SSC-308

CRITERIA FOR HULL–MACHINERY RIGIDITY COMPATIBILITY

SSC-308 CRITERIA FOR HULL-MACHINERY RIGIDITY COMPATIBILITY 1981

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1981

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An Interagency Advisory Committee
Dedicated to Improving the Structure of Ships

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SR-1266

1981

The main propulsion machinery and shafting aboard ships has always required foundations which will limit movement. As hulls have become more flexible and horsepower has increased, the need for rational foundation design which will link these flexible hulls with the more rigid machinery and shafting has increased.

The Ship Structure Committee undertook this effort to study the criteria which the designer may use to adequately address the problems of meeting distortion limits imposed by machinery manufacturers due to bearing loading, misalignment, gear tooth wear, and excessive vibration.

This report presents a proposed methodology for dealing with these problems and gives an example application.

Clyde T. Lusk, Jr.
Rear Admiral, U.S. Coast Guard
Chairman, Ship Structure Committee

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16. Abstract Recent trends in increased ship hull flexibility, particularly in large ships, have given urgency to a host of problems which were not encountered before in naval architecture. This report deals with one of these problems, specifically the compatibility between local hull deflections and distortion limits imposed by the operational requirements of the main propulsion machinery components. The need to conduct this study was felt because very often problems of shaft misalignment, gear wear, excessive vibration and others, were found to be most probably a result of insufficient stiffness in machinery support systems. The evaluation of the relationship between manufacturer's requirements and the structural design of machinery foundations is the goal of this research effort.

The overall objective of this paper is to derive a set of recommendations capable of helping the designer meet the requirements of foundation stiffness and which are necessary for the good performance of machinery components. The design recommendations to be derived essentially concern the structural arrangement of machinery spaces and support systems. Also included are a group of suggested methods and techniques of structural analysis and design which can assist the designer in implementing these recommendations. As a result, it is hoped that the gap between hull flexibility requirements and machinery operational requirements for a ship can be reduced, so that overall design process and the ship's performance can be improved.

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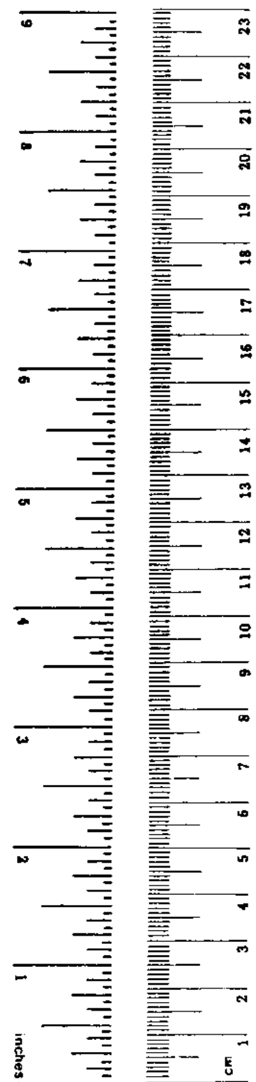
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

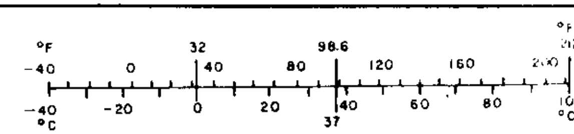
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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INTRODUCTION

Recent trends in increased ship hull flexibility, particularly in large ships, have given urgency to a host of problems which were not encountered before in naval architecture [1]*. This study deals with one of these problems, specifically, the compatibility between local hull deflections and distortion limits imposed by the operational requirements of the main propulsion machinery components. The need to conduct this study was felt because very often problems of shaft misalignment, gear wear, excessive vibration and others, were found to be most probably a result of insufficient stiffness in machinery support systems [2-4], and because of insufficient knowledge of shipboard environment and flexibility by machinery manufacturers. (Ship machinery is usually designed by assuming a concrete foundation). These reasons show clearly the relevance of evaluating in a comprehensive way the relationship between manufacturer's requirements and the structural design of machinery foundations.

In view of unfortunate past experience, manufacturers now attempt to scrutinize carefully the environment in which their equipment must function. In the past, this could be done by experience and by comparison with similar designs. While this procedure worked for many years, it became somewhat inadequate as vessel size grew and economic pressures increased to minimize hull weight and cost. Today more sophisticated methods can be used by the designer to determine structural response. The proposed solution, therefore, requires: (a) the machinery designer to specify reasonable limits within which his equipment can function properly, an area in which as this study indicates a good degree of agreement has already been reached by main propulsion machinery manufacturers in this country, and (b) the hull structural designer to determine that a support system will meet these limits under all normal operating conditions.

In the case of ships built in the U.S., hull-machinery compatibility problems such as those mentioned above have been found to be relevant in large geared-turbine powered ships with units in the size range from approximately 25,000 SHP to 50,000 SHP. In fact, most of the design experience in this country in the case of large ships has traditionally been concerned with turbine-powered vessels. On the other hand, in Europe and Japan, diesel engines have often been used for the propulsion of large ships, and, in Europe, studies on hull-machinery compatibility have also been conducted on diesel-powered ships [5-7]. Because of the current world energy crisis, a growing

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* Square brackets designate references listed before the Appendices.

interest in diesel propulsion is now being felt, and this trend is expected to affect the shipbuilding industry in this country [8-9]. For this reason, the study conducted here also addresses the hull-machinery compatibility problem as it relates to diesel-powered ships. However, the main thrust of this research is concerned with turbine-powered ships. The conclusions and proposed design method can apply to steam as well as gas turbines.

This research program was subdivided for convenience into four main tasks, which followed an extensive computer-aided literature search using the NASIC* Search Service available through the M.I.T. Libraries.

The first task included a survey of major U.S. and foreign machinery manufacturers in order to determine their requirements for rigidity of the main engine supports. Based on this information, a set of general requirements defining maximum foundation deflections, and representing what was felt to be an acceptable industry-wide practice have been defined.

The second task consisted of a review of the design of main engine, gear and thrust-bearing support structures of selected ships, in order to define as much as possible current design practices. This included a study of overall arrangement and scantlings of main support members of machinery, reduction gears, thrust bearing, shaft bearings, and also the dimensions and arrangement of shafting.

The third task was essentially a critical review of available analytical and numerical procedures for studying the coupled response of hull and machinery. Based on this review, it was possible to identify the methods of structural analysis best suited for the study of hull-machinery-compatibility related problems.

Finally, the fourth and last major task was aimed at identifying criteria for defining the structural rigidity of machinery-support systems. This includes recommendations concerning the structural design of these support systems, so that machinery requirements are met, and the possibility of failures due to excessive flexibility is minimized.

The overall objective of this project is to derive a set of recommendations capable of helping the designer meet the requirements on foundation stiffness necessary for the

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* Northeast Academic Science Information Center. The following data bases were accessed by the searchers: MRIS (Maritime Research Information Service) and COMPENDIX (Engineering Index).

good performance of machinery components. The design recommendations to be derived essentially concern the structural arrangement of machinery-support systems and spaces. Also included are a group of suggested methods and techniques of structural analysis and design which can assist the designer in implementing these recommendations. As a result, it is hoped that the gap between strength requirements and machinery operational requirements for a ship can be reduced, so that the overall design process and the ship's performance can be improved.

It can be concluded from the brief overview given above that this project, due to its practical implications involved a considerable information-gathering effort. It included, in addition to the extensive literature survey mentioned earlier, exchange of information with Classification Societies, engine manufacturers, shipyards and shipowners, not only in the U.S. but also abroad. A total of twenty-eight shipyards (twelve in this country, three in Canada, six in Europe and eight in Japan), and nine shipowners (six in the U.S. and three abroad) were contacted. Information was received for twenty-three ships, including fourteen tankers, three LNG carriers, three bulk carriers, one roll-on/roll-off, one container ship and one LASH. The wide cooperation received in the information-gathering effort was an important factor for the successful completion of the proposed work, and the authors are grateful to all those who contributed to this effort.

This report is organized in the following way: Chapter I contains a discussion on the hull/machinery rigidity compatibility problem, including some comments on the causes and effects of excessive hull flexibility, a brief description of the factors which can have a stronger influence on the problem under consideration here, and a review of the various solutions offered in the literature. A case study also is presented, involving a LASH vessel for which considerable data were available.

Chapter II deals with the problem of foundation design. The most relevant structural design parameters are identified, a review of current practice is summarized and some design recommendations are given.

Chapter III presents the result of the survey of machinery manufacturers.

Chapter IV describes a design method proposed by the authors. An example of application is included, involving a 188,500 DWT tanker.

Chapter V contains the main conclusions and gives some recommendations for future work.

CHAPTER I. THE PROBLEM OF HULL-MACHINERY RIGIDITY COMPATIBILITY

1. Strength vs Flexibility

In ship structural design, the most widely used measure of adequacy has traditionally been stress. The strength requirement insures that the stresses never exceed certain prescribed levels, so that the structural integrity is not affected. It is well known that the criterion for hull primary bending strength is section modulus. In reality, the strength criterion cannot be simply stated in terms of section modulus alone, since shear stresses can also be relevant, particularly in the vicinity of the ship's quarter points. Besides, the hull girder is subjected to other forms of loading, such as horizontal and transverse bending and torsion, and in addition to these primary or overall hull response forms, secondary and tertiary effects also have to be considered [10]. In any event, the measure of adequacy can, in general, be expressed in terms of stress or a combination of stresses, and since, at present, various methods of structural analysis can lead to a good estimate of the stresses in a structure, the designer can be reasonably sure of meeting the required strength.

In addition to a strength requirement, a stiffness requirement can also be defined. This implies that the structure must be designed to avoid excessive deformations or deflections which would change excessively the geometry and prevent the structure from withstanding the prescribed loads. In the case of bending stiffness, the stiffness (or flexibility) criterion is obviously moment of inertia, I, since under a given bending moment, curvature is inversely proportional to I. In the case of shear stiffness, the criterion is not so easily defined, since shear deformations can be a rather complex function of the cross-sectional geometry, the shear modulus and Poisson's ratio [11]. In any case, it can easily be shown that stiffness and strength do not necessarily come together, which means that for a given general geometrical configuration the scantlings which lead to maximum strength are not those which imply maximum stiffness. Thus, a compromise between these two objectives is usually necessary [12].

While in the case of strength, relatively simple material tests can lead to clear practical design limits, in the case of stiffness the same is not true. Upper or lower limits on allowable stiffness are not easy to define, even in the most simple structural arrangements, unless very specific operational requirements are to be met. The fact that hull stiffness cannot in practice be changed substantially after the ship is built is another factor which makes the whole problem of required stiffness an important one.

2. Causes and Effects of Excessive Hull Flexibility

As mentioned in [1], the interest in fundamental hull girder stiffness was increased when proposals for building vessels entirely of aluminum were first studied [13]. This is obviously a matter of special relevance in the case of deadweight carriers, where weight saving is a particularly important consideration.

Several factors have caused the recent trend in decreased hull girder stiffness. The most important are [12]:

i. Increased length.

ii. Use of high-strength steels.

iii. Less stringent corrosion or wastage allowances.

iv. Increased knowledge about structural response, encouraging the use of smaller factors of safety and smaller scantlings.

v. Wider use of design optimization techniques, in particular weight minimization, leading also to smaller scantlings.

vi. Use of aluminum for superstructure construction.

As a result of increased hull flexibility or limberness, various detrimental effects can take place, affecting the ship's performance to varying degrees of severity. These can best be defined, as proposed in [14], depending on whether their major impact is of a dynamic or static nature, as follows:

Dynamic

a. Personnel discomfort from propeller-induced or other steady-state vibration and noise.

b. Malfunction of electronic or mechanical equipment, including main shafting, bearing and gear failures from vibration or excessive displacement.

c. Unacceptable high-frequency stress peaks in primary hull structure due to impact loads such as slamming.

d. Fatigue of primary hull structure from the steady-state vibratory response of springing.

Static

e. Excessive curvature causing premature structural instability failure in the primary hull structure.

f. Excessive deformation when loaded resulting in reduced payload capacity in the sagging condition, or lower bottom clearance.

g. Excessive hull deformation imposing structural loads on non-structural items or components, such as joiner bulkheads, piping, propulsion safting, hatch covers, etc.

h. Second-order effects introducing inaccuracies into many of the customary naval architecture calculations.

Some of the aspects listed above have already been the subject of various investigations. In particular, the effects of decreased hull stiffness upon dynamic response from slamming and propeller-induced vibration have been studied in [15], the effects on the whipping bending stress components from slamming, or fatigue from springing, have been considered in [1], and the problem of shipboard vibration and noise control is reviewed in [16].

In the present study, the problem of hull-machinery foundation rigidity compatibility will be studied from a strictly static point of view, so that it essentially falls under (g) above. It is obvious that dynamic effects can also affect the interaction between the hull and the machinery foundations, not only because of the dynamic distortions on the hull caused by ship motions, but also because of the intrinsic dynamic nature of the machinery components [17,18]. This is a subject which will be addressed in more detail at a later stage.

3. Factors Affecting the Hull-Machinery Foundation Compatibility

a. Static primary deformation of the ship's hull girder.

This is the primary ship structural response, in which the ship's hull girder is treated as a simple free-free Bernouli beam. Wave hogging and sagging conditions are usually taken into consideration, and the effect of quartering seas can also be allowed. Normally, the primary concern is vertical bending, but horizontal and transverse bending can also be taken into consideration.

In addition to flexural deformations, shear deformations can also bring an important contribution to the overall hull girder distortions. Taylor [19] found this contribution to be as much as 19% of the total hull deflection, so that it should not be disregarded. The same opinion is expressed in [2].

In [2], it was found that in the case of tankers with machinery aft, the hull girder curvature in the machinery compartment would essentially have an opposite sign as in the remaining part of the ship. Thus, if the ship is in the light condition, the hull would in general deform in hogging while the double bottom in the machinery compartment would deform in sagging. The converse would happen in the fully loaded condition. This indicates how a careful computation of the hull girder deflection can help in detecting the possibility of incompatibility between the hull and the machinery.

b. Dynamic primary deformation of the ship's hull girder

Vibration effects on the hull girder can obviously affect the compatibility between hull and machinery. The same can be stated with respect to hull bottom impact or slamming [17-19].

c. Thermal effects

Thermal effects due to oil, seawater and steam can have a considerable impact on the deflections of double-bottom and foundations of turbines, gear and gear casing. These effects are in general taken into account when designing the machinery support systems [2].

d. Lineshaft alignment and vibrations.

Misalignment and longitudinal, lateral and torsional vibrations induced into the shafting by the propeller and/or the propulsion plant should be considered [2].

e. Shaft stiffness

Due to larger installed horsepower and a tendency toward single-screw ships, shaft diameters have increased and, as a result, lineshafting stiffness has also substantially increased. Since, on the other hand, the hull stiffness has in general decreased, this fact can also be a source of incompatibility between the hull girder and the machinery foundation [2].

f. Ship's beam

The structure of the double bottom is usually transversely framed, so that as the beam increases, its flexibility also suffers an increase, which can only be compensated by increasing the scantlings of the double-bottom structure. If this is not achieved, the machinery-foundation stiffness might be too low, and this can obviously lead to possible incompatibility between the hull and the machinery. Note that this beam effect can be quite relevant, since the deflection is essentially proportional to the fourth power of the span [2].

g. Local deformations

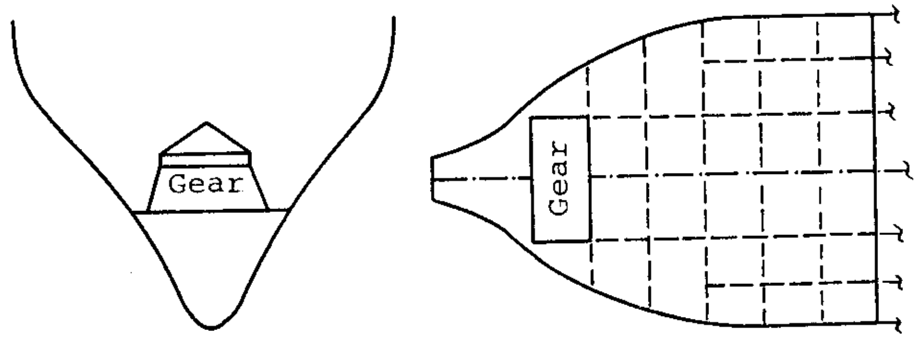
The double-bottom structure is essentially composed of stiffened panels supported by floors and side shell. The hull itself is also an assemblage of stiffened panels supported by transverse bulkheads and web frames. Hydrostatic pressure and dead loads act on these panels and produce local deformations which can also affect the hull-machinery compatibility. Local deformations and insufficient double-bottom stiffness are in part responsible for the motions of rocking and tilting of the thrust block, known to have a very detrimental effect on reduction gears and bearings [2]. These motions are amplified by the fact that the thrust block can be considered as a cantilever beam embedded into the double-bottom structure with an overhung load. This cantilever effect is obviously more pronounced for larger spans, i.e. when the thrust is applied at a greater height from the double bottom, a factor which should carefully be weighed in designing the machinery layout.

h. After body shape

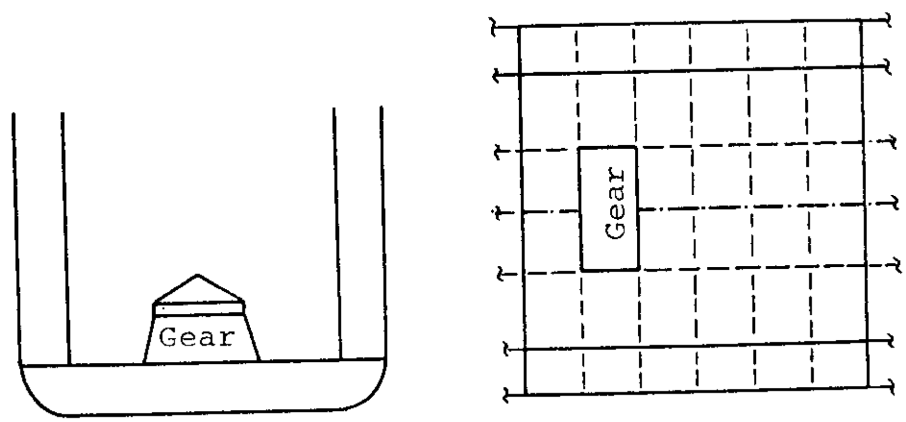
The after body hull shape can have an important impact on the local hydrostatic pressure loading on the hull, and this can also affect the hull-machinery foundation compatibility problem particularly if the machinery spaces are aft. If the stern is full or spoon-shaped, the hydrostatic pressure forces on the side shell are likely to be more important than the corresponding forces on the bottom. In the case of a transom type stern, the opposite is in general true. Thus, the two extreme hull after-body shapes affect differently the overall and local loading on the ship, in the sense that while one normally implies excessive buoyancy on the hull girder and large pressures on the shell plating aft, the other does not.

The after-body shape can have another important impact on the hull-machinery compatibility problem by the way it influences the machinery spaces general shape if located aft. In the case of a tanker, for example, as represented schematically in Fig. 1a, the machinery space can be quite narrow in way of the reduction gear casing. The short floor span is very stiff and can normally provide adequate machinery support. In other ships, such as the LASH (discussed in detail in Section 5) the machinery space is essentially square (Fig. 1b). In way of the reduction gears, the floor span is very large and the stiffness is greatly decreased, particularly if the reduction gear is not close to a transverse bulkhead. This factor is obviously related to the beam effect discussed in (f) above.

AFTER BODY SHAPES



(a)



(b)

Figure 1

i. Machinery characteristics

The machinery type, size and location can also be expected to affect the compatibility between hull and machinery. Larger units produce larger concentrated loads at the supporting points, so that the foundation stiffness becomes critical. The machinery location along the hull is also an important consideration, since the hull stiffness is not constant throughout the ship's length. The shafting length and number of bearings are also important parameters, since they affect directly its stiffness.

j. Draft changes

Local hydrostatic loading on the hull is obviously directly related to the ship's draft. If large draft changes can occur between the fully loaded and light conditions, such as normally happens in the case of tankers, then the local hull deformations can also vary largely, and this can also affect the hull-machinery compatibility problem.

4. Brief Review of the Solutions Proposed in the Literature

In order to reduce the possibility of hull-machinery incompatibility, various solutions have been proposed in the literature. Essentially these can be classified under three main categories as follows:

(a) reduce the stiffness of the shafting, adjusting the equipment to the increased flexibility of the structure;

(b) increase the stiffness of the foundation and double-bottom structure, and, thereby, adapt the structure to the support requirements of the machinery;

(c) modify the design of machinery components so as to adapt them to the increased flexibility of the hull.

In the first group, (a), we can include various possible alternatives, such as the curved alignment of the line shafting. In fact, it is well known that the straight alignment of shafting does not provide a proper operation of the main gears, which leads to the necessity of a rational curved alignment [20]. In addition, factors such as a careful choice of the number of bearings, the rational positioning of the first bearing aft of the main gear with respect to the main gear or diesel engine and the position of the thrust bearing, must also be considered [21]. This subject will be discussed in the next section when describing the modifications introduced in the original LASH design. This case study along with the example described in Chapter IV fully outline the steps the designer should take in situations such as this one.

In the second group, (b) several possible alternatives for increasing the foundation stiffness have been proposed such as reinforcing the thrust-bearing foundation and the adjustment of the web frame thickness [21]. The next Chapter, discusses in detail the subject of foundation design, which is of major importance in problems of this nature.

In the third group, (c), several solutions discussed in the literature can be included. One relates to a new type of bull-gear design termed a "transflex bull gear" [23]. The novel feature in this design is a flexible diaphragm plate which transmits to the gear wheel rim less than 1/30 of those forces and couples transmitted by conventional design. Thus, if this new design is adopted, some of the problems related to hull-machinery compatibility could be reduced.

Another possibility suggested in [2] is the introduction of a flexible coupling between the main gear shaft and the intermediate shaft.

Still another possible solution deals with diesel engines. A box girder design of the machinery base between bedplate and cylinder block, rather than a design based on columns is known to increase substantially the rigidity of the combined engine-hull structure foundation [8,23]. As a result, the double-bottom distortions are reduced by the engine itself with a considerable margin of safety, reducing the possibility of hull-machinery incompatibility. In the case of medium-speed diesel engines, improved designs for reduction gears have also been proposed, with the objective of reducing the detrimental effects caused by excessive hull and machinery flexibility [24]. The subject of diesel engines is considered in detail in Chapter III.

5. A Case Study: The LASH Vessel

5.1 Introduction

In late 1970, the first of twenty large barge/container ships of the LASH type was delivered to its owner after successful trials; however, in the next two years half of these vessels developed machinery troubles that were found to be caused by an incompatibility between the flexibility of the hull structure and the degree of rigidity required for proper support of the machinery. This costly experience, together with problems of a similar nature encountered by some large European vessels, led to recognition of the need for a better and wider understanding of hull/machinery compatibility. The account of the difficulties with the LASH vessels which follows is based on the condensation of a very large amount of test data, experience, and analysis and is not intended to represent a detailed history.

5.2 First Group of Ships

The machinery arrangement, machinery foundations, and hull structure aft of amidships were essentially the same for each of the first eleven LASH vessels. The main particulars of the LASH vessels are summarized in Table I.

The main propulsion machinery consisted of a 32,000 SHP steam turbine driving a single propeller through a standard locked-train, double-reduction gear. All vessels of this first group experienced distress on the reduction gear teeth in varying degrees of severity during their early service life and replacements for several gears were required.

5.3 Second Group of Ships

Modifications made to the machinery arrangement, main shafting, and hull structure of nine vessels comprising the second group eliminated the gear problems. Generally, these changes were retrofitted to the first group and now both groups have operated successfully for many years.

5.4 Gear Distress

Operation of the main machinery in the first three ships was apparently satisfactory when delivered. Following trials of the fourth ship in mid-1971, however, inspection revealed evidence of distress on the second-reduction gear teeth with heavy loading at the forward ends of both helices. Pitting and scuffing led to rapid deterioration and eventual replacement. Subsequent examination of the first three vessels indicated similar distress although very much less severe; a pattern that generally was repeated in the remaining vessels of the first group. There were no signs of distress in the first-reduction gears.

Initially, the reasons for the gear problem were not understood. Attention was focused on the internals of the gear with a detailed analysis of the gear design by the manufacturer, consultants, and shipbuilder. Modifications were made to the gear in those areas that were suspect; however, these internal changes apparently did not eliminate the basic problem and the gears continued to show increasing distress.

Signs of heavy loading on the gear teeth at the forward (or aft) ends of both helices are generally an indication that the gear and pinion axes do not remain parallel during operation. Fig. 2 illustrates how varying amounts of misalignment significantly affect the tooth contact across the mesh.

TABLE I

LASH MAIN CHARACTERISTICS

Principal Dimensions

Length BP	724'
Breadth	100'
Depth	60'
Draft	28'
Displacement	32,650 tons

Machinery

Steam turbine	32,000 SHP

Engine Room Construction

Transverse framing, spacing 7'-4"
Engine room length 73'-4"
Engine room width in way of reduction gear 70'-4"
Web frames at every frame
Tank top - plating thickness = 3/4"
Bottom C.L. girder 3/4" thick
Bottom side girders 9/16" thick
Double bottom depth 8'-9"
Spacing between longitudinals 6' average

Shafting Details

Line shaft diameter 21.88" (original)
Tail shaft diameter 28.56" (original)
Thrust bearing location aft of #2 bearing (original)
Number of line shaft bearings 3
Height of thrust bearing center above inner bottom 7'

LASH 2nd REDUCTION PINION/GEAR MESH
TOOTH CONTACT VS ALIGNMENT

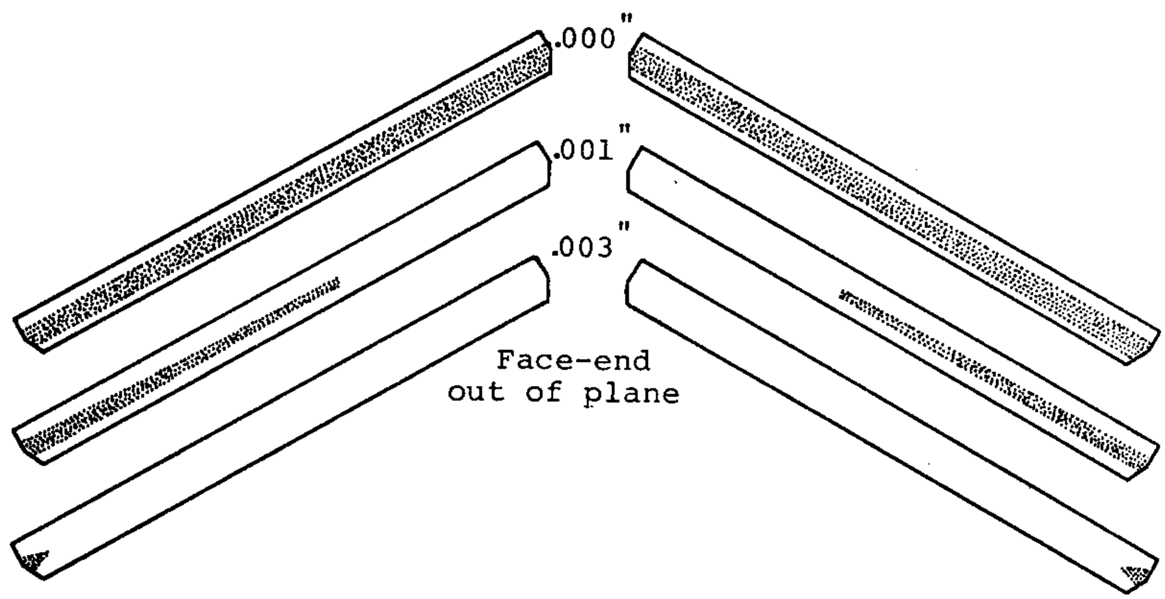


FIGURE 2

The second-reduction, or bull-gear shaft, when connected becomes a part of a continuous beam system supported by a series of bearings. For convenience, these bearings will be numbered from the forward end. It is customary for the gear manufacturer to specify the maximum allowable difference (ΔR) between #1 and #2 bearing static reactions. One manufacturer has based this limit upon a maximum mismatch, or opening between the teeth of meshing pinions and gears, of approximately 0.0002 inches per foot of face width [25]. With approximately five feet between centerlines, this is equivalent to a relative movement of 0.001 inches between #1 and #2 bearings. Generally, ΔR falls between 20-30 per cent of the static reactions [26], and in the case of LASH was established by the manufacturer as 12,400 pounds.

5.5 Bull-Gear Monitoring System

In order to determine what was happening, an electronic system was developed by the manufacturer to continuously monitor the journal position within the oil clearance of each bull-gear bearing. A simplified diagram of the system is shown in Fig. 3. Two proximity probes located in each bearing serve to measure gaps "A" and "B". This enabled the system to display a dot for each journal on an oscilloscope screen, each dot representing the center of the corresponding journal. Electronic magnification permitted movements as small as one half mil to be measured. The display was adjusted initially so that the two dots (forward and aft bearings) were superimposed when both journals were at rest in the bottom centers of their respective bearings. In this position, pinion and gear centerlines were parallel as manufactured and later confirmed by tooth contact tests after installation. Although the journals move to other positions as speeds and loads increase, both journals should move in the same manner if the pinion and gear axes are to remain parallel. Thus, any spread between the dots which develops in operation is a measure of the misalignment of the gear relative to the pinions.

It was found that the bull-gear did in fact skew as power and speed were increased. Accordingly, the position of the first line-shaft bearing was adjusted during operation and the gear could be made to operate in a parallel position at either low power or full power, but no single adjustment would allow proper operation through the entire power range. This suggested that there might be relative movements between the gear bearings and the line-shaft bearings as power was increased.

5.6 Structural Deflection Tests

Test arrangements to measure structural deflections of

AN ELECTRONIC SYSTEM TO MONITOR THE JOURNAL POSITION WITHIN EACH BULL GEAR BEARING DURING OPERATION

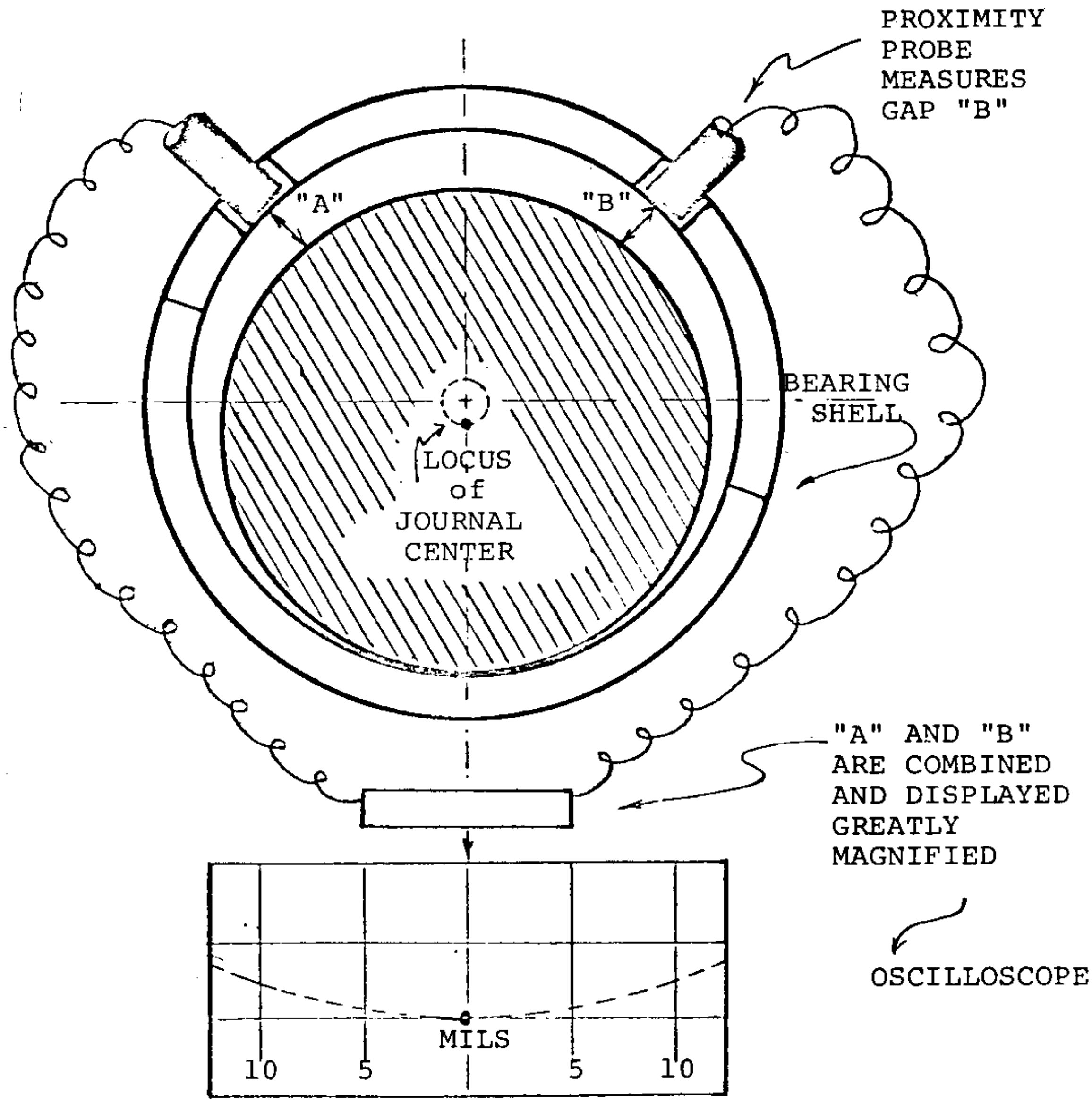


FIGURE 3

gear case and its foundations were developed for the tenth vessel and data were obtained while underway at full power. A second series of tests were made on an earlier vessel while at sea using different test methods and equipment. Finally, the test arrangements of the tenth vessel were applied to the eleventh vessel and data were taken at dockside where full-power torque and thrust were simulated by special hydraulic devices. The deflections measured in these tests by different methods showed reasonably good agreement. Data taken during dockside tests have been chosen for illustrative purposes because it was possible to apply torque and thrust independently.

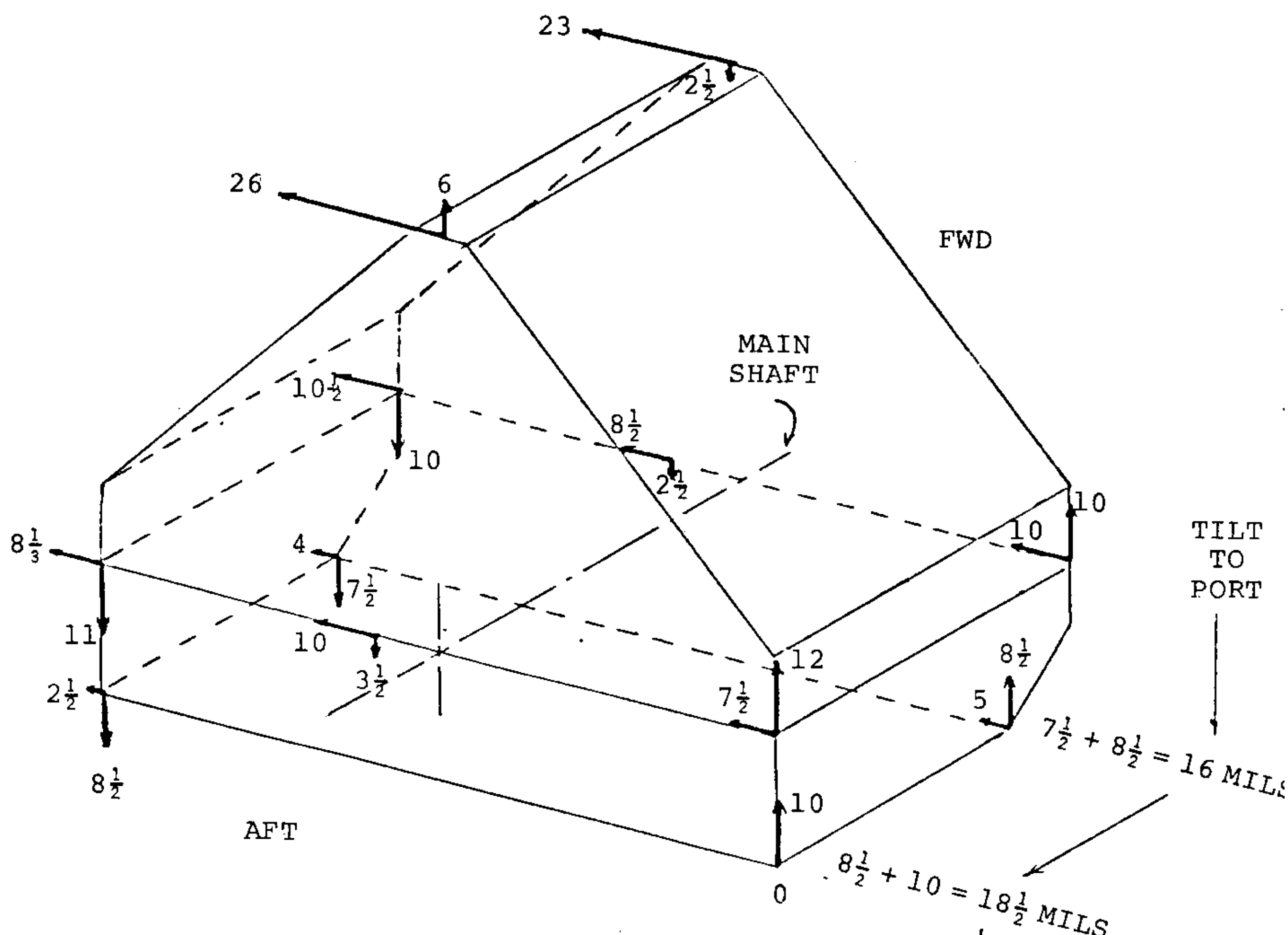
5.7 Torque Test

The direction and magnitude of deflections at selected points on the gear case and its supporting structure while under simulated full-power torque alone are shown in Fig. 4. The forces due to torque reaction are downward on the port side and upward on the starboard side. The structure supporting the gear deflects in a corresponding manner and if the athwartship movements are plotted it will be found that each deflection is approximately proportional to its distance from a longitudinal axis somewhere in the inner bottom. The entire gear case, therefore, rotates to port as shown in the exagerated view of Fig. 5. The movement of the bull-gear bearings relative to the line-shaft bearings is about ten mils and begins to explain why a satisfactory alignment could not be established throughout the power range. The tilt at the foundation is greater at the aft end by about two to three mils, thus the gear case is twisted and the pinion axes are skewed relative to the bull-gear axis.

5.8 Thrust Test

The deflections which were caused by the application of full-power thrust only are shown in Fig. 6. In this test, all movements of the gear case were due to deflection of the foundation because there were no forces or moments applied to the gear case. The three mil readings at the lower aft corners were considered invalid and were assumed to be about six mils in agreement with other data on the aft end of the gear case. The forward movements of the gear case were not harmful since they were parallel to the gear and pinion axes. The five to seven mil depression, however, was significant since it changed the position of the bull-gear bearings relative to the lineshaft bearings.

DEFLECTIONS DUE TO TORQUE



FOUNDATION TWIST------- 2½ MILS

Figure 4

GEAR CASE ROTATION

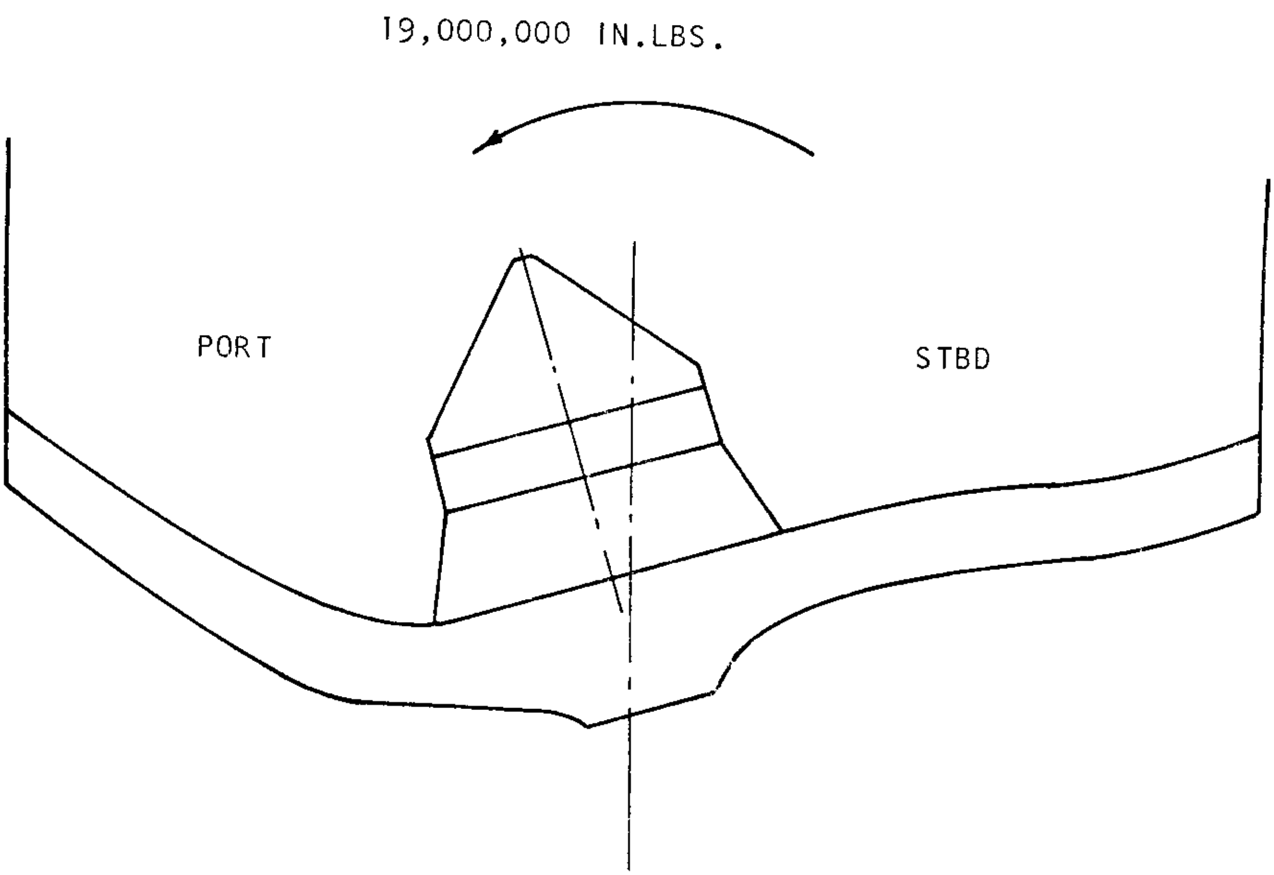
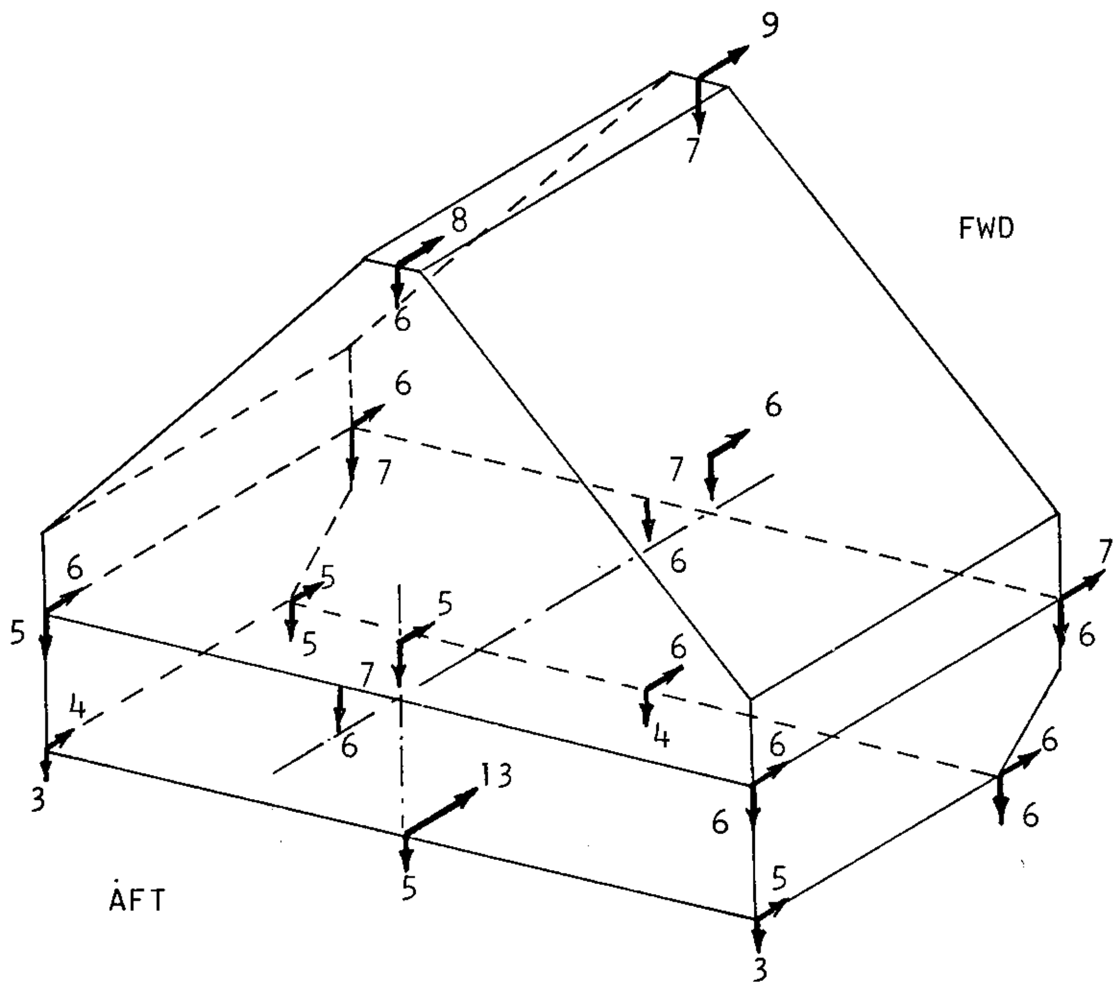


Figure 5

DEFLECTION DUE TO THRUST



3 MIL VALUES AT LOWER AFT CORNERS ARE INVALID DUE TO POOR INDICATOR MOUNTING

ALL MOVEMENTS OF THE GEAR CASE ARE DUE TO FOUNDATION DEFLECTION BECAUSE THERE ARE NO FORCES OR MOMENTS DIRECTLY APPLIED TO THE GEAR CASE

THE UNIT IS DEPRESSED ABOUT 5-7 MILS

Figure 6

5.9 Combined Torque and Thrust

The measured deflections due to torque and thrust may be combined as shown in Fig. 7. The encircled values were developed from the results obtained by a finite-element analysis program and in most cases are in reasonable agreement with the measured results considering the instrumentation problems and the complexity of the calculations.

5.10 Main Thrust Bearing

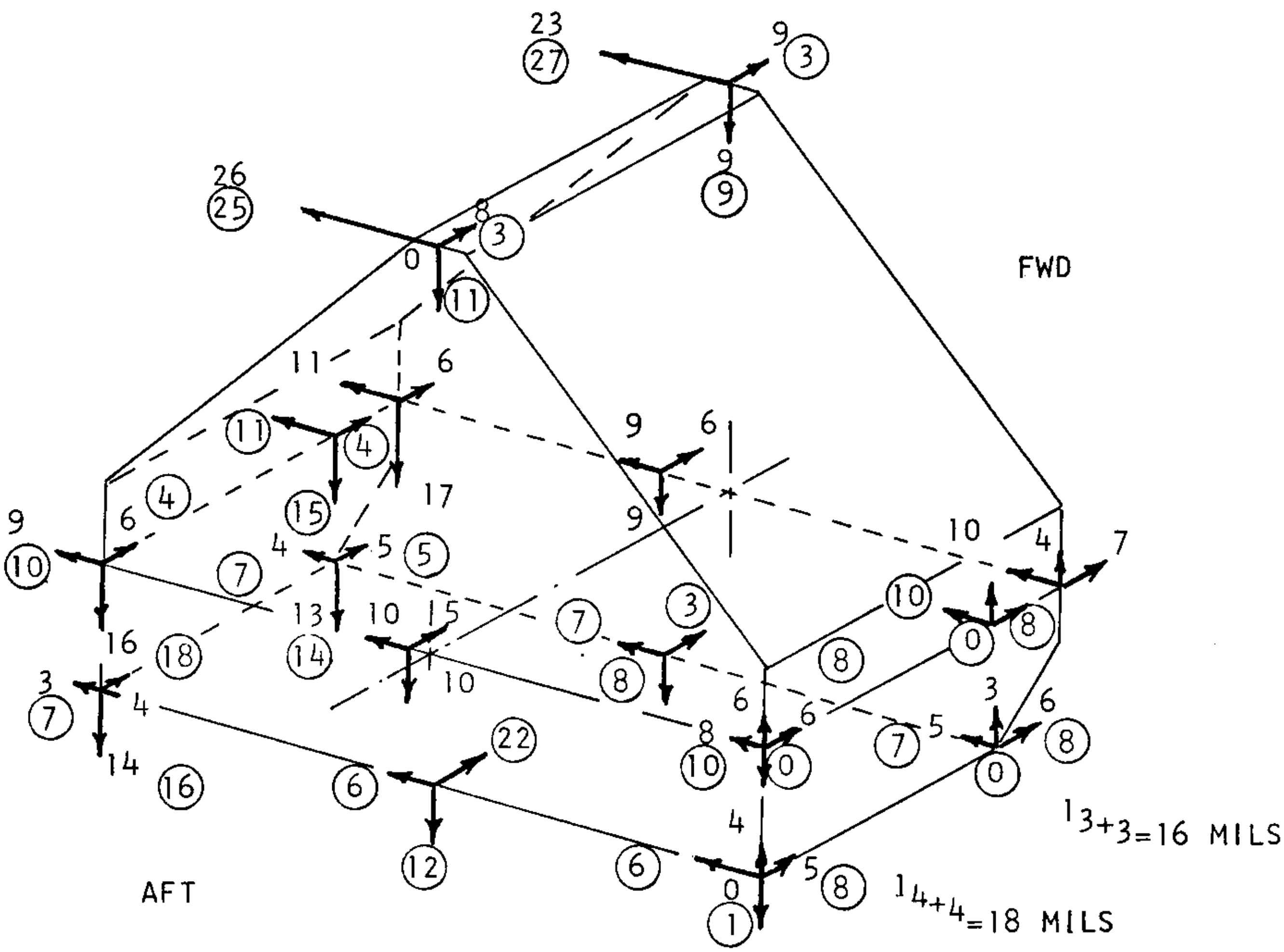
The depression of the bull-gear bearings appears to have been caused by the application of thrust at the main-thrust bearing just aft of the gear. Fig. 8 illustrates the arrangement of the thrust bearing and gear foundations and shows that the main-thrust bearing moved forward 20 mils and downward 5 mils. The motion was essentially rotation as shown in Fig. 9 with the lower thrust shoes becoming more heavily loaded. Multi-shoe thrust bearings have devices which are intended to equalize the loads on the shoes, or pads; however, research has shown that these arrangements are not always effective. Reference [28] states "leveling links are unable to follow shifting of the housing alignment with full thrust load, and force gauges show some pads to be taking nearly the entire load." The tests indicated this effect to be present at loads down to twenty per cent of rated thrust. Failure to equalize the loading of the pads was apparently caused by friction at the pad and link contacts and the attempts to release this friction by applying a vibration shaker to the housing were not successful.

An eccentric load at the thrust collar would introduce a bending moment in the shaft which would tend to unload the #2 bearing. Based upon the measured rise of #2 bearing between zero to full thrust of two mils, and by reference to the shaft flexibility characteristics, it has been estimated that ΔR could be changed by as much as 80 per cent of the maximum allowable value, a significant amount.

5.11 Dynamic Deflection Due to Rolling

The instrumentation shown in Fig. 10 was applied to one vessel of the first group of ships to measure relative deflections of the forward and aft sections of the gear foundation while the vessel was at sea. Dynamic deflections in the athwartship direction of 2-1/2 - 5 mils were recorded with the vessel rolling through a total amplitude of 8-13 degrees. Large roll angles, such as occur in heavy weather, were not encountered during the test and no further measurements are available; however, the data appear to indicate that relative

DEFLECTION DUE TO COMBINED TORQUE AND THRUST

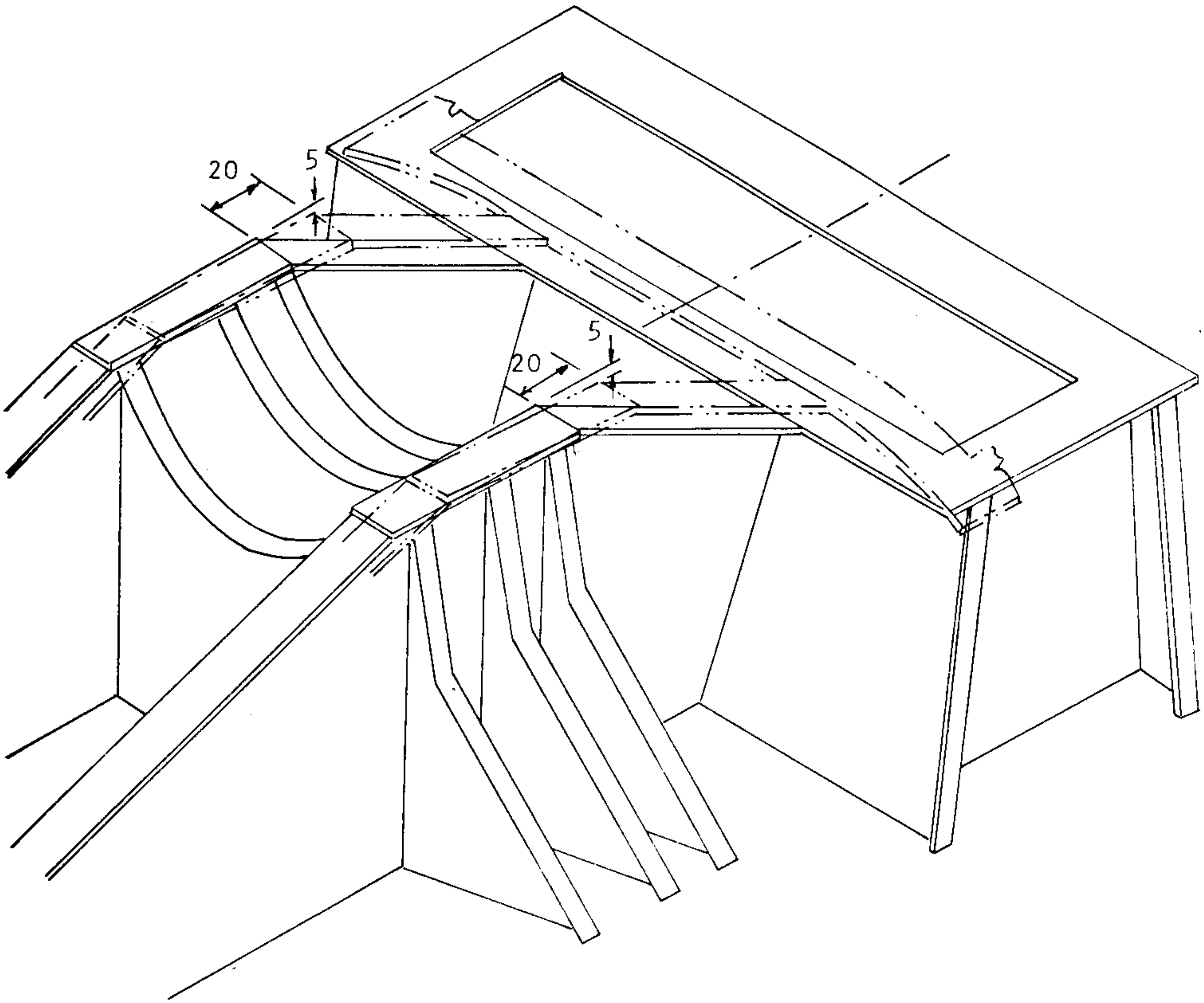


FOUNDATION TWIST________2MILS

CIRCLED FIGURES ARE CALCULATED BY THE ABS PROGRAM "DAISY"
FIGURES WITHOUT CIRCLES ARE BASED ON MEASURED DEFLECTIONS
ALL VALUES ROUNDED TO THE NEAREST MIL

Figure 7

MEASURED DEFLECTION OF THE THRUST BEARING FOUNDATIONS
DUE TO FULL POWER THRUST



ALL VALUES ARE MILS

Figure 8

MAIN THRUST BEARING
FOUNDATION DEFLECTION

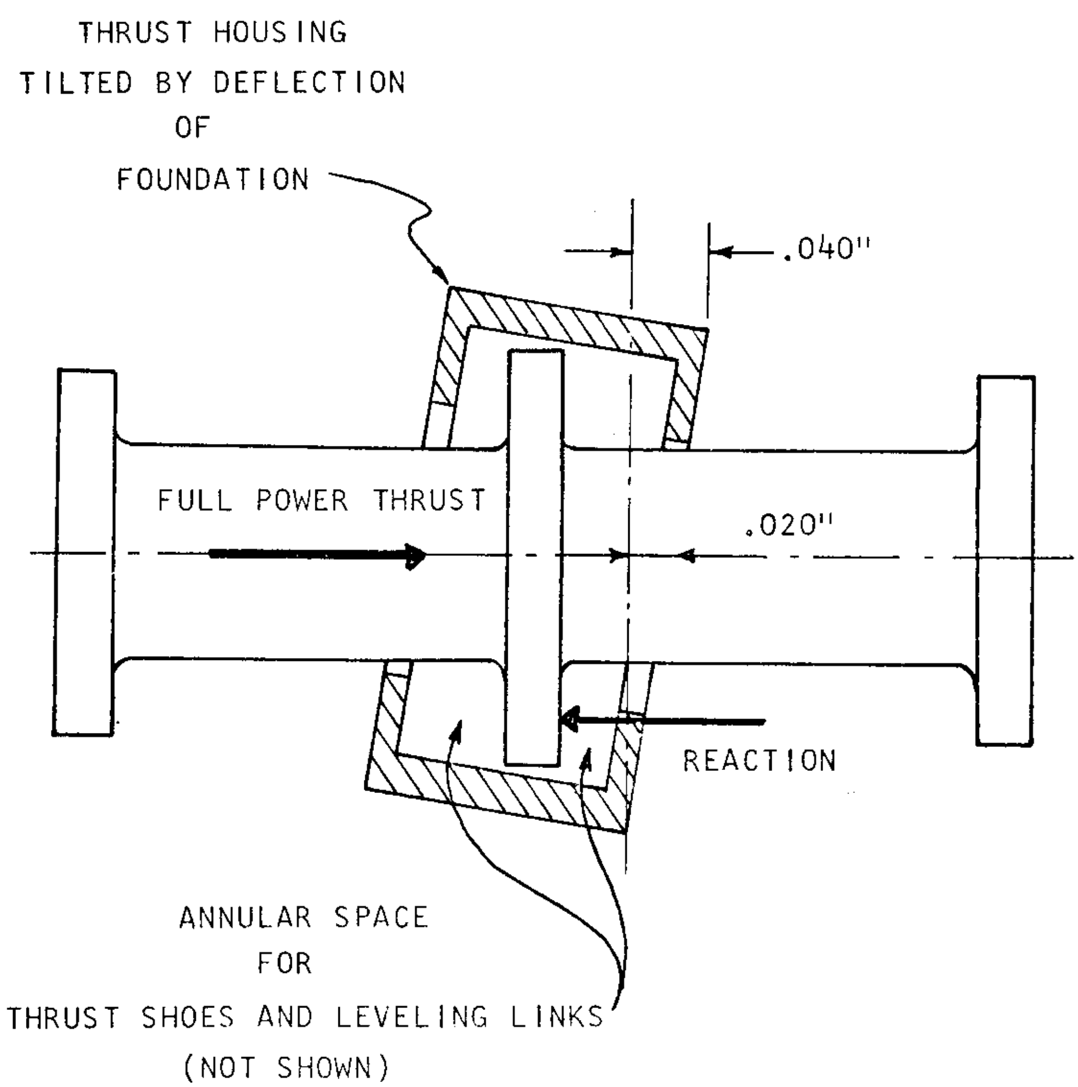
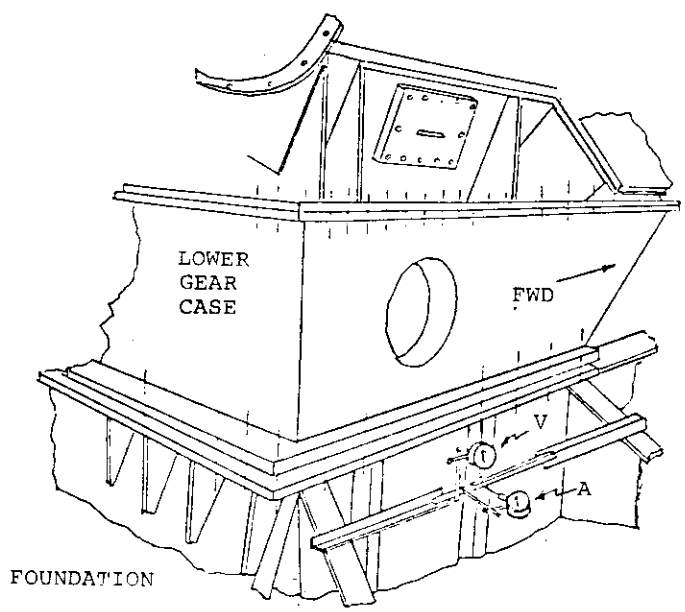


Figure 9

RELATIVE DEFLECTION OF THE FORWARD AND AFT SECTIONS OF THE GEAR FOUNDATION MEASURED AT SEA WHILE THE VESSEL WAS ROLLING



ROLL ANGLE DEGREES	RELATIVE DEFLECTION MILS	
	V	A
5s-3p = 8	$\frac{1}{2}$	$2\frac{1}{2}$-$3\frac{1}{2}$
6s-4p = 10		3-4
$7\frac{1}{2}$s-$5\frac{1}{2}$p = 13		4-5

Figure 10

deflection increases with roll angle. This would imply that very significant deflections may occur with large roll angles.

If the unit is assumed to be in alignment under static conditions, a positive/negative nonparallel condition would occur at the gear mesh in each roll cycle and would be expected to cause heavier contact at both ends of each helix. This condition was reported on several vessels, thus tending to support the dynamic deflection measurements.

When small metallic particles are found on the magnets fitted in the lubricating oil strainers, they generally come from deteriorating tooth surfaces. Such particles were often found on those vessels that suffered severe tooth damage. It was noticed that the rate at which particles collected usually increased during heavy-weather conditions. It is also possible, of course, that some of this effect may have been due to the agitation of the lubricating oil sump which stirred up particles that had been settled at some previous time.

5.12 Shafting System Modifications

Three important changes were made to the eleventh vessel: (Fig. 11)

a. The line-shaft diameter between #2 and #3 bearings was reduced to the minimum allowable with the existing material.

b. The #3 bearing was moved aft

c. The main thrust bearing was relocated to a position aft of #3 bearing.

The effect upon shaft flexibility is illustrated in Fig. 12. Calculations indicated the gear case could now undergo equal vertical movements of #1 and #2 bearings (parallel) of $\pm$ 22 mils instead of $12\frac{1}{2}$ mils without exceeding $\Delta R = \pm 12{,}400$. This method of measuring shaft flexibility has been called "allowable setting error" [29] and should include (a) installation tolerance, (b) hull/foundation deflection, and (c) error in estimating the thermal rise of foundations and gear case. An absolute minimum value of $\pm$ 10 mils is recommended by reference [29]; however, reference [30] lists a number of ships which have operated between $\pm$ 10 mils and $\pm$ 6 mils. Installations with less than $\pm$ 6 mils were generally in difficulty and required modification.

The allowable vertical movement of one gear bearing (nonparallel) is considerably less but increased from $\pm$ 2-1/2 mils to $\pm$ 4 mils. The allowable movement of #3 bearing relative to the gear increased from $\pm$ 10 mils to $\pm$ 14-1/2 mils.

CORRECTIVE MEASURES TO SOFTEN THE SHAFTING SYSTEM

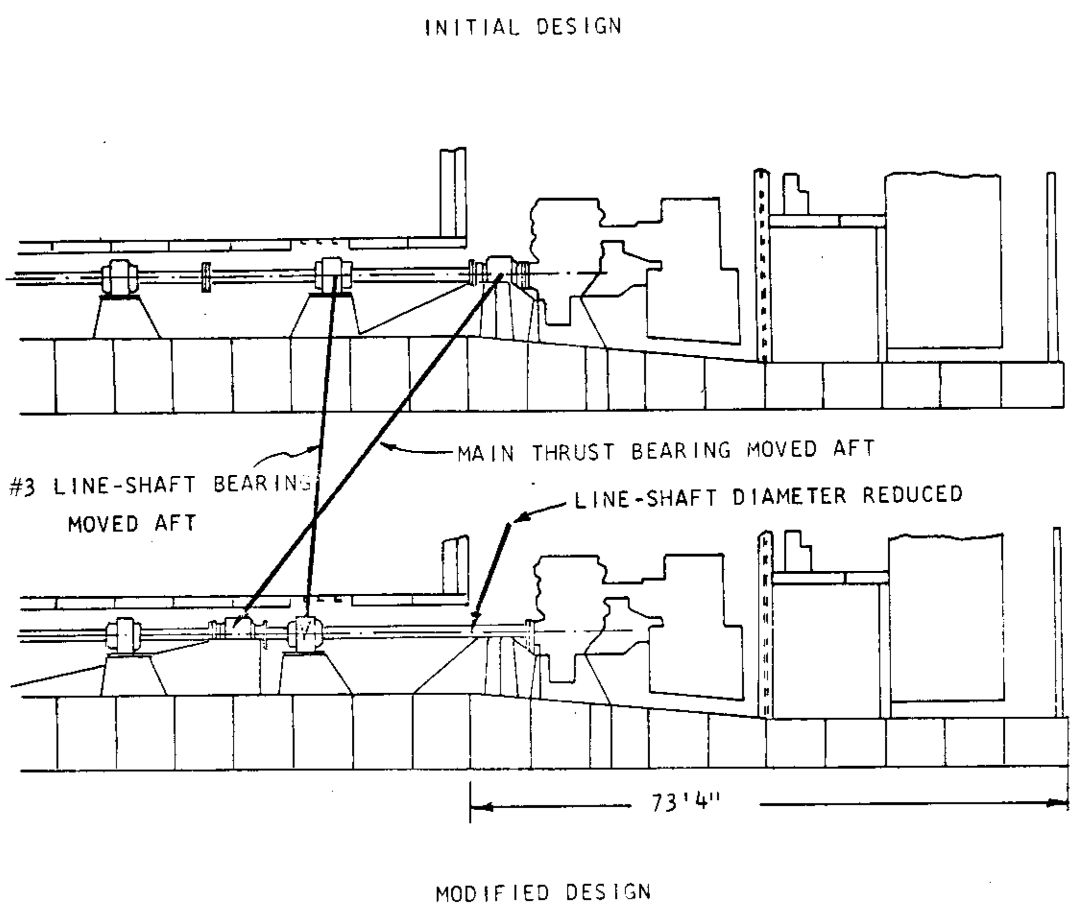
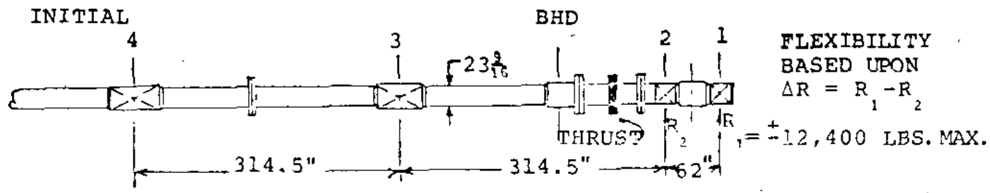


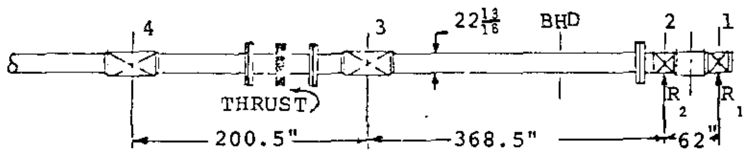
Figure 11

MAIN SHAFTING



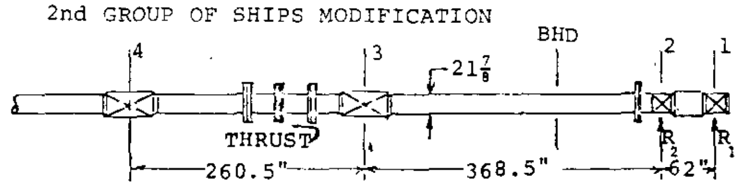
MAXIMUM ALLOWABLE	PARALLEL VERTICAL MOVEMENT OF #1 & #2 BEARINGS (ASE)	$= \pm 12\frac{1}{2}$ MILS
	NON-PARALLEL " " " " " " " "	$= \pm 2\frac{1}{2}$ "
	VERTICAL MOVEMENT OF #3 BEARING	$= \pm 10$ "

1st GROUP OF SHIPS MODIFICATION



MAXIMUM ALLOWABLE	PARALLEL VERTICAL MOVEMENT OF #1 & #2 BEARINGS (ASE)	$= \pm 22$ MILS
	NON-PARALLEL " " " " " " "	$= \pm 4$ "
	VERTICAL MOVEMENT OF #3 BEARING	$= \pm 14\frac{1}{2}$ "

2nd GROUP OF SHIPS MODIFICATION



MAXIMUM ALLOWABLE	PARALLEL VERTICAL MOVEMENT OF #1 & #2 BEARINGS (ASE)	$= \pm 24$ MILS
	NON-PARALLEL " " " " " " " "	$= \pm 4\frac{1}{2}$ "
	VERTICAL MOVEMENT OF #3 BEARING	$= \pm 16$ "

Figure 12

Span ratio (L/D) is used as a rough design guideline for shafting and is defined as the ratio of bearing center distance to shaft diameter. Reference [29] gives values of L/D varying from 12 minimum to 20-22 maximum. The original shaft design had a span ratio of 13 which increased to 16 after modification.

In later vessels, the use of higher strength material permitted a further reduction in shaft diameter and additional flexibility.

It is important to note that the flexibility of the original shaft design, although on the low side, fell within the guidelines based on past practice yet was not sufficient because of the increased flexibility of the machinery supports.

Relocating the main thrust bearing to a position aft of #3 bearing eliminated ninety per cent of the effect upon the gear of a bending moment in the shaft caused by tilt of the thrust bearing housing. In addition, the depression of the gear supporting structure upon application of thrust was eliminated by the increased distance from the gear and, perhaps more important, the bearing was positioned within the shaft alley which, with its sides, overhead deck, inner bottom and shell, formed a stiff girder.

5.13 Main Machinery Foundations

The original arrangement of the propulsion machinery foundations is shown in Fig. 13. The main thrust bearing, located just aft of the reduction gear, was subject to a force of approximately 380,000 pounds at full power. This force was transmitted to the shell via two longitudinal thrust girders which served to spread the load to the tank top and the grid of longitudinal and transverse structure within the inner bottom over a fore and aft span of about 28-30 ft. The moment formed by the force and the distance to the basic hull was responsible for the deflection of the inner bottom, the consequent change in slope or rotary movement of the thrust foundation, and the depression of the gear foundations.

Longitudinal stiffness is required to resist the bending moment and is obtained most effectively by deep girders. Fig. 13 shows that it was necessary to reduce the depth of the thrust girders in order to pass forward under the bull gear; however, aft of the thrust bearing there were no obstructions and the extent of the girders was limited only by the designer's decision.

INITIAL FOUNDATION DESIGN

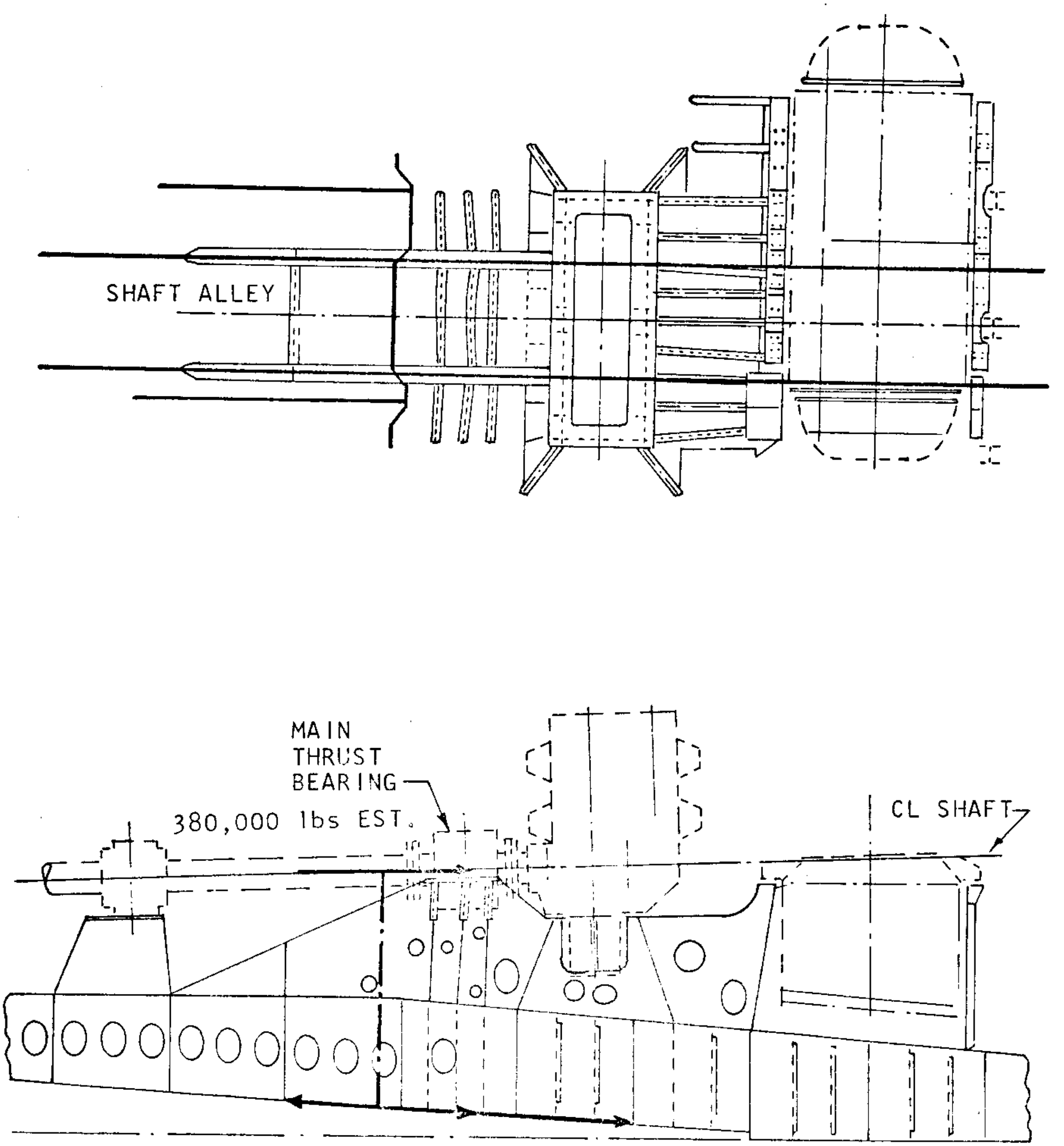


Figure 13

5.14 Structural Modifications

Changes were made to the eleventh vessel with the intent of generally stiffening the machinery supports and in particular reducing the tilt and twist of the gear foundations (Figs. 14 and 15). The extent of the changes was restricted by the practical difficulties of modifying an existing structural arrangement. Consequently, the limited changes which were made probably did not significantly reduce the deflections of the machinery supports. Additional structure was installed in the second group of ships while under construction including two complete floors under the gear foundation in an effort to provide increased resistance to deflection by torque forces (Fig. 16).

No measurements have been made on the vessels having additional structure and, therefore, it is not known to what degree these changes were effective.

5.15 Summary

Incompatibility between the flexibility of the hull structure and the rigidity requirements for support of the machinery caused failures in the reduction gears.

The flexibility of the main shaft, although at the low end of the allowable range, met existing design guidelines but was not sufficient to account for the hull structural guidelines.

No quantitative information was available regarding the flexibility of the hull structure during the design period and no measurements were made following structural changes. It does not appear that the structural changes alone would have been sufficient to correct the problem but instead the major portion of the improvement was due to the more flexible shafting and to the relocation of the main thrust bearing.

MODIFICATIONS TO STRUCTURE

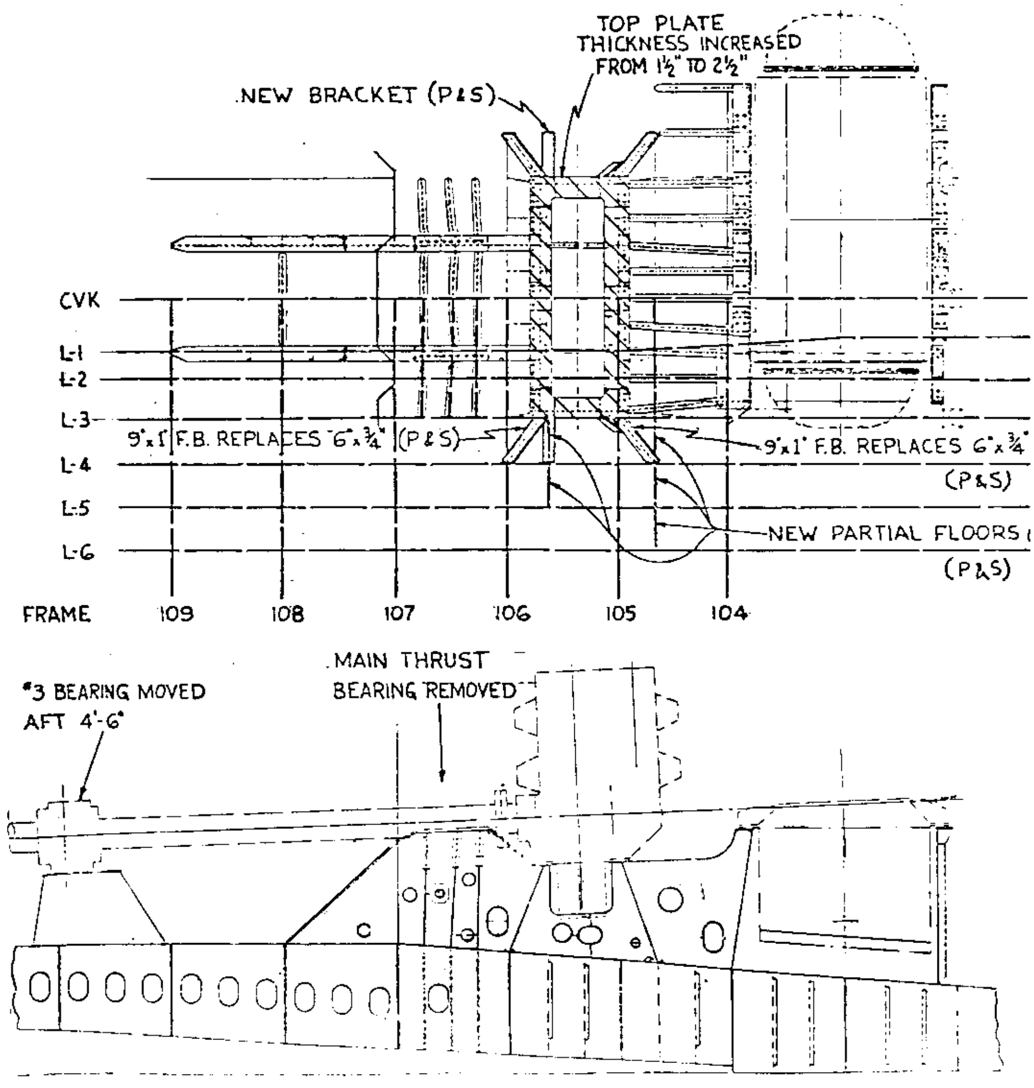
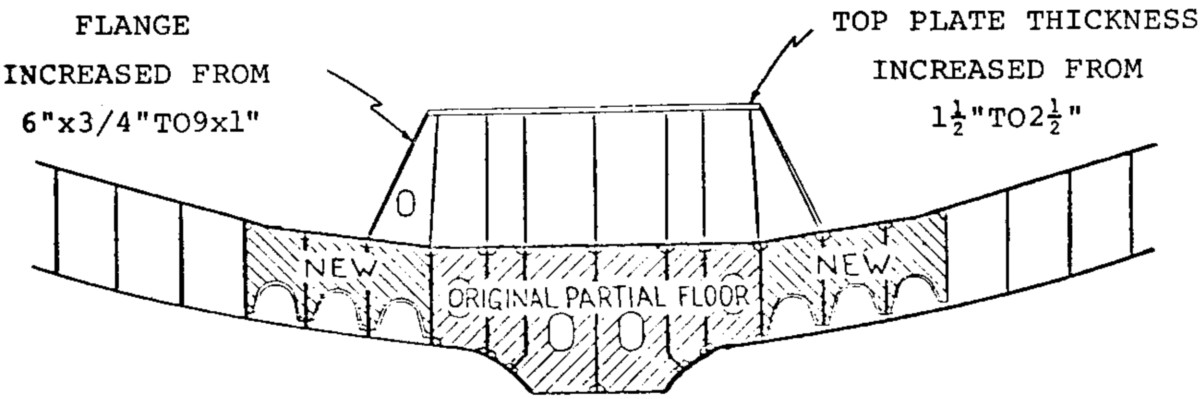


Figure 14

MODIFICATIONS TO STRUCTURE

REDUCTION GEAR FOUNDATION

FORWARD



AFT

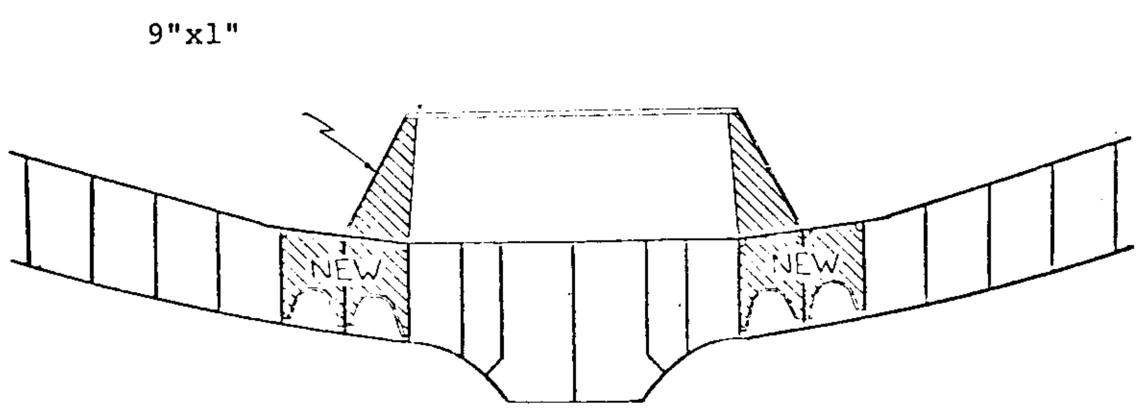
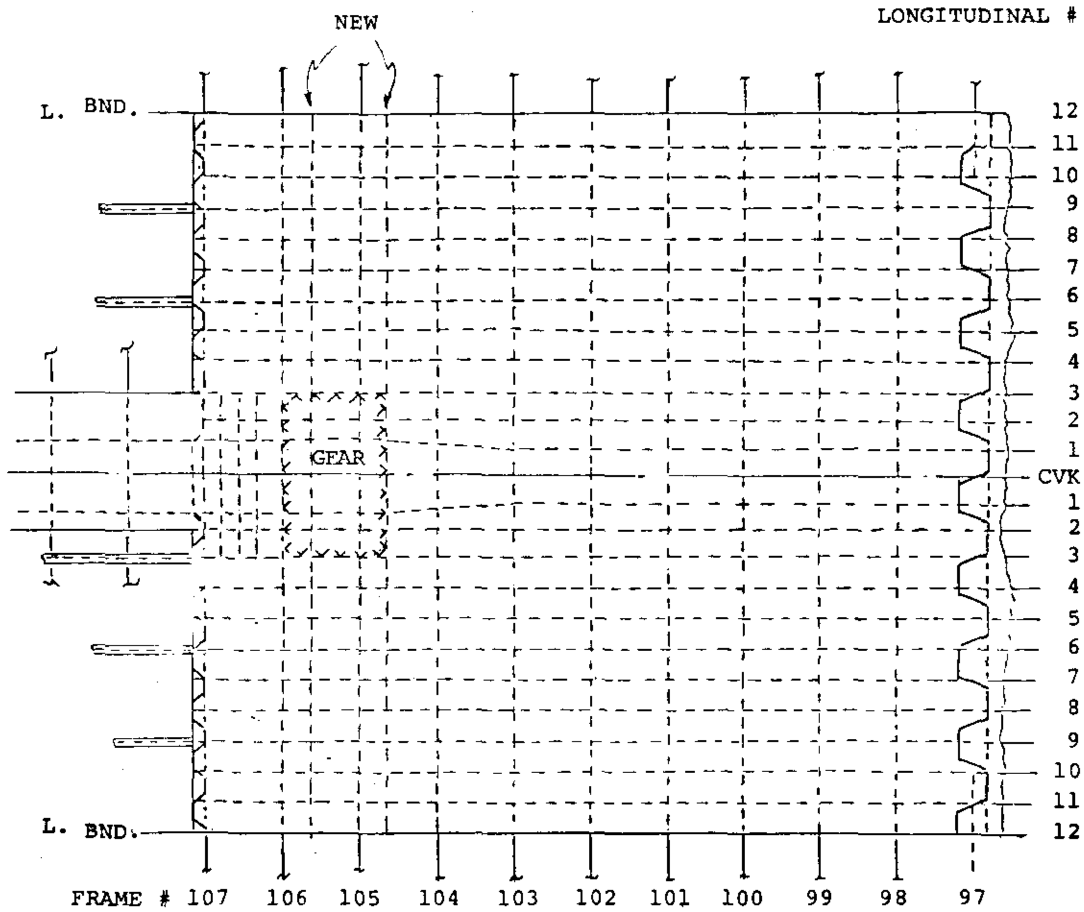


Figure 15

MODIFICATIONS TO STRUCTURE

2nd GROUP OF SHIPS

MACHINERY SPACE INNER BOTTOM



TWO ADDITIONAL FLOORS SUPPORT THE GEAR FOUNDATION
AND RESIST DEFLECTION DUE TO TORQUE

Figure 16

CHAPTER II. FOUNDATION DESIGN

1. Review of Machinery Spaces Structural Arrangement

1.1 Relevant Structural Parameters

In order to study the structural design of machinery spaces of large ships, it is convenient to discuss first the main parameters which characterize the design. Essentially, four groups can be defined, as shown in Table II (refer to Fig. 17). The first four describe the types of structural members present: transverse, longitudinal, vertical and plane. The transverse, longitudinal and vertical members are essentially prismatic, while the plane members are essentially plated structures. The fifth group includes what we may call load-related parameters.

Structural details are not considered here for various reasons. Firstly, because they do not contribute to a large extent to the overall stiffness of the machinery spaces, which is the point of main concern here. Secondly, because it is assumed that structural details are properly designed in order to insure proper joining of the various components, good structural continuity, and in order to avoid stress concentrations and local instabilities such as tripping. Finally, the whole area of ship structural details has already been the subject of extensive research sponsored by the Ship Structure Committee [31,32], so that there is no need to consider it here.

Transverse members include frames, floors and web frames. The important parameters which define frames are spacing and scantlings (web thickness and depth, flange thickness and depth). The side shell can be assumed to be attached to each frame providing an effective breadth based on any acceptable theoretical approach, such as the ones reviewed in [33]. Web frames can have a rather complex geometry, particularly towards the ship ends, and as such cannot easily be defined by a small set of parameters.

Floors can essentially be defined by the average thickness t_f and the location, or the number i_f of frame spacings separating them, assuming a uniform spacing is used throughout the machinery spaces.

Longitudinal members essentially include the bottom center girder, bottom side girders and stringers. The center girder can be defined by the depth d and thickness t_c. The side girders have in general the same depth as the center girder, so that the main parameters are the number, location and thickness of t_s. In addition, bottom girders can be stiffened in order to prevent sidesway or instabilities, and this obviously makes the structure's description more complex.

TABLE II

MAIN STRUCTURAL PARAMETERS

1. Transverse members

   a. frames (s, I)

   b. floors (i_f, t_f)

   c. web frames (i_w, I_w)

2. Longitudinal members

   a. center girder (d, t_{cg})

   b. side girders (i_{sg}, t_{sg})

   c. stringers (s_s, I_s)

3. Vertical members

   a. stanchions

4. Plane Members

   a. inner bottom (t_i)

   b. intermediate decks

   c. longitudinal bulkheads

   d. transverse bulkheads

5. Load-related parameters

   a. point of application of large weights

   b. thrust bearing above base (H)

MAIN STRUCTURAL COMPONENTS
OF MACHINERY SPACE

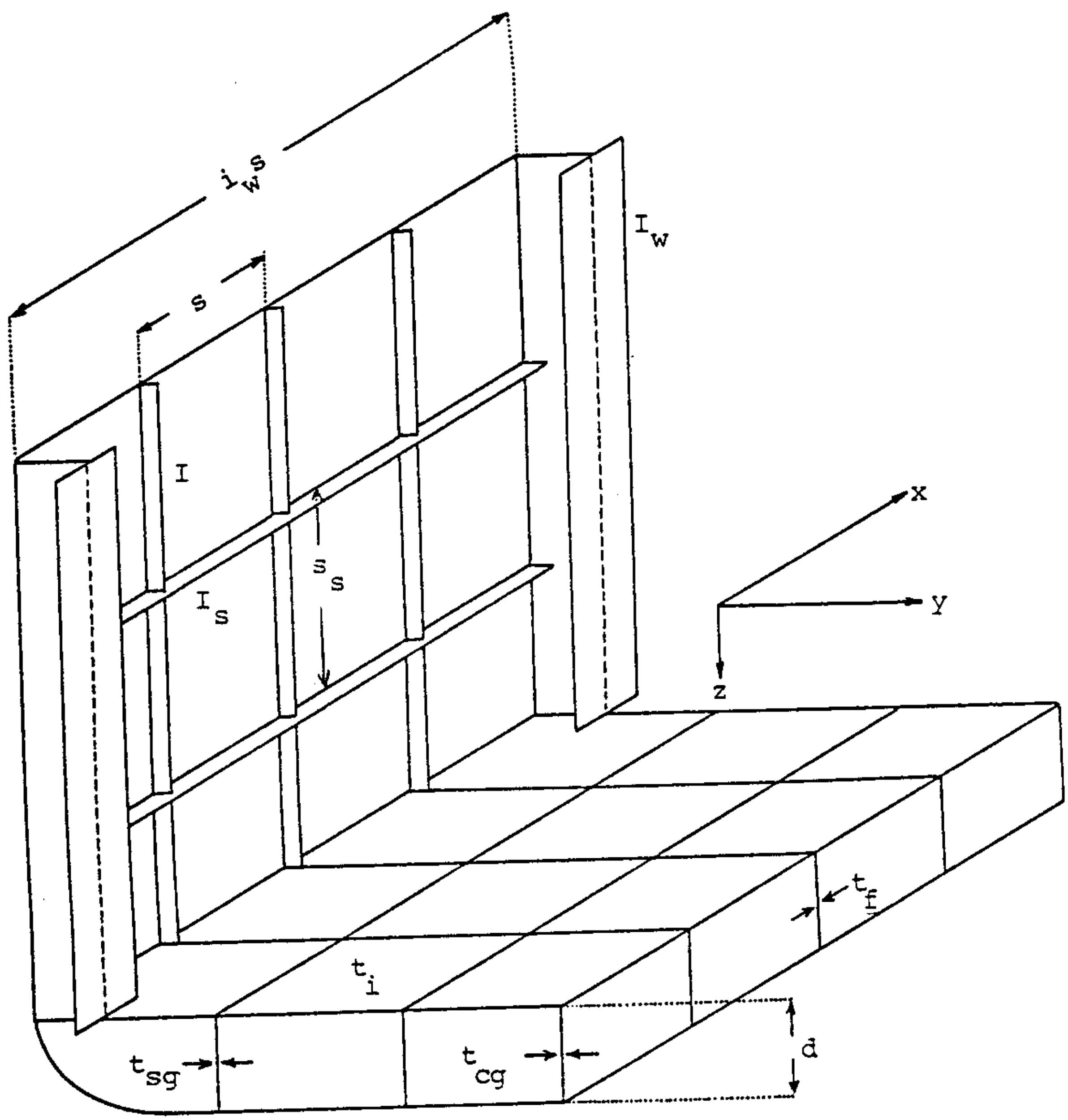


Figure 17

In order to define the stringers, the location and scantlings have to be given. The scantlings include web depth and thickness and flange width and thickness.

Vertical prismatic members are essentially stanchions for which scantlings and location have to be defined.

Plane members include the inner bottom defined by the thickness and stiffening arrangements, intermediate decks or flats and bulkheads. The intermediate decks in the machinery spaces are normally made of orthogonally stiffened panels and they have, in general, large openings. Bulkheads are also, in general, made of orthogonally stiffened panels. Thus, the geometry of intermediate decks and longitudinal bulkheads is not easy to define by a small set of parameters.

The load-related factors in the case of the machinery space include the points of application of large weights, such as the weights of machinery components (turbines, boilers, condensers, reduction gears, etc.) and tanks. These are fixed for a given design and the designer cannot in general modify them. A very important load-related parameter is the height H of the thrust bearing above its foundation base. It is obvious that this height has a minimum permissible value. In some designs, the tank top is penetrated by the gear but there is a limit on how deep this interference can be. As mentioned in Chapter I, the thrust-bearing height above the inner bottom essentially provides a cantilever effect to the applied thrust, which is a very important load acting upon the foundation. As the height increases, the moment transmitted to the foundation becomes larger, and this is one of the major causes for the tilting of the reduction gear and associated failures. Thus, when studying the machinery foundation stiffness, this is a parameter which must certainly be considered.

We can conclude that in order to describe the structural arrangement of the machinery space a very large number of geometrical and structural parameters have to be defined.

The transverse frames are defined in terms of spacing s and moment of inertia I about the x axis (see Fig. 17), assuming they only provide a significant stiffness in the yz plane, and that they are equal and equally spaced. Instead of the moment of inertia, the section modulus could obviously be used. However, since our main concern here is the stiffness rather than the strength, the moment of inertia is the most adequate parameter. If the moment of inertia is fixed, the scantlings can be determined from simple design rules governing proportions, such as web depth/web thickness, as suggested, for example, in [10]. Similarly, the web frames are defined in terms of frame spacings i_W separating them and moment of inertia I_W, and it is again assumed that they are equal and equally spaced. Since the web frame geometry can vary largely in the vertical and longitudinal directions, the moment of inertia can be specified at its lowest span between the inner bottom and the first flat, and essentially at midlength of the engine room space or closer to the reduction gear casing.

As stated earlier, the floors are defined in terms of the number of frame spacings i_f separating them and thickness t_f, and it is also assumed they are uniform and evenly spaced.

The center girder is determined by the depth d and average thickness t_c. The side girders are assumed to be equal and equally spaced between the side and the center girder, so that they can be defined by their number i_{sg} and average thickness t_{sg}. The stringers can also be assumed to be equal and equally spaced by an amount s_s and they are characterized by a certain moment of inertia I_s about the y-axis, since they only provide a significant stiffness in the xz plane (see Fig. 17).

The stanchions can be assumed to be rigid, since their axial stiffness is large and in practice they are designed to preclude the possibility of buckling. In machinery spaces, stanchions are usually used to provide support to large local weights, such as the boiler, or to provide support to deckhouse or superstructure ends. As such they can serve as vehicles to transmit to the foundation large concentrated loads which can induce important deflections. Thus, they should not be neglected when carrying the structural analysis of the machinery spaces. For convenience, they can be taken as rigid struts acting at well-defined locations, and we can assume here that the designer has no freedom in changing their number.

The inner bottom can be defined by an average thickness t_i. In reality, the inner bottom is also a fairly complex structure if all the structural details and stiffening members are taken into account. The problem is simplified here by defining it only in terms of the thickness t_i, and assuming that the stiffening members such as beams and longitudinals can be associated to frames and bottom girders, respectively.

Intermediate decks and bulkheads cannot be treated in detail in any simple mode. The intermediate decks essentially provide lateral support to the side shell, so that the important factors are the number and location in the vertical direction, say height above the inner bottom. Similarly, the longitudinal bulkheads provide support to the bottom shell and decks and as such can be characterized in terms of number and location in the horizontal direction, say distance to the centerline. Transverse bulkheads need not be considered here since, in general, they are only used at the forward and after ends of the machinery space. It should be noted that, in general, the designer does not have very much freedom in choosing the number and locations of bulkheads or intermediate decks, since these are fixed by other considerations such as general arrangement, subdivision requirements, or machinery space requirements. Thus, it can be assumed, when studying the machinery space structural arrangement, that these are given. The designer should then make sure that they are properly stiffened so that they can provide the necessary support to the structure.

If this simplified model of the structural arrangement of the machinery space is adopted, then the foundation stiffness can be expressed as a function of a well-defined number of parameters, as follows:

Stiffness = function (frames, floors, web frames, center girder, side girder, stringers, inner bottom, thrust bearing above inner bottom)

$$= \text{function } (s, I, i_f, t_f, i_w, I_w, d, t_{cg}, i_{sg}, t_{sg}, s_s, I_s, t_i, H)$$

Thus, the stiffness is essentially a function of the fourteen parameters listed above. While the first thirteen can be to a large extent controlled by the designer, the last one H is usually determined from considerations other than stiffness or strength, since it essentially depends on the gear general dimensions and geometry.

In Chapter IV, the model just described is used to perform some parametric variations on the structural design of the machinery space of a given ship, in order to extract some useful information on the best way of meeting the recommended stiffness requirements.

1.2 Review Summary

In order to obtain a reasonable description of current design practice concerning the structural arrangement of machinery spaces, the drawings of various ships were studied and compared, and this section summarizes the most relevant findings of this task. Some of the conclusions reached were used to prepare the set of design recommendations given at the end of this Chapter.

The main particulars of the ships studied are given in Table III. The table includes ship type, deadweight, main dimensions (length between perpendiculars, beam, depth and draft), machinery type and installed horsepower. Each ship is identified by the number given in the left hand column of Table III. Ships #1 through #13 are geared steam-turbine powered, and #14 through #21 are diesel powered. Within

TABLE III

SHIP MAIN CHARACTERISTICS

#	Type	Δ (1)	Main Dimensions (2)				Mech. Type	HP
			L	B	D	T		
1	LNG	125,000	897	144	82	36	Steam	43,000
2	LNG	125,000	906	135	85	36	Steam	40,560
3	LNG	125,000	887	141	94	38	Steam	40,000
4	Tanker	269,574	1050	179	89	69	Steam	40,000
5	Tanker	276,424	1063	178	88	69	Steam	38,000
6	Tanker	290,800	1095	188	94	67	Steam	36,000
7	Tanker	265,000	1060	178	86	67	Steam	35,000
8	Tanker	264,197	1050	176	87	67	Steam	34,000
9	Tanker	249,550	1080	170	84	66	Steam	32,000
10	LASH	40,311	797	100	60	38	Steam	32.000
11	Tanker	188,500	915	166	78	59	Steam	28,000
12	Tanker	164,000	864	173	79	57	Steam	26,700
13	Tanker	120,000	850	138	68	52	Steam	26,000
14	Container	29,194	679	106	62	34	Diesel	36,000
15	Ro-Ro	25,000	652	106	67	33	Diesel	26,800
16	Tanker	135,000	833	143	75	56	Diesel	26,100
17	Tanker	81,283	735	133	43	65	Diesel	15,120
18	Tanker	52,068	658	105	58	39	Diesel	15,000
19	Bulk C.	60,000	702	106	58	40	Diesel	14,000
20	Bulk C.	64,657	715	106	43	60	Diesel	12,960
21	Bulk C.	34,400	548	91	49	37	Diesel	12,600

(1) Δ denotes the deadweight for all ships except LNG's where it denotes the capacity in cubic meter.

(2) L,B,D and T are respectively the length between perpendiculars, breadth, depth and draft in ft (rounded).

each one of these two groups, the ships are listed in decreasing order of installed horsepower. Within the diesel-powered ships group, only #18 and #19 have independent reduction-gear units.

The only two ships that are specifically considered in this study are the LASH #10 and the tanker #11. The LASH vessel was already discussed at length in Chapter I, while the tanker is the subject of detailed calculations described in Chapter IV, which serves as an illustration of the method being proposed here. There are several reasons for concentrating our efforts in these two particular vessels. These two ships have considerably different afterbody shapes, the tanker having a full or "U" form and the LASH an open form. Furthermore, the machinery space configuration for the tanker is essentially as shown in Fig. 1a, while for the LASH it is as shown in Fig. 1b. For the LASH, considerable data were available and an interesting case study could be presented. On the other hand, the tanker has not developed any hull-machinery incompatibility problems, and its conventional design can be considered as quite representative of current practice.

For each ship listed in Table III, shafting particulars are given in Table IV. The identifying number in column 1 of Table IV is the same as used in Table III. The main shaft particulars included in Table IV are the total length of shafting (length from reduction gear coupling to after end), the total number of journal bearings, the mean diameter and length of each shaft section, the distance from the point of support for all bearings at centerline from the reduction gear after bearing or diesel enginer coupling. In addition, the span/diameter ratio, (L/D) for the shaft segment closer to the reduction gear (or diesel engine) is also given. As noted in Chapter I, this is an important parameter since it gives an indication about the shaft stiffness close to the reduction gear, and as such can have a strong impact on the magnitude of the reactions at the gear bearings. In the geared vessels examined (#1 through #13, #18 and #19), the L/D ratio lies between 10.21 and 16.84, which is on the low side of the interval 12 minimum to 20-22 maximum given in [29]. In the case of direct-drive diesel engines, the ratio is substantially smaller, ranging from 2.57 to 8.18.

Table V contains some main-machinery space characteristics, including the engine room main dimensions (length, width aft and width forward), the double-bottom depth and the thrust-bearing height above the inner bottom. The engine room dimensions refer to the tank top level. It can be concluded that all the ships for which dimensions are given have an engine room of trapezoidal shape, such as sketched in Fig. 1a, except #10 for which the engine room essentially has a uniform width and a square shape as in Fig. 1b.

TABLE IV

SHAFT CHARACTERISTICS

#	L_T (ft) (1)	N (2)	Shafting Details (3) Code	Mean Diam. (in)	Length (ft)	Bearing Dtls (4) Code	D_1 (ft)	L/D (5)
1	63.50	2	IS	27.00	16.31	SF	37.13	16.50
			LS	27.00	22.00	SA	54.75	
			TS	33.00	16.19			
2	140.00	4	N/A	N/A	N/A	N/A		N/A
3	106.06	4	LSF	25.00	28.61	L	24.78	11.89
			LSA	24.75	31.31	L	45.44	
			T	32.25	38.25	L	71.44	
						SA	95.68	
4	67.79	3	IS	27.96	30.06	L	30.71	13.18
			TS	37.00	27.89	SF	45.88	
						SA	57.38	
5	61.68	3	IS	26.38	27.26	L	25.10	11.42
			TS	34.00	26.25	SF	41.67	
						SA	50.30	
6	68.89	3	IS	27.17	34.47	L	23.68	10.46
			TS	33.46	29.42	SA	44.31	
						SF	57.30	
7	64.79	2	IS	26.01	28.94	L	33.29	15.36
			TS	32.50	29.85	SA	57.08	

(1) L_T = total length of shafting. (2) N = number of journal bearings.
(3) Code for shafting details: L = lineshaft, T = tail shaft, I = intermediate shaft, S = solid, H = hollow, ST = stern tube shaft, F = forward, A = aft.
(4) Code for bearing details: L = line shaft br., SF = stern tube forward br., SA = stern tube aft br., STB = strut br., D_1 = distance to reduction gear after bearing (or diesel engine coupling).
(5) L = distance between reduction gear after bearing (or diesel engine coupling) and closest bearing, and D = corresponding shaft diameter.

TABLE IV (Continued)

SHAFT CHARACTERISTICS

#	L_T (ft) (1)	N (2)	Shafting Details (3) Code	Mean Diam. (in)	Length (ft)	Bearing Dtls (4) Code	D_1 (ft)	L/D (5)
8	62.28	3	IS	25.71	26.02	L	25.00	11.67
			TS	32.48	28.94	SF	39.75	
						SA	52.75	
9	64.25	3	IS	25.79	27.36	L	21.94	10.21
			TS	32.88	27.89	SF	44.94	
						SA	56.21	
10	160.85	5	LSF	21.88	33.13	L	30.71	16.84
			LSA	23.57	31.48	L	49.91	
			ST	28.56	38.24	L	76.13	
			I	29.50	33.65	SA	103.69	
			T	29.50	18.14	STF	149.41	
11	64.20	3	LS	23.75	27.39	L	29.46	14.89
			TS	29.75	28.48	SF	36.96	
						SA	49.96	
12	62.80	2	IS	24.25	26.72	L	28.19	13.95
			TS	32.50	27.08	SA	52.58	
13	72.88	3	IS	23.63	17.85	L	22.69	11.52
			LS	23.63	17.85	L	40.03	
			TS	30.69	28.85	SA	61.42	
14	120.31	6	IS	25.00	37.73	L	7.22	3.47
			LS	25.00	39.93	L	26.58	
			TS	33.00	42.65	L	45.94	
						L	65.30	
						SF	84.66	
						SA	109.59	

(1) - (5) See footnotes in previous page.

TABLE IV (Continued)

SHAFT CHARACTERISTICS

#	L_T (ft) (1)	N (2)	Shafting Details (3)			Bearing Dtls (4)		L/D (5)
			Code	Mean Diam. (in)	Length (ft)	Code	D_1 (ft)	
15	79.48	4	IS	20.28	9.20	L	13.83	8.18
			LS	20.28	35.00	L	33.02	
			TS	27.01	35.27	SF	49.51	
						SA	69.87	
16	43.11	3	IS	22.44	20.47	L	10.25	5.48
			TS	26.77	22.64	SF	25.72	
						SA	34.09	
17	46.18	4	IS	18.70	25.51	L	5.00	3.21
			TS	24.80	20.67	L	20.75	
						SF	30.79	
						SA	38.24	
18	35.42	2	IH	20.00	16.69	SF	22.53	13.52
			TH	26.00	18.73	SA	29.89	
19	38.16	3	IS	19.06	18.80	L	19.31	12.16
			TS	23.54	19.36	SF	25.91	
						SA	34.52	
20	40.08	4	IS	17.72	20.67	L	3.79	2.57
			TS	22.00	19.41	L	16.91	
						SA	25.60	
						SA	32.64	
21	39.73	3	IS	17.72	19.72	L	10.14	6.87
			TS	22.09	20.01	SF	23.71	
						SA	32.81	

(1) - (5) See footnote in previous page.

TABLE V

MACHINERY SPACE CHARACTERISTICS

#	Engine Room Dimensions (ft)			Double Bottom Depth (ft)	Thrust above inner bottom (ft)
	length	width aft	width fwd		
1	100.0	N/A	N/A	13.33	4.33
2	97.5	N/A	N/A	9.44	N/A
3	100.0	16.0	106.0	14.25	4.00
4	121.0	7.2	84.6	14.76	5.97
5	106.3	9.8	82.7	12.52	7.09
6	115.9	8.2	88.6	13.12	9.43
7	105.0	8.0	86.0	13.42aft 9.00fwd	3.50
8	106.3	8.5	69.9	13.34	8.75
9	93.8	23.0	69.2	8.14	9.84
10	73.3	70.5	70.5	8.75	7.00
11	107.5	8.3	90.0	9.00	9.50
12	95.0	11.0	68.0	11.00	6.50
13	93.0	9.7	72.7	13.00aft 8.00fwd	5.33
14	100.4	24.9	85.3	9.18	5.74
15	120.2	6.9	50.0	6.53	5.11
16	102.2	8.9	82.7	9.15	4.92
17	110.2	7.9	65.6	8.55	4.85
18	76.1	8.9	56.7	6.56	5.90
19	71.8	7.5	59.0	6.86	6.07
20	86.6	9.2	63.0	8.05	4.85
21	81.3	8.0	68.9	7.12	5.00

The double-bottom depth for the steam-powered ships varies from 8.14 to 14.76 ft. For the diesel-powered ships, the depth is, in general smaller, varying from 6.50 ft. to 9.18 ft.

The thrust-bearing height above the inner bottom varies from 3.50 ft. to 9.84 ft. for the steam-powered ships. For the direct-drive diesel-engine ships, it varies from 4.85 ft. to 5.75 ft. For the geared diesel-powered ships #18 and #19, it is equal to 5.90 ft. and 6.07 ft. respectively. In those cases where the height is relatively large, the cantilever effect discussed earlier is reduced by having a sloping tank top which ensures a very gradual transition between the inner bottom and the thrust bearings.

The most relevant information regarding the structural arrangement of the machinery spaces of the ships listed in Table III are summarized in Tables VI and VII.

The frame spacing in way of the machinery spaces can be for convenience expressed in terms of the length between perpendiculars L, by computing the ratio between the frame spacing in inches and L in feet and multiplying by 100.

This ratio is given in the third column of Table IV. For vessels #1 through #8, #11 and #12, this ratio lies between 3.15 and 3.47. For #9, it is equal to 2.92; for #10, it is equal to 11.04 and for #13 to 4.24. For all the diesel-powered ships, the ratio lies between 4.02 and 5.75. Thus, we can conclude that for most steam-powered ships the ratio is smaller than 3.50. For one ship out of thirteen, the ratio is slightly larger and equal to 4.24, while for #10 it is much larger and equal to 11.04. This larger value is due to the fact that the LASH vessel has a strong web at every frame in way of the machinery space. It also has closely spaced stringers or side longitudinals implying a predominantly longitudinally framed structure. All diesel-powered ships have a larger ratio than steam-powered ships (excepting #10, and #16 for which the ratio is 4.02, smaller than what it is for #13).

Regarding floors, all ships have one floor at every frame, except #9 for which the floor spacing in terms of frame spacing can vary between 1 and 3 in the machinery space.

The web spacing in terms of frame spacings varies for most ships between 3 and 5. Ship #10 has web frames at each frame.

The stringer spacing for nine ships is smaller than 40 inches, indicating a clear longitudinal framing arrangement. For eight ships, it lies between 60 and 150 inches and for three ships, it is larger than 200 inches indicating, in this case, a predominantly transverse framing.

The number of side girders is given in way of the reduction gear or engine coupling in the last column of Table VI. For all ships, it varies from 4 to 10, except for #10 where it is equal to 20, which is due to the large span this vessel has at the tank top in way of the machinery spaces.

Table VII lists the average thickness of center girder, side girders, floors and inner bottom in way of the reduction gear or engine coupling. The center girder thickness varies for all ships between 0.57 and 1.00 inches. The thickness of side girders varies from 0.55 to 1.46 inches. This is, in fact, a parameter which is difficult to determine since it can vary quite widely, depending among other factors on the distance to the ship's centerline. The average floor thickness for all steam-powered ships varies from 0.63 to 0.79 inches, except for #10 for which it is equal to 0.56 inches and #13 for which it is 0.60 inches. The inner bottom average thickness varies from 0.57 to 1.10 inches for all ships.

Table VIII lists the moment of inertia in $in.^4$ for frames, web frames and stringers. Referring to Fig. 17, the moment of inertia is given about an axis parallel to the x-axis for frames and web frames, and parallel to the y-axis for stringers. This assumes that frames and web frames essentially bend in the yz plane, while stringers essentially bend in the xz plane. In all cases, the scantlings were taken for members as close as possible to the reduction-gear casing, or engine coupling. As stated earlier, the geometry of web frames can be quite complex, and the figures given correspond to the frame scantlings closer to the point of interest mentioned above. For frames and web frames, shell plating with a width equal to the frame spacing was included in the computation of the moment of inertia given in Table VIII. For stringers, no shell plating is included in the moment of inertia. It can be concluded from Table VIII that the frame moment of inertia, for those ships for which it could be computed, lies between 175 and 2,597 in^4. The web frame inertia lies between 8,166 and 128,006 in^4. The stringers' moment of inertia lies between 68 and 20,934 in^4.

2. Recommendations for Machinery Foundation Design

2.1 General Guidelines

The term "foundation" is often limited to structure above the inner bottom which directly serves to support the machinery; however, to be fully effective the foundation must be properly integrated with the structure of the basic hull and become a part of an overall system.

TABLE VI

MACHINERY SPACE STRUCTURAL PARAMETERS

#	Frame spacing (in)	s/L (1)	Frame spacings between		Stringer spacing (in)	Number of side girders
			floors	web frames		
1	30.00	3.34	1	4	60.00	6
2	30.00	3.31	1	N/A	N/A	6
3	30.00	3.38	1	8	228.00	6
4	35.43	3.37	1	3 and 4	27.56	8
5	35.43	3.33	1	4	230.40	10
6	34.44	3.15	1	4	35.40	8
7	36.00	3.40	1	3 and 4	81.48	6
8	35.43	3.37	1	2 to 4	35.40	6
9	31.50	2.92	1 to 3	3	149.52	6
10	88.00	11.04	1	1	36.00	20
11	30.00	3.28	1	3 to 5	90.00	10
12	30.00	3.47	1	4 and 5	105.00	8
13	36.00	4.24	1	3 and 4	78.00	8
14	35.43	5.22	1	3	33.46	4
15	30.71	4.71	1	3	78.72	4
16	33.46	4.02	1	4	32.64	10
17	31.50	4.29	1	4	125.98	4
18	31.50	4.79	1	4	23.62	4
19	31.89	4.54	1	3	23.64	6
20	31.50	4.41	1	4	379.92	6
21	31.50	5.75	1	3	118.08	6

(1) s/L = frame spacing in inches divided by the length between perpendiculars in feet, multiplied by 100.

TABLE VII

MACHINERY SPACE STRUCTURAL PARAMETERS

#	Thickness (in)			
	center girder	side girders	floors	inner bottom
1	1.00	0.75	0.75	0.88
2	0.88	0.63	0.63	0.75
3	0.81	0.75	0.63	0.88
4	0.87	0.71	0.71	0.98
5	0.94	1.46	0.71	0.94
6	0.98	0.98	0.79	1.10
7	0.94	0.75	0.75	0.86
8	0.85	0.69	0.69	0.93
9	0.98	0.98	0.71	0.87
10	0.75	0.56	0.56	0.75
11	0.81	0.63	0.63	0.72
12	0.75	0.75	0.63	0.75
13	0.81	0.81	0.60	0.79
14	No C.G.	1.48	0.55	0.75
15	0.63	0.79	0.63	0.79
16	0.98	0.67	0.59	0.69
17	0.75	0.96	0.71	0.63
18	0.59	0.59	0.59	0.65
19	0.79	0.98	0.71	0.65
20	0.69	0.55	0.71	0.65
21	0.57	1.02	0.79	0.57

TABLE VIII

FRAME, WEB FRAME

AND STRINGER INERTIAS

#	Moment of inertia (in^4)		
	frames	web frames	stringers
1	907	8,166	
2			
3	175	14,698	5,440
4		128,006	
5		75,812	15,851
6			466
7	2,597	55,727	20,934
8		111,026	475
9	1,901	74,856	4,723
10			179
11	426	23,630	3,336
12	1,143	17,516	8,359
13	1,226	45,893	7,712
14		52,990	
15	311	14,692	1,952
16	No	35,399	415
17	930	25,536	2,462
18	No	30,180	366
19	No	17,179	68
20	1,098	17,369	2,308
21	494	11,731	1,578

Foundations must (1) transfer the propeller thrust and torque to the basic hull and (2) maintain proper alignment of the machinery components.

The main-thrust bearing (1) transfers propeller thrust to its foundations and (2) positions the main-shafting system longitudinally.

Propeller thrust consists of two components (1) steady force and (2) periodically varying force caused by the variable wake in which the propeller blades operate.

Generally, maximum resistance of the thrust bearing foundation to deflection by thrust forces, both static and dynamic, is desirable and results in minimum movement at the bull gear. In certain past cases where a longitudinal natural frequency occurred in the operating range near full power, attempts to lower the frequency by softening the thrust foundation caused the vibratory amplitude at the bull gear to increase beyond safe limits for satisfactory operation of the gear teeth and instead required adjustment of one or more of the other variables.

Symmetrical support of the thrust housing, by using its port and starboard horizontal flanges, sometimes called "centerline support", avoids deflection of the housing which may affect the equal loading of all thrust shoes despite the use of equalizing links or spherical support rings. This also eliminates the bottom vertical support whose bolting to the foundation is generally difficult to access.

The thrust force and distance from the bearing to the basic hull form a large moment applied to the structure. Accordingly, shaft rake should be kept as small as possible, consistent with the configuration of the reduction gear and condenser, in order to minimize the moment arm.

Longitudinal girders are provided to carry the thrust force from the bearing to the basic hull. The primary girders are located as close as possible to the thrust bearing and extended fore and aft to spread the force and moment over as much hull structure as possible. Maintain the girder depth to resist deflection due to the bending moment; however, depth must be reduced as the girders pass under the bull gear. To compensate for the loss of stiffness, secondary girders may be provided to serve a dual purpose. When arranged to line up with the outboard side of the reduction-gear case, they support the gear case and there is no loss of depth. Whenever practical, thrust girders should be extended fore and aft between bulkheads, since each bulkhead provides a vertical anchor point.

Provision must be made to transfer thrust force to the secondary girders by as direct a route as possible. This can be accomplished as shown in Fig. 18, which illustrates a well-engineered foundation. This foundation system was installed in several container ships having 32,000 SHP geared turbine machinery in an aft machinery space. The main thrust bearing is located immediately aft of the reduction gear, an arrangement that usually allows better foundation design and is particularly appropriate for high-powered installations.

The configuration of the centerline and two primary thrust girders is shown in the elevation view of Fig. 18. These girders are carried aft about a dozen frames at essentially full depth; however, forward of the thrust bearing the depth is reduced by both the gear and condenser. The secondary girders are similar except for full depth in way of the gear. In addition to distributing thrust forces, the secondary girders must resist deflection by torque reaction forces from the gear case which are upward on one side and downward on the other.

A portion of the total thrust is transmitted to the secondary girders by (1) sloping plates shown in plan "BB" and section "CC" and (2) a 1-1/2 inch thick horizontal plate below the thrust bearing.

The horizontal plate extends (1) forward to include the reduction gear and the aft turbine and condenser foundations, and (2) aft to include the first lineshaft bearing. The plate, therefore, positions these components in the athwartship direction and minimizes the possibility of athwartship misalingment or relative vibration. Finally, the plate is extended to the shell both port and starboard, which in the case of this vessel are relatively near because of converging waterlines, and therefore, additional paths are available to carry thrust to the shell. Tying to the shell is not always possible and depends upon the location of the machinery space and the hull shape at the stern.

Fig. 19 shows a well-developed foundation for a 36,000 SHP unit in a different type of hull. Here, the stern form consists of a broad flat transom and a relatively flat bottom rising toward the stern with the propeller-shaft-bearing strut supported. The machinery space is almost square in plan and the tank top essentially a large flat panel. With this configuration, excitation of the inner bottom by the alternating component of the thrust force should be avoided by placing the main thrust bearing in way of, or near, a transverse bulkhead.

WELL-ENGINEERED FOUNDATION

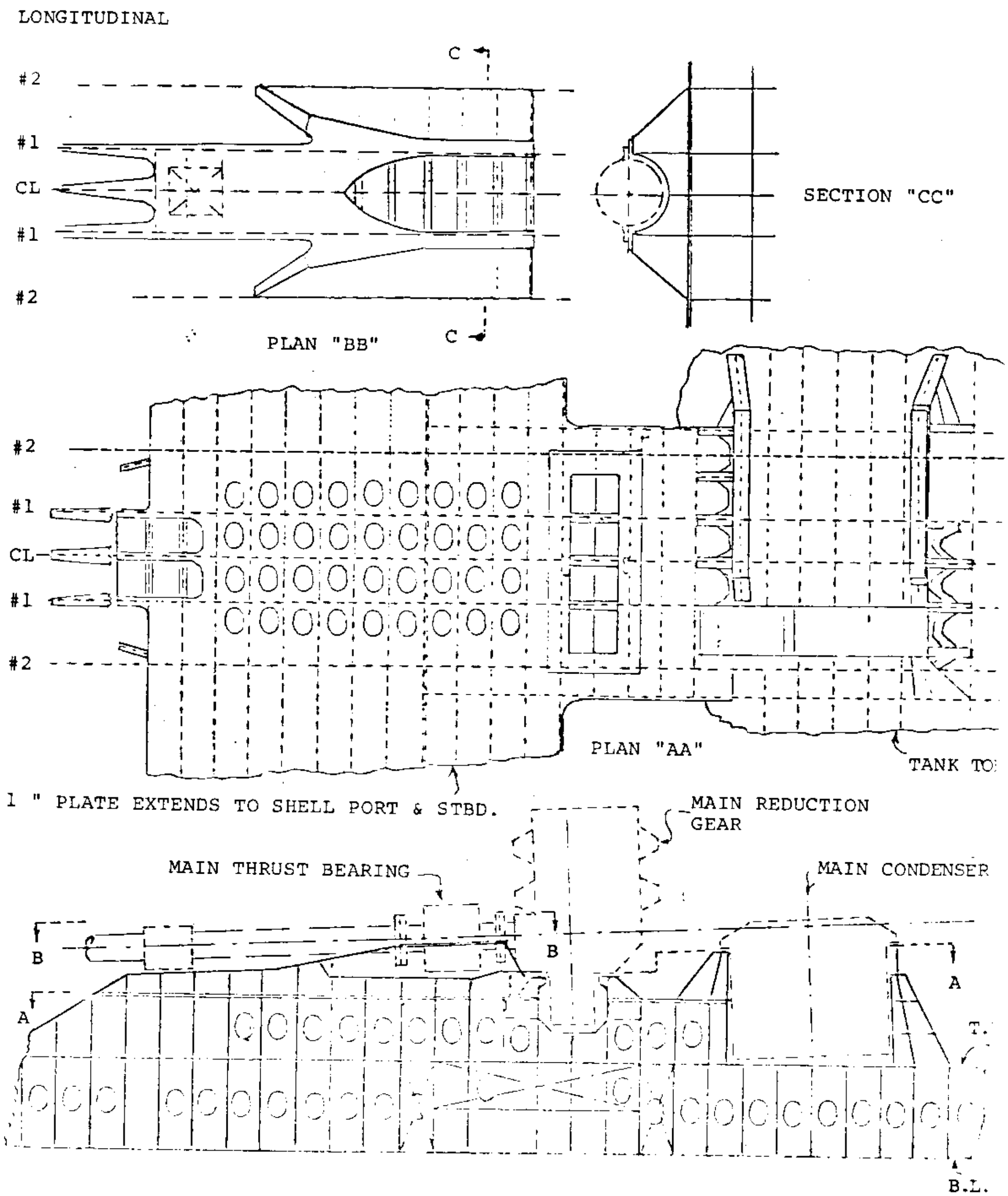


Figure 18

MAIN THRUST-BEARING FOUNDATION

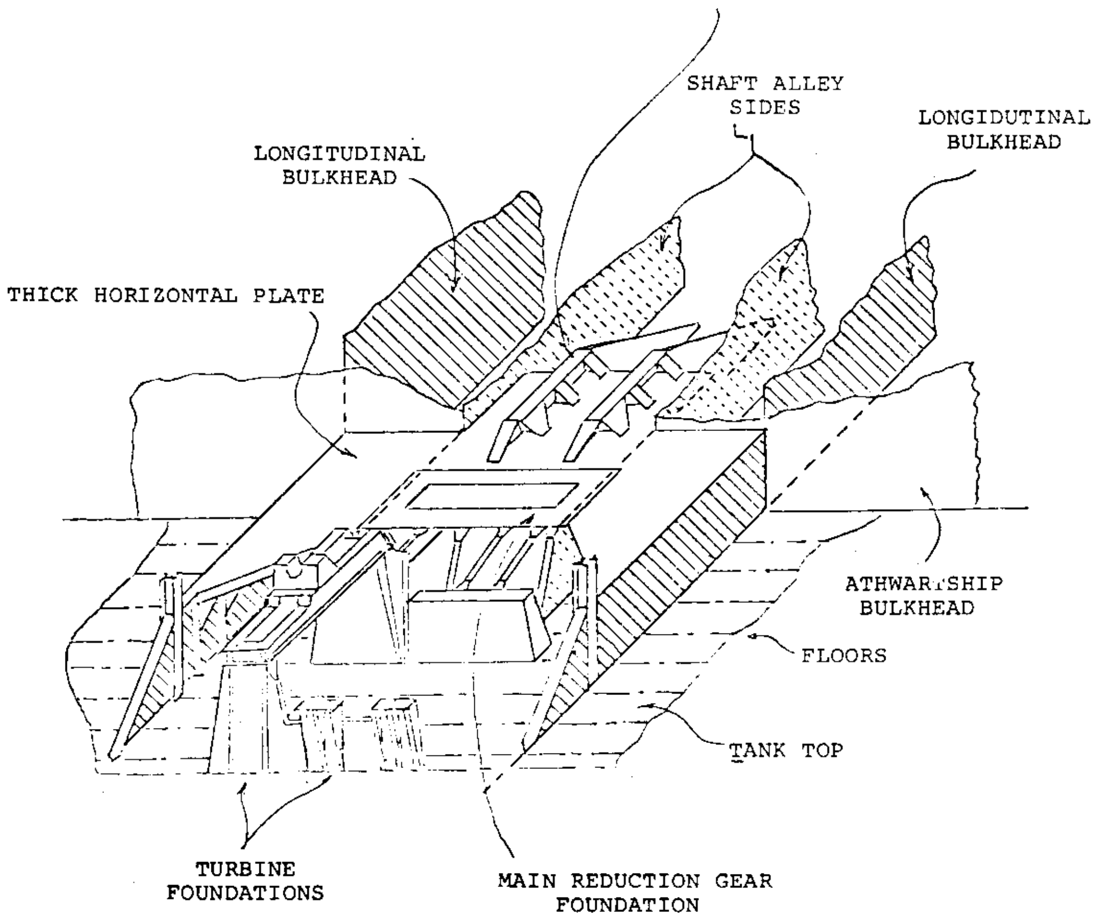


Figure 19

Fig. 19 illustrates excellent use of the ship's basic structure. The thrust bearing is located in a recess of the machinery space at the aft bulkhead. Seven thrust girders are used in conjunction with a thick horizontal plate. The shaft alley sides have been extended forward to carry the outboard sides of the gear. In addition, the horizontal plate is tied at its outboard edges to extensions of the deep longitudinal bulkheads.

Utilization of the stiffness of the shaft alley has been carried out successfully in a number of cases, either by placing the thrust bearing in the shaft alley, or by extending the sides forward as necessary.

Turbines are generally supported at their aft ends by extensions of the gear foundation. The forward turbine supports must maintain the position of the turbine in the vertical and athwartships directions and also provide flexibility in the fore and aft direction to allow for expansion of the turbine casings at increased operating temperatures. To maintain athwartship position, the forward turbine foundation should extend at least between the two secondary-thrust girders, or their extensions in the inner bottom. A separate support structure for the high-pressure turbine having a large height/athwartship width ratio should be avoided. Such a tall, narrow structure is sensitive to changes in the slope of the hull at its base and the movement is magnified by the height so that significant displacement of the turbine may occur with resulting misalignment at the turbine/gear flexible coupling. Several cases have been recorded where coupling or first-reduction gear problems have been due to deflections of the basic hull in an athwartship plane caused by changes in the drafts of the vessels.

Finally, consideration must be given to longitudinal bending of the basic hull and its effect upon alignment of turbine/gear, gear/line shaft, and line shaft bearings.

2.2 Classification Society Rules

An important source of design recommendations concerning the structural arrangement of machinery spaces is provided by the Rules of the Classification Societies, since they are, in general, based on extensive past experience. For this reason, various Rules have been reviewed, and the most relevant aspects of this review are summarized in Appendix A.

The following Classification Societies are included in this study:

(a) American Bureau of Shipping [34]
(b) Lloyd's Register of Shipping [35]
(c) Det norske Veritas [36]
(d) Bureau Veritas [37]
(e) Germanischer Lloyd [38]

The design recommendations provided by the Rules constitute part of the source material on which the summary that follows is based.

2.3 Summary

The following is a brief summary of the most important points regarding the structural arrangement of machinery spaces. They are based on the general guidelines of Section 2.1, on the review of ship drawings summarized in 1.2, on the recommendations included in the rules of the Classification Societies [34-40] and the literature [41-46].

a. It is very important that structural continuity be ensured. Particular attention should be given to the continuity of the double bottom in its connections to the structure forward and aft of the machinery space, and in way of the recess necessary for the installation of the main gear wheel.

b. The double bottom should provide enough stiffness (so that the proposed deflection limits for foundations of geared turbine propulsion units summarized on page 104 are satisfied) and be as deep as possible. If it is deeper than in other compartments, the transition should take place gradually beyond the aft and forward bulkheads of the machinery space, and any abrupt discontinuity should be avoided.

c. The thrust bearing foundations should act as much as possible in shear and tension, rather than bending and compression. The shaft rake should be kept small in order to reduce a "cantilever" type of effect.

d. Thrust girders should be extended fore and aft, if possible a few frame spacings beyond the transverse bulkheads which limit the machinery spaces.

e. Advantage should be taken of machinery casings and shaft tunnels in order to increase the stiffness of the structure in way of the machinery space.

f. A thick horizontal plate below the thrust bearing should extend as much as possible in the fore and aft and athwartship direction to distribute thrust forces over a wide portion of the hull structure.

g. Solid floor plates should be fitted along the line shafting and in the machinery space at every frame, and access holes should be kept to a minimum if the double bottom is transversely framed. If it is longitudinally framed, solid plate floors should be fitted at every frame at least under the main engine.

h. A symmetrical support at the horizontal centerline of the thrust housing should be adopted.

i. Particular attention should be given to the design of machinery spaces located in the extreme afterbody, in which the hull stiffness is decreased and part of the structure may be overhanging.

j. The main-engine seatings should, in general, be integral with the double-bottom structure. In way of the engine foundation, the inner bottom thickness should be substantially increased.

k. Advantage should be taken of strong longitudinal bulkheads which can constitute the walls of deep-tanks in the forward part of the machinery space.

l. Special attention should be given to the structural design of machinery spaces when the width at the tank top level is large, and the shape of the space at that level is rectangular or even square. The decreased stiffness due to the large span of transverse members must be taken into account when determining the scantlings and the structural arrangement, particularly in way of the reduction-gear casing.

CHAPTER III SURVEY OF MAJOR U.S. AND FOREIGN MANUFACTURERS

1. Geared-Turbine Propulsion Machinery

1.1 General

Problems arising from the incompatibility of hull flexibility and machinery support requirements have been associated with large vessels having power plants rated 30,000 SHP or more. The major manufacturers of main propulsion equipment suitable for such vessels were contacted to determine their requirements for support of the machinery. In recent years, only three companies in this country have supplied large geared-turbine units.

Contact with each manufacturer was initiated by a letter which outlined the background and purposes of the project. It was suggested that in view of certain unfortunate past experiences manufacturers in the future will want to scrutinize more carefully the environment in which their equipment must function.

Because of the complex hull and foundation structures, in the past this could only be done qualitatively by visual examination of the design and by comparison with previous successful applications using experience and judgement. Although this procedure worked fairly well for many years, it had become inadequate as vessel size grew and economic pressures increased to minimize hull weight. With the aid of computers and more sophisticated analytical methods, it is now possible to supplement so-called "eyeball engineering" with quantitative measures of structural response and thus determine if the flexibility of the proposed hull design is compatible with limits established for the deflections of machinery supports.

The manufacturers were requested to:

(a) define the critical support points for their standard machinery designs, and

(b) indicate the corresponding allowable deflection limits.

Four specific areas were suggested for consideration:

(a) connections between the prime mover and the reduction gear,

(b) internals of the reduction gear,

(c) main shaft connection to the gear, and

(d) main thrust bearing.

Because of the relatively complex nature of the subject, it was felt that direct discussion, in lieu of questionaires, would be desireable and, therefore, meetings were held with the engineering representatives of the manufacturers. Supplementary data were obtained by correspondence and by telephone conversations. A review of the information that was received indicated good general agreement between the three manufacturers and a summary follows.

1.2 Critical Support Points

Fig. 20 illustrates the location of critical points. The table below the figure indicates for each point the directions in which deflections of the supports are significant.

1.3 Connections Between Prime Mover and Gear

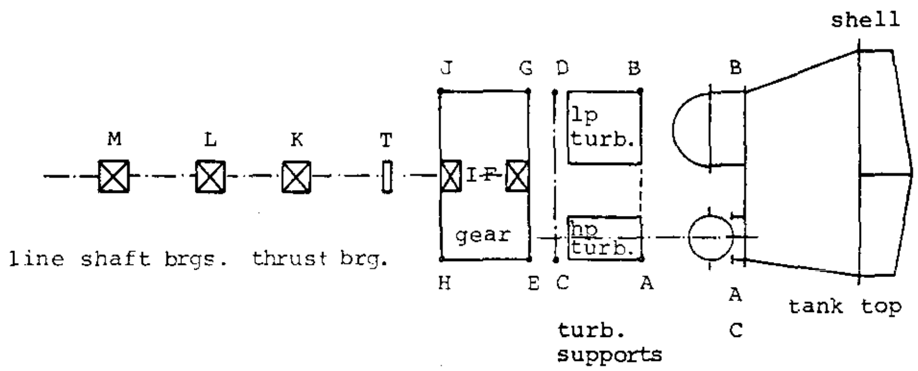
Flexible couplings are provided between the turbine rotors and their drive pinions to accomodate relative longitudinal, parallel, and angular movements of these components. Such movements may be caused by thermal growths in the machinery, or by deflections of the supports. Until recently, marine couplings have been of the dental type, each coupling consisting of two gear tooth elements separated by a length of shafting or a sleeve, as shown in simple form in Fig. 21. Each element includes two meshings rings, one having male teeth, the other female teeth.

Gas turbines and recently some steam turbines use another type of coupling in which the teeth of each element are replaced by a flexible diaphragm. Both types may be treated in a similar manner by considering the maximum allowable angular misalignment of each individual element.

Misalignment occurs when the axes of the turbine rotor and pinion are (a) offset but still parallel, and (b) no longer parallel. Angular misalignment of the individual gear tooth elements is found in each case.

The turbines are usually positioned relative to the gear during installation so that under normal, steady-speed operating conditions misalignment of the turbine and pinion axes will be minimized. Changes in operating conditions and deflections of the machinery support points cause misalignment. The amount of misalignment that can be accepted is dependent upon many design factors such as coupling size, torque, rotational speed, tooth design, lubrication, hardness and finish of the teeth, and relative sliding velocity. When a coupling element operates with misalignment, each pair of meshing teeth slide back and forth longitudinally a small amount proportional to the degree of angular misalignment present in the element. The sliding motion is harmonic at a frequency equal to the rotational speed.

LOCATION OF CRITICAL SUPPORT POINTS



Direction of significant deflections:

Point	Vertical	Athwartship	Longitudinal	Rotational
A	*	*		
B	*	*		
C	*	*		
D	*	*		
E	*	*		
F	*	*		
G	*	*		
H	*	*		
I	*	*		
J	*	*		
K	*	*		
L, etc.	*	*		
T	*	*	*(a)	*(b)

NOTES: (a) important if the arrangement of the foundation permits a rotational deflection of the thrust bearing housing upon application of thrust force.

(b) rotation of the thrust housing in a vertical plane passing through the ship's centerline may cause the thrust shoes to become unequally loaded and introduce a bending moment in the shafting which will affect the distribution of load on the low-speed gear bearings.

(c) uniform nonvibratory longitudinal movement of the gear foundation (points E through J) does not affect the gear internals if within reasonable limits.

(d) the forward turbine support is assumed to be the customary plate structure, sufficiently rigid by reason of vertical depth and stiffening that internal shear deflection may be ignored. Similar assumptions are made with respect to the vertical plates forming the athwartship and longitudinal sides of the reduction gear foundation.

Figure 20

Research and experience has shown that failures of the tooth surfaces tend to occur when the maximum sliding velocity exceeds approximately five inches per second [47]. This criterion is not sharply defined and, therefore, to provide for a factor of safety and account for deflection and thermal growths within the machinery that cannot be avoided, some reduction is required.

Fig. 22 is a diagrammatic representation of a double-element coupling where the turbine rotor and pinion axes are parallel but offset by an amount "m", the maximum allowable parallel displacement. For harmonic motion:

$$m = \frac{60\ v_m L}{RPM\ D}$$

where v_m = maximum allowable sliding velocity assigned to misalignment caused by deflection at the supports

L = Spacing between the coupling elements

RPM = rotational speed

D = pitch diameter of coupling teeth

When "m" is evaluated for typical propulsion units of the size and type under consideration, a value of 0.010 inches is found to be representative. It is convenient to use the special case of parallel displacement for specification purposes because of its simplicity.

In the general case where the turbine and pinion axes are no longer parallel, various alternatives are possible as shown in Fig. 23. The maximum allowable angular misalignment "Θ" of an individual element should be obtained from the manufacturer, or, assuming a properly designed coupling, may be found from the special case as follows:

$$\Theta = \tan^{-1} \frac{m}{L}$$

In each alternative, the centerline of the spacer has been extended and the range of allowable angular misalignment of the turbine rotor and pinion centerlines is "$\pm\ \Theta$".

Fig. 24 represents a typical elevation of turbines, reduction gears, and foundations. For illustrative purposes, assume that the basic hull structure and tank top "XY" bends to a curve "X'Y'" causing deflections "c", "d", "e", and "f" at points "C", "D", "E", and "F". These deflections will cause movements of both turbines and gears which may be assumed as rigid bodies. If the geometry is known, the angular misalignment at each coupling may be calculated and compared with the maximum allowable value "Θ" in order to determine if the deflections cause the coupling limits to be exceeded.

TYPICAL ARRANGEMENT OF DENTAL TYPE FLEXIBLE COUPLING

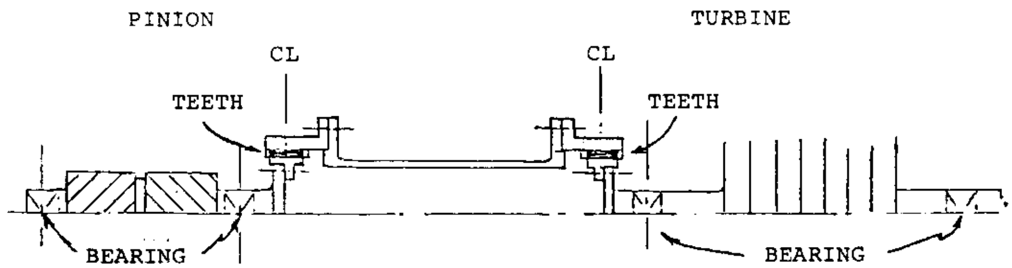


Figure 21

DIAGRAMATIC REPRESENTATION OF A DOUBLE-ELEMENT COUPLING

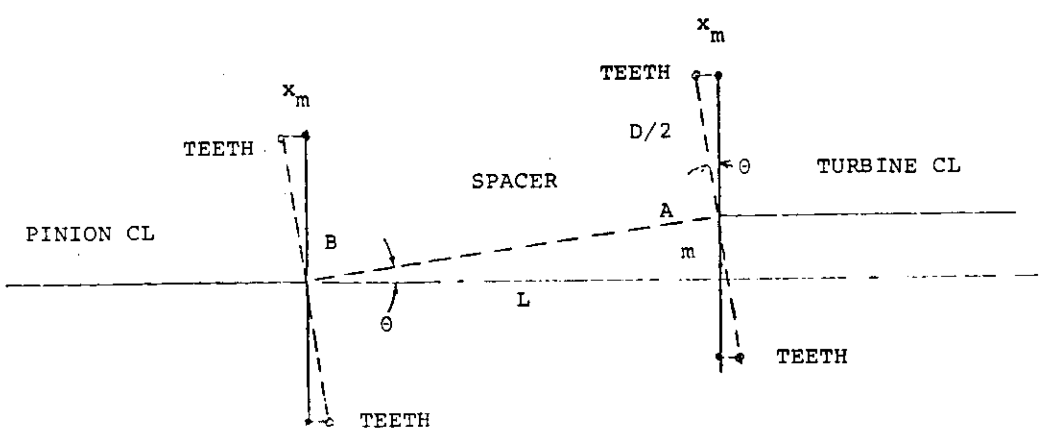
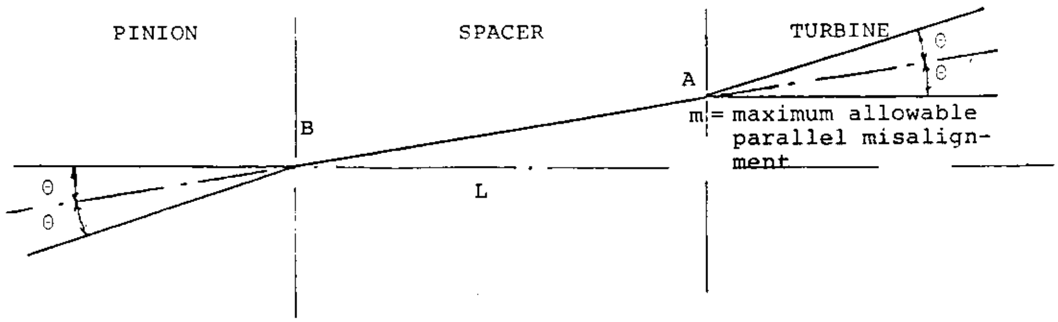


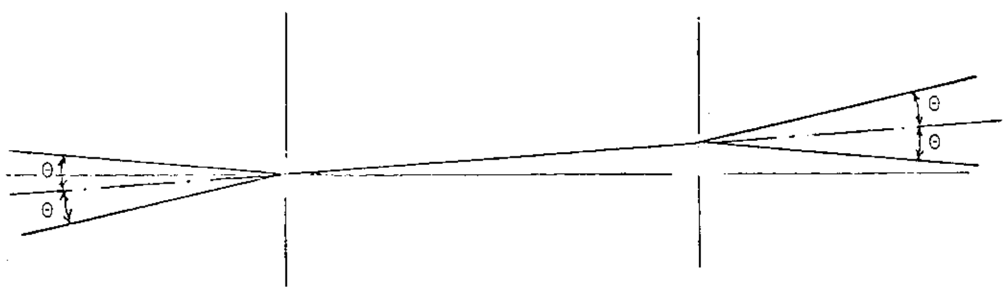
Figure 22

CASES WHERE THE TURBINE AND PINION AXES ARE NOT PARALLEL

DEFLECTION AT "A" RELATIVE TO "B" EQUAL TO "m"



DEFLECTION AT "A" RELATIVE TO "B" LESS THAN "m"



DEFLECTION AT "A" RELATIVE TO "B" GREATER THAN "m"

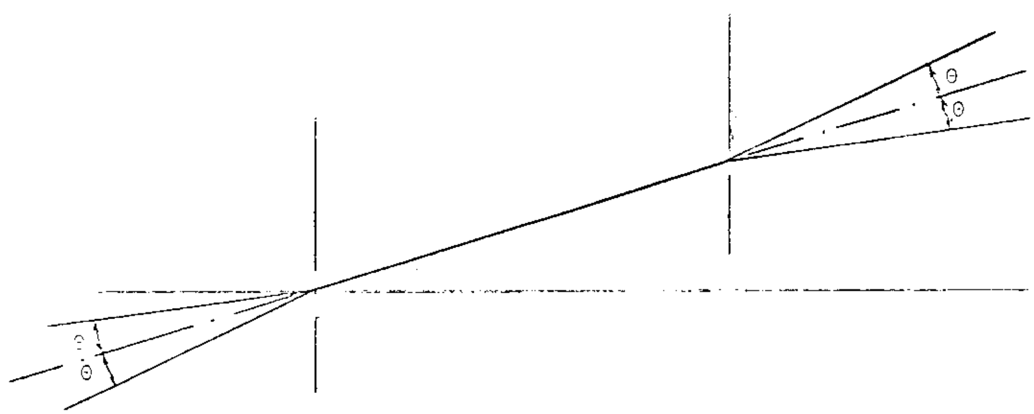


Figure 23

TYPICAL ELEVATION OF TURBINES, REDUCTION GEARS AND FOUNDATION

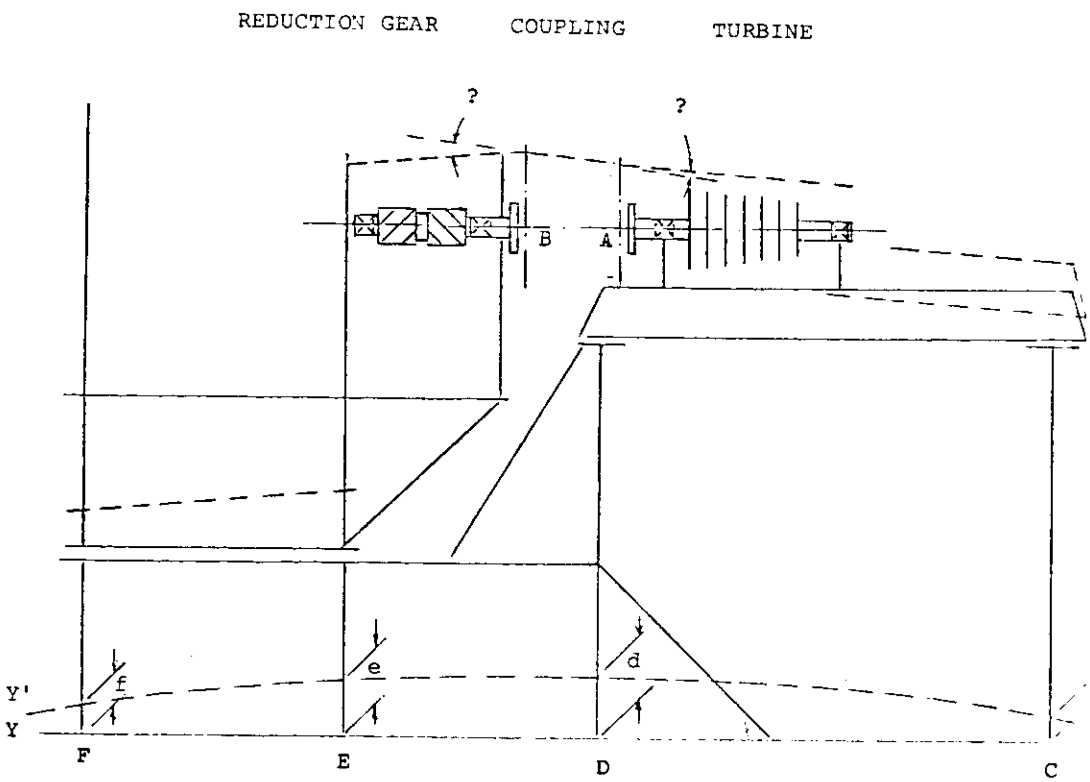


Figure 24

1.4 Internal Alignment of Gears

Most large reduction gears manufactured in this country depend upon the supporting structure to maintain the shape of the gear case and thus preserve the proper alignment of its internal elements. It is impractical to construct a gear case with sufficient stiffness to successfully resist, or prevent, movements of the foundation.

During manufacture, the seating surfaces of the lower case are machined and maintained in a plane while the internal elements are installed in proper alingnment. When the unit is placed on shipboard, it is temporarily supported on jack screws and adjusted to restore it to the factory shape. In addition, tooth contact checks may be made to assure that uniform distribution of load will exist across each mesh under operating conditons. Minor adjustments of a few mils may be made by raising, or lowering, portions of the case. Finally, chocks are fitted between the lower case and foundation, usually at least at the four corners of the gear case and in way of the bull-gear bearings. Thereafter, any deflections of the foundation will be transmitted directly to the gear case and may cause internal misalignment.

Misalignment may consist of either one, or a combination, of two components. "Difference in center distance" occurs when both pinion and gear axes fall in a plane but the axes are no longer parallel. The result is a variation in depth of teeth engagement along the length of the mesh. "Out of plane" occurs when the axes no longer fall in a common plane but when viewed normal to the original plane appear to be parallel. The latter component causes the greater increase in load concentration at the teeth.

Reference [26] published about twenty years ago, stated that the mismatch, or opening, between teeth should not exceed 0.0002 inches per foot of face width. It appears in the light of present knowledge that this limit is conservative and can be increased to as much as 0.0006 inches per foot.

To relate this limit to allowable deflection at the foundation, it is necessary to consider the position of each pinion. For example, if the pinion and gear axes fall in a horizontal plane, then vertical deflection at a corner of the foundation will cause maximum "out of plane" misalignment. In this case, if there is approximately five feet between the second-reduction bearings, the equivalent vertical deflection at a corner of the gear case would be about 0.003 inches. In large gears of the locked-train type, there are four pinions located in various positions intermediate between horizontal and vertical planes passing through the gear axis and, therefore, both components of misalignment are involved. The analysis necessary to establish limits in each case can best be performed by the gear designer, [48]; however, because of the similarity of most modern designs, generalized limitations are possible and are proposed in a following section.

1.5 Main Shaft Connection to the Gear

The bull gear is supported by two large bearings, one forward and one aft. Each journal must be slightly smaller in diameter than its bearing bore in order to provide for lubrication. The operating position of the journals within the clearances are determined by bearing size, oil clearance, rotational speed, and oil viscosity, which in each case is the same for both bearings, and by the magnitude and direction of load on the journals which may differ.

If the design of the bull gear and its shaft were symmetrical and the main shaft disconnected, each bull-gear bearing would be loaded in the same manner. The gear shaft, however, is not symmetrical since it must be extended aft and a flange provided for connection to the main system. The lineshaft, when connected, imposes a direct weight load and introduces a moment at the flange. In this case, the forward and aft bearings are not loaded in the same manner and their journals operate in different positions within the bearing bores. This angular shift of the gear axis with no corresponding movement of the pinions causes misalignment at the meshes and, depending upon the magnitude, danger of tooth wear and failure.

Maximum flexibility in the lineshaft is necessary to minimize effects upon the bull gear. The first lineshaft bearing should be located as far aft of the gear as possible consistent with satisfactory lateral vibration characteristics and bearing loading. In order to achieve additional flexibility, in some cases it may be desirable to select higher strength steel for the line shaft and to reduce its diameter.

"Allowable setting error (ASE)" may be used as a measure of shafting flexibility [29] and is defined as follows:

$$\pm\ ASE = \frac{\Delta R}{I_{11} - I_{22}}$$

where ΔR = allowable difference between two slow-speed gear bearing vertical static reactions

I_{11} = reaction influence number of forward slow-speed gear bearing on itself

I_{22} = reaction influence number of aft slow-speed gear bearing on itself

ASE represents the amount of vertical parallel movement that the gear may be raised, or lowered, relative to the line-shaft bearings without exceeding ΔR. ΔR is established by the manufacturer and may vary from approximately 10,000 - 20,000 pounds for large locked-train units. Reference [29] recommends an absolute minimum ASE value of $\pm$ 0.010 inches, but this is based on experience with smaller vessels with greater hull rigidity. Recent experience indicates that ASE values of 0.025 inches are practical and values less than 0.020 inches should be avoided, particularly in larger hulls. ASE may be used for comparative purposes but the acceptability of the shafting design should be based upon a complete evaluation of the differences in both horizontal and vertical static reactions at the forward and aft gear bearings relative to limits set by the manufacturer.

1.6 Main Thrust Bearing

The arrangement, location, and other consideration associated with the main thrust bearing are dealt with in Chapter II. No additional specifics were developed during the survey.

1.7 Criteria

A review of the information gathered during the survey has led to proposed criteria which is for convenience shown on the next page.

1.8 Alternative Types of Geared-Turbine Propulsion Machinery

Practically all large-reduction gears for turbine drives manufactured in this country during recent years have been of the locked-train design; however, several manufacturers commented on other types including several which incorporated features believed to render them less susceptible to the effects of foundation movements. This increased capability to successfully handle deflections of the supports unfortunately is accompanied by disadvantages.

The alternatives cost more to manufacture and, therefore, it is unlikely that a manufacturer would offer a more expensive unit considering the highly competitive nature of the marine propulsion market unless it was specifically required by bidding specifications. Some types involve major rearrangements of machinery spaces requiring more floor area and longer length. There may be some loss of accessibility for maintenance and repair depending upon the machinery space arrangement. Finally, for each manufacturer there may be development costs and risks associated with a new design.

Before considering the use of such alternatives, the flexibility of the proposed hull should be investigated at an early point in the design schedule to determine if the

PROPOSED DEFLECTION LIMITS FOR FOUNDATIONS

OF

GEARED TURBINE PROPULSION UNITS

The main turbines, reduction gears and condensers shall be designed and coupled to each other so that the deflections of supporting ships structure caused by combinations of elastic and thermal effects resulting from seaway, torque and thrust reactions, cargo and ballast loading, etc., under any operating condition shall not produce excessive stresses or wear, provided these deflections do not exceed the following:

1. Vertical movement of 0.004 inches at the seating surface at the forward port (starboard) corner of the reduction gear foundation relative to a plane passing through the following three points in the seating surface -

   (a) the aft port (starboard) corner

   (b) the forward low-speed gear bearing

   (c) the aft low-speed gear bearing

2. Relative movements of line shaft bearings and gear foundation which cause the difference in static vertical reactions at the bull-gear bearings to exceed the maximum allowable value (ΔR) established by gear manufacturer.

. Relative movements of line-shaft bearings and gear foundation which cause the difference in static horizontal reactions at the bull gear bearings to exceed the maximum allowable value established by the gear manufacturer.

4. Parallel displacement of 0.010 inches, or equivalent angular misalignment, of a turbine rotor to pinion.

movements of the machinery supports under normal operating conditions fall within allowable limits. Should these limits be exceeded, consideration may be given to additional stiffening in the foundations or basic hull structure. In the event that this is impractical, or uneconomical, to provide sufficient stiffness, then the use of alternative machinery types may be warranted and, if so, should be specified after consulting with all potential machinery suppliers.

The articulated double-reduction gear design permits a greater variety of arrangements with essentially the same rotating parts by rolling the pinion and gear centers to various positions and may be classified by the number of horizontal planes in which the pinion and gear axes fall. The "three plane unit" [(1) turbines and high-speed pinions, (2) high-speed gears and low-speed pinions, (3) low-speed gear] is the most common [27]. The gear case is a common structure serving both first and second reductions. The lower gear case usually is chocked and bolted to the foundation at the corners, athwartship wall between first and second reductions, low-speed bearings and often at other points. Deflections of the foundations are transmitted directly to the gear case and may cause misalignment of the internals.

"Singles" and "two plane" arrangements have been built with separate and independent gear cases for the first and second reductions. Flexible couplings between the two reductions allow for relative movements between the casings. The low-speed pinion and low-speed gear axes fall in the same horizontal plane. Such an arrangement is less sensitive to forces and moments introduced by the line shafting. Misalignment of the pinion and gear axes in the horizontal plane (in-plane misalignment) causes a variation in depth of tooth engagement and is less disturbing to the uniformity of tooth contact across the face of each helix. Horizontal misalignment may be caused by differences in horizontal forces and by differences in vertical forces on the low-speed gear bearings.

"Out-of-plane" misalignment is more serious because it results in a greater load concentration at the mesh. It can be eliminated, or minimized, when caused by foundation movements through the use of "three point chocking". Consider each half (port and starboard) of the lower gear case and the corresponding mesh between pinion and gear. Provide chocks at the midpoint of the longitudinal side and in way of the two low-speed bearings. These three support points will always remain in a plane regardless of deflection at any, or all, of these points, and, therefore the alignment of pinion and gear will not be affected. This arrangement leaves the corner of the gear case unsupported and requires careful attention to the design of the case. One European manufacturer has found it desirable to place springs at the corners. In the case of single-plane gears having separate first and second-reduction gear cases, it is possible to treat the first-reduction gear cases in a similar manner.

Geared-turbine propulsion units for naval service have been built on platforms which provide sufficient stiffness and support for the components. The platform is floated on springs, or isolation mounts, which essentially eliminate any effects of deflections at the support points. The principal purpose, however, is to reduce noise transmission to the hull and thus avoid detection by sonar. This arrangement does not appear to be practical for application to large merchant type units. It is costly, not only because of the additional structures but also because the drive and all piping connections must be flexible. Further, the output torque of a large merchant unit is from two to ten times greater than naval units where this arrangement has been applied.

A naval unit that is platform mounted and spring supported requires a flexible coupling between the gear and line shaft to accomodate relative movements. Merchant type units are usually solidly bolted to the lineshaft; however, consideration is currently being given to large flexible couplings designed to handle the much greater torque found with high-powered merchant units. The use of flexible couplings would eliminate the introduction of bending moments at the gear shaft flange leaving only weight loads to be considered.

It is interesting to note that in the early history of marine gears when the current high-accuracy gear cutter were not available, various devices were adopted in attempts to compensate. One arrangement involved a "floating pinion" with bearings that were spring supported. This device, and others of a similar nature, became obsolete as gear cutting became more accurate; however, it should be remembered that at that time ships were smaller and foundations were more rigid.

2. Diesel Engines

2.1 General

Diesel marine propulsion systems are receiving now a great deal of attention. As mentioned in Chapter I in the case of ships equipped with diesel propulsion plants, problems of hull-machinery incompatibility have also been found in the past. With the decrease in power of diesel engines, the dimensions of the webs, crankpins and journals of crankshafts have increased considerably, and as a result the overall crankshaft stiffness has increased. This fact, coupled with an increase in flexibility of hull girder, double bottom and bedplates of propulsion plants, has been the cause of crankshaft and bearing damage.

In an extensive experimental study described in [49] and involving 32 ships, it was found that deformations of the bedplates due to cargo and the effect of sea loads were quite noticeable. The amplitudes measured were of the order of 0.3 to 2 mm in the case of cargo loads, and of the order of 1 to 7 mm in the case of sea loads. These amplitudes were measured relative to the bedplate deformed profile in the light condition and in calm water. This indicates that the vertical positions of the crankshaft bearings vary considerably, according to the loading conditions and sea state. Thermal effects, such as temperature differentials between the cylinder block and its supporting base, can also be responsible for considerable bedplate deformation, and these can induce considerable loads even on the engine structure itself.

The study of the rigidity of the crankshaft and its components is difficult. In [50], a conventional strength of materials approach is adopted, which leads to the distribution of vertical and horizontal reactions in way of each bearing, to the influence coefficients of each bearing of the crankshaft assembly, and to the openings/closings of the webs. In carrying a structural analysis of machinery components, a more exact definition of the structure's geometry can be obtained by using the finite-element method, as suggested in [51].

It was found, as discussed in [49] that when the engine is stationary, and the main bearings are lined in a straight line, because of the fact that the different crankshaft sections have different rigidities, the reactions at the main bearings are unequal. When the engine is running, dynamic and thermal effects can even cause the reactions at some bearings to be reversed, so that as a result, the journals may lose contact with the lower shells.

The inherent flexibility of the crankshaft and the related failures, such as the loss of contact between some of its journals and the corresponding bearings, show how important it is to measure crankshaft deflections and check if they fall within reasonable limits. Part of the problems due to excessive flexibility can be reduced if a curved crankshaft alignment procedure is adopted, as suggested in [49].

An important factor in the hull-engine foundation compatibility is the design of the engine structure itself. In conventional diesel-engine design, the structure of the crankcase is essentially composed of columns extending between the bedplate and the cylinder block. This particular arrangement implies a relatively low shear stiffness, and since large two-stroke engines essentially behave as beams in pure shear, the structure is very much affected by the degree of deformation of the hull and engine-support systems. A possible alternative arrangement consists of box-shaped longitudinal girders comprising a deep-section single-walled bedplate and high cast cylinder jackets [8,24]. As a result of the large bending and shear stiffness of box girder construction, it has been possible to decrease substantially the deformations induced by the ship's girder and double bottom, and no abrupt changes in curvature are observed. Thus the crankshaft line bearing is deformed to a smaller degree, increasing the likelihood of a good engine performance. Numerical studies using the finite-element method and model tests have indicated that this type of arrangement can lead to an increase of about 50 to 80 percent in the standard hull foundation structural stiffness, as compared to an increase of approximately 20 to 30 percent in the case of column type of design [24]. Thus, the double-bottom deformation which occurs in normal operation can be absorbed by an engine of this kind. Furthermore, with large two-stroke engines, the maximum engine flexure produced from double bottom deformation rarely exceeds 1mm. Thus, this indicates that with modern diesel-engine designs, if the structural arrangement of the engine spaces is carefully planned, the problem of hull-foundation rigidity compatibility can to a large extent be minimized.

2.2 Manufacturers' Requirements

Engine manufacturers provide shipyards with recommended foundation structural arrangements for given engines, in order to ensure that the reaction forces between bedplate and foundation are properly transmitted.

Typically, these include recommendations on scantlings for top plate, bottom shell plating and longitudinal members, design and arrangement of lube oil service tank, chocks and foundation bolts. The design of the lube oil service tank is important in order to allow a uniform movement of the whole engine due to thermal expansion. It appears that often the scantlings recommendations provided by the manufacturer are too general, since they don't take into account the ship's longitudinal bending or local deflections within the engine spaces. Thus, the naval architect must use his own judgement in designing the structure in way of the engine spaces. The structural design procedures presented in this report (section IV, Proposed Method) could help the designer to form a basis for judgement. The recommended bolting and their bearing arrangements are, on the other hand, usually more specific, and should be followed.

When defining the deformation limits for the engine foundation, the most important deformation mode to be considered is bending in a vertical plane containing the longitudinal axis of the engine. The deflection profile due to the double bottom deformation in way of the engine closely approximates a circular arc, and a general guideline for large two-stroke engines is that between extreme ship load conditions the radius of curvature of this circular arc should not be less than 30×10^3m.

Engine manufacturers have used for many years the measurement of the crankshaft web deflection as the criterion for the alignment condition of the engine. The crankweb deflection is the difference of the distance between two webs when the crankshaft has been rotated through 180°. The measurement can easily be carried out with a special crankshaft alignment gauge or chocking device.

The web deflection leads to important information regarding the crankshaft stress, the bearing load and the quality of the engine chocking. Taking into account these criteria and based on past experience, the admissible values for the web deflection of each engine can be specified. Values are usually given for two different cases: the case in which the engine is still new and the case in which the engine has been in operation for some time, so that some wear of bearings or chocks has taken place. In the first case, the permissible web deflections can be considerably lower. If in any case, the values measured exceed the corresponding maximum permissible levels the cause must obviously be eliminated and the crankshaft must be realigned. Possible causes of excessive deflection are an uneven wear of the crankshaft bearings, a change in the position of the driven shaft or a change in the support conditions for the engine. It is obviously important that each time the crankweb deflections are measured the loading and thermal conditions of the ship be essentially the same.

Given the allowable crankweb deflections, it is possible to determine the overall admissible crankshaft deflections, using either conventional strength of materials approaches [50], simple graphical methods or more sophisticated finite-element methods [51]. If the permissible crankshaft deflections are known, the maximum support structure distortions can also be defined. As indicated in [8], the use of modern computational techniques for the structural analysis of the entire engine now enables the diesel manufacturers to specify qualitative data on the acceptable deformation for engines under various load conditions.

Due to the variety of diesel engines presently available, no general foundation rigidity criteria will be presented here. This should be supplied for each case by the manufacturer. The naval architect can then essentially use the method suggested in this study in order to make sure that these requirements are met. In particular, a good structural arrangement of the engine spaces, following the general guidelines set forth in Chapter II should always be adopted, as a first step towards minimizing the possibility of hull-machinery incompatibility.

CHAPTER IV. PROPOSED METHODOLOGY

1. Methods for Evaluating the Foundation Stiffness

The methods for evaluating the machinery foundation stiffness are for convenience subdivided here into the following two groups:

- stress-hierarchy method (SHM)
- finite-element method (FEM)

In the stress-hierarchy method [19], the structure is represented by different models with an increased degree of refinement, starting with a beam model for the hull girder, and using two and three-dimensional frame models or grillage models in order to represent the transverse framing system and the double bottom in way of the engine room. In the finite-element method, the structure is discretized in detail so that the actual geometry and configuration can be accurately represented. The advantage of the stress-hierarchy method (SHM) is the capability it offers to the ship structural designer to determine at the early design stages the relative merit of the various possible ways of supporting the machinery components at a reasonable computer cost. Once the design has gone into its more detailed phase, then a FEM is needed in order to get a better estimate of resulting deflections. Clearly the results of the SHM are of great help in reducing the number of computer runs for the FEM calculations. The various methods for evaluating the foundation stiffness, as listed in Table IX, will now be discussed.

2. Stress Hierarchy Method

The total engine structural deflection can be separated into the following components:

- deflections due to hull girder deformation
- deflections due to transverse web frame deformation
- deflections due to engine-room double-bottom deformation

2.1 Deflections Due to Hull Girder Deformation

The hull-girder is considered as a free-free beam subjected to laterally and axially applied loads. The lateral loads are due to the buoyancy and hull weight forces. The axial loads are due to the application of the thrust force at the height of the shaft centerline which, in general, does not

TABLE IX

METHODS FOR EVALUATING FOUNDATION STIFFNESS

<u>Stress-Hierarchy Method (SHM)</u>

- Hull Girder Deformations:
    - a. Bernoulli Beam (no shear effects)
    - b. Timoshenko Beam (shear effects included)
- Transverse-Frame Deformations
    - a. 2D Frame Model
- Double Bottom Deformations:
    - a. Beam Model for Floors
    - b. Grillage Model

<u>Finite-Element Method (FEM)</u>

- Coarse Mesh
- Fine Mesh

coincide with the neutral axis of the hull-girder thus introducing externally applied moments. The total hull girder deflection can be separated into the following components:

- deflections due to bending deformation
- deflections due to shear deformation

2.1.1 Hull Girder Deflections Due to Bending Deformation

The distributed load q per unit length due to buoyancy and weight forces is

$$q = b - w \tag{1}$$

where b is the buoyancy force per unit length and w is the ship weight per unit length. Forces are considered positive when directed upwards. The shear force V at a location x is obtained by integrating equation (1)

$$V = \int_0^x (b-w)\,d\xi \tag{2}$$

where ξ is an integration variable and x is the hull-girder neutral axis which is oriented from the aft to the forward end and originates at the aft end. Integrating the shear force distribution, the bending moment M at location x can be obtained as follows:

$$M = \int_0^x \int_0^\zeta (b-2)\,d\xi d\zeta + M^c < x - a >^o \tag{3}$$

where ξ and ζ are integration variables, $< >^o$ is the Dirac delta function, and a is the distance of the applied concentrated moment of magnitude M^c from the aft end.

The positive direction for the shear force V and bending moment M is given in Fig. 25

SIGN CONVENTION

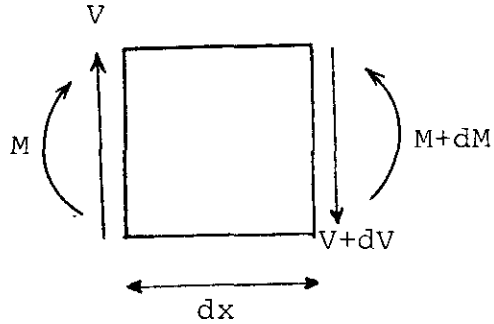


Figure 25

The slope Θ of the deflection curve at location x is

$$\Theta = \int_o^x \frac{M}{EI}\, d\xi - \int_o^L \frac{M}{EI}\, d\xi \qquad (4)$$

where EI is the hull girder flexural rigidity, and L is the length of the hull girder. The positive sense for the slope is in the clockwise direction. The hull girder transverse deflection $(y_B)_{HG}$ due to bending deformation is

$$(y_B)_{HG} = \int_o^x \int_o^\xi \frac{M}{EI}\, d\xi d\xi - \frac{x}{L} \int_o^L \int_o^\xi \frac{M}{EI}\, d\xi d\xi \qquad (5)$$

Deflections are considered positive when directed upwards.

2.1.2. Hull Girder Deflections Due to Shear Deformation

The hull girder transverse deflection $(y_S)_{HG}$ due to shearing deformation is

$$(y_S)_{HG} = f\left\{\int_o^x \frac{V}{GA_w}\, dx - \frac{x}{L} \int_o^L \frac{V}{GA_w}\, dx\right\} \qquad (6)$$

where A_w is the total area of the web (comprising the area of all vertical parts of the shell, longitudinal bulkheads and girders), G is the shear modulus and f is a form factor (equal to 1.2 for rectangular cross-sections).

2.1.3. Calculation Procedure

The total weight w per unit length is essentially given by

$$w = w_{HS} + w_C + w_M \qquad (7)$$

where w_{HS} is the hull steel weight per unit length, w_C is the cargo weight per unit length and w_M is the machinery weight per unit length. For tankers the hull weight distribution depicted in Figure 26 can be adopted.

HULL WEIGHT DISTRIBUTION FOR TANKERS

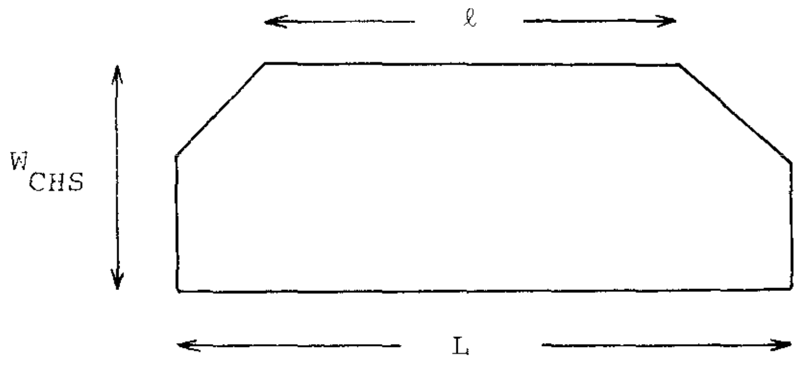


Figure 26

The weight per unit length in the region of the parallel middle body is [52, 53]

$$w_{CHS} = \frac{w_{HS}}{4L}\left(5-\frac{\ell}{L}\right) \tag{8}$$

where ℓ is the length of the parallel middle body. Given the hull steel weight per unit length w_{HS} and the location of the longitudinal center of gravity L_{CG} the determination of the weight distribution can be completed. The area under the weight curve is determined for each loading condition using the trapezoidal rule of integration.

The area under the buoyancy curve is determined by parabolic interpolation between input data points of successive stations. Performing the integrations indicated by equations (2) through (6) the shear force, the bending moment, the slope of the deflection curve and the deflection due to bending and shearing deformations are respectively obtained.

2.2 Deflection Due to Transverse Web Frame Deformation

2.2.1 Symmetric Response

The symmetric response of a transverse web frame due to hydrostatic pressure is sought herein (Fig. 27).

TRANSVERSE WEB FRAME

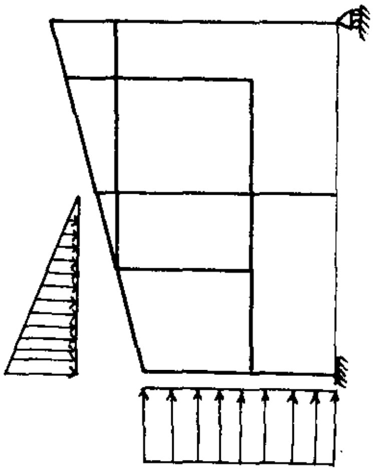


Figure 27

The web frame is symmetrical with respect to the vertical mid-plane. The loading is also symmetrical about the same plane. Therefore, material particles in the plane of symmetry remain in the plane with rotations possible only about the normal to the plane.

The Node Method for plane frames [54] is utilized to determine the required response. The formulation of the node method is based on the following two basic assumptions:

- the plane frame is loaded only at its nodes
- the plane frame has completely "fixed" supports (i.e. at a support both the displacement vector and the rotation are assumed to be zero).

The first assumption requires a method for reducing to nodal loads any externally applied distributed and/or intermediate loads. This is accomplished by decomposing the given loading into the forces required for zero nodal point displacements (fixed-end quantities) and the nodal forces used in the analysis which are negative of the fixed-end quantitites. To obtain the true state of stress in a member having distributed loading, the results obtained from both parts must be superimposed.

The second assumption which requires the use of fixed supports is not a restriction on the range of applicability of the method. Thus, for instance, a frame with hinged support is equivalent to a frame with fixed support with member release.

The nodal displacement vector $\underset{\sim}{\delta}$ and the applied force vector $\underset{\sim}{P}$ is defined for the entire structure:

$$\underset{\sim}{\delta}^T = [\underset{\sim}{\delta}^T{}_1, \underset{\sim}{\delta}^T{}_2, \ldots\ldots \underset{\sim}{\delta}^T{}_J] \text{ and } P^T = [\underset{\sim}{P}^T{}_1, \underset{\sim}{P}^T{}_2, \ldots\ldots \underset{\sim}{P}^T{}_J] \quad (9a,b)$$

$$\text{where } \underset{\sim}{\delta}^T_i = [\delta_{ix}, \delta_{iy}, \Theta_i] \text{ and } \underset{\sim}{P}^T{}_i = [P_{ix}, P_{iy}, M_i] \quad (10a,b)$$

x,y are the global cartesian coordinate axes, δ_{ix}, δ_{iy} are the components of the displacement vector along the x and y axes respectively at node i, Θ_i is the rotation at node i, P_{ix}, P_{iy} are the x and y components of the applied force vector at node i, M_i is the applied moment at node i, the symbol $(\sim)$ denotes a vector quantity, the symbol $(_\sim)^T$ denotes the transpose of a vector quantity and J is the number of movable joints.

The member displacement vector $\underset{\sim}{\Delta}$ and the member force vector F is defined for the entire structure:

$$\underset{\sim}{\Delta}^T = [\underset{\sim}{\Delta}^T, \underset{\sim}{\Delta}^T, \ldots\ldots, \underset{\sim}{\Delta}^T{}_B] \text{ and}$$

$$\underset{\sim}{F}^T = [\underset{\sim}{F}_1{}^T, \underset{\sim}{F}_2{}^T, \ldots\ldots, \underset{\sim}{F}_B{}^T] \quad (11a,b)$$

$$\text{where } \underset{\sim}{\Delta}^T{}_i = [\Delta L_i, \alpha_i{}^+, \alpha_i{}^-] \text{ and } \underset{\sim}{F}_i{}^T = [t_i, m_i{}^+, m_i{}^-] \quad (12a,b)$$

The local coordinate system of the ith member consists of a cartesian coordinate system x^i, y^i where x^i is in line with the neutral axis of the member and directed from the negative end towards the positive end (Fig. 28).

LOCAL COORDINATE SYSTEM

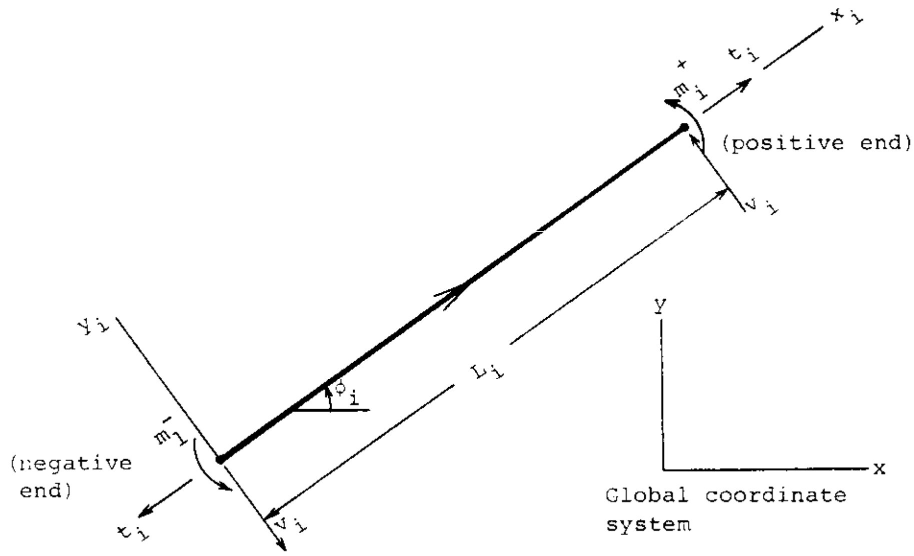


Figure 28

The member bending moment, shear force and axial force at the positive end are m_i^+, v_i, t_i respectively. Corresponding quantities are defined for the negative end. The length change of the ith member, the rotation of the positive end less the rigid body rotation of the member and the rotation of the negative end less the rigid body rotation of the member are ΔL_i, α_i^+ and α_i^- respectively.

For a plane frame structure composed of uniform straight beams, the stiffness matrix $\underset{\sim}{K}^L$ for the entire structure is:

$$K^L = \begin{bmatrix} \underset{\sim}{K}_1 & & \\ & \underset{\sim}{K}_2 & \underset{\sim}{O} \\ & & \ddots \\ \underset{\sim}{O} & & & \underset{\sim}{K}_E \end{bmatrix} \quad \text{where } \underset{\sim}{K}_i = E \begin{bmatrix} A_i/L_i & O & O \\ O & 4I_i/L_i & 2I_i/Li \\ O & 2I_i/L_i & 4I_i/Li \end{bmatrix}$$

where $\underset{\sim}{K}_i$ is the stiffness matrix for the ith member, expressed in its local coordinate system A_i, L_i and I_i are the cross sectional area, the length and the moment of inertia of the ith member, respectively, and E is Young's modulus. The stiffness $\underset{\sim}{K}^G$ expressed in the global coordinate system is

$$\underset{\sim}{K}^G = \underset{\sim}{N}^T \underset{\sim}{K}^L \underset{\sim}{N}$$

where $\underset{\sim}{N}^T = [\underset{\sim}{N}_1, \underset{\sim}{N}_2, \ldots\ldots, \underset{\sim}{N}_B]$

$$\underset{\sim}{N}_{ij} = \begin{cases} \underset{\sim}{N}_i^{\,+}\,\underset{\sim}{R}_i & \text{if } j \text{ is the positive end of member } i \\ \underset{\sim}{N}_i^{\,-}\,\underset{\sim}{R}_i & \text{if } j \text{ is the negative end of member } i \\ 0 & \text{otherwise} \end{cases} \tag{14c}$$

$$\underset{\sim}{N}_i^{\,+} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & -L_i^{-1} & -L_i^{-1} \\ 0 & 1 & 0 \end{bmatrix} \quad \text{and} \quad \underset{\sim}{N}_i^{\,-} = \begin{bmatrix} -1 & 0 & 0 \\ 0 & L_i^{-1} & L_i^{-1} \\ 0 & 0 & 1 \end{bmatrix} \tag{14d,e}$$

$$\underset{\sim}{R}_i = \begin{bmatrix} \cos\phi_i & \sin\phi_i & 0 \\ -\sin\phi_i & \cos\phi_i & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{14f}$$

$\underset{\sim}{R}_i$ is the rotation matrix for the ith member and ϕ_i is the inclination angle ϕ_i for the ith member in the global coordinate system (Fig. 28).

The modal displacement vector $\underset{\sim}{\delta}$ and the applied force vector $\underset{\sim}{P}$ are related through

$$\underset{\sim}{P} = \underset{\sim}{K}^G\,\underset{\sim}{\delta} \tag{15}$$

The problem is solved by constructing the stiffness matrix $\underset{\sim}{K}^G$ using equations (14) and then inverting equation (15) to obtain

$$\underset{\sim}{\delta} = (\underset{\sim}{K}^G)^{-1}\underset{\sim}{P}$$

2.2.2 Antisymmetric Response

When the ship moves obliquely across a wave system, then the slope of the wave changes at each section of the ship, which means that the draft of water on one side is different from that on the other side at a particular section. Thus a horizontal force is generated at each section since the pressures on the two sides of the ship are different. The sign of the horizontal force changes along the length of the ship so that bending in the horizontal plane results.

The web frame is symmetrical with respect to the vertical mid-plane. The hydrostatic pressure loading due to the oblique wave incidence can be decomposed into components symmetrical and antisymmetrical with respect to the vertical plane of symmetry. In the case of the antisymmetric loading, the frame undergoes deformations antisymmetrical with respect to the plane of symmetry of the structure, and the material particles in the plane of symmetry leave the plane along its normal direction with rotations possible only about the lines parallel to the plane. Thus, in this case the boundary conditions are as depicted in Fig. 29.

BOUNDARY CONDITIONS

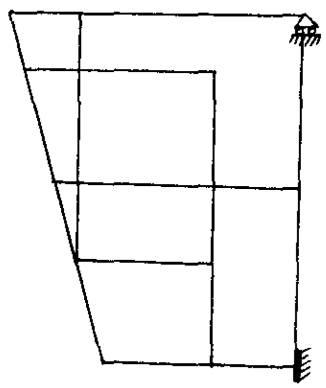


Figure 29

The structural response procedure is similar to the approach outlined to obtain the symmetric response. What remains to be determined is the magnitude of the pressure loading due to the oblique wave incidence.

The following assumptions are made:

- the breadth of the ship to the wavelength ratio is assumed to be small in comparison to one

- no allowance is made for the orbital motion of the particles in the wave, i.e., the problem is treated as a purely static one

- torsion is not considered, i.e. both due to vertical and horizontal pressures

- the decoupled problem is treated, i.e. the horizontal bending problem is considered separately from the vertical bending problem.

The direction of vessel advance has an angle α with the direction of wave propagation (Fig. 30).

HULL POSITION RELATIVE TO WAVES

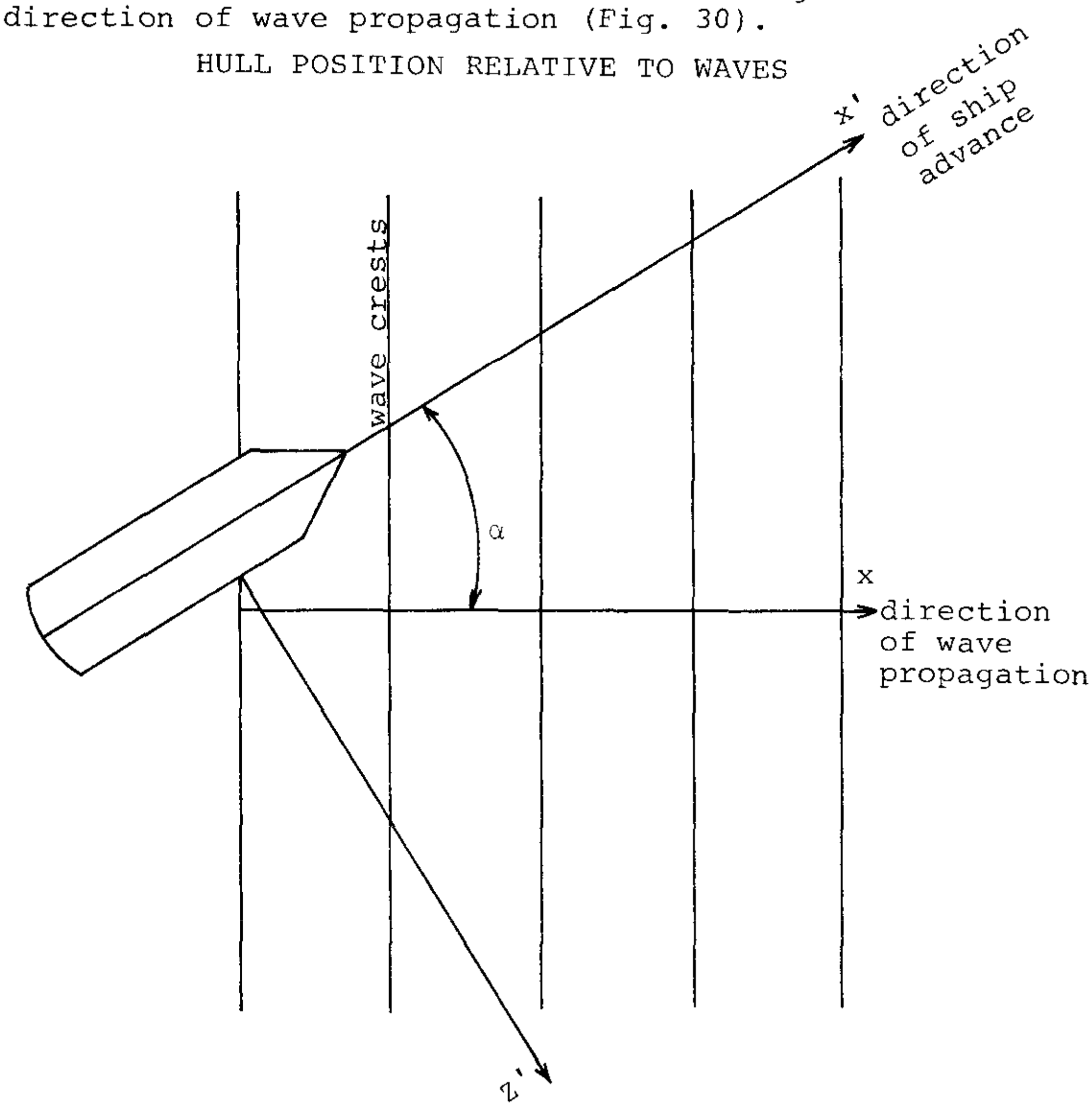


Figure 30

The coordinates x, x' and z' are related through

$$x = x'\cos\alpha + z'\sin\alpha \tag{17}$$

The wave surface elevation η_w (considered to be positive when upwards) is

$$\eta_w = A\cos\left(\frac{2\pi x}{l} - \omega t\right) \tag{18}$$

where A is the wave height

l is the wave length

ω is the wave frequency

t is the time.

Substituting the equation (17) into (18)

$$\eta_w = A\cos\left(\frac{2\pi x'}{l}\cos\alpha + \frac{2\pi z'}{\ell}\sin\alpha - \omega t\right) \tag{19}$$

The wave slope for z' = 0 and t = 0 is

$$\frac{\partial \eta_w}{\partial z'} = A\,\frac{2\pi}{l}\sin\left(\frac{2\pi x'}{l}\cos\alpha\right)\sin\alpha \tag{20}$$

The ordinates of the pressure loading d_1, d_2 (Fig. 31) due to oblique wave incidence are:

$$d_1 = T - \frac{B}{2}\left(\frac{\partial \eta_w}{\partial z'}\right) \tag{21a}$$

and $$d_2 = T + \frac{B}{2}\left(\frac{\partial \eta_w}{\partial z'}\right)$$

where T is the ship draft

and B is the ship breadth.

POSITIVE WAVE SLOPE

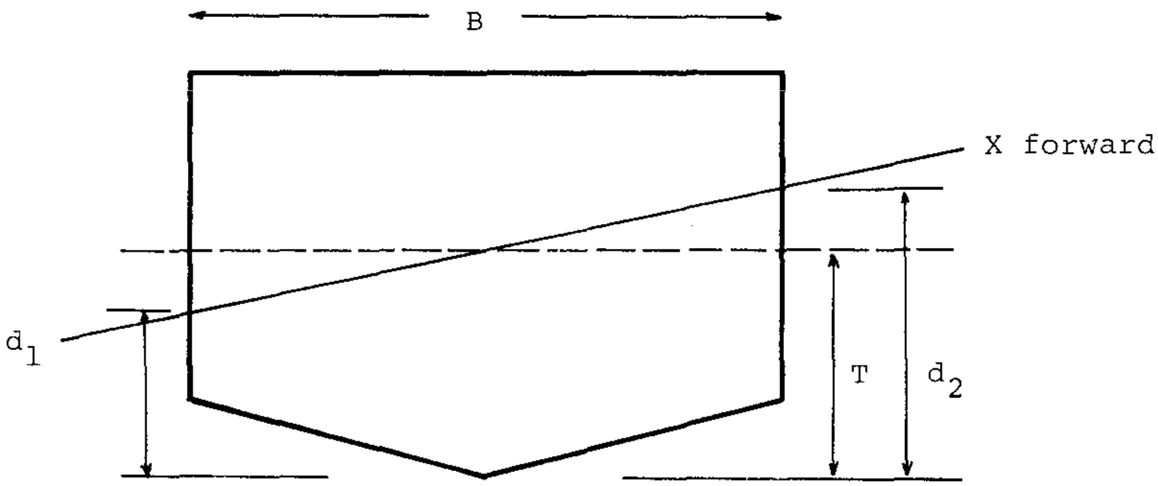


Figure 31

2.3 Deflections Due to Engine Room Double-Bottom Deformation

2.3.1 Grillage Method of Analysis

The engine room double-bottom foundation structure is considered as a grillage consisting of transverse floors and longitudinal girders. Both bending and shear deformations are retained in the analysis. Clamped boundary conditions are considered along the aft and forward engine room transverse bulkhead and the outer shell. The applied loading in the present case consists of the following components:

- hydrostatic pressure
- machinery deadweight loading
- thrust force

The method of approach for solving such a grillage problem is very similar to the one described in section 2.2. In fact the method considered for plane frames can be easily generalized to address three-dimensional frames. The grillage problem is then a particular case of the three-dimensional frame problem. Thus, both deflections, due to transverse web frame deformation and due to engine room double-bottom deformation, can be calculated by using a computer code for space frame problems.

2.3.2 Beam Method of Analysis

A more simplified model results by considering the transverse floor in question as a Bernoulli beam. In such a case, the transverse floor deflections w_{TF} (in the transverse plane) are given by the solution of the following differential equation:

$$\frac{d^4 w_{TF}}{dx^4} = \frac{\rho_w g T s_{TF}}{(EI)_{TF}}$$

where x is the transverse floor beam longitudinal axis $(EI)_{TF}$ is the transverse floor flexural rigidity, assumed constant

ρ_w is the water mass density

T is the ship's draft

and s_{TF} is the transverse floor spacing.

In addition to the transverse floor deflections w_{TF}, the deflections of the engine room double bottom in the vertical plane must also be considered. These deflections can also be estimated using a suitable equivalent beam model [10].

The various components of the stress hierarchy method just described are summarized for convenience in Table X.

3. Finite-Element Method

The Finite-Element Method (FEM) is now a widely adopted tool in ship structural analysis, and it has already been used quite extensively in ship hull-machinery compatibility studies, e.g. [2,5,7,17,19,51]. Due to the very small order of magnitude for deflections which are of concern here, i.e. thousandths

TABLE X

STRESS HIERARCHY METHOD

- **Hull Deflections**
  - **Hull Girder Deformations**
    - Timoshenko beam (shear deformations included)
    - Bernoulli beam (shear deformations not included)

      $$\frac{d^4 w_{HG}}{dx^4} = \frac{q}{(EI)_{HG}}$$

      w_{HG}: hull girder deflections

      x: hull girder longitudinal axis

      $(EI)_{HG}$: hull girder flexural rigidity

      $q = D(x) - P(x)$ - Displacement - Weight

      Basic assumptions:
      - no shear deformations
      - no contribution from elastic foundation
  - **Double Bottom Deformations**
    - Bernoulli beam

      $$\frac{d^4 w_{TF}}{dx^4} = \frac{\rho_w g T s_{TF}}{(EI)_{TF}}$$

      w_{TF}: transverse floor deflections

      x: transverse floor longitudinal axis

      $(EI)_{TF}$: transverse floor flexural rigidity

      ρ_w: water mass density

      T: ship's draft

      s_{TF}: transverse floor spacing

      - $w_{TF} \sim L^4_{TF} \sim [f(B)]^4$

      L_{TF}: length of the floors

      B: ship's beam
    - Grillage consisting of transverse floors and longitudinal girders
      - bending and shearing deformations retained
      - clamped boundary conditions along
        - aft engine room transverse bulkhead
        - forward engine room transverse bulkhead

      ******

      Calculations made for a TANKER

      L = 300m, Beam = 50m, Depth = 25.5m

      Loaded Draft = 19m. Deflections corresponding to a draft difference of 11.5m:

      Bending Deflections (downwards)

      <<

      Shearing Deflections (upwards)
  - **Transverse Web Frame Deformations**
    - frame analysis
    - $w_{TWF} = w'_{TWF}$ (aft body shape)

      $$= w''_{TWF}\left(\frac{\text{length of transverse floors in way of gears}}{\text{length of webs of side shell}}\right)$$

      w_{TWF}: transverse web frame deflections of an inch, the FEM is the only method of structural analysis which can lead to sufficiently reliable and accurate results, and for this reason it is included in the method being proposed here. The method is described in detail in various textbooks, e.g. [54,57] and only those aspects which are of particular relevance in the present study will be briefly addressed in this section.

The FEM has the advantage of enabling the structural engineer to construct a model which can follow very closely the exact geometry and material properties of the structure being examined. The major limitation of the method is the large computer cost it can imply, particularly if a large number of elements is used and if a nonlinear material analysis is to be performed. In addition to a large computer cost, the time involved in input preparation is extremely large and this represents another shortcoming. In the present case, the authors believe a FEM analysis (using a fine mesh) should be used at the last design stage when the structure is well defined, after some parametric studies have been performed with the simpler methods discussed previously. The FEM can in fact be used to tune and verify the results obtained from simpler methods.

A difficulty in implementing the FEM is the choice of the structural model, in particular the types of elements to be used. An exact representation of a complex structure such as a ship would involve so many elements that a certain degree of simplification becomes imperative. In the present case, the results we are looking for are deflections (translations and rotations) within the support structure of the machinery components, in particular at the critical points defined in Fig. 20, in the case of geared steam-powered vessels. In general, in studies of this nature the FEM model can represent the ship structure from the after end to the forward bulkhead of the machinery space. Due to symmetry, only the port or starboard side have to be considered, except if the structural arrangement is not symmetric about the ship's center plane. An initial analysis with a coarse mesh representing the whole machinery space, followed by a fine-mesh analysis of the structure in the area of interest, say in the vicinity of the reduction-gear casing, can be an efficient way of carrying the analysis, as suggested for example in [58].

The element types to be used should ideally be thin-shell elements. These are, however, expensive to implement so that simpler models are usually adopted. A combination of plane stress, truss and beam elements is often used. This implies that some structural components have to be lumped together, which requires a good degree of judgement. This is a subject which is not very deeply covered in the literature, and in which opinions vary quite widely since no exact rules can be established.

Another important decision which has to be made regarding the structural model is the definition of the boundary conditions. At the forward bulkhead of the machinery space, full fixity can, in general, be assumed. Along the center plane, the conditions of symmetry or anti-symmetry are used. If a coarse and a fine-mesh analysis are carried in sequence, the results from the coarse-mesh analysis are used to derive the most appropriate set of boundary conditions to be adopted at the fine-mesh stage.

The application of the FEM to the hull machinery compatibility problem is illustrated at the end of the chapter, when applying the proposed method to a specific ship.

4. Proposed Method

In order that the requirements for hull machinery rigidity compatibility are adequately met, the designer has to devote particular attention to the design of the shafting system and the structural arrangement of the machinery support systems.

In terms of the shafting system, the main criterion, as discussed earlier, is the maximum allowable difference between bearing reactions or ΔR. This is used to estimate the required number of bearings and their corresponding location. The bearing unit loads, the span to shaft diameter ratio, the shafting flexibility and the lateral vibration natural frequencies are additional factors to be considered in determining the number and location of shaft bearings. Using the Boston Naval Shipyard Code [56], described in Appendix B, a continuous-beam-shafting analysis can be made to determine the reactions, slope, deflection, bending moments, and the corresponding influence numbers. The definition of suitable alignment criteria in the vertical and horizontal plane is then used together with the results of the shafting calculations to define suitable alignment tolerances under the various operating conditions. The appliction of the ΔR criterion is summarized in the diagram in Figure 32.

The second main area of concern to ensure adequate hull-machinery compatibility is the structural design of the machinery support systems. The general guidelines given in Chapter II should be followed when designing the foundation. The deflections at the critical support points should then be determined by using sequentially the two methods discussed earlier. The stress-hierarchy method, in particular the frame and the grillage models, should be used to produce parametric variations involving the main structural parameters which define the structural arrangement (see Table II). In particular, the floor and girder scantlings and spacing and the frame and web frame scantlings and spacing should be varied. By doing so, the designer can determine the most efficient way of achieving the stiffness required by the machinery manufacturers. It is

APPLICATION OF THE ΔR CRITERION

$$\underset{\sim}{R}^{\circ} = \underset{\sim}{R}^{A} + \underset{\sim}{I}\,\underset{\sim}{\Delta}$$

$$\underset{\sim}{\Delta} = \underset{\sim}{F}\,\underset{\sim}{P}$$

$$\underset{\sim}{R}^{\circ} = \underset{\sim}{R}^{A} + \underset{\sim}{I}\,\underset{\sim}{F}\,\underset{\sim}{P}$$

$$0 < \underset{\sim}{R}^{\circ} < R^{P}$$

$$-\Delta R < R^{\circ}_{1} - R^{\circ}_{2} < \Delta R$$

$\underset{\sim}{R}^{\circ}$: vector of bearing reactions during operation

$\underset{\sim}{R}^{A}$: vector of bearing reactions during alignment

$\underset{\sim}{I}$: matrix of influence coefficients

$\underset{\sim}{F}$: engine room double bottom flexibility matrix

$\underset{\sim}{\Delta}$: engine room deflections at bearing locations

$\underset{\sim}{P}$: applied loads

$\underset{\sim}{R}^{P}$: limits of magnitude of bearing reactions based upon allowable bearing pressures

ΔR : maximum allowable difference between bearing reactions No. 1 and No. 2

$\underset{\sim}{I} = \underset{\sim}{I}$ (Bearing number and location, shaft rigidity)

$\underset{\sim}{F} = \underset{\sim}{F}$ (Hull (GIRDER) bending and shear rigidity, Engine room transverse web (FRAME) rigidity, Engine room double bottom (GRILLAGE) bending and shear rigidity)

Figure 32

obviously necessary to tune the frame and grillage models so that the results can be accepted with a certain degree of confidence. This can be achieved by using the FEM for which the degree of accuracy is much higher. After the designer is fairly confident that the structural arrangement achieves the required stiffness, then a final check with the FEM can also be undertaken.

The deflections at the critical support points should be computed for different loading conditions, in particular the light loading condition and the full loading condition, with full thrust and torque being applied to the thrust bearing foundation. Symmetrical and antisymmetrical loads should also be considered. All loads are applied statically.

The various components of the SHM have been inplemented in the computer, as described in Appendix C. The software package called ANALYSIS consists of the following two modules:

- GIRDER
- SPACE

The GIRDER module is for hull girder analysis. The SPACE module is for two-dimensional frame analysis and for grillage analysis of the double bottom. A listing of the computer code analysis is given in Appendix C, together with a description of the main variables of the program. It should be noted that the SPACE module is in fact a three-dimensional frame analysis program, so that the structure being analyzed does not necessarily have to be a two-dimensional frame or a grillage. A three-dimensional frame containing all the main parameters given in Table II can in particular be considered, and this certainly represents a very powerful model.

The method briefly described in the foregoing is summarized in Table XI. It can best be understood by considering the specific example discussed in the next Chapter.

TABLE XI

Hull-Machinery Compatibility
Proposed Methodology Summary

Shaft Design

- Meet ΔR criterion (see Figure 32)

Foundation Design

- Follow whenever possible the general guidelines of Chapter II, Section 2.
- Construct SHM models and tune using FEM so that computed deflections are comparable.
- Perform parametric variations with SHM, varying the main parametric variations listed in Table II.
- Choose combination of parameters which meets requirements given in Chapter III. Make this choice on the basis of a suitable efficiency criterion, such as material (weight) savings.
- Check final design with FEM.

CHAPTER V. EXAMPLE OF APPLICATION

1. Ship Main Characteristics

The ship to be considered here is a 188,500-DWT tanker (#11 in Tables III through VIII of Chapter II). For convenience, its main characteristics are summarized in Table XII.

The tanker's main compartments, as shown in Fig. 33 include a forepeak compartment, five cargo tanks, ballast water tank, slop tanks, engine room, and aft peak compartment. A double bottom extends through the vessel's length, and two longitudinal bulkheads 39' off the center line also extend through the length. The framing system is transverse.

As shown in Fig. 33, the engine room is located aft, just forward of the aft peak, and it extends from frames 71 to 114, over a length of 107.5'. At the forward end it has a pump room. Its structural arrangement includes transverse plate floors 0.75" thick at every frame, 30" apart, in way of turbine foundations, reduction gear, thrust bearing, and aft until the stern tube. Forward of frame 88, transverse plate floors 0.63" thick are provided at every frame.

In the engine room, web frames have a maximum spacing of five frames (150"). The scantlings of a typical web frame in way of the reduction gear are: web depth 4.25', web thickness 0.63", flange width 6", flange thickness 1". Transverse frame scnatlings in this same area are: web depth 10", web thickness 0.50", flange width 4", flange thickness 0.75". Side stringers are spaced 7.5' to 9', and typical scantlings have: web depth 39", web thickness 0.50", flange width 5", and flange thickness 0.50".

Flats are 30', 45', and 60' above the base line, extending for the complete length of the engine room (see Fig. 34). There are four stanchions (12" x 12" x 1.59" WF) 15' off center line, port and starboard at frames 88 and 99, extending from the inner bottom to the 60' flat (see Fig. 34 and 35). A sloping flat extending from the engine room aft bulkhead to second reduction gear and close to the thrust bearing is provided. The sloping flat is 3' wide on each side of center line and broadens into the gear foundation at frame 96. A bilge-water oil drain tank is situated below this sloping flat and extends from frame 96 to frame 110.

The transverse watertight bulkhead forming the forward end of the engine room, at frame 71, is stiffened by bulb angles having the following scantlings: above the 45' flat, 16" x 6" x 0.50"/0.75", and below the 45' flat 24" x 8" x 0.50"/1". The plate thickness varies from 0.50" to 0.75".

TABLE XII

TANKER MAIN CHARACTERISTICS

<u>Principal Dimensions</u>	
Length BP	915'
Breadth	166'
Depth	78'
Draft	59'
<u>Displacement</u>	188,500 DWT
<u>Machinery</u>	
Steam turbine	28,000 SHP

<u>Engine Room Construction</u>

Transverse framing, spacing 30"

Engine room length 107.5'

Engine room width in way of reduction gear 41.7'

Web frame spacing 150"

Inner bottom plate 0.72" thick

Bottom C. L. girder 0.81" thick

Bottom side girders 0.63" thick

Double bottom depth 9'

<u>Shafting Details</u>

Line shaft diameter 23.75"

Tail shaft diameter 29.75"

Thrust bearing location 4' aft of reduction gear

Number of line shaft bearings, 1

Height of thrust bearing center above inner bottom 8.93'

TANKER'S MAIN COMPARTMENTS

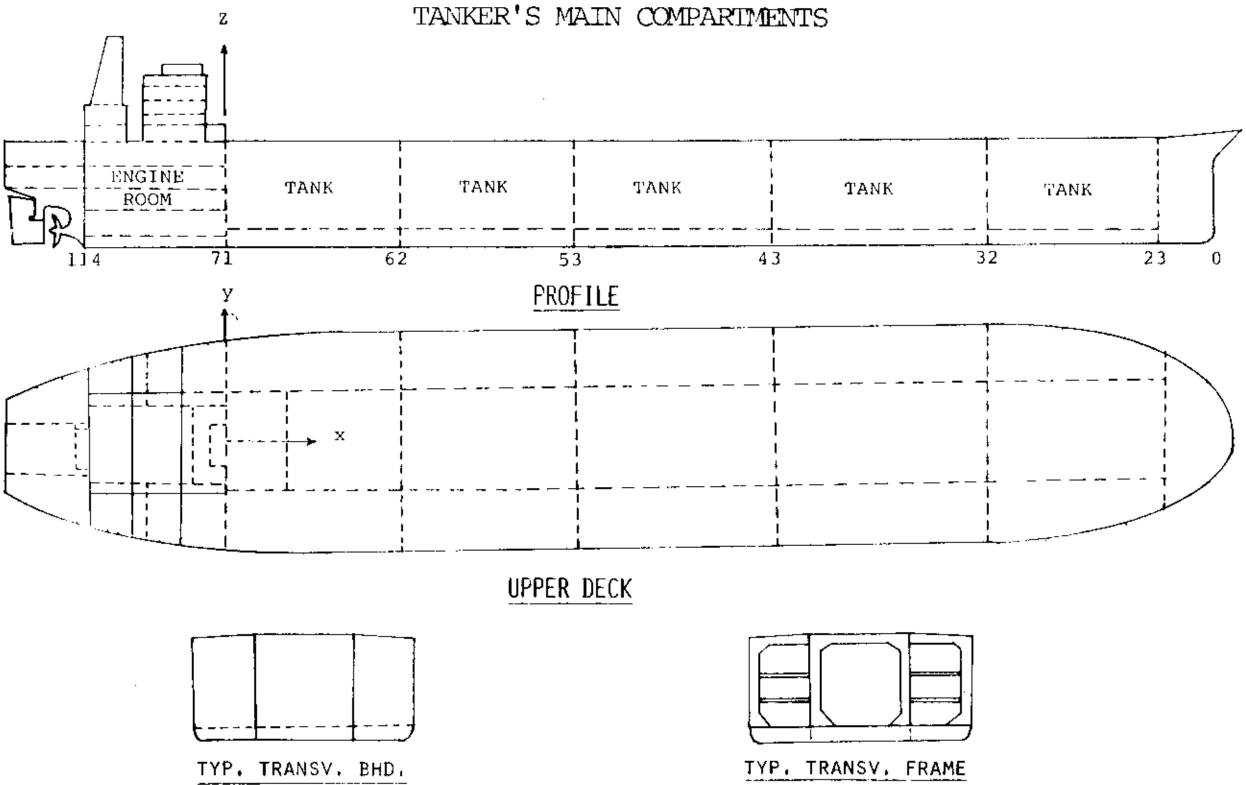


Figure 33

TANKER'S MACHINERY COMPARTMENT GEOMETRY

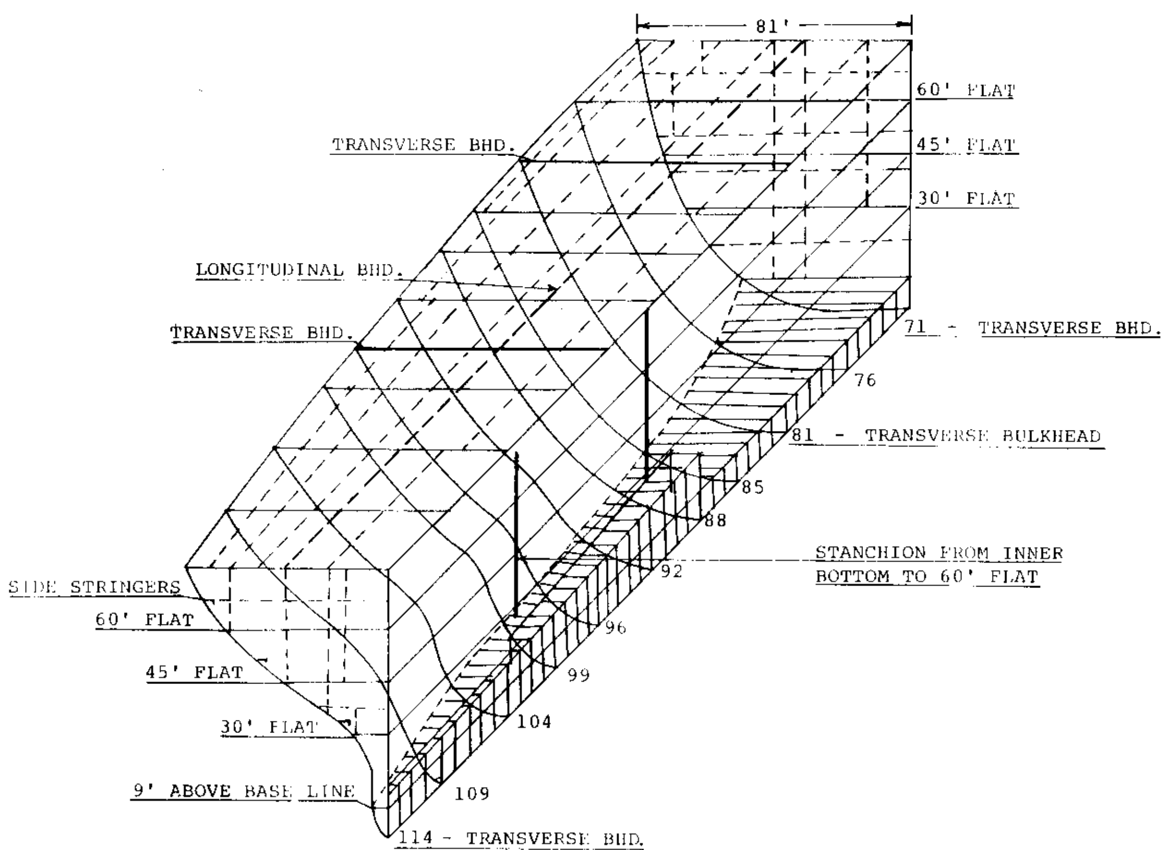
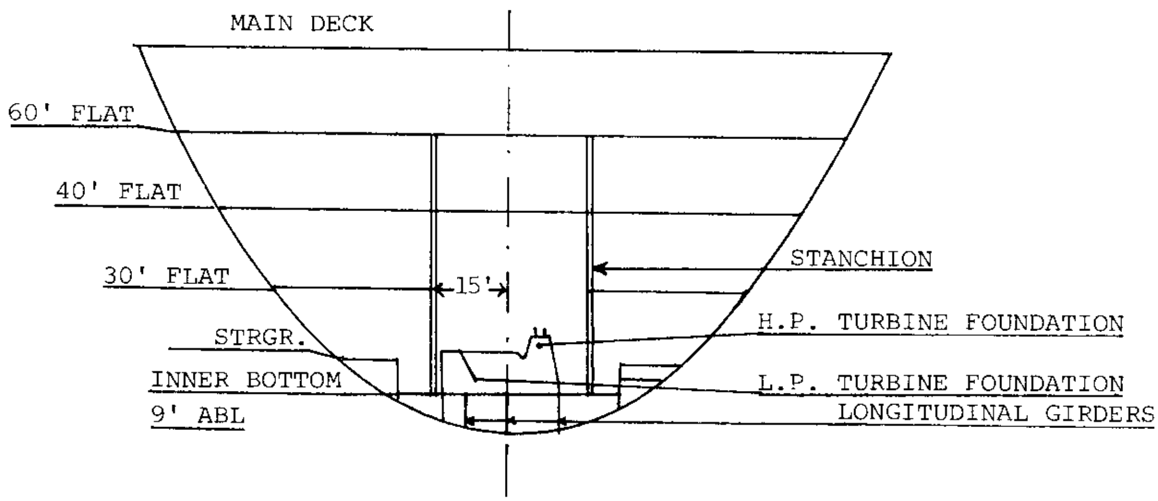
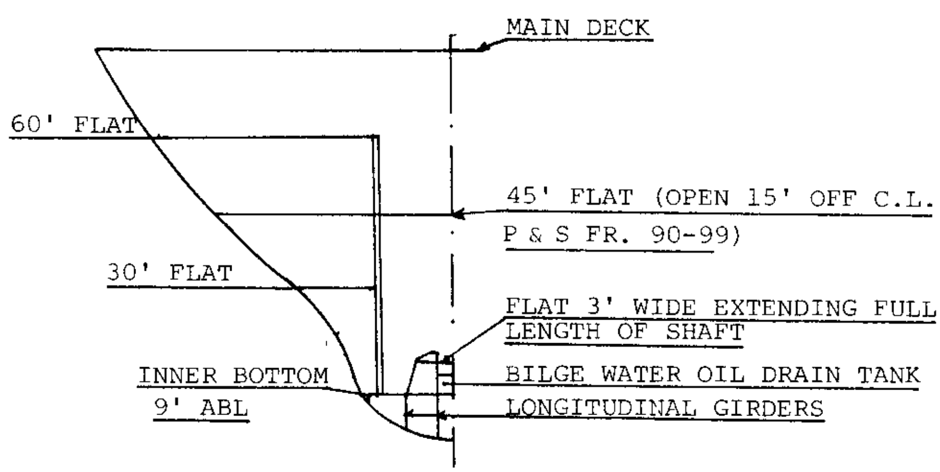


Figure 34



SECTION AT FR.88 - LOOKING FWD.



SECTION AT FR. 99 - LOOKING FWD.

(PORT SHOWN - STBD. SIMILAR)

Figure 35

The transverse bulkhead forming the aft end of the engine room, at frame 114, is stiffened by bulb angles having the following scantlings: 14" x 4" x 0.50"/0.75". The plate thickness varies from 0.44" to 1". The side shell plating thickness varies from 0.69" to 1.13", the main deck plating forward of frame 90 is 0.88" thick, and aft of this frame it is 0.72" thick.

In way of the reduction gear, there are a total of five longitudinal bottom girders, at center line and 9' and 18' off center line on port and starboard. The girder plate thickness is 0.81" for the center girder and 0.63" for the side girders. The inner bottom plating is 0.72".

2. Finite-Element Model

A FEM for the hull structure aft of the forward bulkhead of the machinery space was developed for the purpose of obtaining accurate results for the deflections at the critical support points defined in Fig. 20, in particular at the corners of the reduction gear foundation. The program ADINA (Automatic Dynamic Incremental Nonlinear Analysis) was used for implementing the FEM analysis [59]. A special graphics package to display the input mesh and to check the grid geometry was developed by the authors as part of this research effort.

The complete hull structure from frame 71 to the transom was included in the model. Vertically, the model extends from the ship's bottom to the main deck and in the transverse direction from the center plane to the port side shell plating. Only the portside half of the afterbody was modelled, since the structure can be for any practical purposes considered symmetric. The superstructure was not included, since it does not affect to a large extent the deflections in the region of interest. Other structural elements having a negligible influence on the double-bottom deflections were also omitted. This was necessary in order to keep the computer time required for the analysis within reasonable limits.

The FEM contains a total of 765 elements subdivided among the following groups: 6 truss elements, 405 plane stress three-dimensional orthotropic elements, and 354 beam elements.

The truss elements were used to model the stanchions. The plane-stress elements were used to model the transverse and longitudinal bulkheads, decks, and flats. The linear-orthotropic-elastic model was used in order to represent in a simple way the effect of the stiffeners, and by using this approach it was possible to decrease substantially the number of elements needed to represent adequately the structure. The computation of the equivalent orthotropic material constants was done in accordance with the classical orthotropic-plate theory [60].

The beam elements were used to model the double-bottom structure and side shell, including transverse web frames, bottom plating, inner bottom plating, transverse floors, and longitudinal girders. The side shell plating was accounted for in the stiffness of the side stringers, by considering an effective breadth of plating attached to a beam element.

The finite-element grid includes a total of 513 nodes. The mesh is shown in Fig. 36, as plotted by the graphics package developed by the authors. Fig. 37 shows a plot of the longitudinal bulkheads and Fig. 38 a plot of the flats.

The following boundary conditions were adopted: full fixity at the forward engine room bulkhead (frame 71) and symmetry conditions along the center line (zero y displacement and zero rotations about the x and z axes, the coordinate system being the one shown in Fig. 33). The total number of degrees of freedom is 1806.

Two static-loading conditions were considered: full-load condition at full power (full thrust and torque), and light-load condition. The hydrostatic pressures, machinery component weights, fuel weight, etc., were translated into statically equivalent forces and moments to be applied at the grid's nodal points. ADINA has provisions for gravity loading due to the mass of the elements, and the material specific gravity was adjusted so as to provide the total correct structural weight.

3. Results

3.1 Shaft Bearing Reactions

The shafting system of the tanker under consideration has been studied using the Boston Naval Shipyard Computer Code [56]. The shafting arrangement is shown in Figure 39. The bearing reaction influence numbers, as well as the straight line bearing reactions have been obtained for various magnitudes of the shaft length L and diameter D. The corresponding numerical values for the particular case of D = 23.75" and L/D = 15 are presented in Table XIII. This information is useful in order to examine the influence of the movements of the bearing locations on the changes of the magnitude of the bearing reactions. In the alignment condition, the equality of the magnitude of the reactions at bearings No. 1 and 2 is sought.

TANKER FINITE-ELEMENT GRID

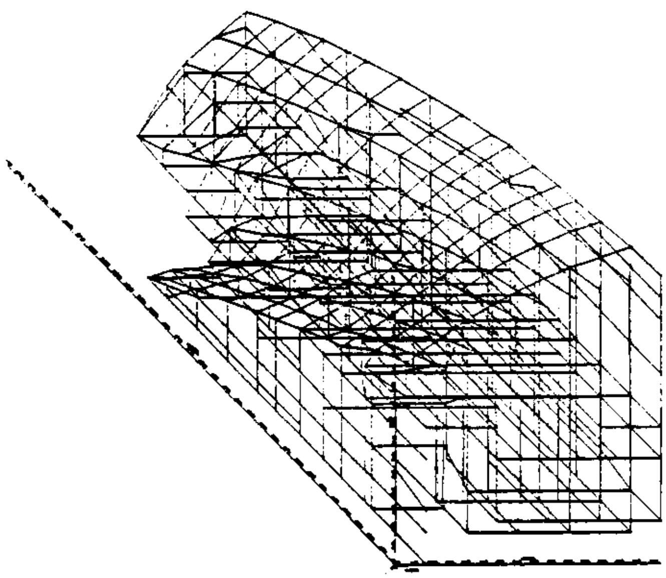


Figure 36

FINITE-ELEMENT GRID - FLATS

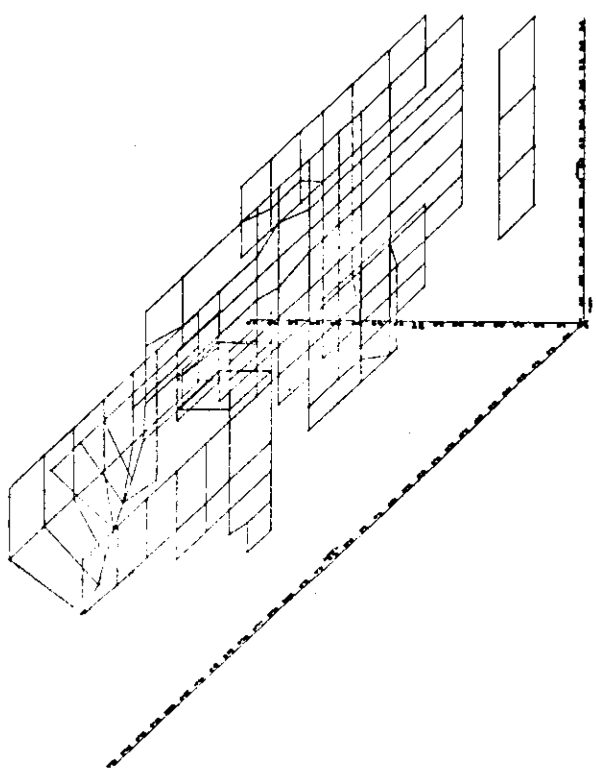


Figure 37

FINITE-ELEMENT GRID - LONGITUDINAL BULKHEADS

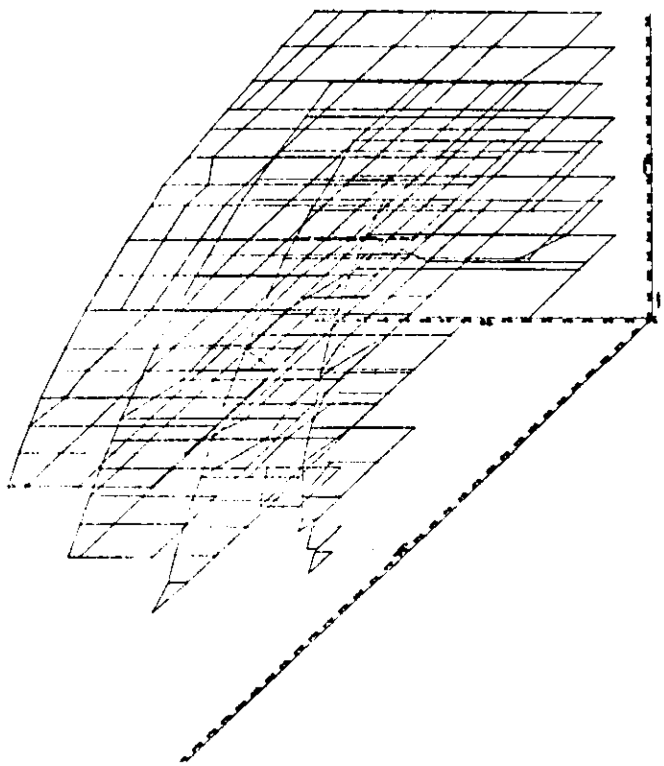


Figure 38

SHAFTING ARRANGEMENT

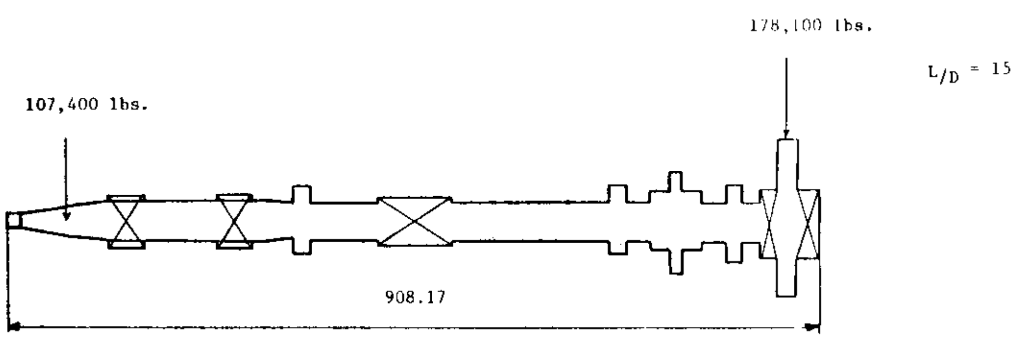


Figure 39

TABLE XIII

TANKER L/D=15.00 D=23.75

Bearing Reaction Influence Numbers
(lb. per 0.001 inch rise of bearings)

Bearing No.	1	2	3	4	5
1	1524.2	-1942.9	718.0	-397.5	98.2
2	-1942.9	2500.2	-1010.8	602.3	-148.8
3	718.0	-1010.8	1068.6	-1274.5	498.8
4	-397.5	602.3	-1274.5	1919.3	-849.6
5	98.2	-148.8	498.8	-849.6	401.5

Straight Line Bearing Reactions
(lbs.)

	1	2	3	4	5
	45625.2	174149.4	54682.9	-50552.5	207918.5

Tables XIV, XV and XVI list the values of the straight line bearing reactions R_1 and R_2, the influence coefficients I_{11} and I_{22} and the magnitude of the ASE (Allowable Setting Error) based on an assumed ΔR equal to 15,000 lbs for various values of the L/D ratio. The bearing reactions are given in lbs and the influence coefficients in lbs per unit of an inch. The ASE is computed in mils of an inch. To avoid incompatibility problems the value of the actual $(R_1 - R_2)/(I_{11} - I_{22})$ must be kept smaller than the corresponding value of ASE for a given allowable ΔR.

3.2 <u>Hull Girder</u>

The four loading conditions (for the tanker under consideration) presented in Table XVII have been analyzed. They are:

- the lightship
- the lightship plus segregated ballast
- Martinez departure (ballast)
- Alaska departure (full load)

Table XVII also lists the cargo carried, the mean draft, the maximum bending moment and the bending deflections amidships for each of the four loading conditions listed above. The hull girder deflections are considered positive when upwards from that reference line. The hull girder deflections amidships obtained from the proposed procedure are compared with the predictions based on the semiempirical method suggested in Reference [12]. It can be seen from Table XVII that the two results compare fairly well.

The hull girder deflections are also plotted in Figure 40 within the extent of the machinery compartment from frame 71 to frame 114. It can be seen from Figure 40 that the lightship, the lightship plus segregated ballast and the Martinez departure are in a hogging condition, whereas the Alaska departure is in a sagging condition. The hull girder deflections plotted in Figure 40 are total deflections (bending plus shear deflections).

TABLE XIV

ALLOWABLE SETTING ERROR (D-23.75 in.)

L/D	R_1	R_2	I_{11}	I_{22}	ASE
10	67425.4	138000.9	2339.7	4468.5	- 7.046
12	58848.7	153234.1	1936.6	3435.7	-10.006
13	54455.7	160429.8	1777.9	3062.5	-11.677
14	50044.9	167387.3	1641.6	2755.4	-13.467
15	45625.2	174149.4	1524.2	2500.2	-15.369
16	41205.2	180744.8	1422.9	2287.0	-17.359
16.16	40501.6	181781.9	1408.1	2256.4	-17.682
17	36817.8	187154.6	1336.4	2109.9	-19.392
18	32619.4	193180.0	1265.1	1967.6	-21.352

R_1, R_2 straight line bearing reactions in LBS
I_{11}, I_{22} LB. per 0.001 rise of bearings

TABLE XV

ALLOWABLE SETTING ERROR (D-22.5625 in.)

L/D	R_1	R_2	I_{11}	I_{22}	ASE
10	69333.4	134276.6	2423.3	4721.0	- 6.528
12	61352.9	148742.1	1987.2	3594.5	- 9.332
13	57222.3	155586.3	1814.3	3185.8	-10.937
14	53082.0	162175.8	1665.8	2849.2	-12.675
15	48948.5	168547.6	1537.6	2569.2	-14.541
16	44823.8	174743.2	1426.7	2334.6	-16.522
17	40703.0	180797.7	1331.0	2137.6	-18.597
18	36617.0	186690.6	1249.5	1974.0	-20.704
19	32741.4	192192.2	1183.6	1844.4	-22.700

R_1, R_2 straight line bearing reactions in LBS
I_{11}, I_{22} LB. per 0.001 rise of bearings

TABLE XVI

ALLOWABLE SETTING ERROR (D-21.375 in.)

L/D	R_1	R_2	I_{11}	I_{22}	ASE
10	71022.5	130887.6	2520.8	5013.1	- 6.019
11	67455.7	137817.3	2266.9	4330.5	- 7.269
12	63668.5	144570.1	2047.4	3779.1	- 8.662
13	59789.3	151088.7	1858.4	3329.3	-10.198
14	55897.8	157348.6	1695.8	2958.8	-11.876
15	52030.3	163363.9	1555.6	2650.7	-13.697
16	48197.1	169166.9	1434.2	2392.3	-15.656
17	44380.6	174815.1	1328.9	2174.3	-17.743
18	40568.8	180348.9	1237.9	1990.4	-19.934
19	36770.9	185771.1	1160.5	1837.4	-22.160
20	33160.6	190852.1	1098.7	1717.4	-24.244

$R_1 R_2$ straight line bearing reactions in LBS

I_{11}, I_{22} LB. per 0.001 rise of bearings

HULL GIRDER DEFLECTIONS

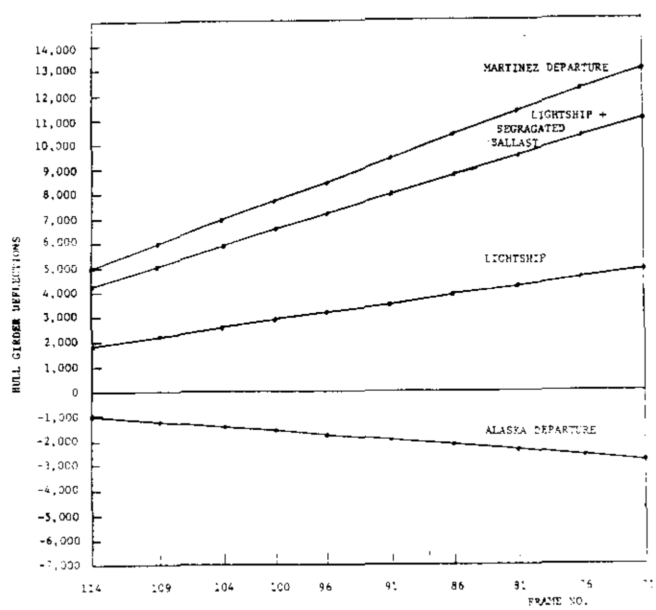


Figure 40

TABLE XVII

HULL GIRDER DEFLECTIONS

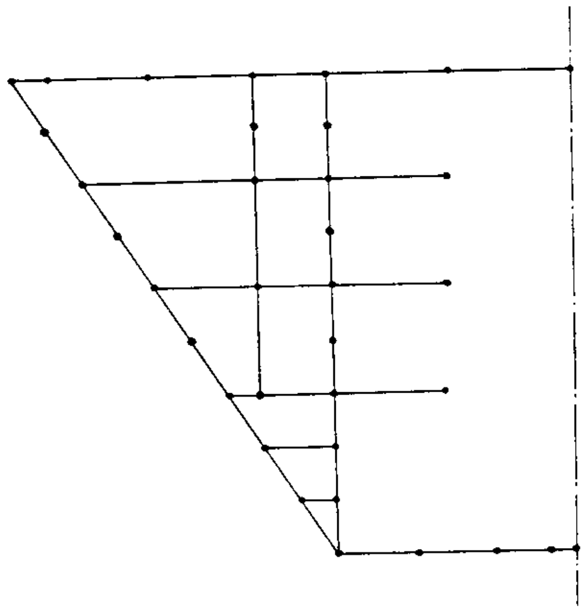
Loading Condition	Cargo Tons	Mean Draft in Ft	Maximum Bending Moment FT-TONS	Bending Deflection Amidships* MILS of IN	Bending Deflection Amidships** MILS of IN
Lightship	-------	9.06	-790,346	8,005	9,097
Lightship + Segregated Ballast	To meet I.M.C.O. Require-ments	25.64	-1,705,970	17,278	19,241
Martinez Departure	59,857 S.W.	26.87	-2,077,382	21,040	23,115
Alaska Daparture	182,346 Oil	59.01	657,075	-6,655	-5,450

* estimated from REF [12].

** from proposed SHM.

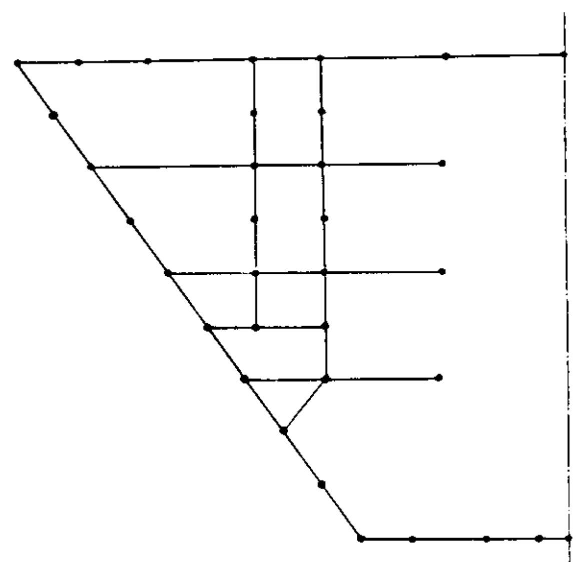
3.3 Transverse Frames

The deflections of the transverse frames 92, 96 and 109 have been obtained due to weight and hydrostatic pressure loading. The geometry of the frame 92, 96 and 109 is depicted in Figure 41, 42 and 43 respectively. The centerline nodal points are not allowed to displace horizontally for symmetric loading conditions. Furthermore, the rotations at the centerline nodal points are zero for symmetric loading. The deflections of the lowest centerline nodal point are presented in Table XVIII for three waterlines with mean draft equal to 59.33, 65.27 and 53.40 feet. The deflections reported in Table XVIII are with respect to the deflections of the highest centerline nodal point. Deflections are given in mils of an inch and are considered positive upwards. The transverse web frame structure is compressed by the hydrostatic pressure from the bottom and as a result the double-bottom structure



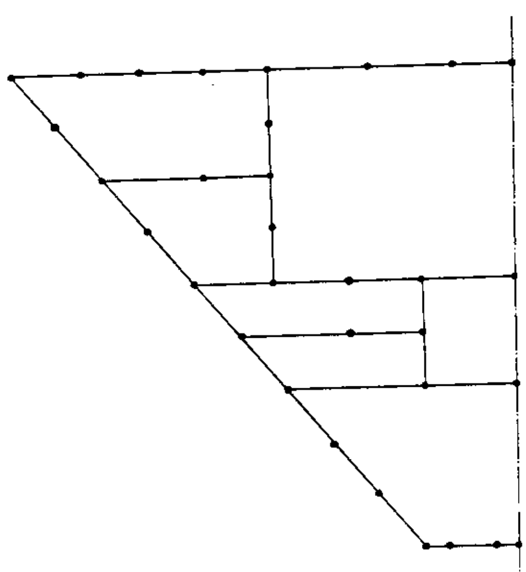
TRANSVERSE FRAME 92
GEOMETRY

Figure 41



TRANSVERSE FRAME 96
GEOMETRY

Figure 42



TRANSVERSE FRAME 109
GEOMETRY

Figure 43

has a tendency to displace upward. However, the hydrostatic pressure at the side plating tends to produce downward displacements at the bottom structure. It can be seen from Table XVIII that for the frames 92 and 96 the centerline displacements calculated are directed downwards indicating that the side plating pressure deformation mechanism is the dominant one, in this case. The centerline deflection results for frame 109 suggest that, in that case, the hydrostatic bottom pressure deformation mechanism is dominant.

TABLE XVIII

BEARING REACTION INFLUENCE NUMBERS

Frame No. / Mean Draft ft	59.33	65.27	53.40
92	-20,230*	-26,033	-13,152
96	- 7,739	-11,894	- 2,873
109	5,654	5,716	6,015

* deflections given in MILS of an INCH and considered positive upwards

Deflections for a sample case have been computed to consider the case of asymmetric pressure distribution (the theoretical procedure is presented in section 2.2.2). The port and starboard water levels considered are $d_1 = 53.40'$ and $d_2 = 65.27'$ respectively (Fig. 31) with a mean draft equal to 59.33'. The centerline vertical deflection for frame 96, under the loading condition mentioned above, has been found to be equal to 4,970 mils of an inch.

3.4 Double Bottom

The engine-room double-bottom structure, for the tanker under consideration, has been modelled as a grillage consisting of interesecting beams (transverse floors and longitudinals). Fig. 44 presents the geometry of the grillage structure analyzed by the SHM. The deflections of the grillage structure have been obtained due to weight and hydrostatic loading. The grillage centerline vertical deflections are plotted in Fig. 45 for three different draft levels. The deflections are considered

ENGINE ROOM DOUBLE-BOTTOM GRILLAGE STRUCTURE

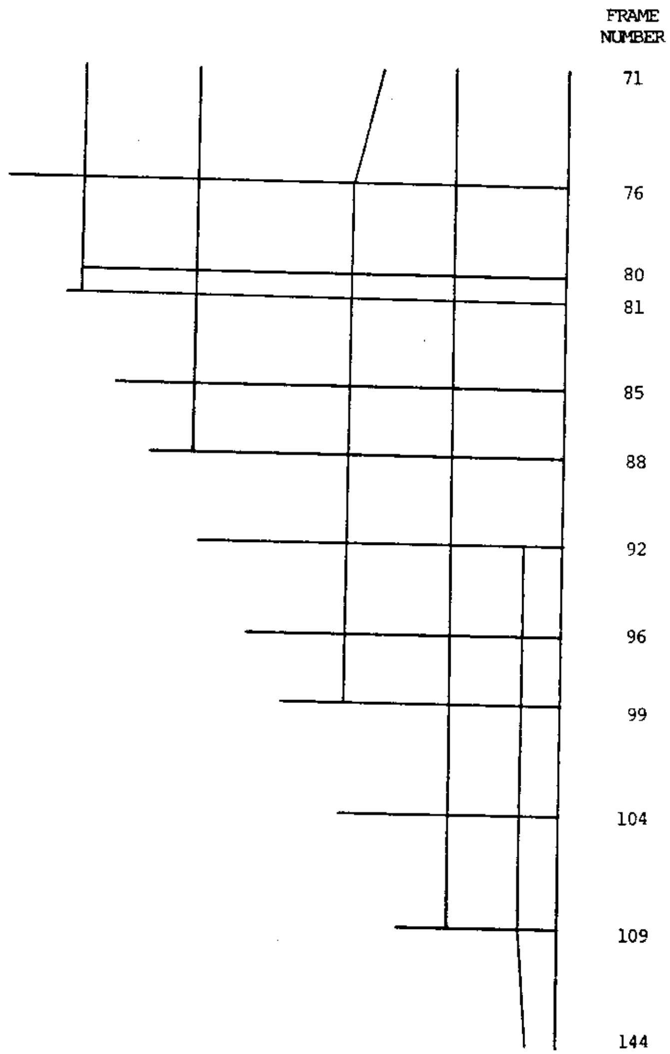


Figure 44

GRILLAGE CENTERLINE VERTICAL DEFLECTIONS

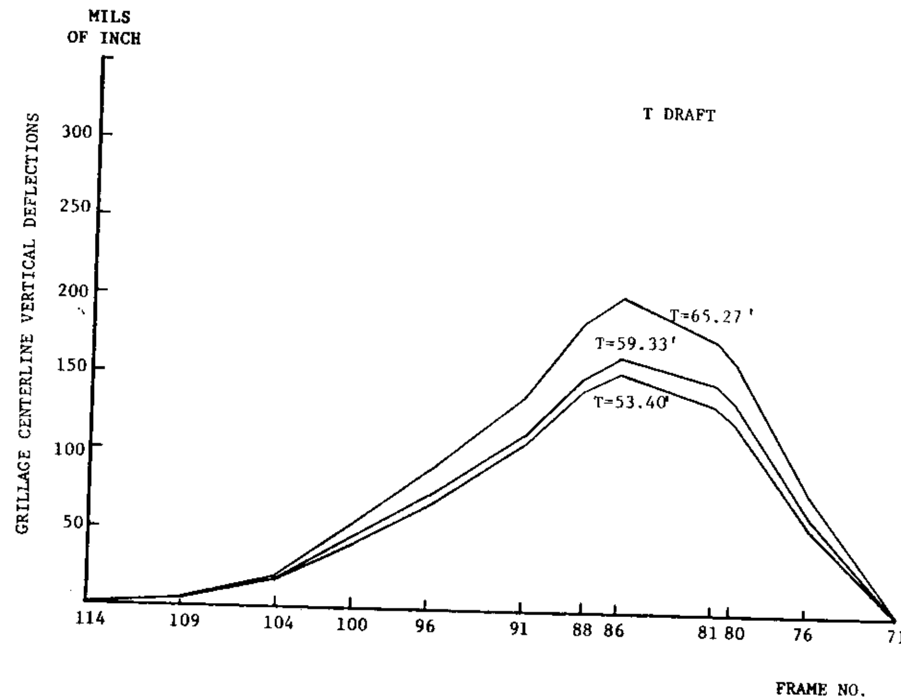


Figure 45

positive when upwards. The magnitude of engine-room double-bottom centerline deflections can be obtained from Fig. 45 as well as its dependence on waterline draft variations.

3.5 Finite-Element Model Results

The vertical deflections along the ship's centerline as determined by the finite-element analysis are shown in Fig. 46. These deflections are measured relatively to the forward bulkhead of the machinery space, since in the analysis a perfect clamped condition was assumed along this bulkhead. It can be concluded from Fig. 46 that in the light condition the hull structure along the machinery space deforms in hogging. In the full load condition, the forward part of the machinery space deforms in sagging (from frames 71 to 76), while the after part (aft of frame 76) deforms in hogging. This inversion of curvature within the machinery space in the full load condition was also observed in the studies reported in [2], as mentioned in Chapter I.

The most relevant result the finite-element analysis provided concerns the deformations at the critical points defined in Fig. 20, in particular at the corners of the reduction gear foundation. The relative deflections at these points between the full and light-load conditions are shown in Fig. 47. The results indicated in this figure show that due to the increase in draft and the applied thrust and torque, the four critical points (forward and aft port corners and low-speed gear bearings), suffer a translation forward, along the x-axis, which does not depart significantly from an average of 0.037".

The translation along the z-axis is larger for the forward low-speed gear bearing (0.0699"), and much smaller for the after corner (0.0341"), while for the remaining two points the values are closer (0.0466" and 0.0632"). Due to the condition of symmetry, the points along the center line do not move in the y direction, while the outer corners move by practically the same amount (0.0057" and 0.0058").

As discussed in Chapter III, the proposed deflection limits for foundations of geared-turbine-propulsion units, specify a maximum allowable value of 0.004" for the vertical movement at the seating surface of the forward port corner of the reduction-gear foundation relative to a plane passing through the remaining three critical points (the aft port corner, the forward low-speed gear bearing and the aft low-speed gear bearing). The finite-element analysis provides for each loading condition the distorted positions for the four points mentioned above. If the x, y and z coordinates for the forward and aft port corners of the reduction gear foundation and the forward and aft low speed gear bearings are denoted by x_i, y_i, z_i with $i = 1, 2, 3, 4$ respectively, then it can easily be shown that the vertical distance mentioned above can be obtained

VERTICAL DEFLECTIONS ALONG CENTERLINE (MILS)

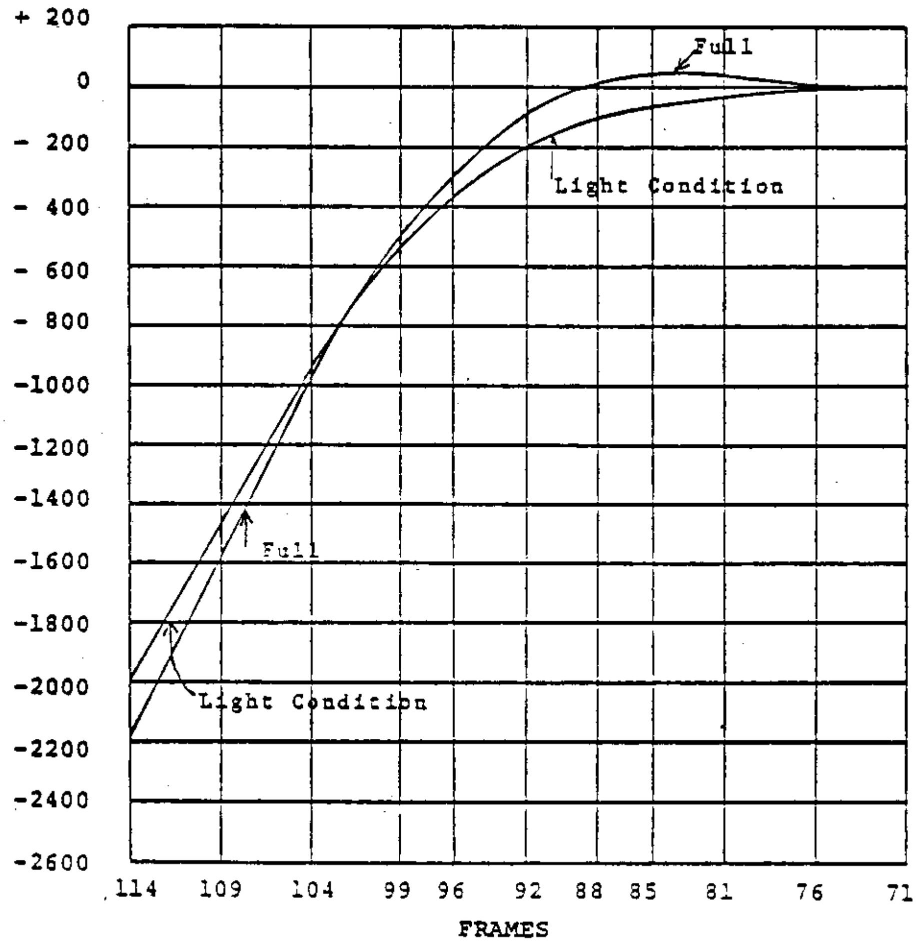


Figure 46

RELATIVE DEFLECTION DUE TO CHANGE IN DRAFT (FULL-LIGHT)

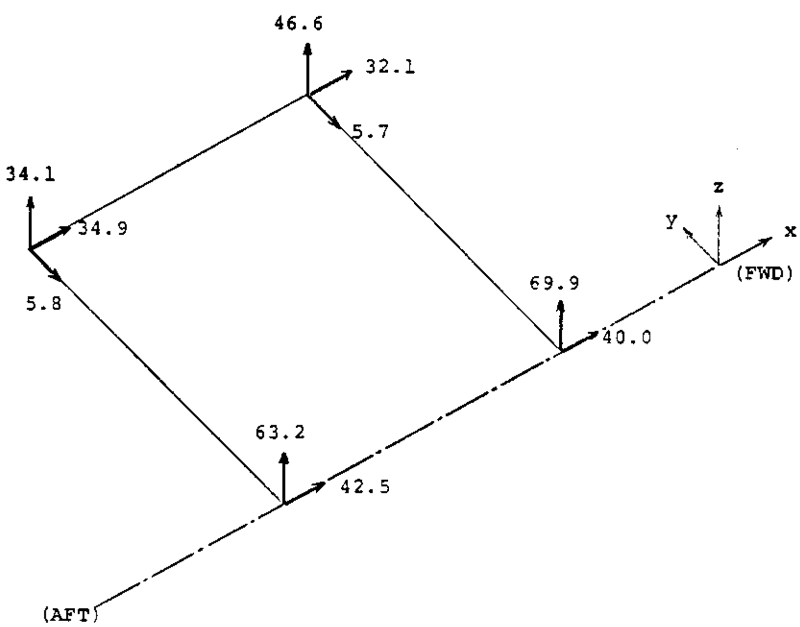


Figure 47

from the following equation:

$$d = \frac{Ax_1 + By_1 + Cz_1 + D}{\sqrt{A^2 + B^2 + C^2}}$$

where

$$A = (y_3 - y_2)\ (z_4 - z_2) - (y_4 - y_2)\ (z_3 - z_2)$$

$$B = (z_3 - z_2)\ (x_4 - x_2) - (x_3 - x_2)\ (z_4 - z_2)$$

$$C = (x_3 - x_2)\ (v_4 - y_2) - (v_3 - y_2)\ (x_4 - x_2)$$

$$D = -x_1A - y_1B - a_1C$$

The value of d as determined from the equation defined above was computed for the two loading conditions considered here. It was found that in the light condition d is equal to 0.018", while in the full-load condition it is equal to 0.017". Thus, the difference between the two loading conditions gives a difference of 0.001" in the out-of-plane deformation suffered by the reduction-gear seating surface, which means that the design is well within the maximum limit of 0.004" suggested in Chapter III.

VI. CONCLUSIONS AND RECOMMENDATIONS

The potential for incompatibility of hull and foundation deflections at the support points of gear-turbine and diesel machinery relative to the rigidity requirements for the machinery should be given careful attention during the design phase, particularly in large vessels.

The problem is strongly influenced by the hull shape and beam in way of the machinery space. A machinery space located aft with narrow beam, "U" or "V" sections and waterlines which converge is inherently a relatively stiff arrangement, particularly in the vertical direction. In contrast, hull shapes similar to those of cruisers and destroyers, with gradually rising buttocks and strut-supported propeller bearings, place the machinery further forward, generally on a relatively wide, flat bottom. Such vessels appear to have a greater risk of hull-machinery incompatibility.

Conformance with the rules of the Classification Societies per se does not guarantee freedom from the problem of hull-machinery incompatibility and each case requires specific analysis of the design.

Location of the machinery space and arrangement of the machinery foundations are extremely important. The integration of machinery foundations with basic hull structure must be carefully planned. Bulkheads, both transverse and longitudinal, as well as decks and shaft alleys may be utilized to improve overall stiffness and particularly to distribute thrust forces to the hull over as wide an area as possible.

Analytical methods are available to determine the estimated maximum deflections at critical machinery support points. These deflections represent movements between the condition when machinery is aligned (usually lightship, zero power output) and some other operating condition (usually maximum draft, maximum power) that produces the greatest deflections.

Proposed limits for the deflections of machinery support points have been included in this report and may be used for comparison with estimated structural deflections. Whenever possible, all potential machinery suppliers should be consulted regarding their individual requirements.

In the event that deflection limits are exceeded, consideration should be given to increasing the structural stiffness as necessary. Should this become impractical, or uneconomical, then steps may be taken to render the machinery less susceptible to foundation movements. This may involve trade-offs between structure and machinery costs. Although it is impractical to construct gear cases that can resist foundation movements, special designs, chocking arrangements, and other devices may minimize the effects. Similarly the box girder type of construction for large diesels adds significantly to the vertical stiffness in way of the engine.

It will be of interest that some further work is performed in the future to include the following important aspects of the hull-machinery coupled response:

- nonsymmetric response using the finite-element method
- more work on diesel-engine-propulsion support systems
- dynamic response
- include plane-stress analysis subroutine in stress hierarchy computer code to treat deflections of transverse bulkheads
- thermal effects
- perform parametric variations

VII. REFERENCES

1. J.H. Evans and R.G. Kline, Effect of Hull Girder Stiffness Variations on Ship Structural Performance, SNAME Transactions, Vol. 86, 1978, pp. 101-139.

2. G.C. Volcy, Reduction Gear Damages Related to External Influences, Marine Technology, Vol. 12, No. 4, 1975, pp. 335-366.

3. G.C. Volcy, Deterioration of Marine Gears Teeth; Cases Encountered and Solved, Bureau Veritas, Paris, 1977.

4. G.C. Volcy, Modern Treatment of Stern Gear Behavior in View of Difficulties Encountered in the Past, ISME, 1978.

5. G.C. Volcy, H. Garnier and J.C. Masson, Deformability of the Hull Steel-Work and Deformations of the Engine-Room of Large Tankers, Schiff and Hafen, Vol. II, 1973, pp. 3-15.

6. G.C. Volcy and M. Nakayama, Studies Leading to Vibration and Noise Free Ships, Bulletin Technique du Bureau Veritas, June 1977, pp. 58-62.

7. G. Volcy, Interaction and Compatibility Between Machinery and Hull from a Static and Vibratory Point of View, SNAME Ship Vibration Symposium, Arlington, Va., October, 1978.

8. F. Böhn and J. Simon, Diesel Propulsion Systems of the Future, SNAME Transactions, Vol. 87, 1979, pp. 383-403.

9. Diesels Replace Steam Turbine in Large Tanker, The Naval Architect, January 1980, p. 10.

10. J.H. Evans, Editor, <u>Ship Structural Design Concepts</u>, Cornell Maritime Press, Inc., 1975.

11. G.R. Cowper, The Shear Coefficient in Timoshenko's Beam Theory, J. Applied Mechanics, June 1966, pp. 335-340.

12. H.A. Schade, Hull Strength, Chapter III in <u>Ship Design and Construction</u>, A.M. D'Arcangelo Editor, SNAME, 1969.

13. D. MacIntyre, Aluminum Ore Carriers, SNAME Transactions, Vol. 60, 1952, pp. 544-565.

14. J.H. Evans, Hull Girder Deflections and Stiffness, Chapter 5 in Second Cycle Ship Structural Design Concepts, (in preparation).

15. M.L. Sellers and R.G. Kline, Some Aspects of Ship Stiffness, SNAME Transactions, Vol. 75, 1967, pp. 268-288.

16. E.F. Noonan and S. Feldman, State of the Art for Shipboard Vibration and Noise Control, SNAME Ship Vibration Symposium, Arlington, VA., October 1978.

17. G.C. Volcy, H. Garnier and J.C. Masson, Chain of Static and Vibratory Calculations of Propulsive Plants and Engine Rooms of Ships, Bulletin Technique du Bureau Veritas, June 1975, pp. 60-66, and October 1975, pp. 93-104.

18. G.C. Volcy, M. Baudin and P. Morel, Integrated Treatment of Static and Vibratory Behavior of Twin-Screw 553,000 dwt Tankers, The Naval Architect, July 1979, pp. 197-217.

19. J.L. Taylor, The Theory of Longitudinal Bending of Ships, Transactions NECIES, 1924-25, pp. 123-144.

20. R.T. Bradshaw, The Optimum Alignment of Marine Shafting, Marine Technology, July, 1974, pp. 260-269.

21. Recommendations Designed to Limit the Effects of Vibrations on Board Ships, Bureau Veritas Guidance Note NI 138A - RD3, June 1979.

22. T.W. Bunyan, A Shipowner's Approach to Some Shafting Machinery and Hull Problems, Europort Technical Congress, November 1971.

23. M. Wokik, G. Volcy and F. Benoit, Analytical and Experimental Investigation into a New Crosshead Two-Stroke Engine Frame Structure, 13th International Congress on Combustion Engines, Vienna, 1979.

24. W. Pinnekamp and E. Pollak-Banda, Marine Reduction Gears for Medium Speed Diesel Engines: A New Design Based on the Investigation of Alignment Problems, International Symposium on Marine Engineering, Tokoyo, 1973.

25. H.C. Andersen and J.J. Zrodowski, Co-Ordinated Alignment of Line Shaft, Propulsion Gear and Turbines, SNAME Transactions, Vol. 67, 1959, pp. 449-523.

26. H.W. Semar, Reduction Gears, Chapter IX in Marine Engineering, R.L. Harrington Editor, SNAME, 1971.

27. Investigation of the S.S. "Philippine Bear" for a Comparative Evaluation of the Main Reduction Gear Difficulties as Experienced by the LASH Vessels, American Bureau of Shipping, Final Report RD-74002, February 1975.

28. R.E. Gustafson, Behavior of a Pivoted-Pad Thrust Bearing During Start Up, Journal of Lubrication Technology, Vol. 89, 1967, pp. 124-142.

29. C.L. Long, Propellers, Shafting and Shafting System Vibration Analysis, Chapter XI in Marine Engineering, R.L. Harrington Editor, SNAME, 1971.

30. W.E. Lehr, Jr. and E.L. Parker, Consideration in the Design of Marine Propulsion Shaft Systems, discussion by H.C. Andersen, SNAME Transactions, Vol. 69, 1961, pp. 555-601.

31. R. Glasfeld, D. Jordan, M. Kerr and D. Zoller, Review of Ship Structural Details, Ship Structure Committee Report SSC-266, 1977.

32. C.R. Jordan and C.S. Coshran, In-Service Performance of Structural Details, Ship Structure Committee Report SSC-272, 1978.

33. D. Faulkner, A Review of Effective Plating for Use in the Analysis of Stiffened Plating in Bending and Compression, J. Ship Research, Vol. 19, No. 1, 1975, pp. 1-17.

34. Rules for Building and Classing Steel Vessels, American Bureau of Shipping, New York, 1980.

35. Rules and Regulations for the Classification of Ships, Lloyd's Register of Shipping, London, 1978.

36. Rules for the Construction and Classification of Steel Ships, Det norske Veritas, Oslo, 1976.

37. Rules and Regulations for the Construction and Classification of Steel Vessels, Bureau Veritas, Paris, 1977.

38. Rules for the Classification and Construction of Seagoing Steel Ships, Germanischer Lloyd Hamburg, 1980.

39. Rules and Regulations for the Construction and Classification of Ships, Nippon Kaiji Kyokai, Tokyo, 1978.

40. Rules for the Classification and Construction of Seagoing Ships, USSR Register of Shipping, Leningrad, 1978.

41. D.J. Eyres, Ship Construction, Heinemann, London, 1978.

42. J.H. Wright, Longitudinal Stiffness of Marine Propulsion Thrust Bearing Foundations, ASNE Journal, February 1959, pp. 103-106.

43. R.M. Cashman, Design of Marine Machinery Foundations, SNAME Transactions, Vol. 70, 1962, pp. 723-748.

44. A.J. Johnson and W. McClimont, Machinery Induced Vibrations, IME Transactions, 1963.

45. J.R. Kane and R.T. McGoldrick, Longitudinal Vibratoins of Marine Propulsion - Shafting System, SNAME Transactions, Vol. 57, 1949, pp.

46. L. Vassilopoulos and F.M. Hamilton, Longitudinal Stiffness Analysis for the Propulsion Shafting Systems of th Polar Class Icebreakers, Presented at the ASNE Annual Meeting, May 1980.

47. W. Boylan, Marine Application of Dental Couplings, SNAME Spring Meeting, Marine Power Plants, 1966.

48. V.A. De George, Allowable Lineshaft Alignment in Both the Vertical and Horizontal Directions Based on their Combined Effect on Bull Gear to Driving Pinion Tooth Loads, General Electic Company, April 1980.

49. G. Bourceau and G. Volcy, Some Aspects of the Behavior in Service of Crankshafts and their Bearings,ATMA, Paris 1966.

50. G.C. Volcy and A. Trivouss, The Crankshaft and Its Curved Alignment, Joint Session S.T.G. and A. Te. N. Venice, Trieste 1972.

51. M. Lanballe, E. Aasen and T. Mellem, Application of the Finite Element Method to Machinery, Computers and Structures, Vol. 4, 1974, pp. 149-192.

52. J.P. Comstock, Editor, Principles of Naval Architecture SNAME 1967.

53. M. Benicourt, Methode de Calcul Classique: Efforts sur le Navire en Eau Calme, Charpente du Navire, Ecole Nationale Superieure de Techniques Avances, 1972.

54. W.R. Spillers, Automated Structural Analysis: An Introduction, Pergamon Press Inc., 1972.

55. Longitudinal Stiffness of Main Thrust Bearing Foundations, Technical and Research Report R-15, SNAME, September 1972.

56. J.Luvisi, Calculation of Ship Propulsion Shafting, Bearing Reactions on a Datatron 205 Computer, Development Report No. R-24, Boston Naval Shipyard, Boston, Massachusetts, 1960.

57. R.D. Cook, Concepts and Applications of Finite-Element Analysis, John Wiley & Sons, 1974.

58. S.G. Stiansen, H.Y. Jan, and D. Liu, Dynamic Stress Correlation for the S1-7 Containership, SNAME Transactions, Vol. 87, 1979 pp. 64-98.

59. K.J. Bathe, ADINA, A Finite-Element Program for Automatic Dynamic Incremental Nonlinear Analysis, M.I.T. Acoustics and Vibration Laboratory Report 82448-1, 1978.

60. R. Szilard, Theory and Analysis of Plates; Classical and Numerical Methods, Prentice-Hall, Inc., 1975.

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APPENDIX A: REVIEW OF CLASSIFICATION SOCIETY RULES

The rules of the following Classification Societies are included in this study:

(a) American Bureau of Shipping [34]
(b) Lloyd's Register of Shipping [35]
(c) Det norske Veritas [36]
(d) Bureau Veritas [37]
(e) Germanischer Lloyd [38]

The review presented here does not obviously include an exhaustive discussion of the various rules. It deals strictly with requirements concerning the ship structure in way of machinery spaces, and only those aspects considered to be of particular interest to the specific problem of hull-machinery compatibility are included, sometimes in condensed form. A direct attempt to compare the rules will not be made, since it is well known that the philosophy of design on which each Classification Society bases its requirements is different and has its own merits.

Only double-bottom construction will be discussed, since this is the type of structural arrangement used in large ships. The rules usually give requirements for high-strength materials, but these will not be considered here, since these materials are not widely used in the machinery spaces of large commercial vessels. The structural arrangement of shaft tunnels and machinery casings, a subject covered in all the rules, will not be discussed since it is not particularly relevant for this study. The same happens with respect to watertight bulkheads, which are required by all the rules to form the forward and after boundaries of the machinery spaces, and the structural strengthening of openings in way of machinery spaces. Similarly, structural details such as those involved in the attachment of web frames to the inner bottom or the machinery bolting arrangements will not be considered, since they do not affect to a large extent the overall structural rigidity of the foundation. Besides, as stated earlier, the whole area of ship structural details has already been the subject of extensive research [31, 32], so that there is no need to review it again here.

Whenever possible, the main sections in which specific requirements are defined are identified in parenthesis, so that if desired the reader can refer to them for complete information.

A.1 American Bureau of Shipping

In view of the effect upon the structure of the necessary openings in the machinery space, the difficulty of securing adequate support for the decks, of maintaining the stiffness of sides and bottom and of distributing the weight of the machinery, special attention is directed to the need for arranging, in the early stages of design, for the provision of plated through beams and such casing and pillar supports as are required to secure structural efficiency; careful attention to these features in design and construction is to be regarded as of the utmost importance. All parts of the machinery, shafting, etc., are to be efficiently supported and the adjacent structure is to be adequately stiffened. In twin-screw vessels and in other vessels of high power, it will be necessary to make additions to the strength of the structure and the area of attachments, which are proportional to the weight, power and proportions of the machinery, more especially where the engines are relatively high in proportion to the width of the bed plate. A determination is to be made to assure that the foundations for main propulsion units, reduction gears, shaft and thrust bearings, and the structures supporting those foundations are adequate to maintain required alignment and rigidity under all anticipated conditions of loading (19.1)

The engines are to be seated directly upon thick inner bottom plating or upon thick seat plates on top of heavy foundations arranged to distribute the weight effectively. Additional intercostal girders are to be fitted within the double bottom to ensure the satisfactory distribution of the weight and the rigidity of the structure (19.3.2).

Boilers are to be supported by deep saddle-type floors or by transverse or fore and aft girders arranged to distribute the weight effectively. If the boilers are supported by transverse saddles or girders the floors in way of boilers are to be suitably increased in thickness and specially stiffened. Proper accessibility and ventilation have to be ensured, and the thickness of adjacent material is to be increased as required, where the clear space is less than recommended (19.5).

Thrust blocks are to be bolted to efficient foundations extending well beyond the thrust blocks and arranged to distribute the loads effectively into the adjacent structure. Extra intercostal girders, effectively attached, are to be fitted in way of the foundations as may be required (19.7).

Shaft stools and auxiliary foundations are to be of ample strength and stiffness in proportion to the supported weight (19.9).

Special provisions are given in the rules regarding the arrangement and scantlings of bottom structure in way of machinery spaces.

Solid floors are to be fitted on every frame under machinery and transverse boiler bearers. Their minimum thickness t is

$$t = 0.036L + 6.2 \text{ mm} \tag{1}$$

where L, the rule length is m in such that $L \leq 427$ m. Where boilers are mounted on the tank top the thickness of the floors and intercostals in way of the boilers is to be increased 1.5 mm above engine space requirements (7.3.4).

The inner bottom plating minimum thickness t in way of engine spaces is given by

$$t = 0.037L + 0.009s + 1.5 \text{ mm} \tag{2}$$

where the rule length is $L \leq 427$ m and s is the frame spacing in mm (7.5.1).

Under boilers, there is to be a clear space of at least 457 mm. Where the clear space is necessarily less, the thickness of the plating is to be increased as may be required (7.5.3).

In way of engine-bed plates or thrust blocks which are bolted directly to the inner bottom, the plating is to be at least 19 mm thick. The thickness is to be increased according to the size and power of the engine (7.5.4).

Side girders with minimum thickness t given by equation (1) are to be so arranged that the distance from the center girder to the first girder, between the girders or from the outboard girder to the center of the margin plate does not exceed 4.57 m. Additional full or half-depth girders are to be fitted beneath the inner bottom as required in way of machinery and thrust seatings and beneath wide-space pillars. Where the bottom and inner bottom are longitudinally framed, this requirement may be modified (7.9).

The rules stress the need to provide sufficient transverse strength and stiffness in the machinery space by means of webs, plated through beams, and heavy pillars in way of deck openings and casings (8.15).

'Tween-deck webs are to be fitted below the bulkhead deck over the hold webs as may be required to provide continuity of transverse strength above the main webs in the holds and machinery space (9.7).

Special support provided by stanchions or pillars or by means which are not less effective is to be arranged at the ends and corners of deckhouses, in machinery spaces, at ends of partial superstructures and under heavy concentrated weights (11.1).

Under boilers, the plating of effective deck is to be at least 15 mm in thickness (16.5.8).

For tankers, machinery spaces aft are to be specially stiffened transversely. Longitudinal material at the break is also to be specially considered to reduce concentrated stress in this region. Longitudinal wing bulkheads are to be incorporated with the machinery casings or with substantial accomodation bulkheads in the 'tween decks and within the poop (22.15).

A.2 Lloyd's Register of Shipping

Except where otherwise noted the rule sections given in parenthesis are taken from Part 3, Chapter 7.

The rules make a distinction between three basic locations for machinery spaces, namely: midship region, aft region with a cargo compartment between it and the after peak bulkhead, and aft region with the after peak bulkhead forming the aft end of the machinery space (1.1.2).

If the machinery spaces are amidships and the shell and decks outside the line of openings are longitudinally framed in way of adjacent cargo spaces, the machinery space is also to be longitudinally framed (1.2.2).

In longitudinally framed machinery spaces the maximum spacing S_{max} of transverses is given by

$$S_{max} = 3.8 \text{ m} \qquad L \leq 100 \text{ m}$$

$$S_{max} = (0.006L + 3.2)\text{m} \qquad L \geq 100 \text{ m}$$

where L is the rule length in m. In way of a machinery space situated adjacent to the after peak, the spacing is not to exceed five transverse frame spacings (1.2.4).

The rules emphasize the need for suitable structural continuity of the machinery spaces, suitable deck strengthening in way of machinery openings, and suitable support systems for deck beams (in transversely framed ships), deck longitudinals (in longitudinally framed ships) and machinery casings. Also, in way of concentrated loads such as those from boiler bearers or heavy auxiliary machinery, the scantlings of lower decks or flats must be specially considered taking the actual loading into account (1.3, 2.1, 2.2 and 2.3).

If the machinery space is amidships, web frames should be fitted and spaced not more than five frame spaces apart, and extending from the tank top to the level of the lowest deck above the load waterline. The scantlings should be such that the combined section modulus of the web frame and the main or 'tween deck frames is 50% greater than that required for normal transverse framing. These web frames can be omitted if the section modulus of the ordinary main or 'tween deck frames is to be increased by 50%, up to the level of the lowest deck above the load waterline. Where fully effective stringers supported by web frames are fitted, the stringers may be considered as decks for the purpose of calculating the modulus of the frames (3.1.1 and 3.2.3).

Except where the machinery is adjacent to the after peak, longitudinal framing should have the same scantlings as for cargo spaces. For machinery spaces adjacent to the after peak, the section modulus Z of side longitudinals is given by

$$Z = 0.0065\ ks\ell_e^{\,2}\ \ (h+0.167D)\ \ cm^3$$

where k is higher tensile steel factor (equal to unity for mild steel), s is the spacing of floors and longitudinals in mm, ℓ_e is the effective length of the stiffening member in m, h is the load head in m, from mid-span to upper deck at side amidships (not less than 0.9m) and D is the rule depth (not more than 20 m) (3.1.2).

If the space is situated in the aft region and transverse framing is adopted, web frames are to be fitted in general not more than five frame spaces apart, extending from the tank top to the upper deck. A spacing up to seven transverse frame spaces is also acceptable if the ordinary frames are substantially strengthened to satisfy the overall modulus and inertia requirements. The scantlings of web frames below lowest deck and not supporting effective stringers are to be governed by the following minimum section modulus:

$$Z = 5KSh\ell_e^{\,2}$$

Above the lowest deck Z is given by:

$$Z = 1.68\ CkTS\ell_e\ \ \sqrt{D}$$

where S is the spacing or mean spacing of web frames or transverse in m, C is a parameter taking the value 2.2 for a lower 'tween deck and 2.0 for an upper 'tween deck, T is the rule draft and the remaining symbols have already been defined. The minimum web depth to be used in conjunction with these two expressions for Z is 2.5 times the depth of adjacent main frames (3.2.1 and Table 7.3.1).

If the span of ordinary frames below the lowest deck or flat exceeds 6.5 m, one or more fully effective side stringers are to be fitted to support the frames. The scantlings are to satisfy the following minimum section modulus Z:

$$Z = 7.75\ kSH\ell_e^{\,2}$$

where all the symbols have already been defined. The minimum web depth in this case is two and a half times the depth of adjacent main frames (3.2.1 and Table 7.3.1). An arrangement of light stringers spaced about 2.5 m apart may also be accepted as an alternative to the fully effective stringers just defined (3.2.2).

If the machinery space is not in the after end region, the web frames below the lowest deck supporting effective stringers are to be found from the following assumptions: fixed ends, point loadings, head to upper deck at side, bending stress 93.2 N/mm^2 and shear stress 83.4 N/mm^2. Again the minimum web depth is equal to 2.5 times the depth of adjacent main frames (Table 7.3.1).

If the machinery is longitudinally framed, side transverses are to be fitted. Below the lowest deck, their scantlings are governed by the following minimum section modulus.

$$Z = 10k\ Sh\ell_e^{\,2}$$

and above the lowest deck by

$$Z = 2.1\ CkTS\ell_e\ \sqrt{D}$$

where all the parameters have already been defined. In both cases, the minimum web depth is 2.5 times the depth of the longitudinals. Suitable connections at top and bottom are to be provided to the web frames (3.3.1 and Table 7.3.1).

The minimum depth of the center girder d_{DB} is

$$d_{DB} = 28\ B + 205\ \sqrt{T}$$

where B is the rule breadth and T the draft. d_{DB} should not be less than 650 mm (Part 4, Chapter 1, 8.3.1). A greater depth is recommended in way of large engine rooms when the variation in draft between light and loaded conditions is considerable (4.1.1).

The minimum center girder thickness t is

$$t = (0.008\ d_{DB} + 4)\ \sqrt{k}$$

t should not be less than 6 mm (Part 4, Chapter 1, 8.3.1).

In machinery spaces adjacent to the after peak, the double bottom is to be transversely framed. Elsewhere, transverse or longitudinal framing may be adopted, but for ships exceeding 120 m in length and for ships strengthened for heavy cargos, longitudinal framing is in general to be used (4.1.2 and Part 4, Chapter 1, 8.2.1).

If the double bottom is transversely framed, plate floors are to be fitted at every frame in the engine room. In way of boilers, plate floors are to be fitted under the boiler bearers (4.1.3). If the double bottom is longitudinally framed, plate floors are to be fitted at every frame under the main engines and thrust bearings. Outboard of the engine seating floors may be fitted at alternate frames (4.1.4).

The scantlings of floors clear of the main engine seatings are generally to be as required in way of cargo spaces. In way of engine seatings, the floor minimum thickness is given by:

$$t_3 = (10 + 1.5\ f)\ \text{mm}$$

where f is the engine factor given by $P/R\ell$. P is the power of one engine at maximum service speed in KW, R is the rev/min of engine at maximum service speed, and ℓ is the effective length of engine foundation plate in m required for bolting the engine to the seating. In determining ℓ, the thrust and gearcase seating is to be considered as a separate item (4.1.5, 6.2.1 and Table 7.6.1).

Sufficient fore and aft girders are to be arranged in way of the main machinery to effectively distribute its weight and to ensure adequate rigidity of the structure. In midship machinery spaces, these girders are to extend for the full length of the space and are to be carried aft to support the foremost shaft tunnel bearing. This extension beyond the after bulkhead of the machinery space is to be for at least three transverse frame spaces, aft of which the girders are to scarf into the structure. Forward of the engine room forward bulkhead, the girders are to be tapered off over three frame spaces and effectively scarfed into the structure. In machinery spaces situated at the aft end, the girders are to be carried as far aft as practicable and the ends effectively supported by web frames or transverses (4.1.6).

Outboard of the engines, side girders are to be arranged where practicable to line up with the side girders in adjacent cargo spaces (4.1.7).

Where the double bottom is longitudinally framed and transverse floors are fitted in way of the engine seatings as required by the rules, no additional longitudinal stiffening is required in way of the engines other than the main engine girders, provided that the spacing of girders does not exceed 1.5 times the normal spacing of longitudinals. Where this spacing of girders is exceeded, shell longitudinals are to be fitted, and these are to scarf into the longitudinal framing clear of the machinery spaces (4.1.8).

The minimum thickness t of inner bottom plating in engine rooms clear of engine seatings is

$$t = 0.0015\ \sqrt[4]{Ltk^2}\ (S + 660)\ \text{mm}$$

t should not be less than 7.0 mm (4.1.9). In way of engine seatings integral with the tank top, the minimum thickness as given by Table 7.6.1 is

$$t = (19 + 3.4\ f)\ \text{mm}$$

if two girders are fitted and

$$t = (25 + 3.4\ f)\ \text{mm}$$

if one girder is fitted. f is the engine factor already defined, and A is the area of top plate in cm^2 for one side of seat, given by:

$$A = (120 + 44.2f + 4.07f^2)\ \text{cm}^2$$

The main engine girder total thickness for the case where two girders are fitted is

$$t_1 + t_2 = (28 + 4.08f)\ \text{mm}$$

If one girder only is fitted, we have

$$t_1 = (15 + 4.08f)\ \text{mm}$$

In general, one single girder can be accepted when all the following conditions apply (Table 7.6.1): $f < 1.84$, $P < 5900$ KW and $L < 100$ m.

Where the height of inner bottom in the machinery space differs from that in adjacent spaces, continuity of longitudinal material is to be maintained by sloping the inner bottom over an adequate longitudinal extent. The knuckles in the plating are to be arranged close to plate floors (4.1.10).

A.3 Det norske Veritas

All the rule sections given in parenthesis are taken from Chapter II.

The height of the center girder is to be sufficient to give good access to all parts of the double bottom and it is not to be less than:

$$h = 600 + 9B\sqrt{d}\ \text{mm}$$

where B is the greatest moulded breadth in m and d the mean moulded summer draft in m. In the engine room, the height of the tank top above the keel should be 45% and 30% greater than the required center girder height, respectively, with and without a sump in way of the main engine (Section 10, A305).

The thickness of inner bottom plating in engine and boiler room is not to be less than

$$t = \frac{(3.5 + 0.023L) \quad (S + 0.8)}{\sqrt{f_1}} \text{ mm}$$

where L is the rule length, s is the frame spacing in m and f_1 a material factor depending on the material (f_1 is equal to unity for mild steel) (Section 10, B501).

The thickness of the center girder in the engine room is not to be less than (Section 10, Table D103):

$$\frac{6.5 + 0.05L}{\sqrt{f_1}}$$

The thickness of side girders, floors and brackets in the engine room is not to be less than (Section 10, Table D103):

$$\frac{6 + 0.035L}{\sqrt{f_1}}$$

In general, side girders are to be fitted so that the distance between the side and center girders or the margin plate or between side girders does not exceed the following values: 5 m in longitudinally stiffened double bottom and 4 m in transversely stiffened double bottom (Section 10, D201).

Girders are to be fitted under the machinery extending from the bottom to the engine-seating top plate. If the engine bed plates are bolted directly to the inner bottom, the thickness of plating under the engines is to be at least twice the rule thickness of inner bottom plating. At least one side girder is to be fitted outside the engine seating girders (Section 10, D202).

In the engine room, if the double bottom is longitudinally stiffened, floors are to be fitted at every second side frame. Bracket floors are to be fitted at intermediate frames extending to the first ordinary side girder outside the engine seating. In way of thrust bearing and below pillars, additional strengthening is to be provided (Section 10, D301).

If the double bottom is transversely stiffened, floors are to be fitted at every frame. In way of thrust bearing and below pillars, additional strengthening is to be provided (Section 10, D304).

Verticals in the engine room are to have a depth not less than 200 ℓ_1 mm where ℓ_1 is the span of the girder in m (Section 12, D101).

Girder flanges are to have a thickness not less than 1/30 of the flanges width when the flange is symmetrical, and not less than 1/15 when the flange is asymmetrical. The total flange width in the engine room is not to be less than 35ℓ mm (Section 12, D102).

In the engine and boiler room, side verticals are normally to be fitted at every fifth frame. For diesel machinery with a large number of cylinders and for turbine machinery, every fourth side vertical is normally to be replaced by a bulkhead between the ship's side and the supports under the casing side from the bottom to the lowest continuous deck (Section 12, D105).

A.4 Bureau Veritas

The rules first consider the particular case of cargo ships. If the double bottom is transversely framed, the minimum thickness of strakes in the inner bottom in way of the engine room is

$$e = 0.75 \sqrt{L + 10T_2} + 1.5 \text{ mm (minimum 7 mm)}$$

and in way of the boiler compartment:

$$e = 0.75 \sqrt{L + 10T_2} + 3.5 \text{ mm (minimum 9 mm)}$$

where L is the rule length and T_2 the draft. If the ship is longitudinally framed, the thickness is that given above but reduced by 0.5 mm (6.33.11).

If the spacing E of stiffeners (floors or longitudinals in m) is greater (or smaller) than the basic spacing E_o, the values are to be increased (or reduced) by $20(E-E_o)/3$ (6.33.12). E_o is the rule frame spacing in m, given by (6.12.11)

$$E_o = 0.72 \left(\frac{L}{100}\right)^{1/4}$$

In no case, should the thickness of the inner bottom plating be less than:

transverse framing: $e = 5.26\ E\ \sqrt{h}$

longitudinal framing: $e = 4.45\ E\ \sqrt{h}$

where $h = C_1 - H_D$, C_1 is the depth of ship in m to the deck below the top of overflow and H_D is the double-bottom depth in m (6.33.13).

In the engine and boiler space, the margin plate thickness is not to be less than that required for adjoining inner bottom plates (6.33.15).

The depth of the center girder is generally equal to (6.33.21):

$$b = 0.1\ \sqrt{L}$$

and the thickness is not to be less than (6.33.22):

midship region: $e = 0.95\ \sqrt{L + 10T_2}$ (minimum 7mm)

ends: $e = 0.80\ \sqrt{L + 10T_2}$ (minimum 6mm)

The side girder thickness is not to be less than (6.33.24):

$$e = 0.7\ \sqrt{L + 10T_2} + 1 \text{ (minimum 7mm)}$$

Under the engine bed plates, additional girders are to be included. Under the thrust block, side girders are to be arranged to the satisfaction of the Head Office. Under the engine seatings, additional girders are to be arranged (6.33.25 and 6.33.75).

In the case of transverse framing, the thickness of plate floors is not to be less than

$$e = 0.7\ \sqrt{L + 10T_2} + 1 \text{ (minimum 7mm)}$$

In the case of longitudinal framing, the thickness is that given above increased by 10% (6.33.31).

In ships over 100 m in length or where the depth of floors exceeds 0.95m, floors are to be fitted with vertical stiffeners (6.33.84).

The number of side girders is to be such that the distance separating them from either another or the center girder or margin plate does not exceed 4.2m (6.33.74).

In transversely framed systems, plate floors are to be fitted at every frame within the machinery space, under the thrust block and under the boiler bearers. Under the main engines and auxiliaries, care must be taken to ensure that all the double-bottom items are well fitted (6.33.8).

In longitudinal systems, the floor spacing should be two frame spaces within the machinery and boiler spaces, one frame space in way of the main internal combustion engine and thrust blocks. In this case, bottom longitudinals may be reduced in scantlings (6.33.9).

In the engine room, web frames are to be generally fitted every fifth frame. The web frame depth is to be not less than twice that of the frame replaced and to have a section modulus not less than four times that of the frame (6.44.13).

The section modulus of frames is not to be less than that of the 'tweendeck or superstructure frame located just above nor than (6.43.21):

$$w = 3.5hE\ell^2$$

where h is the design load height in m, E is the frame spacing in m, and the span ℓ in m is to be measured between the level of the top of floors or tank top and the lowermost deck line. The loading height h is given by (6.42.11.):

$$h = h_s + 0.4\ h_o$$

with $h_o = 8.143 - 0.714\ \left(\frac{300-L}{100}\right)^{3/2}$ if $L \leq 300$

$h_o = 8.143$ if $L > 300$

$h_s = 0.6\ \frac{b^2}{\ell_1}$ where the frame is partly immersed

$h_s = b - 0.4\ell_1$ when the frame is entirely immersed

ℓ_1 and b_1 are vertical distances in m, measured between the intersection of the side shell, with the extension of the top of floor or the inner bottom plating, and respectively the lower deckline side and the rule waterline.

If web frames are not provided, the section modulus should not be less than the value obtained from the formula given above increased by 15% where the engine room is not aft, or 40% if the engine room is aft (6.43.51).

If the particular case of tankers, the rules indicate that attention should specially be directed to the rigidity of the framing in the machinery space, particularly when high-power diesel engines are used (8.51.23).

In the double bottom, floors are to be provided at each frame, and girders in line with the bottom longitudinals in the cargo tank space are also to be provided. Where necessary, additional girders extending over a few frames are to be provided every two or three longitudinals, so as to ensure structure continuity (3.54.21).

The thickness of the inner bottom plating is not to be less than (8.54.22):

$$e = 0.75 \sqrt{L + 10T} + 2.5$$

The thickness derived from the formula is not to exceed 17mm unless the overflow depth of the double bottom tanks justifies a greater thickness.

The thickness of ordinary floors is not to be less than (8.54.23):

$$e = 0.7 \sqrt{L + 10T} + 1$$

and it need not exceed 16 mm.

The thickness of watertight floors may be taken equal to that of ordinary floors increased by 1.5 mm, provided there are stiffeners spaced about 0.76 m and having a section modulus at least equal to

$$w = 6.5 H_D^2 h$$

where H_D is the double bottom depth in m and h is the distance in m between the tank top and the top of the overflow (8.54.24).

Girders, floors and inner bottom plating are to be suitably strengthened in way of the engines, reduction gears, thrust blocks, etc. (8.54.25).

As a rule, on the side shell plating, web frames are to be provided four frame spaces apart. The section modulus of web frames is not to be less than

$$w = 6E\ell^2 d$$

where ℓ is the span of web frames in m measured between flats or between the lower flat and the tank top, and d is the vertical distance in m between midspan and the main deck (8.54.31 and 32).

Where the side shell is framed transversely, the scantlings of the stringers are such that the section modulus is not to be less than

$$w = 10E^2 bd$$

where E is the web frame spacing in m, b is the width supported by the stringers in m and d is the vertical distance in m from the center of the area supported by the stringer to the deckline at side, without this distance being taken less than 4 (8.54.34 and 8.53.24).

A.5 Germanischer Lloyd

The rule sections mentioned below are taken from Chapter 2 (Construction Rules for Hull).

The rules specify that lightening holes in way of the engine foundation are to be kept as small as possible while keeping good accessibility. Where necessary, the edges of lightening holes are to be strengthened by means of face bars or the plate panels are to be stiffened (8.C.2.1.1).

Local strengthenings are to be provided beside the following minimum requirements according to the construction and the local conditions (8.C.2.1.2).

Plate floors are to be fitted at every frame. The minimum floor thickness in compartments other than machinery compartments is given by (8.B.7.2.1).

$$t = h/100 - 1.0 \text{ mm for } h \leq 1200 \text{ mm}$$

$$t = h/120 + 1.0 \text{ mm for } h > 1200$$

t need not exceed 16.0 mm. h is the depth of center girder in mm and its minimum value is given by (8.B.2.2).

$$h = 350 + 0.45B \text{ mm} > 600 \text{ mm}$$

where b is the greatest moulded breadth.

In machinery spaces, the floor thickness as given by the above expressions is to be strengthened as follows:

$$3.6 + N/500 \text{ (per cent)}$$

minimum 5%, maximum 15%, where N is the single engine output in KW (8.C.2.2).

At least one side girder shall be fitted in the engine room. The distance of the side girders from each other and from center girder and margin plate respectively shall not be greater than 1.8 m in the engine room within the breadth of engine seatings (8.B.3.1).

The thickness of side girders under an engine foundation top plate inserted into the inner bottom is to be similar to the thickness of side girders above the inner bottom as defined below (8.C.2.3.1).

Side girders with the thickness of longitudinal girders are to be fitted under the foundation girders in full height of the double bottom. Where two side girders are fitted on either side of the engine, one may be a half-height girder under the inner bottom for engines up to 3000 KW (8.C.2.3.2).

Side girders under foundation girders are to be extended into the adjacent spaces and to be connected to the bottom structure. This extension aft and forward of the engine room bulkheads shall be 2-4 frame spaces if practicable (with machinery aft, only forward of the engine room) (8.C.2.3.3).

No center girder is required in way of the engine seating but intercostal docking profiles are to be fitted instead. The sectional area of the docking profiles is not to be less than (8.C.2.3.4 and 8.C.1.4).

$$f = 10 + 0.2L$$

where L is the rule length. Docking profiles are not required where a bar keel is fitted. Brackets connecting the floor plates to the bar keel are to be fitted on either side of the floors.

Between the foundation girders, the thickness of the inner bottom plating is to be increased by 2 mm over the value it has in other locations. The strengthened plate is to be extended beyond the engine seating by three to five frame spacings (8.C.2.4).

Regarding the design of engine seatings, the rules give some recommendations which apply to low-speed engines. Seatings for medium and high-speed engines as well as for turbines must be specially considered (8.C.3.1.1).

The rigidity of the engine seating and the surrounding bottom structure must be adequate to keep the deformations of the system due to the loads within the permissible limits. In special cases, evidences may be required of deformations and streses (8.C.3.1.2).

Regarding the foundation of diesel engines, the rules offer the following guidance (8.C.3.1.2):

At the draught resulting in the maximum deflection in way of the foundation, the deflection of two stroke, cross-head engines including foundation ought to be less than 1 mm over the length of the engine. In addition to the deflection of engine and foundation, the crank web deflections by which the admissible engine deflection may be limited to values much less than 1 mm have to be considered as well. For medium-speed and high-speed engines, not only the deflections of crank webs have to be taken into account, but for assuring trouble-free bearing conditions of the crank shaft the bending deflections of the engine is to be limited.

Due regard is to be paid, at the initial design stage, to a good transmission of forces in transverse and longitudinal direction (8.C.3.1.3).

The foundation bolts for fastening the engine at the seating shall be spaced no more than 3 d apart from the longitudinal foundation girder. Where the distance of the foundation bolts from the longitudinal foundation girder is greater, proof of equivalence is to be provided. d is the diameter of the foundation bolts (8.C.3.1.4).

In the whole speed range of main propulsion installations for continuous service, resonance vibrations with inadmissible vibration amplitudes must not occur; if necessary structural variations have to be provided for avoiding resonance frequencies. Otherwise, a barred speed range has to be fixed. Within a range of -10% to +5% related to the rated speed no barred speed range is permitted. The Society reserves the right to demand in special cases a proof of vibration-free service (8.C.2.1.5).

The thickness of the longitudinal girders above the inner bottom is not to be less than:

$$t = \sqrt{N/15} + 6 \text{ mm for } N < 1500 \text{ KW}$$

$$t = N/750 + 14 \text{ mm for } 1500 \leq N \quad 7500 \text{ KW}$$

$$t = N/1875 + 20 \text{ mm for } N \geq 7500 \text{ NW}$$

where N is the single engine output in KW (8.C.3.2.1).

Where two longitudinal girders are fitted on either side of the engine, their thickness as given by the formulas above may be reduced by 4 mm (8.C.3.2.2).

The sizes of the top plate (width and thickness) shall be sufficient to attain efficient attachment and seating of the engine and depending on seating height and type of engine adequate transverse rigidity.

The thickness of the top plate shall approximately be equal to the diameter of the fitted-in bolts. The cross-sectional area of the top plate is not to be less than:

$$F_T = N/15 + 30 \text{ cm}^2 \text{ for } N \leq 750 \text{ KW}$$

$$F_T = N/75 + 70 \text{ cm}^2 \text{ for } N > 750 \text{ KW}$$

Where twin engines are fitted, a continuous top plate is to be arranged in general if the engines are coupled to one propeller shaft (8.C.3.2.3).

The longitudinal girders of the engine seating are to be supported transversely by means of web frames or wing bulkheads (8.C.3.2.4).

Top plates are preferably to be connected to longitudinal and transverse girders thicker than approximately 15 mm by means of a double bevel butt joint (8.C.3.2.5).

In the engine and boiler room, web frames are to be fitted. Generally, they should extend up to the uppermost continuous deck. Where the depth is 4 m, the web frames are to be spaced 3.5 m apart on an average, where the depth if 14 m, they are to be spaced 4.5 m apart on an average (9.A.8.1.1).

For combustion engines up to about 400 kW, the web frames shall generally be fitted at the forward and aft ends of the engine. For combustion engines of 400 to 1500 kW, an additional web frame is to be provided at half length of the engine, and for engines with higher outputs, at least two further web frames are to be provided (9.A.8.1.2).

Where combustion engines are fitted aft, stringers spaced 2.6 m apart are to be fitted in the engine room, in alignment with the stringers in the after peak, if any, or else, the main frames are to be adequately strengthened. The scantlings of the stringers shall be similar to those of the web frame. At least one stringer is required where the depth up to the lowest deck is less than 4 m (9.A.8.1.3).

The section modulus of the web frames is not to be less than (9.A.8.2):

$$w = k\ 0.8\ e\ \ell^2\ p_s\ \mathrm{cm}^3$$

where k is a material factor (equal to unity for ordinary hull structural steel), e is the web frames spacing in m, ℓ is the span and p_s is the load in KN/m^2 on the ships side.

The moment of inertia of the web frame is not to be less than:

$$J = H(4.5H - 3.75)\ c\ 10^2\ \mathrm{cm}^4$$

where $3m \leq H \leq 10m$

$$J = H(7.25H - 31)\ c\ 10^2\ \mathrm{cm}^4$$

where $H > 10m$

$$c = 1 + (H_u - 4)\ 0.07$$

where H is the rule depth and H_u is the depth measured to the lowest deck in m. The scantlings of the webs (depth h and thickness t) are to be calculated as follows:

$$h = 50.H\ \mathrm{mm} > 250\ \mathrm{mm}$$

$$t = h/(32 + 0.03h)\ \mathrm{mm} > 8.0\ \mathrm{mm}$$

Ships with a depth of less than 3 m are to have web frames with web scantlings not less than 250 x 8 mm and a minimum face sectional area of 12 m^2.

APPENDIX B: BEARING REACTIONS COMPUTER PROGRAM

The Boston Naval Shipyard (BNS) Computer Code [56] is used to compute the bearing reactions of a ship's propulsion shafting system. Bearing reactions are computed when all the bearings are on a straight line. Alternatively, the reactions for other than the straight line conditions can be calculated using a matrix of influence numbers which are computed by the BNS Computer Code. Thus, the effect on the bearing reactions magnitude of raising or lowering any particular bearing can be found using the influence number table. The BNS Computer Code also computes at given points of the shafting system, the shear force, bending moment, slope and transverse deflection value. The details of the theoretical procedure for the Shafting calculation can be found in Reference [56].

In order to prepare the input data for the BNS Computer Code, the shafting system must be divided into a number of uniform sections. For each section of the shaft the following must be specified:

- length of shaft section
- outside diameter
- second diameter
- material

The computer output consists of the following:

- shear force
- bending moment
- slope
- transverse deflection

Figure 32 illustrates the usefulness of the computer code ANALYSIS for hull-machinery compatibility studies. The engine-room double-bottom flexibility matrix $\underset{\sim}{F}$ is computed by the ANALYSIS code. The matrix $\underset{\sim}{I}$ of influence coefficients is computed using the Boston Naval Shipyard Computer Code [56].

APPENDIX C: COMPUTER CODE ANALYSIS

The deformation of the engine room structure at any point has the following contributions:

- deformation of hull girder
- deformation of 2-D frame
- deformation of double-bottom grillage structure

The software computer code developed consists of the following modules:

- GIRDER for hull girder analysis
- SPACE for 2-D frame and grillage analysis of double bottom

The computer code ANALYSIS contains the modules GIRDER and SPACE. Any of the modules can be executed any number of times and in the desired order. The modules are written in Fortran IV.

The input parameters to the ANALYSIS Computer code are described in the following. A computer listing is provided thereafter.

ANALYSIS can accept input in any dimensions. However, the input should be dimensionally compatible and the same dimensions should be used for the entire package.

Below the main heading, a 'check for dimensions' is printed out. The user should make sure the dimensions are compatible:

ALL INPUT IS FORMAT FREE EXCEPT WHERE GIVEN.

Card 1: DLEN, DWET

DLEN: dimension for length (4 characters) $- \left\{ \begin{array}{l} \text{FEET} \\ \text{INCH} \\ \text{METER} \end{array} \right.$

DWET: dimension for weight (5 characters) $- \left\{ \begin{array}{l} \text{POUND} \\ \text{TON} \\ \text{KG} \end{array} \right.$

CARD 2: PRM

PRM: Program module to be executed

'GIRDER' for girder analysis
'FRAME' for frame analysis
'GRILLAGE' for grillage analysis

Depending on the value of PRM - go to the appropriate section for input parameters.

GIRDER - Input Parameters

CARD 1: ALBP, PMEA, PMEF, WTLS, CGLS

ALBP: Length between perpendiculars
PMEA: Aft end of parallel middle body from AP
PMEF: Forward end of parallel middle body from AP
WTLS: Distributed hull weight
CGLS: Distributed hull weight C.G. from AP

CARD 2: YEM, GEE, FØF

YEM: Young modulus of elasticity
GEE: Shear modulus of elasticity
FØF: Form factor for shear deflection

CARD 3: THM, TMLC

THM: Thrust moment
TMLC: Location of thrust moment from AP

CARD 4: NBP

NBP: No of points for which buoyancy values are input - max 20

If there is a discontinuity in buoyancy curve input 2 points for the same location

eg:
TREATMENT OF DISCONTINUITY IN BUOYANCY CURVE

x = input 2 points here.

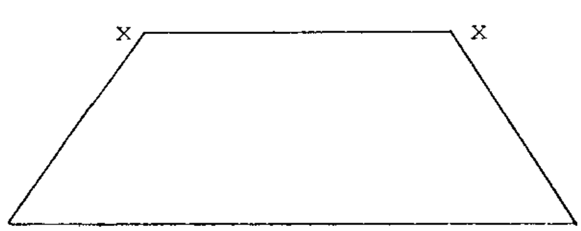

Figure 48

CARD 5: BPL

BPL: Distance of buoyancy points from AP

CARD 6: BPV

BPV: Buoyancy per unit length value at each point

CARD 7: NS

NS: No. of semi-distributed weight items - max 50

CARD 8: NU, ANL1, ANL2, ANL3, ANL4, ANL5, WL, GLL, DLL, AL - (formatted input)

NU: Serial no. of weight item - I2
ANL1, ANL2, ANL3, ANL4, ANL5 - alphanumeric description of item: 20 characters
WL: Weight of item F8.2
GLL: Distance of CG of weight item from AP F8.2
DLL: Length over which weight item is distributed F8.2
AL: Distance of aft end of weight from AP F8.2

SPACE - Input Parameters

Preparation of input data for this program should be accomplished in the following sequence:

1. Sketch the structure and number the joints and members as indicated in Fig. 49, remembering to observe the geometry of the structure in order to determine the joint sequence that will keep the half-band width of the stiffness matrix as narrow as possible.

2. Establish the reference coordinate system and label the joints with the proper coordinate values.

3. Define the different load cases to be considered.

4. With the aid of items 1 through 3 above, prepare data cards according to the formats indicated in the following descriptions.

5. The units should be dimensionally compatible, i.e. for eg:
   Length: feet
   Area: $feet^2$
   Inertia: $feet^4$
   Modulus of elasticity: $KIPS/feet^2$
   Load: KIPS
   Moment: KIPS - feet
   Distributed load: KIPS/feet

STRUCTURE DATA

Card Columns	*Identifier Name Used in Program*	*Data Description*	*FORTRAN Format*
1	KODE	The letter 'S' for structure card	A1
2-4	M	Number of members in structure.	I3
5-7	NJ	Number of joints in structure.	I3
8-10	NL	Number of loading conditions for this problem.	I3
11-13	MUD	The half-band width of the stiffness matrix (estimated).	I3
14-23	EN	Global modulus of elasticity for this problem	F10.0
24-75	—	Blank.	52X
76-80	JOBNO	Job identification number (any one - to five-digit number).	76-80

JOINT DATA

Card Columns	*Identifier Name Used in Program*	*Data Description*	*FORTRAN Format*
1	KODE	The letter 'J' for joint data card.	A1
2-4	J	The joint number of this joint.	I3
5	IXT	X-coordinate translational restraint of this joint. Leave blank if this joint is unrestrained in X-coordinate direction. Place '1' in this column if this joint is restrained in X-coordinate direction.	I1
6	IYT	Same as IXT above, except in Y-coordinate direction.	I1
7	IZT	Same as IXT above, except in Z-coordinate direction.	I1
8	IXR	Rotational restraint of this joint in X-coordinate direction. Leave blank if joint is.	I1

		unrestrained in X-coordinate direction. Place '1' in this column if this joint is restrained in X-coordinate direction. *Note:* For space truss problems, place '1' in this column.	
9	IYR	Same as IXR above, except in Y-coordinate direction.	I1
10	IZR	Same as IXR above, except in Z-coordinate direction	I1
11-20	XCOOR	X-coordinate of this joint.	F10.2
21-30	YCOOR	Y-coordinate of this joint.	F10.2
31-40	ZCOOR	Z-coordinate of this joint.	F10.2
41-80	———	Blank.	——

MEMBER DATA

Card Columns	*Identifier Name Used in Program*	*Data Description*	*FORTRAN Format*
1	KODE	The letter 'M' for member data card.	A1
2-4	I	Member number.	I3
5-7	J	Joint number of end j of member.	I3
8-10	K	Joint number of end k of member.	I3
11	MT	Member type. Leave column 11 blank if space frame member, place '1' in column 11 if space truss member.	I1
12-20	QIX	Moment of inertia about member X-axis.	F9.2
21-30	QIY	Moment of inertia about member Y-axis.	F10.0
30-40	QIZ	Moment of inertia about member Z-axis.	F10.0
41-50	QA	Cross-sectional area of member.	F10.0
51-60	G	Shear modulus of elasticity.	F10.0

61-70	SI	Angle of roll Ψ in degrees.	F10.0
71	ISI	If the angle of roll is specified for rotation YZX, leave this column blank. If specified for rotation ZYX, place '1' in this column.	I1
72-80	E	Modulus of elasticity of this member if different from global modulus assigned on structure data card. If same as global modulus, leave this field blank.	F9.0

MEMBER LOAD DATA

Card Columns	*Identifier Name Used in Program*	*Data Description*	*FORTRAN Format*
1	KODE	The letter 'L' for load data card.	A1
2-4	IB1	Member number of this number.	I3
5	IB2	Plane of loading. If the load lies in the member axis x_m-y_m plane, leave this column blank. (See Fig. 51, load P.) If the load lies in the member axis x_m-z_m plane, place '1' in this column. (See Fig. 51, load Q.)	I1
6-10	——	Blank.	5X
11-20	AB1	Value of load/length if uniform load is specified; load if a concentrated load is specified.	F10.3
21-30	AB2	Distance from joint j of member to beginning of load.	F10.3
31-40	AB3	Distance from joint j of member to termination of load.	F10.3

41-50	AB4	The angle the load makes with a normal line in degrees - i.e., α in Fig. 51.	F10.3
51-60	AB5	Blank (used in reading joint load data).	F10.3
61-70	AB6	Blank (used in reading joint load data).	F10.3
71-80	---	Blank.	-----

JOINT LOAD DATA

Card Columns	*Identifier Name Used in Program*	*Data Description*	*FORTRAN Format*
1	KODE	The letter 'P' for joint load card.	A1
2-4	IBI	The joint number.	I3
5	IB2	Blank (used in reading member load data).	I3
6-10	---	Blank.	5X
11-20	AB1	Applied force in X-coordinate direction at this joint	F10.3
21-30	AB2	Same as AB1 above, except in Y-coordinate direction.	F10.3
31-40	AB3	Same as AB1 above, except in Z-coordinate direction.	F10.3
41-50	AB4	Applied moment about N-axis at this joint.	F10.3
51-60	AB5	Applied moment about Y-axis at this joint.	F10.3
61-70	AB6	Applied moment about Z-axis at this joint.	F10.3
71-80	---	Blank.	-----

DUMMY LOAD DIVIDER CARD

Card Columns	*Identifier Name Used in Program*	*Data Description*	*FORTRAN Format*
1	KODE	The letter 'N' to indicate termination of this loading condition and the beginning of a new loading condition. The letter 'E' to terminate the last loading condition for this problem.	A1
2-80	----	Blank	--

PROGRAM TERMINATION CARD

Card Columns	*Identifier Name Used in Program*	*Data Description*	*FORTRAN Format*
1	KODE	The letter 'Q' to tell the program to quit execution. This is the last card in the data deck.	A1
2-80	----	Blank	----

NUMBERING SCHEME FOR SPACE FRAME

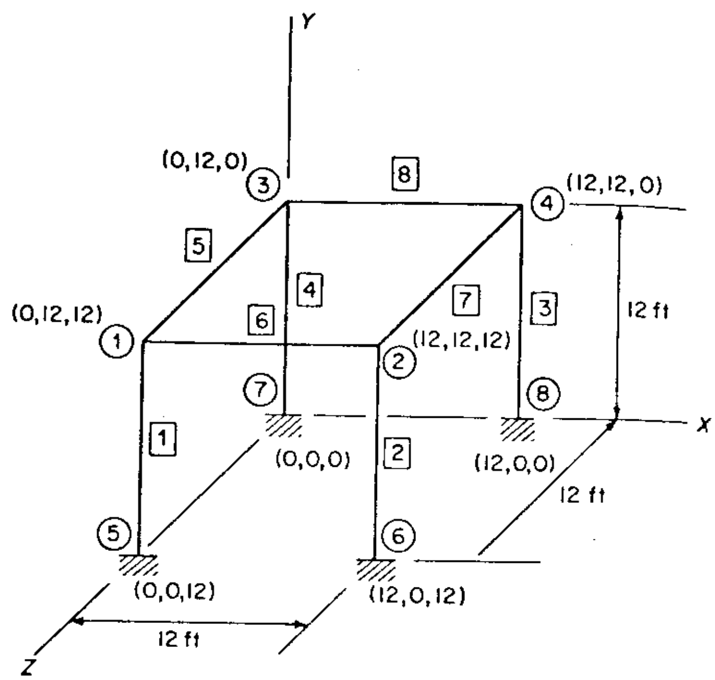

FIGURE 49

DISPLACEMENT DESIGNATION SEQUENCE FOR SPACE FRAME JOINT

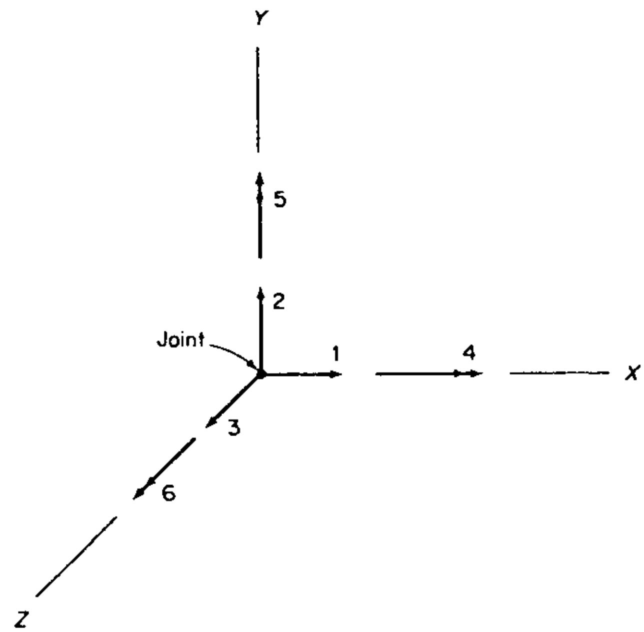

FIGURE 50

SIGN CONVENTION FOR MEMBER LOADS

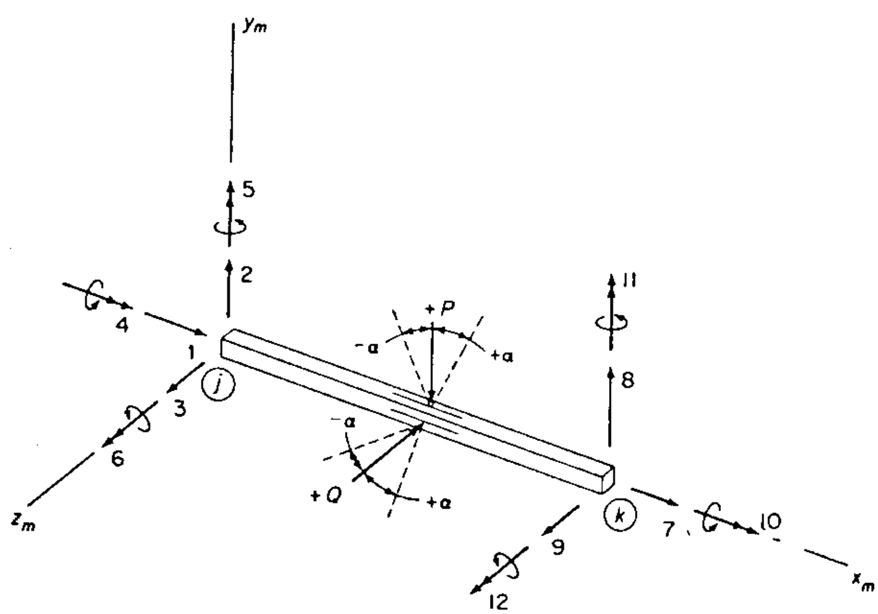

FIGURE 51

APPENDIX D: STRUCTURAL ANALYSIS OF ENGINE ROOM COMPUTER CODE LISTING

```
************************************************************************
C MAIN PROGRAM FOR STRUCTURAL ANALYSIS OF ENGINE ROOM
C CONSISTS OF THREE SECTIONS: 1) GIRDER ANALYSIS
C                             2) FRAME ANALYSIS
C                             3) THRUST BEARING FOUNDATION ANALYSIS
C                             4) GRILLAGE ANALYSIS
************************************************************************
      CHARACTER*4 PRM,DLEN
      CHARACTER*5 DWET
      COMMON/A1/PRM
      READ(5,1010)DLEN,DWET
      WRITE(6,1015)
      WRITE(6,1016)DLEN,DWET,DWET,DLEN,DWET,DLEN,DLEN,DWET,DLEN
 1000 READ(5,1001,END=1111)PRM
      IF(PRM.EQ.'GIRD')CALL GIRDER
      IF(PRM.EQ.'FRAM')CALL SPACE(PRM)
      IF(PRM.EQ.'GRIL')CALL SPACE(PRM)
      IF(PRM.EQ.'ROTA'.OR.PRM.EQ.'SHEA'.OR.PRM.EQ.'BEND') CALL TBSTIFF
      IF(PRM.NE.'GIRD'.AND.PRM.NE.'FRAM'.AND.PRM.NE.'ROTA'.AND.
     1PRM.NE.'SHEA'.AND.PRM.NE.'BEND'.AND.PRM.NE.'GRIL')THEN
      TYPE 1002
      GO TO 1111
      END IF
      GO TO 1000
************************************************************************
C FORMAT SECTION
************************************************************************
 1001 FORMAT(A)
 1010 FORMAT(2A)
 1002 FORMAT(5X,'***** ANALYSIS MODULE NAME NOT CORRECT',/,5X,
     1'***** SHOULD BE --- GIRDER OF FRAME OR FOUNDATION',/,50('*'))
 1015 FORMAT(16(/),132('I'),/,132('I'),/,'I',130X,'I',/,'I',130X,'I',
     1/,'I',15X,'ANALYSIS OF ENGINE ROOM STRUCTURE USING SIMPLIFIED ',
     2'METHODS',57X,'I',/,'I',130X,'I',/,'I',15X,'DEVELOPED AT M.I.T.',
     3'   DEPARTMENT OF OCEAN ENGINEERING  JUNE 1980',51X,'I',/,'I',
     4130X,'I',/,'I',130X,'I',/,132('I'),/,132('I'))
 1016 FORMAT(5(/),15X,'CHECK FOR DIMENSIONS IN THIS EXECUTION',/,
     115X,38('-'),/,15X,38('-'),//,15X,'LENGTH:',A,/,15X,'WEIGHT:',A,/,
     215X,'FORCE:',A,/,15X,'MOMENT:',A,1X,A,/,15X,'AREA:',A,'** 2',/,15X
     3,'INERTIA:',A,'**4',/,15X,'MODULUS:',A,'/',A,'**2')
************************************************************************
 1111 STOP
      END
************************************************************************
C END OF MAIN PROGRAM
************************************************************************

      SUBROUTINE GIRDER
************************************************************************
C ANALYTICAL METHODS PACKAGE FOR ANALYSIS OF ENGINE ROOM STRUCTURE
************************************************************************
C PART 1 HULL GIRDER ANALYSIS
************************************************************************
C INPUT AND ECHO SECTION
************************************************************************
      DIMENSION BPL(20),BPV(20),NU(20),ANL1(50),ANL2(50),ANL3(50),
     1WI(50),GI(50),DLL(50),AL(50),X(105),DMP(101),ANL4(50),
     2WD(101),BD(101),BCV(101),V(101),BM(101),SI(101),DLTA(101),
     3SDF(101),TDF(101),TCK(40),ANL5(50)
      WRITE(6,301)
      READ(5,*)ALBP,PMEA,PMEF,WTLS,CGLS
      READ(5,*)YME,GEE,FCF
      IF(FCF.EQ.0.0)FCF=1.0
      WRITE(6,302)ALBP,PMEA,PMEF,WTLS,CGLS
      READ(5,*)THM,THLC
      WRITE(6,303)THM,THLC
      WRITE(6,388)YME,GEE
      READ(5,*)NBP
      READ(5,*)(BPL(I),I=1,NBP)
      READ(5,*)(BPV(I),I=1,NBP)
      WRITE(6,304)
      WRITE(6,305)(BPL(I),BPV(I),I=1,NBP)
      READ(5,*)NS
      READ(5,206)(NU(J),ANL1(J),ANL2(J),ANL3(J),ANL4(J),ANL5(J),WI(J),
     1GI(J),DLL(J),AL(J),J=1,NS)
      WRITE(6,307)
      WRITE(6,308)(NU(J),ANL1(J),ANL2(J),ANL3(J),ANL4(J),ANL5(J),WI(J)
     1GI(J),DLL(J),AL(J),J=1,NS)
************************************************************************
C DETERMINE WEIGHT DISTRIBUTION
************************************************************************
      PBL=PMEF-PMEA
      FOL=ALBP-PMEF
      VI=5.0-PBL/ALBP
      PC=WTLS*VI*0.25/ALBP
      WMT=WTLS*CGLS
      REC1=WTLS-PC*PBL
      REC2=WMT-PC*PBL*(PBL*0.5+PMEA)
      RHS1=2.0*REC1-PC*(PMEA+FOL)
      RHS2=6.0*REC2-PC*2.0*(PMEA**2)-PC*(FOL**2)-PC*PMEF*FOL*3.0
      C1=(2.*(FOL**2))+(3.0*FOL*PMEF)+(PMEA*FOL)
      FCRD=(RHS2+(RHS1*PMEA))/C1
      ACRD=(RHS1-FCRD*FOL)/PMEA
************************************************************************
C AREA UNDER WEIGHT CURVE AND DISTRIBUTION MAPPING
************************************************************************
      CIL=ALBP/100.
      X(1)=0.0
      WD(1)=0.0
      DMP(1)=ACRD/PC
      DA=(PC-ACRD)/PMEA
      DF=(FCRD-PC)/FOL
      BLK1=PMEA*(ACRD+PC)*0.5
      BLK2=PBL*PC+BLK1
      DO 401 I=2,101
      X(I)=(I-1)*CIL
      IF(X(I)-PMEA)402,403,403
  402 WD(I)=X(I)*(ACRD+DA*0.5*X(I))
      DMP(I)=DMP(I-1)+(CIL*DA/PC)
      GO TO 406
  403 IF(X(I)-PMEF)404,404,405
  404 WD(I)=(X(I)-PMEA)*PC+BLK1
      DMP(I)=1
      GO TO 406
  405 VAR=X(I)-PMEF
      WD(I)=VAR*(PC+DF*0.5*VAR)+BLK2
      DMP(I)=DMP(I-1)+(CIL*DF/PC)
  406 DO 407 J=1,NS
      IF(X(I).LT.AL(J))GO TO 407
      EXT=X(I)-AL(J)
      IF(EXT-DLL(J))408,409,409
  408 RHS1=2.0*WI(J)/DLL(J)
```

```
      RHS2=WL(J)*(GL(J)-AL(J))*6.0/(DLL(J)**2)
      ORDR=RHS2-RHS1
      ORDL=2.0*RHS1-RHS2
      SLP=(ORDR-ORDL)/DLL(J)
      WLLT=(ORDL+EXT*SLP*0.5)*EXT
      WD(L)=WD(L)+WLLT
      GO TO 407
  405 WD(L)=WD(L)+WL(J)
  407 CONTINUE
  401 CONTINUE
*****************************************************************
CAREA UNDER BUOYANCY CURVE
*****************************************************************
      BD(1)=0.0
      BOV(1)=BPV(1)
      L=1
      KD=0
      NC=NBP-1
      I=1
  435 IF(I.NE.NC)GO TO 427
      KD=1
      GO TO 411
  427 Y1=BPV(I+1)-BPV(I)
      Y2=BPV(I+2)-BPV(I)
      X1=BPL(I+1)-BPL(I)
      X2=BPL(I+2)-BPL(I)
      TEMP=X1-X2
      IF(TEMP.EQ.0)GO TO 431
      IF(X1-0.0)430,432,430
  431 CCA=0.0
      CCC=BPV(I)
      COB=Y1/X1
      GO TO 411
  432 CCA=0.0
      COB=Y2/X2
      CCC=BPV(I)
      GO TO 411
  430 CCA=(Y1*X2-X1*Y2)/(X1*X2*(X1-X2))
      CCB=(Y1/X1)-CCA*(BPL(I+1)+BPL(I))
  433 COC=BPV(I)
  411 IF(L.EQ.101)GO TO 410
      IF(X(L+1).GT.BPL(I+1))GO TO 410
      L=L+1
      VAR=X(L)-BPL(I-KD)
      BCV(L)=QUAD(CCA,COB,CCC,VAR)
      BD(L)=(BCV(L)+BCV(L-1))*0.5*CIL+BD(L-1)
      GO TO 411
  410 I=I+1
      IF(I.NE.NBP)GO TO 435
*****************************************************************
C SHEAR FORCE
*****************************************************************
      DO 412 L=1,101
      V(L)=(BD(L)-WD(L))
  412 CONTINUE
*****************************************************************
C BENDING MOMENT
*****************************************************************
      BM(1)=0.0
      DO 413 L=2,101
      BM(L)=(V(L)+V(L-1))*0.5*CIL/DMP(L)+BM(L-1)
  413 CONTINUE
      ADM=THM
      IF(TMIC.GT.PMEA)GO TO 414
      TMI=DMF(1)+TMIC*DA/PC
      GO TO 416
  414 IF(TMIC.GT.PMEF)GO TO 415
      TMI=1.
      GO TO 416
  415 TMI=1.0+((TMIC-PMEF)*DF/PC)
  416 ATMI=ADM/TMI
*****************************************************************
C SLOPE CALCULATIONS
*****************************************************************
      SL(1)=0.0
      AMTA=0.0
      DO 417 L=2,101
      TDED=TMLC
      IF(X(L).GT.TMLC)AMTA=ATMI
      IF(X(L-1).GT.TMLC)TDED=X(L-1)
      SL(L)=(BM(L)+BM(L-1))*0.5*CIL+AMTA*(X(L)-TDED)+SL(L-1)
  417 CONTINUE
*****************************************************************
C DEFLECTION CALCULATIONS
*****************************************************************
      DLTA(1)=0.0
      DLTMAX=0.0
      DO 418 L=2,101
      DLTA(L)=(SL(L)+SL(L-1))*0.5*CIL+DLTA(L-1)
  418 CONTINUE
      SUB=DLTA(101)
      DO 419 L=2,101
      DLTA(L)=(DLTA(L)-(X(L)*SUB/ALBP))/YME
      IF(ABS(DLTA(L)).GT.DLTMAX)DLTMAX=ABS(DLTA(L))
  419 CONTINUE
*****************************************************************
C SHEAR DEFLECTION
*****************************************************************
      SDF(1)=0.0
      DO 420 L=2,101
      SDF(L)=(V(L)+V(L-1))*SDF*0.5*CIL/(GEE*DMP(L))+SDF(L-1)
  420 CONTINUE
      DO 421 L=2,101
      IF(ABS(SDF(L)).GT.DLTMAX)DLTMAX=ABS(SDF(L))
  421 CONTINUE
*****************************************************************
C TOTAL DEFLECTION
*****************************************************************
      DO 422 L=1,101
      TDF(L)=SDF(L)+DLTA(L)
      IF(ABS(TDF(L)).GT.DLTMAX)DLTMAX=TDF(L)
  422 CONTINUE
*****************************************************************
C OUTPUT SECTION
*****************************************************************
      WRITE(6,352)V(101),WD(101)
      WRITE(6,353)
      WRITE(6,306)
      WRITE(6,309)
      WRITE(6,310)
      DATA BLANK,AXIS,DOT,STAR,PLUS/' ','I','.','*','+'/
      DO 425 L=1,101
```

```
      DO 426 IP=1,40
      TOK(IP)=BLANK
  426 CONTINUE
      TOK(20)=AXIS
      LD=(1.0+(DLTA(L)/DLTMAX))*20.0+0.5
      LS=(1.0+(SDF(L)/DLTMAX))*20.0+0.5
      LT=(1.0+(TDF(L)/DLTMAX))*20.0+0.5
      TOK(LD)=DOT
      TOK(LS)=STAR
      TOK(LT)=PLUS
      WRITE(6,311) X(L),DLTA(L),SDF(L),TDF(L),(TOK(IP),IP=1,40)
  425 CONTINUE
*****************************************************************
C FORMAT SECTION
*****************************************************************
  206 FORMAT(I2,5A4,4F8.2)
  301 FORMAT(1H1//,5X,'ANALYTICAL METHODS PACKAGE FOR ENGINE ROOM',
     1' STRUCTURE',//,5X,'LONGITUDINAL HULL GIRDER ANALYSIS',/,
     2132('*'),/,132('*'),//)
  302 FORMAT(/,15X,'LENGTH BETWEEN PERPENDICULARS',3X,F8.2,/,15X,
     2'EXTENT OF PARALLEL MIDDLE BODY',/,41X,'FROM',2X,F8.2,2X,
     3' FWD OF AP',/,43X,'TO',2X,F8.2,2X,' FWD OF AP',/,
     415X,'DISTRIBUTED HULL WEIGHT',7X,F10.2,/,15X,
     5'DISTRIBUTED WEIGHT CG FROM AP',3X,F8.2)
  303 FORMAT(15X,'THRUST MOMENT',17X,F10.2,/,
     115X,'LOCATION OF THRUST MOMENT',7X,F8.2,2X,' FROM AP')
  304 FORMAT(//,15X,'BUOYANCY DISTRIBUTION',//,15X,'LOCATION FROM AI
     11CX,'BUOYANCY VALUE',/)
  305 FORMAT(18X,F8.2,17X,F8.2)
  306 FORMAT(//,132('-'),/,132('-'))
  307 FORMAT(//,15X,'SEMI-CONCENTRATED WEIGHT ITEMS',//,15X,
     1'NOTE',/,20X,'D.L. STANDS FOR DISTRIBUTED LENGTH',/,20X,
     2'LOCATION IS DISTANCE FROM AP TO AFT END OF WEIGHT',//,15X,
     3'NO',10X,'ITEM',10X,'WEIGHT',6X,'LCG',6X,'D.L.',4X,
     4'LOCATION',/)
  308 FORMAT(15X,I2,2X,5A4,4F10.2)
  386 FORMAT(/,15X,'MODULUS OF ELASTICITY= ',E14.8,/,15X,
     1'SHEAR MODULUS OF ELASTICITY= ',E14.8)
  309 FORMAT(1H1,//,5X,'CODE**  . BENDING DEFLECTION',4X,'* SHEAR',
     1' DEFLECTION',4X,'+ TOTAL DEFLECTION')
  310 FORMAT(//,4X,'DISTANCE',15X,'DEFLECTION',/,4X,'FROM AP',8X,
     1'BENDING',12X,'SHEAR',11X,'TOTAL',/,132('-'))
  311 FORMAT(3X,F7.2,3E18.8,10X,40A1)
  352 FORMAT(//,15X,'RESIDUAL FORCE=',F8.4,10X,'TOTAL INTEGRATED',
     1' DISPLACEMENT=',F12.4)
  353 FORMAT(15X,'RESULTS ARE FOR UNIT VALUES OF:',/,45X,
     1'INERTIA OF MIDSHIP SECTION ',/,45X,'AREA OF WEB OF',
     2' MIDSHIP SECTION ',/,15X,'DISTRIBUTION OF M.I. AND AREA',
     3' OF WEB TAKEN SAME AS DISTRIBUTED STEEL WEIGHT',/,15X,
     4'FOR ANY PARTICULAR VALUE OF M.I. AND AREA OF WEB OF',
     5' MIDSHIP SECTION',/,15X,'DIVIDE BENDING DEFLECTION BY M.I.',
     6' AND SHEAR DEFLECTION BY AREA OF WEB OF MIDSHIP SECTION',/,
     715X,70('-'))
*****************************************************************
      RETURN
      END

*****************************************************************
C FUNCTION QUAD
*****************************************************************
      FUNCTION QUAD(A,B,C,X)
      QUAD=C+X*(A*X+B)
      RETURN
      END

*****************************************************************
C END OF PROGRAM GIRDER
*****************************************************************
```

```
      SUBROUTINE SPACE(NAME)
C*****************************************************************
C*** THE DIMENSION OF ARRAY 'STIFF' MUST BE EQUAL TO THE VALUE
C***      GIVEN 'NDIM1'
C*** SET DEVICE ASSIGNMENTS AS FOLLOWS:
C***     1. IN= CARD READER
C***     2. IPRINT = PRINTER OR OUTPUT FILE
C***     3. ITAPE = TAPE UNIT OR EQUIVALENT
C*** THE DIMENSION OF ARRAYS TJ,RJ,GJ,CJ,EJ,PJ,UK,SK,HK,DK,FK,
C***   OK,MT2, AND CL1 MAY BE MODIFIED TO CHANGE THE MAXIMUM
C***   NUMBER OF MEMBERS THAT CAN BE PROCESSED.THE PROGRAM AS LISTED
C***   PERMITS NO MORE THAN 100 MEMBERS IN A STRUCTURE.
C*** THE DIMENSION OF ARRAYS IRX,IXD,IYD,IZD,X,Y,Z,IRY AND IRZ
C*** MAY BE MODIFIED TO CHANGE THE MAXIMUM NUMBER OF JOINTS
C*** THAT CAN BE PROCESSED.THE PROGRAM AS LISTED PERMITS NO MORE
C*** THAN 30 JOINTS IN A STRUCTURE.
C*** THE ARRAY 'STIFF' IS USED TO STORE THE STIFNESS MATRIX
C*** PLUS ONE LOAD VECTOR.THE STIFNESS MATRIX IS STORED
C*** IN HALF BAND FORMAT.AFTER OTHER DIMENSIONS ARE SET
C*** ARRAY 'STIFF' SHOULD BE GIVEN THE MAXIMUM DIMENSION THAT
C*** STORAGE WILL PERMIT
C*****************************************************************
C FRAME AND GRILLAGE ANALYSIS
C*****************************************************************
      DIMENSION AFM(12),DM(12),FEM(12),FEMT(12),T(12,12),
     1AK(12,12),AKT(12,12),TJ(100),RJ(100),GJ(100),CJ(100),
     2EJ(100),PJ(100),UK(100),SK(100),HK(100),DK(100),FK(100),OK(100
     3MT2(100),CL1(100),IRX(99),IXD(99),IYD(99),IZD(99),X(99),Y(99),
     4STIF(9000),Z(99),IRY(99),IRZ(99),CX(100),CY(100),CZ(100),
     5SI(100),ISI(100),E(100),OIX(100),OIY(100),OIZ(100),OP(100),
     6G(100),K1(100,12),KT(12)
      DATA IC1,IC2,IC3,IC4,IC5,IC6,IC7,IC8/'S','J','M','L','F','N','
     1'C'/
      NAME='FRAM'
      NDIM1=9000
      IN=5
      IT=6
      LTAPE=3
      WRITE(IT,921)
  921 FORMAT(1H1,//,5X,'ANALYTICAL METHODS PACKAGE FOR ENGINE ROOM',
     1'STRUCTURE')
      IF(NAME.EQ.'FRAM')WRITE(IT,922)
      IF(NAME.EQ.'GRIL')WRITE(IT,922)
  922 FORMAT(//,5X,'FRAME ANALYSIS',/,2X,130('*'))
  923 FORMAT(//,5X,'DOUBLE BOTTOM GRILLAGE ANALYSIS',/,2X,130('*'))
  977 CONTINUE
      MX=1
      N=0
    3 READ(IN,105)KODE,M,NJ,NL,MUD,EN,JOBNO
  105 FORMAT(A1,4I3,F10.0,52X,I5)
      IF(KODE.EQ.IC8)CALL EXIT
      IF(KODE.NE.IC1)GO TO 3
      WRITE(IT,901)JOBNO
  901 FORMAT(26X,'STRUCTURE DATA FOR JOB NO',I6/)
      WRITE(IT,902)M,NJ,NL,EN
  902 FORMAT(15X,I3,3X,'MEMBERS',I6,3X,' JOINTS',I6,3X,' LOADINGS',3
     1'GLOBAL E =',F11.0,//,132('-'))
      WRITE(IT,924)
  924 FORMAT(40X,'JOINT DATA',//,15X,'JOINT',3X,'IXT   IYT   IZT   I
     13X,'IYR   IZR',5X,'XCOOR',8X,'YCOOR',8X,'ZCOOR',/,132('-'))
      DO 201 I=1,NJ
```

```
      READ(IN,106)KODE,J,IXT,IYT,IZT,IXR,IYR,IZR,XCOOR,YCOOR,ZCOOR
  106 FORMAT(A1,I3,6I1,3F10.2)
      IF(KODE.NE.IC2)GO TO 13
      WRITE(IT,925)J,IXT,IYT,IZT,IXR,IYR,IZR,XCOOR,YCOOR,ZCOOR
  925 FORMAT(16X,I3,5X,I1,5X,I1,5X,I1,5X,I1,5X,I1,5X,I1,3F12.2)
      X(J)=XCOOR
      Y(J)=YCOOR
      Z(J)=ZCOOR
      IF(IXT)202,203,202
  203 N=N+1
      IXD(J)=N
      GO TO 204
  202 IXD(J)=NDIM1
  204 IF(IYT)205,206,205
  206 N=N+1
      IYD(J)=N
      GO TO 207
  205 IYD(J)=NDIM1
  207 IF(IZT)208,209,208
  209 N=N+1
      IZD(J)=N
      GO TO 210
  208 IZD(J)=NDIM1
  210 IF(IXR)211,212,211
  212 N=N+1
      IRX(J)=N
      GO TO 213
  211 IRX(J)=NDIM1
  213  IF(IYR)214,215,214
  215 N=N+1
      IRY(J)=N
      GO TO 216
  214 IRY(J)=NDIM1
  216 IF(IZR)217,218,217
  218 N=N+1
      IRZ(J)=N
      GO TO 201
  217 IRZ(J)=NDIM1
  201 CONTINUE
      IF(MUD.EQ.0)GO TO 307
      IF(MUD.GT.N-1)GO TO 14
      MM=MUD+1
      NM=N*MM-(MUD*MM)/2
      NTEM=NM+N
      IF(NTEM.GE.NDIM1)GO TO 15
      DO 150 I=1,NM
  150 STIF(I)=0.
  307 WRITE(IT,905)
  905 FORMAT(1H0,40X,'MEMBER DATA',//,1H ,13X,'MEM NO',3X,'J    K',8X,
     1'IX',8X,'IY',8X,'IZ',7X,'AREA',4X,'MODULUS',8X,'G',6X,'LENGTH',(
     2'TYPE',4X,'SI',3X,'ISI',/,132('-'))
      DO 10 IJK=1,M
      READ(IN,107)KODE,I,J,K,MT,QIX(IJK),QIY(IJK),QIZ(IJK),QA(IJK),
     1G(IJK),SI(IJK),ISI(IJK),E(IJK)
  107 FORMAT(A1,3I3,I1,F5.2,5F10.0,I1,F9.0)
      IF(KODE.NE.IC3)GO TO 13
      IF(E(IJK).EQ.0.0)E(IJK)=EN
      K1(IJK,1)=IXD(J)
      K1(IJK,2)=IYD(J)
      K1(IJK,3)=IZD(J)
      K1(IJK,4)=IRX(J)
      K1(IJK,5)=IRY(J)
      K1(IJK,6)=IRZ(J)
      K1(IJK,7)=IXD(K)
      K1(IJK,8)=IYD(K)
      K1(IJK,9)=IZD(K)
      K1(IJK,10)=IRX(K)
      K1(IJK,11)=IRY(K)
      K1(IJK,12)=IRZ(K)
      DO 90 LP=1,12
      DO 90 MP=LP,12
      IF((K1(IJK,LP).GE.NDIM1).OR.(K1(IJK,MP).GE.NDIM1))GO TO 90
      MDI=K1(IJK,LP)-K1(IJK,MP)
      IF(MDI)75,76,76
   75 MDI=-MDI
   76 IF(MDI-MX)90,90,85
   85 MX=MDI
   90 CONTINUE
      QI=(X(K)-X(J))*(X(K)-X(J))
      QI=QI+(Y(K)-Y(J))*(Y(K)-Y(J))
      QI=QI+(Z(K)-Z(J))*(Z(K)-Z(J))
      QI = SQRT(QI)
      QI2=QI
      WRITE(IT,908)I,J,K,QIX(IJK),QIY(IJK),QIZ(IJK),QA(IJK),E(IJK),
     1G(IJK),QL2,MT,SI(IJK),ISI(IJK)
  908 FORMAT(16X,I3,2X,I3,1X,I3,1X,7E10.2,6X,I3,2X,E7.2,I4)
      MT2(I)=MT
      QI1(I)=QI
      CX(IJK)=(X(K)-X(J))/QI
      CY(IJK)=(Y(K)-Y(J))/QI
      CZ(IJK)=(Z(K)-Z(J))/QI
      IF(MUD.EQ.0)GO TO 10
      CALL ROTA(T,CX(IJK),CY(IJK),CZ(IJK),SI(IJK),ISI(IJK))
      CALL MEMBER(T,AK,AKT,E(IJK),QIX(IJK),QIY(IJK),QIZ(IJK),QA(IJK),
     1G(IJK),QL,MT)
      DO 60 L=1,12
   60 KT(L)=K1(IJK,L)
      CALL STORE(KT,AKT,STIF,MUD,NDIM1)
   10 CONTINUE
      IF(MUD.GE.MX)GO TO 305
      MUD=MX
      MM=MUD+1
      NM=N*MM-(MM*MUD)/2
      DO 306 L=1,NM
  306 STIF(L)=0.0
      DO 310 IJK=1,M
      QI=QI1(IJK)
      MT=MT2(IJK)
      CALL ROTA(T,CX(IJK),CY(IJK),CZ(IJK),SI(IJK),ISI(IJK))
      CALL MEMBER(T,AK,AKT,E(IJK),QIX(IJK),QIY(IJK),QIZ(IJK),QA(IJK),
     1G(IJK),QL,MT)
      DO 61 L=1,12
   61 KT(L)=K1(IJK,L)
  310 CALL STORE(KT,AKT,STIF,MUD,NDIM1)
  305 NR=NM+1
      NEND=NM+N
      DO 274 IJJ=1,NL
      L38=0
      L39=0
      DO 234 I=1,M
      TJ(I)=0.0
      RJ(I)=0.0
```

```
      GJ(I)=0.0
      CJ(I)=0.0
      EJ(I)=0.0
      PJ(I)=0.0
      UK(I)=0.0
      SK(I)=0.0
      HK(I)=0.0
      DK(I)=0.0
      FK(I)=0.0
  234 QK(I)=0.0
      DO 235 I=NB,NEND
  235 STIF(I)=0.0
 8021 READ(IN,108)KODE,IB1,IB2,AB1,AB2,AB3,AB4,AB5,AB6
  108 FORMAT(A1,I3,I1,5X,6F10.3)
      IF(KODE.EQ.IC4)GO TO 8006
      IF(KODE.EQ.IC5)GO TO 8732
      IF(KODE.EQ.IC6.OR.KODE.EQ.IC7)GO TO 8008
      GO TO 13
 8006 IF(L39.GT.0)GO TO 393
      WRITE(IT,914)IJJ
  914 FORMAT(1H1,30X,'MEMBER LOAD DATA FOR LOADING NUMBER',I4,//,25X,
     1'MEMBER',4X,'LOAD',7X,'X BEGIN'5X,'X END',5X,'ALPHA',3X,'TYPE'/,
     2132('-'))
  393 KL=IB1
      L39=1
      MI=MT2(KL)
      W=AB1
      XX1=AB2
      XX2=AB3
      ALPHA=AB4
      WRITE(IT,912)KL,AB1,AB2,AB3,AB4,IB2
  912 FORMAT(26X,I3,2X,E10.2,3E11.2,I4)
      IF(ALPHA)8009,8010,8011
 8010 COSA=1.
      SINA=0.
      GO TO 8020
 8009 IF(ALPHA+90.)8012,8013,8012
 8013 COSA=0.
      SINA=-1.
      GO TO 8020
 8011 IF(ALPHA-90.)8012,8015,8012
 8015 COSA=0.
      SINA=1.
      GO TO 8020
 8012 ALPHA=ALPHA*3.14159/180.
      COSA=COS(ALPHA)
      SINA=SIN(ALPHA)
 8020 QL=QL1(KL)
      IF(MI.EQ.0)CALL FIX(W,XX1,XX2,QL,SINA,COSA,P11,P22,P33,P44,P55,P
     1)
      IF(MI.GT.0)CALL FIX3(W,XX1,XX2,QL,SINA,COSA,P11,P22,P33,P44,P55,
     1P66)
      TJ(KL)=TJ(KL)+P11
      UK(KL)=UK(KL)+P44
      IF(IB2.EQ.1)GO TO 7681
      RJ(KL)=RJ(KL)+P22
      PJ(KL)=PJ(KL)+P33
      SK(KL)=SK(KL)+P55
      QK(KL)=QK(KL)+P66
      GO TO 8021
 7681 GJ(KL)=GJ(KL)+P22
      EJ(KL)=EJ(KL)-P33
      HK(KL)=HK(KL)+P55
      FK(KL)=FK(KL)-P66
      GO TO 8021
 8732 IF(L38.GT.0)GO TO 37
      WRITE(IT,9645)IJJ
 9645 FORMAT(1H1,32X,'JOINT LOAD DATA FOR LOADING',I4,//,10X,'JOINT',5X
     1'X-FORCE',5X,'Y-FORCE',5X,'Z-FORCE',4X,'X-MOMENT',4X,'Y-MOMENT',
     24X,'Z-MOMENT',/,132('-'),/)
   37 J=IB1
      L38=1
      K11=IXD(J)+NM
      K22=IYD(J)+NM
      K33=IZD(J)+NM
      K44=IRX(J)+NM
      K55=IRY(J)+NM
      K66=IRZ(J)+NM
      IF(K11.GT.NDIM1)K11=NDIM1
      IF(K22.GT.NDIM1)K22=NDIM1
      IF(K33.GT.NDIM1)K33=NDIM1
      IF(K44.GT.NDIM1)K44=NDIM1
      IF(K55.GT.NDIM1)K55=NDIM1
      IF(K66.GT.NDIM1)K66=NDIM1
      STIF(K11)=STIF(K11)+AB1
      STIF(K22)=STIF(K22)+AB2
      STIF(K33)=STIF(K33)+AB3
      STIF(K44)=STIF(K44)+AB4
      STIF(K55)=STIF(K55)+AB5
      STIF(K66)=STIF(K66)+AB6
      WRITE(IT,931)J,AB1,AB2,AB3,AB4,AB5,AB6
  931 FORMAT(1H ,12X,I3,6F12.2)
      GO TO 8021
 8008 CONTINUE
      IF(L39.EQ.0)GO TO 36
      DO 8031 LLL=1,M
      CALL ROTA(T,CX(LLL),CY(LLL),CZ(LLL),SI(LLL),ISI(LLL))
      FEM(1)=TJ(LLL)
      FEM(2)=RJ(LLL)
      FEM(3)=GJ(LLL)
      FEM(4)=CJ(LLL)
      FEM(5)=EJ(LLL)
      FEM(6)=PJ(LLL)
      FEM(7)=UK(LLL)
      FEM(8)=SK(LLL)
      FEM(9)=HK(LLL)
      FEM(10)=DK(LLL)
      FEM(11)=FK(LLL)
      FEM(12)=QK(LLL)
      DO 9997 I=1,12
      FEMT(I)=0.0
      DO 9997 J=1,12
 9997 FEMT(I)=FEMT(I)+T(J,I)*FEM(J)
      DO 8291 I=1,12
      K11=K1(LLL,I)+NM
      IF(K11.GE.NDIM1)K11=NDIM1
 8291 STIF(K11)=STIF(K11)-FEMT(I)
 8031 CONTINUE
   36 CALL CHOLES(STIF,N,MM,0,IJJ,IT,&977)
      STIF(NDIM1)=0.0
      WRITE(IT,909)IJJ
  909 FORMAT(1H1,30X,'JOINT DEFLECTIONS AND ROTATIONS FOR LOADING',I4,
```

```
     1//,10X,'JOINT',5X,'X-DEFL',6X,'Y-DEFL',6X,'Z-DEFL',4X,'ROTA X-AX
     24X,'ROTA Y-AX',4X,'ROTA Z-AX'/,132('-'))
      DO 100 I=1,NJ
      K11=IXD(I)+NM
      K22=IYD(I)+NM
      K33=IZD(I)+NM
      K44=IRX(I)+NM
      K55=IRY(I)+NM
      K66=IRZ(I)+NM
      IF(K11.GT.NDIM1)K11=NDIM1
      IF(K22.GT.NDIM1)K22=NDIM1
      IF(K33.GT.NDIM1)K33=NDIM1
      IF(K44.GT.NDIM1)K44=NDIM1
      IF(K55.GT.NDIM1)K55=NDIM1
      IF(K66.GT.NDIM1)K66=NDIM1
  100 WRITE(IT,910)I,STIF(K11),STIF(K22),STIF(K33),STIF(K44),STIF(K55),
     1STIF(K66)
  910 FORMAT(1H ,12X,I3,6E12.5)
      WRITE(IT,911)IJJ
  911 FORMAT(1H1,45X,'FINAL MEMBER END ACTIONS LOADING',I4,//,2X,
     1'MEMBER  J X-FCE  J Y-FCE  J Z-FCE  J X-MOM  J Y-MOM  J Z-MOM  K'
     2' X-FCE  K Y-FCE  K Z-FCE  K X-MOM  K Y-MOM  K Z-MOM',/,132('-'))
      DO 120 IJK=1,M
      QI=QL1(IJK)
      CALL ROTA(T,CX(IJK),CY(IJK),CZ(IJK),SI(IJK),ISI(IJK))
      CALL MEMBER(T,AK,AKT,E(IJK),QIX(IJK),QIY(IJK),QIZ(IJK),QA(IJK),
     1G(IJK),QL,MT)
      DO 117 II=1,12
      DM(II)=0.0
      DO 117 J=1,12
      K11=K1(IJK,J)+NM
      IF(K11.GE.NDIM1)K11=NDIM1
  117 DM(II)=DM(II)+T(II,J)*STIF(K11)
      DO 130 J=1,12
      FEM(J)=0.
      DO 130 K=1,12
  130 FEM(J)=FEM(J)+AK(J,K)*DM(K)
      AFM(1)=FEM(1)+TJ(IJK)
      AFM(2)=FEM(2)+RJ(IJK)
      AFM(3)=FEM(3)+GJ(IJK)
      AFM(4)=(FEM(4)+CJ(IJK))
      AFM(5)=(FEM(5)+EJ(IJK))
      AFM(6)=(FEM(6)+PJ(IJK))
      AFM(7)=FEM(7)+UK(IJK)
      AFM(8)=FEM(8)+SK(IJK)
      AFM(9)=FEM(9)+HK(IJK)
      AFM(10)=(FEM(10)+DK(IJK))
      AFM(11)=(FEM(11)+FK(IJK))
      AFM(12)=(FEM(12)+QK(IJK))
      WRITE(IT,9327)IJK,(AFM(N3),N3=1,12)
  120 CONTINUE
 9327 FORMAT(3X,I3,2X,12E9.2)
  274 CONTINUE
      GO TO 977
   13 WRITE(IT,1333)
 1333 FORMAT(15X,'ERROR IN INPUT DATA,GONE TO NEXT PROBLEM'//)
      GO TO 977
   14 WRITE(IT,943)MUD,N
  943 FORMAT(' BANDWIDTH TOO LARGE FOR MATRIX   MUD=',I3,3X,'N=',I3)
      GO TO 977
   15 WRITE(IT,944)NTEM,NDIM1
  944 FORMAT('1 STORAGE AVAILABLE FOR STIFFNESS MATRIX AND ONE LOAD',
     1' VECTOR IS INSUFFICIENT',/,20X,'STORAGE REQUIRED=',I6,2X,
     2'STORAGE AVAILABLE=',I6)
      GO TO 977
      END
C***********************************************************************

C***********************************************************************
      SUBROUTINE FIX3(W,A,B,QL,SINA,COSA,P11,P22,P33,P44,P55,P66)
      P33=0.
      P66=0.
      C=B-A
      IF(ABS(C)-0.05)1,1,2
    1 WU=W*COSA
      P11=W*SINA*(QL-A)/QL
      P44=W*SINA-P11
      GO TO 3
    2 P44=W*SINA*(A-B)*C/(2.*QL)
      P11=W*SINA*C-P44
      WU=W*C*COSA
    3 D=QL-C.5*(A+B)
      P22=WU*D/QL
      P55=WU-P22
      RETURN
      END
C***********************************************************************

C***********************************************************************
      SUBROUTINE FIX(W,XX1,XX2,QL,SINA,COSA,P11,P22,P33,P44,P55,P66)
      C=XX2-XX1
      IF(ABS(C)-.05)1,1,2
    1 WU=W*COSA
      P11=W*SINA*(QL-XX1)/QL
      P44=W*SINA-P11
      GO TO 3
    2 WU=W*C*COSA
      P44=W*SINA*C*(XX1-XX2)/(2.*QL)
      P11=W*SINA*C-P44
    3 D=QL-XX1/2.-XX2/2.
      TS1=WU/(24.*QL)
      TS2=24.*D**3/QL-6.*XX2*C**2/QL+3.*C**3/QL
      TS3=4.*C*C-24.*D*D
      P33=-TS1*(TS2+TS3)
      TS3=2.*C*C-48.*D*D+24.*D*QL
      P66=-TS1*(TS2+TS3)
      TS1=TS1*6./QL
      TS2=12.*D*D-8.*D**3/QL+2.*XX2*C*C/QL-C**3/QL-C*C
      P22=TS1*TS2
      P55=WU-P22
      RETURN
      END
C***********************************************************************

C***********************************************************************
      SUBROUTINE ROTA(T,CX,CY,CZ,SI,ISI)
      DIMENSION T(12,12)
      REAL*8 S,D1,D2
      D1=22./7.
      D2=180.
      D1=D1/D2
      DO 9991 I=1,12
```

```
      DC 9991 J=1,12
 9991 T(I,J)=0.C
      T3=SQRT(CX**2+CZ**2)
      IF(SI.EQ.0.)GO TO 70
      IF(SI.EQ.90.)GO TO 71
   7C CS=1.
      SN=C.
      GC TC 75
   71 CS=C.
      SN=1.
      GC TC 75
   6C S=SI*D1
      SN=DSIN(S)
      CS=DCOS(S)
   75 CONTINUE
   63 IF(ISI.EQ.C)GO TO 64
      T3=SQRT(CX**2+CY**2)
      T(1,1)=CX
      T(1,2)=CY
      T(1,3)=CZ
      T(2,1)=(-CX*CZ*SN-CY*CS)/T3
      T(2,2)=(-CY*CZ*SN+CX*CS)/T3
      T(2,3)=T3*SN
      T(3,1)=(-CX*CZ*CS+CY*SN)/T3
      T(3,2)=(-CY*CZ*CS-CX*SN)/T3
      T(3,3)=T3*CS
      GC TC 47
   64 T(1,1)=CX
      T(1,2)=CY
      T(1,3)=CZ
      T(2,1)=(-CX*CY*CS-CZ*SN)/T3
      T(2,2)=T3*CS
      T(2,3)=(-CY*CZ*CS+CX*SN)/T3
      T(3,1)=(CX*CY*SN-CZ*CS)/T3
      T(3,2)=-T3*SN
      T(3,3)=(CY*CZ*SN+CX*CS)/T3
   47 DC 62 K=3,9,3
      DC 62 I=1,3
      DC 62 J=1,3
      IK=I+K
      JK=J+K
   62 T(IK,JK)=T(I,J)
      RETURN
      END
C*************************************************************

C*************************************************************
      SUBROUTINE MEMBER(T,AK,AKT,E,QIX,QIY,QIZ,QA,G,QL,MT)
      DIMENSION T(12,12),AK(12,12),AKT(12,12),TS(12,12)
      DC 65 J=1,12
      DC 65 I=1,J
   65 AK(I,J)=0.0
      C1=2.*E*QIY/QL
      C2=2.*E*QIZ/QL
      C3=3.*C1/QL
      C4=3.*C2/QL
      C5=2.*C3/QL
      C6=2.*C4/QL
       C7=G*QIX/QL
      C8=E*QA/QL
      AK(1,1)=C8
      AK(1,7)=-C8
      AK(7,7)=C8
      IF(MT.NE.0)GC TO 7499
      AK(2,2)=C6
      AK(3,3)=C5
      AK(4,4)=C7
      AK(5,5)=2.*C1
      AK(6,6)=2.*C2
      AK(8,8)=C6
      AK(9,9)=C5
      AK(10,10)=C7
      AK(11,11)=2.*C1
       AK(12,12)=2.*C2
      AK(2,6)=C4
      AK(2,8)=-C6
      AK(2,12)=C4
      AK(3,5)=-C3
      AK(3,9)=-C5
      AK(3,11)=-C3
      AK(4,10)=-C7
      AK(5,9)=C3
      AK(5,11)=C1
      AK(6,8)=-C4
      AK(6,12)=C2
      AK(8,12)=-C4
      AK(9,11)=C3
 7499 DC 66 J=2,12
      JI=J-1
      DC 66 I=1,JI
   66 AK(J,I)=AK(I,J)
      DC 9994 I=1,12
      DC 9994 L=1,12
      TS(I,L)=C.C
      DC 9994 J=1,12
 9994 TS(I,L)=TS(I,L)+AK(I,J)*T(J,L)
      DC 9995 I=1,12
      DC 9995 L=1,12
      AKT(I,L)=C.
      DO 9995 J=1,12
 9995 AKT(I,L)=AKT(I,L)+T(J,I)*TS(J,L)
      RETURN
      END
C*********************************************

C*********************************************
      SUBROUTINE STORE(K1,AKT,STIF,MUD,NDIM1)
      DIMENSION K1(12),AKT(12,12),STIF(9000)
      MM=MUD+1
      NS=(MUD*MM)/2
      DC 621 L=1,12
      I=K1(L)
      DC 621 K=1,12
      J=K1(K)
      IF(I.LT.J)GO TC 621
      IF(I.EQ.NDIM1.OR.J.EQ.NDIM1)GC TO 621
      LL=J+(I-MM)*MUD+NS
      IF(I.LE.MUD)LL=J+((I-1)*I)/2
      STIF(LL)=STIF(LL)+AKT(L,K)
  621 CONTINUE
      RETURN
      END
```

```
C***************************************************

C*************************************************************
      SUBROUTINE CHOLES(A,N,MM,IB,NT,IPRNT,*)
      DIMENSION A(9000)
      REAL*8 S1,T1,TMP
C
C*** A IS THE ARRAY CONTAINING THE ELEMENTS OF THE LOWER HALF BAND
C*** OF THE SYMMETRIC STRUCTURE STIFFNESS MATRIX.
C*** N=NUMBER OF EQUATIONS OR NUMBER OF UNKNOWN DISPLACEMENTS
C*** MM= HALF BANDWIDTH +DIAGONAL ELEMENT OR MUD + 1
C*** IB+1 = NO. OF RIGHT HANDSIDE VECTORS
C*** NT=- IF 1 OR LESS MATRIX IS FACTORED.IF GREATER THAN 1
C*** FORWARD AND BACKWARD SUBSTITUTION PERFORMED.
C*** S1 AND T1 ARE TEMPORARY DOUBLE PRECISION VARIABLES
C
      MUD=MM-1
      NS=MUD*MM/2
      NN=N*MM-NS
      IF(NT-1)30,30,31
C
C*** BEGIN CHOLESKY ALGORITHM FOR FACTORING MATRIX
   30 DO 20 J=1,N
      IF(J-MUD)1,1,2
    2 IN=J-MUD
      L=IN+(J-MM)*MUD+NS
      GO TO 7
    1 IN=1
      L=IN+(J-1)*J/2
    7 IF(J-N+MUD)103,103,105
  105 M5=N
      GO TO 104
  103 M5=J+MUD
  104 S1=0.0
      J1=J-1
      J2=J+1
      IF(J1)4,4,3
    3 DO 6 K=IN,J1
      T1=A(L)
      S1=S1+T1*T1
    6 L=L+1
    4 T1=A(L)
      IF(T1-S1.LT.0.)GO TO 100
      TMP=T1-S1
      T1=DSQRT(TMP)
      A(L)=T1
      IF(J-N)19,20,20
   19 DO 18 I=J2,M5
      SUM=0.0
      IF(I-MUD)68,68,71
   71 IN=I-MUD
      LI=IN+(I-MM)*MUD+NS
      GO TO 5
   68 IN=1
      LI=IN+(I-1)*I/2
    5 IF(J1)18,18,8
    8 IF(IN-J1)53,53,18
   53 DO 17 K=IN,J1
      LM=I+K-J
      SUM=SUM+A(LI)*A(LM)
   17 LI=LI+1
   18 A(LL)=(A(LL)-SUM)/A(L)
   20 CONTINUE
C
C***  BEGIN FORWARD SUBSTITUTION
   31 NP=IB+1
      NB=NN+1
      DO 65 K=1,NP
      A(NB)=A(NB)/A(1)
      DO 60 I=2,N
      IF(I-MUD)21,21,22
   22 IN=I-MM
      KS=IN*MUD+NS
      M5=MUD
      GO TO 27
   21 IN=0
      M5=I-1
      KS=M5*I/2
   27 SUM=0.0
      DO 61 J=1,M5
      JP=J+IN
      L=JP+KS
      JR=JR+NB-1
   61 SUM=SUM+A(L)*A(JR)
      II=I+KS
   60 A(JR+1)=(A(JR+1)-SUM)/A(II)
   65 NB=NB+N
C
C*** BEGIN BACKWARD SUBSTITUTION
      NB=NN+N
      DO 75 K=1,NR
      A(NB)=A(NB)/A(NN)
      DO 80 II=2,N
      I=N-II+1
      IF(I-MUD)41,41,95
   95 ID=I+(I-MM)*MUD+NS
      GO TO 42
   41 ID=I+(I-1)*I/2
   42 IF(I-N+MM)43,43,45
   45 M5=II-1
      GO TO 76
   43 M5=MUD
   76 SUM=0.0
      DO 81 J=1,M5
      JP=I+J
      IF(JR-MUD)98,98,99

   99 L=I+(JP-MM)*MUD+NS
      GO TO 82
   98 L=I+(JP-1)*JR/2

   82 JR=NB-N+JP
   81 SUM=SUM+A(L)*A(JR)
      JR=NB-N+I
   80 A(JR)=(A(JR)-SUM)/A(ID)
   75 NB=NB+N
      RETURN
  100 WRITE(IPRNT,901)
  901 FORMAT(' STIFFNESS MATRIX IS NOT POSITIVE DEFINITE HALT THIS
     1 PROBLEM')
      RETURN 1
      END
```

```
************************************************************************
C END OF SECTION FOR FRAME AND GRILLAGE ANALYSIS
************************************************************************

      SUBROUTINE TBSTIFF
      COMMON/A1/DUM
************************************************************************
C CALCULATION OF THRUST BEARING FOUNDATION STIFNESS AND STIFNESS OF
C INNER BOTTOM TO THRUST FORCE --AS PER SNAME BULLETIN R 15
C SECTION NAME TBSTIFF
************************************************************************
      COMMON/B1/DL(25),DU(25),NPP,VL(25),VU(25)
      COMMON/B2/FLAG
      DIMENSION VM(50),RP(10),RL(10),A(50),B(50)
************************************************************************
C INPUT SECTION
************************************************************************
      KO=2
      IF(DUM.EQ.'RCTA')IK=1
      IF(DUM.EQ.'SHEA')IK=2
      IF(DUM.EQ.'BEND')IK=3
      FCF=1.0
      IF(IK.NE.2)READ(5,*)ALBP,EM,NPP,NRP
      IF(IK.EQ.2)READ(5,*)ALBP,EM,NPP,NRP,FCF
      READ(5,*),(DL(I),DU(I),VL(I),VU(I),I=1,NPP)
      READ(5,*)TF,TL,SLNA
      IF(IK.EQ.1)READ(5,*),(RP(M),RL(M),M=1,NRP)
      IF(IK.NE.1)READ(5,*),(RP(M),M=1,NRP)
************************************************************************
C INPUT ECHO
************************************************************************
      IF(IK-2)410,411,412
  410 WRITE(6,201)
      WRITE(6,202)ALBP,EM
      WRITE(6,207)
      WRITE(6,203)(I,DL(I),DU(I),VL(I),VU(I),I=1,NPP)
      WRITE(6,204)TF,TL,SLNA
      WRITE(6,205)
      GO TO 413
  411 WRITE(6,211)
      WRITE(6,212)
      WRITE(6,208)ALBP,EM
      WRITE(6,213)
      WRITE(6,203)(I,DL(I),DU(I),VL(I),VU(I),I=1,NPP)
      WRITE(6,204)TF,TL,SLNA
      WRITE(6,214)
      GO TO 413
  412 WRITE(6,211)
      WRITE(6,215)
      WRITE(6,202)ALBP,EM
      WRITE(6,207)
      WRITE(6,203)(I,DL(I),DU(I),VL(I),VU(I),I=1,NPP)
      WRITE(6,204)TF,TL,SLNA
      WRITE(6,216)
************************************************************************
C END REACTIONS & M/EI CURVE OR V/GAW CURVE
************************************************************************
  413 TM=TF*SLNA
      VA=-TM/ALBP
      VB=VA
      IF(IK.EQ.2)VA=0
      TAD=0.0
      K=0
      IF(IK.EQ.2)TAD=VB
      DO 401 I=1,NPP
      K=K+1
      VM(K)=(VA*DL(I)+TAD)/VL(I)
      K=K+1
      VM(K)=(VA*(DL(I)+DU(I))*0.5+TAD)/((VL(I)+VU(I))*0.5)
      K=K+1
      VM(K)=(VA*DU(I)+TAD)/VU(I)
  401 CONTINUE
************************************************************************
C CALCULATION OF AREA AND RESULTS
************************************************************************
      DV=EM/FOF
      IF(IK.EQ.2)GO TO 420
      FLAG='A'
      CALL AREVAL(ALBP,VM,A,TIT,TM,TL,EM)
      CALL AREVAL(ALBP,A,B,BOD,0.0,ALBP,1.)
  420 DO 402 L=1,NRP
      IF(IK-2)415,416,417
  415 FLAG='A'
      CALL AREVAL(RP(L),A,B,TIT,0.0,ALBP,1.)
      VDF=TIT-(RP(L)*BOD/ALBP)
      FLAG='M'
      CALL AREVAL(RP(L),VM,A,TAB,TM,TL,EM)
      ARG=(VDF-TAB)/RP(L)
      THETA=ATAN(ARG)
      DEF=THETA*RL(L)
      WRITE(6,206)L,RP(L),RL(L),VDF,THETA,DEF
      GO TO 402
  416 FLAG='A'
      CALL AREVAL(RP(L),VM,A,TIT,0.0,ALBP,DV)
      WRITE(6,209)L,RP(L),TIT
      GO TO 402
  417 FLAG='A'
      CALL AREVAL(RP(L),A,B,TIT,0.0,ALBP,1.)
      VDF=TIT-(RP(I)*BOD/ALBP)
      WRITE(6,209)L,RP(L),VDF
  402 CONTINUE
************************************************************************
C FORMAT SECTION
************************************************************************
  201 FORMAT(1H1,/,5X,'ANALYTICAL METHODS PACKAGE FOR ENGINE ROOM',//,
     15X,'INNER BOTTOM STIFNESS TO THRUST FORCE',/,2X,130('*'),/,2X,
     2130('*'))
  202 FORMAT(//,15X,'LENGTH OF EQUIVALENT BEAM =',F8.4,/,15X,
     1'MODULUS OF ELASTICITY =',E14.8)
  203 FORMAT(10X,I2,6X,F10.2,5X,F10.2,7X,F10.2,11X,F10.2)
  204 FORMAT(//,15X,'THRUST FORCE =',F10.2,/,15X,'LOCATION OF THRUST ='
     1F10.2,'FROM AFT END',/,15X,'SHAFT CL TO SECTION NA =',F10.2,//)
  205 FORMAT(//,15X,'OUTPUT VALUES',//,10X,'NO',5X,'POSITION FROM',
     15X,'POSITION ABOVE',10X,'VERTICAL',10X,'SLOPE',10X,'AXIAL',/,20X,
     2'AFT END',8X,'SECTION NA',13X,'DEFLECTION',6X,'DEFLECTION',6X,
     3'DEFLECTION',/,132('-'))
  206 FORMAT(10X,I2,3X,F10.2,9X,F10.2,4X,E14.8,8X,E14.8,3X,E14.8)
  207 FORMAT(//,15X,'**** BEAM PROPERTIES',/,10X,
     1'NO',5X,'DISTANCE FROM',5X,'FORWARD',5X,'VALUE OF INERTIA',5X,
     2'VALUE OF INERTIA',/,22X,'AFT END',6X,'EXTENT',11X,'AFT END',12X,
     3'FORWARD END',/,130('-'))
  208 FORMAT(//,15X,'LENGTH OF EQUIVALENT BEAM =',F8.4,/,15X,
```

```
     1'SHEAR MODULUS OF RIGIDITY =',E14.8)
 209 FORMAT(10X,I2,5X,F10.2,5X,E14.8)
 211 FORMAT(1H1,/,5X,'ANALYTICAL METHODS PACKAGE FOR ENGINE ROOM',/,
     15X,'THRUST BEARING FOUNDATION STIFNESS ')
 212 FORMAT(/,5X,'STIFNESS IN SHEAR DEFLECTION',/,2X,130('*'),/,2X,
     1130('*'))
 213 FORMAT(//,15X,'BEAM PROPERTIES',/,10X,'NO',5X,'DISTANCE FROM',5X,
     1'FORWARD',7X,'SECTION AREA',9X,'SECTION AREA',/,22X,'AFT END',
     26X,'EXTENT',11X,'AFT END',12X,'FORWARD END',/,132('-'))
 214 FORMAT(//,15X,'OUTPUT VALUES',//,10X,'NO',5X,'POSITION FROM',10X,
     1'SHEAR',/,20X,'AFT END',10X,'DEFLECTION',/,132('-'))
 215 FORMAT(/,5X,'STIFNESS IN BENDING DEFLECTION',/,2X,130('*'),/,2X,
     1130('*'))
 216 FORMAT(//,15X,'OUTPUT VALUES',//,10X,'NO',5X,'POSITION FROM',10X,
     1'BENDING',/,20X,'AFT END',10X,'DEFLECTION',/,132('-'))
*****************************************************************
      RETURN
      END
*****************************************************************

      SUBROUTINE AREVAL(PT,Y,A,THETA,ADM,ADL,BM)
*****************************************************************
C SUBROUTINE TO FIND DISTRIBUTION OF VALUES & CALLS FUNCTION FOR AREA
C OR MOMENT CALCULATION
*****************************************************************
      COMMON/B1/XL(25),XU(25),NPP,VL(25),VU(25)
      DIMENSION Y(50),A(50)
      THETA=0.0
      A(1)=0.0
      DO 403 I=1,NPP
      ADD1=0.0
      ADD2=0.0
      N=I*3-1
      TKI=(XU(I)-XL(I))/4.0
      DCR1=(XU(I)+XL(I))*0.5
      DCR2=XU(I)
      T1=((VU(I)-VL(I))*(ADL-XL(I))/(XU(I)-XL(I)))+VL(I)
      T2=(VU(I)+VL(I))*0.5
      IF(DCR1.GT.ADL)ADD1=TAOM(ADL,DCR1,ADM,ADM,T1,T2,BM)
      IF(DCR2.GT.ADL)ADD2=TAOM(ADL,DCR2,ADM,ADM,T1,VU(I),BM)
      IF(DCR1.GT.ADL)ADD2=TAOM(DCR1,DCR2,ADM,ADM,T2,VU(I),BM)
      IF(XL(I).GT.ADL)ADD1=TAOM(XL(I),DCR1,ADM,ADM,VL(I),T2,BM)
      IF(N.NE.2)A(N-1)=A(N-2)
      IF(XU(I).GT.PT)GO TO 404
      A(N)=A(N-1)+TAOM(XL(I),DCR1,Y(N-1),Y(N),1.,1.,BM)+ADD1
      A(N+1)=A(N)+TAOM(DCR1,XU(I),Y(N),Y(N+1),1.,1.,BM)+ADD2
      THETA=A(N+1)
  403 CONTINUE
      GO TO 405
  404 DIS=PT-XL(I)
      ADD1=0.0
      DA=XU(I)-XL(I)
      T3=(VU(I)-VL(I))*(PT-XL(I))/DA+VL(I)
      COB=Y(N-1)
      COA=(Y(N+1)-Y(N-1))/DA
      YTU=DIS*COA+COB
      IF(PT.GT.ADL)ADD1=TAOM(ADL,PT,ADM,ADM,T1,T3,BM)
      IF(XL(I).GT.ADL)ADD1=TAOM(XL(I),PT,ADM,ADM,VL(I),T3,BM)
      AREA=TAOM(XL(I),PT,Y(N-1),YTU,1.,1.,BM)+ADD1
      THETA=THETA+AREA
  405 RETURN
      END
*****************************************************************

      FUNCTION TAOM(X,Y,V1,V2,T1,T2,EM)
*****************************************************************
C FUNCTION FOR AREA OR MOMENT CALCULATION DEPENDING ON FLAG)
*****************************************************************
      COMMON/B2/FLAG
      H=Y-X
      VAR=((V1/T1)+(V2/T2))/EM
      VAS=((V1/T1)+(2.0*V2/T2))/EM
      TAOM=H*0.5*VAR
      IF(FLAG.EQ.'M')TAOM=((H**2)*VAS/6.0)+TAOM*X
      RETURN
      END
*****************************************************************

*****************************************************************
C END OF SECTION FOR THRUST BEARING STIFNESS AND INNER BOTTOM STIFNESS
C TO THRUST FORCE
*****************************************************************
CEND OF PROGRAM
*****************************************************************
```

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SHIP STRUCTURE COMMITTEE PUBLICATIONS

These documents are distributed by the National Technical Information Service, Springfield, VA 22314. These documents have been announced in the Clearinghouse Journal U. S. Government Research & Development Reports (USGRDR) under the indicated AD numbers.

SSC-300, *Summary of Nondestructive Inspection Standards for Heavy Section Castings, Forgings, and Weldments* by R. A. Youshaw. 1980. AD-A099119.

SSC-301, *Probabilistic Structural Analysis of Ship Hull Longitudinal Strength* by J. C. Daidola and N. S. Basar. 1981. AD-A099118.

SSC-302, *Computer-Aided Preliminary Ship Structural Design* by A. E. Marsow and A. Thayamballi. 1981. AD-A099113.

SSC-303, *Fatigue and Fracture Toughness Characterization of SAW and SMA A537 Class I Ship Steel Weldments* by J. F. Souak, J. W. Caldwell, and A. K. Shoemaker. 1981.

SSC-304, *SL-7 Extreme Stress Data Collection and Reduction* by E. T. Booth. 1981.

SSC-305, *Investigation of Steels for Improved Weldability in Ship Construction - Phase II* by B. G. Reisdorf and W. F. Domis. 1981.

SSC-306, *Experimental Program for the Determination of Hull Structural Damping Coefficients* by P. Y. Chang and T. P. Carroll. 1981.

SSC-307, *Evaluation of Fracture Criteria for Ship Steels and Weldments* by A. W. Pense. 1981.

SSC-308, *Criteria for Hull-Machinery Rigidity Compatibility* by W. I. H. Budd, S. V. Karve, J. G. de Oliveira, and P. C. Xirouchakis. 1981.

SSC-309, *A Rational Basis for the Selection of Ice Strengthening Criteria for Ships - Vol. I* by J. L. Coburn, F. W. DeBord, J. B. Montgomery, A. M. Nawwar, and K. E. Dane. 1981.

SSC-310, *A Rational Basis for the Selection of Ice Strengthening Criteria for Ships - Vol. II - Appendices* by J. L. Coburn, F. W. DeBord, J. B. Montgomery, A. M. Nawwar, and K. E. Dane. 1981.

SSC-311, *Evaluation of SL-7 Scratch-Gauge Data* by J. C. Oliver. 1981.

SSC-312, *Investigation of Internal Corrosion and Corrosion-Control Alternatives in Commercial Tankships* by L. C. Herring, Jr. and A. N. Titcomb. 1981.

SSC-313, *SL-7 Research Program Summary, Conclusions and Recommendations* by K. A. Stambaugh and W. A. Wood. 1981.