to a voluntary reduction in speed by the Captain of the ship. Hence, the short-term trends must consist of high speed curves at the low sea states and reduced speed at the higher sea states. The long-term prediction will therefore represent a more realistic extreme value.

One of the problems facing the designer is how to combine two or more long-term responses such as vertical and horizontal bending moments which can occur simultaneously or vertical and lateral accelerations which can also occur together. It is evident that the maximum vertical acceleration and the maximum horizontal acceleration would not occur under the same conditions, yet an increase in the maximum vertical acceleration due to a horizontal component is likely to occur, even though the horizontal component is less than the maximum value.

Based on the statistical law that the variance of a sum of independent variables is equal to the sum of the variances, the square root law can be applied as shown in Section IV. For variables which are not completely independent, such as vertical and lateral acceleration in oblique seas, the coefficient of correlation $\rho_{1,2}$ between the accelerations must be determined and the combined acceleration can be written as follows:

$$a_{z+y} = (a_z^2 + a_y^2 + 2\,\rho_{1,2}\, a_z a_y)^{\frac{1}{2}} \tag{19}$$

Correlation coefficients are best obtained from full-scale measurements of a_{z+y}, a_z and a_y using the above equation to determine $\rho_{1,2}$.

A different problem occurs when the gravitational acceleration component must be added to the vertical wave-induced acceleration so that the instantaneous trim and heel of the ship at the instant of maximum acceleration must be known. Since no direct phase relationship between the various long-term responses is available, the method of equivalent regular waves is sometimes used.

$$\text{Equivalent Wave Height} = \frac{\text{Max. Long-Term Acc.}}{\text{Max. Acc. due to unit wave height}} \tag{20}$$

While the long-term acceleration is a single value at each location on the ship, the maximum acceleration due to unit wave height may have several close values depending on the specific heading. While the actual maximum amplitude may be close for several headings, the phase angles may not necessarily be so close. To determine the instant at which the acceleration is maximum, the following steps are taken:

$$a = a_o \sin(\omega_e t + \epsilon_a) \tag{21}$$

when

a_o = acceleration amplitude

ϵ_a = acceleration phase angle at maximum acceleration

ω_e = encounter frequency at maximum acceleration.

For $a = a_o$,

$$\sin(\omega_e t + \epsilon_a) = 1$$

i.e., $\omega_e t + \epsilon = 90$ or 270.

Hence, $\omega_e t = 270 - \epsilon_a$ or $90 - \epsilon_a$

If $\omega_e t$ is known, both the pitch and roll amplitudes can be determined provided the phase at that ω_e value is available, i.e.,

$$\begin{aligned} \phi &= \phi_o \sin(\omega_e t + \epsilon_\phi) \\ \theta &= \theta_o \sin(\omega_e t + \epsilon_\theta) \end{aligned} \tag{22}$$

where ϕ_o and θ_o represent the roll and pitch amplitudes at that ω_e value and ϵ_ϕ and ϵ_θ are the phase angles respectively.

This method of approximating the instantaneous roll and pitch at the time of maximum acceleration is obviously very vague and has not been substantiated. However, if for each case the procedure is repeated for the maximum acceleration at several different headings, a good feeling for the sensitivity of the roll and pitch angles can be obtained and the order of magnitude can be determined. Hence, the g component can be adjusted for the roll and pitch angles as follows:

$$\begin{aligned} a_z &= a_z \pm g \cos\phi \cos\theta \\ a_y &= a_y \pm g \sin\phi \end{aligned} \tag{23}$$

A more exact solution to the above problems can only be obtained from a time domain model; however, a very large record will be required to predict long-term trends.

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A study of LNG tank loading criteria is presented that includes a survey and review of load criteria presently employed in the design of cargo tanks for LNG carriers. Motion and acceleration values as determined from these criteria are compared to ship motion calculations and available full-scale data. A comparison of LNG tank loads, as predicted by current classification society and regulatory agency criteria, is given along with recommended updated criteria in each of seventeen load categories. Model tests and full-scale measurement programs to provide adequate data for verification of load and acceleration criteria are also outlined.

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Advisory Group I, "Ship Response and Load Criteria", prepared the project prospectus and evaluated the proposals for this project:

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PROF. J. L. BOGDANOFF, *School of Aeronautics & Astronautics, Purdue University*
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