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Yesterday's Technology—Today's Ships—Some Tanker Experience

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ABSTRACT

This paper describes, in a practical manner, some operating experience with tankers built in the mid 1960's when finite element analysis techniques were not broadly applied. A comparative review of the transverse strength of certain vessels is presented from the view of rules in effect and design approaches available at the time of design versus rules and analysis techniques subsequently developed and applied. As a result of the experience described it is recommended that proper attention be paid to all critical areas of ship's structure. A plea is made for the use of realistic design loads reasonably representative of practical conditions to be expected at sea. A few rather obvious but often ignored recommendations regarding details are presented. The paper further describes some experience with engine room double bottom deflections and how they were quantified.

INTRCDUCTION

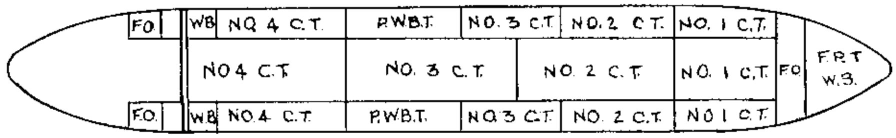
The drawing boards normally contain more interesting products than the assembly lines, especially in periods of rapidly expanding technology. For obvious reasons the latest conceptual design is not immediately available . We buy yesterday's product in today's marketplace.

This is particularly true of ships. A contract signed in mid 1974 promises delivery in mid 1977 of a ship built in accordance with rules in effect at the time of contract signing. Detail design and plan development, material procurement and vessel assembly: not only is each of these phases of ship construction time consuming, but one follows another. Meanwhile, researchers and designers find more rigorous application of design theory and technology. Sometimes prudence requires immediate adoption of such developments when the new technology discovers and overcomes some undesirable feature previously, though unwittingly, incorporated in a construction contract. Except where safety or design deficiency considerations arise, it is impractical to continually update the design in accordance with these latest technological developments since cost overruns and late deliveries would be guaranteed.

Generally speaking ships are built to meet requirements of a classification society. Classification society rules are usually based on past history and there is an understandable lag in updating these rules to reflect the latest technology. Ship designers as well as the technical staffs of the classification societies are usually a step ahead of the rules changes. The existing rules, therefore, should represent the minimum design criteria.

As requirements expand and applications develop which precede the experience factors of the rules, basic theory becomes more important in solving design problems. These designs become the basis of the experience needed to update the regulations and the standard design techniques.

But what of the innovative design? What is to be done to assure that the theories applied are indeed practical? What factor of safety should be used? More importantly to a shipowner, what should be done with existing designs that exhibit problems? One possible approach is to instrument ships to evaluate existing designs and to obtain or refine data for design use. Researchers and designers often meet with opposition from shipowners and operators on this approach. Like the author in Reference 1 they bemoan the scarcity



CASE I - Plan of Cargo Area

Principal Particulars	Metric	English
Length B.P.	221.0	725'-0"
Breadth MLD	33.2	109'-0"
Depth MLD	16.6	54'-6"
Draft MLD	12.2	40'-0"
Deadweight	60,000 T	

Figure 1 - Plan and Particulars of 60,000 DWT Tanker

of volunteers, but probably for different reasons.

Commercial ships are built with a very definite purpose in mind and most owners of such ships are not interested in transforming these into design laboratories due to fear of possible interference with operations and consequent costs. Those members of the R & D community interested in full scale data gathering must convince the shipowners of the benefits the owners will derive from volunteering the use of their ships for basic research, design development, or whatever other legitimate reasons might exist. Well laid out experimentation where the researchers recognize the vagaries of real life at sea and are willing to properly plan and coordinate the execution of data gathering would be accepted by most shipowners.

For the most part if problems develop in existing designs analytical approaches are needed to understand and overcome them. This paper deals with certain structural problems which occurred in specific tankers designed and constructed in the mid 1960's. It is the intent of this paper to describe, in strictly a practical manner, the nature of the problems and how they were handled. The theory and the details of its application are not presented but are referenced where appropriate. A comparative review of the problem structures is presented from the point of view of rules in effect and design approaches available at the time of design versus rules and analysis techniques available at the time of the vessel repair.

Two examples referred to as Case I and Case II deal with transverse ring strength. Case III describes a problem of double bottom flexibility in large tankers.

CASE I

Rapid development in tanker size from about 50,000 DWT tons to in excess of 300,000 DWT tons during the 1960's was fostered by a combination of various developing technologies including shipyard production methods, welding engineering, materials application, computer sciences and structural design techniques. In the design and construction of many tankers built during this time various stages of these technologies were applied sometimes with less than desirable results.

The lag in properly applying available technology often manifests itself in the form of operating problems. Numerous tankers were reported to have developed cracks particularly in the transverse ring structure. These cracks had been attributed to various causes including improper stiffening against buckling, inadequate shear area, insufficient weld area and poor detail arrangement, often compounded by poor workmanship.

Vessel

The first example to be sited involves a crude oil tanker delivered from the builder in

1966. The design and construction were in accordance with the 1964 rules of a major classification society.

The vessel was built as a 60,000 DWT ton ship, accomodations aft. The dimensions and general tank plan are indicated in Figure I.

Typical tankers of that era had few cargo tanks, each quite large. This vessel had twelve cargo tanks and two midship permanent water ballast tanks. Permanent water ballast tanks were specially coated as were the bottom four feet of all cargo tanks intended to carry sea water ballast.

Incident

In mid 1968 the vessel, enroute from the Persian Gulf to Europe with cargo tanks full and midship ballast tanks empty, encountered extreme sea conditions and winds of hurricane force. The storm was so intense in the area that at least one vessel was lost and numerous others were reported in trouble. Wave heights of 20-30 ft. above the deck were reported. Considerable nuisance damage occurred on deck and above: derrick booms were bent, accomodation ladders and davits were damaged, pipe supports were distorted and items on top of the midship deck house were severely damaged.

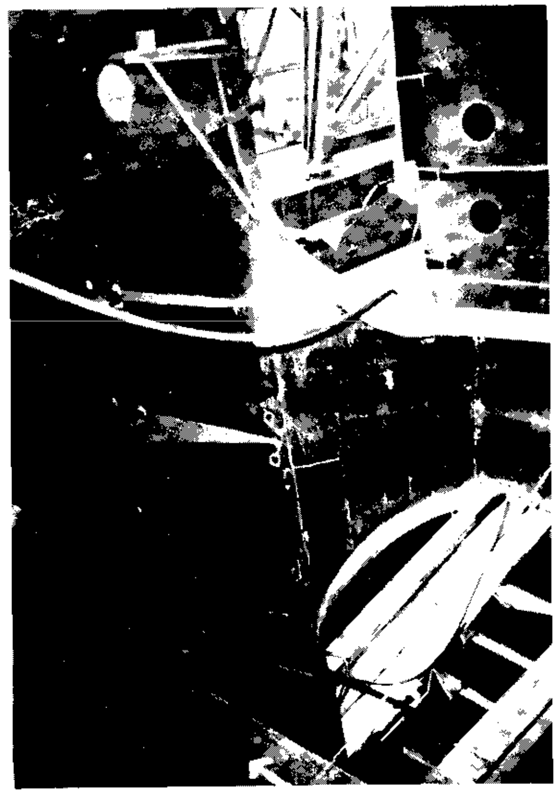


Figure 3 - Strut

Figure 2 - Face Plate on Deck Transverse

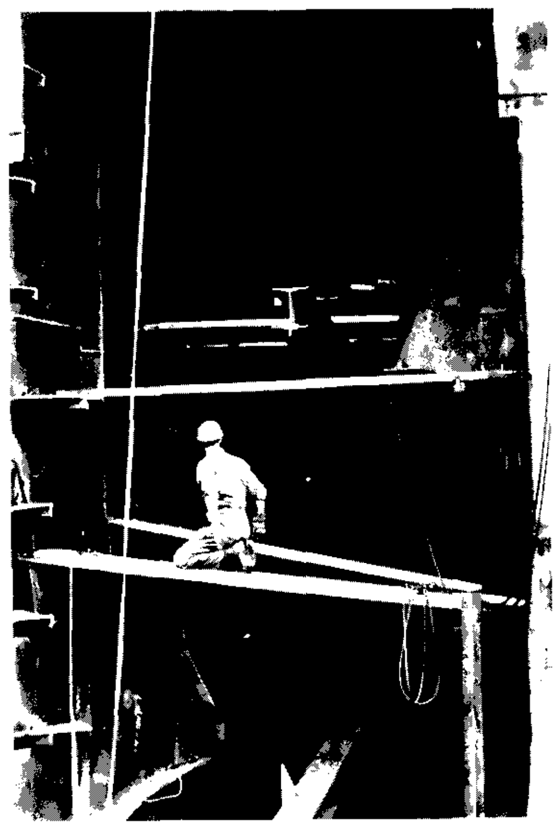


Figure 4 - Strut-Swash Bulkhead

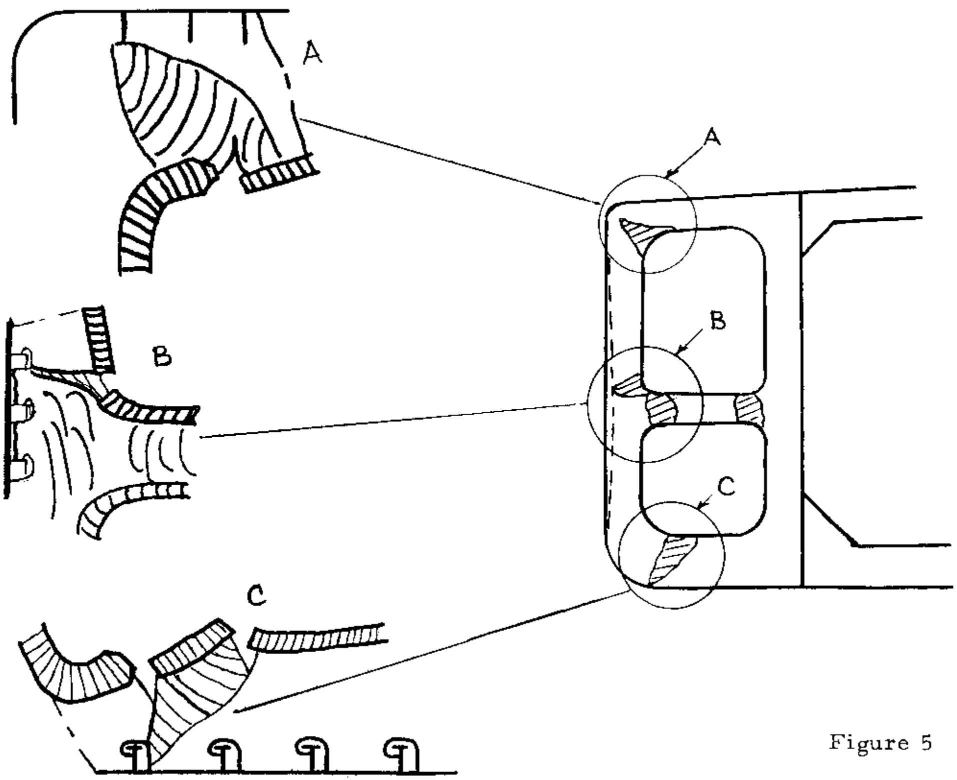


Figure 5

Typical Damage to Transverse Ring

Figure 6 - Face Plate on Bottom Transverse

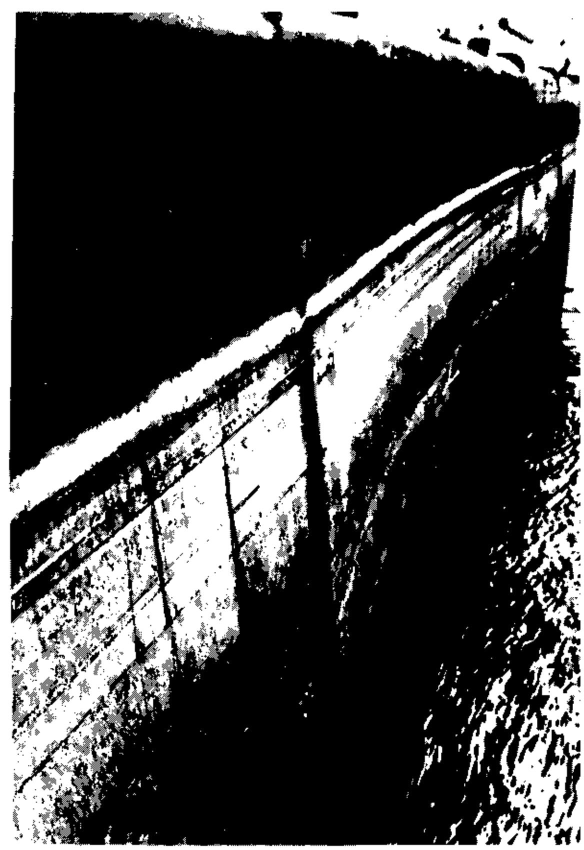


Figure 7 - Sideshell Set -In

Because of the height to which the damage was in evidence it is concluded that significant green water washed at least 4 meters or more above the main deck. The height of the waves relative to the deck was probably influenced by heavy rolling of the vessel during the storm.

Damage

The serious damage which instigated detailed analysis of the structure of the ship occurred in the port midship permanent water ballast tank. All transverse web frames in this tank were damaged. The web plates and the face plates at the gunwale corner, the strut, and the bottom transverse near the bilge corner were deformed and cracked. The extent of the damage to these transverse members can be seen in the sketches and photos of Figures 2 thru 6.

Figure 7 shows the external evidence of the damage, the side shell having been set in about one meter.

Design and Construction - 1964

This vessel was designed according to methods and rules in existence in 1964. The scantlings of the transverse ring according to 1964 rules and as built as are indicated in Figure 9. At the time of plan development the scantlings were checked by beam theory methods and found to be within acceptable limits.

The transverse strength of the hull was based, generally, on the assumptions that the vessel would be in the following conditions:

1. The vessel afloat with one center tank in structural test condition.

2. The vessel afloat with wing tanks, port & starboard, in structural test condition.

3. The vessel would be operated in accordance with the conditions set out in the vessel's loading manual, encountering waves presumed by the rules of the classification society.

The height of the waves supposed by the classification society rules at the time of design were based on the following:

$$Hw = 1.026 \times Lw^{0.4}$$

Where: Hw = Wave Height
Lw = Length of Ship

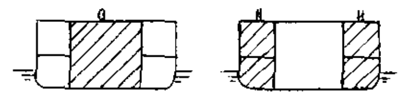
For this vessel whose Lw is 221 meters,

$$Hw = 8.9 \text{ meters}$$

Therefore the draft used for the study of transverse strength was the mean draft in each loading condition plus or minus 4.45 meters. With a full load draft of 12.2 meters, the effective draft becomes 16.65m or wave crest at deck at side.

The loading distributions used for analysis of the two test conditions and at sea conditions noted above are illustrated in Figure 8.

Structural Test Condition 4.15 Meter Effective Head



Operating Condition - 16.6 Meter Effective Head (at Deck at Side)

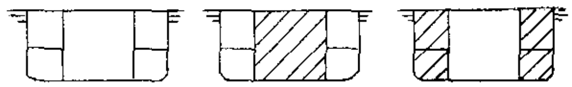


Figure 8 - Loading Distributions Assumed at Time of Design

Analysis and Repair - 1968

Are the failures briefly described above attributable to faulty design criteria? Were the assumptions regarding head and vessel loading proper? Was the basic design and analysis method adequate? In short, was the technology of the time up to the task of producing a vessel suitable for its time?

As a result of the aforementioned damage a detailed reanalysis of the transverse structure was accomplished. During the four brief years from the vessel's design until the damage, considerable progress had been made in the application of matrix methods to the analysis of complex structures. The reanalysis was accomplished by two-dimensional frame analysis utilizing "Stiffness method" of matrix analysis. Computations were performed using "Ices Strudl - I" and "Stress" computer programs. The background of the analytical technique can be found in References 2 through 11. Three dimensional analysis was also conducted taking into account the relative

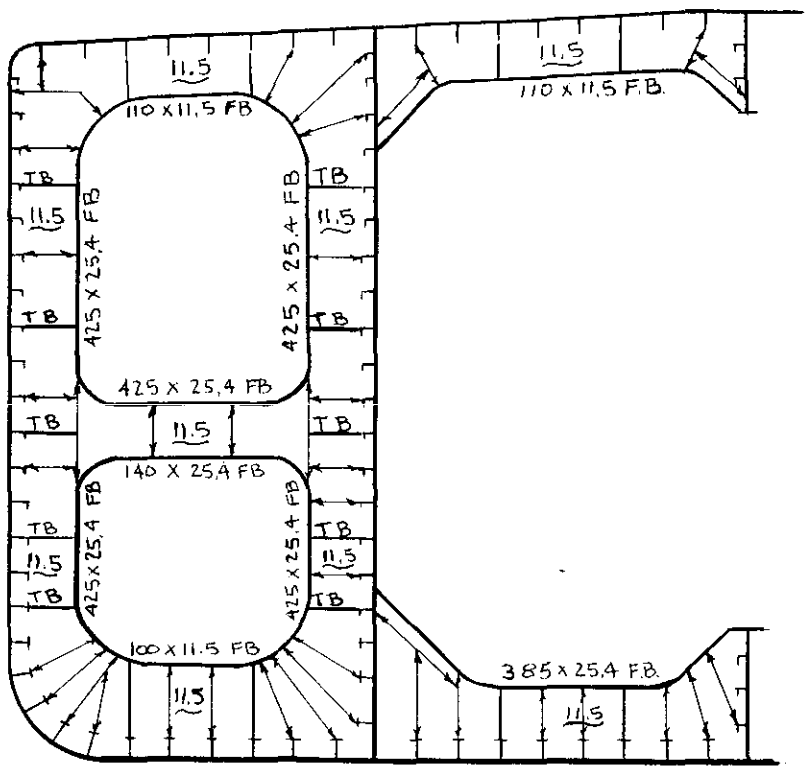


Figure 9 - Case I - Midship Section As Built

displacement of the longitudinal strength members in determining the stresses in each structural member of the transverse ring. This was done according to Reference 12.

Loading conditions used in analysis included also the actual sea conditions reported by the vessel during the storm that caused the damage in order to obtain a reasonable failure analysis as well as transverse strength reanalysis.

It has already been noted that during the extreme sea conditions damage was caused to the vessel which indicated green water in excess of 4 meters above the deck. The Master's report had indicated that seas as high as almost 9 meters had occurred. An average head of 4.88 meters (16 feet) of water, acting on the full beam of the vessel was used in the reanalysis. The maximum pressure head in this condition is 21.48 meters. The wave height in this case can be expressed as:

$$Hw = (\text{Depth} + 4.88 - \text{Draft}) \times 2$$
$$= (16.6 + 4.88 - 12.2) \times 2 = 18.56\text{m}$$

When compared with the previously mentioned wave height based on class rules of 8.9 meters, this is indeed an extremely abnormal sea condition.

Not yet considered are the dynamic effects which undoubtedly exist. Generally speaking, whenever a vessel encounters abnormally extreme weather conditions such as a hurricane necessary measures would be taken without delay to properly orient the vessel to avoid washing of waves over the broadside of the vessel. However, it is conceivable that a vessel, when struck by a large wave from one side, may heel greatly and be struck again by another large wave before proper corrective action can be taken. In a case where this vessel encounters a classification society assumed wave height of 8.9 meters when it is heeling 20° to one side in full load condition, its draft is:

$$\text{Draft} = 12.2 + \frac{B}{2} \tan 20^\circ + 4.45 = 22.7\text{m}$$

When the vessel is struck by a wave 6.4 meters in height when it is heeling 20° the effective head is 21.48 meters, the same as that used in the reanalysis.

The above brief example was presented to show the reasonableness of using an effective head of about 5 meters over the deck in structural analyses of this type.

The most significant loading conditions used in the reanalysis were those of Figure 8 and those of Figure 10.

The transverse strength of the ship's hull in the heeled state was examined by the use of a sophisticated form of three dimensional analysis of Reference 13.

Draft - Rough Sea Condition, Full Load, Upright & Heeling Conditions

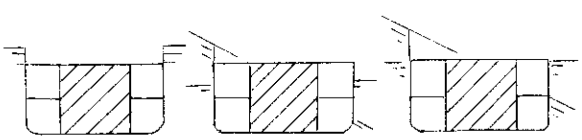


Figure 10 - Loading Conditions Assumed At Time of Reanalysis

The following results of the reanalysis indicated the following:

1. The assumed classification wave height was probably unrealistic considering the extreme seas to which vessels are subjected.
2. The computed stresses which resulted from the assumed extreme loading condition indicated that the lack of continuity of structure in the face plates of the bottom and deck transverse members and in the strut was the most important cause of failure when the vessel was exposed to repeated severe impact loads caused by the force of large seas breaking against the ship.
3. The unbalanced design of the strut, the upper face plate width being three times the width of the lower face plate, was presumed to have added to the fixed end bending moments which are, theoretically, considered to be larger than for a uniform cross section. This is discussed in Reference 10.

At the time of reanalysis the 1968 classification society rules and recent notices were applied to this design to compare the existing vessel to the then current design criteria. This is shown in Figure 11. When compared with the original scantlings of Figure 9 it may be seen that the principal changes occur in the size of web plate thickness, relocation of tripping brackets at the strut, and a balanced design of the strut. The lack of continuity in the face plate arrangement remains, however.

It is understood that the classification societies had considered the problem of practical design head criteria and concluded that a design condition using a draft equal to the depth at side would be adequate. As a result external pressures greater than the depth at side were not used when meeting class criteria.

The reanalysis of the transverse structure was completed, of course, with the idea in mind of repairing and preventing a reoccurrence of the damage. Utilizing as much of the existing design as possible, reinforcement of the transverse ring was accomplished according to Figure 12.

The principal points to note in the reinforcement are the removal of the face plate thickness discontinuities, the balancing of the strut design, the use of flat bar panel breakers to reduce buckling possibilities, and the relocation of the tripping brackets at the strut ends.

The vessel has been operating free of structural problems since the reinforcements were completed.

CASE II - TRANSVERSE STRENGTH

This second example refers to the same general subject of transverse strength of a tanker built in the mid 1960's. Unlike the example in Case I, this vessel did not fail prior to structural reanalysis and reinforcement. As a result of the Case I failure and numerous reports of structural failures in tankers of this vintage it was deemed prudent to conduct an investigation into the vessel's structural adequacy using the latest available techniques and applying extreme, but realistic assumptions regarding design head.

Vessel - 1965

This is an example of a class of 95,000 DWT Tankers constructed in Japan in 1966 according to classification society rules of 1965.

The plan view and principal dimensions are given in Figure 13.

The design reflects the typical trends at the time of few large cargo tanks, midship permanent ballast tanks, and reduced scantling notation in the record. The design differed somewhat from the normal tanker arrangement in that the center and wing tanks were all of the same width.

The scantlings of the transverse ring, as built, are shown in Figure 15.

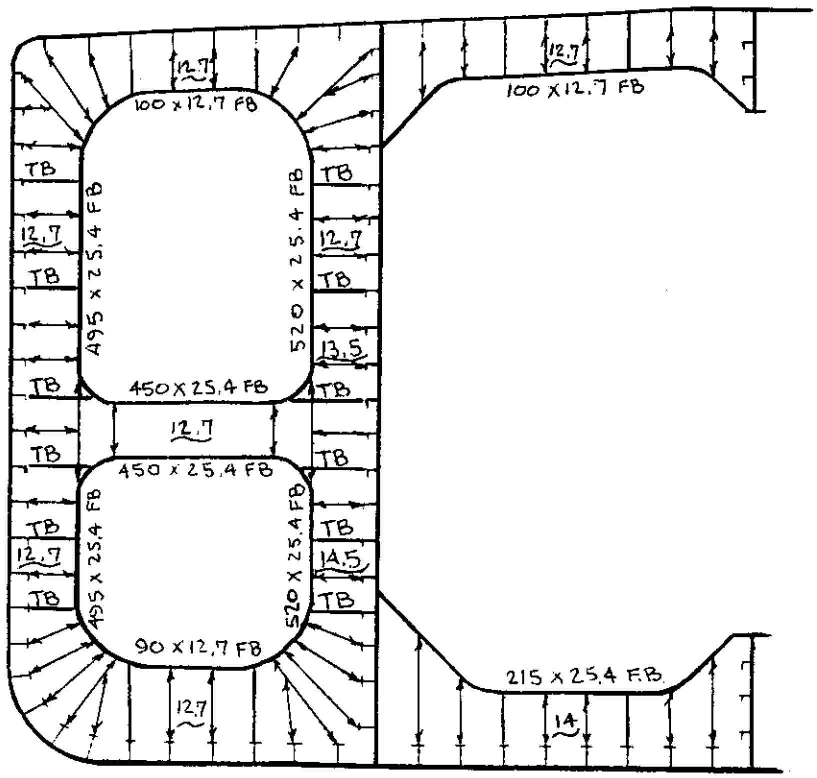


Figure 11 - Case I - Midship Section Utilizing 1968 Rules and Notices

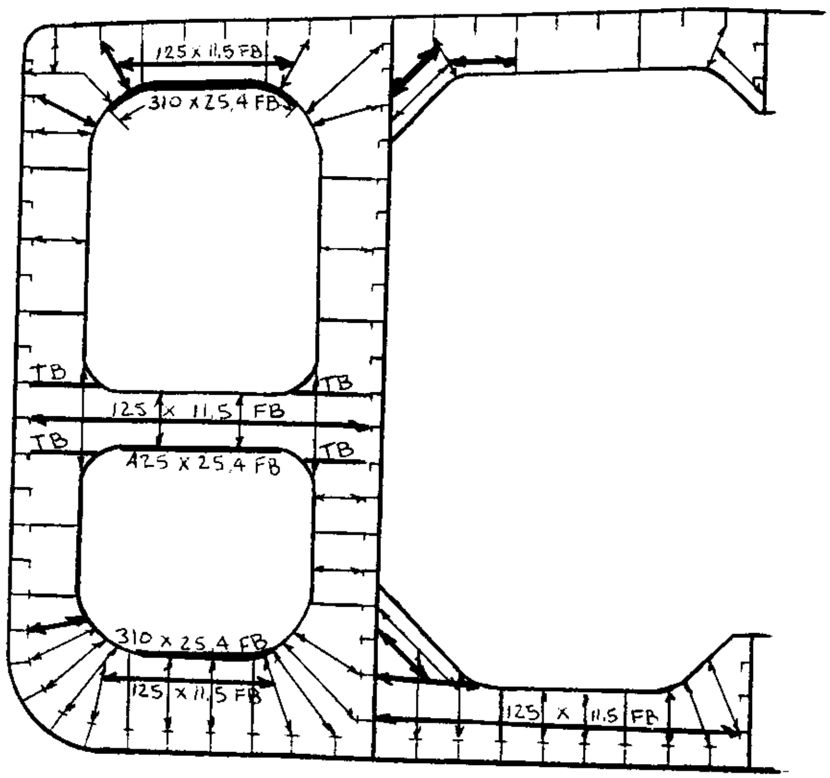
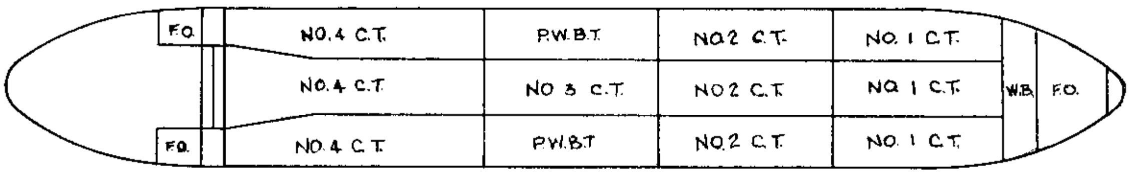


Figure 12 - Case I - Midship Section As Repaired



Case II Plan of Cargo Area

Principal Particulars	Metric	English
Length (B.P.)	264.6	868'-0"
LWL	269.2	883'-0"
Beam (MLD)	38.95	127'-9"
Depth (MLD)	18.90	62'-0"
Draft (extreme)	13.3	43'-7 1/2"
Deadweight	95,000	

Figure 13 - Plan and Particulars of a 95,000 DWT Tanker

The criteria for checking the transverse strength of this design at the time it was developed was essentially the same as that described in Case I. The structural test conditions as well as the most critical cases of the vessel's expected loading pattern were analyzed in accordance with classification society recommendations.

The class wave height formula $H_w = 1.026 \times L_w^{0.4}$ resulted in a wave height of 9.3 meters. The effective head during the worst assumed operating condition fully laden applying the class wave is:

$$\text{Head} = \text{Draft} + \frac{H_w}{2}$$

$$= 13.3 + \frac{9.3}{2} = 18 \text{ meters}$$

This assumed worst effective head is almost one meter below the deck at side.

Structural Analysis - 1968

The detailed frame analyses by the methods previously outlined in Case I were performed. These included the two dimensional "displacement method" and the three dimensional analysis considering the relative deflection of the wing tanks. The analysis of the vessel in the heeled condition was not done.

Rough Sea Condition - 4.88 Meter Head Over Main Deck

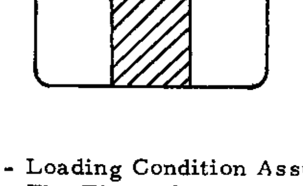


Figure 14 - Loading Condition Assumed At The Time of Reanalysis

The various loading conditions studied were identical to Case I assuming water to be 4.88m or 16 feet over the main deck, center tanks full and wing tanks empty. Figure 14 shows the worst condition considered.

The results of the reanalyses indicated that high combined stresses existed in the deck transverses and high shear stresses existed in the bottom web near the bilge. The strut indicated a propensity to buckle.

Corrective reinforcement was undertaken as indicated in Figure 16.

The points to note are the correction of the discontinuous face plate of the deck transverse, panel stiffeners in the strut and the installation of more shear area in the form of a diagonal bilge bracket.

CASE III - DOUBLE BOTTOM FLEXIBILITY

This third example of the lag in the application of latest technological developments to the practical realm deals with the problem of engine room double bottom flexibility in large tankers. As tankers rapidly increased in deadweight tonnage during the 1960's, ship's length, beam, depth, draft and even block coefficient were increased. The increase in ships beam and draft, slight increase in block coefficient and the refinement of welding technology resulting in more efficient and lighter structures, all contributed toward greater flexibility in engine room

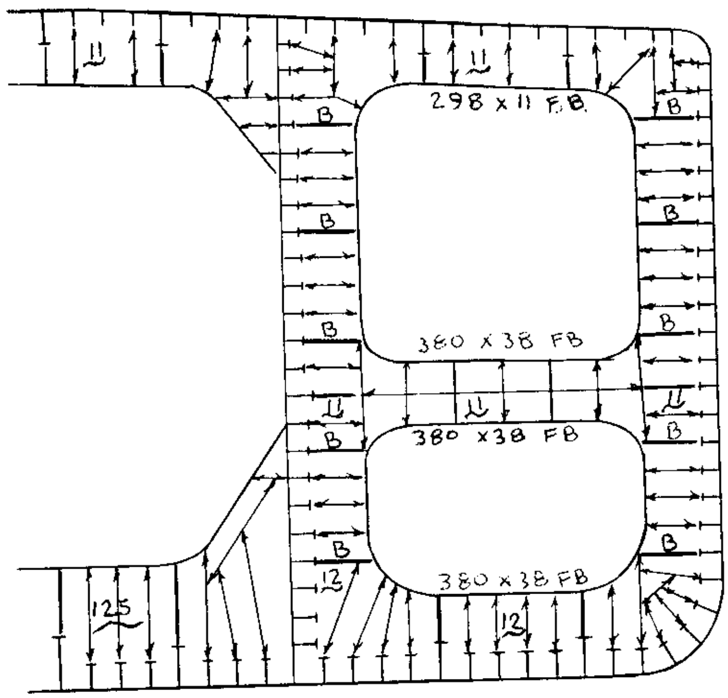


Figure 15 - Case II - Midship Section As Built

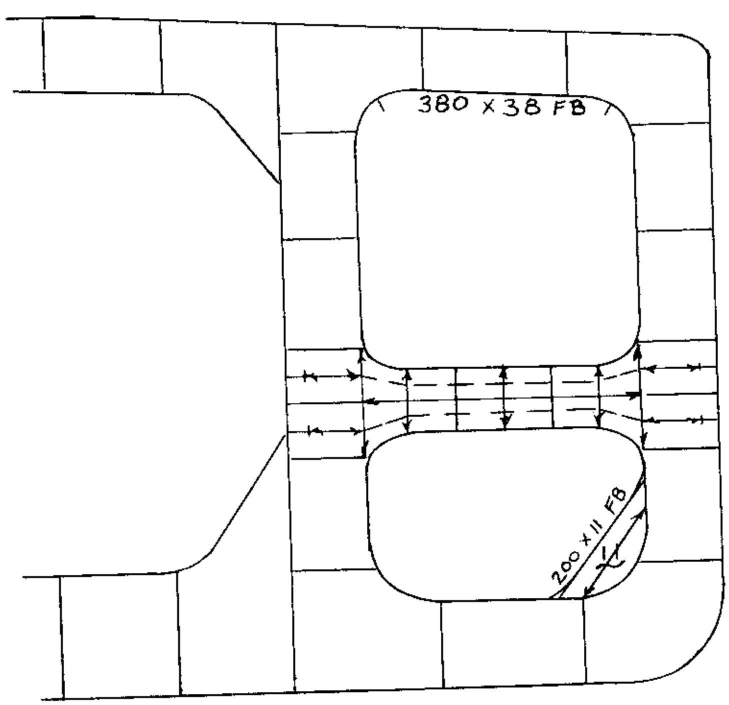


Figure 16 - Case II - Midship Section Reinforcements

steel work.

Contrary to the trend of greater flexibility of ship's structure was the trend in greater stiffness of the main propulsion machinery associated with higher powers applied to single shafts and lower main engine RPM. The relative incompatibility became more and more significant as the size and power of vessels increased. Serious physical problems were the result: improper shafting alignment, damaged main reduction gears and main engine crankshaft and bearing damage. These problems are carefully analyzed and reported upon in References 14 and 15.

This example refers to the class of 95,000 DWT tankers described in Case II above. The plan view and dimensions are given in Figure 13.

Shortly after delivery signs of pitting were noted on the forward helixes of the main bull gear and low speed pinions. The wear was attributable to faulty shaft alignment. Only after considerable investigation could it be confirmed that the double bottom deformation was markedly greater than had been presumed during the design and construction stages of these vessels.

Design & Construction - 1965

During the design stages of this class of vessels the anticipated hull deflection was calculated using the conventional longhand longitudinal strength computation methods for different conditions of loading. The results are illustrated in Figure 17 and show a fairly typical tanker deflection curve. It can be seen that the characteristic hogging conditions exist in light ship and ballast conditions. The full load condition shows almost no deflection aft, but some sagging forward. The effect of empty midship ballast tanks is evident in this figure.

An enlargement of the engine room portion of the hull girder deflection curve is presented in Figure 18. From the figure it can be seen that the engine room deforms essentially linearly. There is indicated up to 2mm upward deviation from a linear projection from the stern frame to the engine room forward bulkhead in the fully loaded and ballast conditions. However, the relative deflection of the engine room double bottom from the linear between those two operating conditions doesn't change. For example, forward of Frame 29, the location of the main reduction gears, the deviation from the linear is 2mm in both loaded and ballast conditions. The conclusion was to neglect the change in deformation of the hull as a girder for practical operating conditions. Proper shaft alignment for the ballast condition was considered to be valid for all the operating conditions between ballast and full load.

Local deformations due to variations in draft were neglected. It has properly been pointed out in References 14 and 15 that as ship sizes increased from 50,000 DWT to 100,000 DWT and upwards little consideration was given to the draft-engine room beam relationship. The length of the double bottom floors is a direct function of the beam of the ship. When treated as a beam with an evenly distributed load, hydrostatic pressure due to draft, the deflection is proportional to the fourth power of the floor length. All other things being constant, doubling the beam of a vessel results in deflections being increased 16 times.

With the classification society rules requiring double bottom depth to increase as a function of vessel beam and draft, actual deflections probably vary somewhat less than the cube of the beam. With the doubling of vessel's breadth this still represents an "unhealthy" increase in deflection by a factor of almost 8.

Analysis - 1970

Investigation into the gear problems and rationalization of shafting alignment was directed toward determination of steel work deformation. This was carried out on a two pronged front: 1) analytical and 2) full scale instrumentation.

The importance of determining the local deflection of the double bottom under the bearing foundations was recognized. A finite element analysis technique was applied utilizing a newly developed computer programmed Matrix Method of Structural Analysis of Framed Structures. The result of this analysis of the steel work as a complex box structure indicated an expected deflection pattern at the ship's centerline to be as indicated in Figure 19 for a uniform load equivalent to 6.1 meters (20 feet) draft variation. The calculated deflection increased from 0 at the afterpeak bulkhead to 0.47mm at the after end of the gear case, 0.85mm at the bull gear forward bearing to a maximum of 3.88mm at Frame 43, well forward of the gearing and line shafting. Expressed in terms of deflection at the bull gear forward bearing per foot of vessel draft aft the expected deformation amounted to 0.42mm/10 feet.

In order to corroborate results of the finite element analysis so that adequate compensation would be made for local deformation in way of the line shaft support points, actual measurement of the deflection of the structure was undertaken. This was accomplished by two independent methods:

1) A piano wire was strung from the forward bulkhead of the engine room to the

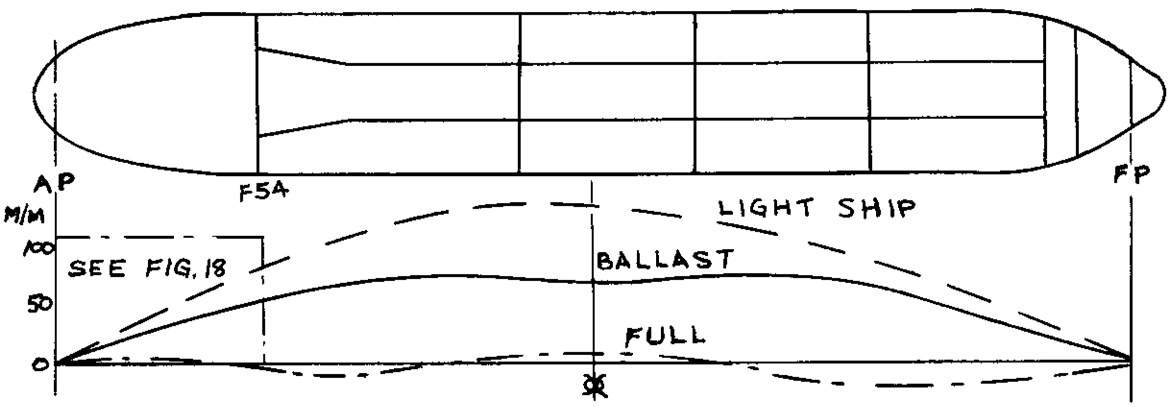


Figure 17 - Deflection of Hull Girder

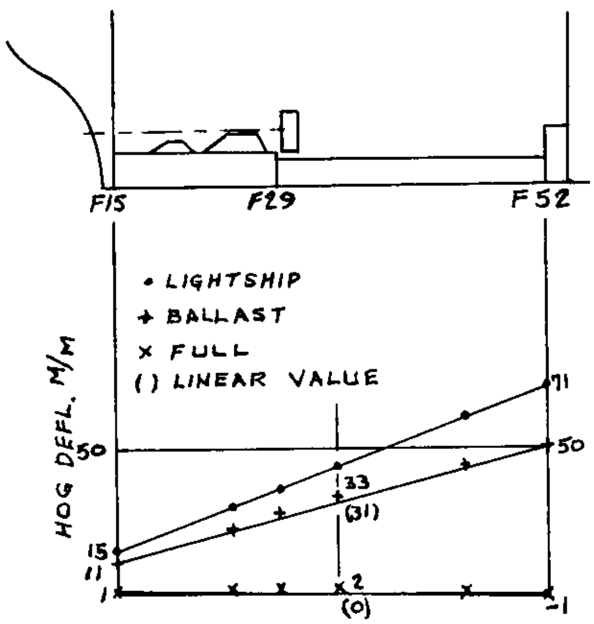


Figure 18 - Deflection of Hull Girder in Way of Engine Room

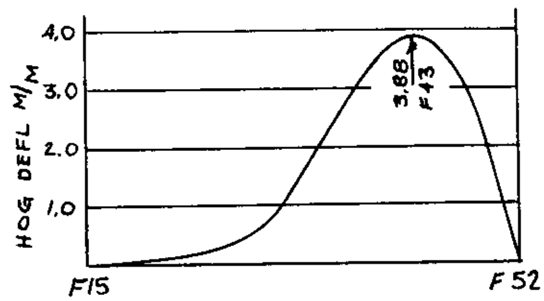
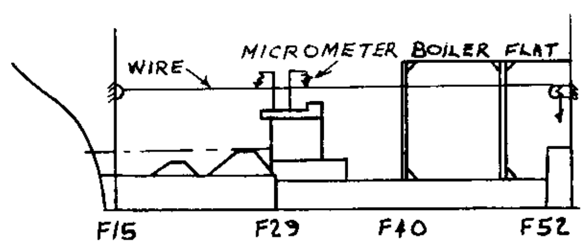


Figure 19 - Deflection at ℄ Due to 6.1 Meter Draft Change



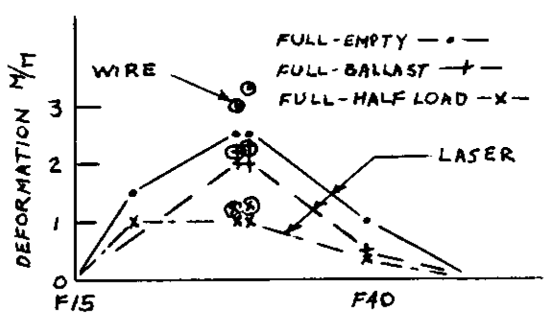


Figure 20 - Measured ℄ Deflection

after peak bulkhead at the after end of the engine room along the vessel centerline. This set-up is illustrated in Figure 20. Micrometers and clock gages were positioned to measure movement in way of the gear casing as the vessel draft varied during loading and discharging.

2) A laser instrument was installed on the vessel centerline at the engine room after bulkhead and directed toward the engine room forward bulkhead. At intermediate points along the vessel's centerline the deflection of the double bottom in relation to the laser beam was measured. The laser device proved to be highly sensitive to vessel vibrations and the results were somewhat less repeatable than for the piano wire. Not withstanding this, the laser proved useful in corroborating hull deflection trends.

The results of the shipboard measurements are given in Figure 20. Not only were the deflections considerably greater than

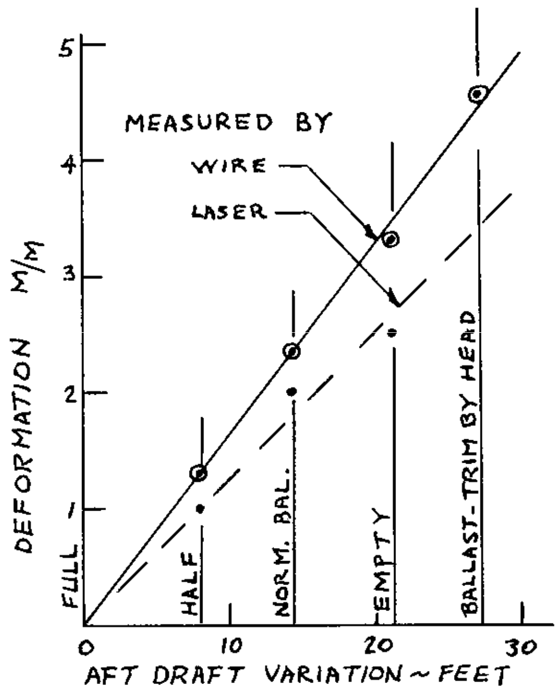


Figure 21 - Deflection at Bull Gear Forward Bearing

calculated, but the point of maximum deflection was much further aft than anticipated. In this case the maximum deflection occurred at the main gear casing.

The deformation at the bull gear forward bearing as a function of the after draft variation is plotted in Figure 21. The slope of 1.65mm per 10 feet of draft is almost four times the calculated deformation of 0.42mm/10 feet previously mentioned. The causes for the poor analytical results are not known but are assumed to be primarily due to faulty boundary condition assumptions and disregard of the effects of interconnecting structures and pillars.

The line shafting of this class of vessels was realigned taking into account reactions due to vertical deformation of the local structure as well as the whole ship as a girder.

Today's Tankers

In view of these various parameters working to create certain incompatibilities between the hull on one hand and the main propulsive equipment on the other, one of two choices seems available:

a) Reduce the stiffness of the shafting and thereby adjust the equipment to the flexibility of the structure, or

b) increase the stiffness of the foundations and double bottom structure and thereby adapt the structure to the reduced flexibility of the machinery.

The latter method is recommended. Serious consideration should be given to increasing the depth of the engine room double bottom floors and to increase the effective rigidity of the engine foundations. Mounting the propulsion plant as far aft as practicable will locate the equipment where the floor breadth is less and thus reduce the effect of double bottom deflection. References 14 and 16 have pertinent discussions on this point.

Summary and Recommendations

Continued cooperation and communication among builders, operators, designers and researchers are needed particularly in areas of full scale research to continue practical application of advanced technology.

Three specific cases giving two basic problems encountered with tankers designed and built in the mid-1960's have been presented. With reference to transverse strength one case of severe damage was described, evaluation of which led to reinforcement of the structure. Utilizing advanced structural analysis techniques reanalysis of other existing vessels led to their reinforcement. As a result of the experiences described, the following are indicated:

1) It is recommended during vessel design to utilize the latest finite element analysis techniques to determine stress levels in all major areas of vessel structure.

2) Use realistic loading assumptions, being cognizant of the extreme conditions to which the ocean environment subjects vessels. Pressure head assumptions approximately 5 meters above the main deck are considered appropriate.

3) Major problems often originate due to lack of proper attention to rather basic and apparently minor details.

a) Maintain continuity of structure and avoid abrupt changes in section.

b) Strut designs should favor balanced or uniform cross sections.

c) Adequate panel stiffness against buckling should be provided.

The relative flexibility of large ship engine room double bottoms coupled with high power, low RPM main propulsion equipment results in a basic incompatibility between ships machinery and its supporting structure. It is recommended that the double bottoms of large tankers be designed to minimize hull deflections by deepening double bottoms and increasing the effective rigidity of foundations.

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DISCUSSION

Huynh duc Bau, Visitor

This paper is, in several aspects, very laudable. The author should be particularly commended for the very practical manner in which the damages have been analyzed and the corrective measures explained. Indeed, most unfortunately, today's ship designers having to deal with a tremendous amount of computer's print outs, seldom can indulge into lengthy meditation over causal relationships between loads and structural behavior.

In regard to different questions raised by the author a few answers can be provided. First of all the shipowners should not be over-anxious of buying "yesterday's product in today's market place." Indeed, Classification Societies always place the highest priority into structural integrity regardless of Rules changes. For obvious practical reasons Classification Rules cannot be updated on a daily basis to reflect the latest in house or otherwise acquired technological improvements. However, when reviewing the reasonableness of a given design, Bureau Veritas members always apply the most up to date technique known to them. As a matter of fact, the "Shadow Rules" reflecting the latest changes (to be published) are simultaneously used with the existing Rules for comparison purposes. This practice most certainly is also adopted by other Societies.

With references to Case I transverse strength damages this discusser would like to:

(a) fully agree with the author's judgement regarding the unbalanced design of the strut.

(b) seek further classification concerning conclusion (2) of page G7. Did the 1968 reanalysis, include investigation of the transverse ring's behavior under dynamic loads (shocks)? What was the author's judgement of the quality of workmanship particularly in areas of discontinuities in deck and bottom transverses face plates?

(c) Call the author's attention to the usually lack of significance of the mentioned test condition (center tank full, wing tank empty) regardless of the vessel's draft over the behavior (stress) of transverse members. This loading condition is primarily aimed at checking the scantlings of the strut(s). It thus would be interesting to know whether a rerun of the transverse analysis with reinforcements made only to the strut's scantlings would modify the stress distribution.

(d) In connection with the above, determine whether the damages presented the same degree of severity in way of transverse rings where the struts may possibly be reinforced by transverse end stringers of the wing tanks transverse bulkheads.

(e) disagree, however respectfully, with the author regarding the absolute necessity of avoiding discontinuity in the face plates. Indeed, several designs of this type, have been proven successful in service. The decks and bottom transverses are stressed differently than the side's or bulkhead's transverses. In obvious areas, more section is needed for shear (either by effect of external pressure or by forces due to the relative deflection between the side shell and the longitudinal bulkhead). Thus weight and cost constraints compel one to optimize the scantlings of the face plates. The important aspect is to carefully provide for a smooth stress flow by proper tapering as well as adequate tripping brackets. The discontinuity is somewhat dramatized in this "rounded face plate" design as opposed to the European straight design (with transition brackets). Each of both designs has its own merit. The obvious inconvenience in the design of this vessel resides in a difficult stabilization of the strut in way of the connection with the side's and bulkhead's transverses.

(f) ask the author to provide for more information regarding the percentage of stress increase owing to higher pressure head, particularly shearing stress imposed by higher relative deflection shell/bulkhead in bottom, strut and deck transverses. Indications relative to the heeled condition should be very much instructive. Similar calculations performed by this discusser on OBO of 250,000 dwt range have failed to indicate needed reinforcements, taking into account the scantlings as required by other loading conditions (full or light ballast).

Finally the author's recommendation (3c) page G13 is wholeheartedly agreed to. It is somewhat strange to notice in several designs that approach to panel stiffening against buckling has been conducted in all but the most natural way consisting (as was adopted in the author's repairs) in fitting stiffeners parallel, close to the face plate (1/3 of web's height for instance) where axial stress is expected to be high and vertical flat bars in areas where shear stress is at its peak;eg. between the above "panel breaker" and the primary longitudinal stiffeners.

AUTHORS' CLOSURE

Mr. Huynh duc Bau's comments are appreciated.

With reference to Item (b) of his discussion, no shock analysis was done during the re-analysis of either of the subject tankers. Static equivalents were used reflecting the estimated head experienced by vessel A in heavy seas. Much work has been done and a lot has been written about assumed practical heads for primary strength determination. Usually measured stresses are compared against stresses calculated from theoretical waves. The author believes more work can and should be done in the area of assumed practical head for local strength wherein actual stresses are similarly compared to those theoretically derived.

The structural test conditions mentioned in paragraph (c) is considered a useful and inexpensive way to check the structural design. Obviously an 8 foot head in the center tank with wing tanks empty is not quite the same as such a head loading the vessel externally as is accomplished by wave action.

To answer paragraph (d) all webs were fractured. The influence of the transverse bulkheads could be noted only inasmuch as the forwardmost and aftermost webs showed less displacement than the center web. The author agrees with the discusser's comments regarding the importance of proper tapering. It is, in the author's opinion, a method recommended for avoiding discontinuities.

Flat bar stiffeners in the web were determined necessary as a result of the heeled condition by a factor of 1.2 to 2.0 than in similar loading conditions with the vessel unheeled.