THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS
74 Trinity Place, New York, N.Y., 10006
Paper to be presented at the Ship Structure Symposium
Washington, D.C., October 6-8, 1975

Instrumentation–the Only Way

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ABSTRACT

Communication between the theoretical analyst of ship structures and the practicing naval architect can be improved by the mutual use of experimental full-scale data from instrumented ships. Such data can provide information on seaway loads, ship responses, and the transfer function between them. In addition, full-scale data can be used to verify or modify theory, investigate operational problems, and determine, through a calibration experiment, ship responses to applied loads in structural regions where calculations are difficult or impossible. Three examples taken from instrumentation projects undertaken on the SL-7 Class Containerships are presented.

INTRODUCTION

The title of this paper and today's general heading of "Yesterday's Technology, Today's Ships" probably sound a bit extreme. While these titles may be overdramatic, the authors feel that they do express the sense of frustration over the time required for current theory to be placed in practice, and over the lack of data available to assist a ship designer in making judgments as to which analytical technique will most accurately predict the characteristics of new construction.

The main problem, of course, is communication. The theoretical analyst usually has some disdain for the ship design practitioner who wants the analyst's formulae reduced to plug-in, tabular form so that he, the practitioner, can use them. On the other hand, the practitioner may not grasp the significance of the analyst's advanced theories, and may lack the motivation to understand them. A median ground must be sought, therefore, where the analyst brings himself to a semi-practical level and the practitioner attempts to pull himself out of the handbooks. One excellent common ground where better communication can result is in the use of experimental data. Full-scale data collected from a properly instrumented ship presents a set of numbers which should be understandable by both parties.

Collection of full-scale data is necessary from a number of standpoints. Any predictions resulting from mathematical analyses of experimental models must accurately characterize the actual structure, or must be correctable in a known way to correlate the technique to the actual structure. Full-scale data, properly interpreted, provide the criteria against which all predictive techniques of structural response must be judged. A second but equally important use of full-scale data is to provide an estimate of the input loads which form the basis of the rational design. Such loading data can be gathered directly from a characterization of observed service conditions, such as wind and wave probability distributions, or inferred from the response of the vessel to the combination of these conditions. The latter scheme requires a knowledge of the structure's input-output or transfer function which again can be provided by adequate full-scale data describing actual loads and responses. In sum, full-scale data can provide three indispensable parts of rational design: input loads, responses, and the derived characteristics of the link between the two.

Full-scale data collection sounds easy, but as with any research project certain basics must be applied in order to obtain credible data. We list these general steps that should be taken in any instrumentation project:

1. The analytical community should specify where the theoretical modeling may be weak and what data are needed for verification or for use as inputs to the model. In conjunction with the practitioner a useful program can then be formulated.

2. An experienced instrumentation team should design and install

the data acquisition system and follow the project through data reduction.

3. Enough data should be collected to answer the questions posed within some agreed-upon limits of accuracy, and to eliminate secondary or extraneous influences.

This sequence is exactly what the Ship Structure Committee has done in the past and is doing today. Every Ship Structure Committee project receives technical supervision from a Project Advisory Committee of the National Academy of Sciences/National Research Council, which provides inputs from many related disciplines. This paper deals with one particular current project of SSC: the instrumentation program for the SL-7 Containerships (1, 2).

This program, a jointly funded undertaking of Sea-Land Service, Inc., the American Bureau of Shipping and the Ship Structure Committee, represents an excellent example of cooperation between private industry, regulatory authority and government. The goal of the program is to advance understanding of the performance of ship hull structures and the effectiveness of the analytical and experimental methods used in their design. While the experiments and analyses of the program are keyed to the SL-7 Containership and a considerable body of data will be developed relating specifically to that ship, the conclusions of the program will be completely general, and thus applicable to any surface ship structure.

The program includes measurement of hull stresses, accelerations and environmental and operating data on the SS SEA-LAND McLEAN, development and installation of a microwave radar wavemeter for measuring the seaway encountered by the vessel, a wave tank model study and a theoretical hydrodynamic analysis which relate to the wave-induced loads, a structural model study and a finite element structural analysis which relate to the structural response, and installation of long-term stress recorders on each of the eight vessels of the class. In addition, work is underway to develop the initial correlations of the results of the several program elements.

It will be some time before the various correlations and comparisons are made and the final judgments are in concerning the relationships of the various predictive techniques to the behavior of the real ship. The intent of this paper

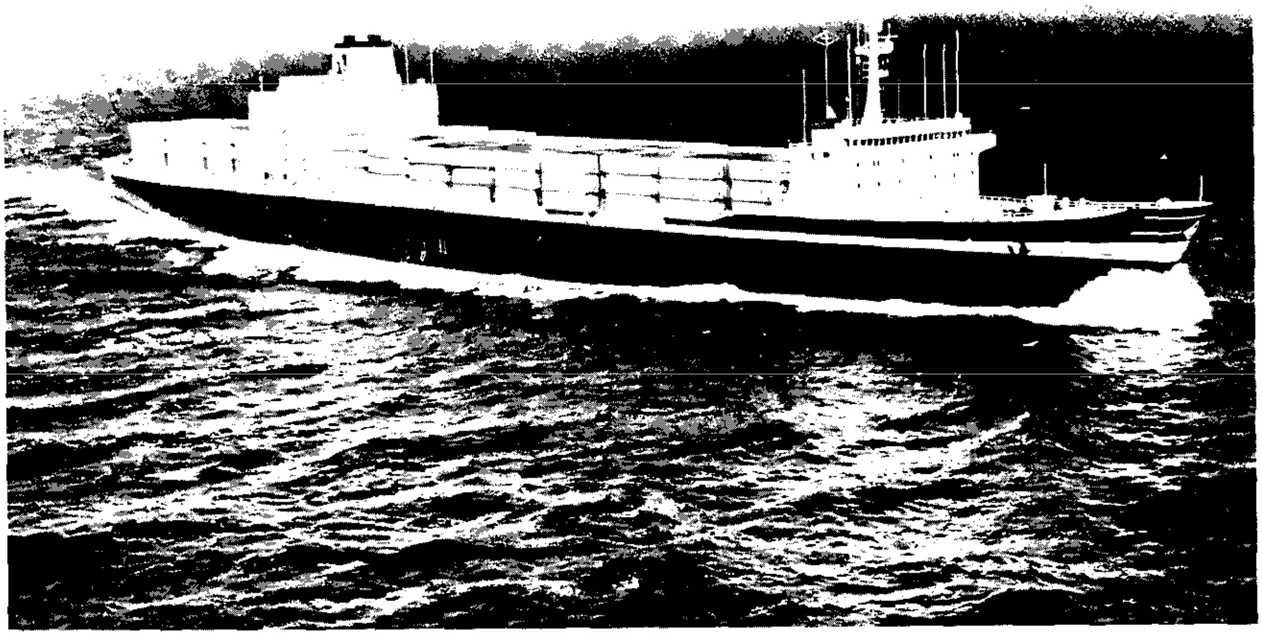


Fig. 1 SS SEA-LAND McLEAN - First SL-7 Containership in a Class of Eight

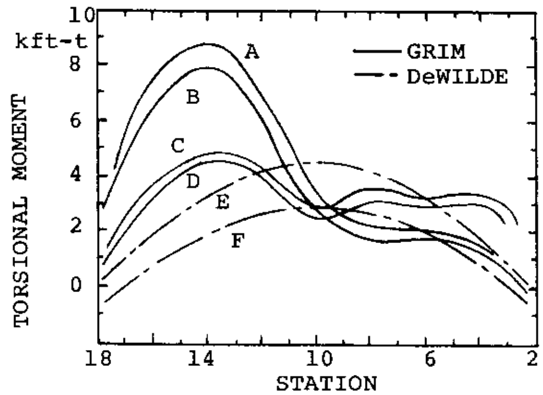
is to provide a practical perspective on the use of full-scale experimental data as an aid to ship design by discussing several examples.

THEORY VERIFICATION

One of the prime uses of instrumentation is the verification of theory. The analytical community tends to look upon experimental data as "good" if it agrees with theory, and "bad" if it does not. It is therefore important, as mentioned, that the analytical community understand and accept the method of instrumentation and its application to the particular design problem involved.

In order to illustrate the designer's concern with theory, we would like to present an actual design problem associated with containerships of the SL-7 Class illustrated in Figure 1. One of the peculiar design problems associated with the containership open hull structure is the consideration of torsional moment effects. While the hull torsional moment does have a minor effect on the total vertical and lateral bending stress experienced by the upper structure in the ship in open areas, it causes major detail design problems at the intersection of transverse hatch girders and the longitudinal box girders, and at the ends of the ship. The generally unrestrained midsection of the vessel free to deflect in torsion is, of course, restrained by the decked-over ends of the ship. Severe restraint-of-warping stresses occur at the ends of the ship and similar high stresses will be found at the intersection of longitudinal and transverse deck structures. Aside from the high local stresses, deflection of the hull structure with relation to the hatch coaming/longitudinal hatch covers produces an extremely complicated design problem. An excellent RINA paper (3) describes some of these problems as they applied to the OCL ships. Most of these stresses and deflections are directly related to the magnitude of the torsional moment and, more importantly, its absolute value at any point along the length of the ship.

With this in mind, we call your attention to Figure 2 in which the calculated values of the torsional moment for the SL-7 Containership based on the methods of De Wilde and Grim are shown. Grim's equation allows more relative wave directions to be calculated, and his results seem more plausible with regard to the shape of the ship, while De Wilde gives only a simple cosine distribution. These curves have been plotted for wave height of 23.3 feet (the largest wave height required by ABS at the time of the original calculation) and 38.0 feet for demonstration purposes. Waves have been encountered in excess of 38.0 feet, so that additional curves



Key: A-Bow Sea, 38 ft Wave
B-Bow Sea, 23.3 ft Wave
C-Quartering Sea, 38 ft Wave
D-Quartering Sea, 23.3 ft Wave
E-Bow Sea, 38 ft Wave
F-Bow Sea, 23.3 ft Wave

Fig. 2 Some Predictions of Torsional Moment Distributions for the SL-7 Class Ships

could be drawn to cover all cases. Using the top curve (Grim, bow sea, 38.0 feet) as an envelope curve, we think it can be agreed that this is a gross approximation at best. With a good finite element model and a proper torsional moment distribution, restraint-of-warping stresses and hull deflections could be fairly accurately predicted. As it stood some 5 years ago, however, the finite element model predictions could be off by a factor of 3 depending on what theory and wave height assumption was used.

During the second season of data acquisition from the SS SEA-LAND McLEAN, a severe storm with waves up to 50 feet height was encountered. In response to a large slam, a midship torsional moment in excess of 100 Kft-tons was measured as determined by extrapolation from the calibration data. This was an instance of a dynamic stress-producing load measured in service but not covered by the theoretical predictions. Under the same conditions, stresses in the deck just forward of the Aft House (Frame 142) were recorded which were close to the highest levels predicted by a Finite Element Method (FEM) analysis (see Figure 3). The FEM analysis (4) found this to be the most highly stressed area on the ship. Seaway measurements, however, found the hatch corner just aft of the Forward House (Frame 290), which has a geometrically similar cut-out, to exhibit an even higher stress. It is interesting to note that although no analysis combining vertical, lateral, and torsional loading predicted high deck stresses in this area, it is the only point of structural hull failure encountered on all eight vessels of the class. There are several reasons why these stresses are higher on the real

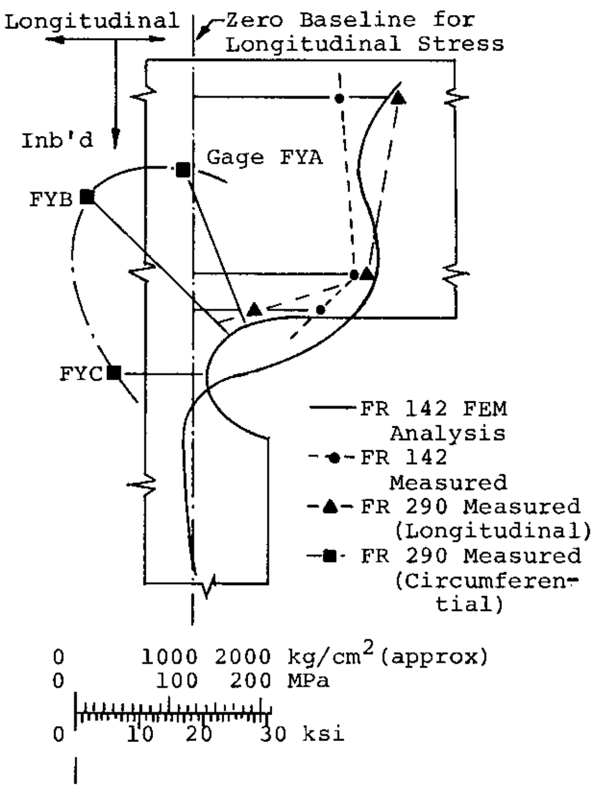


Fig. 3 A Sample Comparison Between FEM Stress Prediction and Measured Stresses for Two Hatch Corners

ships than in the analysis:

1. Secondary dynamic wave loadings due to high ship speed in heavy seas may produce high short-term stresses throughout the bow area. Since the forward end of Hatch No. 1 has the least modulus in this area because the box girder area is reduced, momentary stress extremes do occur.

2. Warping stresses due to the roll of the vessel in a seaway may be increased by superposition of lateral bending mode stresses.

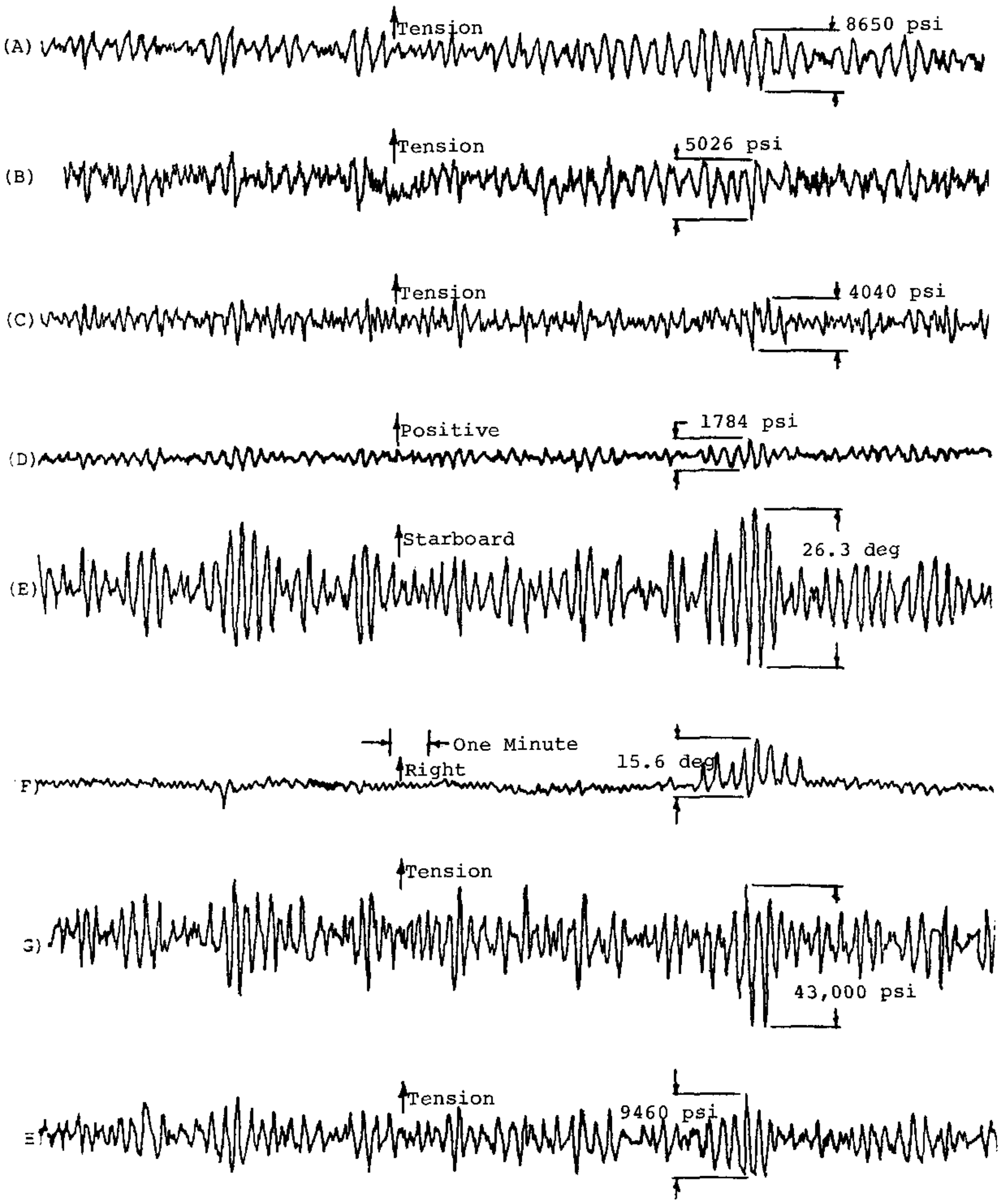
3. Although more prevalent in a quartering sea, the directional shearing of the bow and resulting compensating steering maneuver may add to lateral and torsional moment.

With these factors in mind, the third season data acquisition program was restructured to investigate some of these areas. Very high circumferential stresses at the hatch corners were measured as shown in Figure 3. Perhaps even more significant is the frequency of occurrence of these high stress levels, and their correlation to factors other than wave height and direction. Simultaneous analog magnetic tape recording allows expansion (or compression) of the timebase of such data for interpretation. An example of such a record from the third season data is presented in Figure 4. During this period quartering waves and swells were only three to five feet in height. Yet, peak-to-trough stress levels (excluding still-water components) were frequently measured in excess of 30 Ksi. Such high stresses, so frequently experienced, could induce fatigue cracking failures even though their absolute levels may be well below yield. When cracking did appear in these areas, it was possible, based on the knowledge of the recorded data for outboard locations, to demonstrate that these were self-limiting localized phenomena and not the first indication of a possible catastrophic failure.

A further review of data such as presented in Figure 4 provides insight relative to what is causing the high deck stress levels. Notice the high correlation between roll, rudder angle, and stress. Apparently, due to the quartering sea, course corrections induce rolls which, due to the extremely fine bow, cause significant combinations of torsional and longitudinal horizontal bending. It is expected that an understanding of this interaction mechanism will result, not only in lower stress levels, but in revised operating procedures for improved motion control and steering.

The authors feel, therefore, that even with the degree of analytical sophistication available today not enough is known about the dynamic loading any one particular ship encounters. Moreover, it is doubtful that the complete loading spectrum of any one vessel will ever be known. If the designer has detailed design problem areas, instrumentation of preceding ships is "the only way" to obtain order-of-magnitude stresses and deflections. New ship designs can then be estimated more accurately by using recorded experimental data and modifying to fit, or by theoretical extrapolation of the recorded data. With sufficient data, of course, new theory can be postulated with increased confidence.

The cost, carrying capacity, speed, and other parameters of today's ships are so sensitive to ship weight that the designer can no longer afford to design on yield strength with a safety factor of 5.0. The stress values shown in this paper indicate that on the SL-7, in terms of stress level, not too much material has been wasted in the upper girder structure of the ship.



Key: (A) Midship Longitudinal Vertical Bending Stress
(B) Forward Longitudinal Vertical Bending Stress
(C) Midship Longitudinal Lateral Bending Stress
(D) Torsional Midship Shearing Stress (+ CW Fwd)
(E) Roll Angle
(F) Rudder Angle
(G) Hatch Corner Gage FYB
(H) Hatch Corner Gage FYC

Fig. 4 Sample Analog Traces for One Instant of Ship Response to Quartering Sea with 3-5 Foot Waves and Swells

INVESTIGATION OF OPERATIONAL PROBLEMS

A second general use of instrumentation is the determination of the dominant variables in a complex design problem. The example we would like to use here is the strut bearing and shaft problems encountered with these new SL-7 containerships.

As originally designed, the intermediate strut bearings (see Figure 5) were a water-lubricated phenolic type, while the main strut bearings were (and are) oil-lubricated Babbitt. Some wiping of the main strut bearings after only short service was attributed to an overloading of this oil bearing caused by excessive wear of the intermediate phenolic bearing and the resulting change in shaft alignment. The intermediate bearing was then converted to an oil-lubricated Babbitt type. Although this particular problem was solved in this manner for some time, considerable oil leakage from the strut bearing seals is now experienced, and thus major bearing problems still exist.

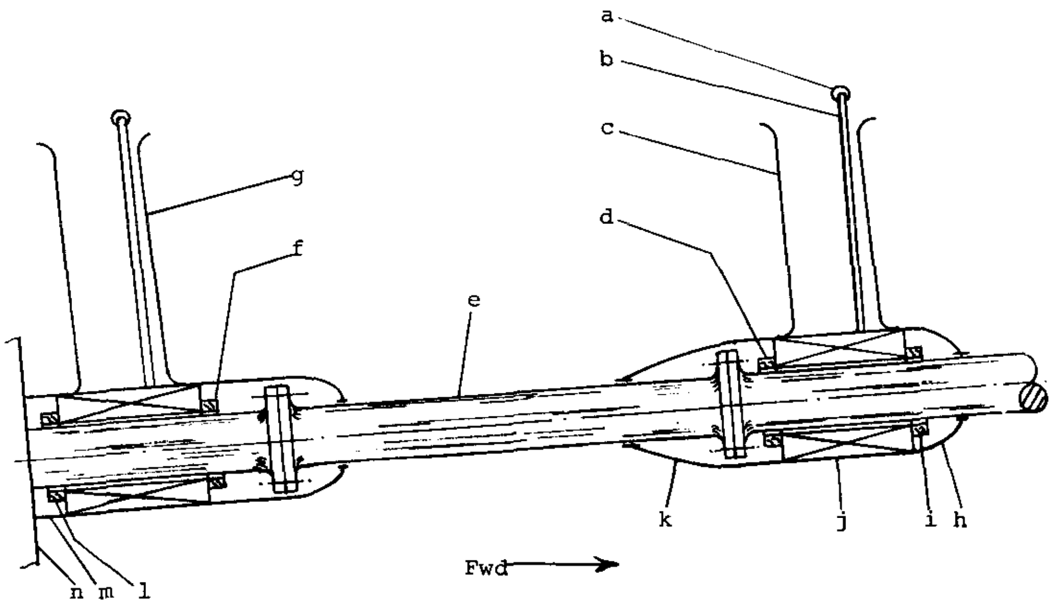
There are an unbelievable number of variables which one could speculate as being associated with this problem, such as:

1. The change in shaft alignment between low- and high-power operation, and the change in bearing loads, strut and shaft stresses, shaft and strut deflections and bearing seal pressures associated with this change in alignment.

2. The dynamic load and deflection in the shafting and bearings resulting from resonant and off-resonant response of the shafting system to the harmonic forces and moments generated at the propeller.

3. The relative movement of the shaft seals resulting from the fore-and-aft dynamic motions of the strut.

4. The effects of ship motion and deflections in the seaway upon



Key:
(a) Hull Penetration (2 places)
(b) Ribbon Cable Bonded to Strut
(c) Intermediate Strut
(d) Aft Bearing Seal
(e) Intermediate Shaft
(f) Fwd Main Bearing Seal
(g) Main Strut
(h) Fwd Fairwater
(i) Fwd Bearing Seal
(j) Bearing Housing
(k) Aft Fairwater
(l) Aft Main Bearing Seal
(m) Rope Guard
(n) Propeller

Fig. 5 Schematic of Outboard Instrumentation for Shaft Seal Investigation bearing reactions and shaft stresses.

5. The influences of the hull structure to which the struts are attached upon the stiffness of the struts.

6. The changes in bearing pressures and journal clearances resulting from the shaft carrying different steady radial loads and steady moments and deflections at slow speed and at rated speed.

7. The shaft motions and bearing pressures generated by the rotating shaft excited by harmonic radial shaft forces and harmonic moments as applied to the main and intermediate bearings.

8. The thermal stresses associated with a shaft having a high heat input for a given length and cooled by water on both sides of the heated section.

9. The dynamics at a bearing of suddenly stopping the shaft, and the stresses in shafting associated with such sudden stopping.

Analyzing all of these variables and conservatively coupling (probably adding) them would produce loads, deflections and enough suspected problem areas to throw terror into the hearts of owners. It is possible that major problem areas would be identified, but the actual values needed for corrective design could not be obtained. Instrumentation showed that many of the above factors were minimal. More importantly, it showed an internal lube oil pressure oscillation magnitude that could not be explained initially.

As noted above, many hypotheses had been advanced to attempt to explain the failure of the seals and bearings. Among the most reasonable of these was the existence of various critical vibratory modes of the shaft and various combinations of bearing oil pressure and varying seawater pressure. Based on these hypotheses an instrumentation system was designed (see Figure 5) which incorporated extensive pressure instrumentation and some displacement sensors.

Although the conditions which were thought to exist in the area of the bearings could be monitored by conventional gauging, it was decided that transducers capable of extended frequency response and a larger dynamic range would be good insurance against

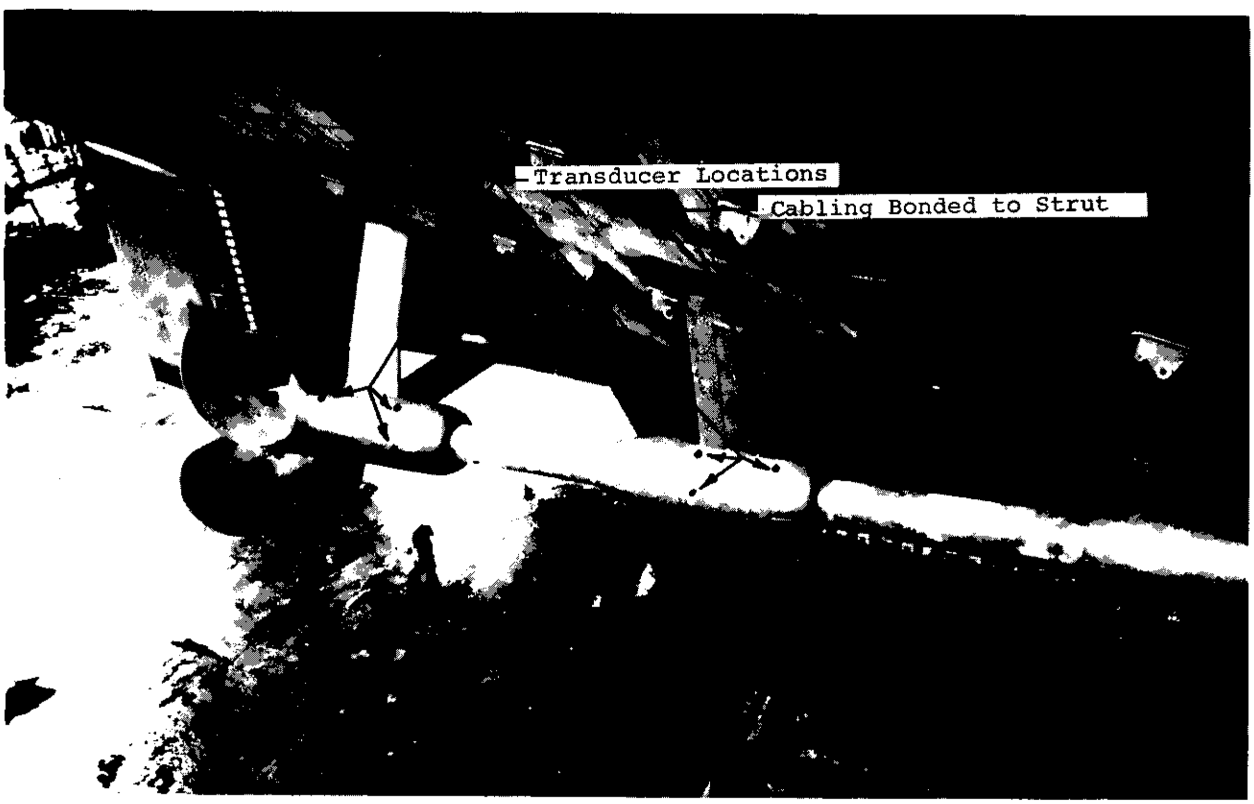


Fig. 6 Completed Transducer Installation on Struts

unexpected conditions. Pressure transducers capable of responding to frequencies ten times greater than expected and pressure five times greater than expected were selected. In addition, the noncontacting proximity sensors used had a linear range twice the nominal bearing clearance, and a usable frequency response in excess of ten times greater than anything which could be encountered in service.

Since the struts were external to the hull in an area which could be subjected to cavitation due to the high ship speed, a special method of hardwiring the transducers to the instrumentation inside the hull was required. After some laboratory experimentation, it was judged that adhesive bonding of ribbon-type cable along the bearing housing and up the strut to a hull penetration away from critical cavitation areas would offer the best tradeoff between reasonable protection for the wiring and minimum surface profile disturbance. The completed installation is shown in Figure 6.

In addition, a system of installation techniques and sealing was developed which met all requirements for adequate protection, schedule considerations and operational security.

Data were taken during a regular Pacific crossing from which the following conclusions were developed:

1. The lube oil pressure, at the ends of the main bearing, oscillated at a frequency exactly five times the shaft rotational rate (propeller blade frequency). The magnitude of the oscillation varied with shaft speed and the applied oil bearing pressure. Below 80 RPM the oscillations were not significant (see Figure 7).

2. The seawater pressures observed on the forward face of the bearing seal were greater than those on the after face, and this differential increased with increasing ship speed.

3. Oscillatory transverse and vertical deflections of the shaft relative to the main bearing seal housing were small compared to the bearing clearance and no resonant modes were noted. The motion of the shaft center was determined at all speeds including the condition of the shaft on jacking gear.

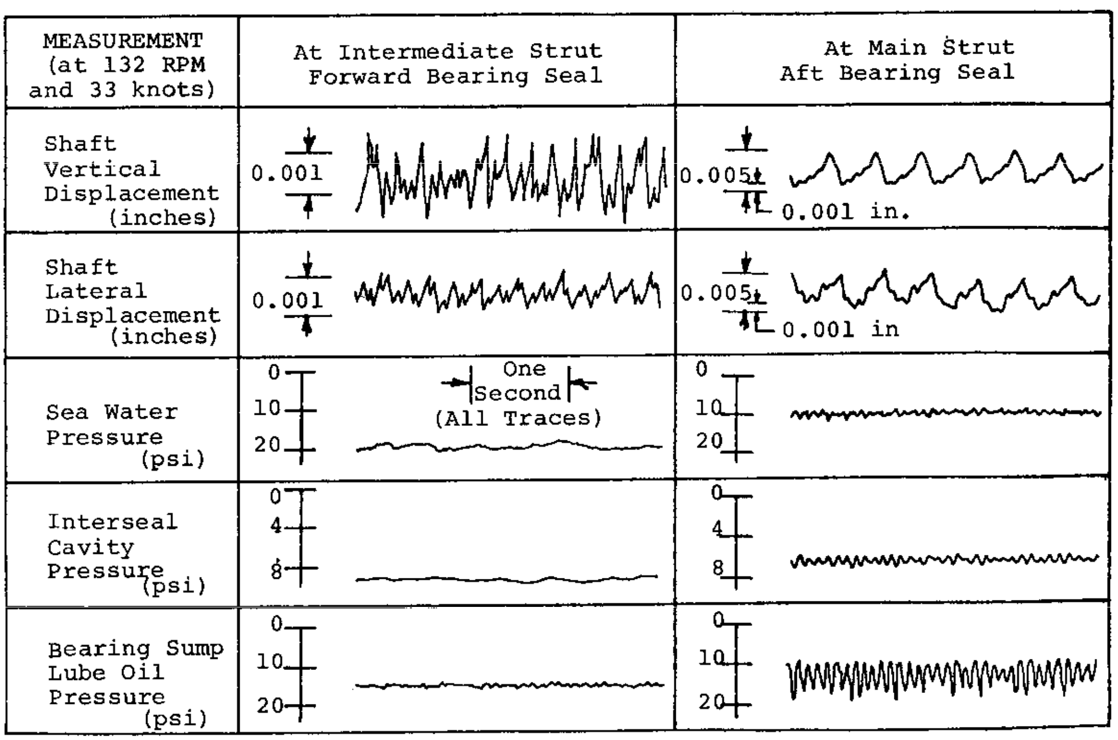


Fig. 7 Measured Displacement and Pressure Data During 33-knot Run

4. Steady state shaft position is significantly affected by ship motion and changes of shaft speed in the lower RPM ranges.

The experimental data showed much higher pressure fluctuations in the main strut as opposed to the intermediate strut. As longitudinal movement of the shaft would have an equal effect in both bearings, it was deduced that an unexpected condition existed: that the main bearing housing oscillated with respect to the shaft in the axial direction. Due to the geometry of the bearing assembly, severe cyclic fluctuations in the bearing oil pressure were developed. Such conditions were apparently beyond the acceptable range of pressures which could be endured by the seal. In addition, due to the design of the lubrication system it was not possible to diminish these pressure fluctuations. Decreasing the internal oil pressure would allow water into the bearings; increasing the oil pressure only resulted in a heavy loss of oil.

As a result of these findings, a further experimental effort was undertaken aimed at determining the loads and deflections which existed at the main bearing. A set of strain gages was used to instrument the main strut during a later drydocking of another ship in the series.

Before this installation was made, however, a retrofit of the problem area was designed which took out the strut oscillatory forces by connecting all of the struts to the stern tube. A similar arrangement of instrumentation of pressure transducers and strain gages was installed on this second vessel. As expected, the previously observed oil pressure fluctuation conditions were found not to exist when this vessel was again placed in service. After a week in service, however, the weld holding this new tube restraining assembly failed allowing again partial movement of the main strut. Upon monitoring the reactivated instrumentation, it was found that the oscillations in lube oil pressure had returned (Figure 8). The strain gages provided additional valuable information.

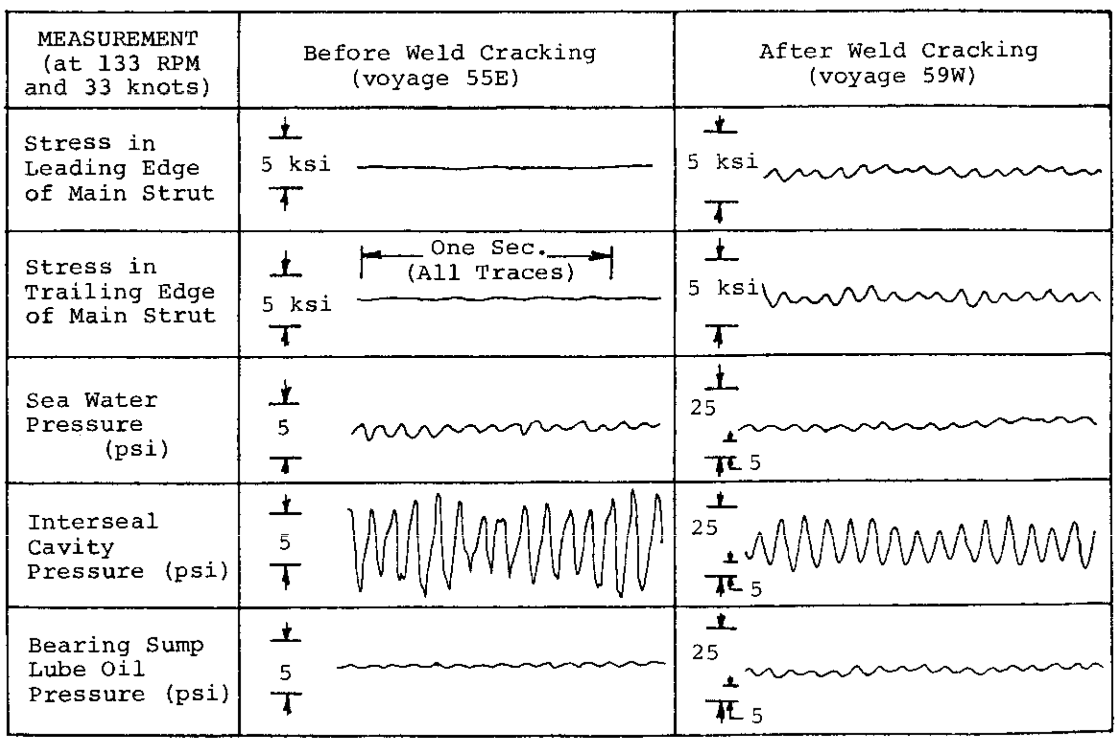


Fig. 8 Comparison of Measured Data Taken Before and After Weld Failure in Restraining Tube

Lessons to be drawn from the above experience include:

1. Unless the problem is absolutely defined, an excessive specification in instrumentation can be valuable.

2. When laying out an instrumentation system, as much data as possible should be gathered. The additional cost, once any installation is made, in added channels or capability is usually minimal.

3. The recording medium should be as flexible as possible. Although a manual reading of a meter is sometimes adequate, any waveform or dynamic data is lost. Similarly, although an oscillograph will record relative phase and waveform data, no time-base expansion is possible later, and such a format does not lead itself to further automated data analysis. Analog magnetic tape recording is an excellent method when the number of data channels is relatively small and there is uncertainty about the nature of the data.

4. A thorough knowledge of state-of-the-art in instrumentation techniques along with an appreciation of the system being measured is essential for good results and a cost-effective program.

5. An on-board observer (for repair, maintenance, and environmental observations) is useful, if not essential, for one-shot efforts involving a planned experimental program. Programs can also be conducted fully automatically if plans provide for the acquisition of the other data (shaft RPM, lube oil feed conditions, etc.).

CALIBRATION OF THE SS SEA-LAND McLEAN

This experiment is a third illustration of the uses of instrumentation data in ship design. Comparing measured ship stress variations with values calculated to result from a known change in loading can provide basic information of great value. And, measuring the response in areas where calculations are impossible can provide information otherwise unobtainable. In addition, a calibration of an instrumented ship will provide verification of instrumentation system performance, and thus of the accuracy of seaway data.

The McLEAN calibration experiment was conducted on 9-10 April 1973 in Rotterdam (5). Six loading conditions were specified in the course of unloading the ship in a manner which maximized both vertical bending and torsional load changes. At each loading condition all strain gages were recorded, as were the existing container weights and locations, and the drafts along the ship. Calculations of vertical bending moment and torsional moment for each condition were provided by the American Bureau of Shipping, and the related stress changes have been compared with the measured values in Figures 9 and 10.

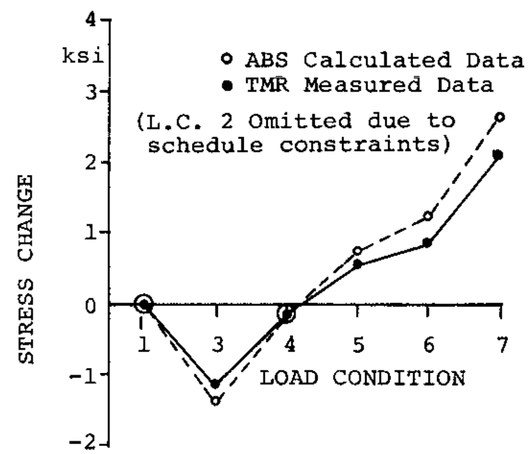


Fig. 9 Change in Measured and Calculated Midship Longitudinal Vertical Bending Stress vs. Load Condition

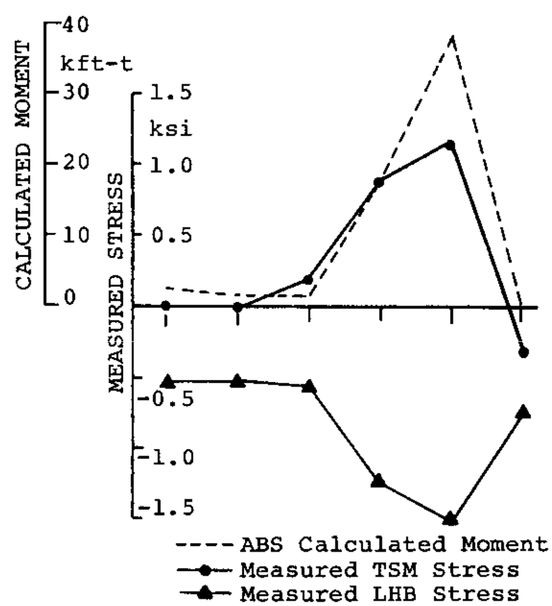


Fig. 10 Measured Torsional Shear Midships (TSM), Longitudinal Horizontal Bending (LHB) and Calculated Torsional Moment vs. Load Condition

In the absence of detailed sectional information suitable for calculating shear stresses using the calculated torsional moments, the moments themselves have been plotted along with the measured shear data. The comparison is generally good. Virtually no output is indicated until the start of the torsional loading condition. Although there is no change in the horizontal bending sensor output for Conditions 1 through 4, an increasing output is indicated for Conditions 5 and 6. This corresponds to the torsional stress distribution (restraint of torsional warping resulting in symmetrically opposite normal stresses about the centerline and torsional neutral axis). It is also possible that some of this is due to thermally-induced horizontal bending which is restrained by the constant-temperature ship bottom.

There is a linear relationship between calculated vertical bending stress changes and measured values, with measured values consistently about 80 percent of these calculated, as shown in Figure 11.

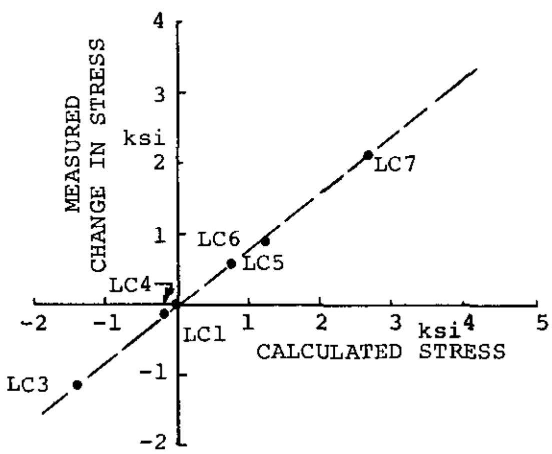


Fig. 11 Midship Longitudinal Vertical Bending Stress: Measured vs. Calculated

The instrumentation of the McLEAN was planned so that the full-scale measurements could be compared with various experimental and analytical models. The calibration calculations, however, provided data for comparison at only a few transducer locations. Nevertheless, the response of other strain gages to the applied loadings is of great interest.

Most of these gages were placed in the regions of hatch corners, especially at transitions from a wider to a narrower hatch width. For example, a set of strain gage rosettes was placed port and starboard just aft of the Forward House, near the Hatch No. 1 corners. Since all of the cargo used to apply the vertical bending and torsional moments was aft of this section, one might expect negligible stress changes. Relatively significant longitudinal stress changes were exhibited, however. These are associated more with the restraint-of-warping stresses than with the bending moment changes. Both the Forward and Aft Houses restrain the free action of the open cell torsional deflections, thus giving rise to significant (in comparison with those induced by vertical bending) longitudinal stress components. These components are especially important at hatch corners near the house structures because the house structure geometries further increase their magnitudes.

Three gages (SY) were located circumferentially about the hatch corner reinforcement on the starboard side just forward of the Aft House (Hatch No. 9). The first of these gages, SYA, displayed the highest recorded strain change of any gage during the calibration. This gage was located 22 1/2 degrees from the longitudinal direction around the cutout ring. These gages were installed especially for the calibration, and were read with a strain indicator.

As presented above in this paper, significant stress variations have been recorded underway from a number of new gages installed around the periphery of the hatch corners during the third season of data acquisition which ended in March 1975. Complete information about all underway data will be published by the Ship Structure Committee in the report of the data reduction from the third season.

The calibration of the McLEAN has generated data which not only gives confidence in the seaway data, but also provides information on loads and responses not otherwise obtainable. The relatively high stresses measured at the hatch corner during the calibration accentuated the concern for the structure in this region, concern which was justified by a subsequent history of cracking. The available seaway data now provide a measurement of the actual stress levels being experienced, values which the theoretical predictions did not show.

CONCLUSION

This paper has presented three examples indicating that there are several uses of experimental data in predicting and understanding the behavior of real ships. No amount of computer or model studies can ever provide the level of confidence in the answers that such real data under real conditions provide. We firmly believe that instrumentation is "the only way" in many cases, and urge that experimental data be incorporated as swiftly as possible into the notebooks of both the academic researcher and the practicing naval architect.

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DISCUSSION

Pin Yu Chang, Member

The authors as well as the members in the Ship Structure Committee have conducted a much needed and useful investigation for the Society of Naval Architects in the Sea Land 7 instrumentation project. This paper is a useful addition to the literature and I expect that much more information will be forthcoming as the project proceeds.

I fully agree with the authors that communication between the theoretical analyst of ship structures and the practicing naval architects can be improved by using experimental full-scale data and that full-scale data properly interpreted provides the criteria against which all predictive techniques of structural response must be judged. There are, however, several points I would like to bring out for further discussion.

First, I do not agree that the theoretical analyst has disdain for the designer who prefers simple formulae. Members of Panel HS-3 are now working on a manual for structural stability. The main goal is to provide simple formulae, tables or charts for the designers.

Secondly, I do not agree with the authors that the analytical community tends to look upon experimental data as "good" if it agrees with theory or as "bad" if it does not agree. A typical example is the stability of shells. Theoretical stress predictions are usually much higher than experimental results. But no analyst has ever said that the experiments are bad.

In my opinion, an experiment is bad if it is planned improperly and/or executed improperly, not because the correlation is poor.

At the present stage of ship strength analysis, we still don't know much about a number of important factors about the structural loading and response of ships. The only way to increase our knowledge about these important factors is for both the analysts and the experimentalists to closely work together. Experiments or instrumentation without the guidance of the theory has little value. Theory without the guidance of experiments is impractical. Detailed and careful interpretation of the collected data is also essential to obtain correct conclusions.

According to the authors "It will be some time before the various correlations and comparisons are made and the final judgements are in concerning the relationships of the various predictive techniques to the behavior of the real ship." It is difficult to understand how the authors can conclude that instrumentation is the only way if the results have not been analyzed.

One of the reasons which the authors used to draw this conclusion is that the theoretical predictions by ABS are different from the seaway measurements. I would like to ask the authors whether these differences occur under the same loading conditions? It seems to me that only 6 loading conditions have been considered in the predictions and the experiments cover many more loading conditions.

In fact very good correlation has been obtained between finite element analysis and strain-gage measurements in the past by ABS and Det Norske Veritas for other ships. The Esso Norway is one example.

It is true that the theory for predicting the seaway loading is not yet very accurate. But the question is, can instrumentation alone solve this problem?

In claiming that instrumentation is the only way, the authors seem to have neglected the difficulties and limitations of experiments.

First, all the measurements from strain gages are only relative. The data shows only the change in strain from the calibration condition. Secondly, if the absolute stress is beyond the yield point of the steel, the stress-strain relation depends also on the loading history. In other words, if the absolute stress at one joint was -25,000 psi, then what is indicated by the measurements to be 50,000 psi is really only 25,000 psi tensile. Similarly a point with measured stress of 10,000 psi may be subjected to an absolute stress of 30,000 psi. This is on the assumption that all strain gages performed properly.

Thirdly, there is the factor of residual stress which cannot be predicted by the theory nor by the instrumentation. With these limitations it seems to be too early to say that instrumentation is the only way.

In view of these difficulties and uncertainties, it is recommended that all the collected data be subjected to careful study and proper interpretation not only from the theoretical viewpoint but also from the practical point of view. This is no easy task. The interpreter must be able to separate the effect of many factors, and must have profound knowledge about the relationships between all the factors involved. The SSC, Sea Land Service, and Teledyne have done a service to the industry in providing us with these valuable data. But we have to interprete and organize them properly before we can use them for the benefit of the shipbuilding industry.

AUTHORS' CLOSURE

We wish to thank the contributors of both the written and oral comments on our paper. Their remarks present an opportunity to clarify and amplify the points which were not adequately covered in the paper due to the requirements of brevity.

It was not our intention to imply an adversary relationship between the theorist and the practicing naval architect. We wish to emphasize, however, that the ship designer requires timely answers that have a high degree of confidence. The work of the HS-3 Panel in rationalizing design data will be a large step in the right direction.
When analysis technique is used, designer's requirements are not being met. The example of the shafting instrumentation given in the paper is such a case. Experimental full-scale measurements do not involve modeling assumptions, do not rely on theoretical approximations and do include all factors whether explicitly recognized or not. Such was the case in the example presented; a phenomenon was present which had not been predicted or even included in the models. Of course, if poorly designed or if the scope of the problem is not appreciated, even full-scale instrumentation can be misleading. Here is the place for close cooperation between the theorist and experimenter.

In the matter of shell theory, the consistently lower buckling loads observed in experiments is a reflection of the "real world", or what is attainable in practice, regardless of the assumptions made by theory with respect to initial straightness, end fixity, uniformity of loading, etc. In the final analysis, theory should predict what is observed in real life in a way that is useful. If theory cannot so predict and the actual structure is available, then instrumentation is the only way to obtain realistic data.

Whether or not the loading conditions assumed in the finite-element analysis matched that measured in a seaway is not the point. The location and magnitude of the maximum stresses predicted were different from those actually observed. Additionally, the magnitude and frequency of the cyclic stresses near the hatch corners were high enough to induce fatigue cracking even under sea conditions much lower than considered in the FEM solution. If the designer had relied exclusively on such predictions, without regard for their limitations and shortcomings, serious errors would have been introduced. While the FEM is a powerful and invaluable tool, it, too, must be tested against the criteria of its ability to reliably predict design parameters, regardless of the assumptions inherent in making the analysis. Here again instrumentation provides the answers for the judgments and feedback for further development of theory.

A reference by a discusser was made to residual stress which, it was said, could not be predicted by theory or instrumentation. In fact, certain techniques, such as trepanning, can determine the residual state of stress. Some NDT techniques can also be used to measure residual surface stresses. Perhaps this is an example of where knowledge of available techniques by an experimenter could aid the theorist.

In sum, the authors do not refrain from using theory, especially when it indicates the parameters influencing a problem. And, of course, much theoretical analysis is well-proven (e.g. calculation of "static" midship bending moments) and in no need of further refinement. We do maintain, however, that for many of the problems confronting the naval architect, theory in many cases is insufficient or too cumbersome, and past experience or current instrumentation is, in practice, the "Only Way".

ERRATA

On page E-3 of our paper, the units of the vertical axis of Fig. 2 should be in tens of Kft-t or 10 Kft-t (instead of Kft-t). On page E-10, the horizontal axis of Fig. 10 should be labeled in a manner identical to that in Fig. 9.