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Highly Skewed Propellers—Full Scale Vibration Test Results and Economic Considerations

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ABSTRACT

This paper presents full-scale test results and discusses the effectiveness of three different highly skewed propeller designs in reducing ship vibration. The test results cover three merchant ship designs, including one OBO vessel and two RO/RO vessels, having installed horsepower ranging from 24,000 to 37,000 SHP. Limited discussion is presented on the relative effectiveness of installing a highly skewed propeller versus adding additional structural reinforcement on a vessel to reduce excessive vibration. Some economic factors that should be considered by a ship owner contemplating a highly skewed propeller for a new ship design are presented.

INTRODUCTION

The subject matter of this paper is concerned with use of highly skewed[1] propeller designs to reduce shipboard vibration. One might ask considering the history of propeller technology wherein propellers have been installed on ships since 1837 why attention should be focused on this old and proven propulsive device? The answer of course is that while more than 140 years of development work have gone into increasing the efficiency and service life of propellers and on a overall basis the technology is well developed, there still remains the possibility for further improvements from standpoints other than propulsive efficiency. Also the trends of modern ship designs wherein greater horsepower is being placed in single screw ships raises the possibility that perhaps other design criteria besides "efficiency" may be equally important.

There is a second reason for exploring the use of highly skewed propellers on ships and that is to attempt to cure instances of excessive vibration after all conventional attempts at cures have failed. This reason was really the primary one for the Maritime Administration becoming interested in the possibility of using highly skewed propeller designs on merchant vessels starting in 1969.

While there have been many instances of severe vibration on ships over the past decades, the difficulty in solving any specific problem has not been a lack of ideas concerning the various potential remedial measures, but rather the uncertainty of success and economic costs associated with each alternative. This paper is concerned with the actual full scale performance of highly skewed propellers installed on three relatively new merchant ship designs, including one Ore/Bulk/Oil (OBO) vessel and two Roll-On/Roll-Off (RO/RO) vessels. It should be noted that one propeller was specifically designed to solve a known vibration problem, another designed to solve a suspected vibration problem and the third merely to demonstrate the full scale performance characteriestics of this old but novel propeller concept.

Within the past five years there have been several technical papers discussing the design aspects of highly skewed propellers. For example papers by Boswell and Cox (1)[2] and Cumming, et al(2) have addressed the numerous design considerations and potential benefits. Thus far however only one paper, that by Dashnaw and Valentine (3) and one report (4) have presented full-scale performance data test results. Throughout this period, however, many potential advantages have been cited. Others, noting the absence of full-scale evidence have cited an equal number of disadvantages associated with highly skewed propellers. Although the basic concept of highly skewed propellers dates back to 1851, no full-scale highly skewed propellers for merchant

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[1] See Appendix A for explanation of propeller terminology.

[2] Number in brackets designate References at end of paper.

Table 1 - "Asserted" Advantages and Disadvantages of Highly Skewed Propellers

Advantages	Disadvantages
• Reduced ship vibration levels	• Costs more than conventional propellers
• Improved shipboard crew comfort	• More susceptible to damage
• Reduced M&R cost of navigation equipment	• Lose 5% propeller efficiency therefore greater fuel bill
• Increased equipment life	• Added propeller weight requires larger diameter tail shaft, stern tube
• Greater propeller service life due to decreased blade cavitation erosion	• Shorter propeller service life due to increased cavitation erosion
	• Inadequate strength

ships were constructed until 1974, some one-hundred and twenty-three years later.

While some of the performance capabilities of the first completed highly skewed propeller were presented to the maritime industry at the First Ship Technology and Research (STAR) Symposium held August 1975, there still remains some uncertainty regarding the advantages and disadvantages of these unique appearing propellers. Table 1 summarizes some of the various advantages and disadvantages that have been cited within recent years. It is hoped that this paper will give more insight into the validity of the advantages and disadvantages outlined in the table.

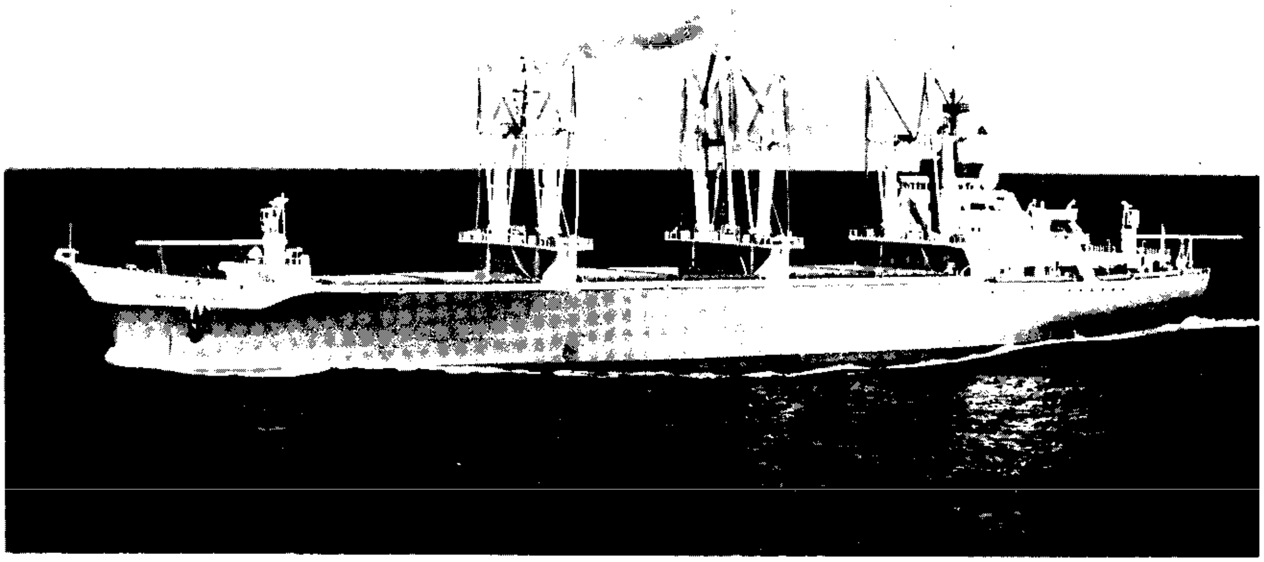

Figure 1 - Sea Bridge Class (MarAd Design C5-S-78a)

Table 2 - Sea Bridge Class Principal Characteristics

Length Overall	602'-0"
Length Between Perpendiculars	560'-0"
Beam, Molded	90'-0"
Depth To Main Deck At Side ℄, Molded	59'-0"
Draft Full Load (Scantling), Molded	34'-0"
Light Ship	10,900 L. Tons
Passengers, Crew Effects, & Stores	150 L. Tons
Fuel Oil	2,415 L. Tons
Anti-Roll Tank	245 L. Tons
Fresh Water	431 L. Tons
Refrigerated Cargo (In Containers)	500 L. Tons
Liquid Cargo	2,300 L. Tons
General Cargo	10,789 L. Tons
Total Deadweight	16,830 L. Tons
Displacement, Full Load (Scantling), Draft	27,580 L. Tons
Cargo Volume, Bale, Cu. Ft.	1,300,000
Containers In Hold (40' Cont.)	245
Containers on Deck	167
Passenger Accommodations	12
Crew Accommodations	39
Shaft Horsepower, A.B.S.	30,000
Speed, Knots	24
Propeller 6 Blades	23'-0"
Propelling Machinery, Cross Compound, Double Reduction, Geared Turbine	

THREE MERCHANT SHIP DESIGNS

The three merchant ship designs discussed in this paper in order of actual ship delivery sequence are: (a) the Sea Bridge class (RO/RO) vessels constructed by Ingalls Shipyard, (b) the San Clemente class (OBO) vessels constructed by National Steel and Shipbuilding Company and (c) the Maine class (RO/RO) vessels constructed by Bath Iron Works.

It should be noted that the highly skewed propeller installed on one vessel of the Sea Bridge class was undertaken to solve a known vibration problem. Installation of the first highly skewed propeller on the San Clemente class vessels was undertaken merely to demonstrate the full scale performance characteristics of the propeller concept. Whereas highly skewed propellers were installed on the Maine class vessels to solve a suspected vibration problem that ultimately never materialized. Principal ship characteristics of the three merchant ship designs are outlined in Tables 2, 3, and 4. Photographs from one ship of each design are shown in Figures 1, 2, and 3.

Figure 4 has been prepared to display the trend of maximum horsepower levels for single screw merchant ships and inspection of this figure reveals that approximately a 20,000 horsepower level was maintained as a maximum plateau for the period of 1951-1966.[3] Starting in 1967/1968 however the maximum horsepower started to climb reaching a new maximum plateau of 50,000 horsepower in 1973. Plotted on Figure 4 are the horsepower levels for each of the subject three merchant ship designs. It will be noted that the first Sea Bridge vessel with a 30,000 SHP machinery plant represented a new high horsepower level at time of delivery of the first ship in 1969. With regard to the San Clemente class and Maine class vessels, these designs entered service approximately seven years after the horsepower levels had been established by other vessels. The timing of the delivery of each ship design relative to the maximum horsepower "line" is important because it indicates in an approximate way the degree of risk being undertaken to design a precedent setting vessel. This matter will be discussed in more detail later in the section of the paper concerning economic and risk aspects.

At the present time 7 highly skewed propellers have been constructed and installed on merchant ships with 5 additional propellers under construction. Table 5 indicates information on the ship design, type ship, and the corresponding number of propellers. It will be noted from inspection of this table that the distribution of highly

[3] See Appendix B for information on historical trends.

Table 3 - San Clemente Class Principal Characteristics

Length Overall	892'-6"
Length Between Perpendiculars	855'-0"
Length On Designated Water Line	875'-0"
Beam, Max. Molded	105'-9"
Depth To Main Deck Aside Molded	62'-6"
Draft Full Load Molded	45'-10"
Displacement At Full Load Draft	99,210 L. Tons
Gross Tonnage, U.S. (Approx.)	43,000
Net Tonnage, U.S. (Approx.)	37,000
Lightship	18,710 L. Tons
Fuel Oil	4,845 L. Tons
Diesel & Lube Oil	50 L. Tons
Fresh Water	350 L. Tons
Crew & Stores	50 L. Tons
Cargo - Ore, Bulk or Oil	75,250 L. Tons
Clean Ballast	30,250 L. Tons
Total Cargo (Deadweight)	75,250 L. Tons
Total Deadweight (45'-11" Mean, S.W.)	80,500 L. Tons
Total Deadweight (39'-0" Mean, F.W.)	62,240 L. Tons
Crew Accommodations	27
Total Accommodations	31
Shaft Horsepower	24,000
Sea Speed In Knots	16.5
Propeller, 5 Blades	26'-0"
Propelling Machinery, Cross Compound, Double Reduction, Geared Turbine	

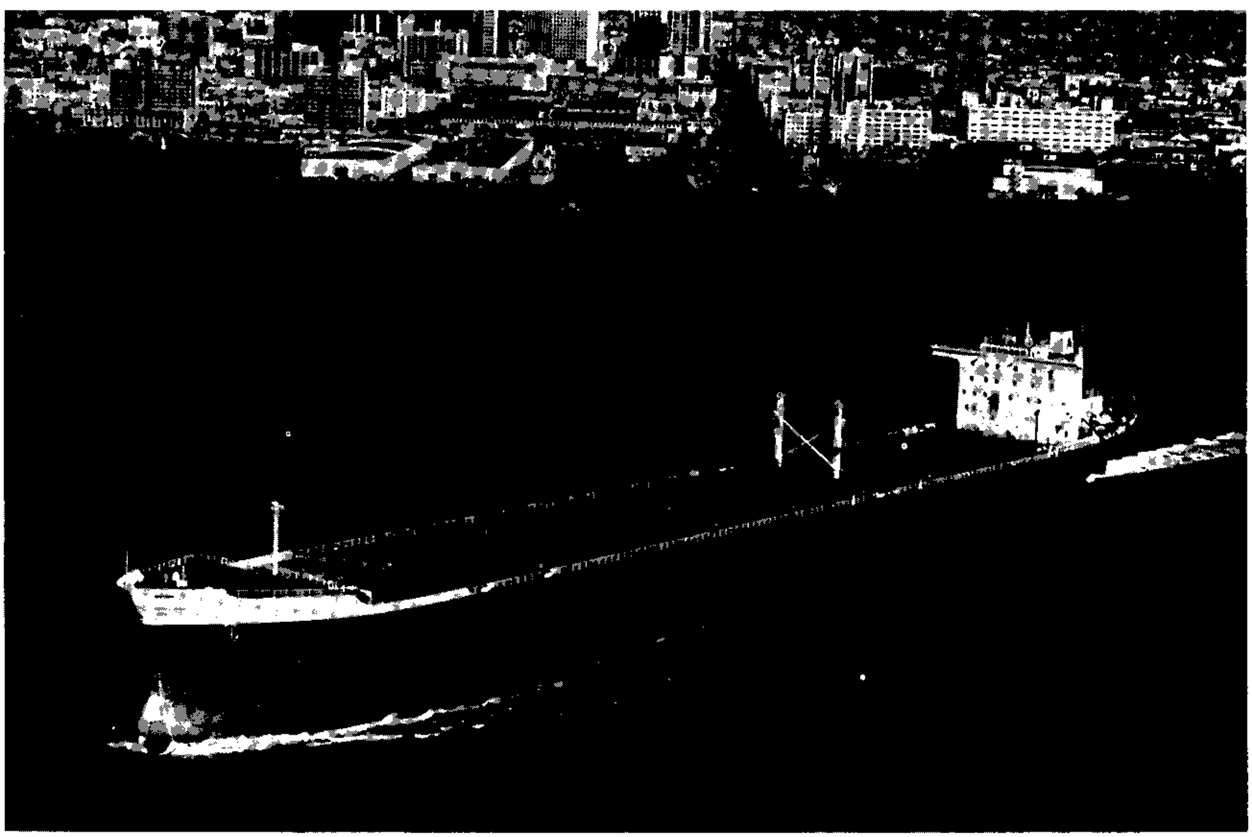

Figure 2 - San Clemente Class (MarAd Design OB8-S-90a)

Table 4 - Maine Class Principal Characteristics

Length Overall	684'-0"
Length Between Perpendicular	640'-0"
Beam, Molded	102'-0"
Depth To "A" Deck, Molded	69'-6"
Depth To "B" Deck, (Bulkhead Deck) MLD.	57'-2"
Draft, Design Molded	32'-0"
Draft, Scantling	34'-0"
Lightship, EST.	14,222 L. Tons
Liquid Cargo	728 L. Tons
Fuel Oil	3,648 L. Tons
Salt Water Ballast	6,221 L. Tons
Fresh Water	243 L. Tons
Total Deadweight	19,535 L. Tons
Displacement Molded, At Design Draft	33,640 L. Tons
Displacement, Total At Design Draft	33,765 L. Tons
Cargo Volume, Bale, Cu. Ft. Holds	1,643,000
Cargo Volume, Bale, Cu. Ft. Ramps	92,660
Area Sq. Ft. (Including "A" Dock Ramps)	146,932
Basic Manning	41
Crew Accommodations (Incl. Spares)	42
Shaft Horsepower Max, Cont.	37,000
Speed, Knots	23
Propeller, 6 (Skewed) Blades	22'-0"
Propelling Machinery, Cross Compound, Double Reduction, Geared Turbine	

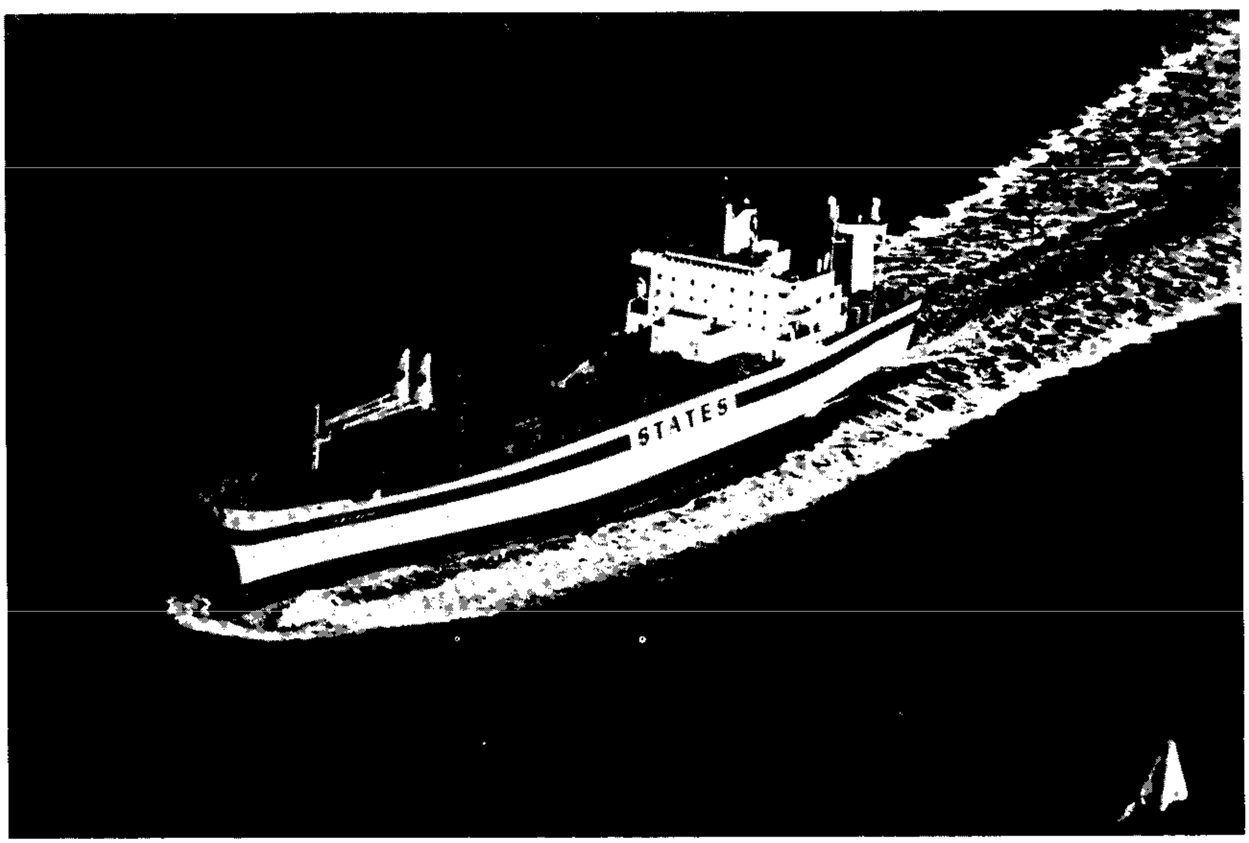

Figure 3 - Maine Class (MarAd Design C7-S-95a)

Table 5 - Summary of U.S. Merchant Ship Highly Skewed Propellers

I. Propellers Constructed to Date

Ship Design	Ship Type	Number of Propellers
Sea Bridge	RO/RO	1
San Clemente	OBO	1
San Clemente	Tanker	1
Maine	RO/RO	4
		7

II. Propellers Under Construction

Ship Design	Ship Type	Number of Propellers
Enterprise (Matson)	Container	1
Navy AO-177	Tanker	4
		5

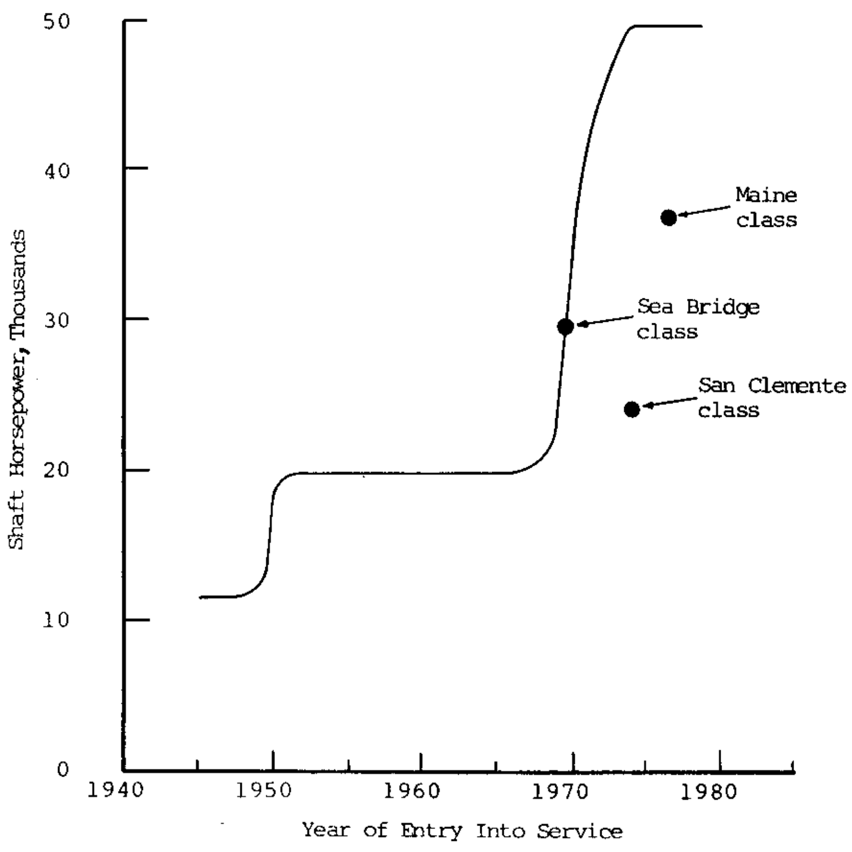

Figure 4 - Maximum Shaft Horsepower Single Screw Merchant Ships

Table 6 - Propeller Characteristics

	Sea Bridge		San Clemente		Maine	
	Original	Skew	Original	Skew	Original	Skew
Shaft Horsepower	30,000	30,000	24,000	24,000	37,000	37,000
RPM, Design	110	107	92	92	120	120
No. of Propellers	4	1	2	2	1	4
No. of Blades	6	6	5	5	6	6
Diameter, Feet	23	23	26	26	22	22
Skew, Degrees	N/A	60°	N/A	72°	10°	30°
Skew, Percent	N/A	100	N/A	100	17	50
Weight, Pounds	75,700	80,000	106,000	116,000[4]	80,000	90,000
Material	NiAlBz	NiAlBz	MnBz	MnBz	NiAlBz	NiAlBz

skewed propellers already includes a broad range of merchant ship types such as: RO/RO, OBO, tanker and a containership.

This trend is expected to continue as more experience is gained with these interesting propellers. However for purposes of this paper it should be noted that the information presented herein is considered to reflect full scale performance results of virtually all merchant ships now operating with highly skewed propellers and represents results from approximately 60 percent of the ship designs listed in Table 5.

Once the decision has been made to move forward with the design and installation of a new shipboard feature, such as a highly skewed propeller, the test and evaluation should be relatively straight forward. One first measures the baseline performance of the ship fitted with the conventional propeller and then replaces the unit with the new propeller repeating loading conditions and test measurements. Normally in obtaining measurements of full scale vibration performance, one is concerned with: (a) hull girder motion, (b) longitudinal machinery vibration and (c) superstructure vibration. SNAME Codes C-1 and C-4 (5) and (6) respectively, outline the basic test procedures, instrumentation requirements, analysis procedures to insure the quality of the recorded data.

Comparison of the "baseline" results with the "after" results then indicates the degree of improvement that was achieved. Since any full-scale data measurement program seems inherently to always leave some gaps of information due to equipment failures, less than ideal test conditions, omissions of data recording, the perfect comparison is seldom achieved in practice.

For each of the three ship designs discussed in the following pages, the basic effort has been to first obtain the baseline measurements. The next step of repeating the measurements with the highly skewed propeller has however varied somewhat from one project to another. In the instance of the Sea Bridge class, one ship the DEFIANCE (ex MORMACSEA) was used for all tests. With regard to the San Clemente class and the Maine class, here sister ships were used for test purposes. These vessels were the ULTRAMAR and ULTRASEA, and the MAINE and NEVADA.

Design information on the conventional and highly skewed propellers for the three different ship designs are outlined in Table 6. The discussion presented in the following pages outlines additional background information concerning each vessel, describes the propeller design test programs, presents the full scale vibration results and lastly mentions any special noteworthly incidents that have occurred with the highly skewed propellers.

I. Sea Bridge Class

Overview/Background. Early in 1969 the SS MORMACSEA was delivered to Moore-McCormack Lines, the first of four identical ships being constructed by Ingalls Shipyard. This 30,000 shaft horsepower vessel shown in Figure 5 was at time of delivery the highest horsepower single screw merchant vessel that had ever been constructed in the U.S. Excessive vibration was experienced at the outset and a number of remedial

[4] Second propeller has slight pitch correction and change in material from MnBz to NiAlBz. Weight reduced to 109,000 pounds.

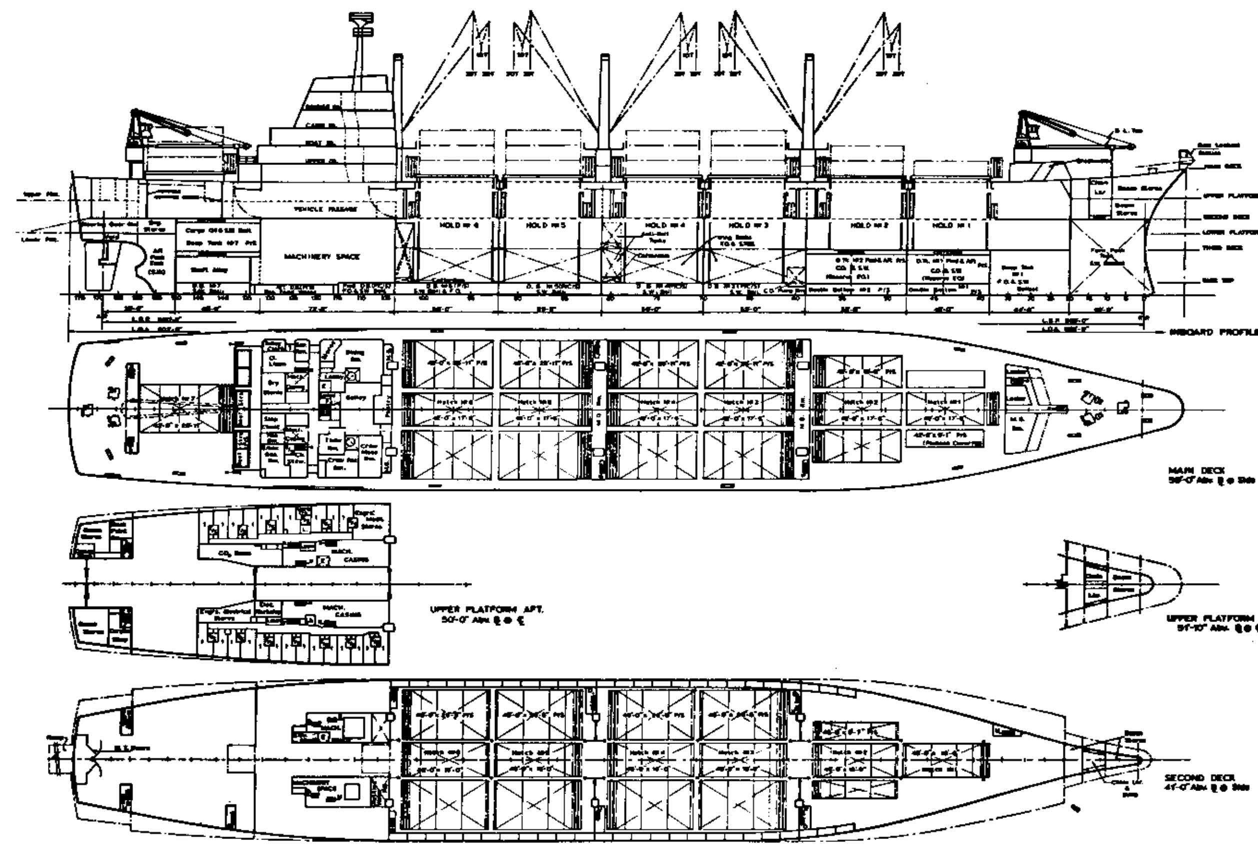

Figure 5 - General Arrangements, Sea Bridge Class

corrective measures were undertaken to improve the operating performance. Although a variety of work was considered at that time with some structural modifications incorporated in the vessel successfully curing some of the lesser problems, the MORMACSEA and her three sister ships were utlimately accepted by the owners without a complete cure to the vibration problem ever being achieved. Concern also had surfaced about the possibility of high propeller blade erosion due to cavitation (7, 8, 9, 10). During 1970/1971 work was carried out at the David Taylor Naval Ship Research and Development Center (DTNSRDC) to explore the possibility of installing a highly skewed propeller on one of the "Sea Bridge" vessels in order to reduce the level of ship vibration and minimize the effects of cavitation. All four ships of the design were subsequently sold to American Export Lines and renamed.[5] The highly skewed propeller work however resumed after a pause of one year and was further developed. While utlimately another ship, the SS ULTRASEA discussed later in the paper, was to be the first U.S. merchant vessel actually fitted with a highly skewed propeller, the work started and still ongoing for the "Sea Bridge" vessels served as the basic foundation work for all the other highly skewed propeller projects discussed in this paper. Because of the special importance of the MORMACSEA/ DEFIANCE work, the history of events and nature of the vibration problems and overall program will be discussed in some detail.

Initial Problems and Corrective Steps - (1969-1971). Early in 1969 the MORMACSEA was nearing completion and extensive analytic and model work had been completed aimed at avoiding possible vibration problems. Due to the high power of the vessel this work was much more than normally undertaken. The propeller design had been tested several times to check performance characteristics and the extent of cavitation and also special blade erosion tests had been completed. As the vessel neared the time for builders trial, three separate vibration shaker tests were made on the vessel in the months of February and March to locate and correct items subject to local vibration. At the end of March the vessel was taken out to sea for the first time on a builder's trial and observers posted to note vibration problem areas. After completion of the first official sea trial in early April, approximately twenty-five structural modifications were made upon returning to the shipyard. A second official sea trial was held some two weeks later and five more structural changes were made.

Shortly after taking delivery of the MORMACSEA the owner indicated that there were still several shipboard locations that appeared to be vibrating. In June a vibration survey was made during a voyage from Baltimore, Maryland to Elizabeth, New Jersey and numerous shipboard locations were indicated as being of concern. A number of structural modifications were made in the summer and during a coastwise passage in water depths of ten to thirteen fathoms at full power the vibration was judge by the crew as being excessive. During the trans-Atlantic passage under full power new locations of vibration were discovered and passengers complained of doors rattling and beds shaking. The after end of the unlicensed crew rooms on the Upper Platform deck and master's stateroom displayed increased vibration.

In November, the MORMACSEA was at an East Coast shipyard undergoing repairs and in order to examine the effect of added structural stiffening on the Bridge Deck vibration, eight heavy vertical stanchions were installed between the Main Deck and the 2nd Deck, between frames 107 and 126. After these structural modifications were completed another vibration survey was conducted between Baltimore and New York. Two series of tests were conducted: a shallow water series during which the depth of water did not exceed 100 feet and a deep water series in which the depth was always greater than 300 feet.

Although a marked decrease in amplitudes at 105 RPM was noted, previously a critical speed for the house structure, amplitudes at top speed remained about the same. Overall house structure behavior was such that amplitudes tended to increase at the higher deck levels. Also, amplitudes tended to be greater at the ship centerline and decrease closer to the deck edge and boundary of the house structure.

Based on the minor effect of the structural reinforcement it was generally concluded that extensive structural modifications would be needed to achieve further improvement. However, this was not considered a practical solution because extensive structural work would be required in fully outfitted spaces. The only remaining solution considered at that time was to design and install a 5-bladed conventional propeller in lieu

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[5] The names of the four Sea Bridge class vessels are: DEFIANCE (ex MORMACSEA), GREAT REPUBLIC (ex MORMACSKY), RED JACKET (ex MORMACSTAR) and YOUNG AMERICA (ex MORMACSUN).

of the 6-bladed propeller that was on the vessel. This work was never completed. Concurrent with the vibration surveys, structural modifications, and special studies being made on the MORMACSEA, following sister ships were also undergoing vibration tests and structural modifications. During the month of June 1969, a vibration survey was made on the MORMACSKY (second ship). Several structural modifications had been made on this vessel and reports from personnel abroad during the sea trials contained numerous subjective impressions. For example one person indicated:

> "the stern area although noisy (indicating propeller cavitation) was remarkably free of vibration, being less than that encountered on the Mariner's at 22,000 SHP. Certain areas in the deck house however present problems. The deck in way of the Master's office, particularly at his desk, shakes in most disagreeable manner. The single person stateroom No. 7 on the Cabin Deck also shakes uncomfortably."

During Mid-September 1969 a vibration survey was conducted on the MORMACSTAR, (third ship) and distinct resonance of the bridge structure was observed at about 105 RPM in the vertical direction. The main difference observed between the MORMACSTAR and the MORMACSEA was the reduction in amplitude at 105 RPM compared with that measured on the first ship. However, the same overall house structure behavior was observed on both ships, namely increasing amplitudes at the higher deck levels and greatest vertical movement at the ship centerline, decreasing as one neared the house exterior boundaries.

Near the end of 1969 serious consideration was being given to the possibility of installing a new highly skewed propeller design on one of the vessels. Work was subsequently undertaken in 1970 to explore the potential of using a highly skewed propeller design to solve the problems experienced on the Sea Bridge vessels. A program was undertaken encompassing the following work: (a) design of a highly skewed propeller for the MORMACSEA, (b) manufacture of a model propeller, (c) open-water characterization tests, (d) self propulsion tests with the skewed propeller and the existing model hull, (e) cavitation tests behind the existing water screen, and (f) construction and testing from a strength standpoint of a two-bladed model propeller using the same material as for the prototype. This work was completed in 1971 and the results documented (11). Essentially the following conclusions were reached based on the extensive model test program:

- Design propulsion characteristics were met.
- Propulsion performance was comparable to the conventional design for the same SHP.
- The highly skewed design vis a vis conventional design delayed cavitation inception by approximately two knots.
- The highly skewed design had less tendency towards cavitation erosion.
- The highly skewed propeller had adequate strength for the proposed application.

At that point in time all that remained to be accomplished was the construction and installation of the prototype propeller and full-scale testing. However, the four ships were subsequently sold to American Export Lines and the highly skewed propeller work ceased at that point until the following year.

<u>Propeller Redesigns and New Test Program - (1972-1976)</u>. In 1972 American Export Lines (AEL), new owners of the vessels expressed interest to the Maritime Administration in moving forward with the highly skewed propeller work. While the vertical vibration of the accommodation space house structure was a discomfort problem to the crew, extensive propeller blade erosion had appeared indicating decreased efficiency and very short service life for the propellers.

By September 1973, plans had reached the point where AEL was authorized to proceed with the performance of the first task of a renewed and somewhat expanded highly skewed propeller research and development project pending completion of a final contract. This first task of the renewed project was to review the earlier 1971 propeller design and ascertain that improvements developed over the now two year time interval would be incorporated into the final highly skewed propeller that would be constructed and installed on one ship of the class. Key elements of the overall full-scale verification test program included the following: (a) measurement of hull and superstructure vibration, (b) speed, power and thrust measurements, (c) cavitation erosion tests, (d) visual and photographic tests of propeller performance and (e) progressive speed trials. The overall objectives of the highly skewed propeller test program were as listed on the following page:

- Verify the benefits of a highly skewed propeller versus a conventional propeller on a high power and high speed vessel.
- Reduce the level of accommodation space vibration on the Sea Bridge class ships.
- Increase propeller service life from 5 years to 10 years.
- Advance the state-of-the-art of propeller design.
- Investigate the effects of highly skewed propeller design on cavitation erosion.

The David Taylor Naval Ship Research and Development Center embarked on a review of the 1971 propeller design. While the original 1971 propeller design had been based on linear skew and no rake, more recent NSRDC work indicated that different skew distributions and also forward rake should be considered. The harmonic content of the Sea Bridge wake was reexamined and estimates made of the alternating thrust, vertical bearing forces and horizontal bearing forces. Since the predominant direction of vibratory motion on the ship needing correction was vertical it was hoped that a propeller designed to produce low vertical bearing forces would improve the overall ship performance. Recognition was given to the fact that the total forces generated by the propeller consist of both "bearing" and "pressure" forces with some uncertainty remaining as to the conditions when either bearing or pressure forces are dominant.

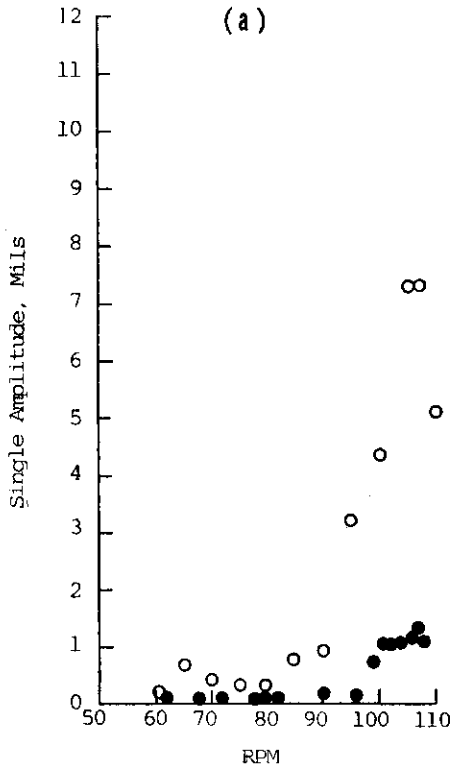
A model propeller conforming to the new design parameters was constructed and tested in August 1974 and when the test results became known it was discovered that full power would be reached at 102.9 RPM in lieu of the original target of 107 RPM. Since the machinery plant of the ships, i.e. gears, shafting, etc. could not tolerate developing the full 30,000 SHP at 102.9 RPM, (a higher torque level) the newly designed model propeller was unfortunately unacceptable.

Three possible alternative courses of action were then considered as follows: (a) construct the full scale propeller based on the current model propeller design but make a pitch correction to increase the RPM, (b) revert back to the earlier 1971 skewed propeller design, and lastly, (c) construct a new model propeller based on the current design (with pitch change only) and perform new model tests to verify the design.

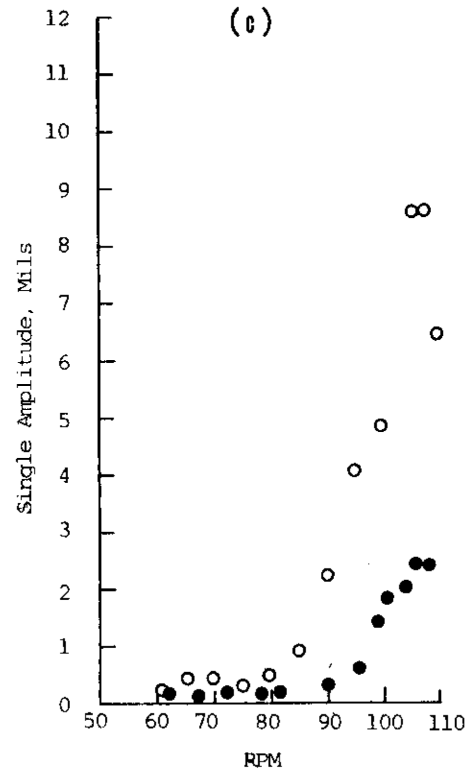
Figure 6 - Sea Bridge Class Highly Skewed Propeller Installation

The first option was judged very risky and also a situation would be created wherein every propeller in the entire project was slightly different (i.e. model propellers different than the full scale propellers) thereby complicating the correlation of model work with full scale work.[6] The second option would revert back to the 1971 skewed propeller design, however, this design posed the greatest risk of full scale blade failure due to higher predicted stresses and also would have added complications to propeller installation and removal. This option would also have adversely affected the propeller blade erosion work then being carried out the most of any of the three options. The third option, that of building another model propeller for more tests would increase costs and pose problems for timing, however this was the option selected. A new model was constructed and tested and this, the third highly skewed model propeller design for the vessels, proved to have

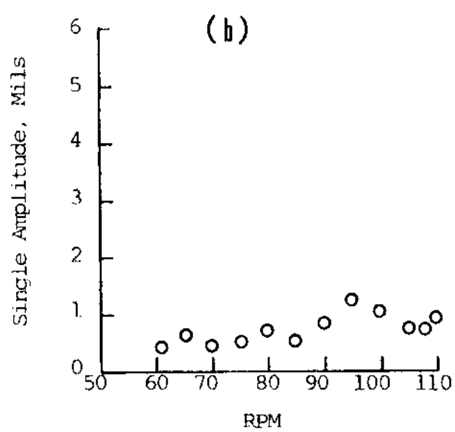
6 Results of the ULTRASEA tests were known at this time and there was doubt about model tests giving accurate RPM predictions.



Vertical Vibration of Elevator
Casing at Upper Deck



Vertical Vibration of Elevator
Casing at Cabin Deck



Athwartship Vibration of Elevator
Casing at Cabin Deck

KEY

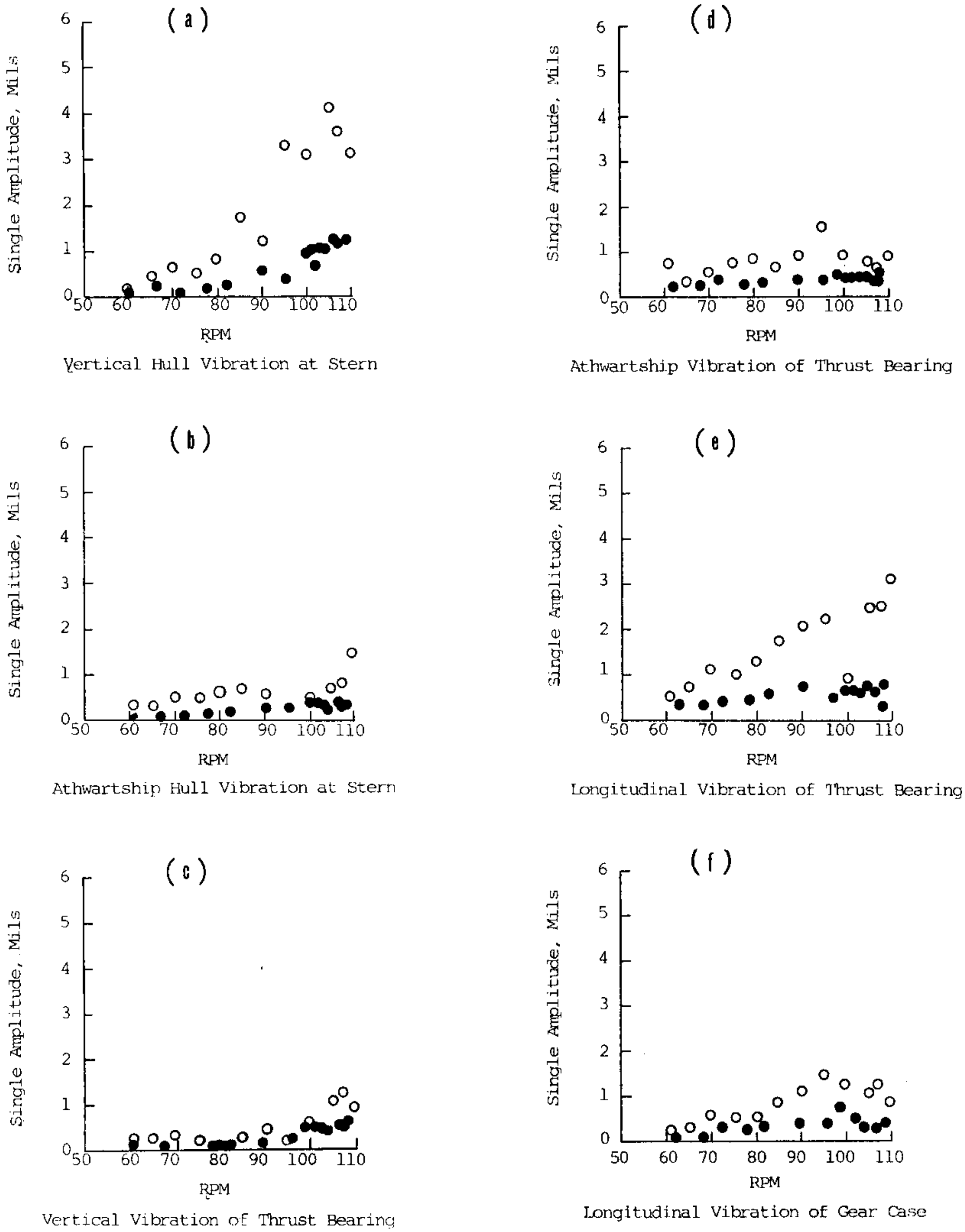
o Blade Frequency Components
Conventional Propeller

● Blade Frequency Components
Skewed Propeller

Figure 7 - Elevator Casing Vibration

satisfactory performance test results (including RPM) and was used as the basis for the prototype.

By now the DEFIANCE (ex MORMACSEA) had been selected as the test ship to receive the new propeller and in July 1974 the baseline hull and superstructure vibration performance measurements were obtained on a coastwise voyage from Norfolk, Virginia to Staten Island, New York. Work proceeded and the full-scale propeller was constructed and installed on the DEFIANCE in Newport News, Virginia in May 1976. Figure 6 shows the highly skewed propeller installation. Although the original schedule called for the full-scale test measurements to be made on a voyage from Norfolk, Virginia to Staten Island, New York, the ship was rerouted to first make a stop in Baltimore, Maryland due to cargo commitments. During the short voyage up the Chesapeake Bay



KEY

○ Blade Frequency Components
Conventional Propeller

● Blade Frequency Components
Skewed Propeller

Figure 8 - Hull, Thrust Bearing and Gear Case Vibration

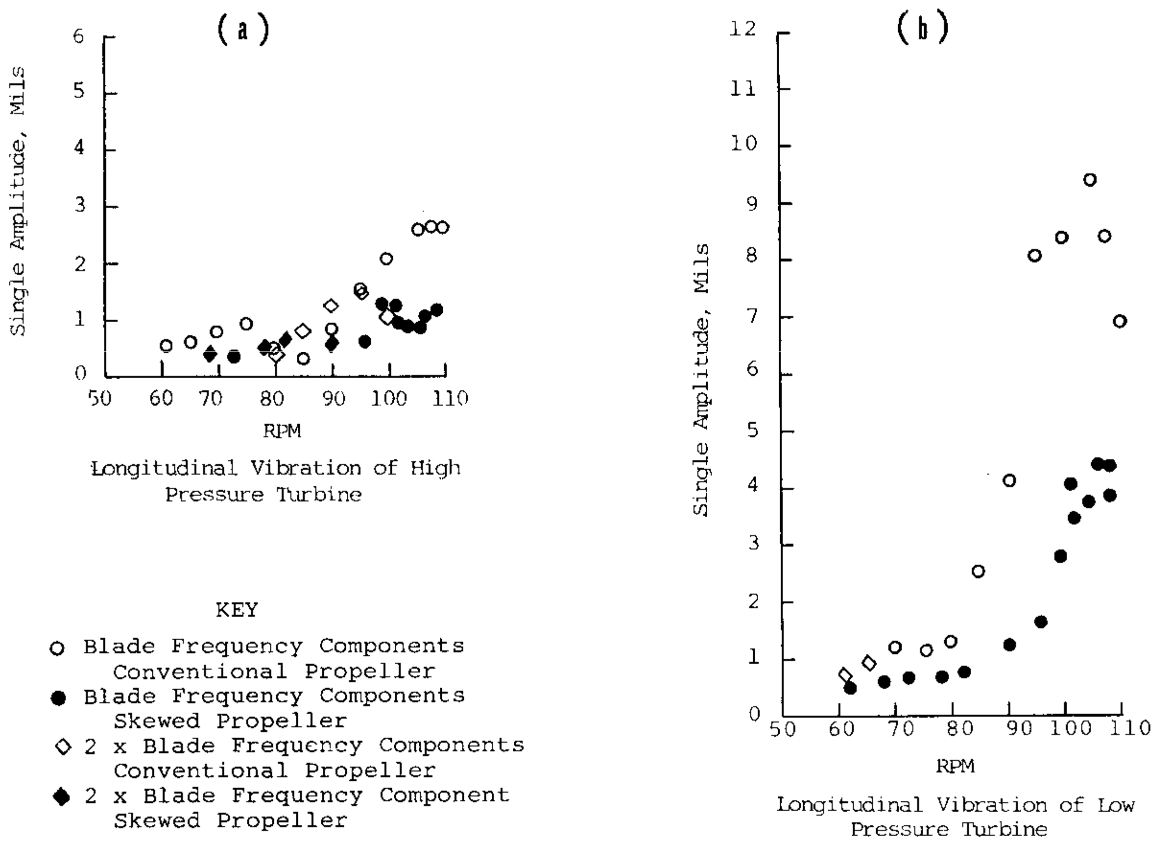


Figure 9 - High Pressure and Low Pressure Turbine Vibration

while operating at full power in shallow water the ship crew was very aware of the reduced vibration levels. While no test measurements had been made during the short voyage, the new propeller had already received crew acceptance by the time the vessel reached Baltimore. In fact comments such as: "this was the first trip I have slept soundly coming up the bay", were heard from numerous crewmembers. On June 4, 1976, full scale vibration measurements were made on the DEFIANCE confirming the remarkable improvement of the accommodation house structure. Finally after seven years from the builders trial in 1969, the overall discomfort problem had been eliminated.

Vibration Measurements and Results. If one examines Figure 5 it will be noted that the vessel arrangement features a continuous fore and aft RO/RO deck for vehicle traffic on the Second Deck level. As has been discussed vertical vibration of the accommodation house was the primary vibration problem encountered on the Sea Bridge class vessels needing correction. Although not apparent from the arrangement plans, structurally there is little continuous vertical support throughout the center core of the accommodation house with the vast majority of the interior bulkheads being of the joiner nonsupport type. The machinery casing, while extending from the machinery space to the Bridge Deck level, is divided into port and starboard sections located outboard of the vehicle passageway at the Second Deck level and does not merge into one open casing until the Upper Deck level is reached. Thus, the rigid vertical and athwartship support normally provided by machinery casing boundaries on conventional ship designs was not possible on the Sea Bridge class vessels. In fact, the elevator shaft boundary bulkheads are the only structural bulkheads in perfect alignment extending from the machinery space up to the Bridge Deck level. Because of unique aspect of the elevator casing bulkheads, two locations were selected as the reference points for measuring "before" and "after" house vibration.

Figure 7 displays the results of vibration measurements obtained on selected points on the elevator casing bulkheads and inspection of Figure 7c reveals that maximum amplitudes of approximately 8.5 mils were reached in the vertical direction at the Cabin Deck level at speeds corresponding to 107/108 RPM. Athwartship measurements at the same location and speed, Figure 7b, indicates that vibration levels were approximately 1.0 mils or less. Vertical vibration of the elevator casing at the Upper Deck level as shown on Figure 7a, was 7.5 mils, somewhat less than that measured two decks higher. Superimposed on Figures 7a and 7c are the results of measurements made on the DEFIANCE after the highly skewed propeller had been installed on the vessel and inspection of these figures shows the remarkable improvement achieved. Vertical vibration at the Cabin Deck once 8.5 mils has been reduced to 2.5 mils for a 66 percent reduction. Vertical vibration at the Upper Deck level initially 7.5 mils being reduced to 1.5 mils for a 80 percent improvement.

Figures 8a through 9b, display the results of "before" and "after" measurements at the ship stern, thrust bearing foundation, gear case, high pressure turbine, and low pressure turbine, respectively. These locations conform to the standard locations outlined in SNAME Code C-1. Inspection of Figures 8a and 8b pertaining to hull vibration reveals substantially reduced vibration levels after the highly skewed propeller had been installed. Vertical hull vibration at the stern initially 4.0 mils at 105 PRM has been reduced to 1.5 mils or less, a 62 percent improvement. Clearly a 50 percent reduction or greater has been achieved at other shaft speeds.

With regard to the other standard locations, namely, thrust bearing foundation, gear case, HP and LP turbines, Figures 8c through 9b also reveal the substantial improvement achieved. In fact at every measured location the highly skewed propeller has reduced the vibratory movement and if one searches for general trends it can be observed that the greatest improvement consistently is measured at shaft speeds corresponding to localized resonant frequencies. This appears to be an important characteristic of highly skewed propeller performance and the reader is encouraged to examine the data and form independent conclusions from the measured results.

II. San Clemente Class

Background. While substantial work had been completed exploring the potential benefits of installing a highly skewed propeller on the Sea Bridge vessels by 1971, the highly skewed propeller work came to a halt until a shipowner was found that would be willing to install one on his ship. Prospective shipowners were therefore contacted regarding their willingness. to consider installation of a highly skewed propeller.

Aries Marine Shipping Company who had signed contracts in June 1971 with National Steel and Shipbuilding Company (NASSCO) to construct two 80,500 DWT San Clemente Ore/Bulk/Oil (OBO) carriers, indicated a willingness to try the highly skewed propeller concept. In April 1973 the Maritime Administration awarded an R&D contract to install the first highly skewed propeller on one of these 24,000 SHP merchant vessels. Arrangements of the San Clemente class vessel are shown on Figure 10.

In contrast to the situation on the Sea Bridge class vessels here was a new vessel design not yet constructed with no known or predicted vibration problems. Rather this was the first shipowner willing to accept the risks involved with the development of an old but yet new and highly innovative propeller design.

The basic model test program for the San Clemente OBO was similar to that carried out for the first Sea Bridge highly skewed propeller design, namely: (a) a wake survey was conducted at full load displacement, (b) a highly skewed propeller designed, (c) a model propeller

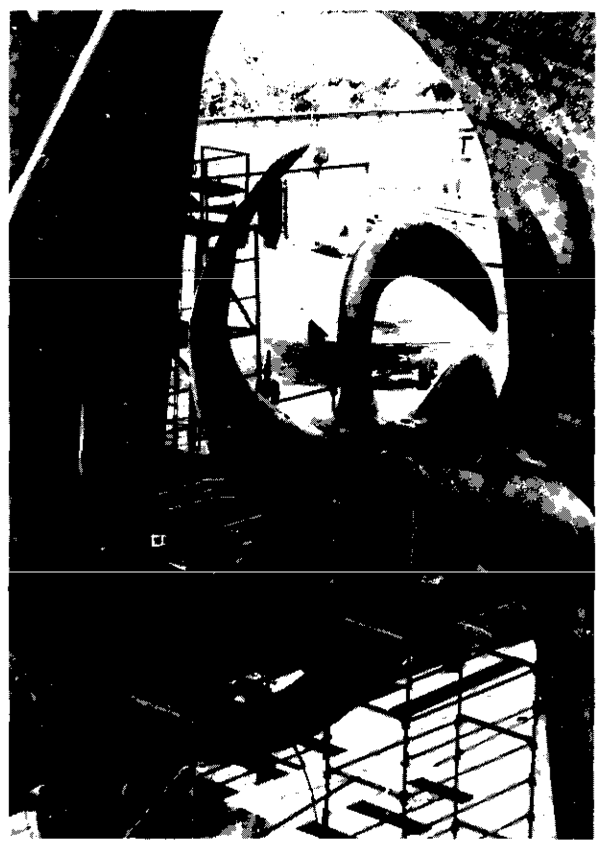


Figure 11 - San Clemente Class Highly Skewed Propeller Installation

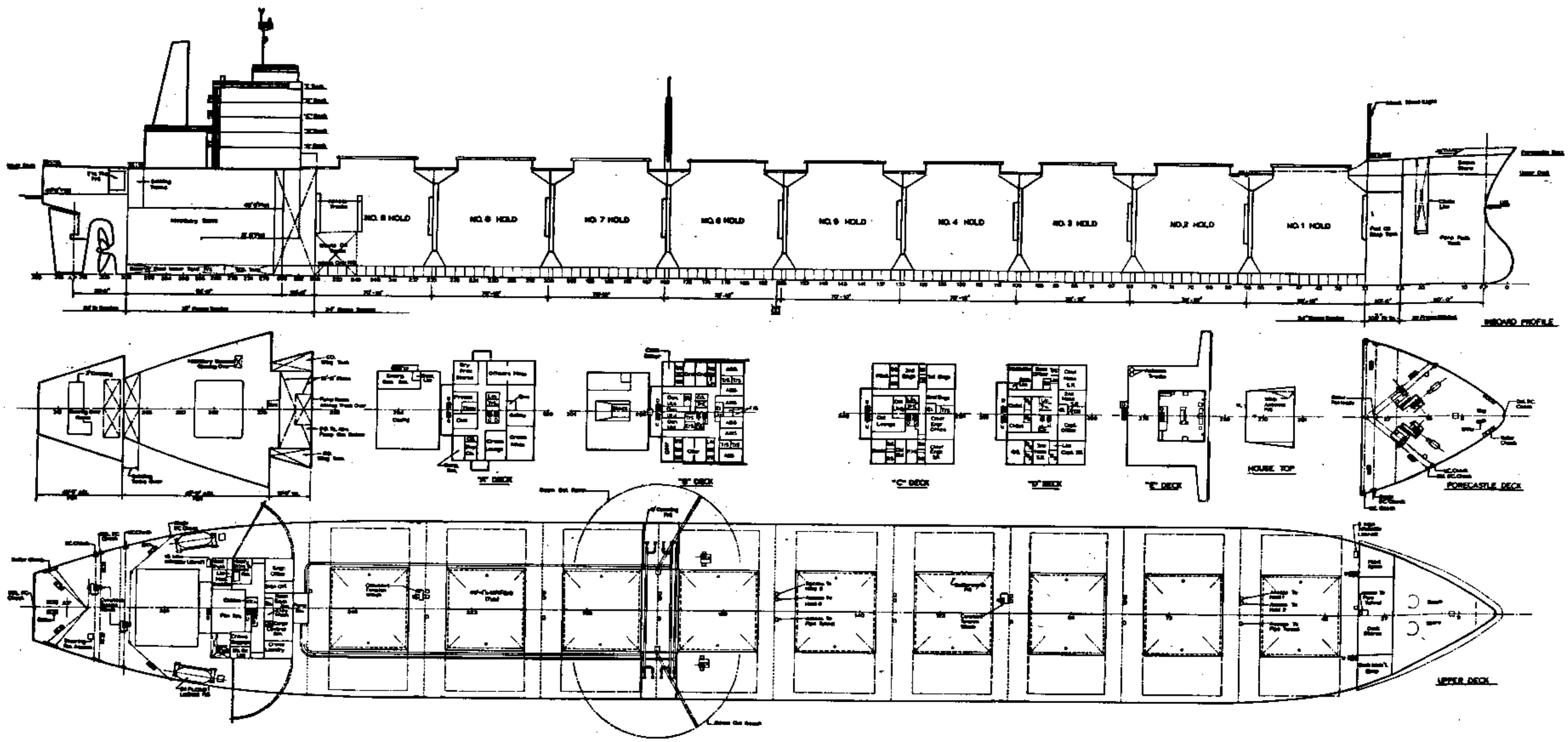


Figure 10 - General Arrangements, San Clemente Class

manufactured, and (d) open-water experiments and resistance and propulsion experiments conducted. Some additional model testing work was performed involving measurements of the propeller induced pressures on the model hull surfaces above the propeller location at the full load displacements for both the conventional and highly skewed propellers.

Overall the model test work progressed without any special problems being encountered. The propeller was designed for the best fit in the aperture and such that it could be removed without pulling the tail shaft or removing the rudder. The highly skewed wheel was also designed to have the same number of blades and tip clearances as the conventional propeller.

Baseline vibration and performance measurements were made in July 1973, on the ULTRAMAR (first ship) fitted with the conventional propeller. Although no special vibration problems were encountered, the vessel was found to develop the full 24,000 shaft horsepower at 88 RPM, some 4 RPM below the design value of 92 RPM. The ULTRAMAR was subsequently delivered and limited to operating at speeds no greater than 80 RPM until 6 inches could be removed from the propeller tips.

Approximately eight months later the ULTRASEA (second ship) fitted with the highly skewed propeller was taken to sea and another set of full scale performance measurements were taken. Figure 11 shows the highly skewed propeller installation. On this vessel it was found that the full 24,000 shaft horsepower was developed at 98 RPM, or 6 RPM greater than the target of 92 RPM. Both conventional and highly skewed propellers had missed the design RPM by a substantial amount.

While the historical events leading up to the installation of a highly skewed propeller on the DEFIANCE were complex and of special interest, the interesting aspects of the San Clemente class propeller project were about to begin. On March 5, 1974, the ULTRASEA was off the coast of San Diego, California and almost finished with the last remaining sea trial test items involving forward and astern crash test maneuvers. After a successful crash ahead had been completed and full ahead speed reached, a crash astern command was given. The engines responded, rapidly coming to a halt and started to build up astern power. As the seconds and minutes passed both boilers were noted to be operating in the overload condition. A shaft rate impulse and sound was detected and leaning over the stern rail, one could view an erratic bubble-filled wake. Propeller shaft torque readings fluctuated and as the test continued it became apparent that one propeller blade had failed. No one aboard was certain of exactly what had happened. The sea trial was terminated and the vessel proceeded back to the shipyard.

Upon deballasting and reaching the shipyard the propeller shaft was slowly rotated by the jacking gear and it became clear that one blade had been substantially bent aft after striking a hard metallic object. The very tip of the damaged blade was curled aft for about a 3"x3" area and Figure 12 shows a closeup view of the tip damage.

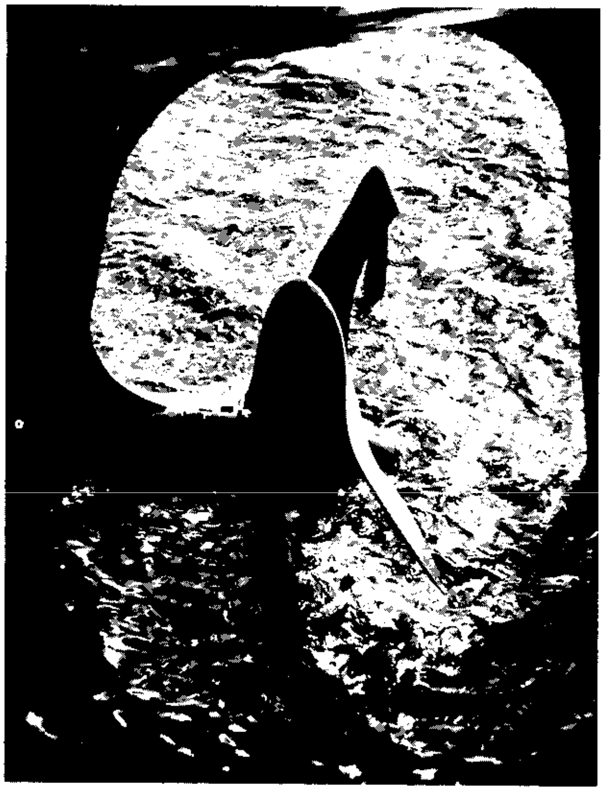
Figure 12 - Tip of Damaged Propeller Blade

Subsequent examination later showed the entire blade set aft and distorted with the bend located about 4 to 5 feet from the tip.

Figure 13 indicates two views of the extent of the damage.[7] The damaged propeller blade was subsequently heated and faired in place and returned to the design pitch and rake, following by stress relieving. Figures 14 and 15 respectively, show the damaged propeller blade partially refaired and completely repaired.

The ULTRASEA was subsequently taken out to sea again to complete the remaining test items and ultimately delivered to the owners at the end of March 1974. Thereafter the ULTRASEA was engaged in worldwide service and approximately six months later while in

[7] Numerous submarines were operating on the surface in the near vicinity of the ship and some months later it was learned that the propeller damage may have been caused by striking a submerged submarine.



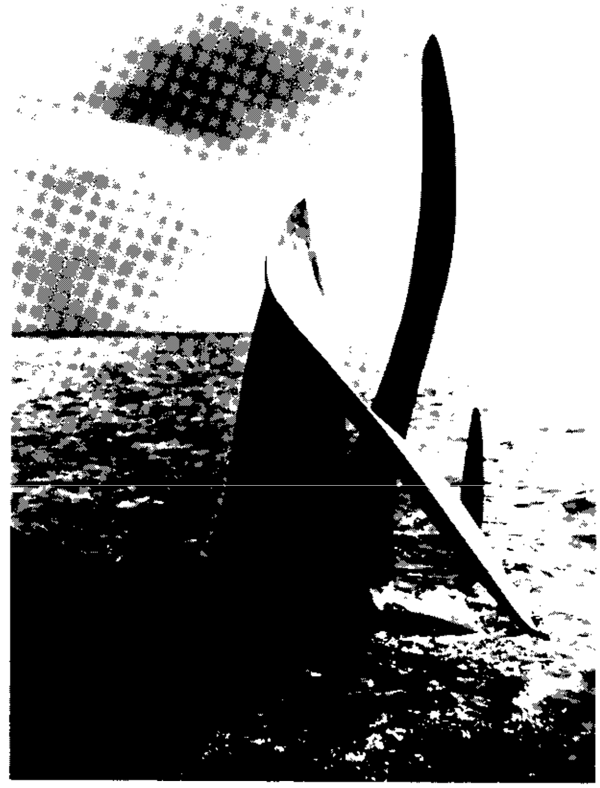


Figure 13 - Overview of Damaged Propeller

port it was observed that approximately 4 feet was missing from one propeller blade.[8] A photograph of the broken propeller is shown in Figure 16. The missing tip was discovered by a mate reading vessel draft marks and the exact date of the failure is not known.

In November 1974 the highly skewed propeller was removed from the ULTRASEA and replaced with a conventional propeller. The damaged highly skewed propeller was then returned to the manufacturer and a new tip section installed.

Figure 14 - Partially Refaired Propeller Blade

After completion of repairs the propeller was then installed in October 1975 on the ULTRAMAR, the sister ship to the ULTRASEA, and the propeller again returned to service. Sometime in October 1977 some two years later the ULTRAMAR struck a submerged object while docking in the port of Corpus Christi, Texas. The vessel departed for New Orleans and while underway at low RPM an unusual noise and vibration was noticed. During the voyage the vessel stopped due to a machinery problem and in the maneuver involving reversing of the engines from 60 RPM ahead to approximately 20 RPM astern the propeller nut backed off and the propeller was lost at sea where it still remains.

Vibration Measurement Results. Baseline hull girder and machinery vibration measurements made on the ULTRAMAR (first ship) fitted with the conventional propeller are shown on Figures 17a through 18b. Limited superstructure measurements made on the "E" deck of this vessel are shown on Figure 19. Superimposed on Figures 17a through 19c are the results of repeat measurements made on the ULTRASEA fitted with the highly skewed propeller.

[8] The propeller blade broke approximately at the knuckle shown in Figure 16.

Although these vessels encountered no vibration problems, examination of Figures 17f through 18d and Figure 18c for the LP turbine, reveals that the main machinery components of the ULTRAMAR were approaching a longitudinal resonant condition. When the highly skewed propeller was tested on the ULTRASEA one can observe that the vessel passed through the critical speed at approximately 96 RPM and that the vibration levels dimenished thereafter.

In the case of the San Clemente OBO vessels even though all tests have been completed there still remains uncertainty regarding a determination of the amount of improvement due to the highly skewed propeller. The reason for this assessment is that the full-scale highly skewed propeller operated at some 6 RPM higher than the design value, while the full-scale conventional propeller operated some 4 RPM on the low side resulting in a 10 RPM difference between the two propellers. Essentially this means if one attempts to make a comparison at constant RPM

Figure 15 - Propeller After Completion of Repairs

Figure 16 - Broken Propeller Blade

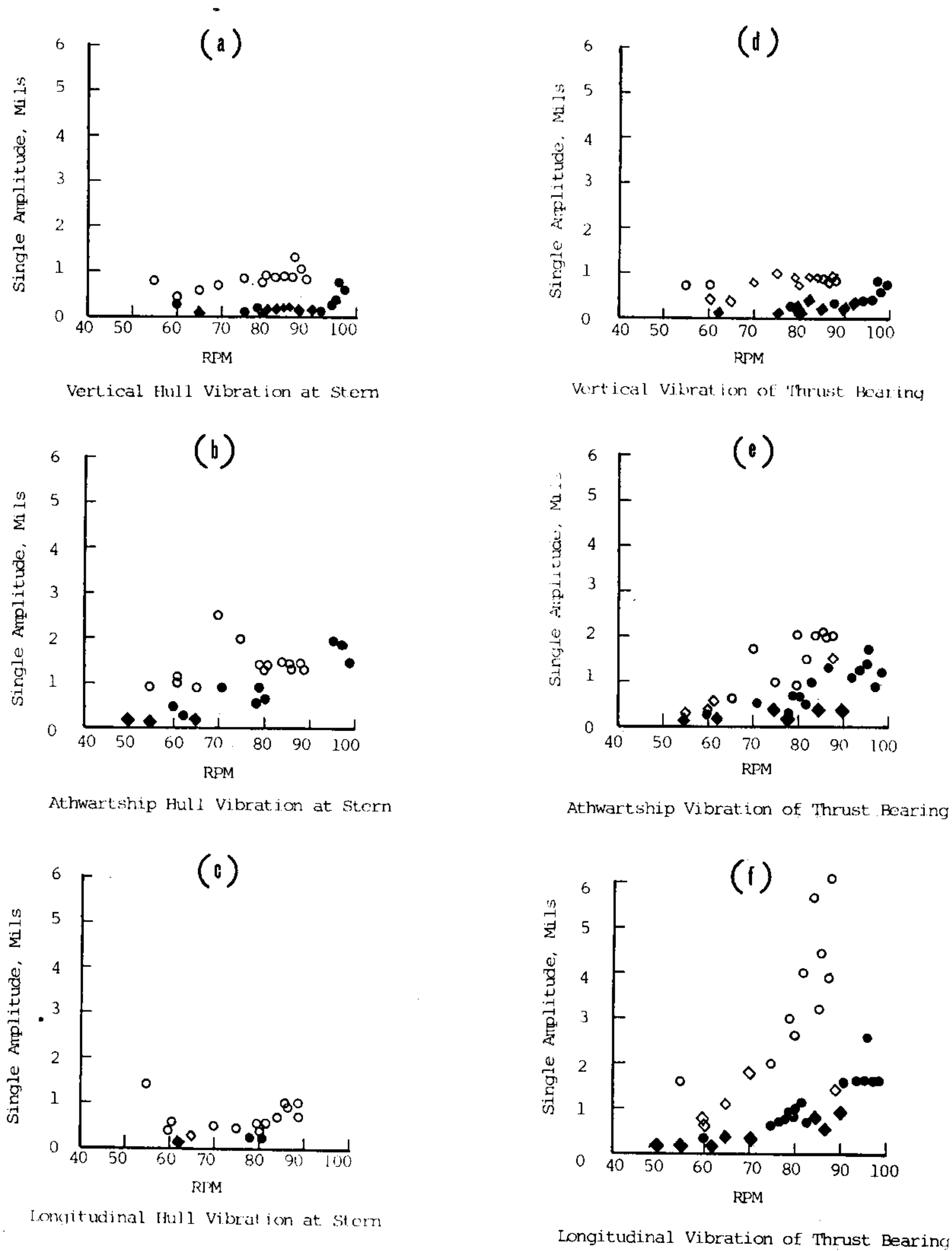
values, propeller induced power levels will be substantially higher for the conventional propeller than the skewed propeller thus precluding a true comparison. On the other hand if one attempts to make a comparison on the basis of equal shaft horsepower, the propeller RPM (and blade frequency) for the highly skewed propeller will be greater and inasmuch as some ship locations and components may be highly frequency dependent, this also voids a true comparison.

While there cannot be a positive determination of the degree of improvement made by the highly skewed propeller, the authors' are of the opinion that vibration levels were reduced by one-half overall. It should be noted that in every location where measurements were taken, the highly skewed propeller always resulted in reduced levels of vibration regardless of RPM and direction of vibratory motion. In some instances, where the ship was operating near to, or at critical frequencies, reductions were one-fourth to one-sixth of the original value. Again it should be observed that the highly skewed propeller shows greatest improvement vis a vis the conventional propeller at resonant conditions of vibration. The reader is invited to examine the test results and form independent conclusions.

III. Maine Class

Overview/Background. In contrast to the rather complicated history of events associated with the Sea Bridge and San Clemente class highly skewed propeller projects, the design and testing of the highly skewed propellers for the Maine class vessels as well as service experience has been very routine in nature.

After construction contracts had been awarded for the subject ships and about midway through development of the detailed engineering phase concern was expressed about the rigidity of the aft



Vertical Hull Vibration at Stern

Athwartship Hull Vibration at Stern

Longitudinal Hull Vibration at Stern

Vertical Vibration of Thrust Bearing

Athwartship Vibration of Thrust Bearing

Longitudinal Vibration of Thrust Bearing

KEY

○ Blade Frequency Components Conventional Propeller
● Blade Frequency Components Skewed Propeller
◇ 2 x Blade Frequency Components Conventional Propeller
◆ 2 x Blade Frequency Components Skewed Propeller

Figure 17 – Hull and Thrust Bearing Vibration

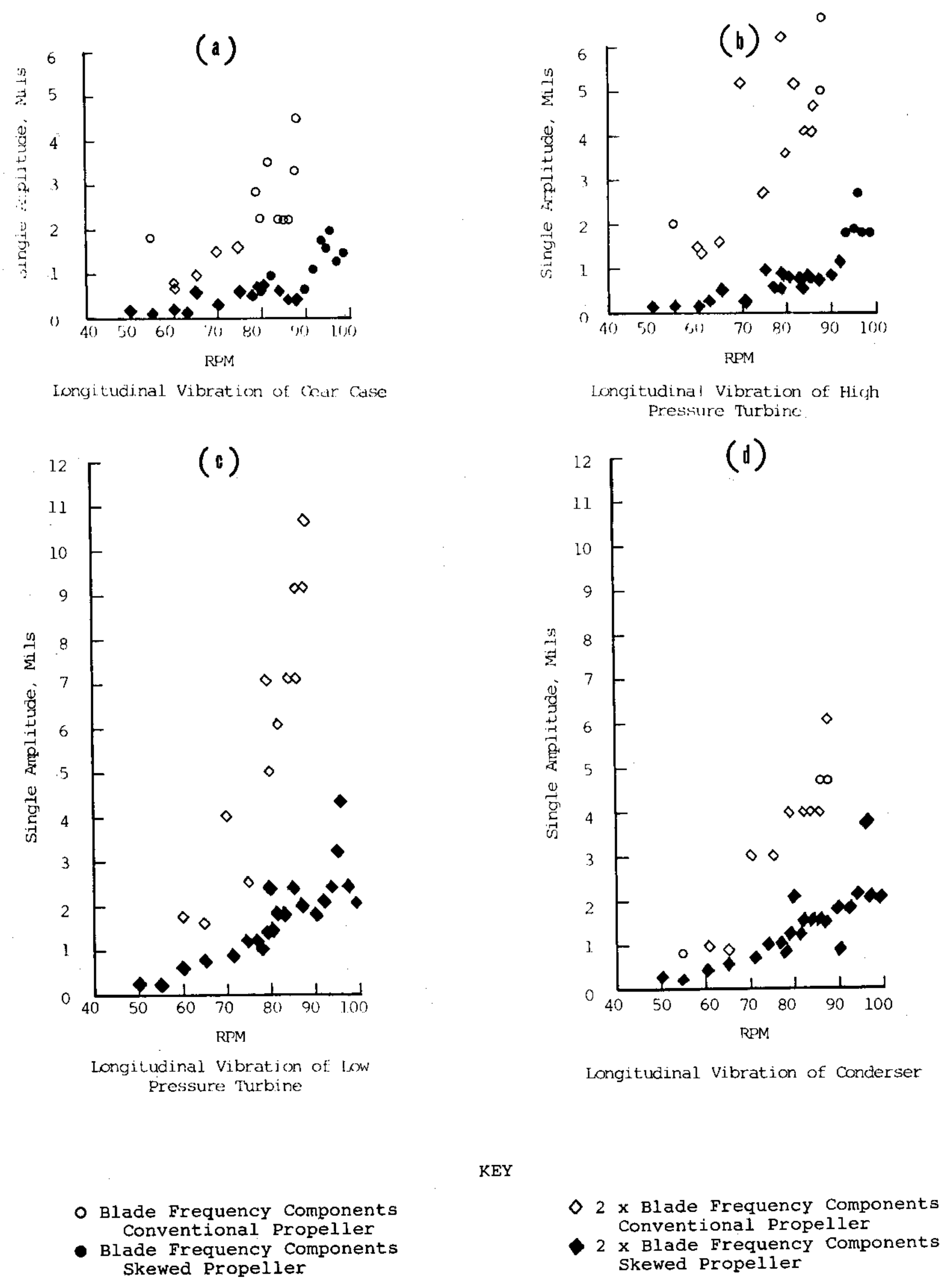


Figure 18 - Gear Case, High Pressure Turbine, Low Pressure Turbine, and Condenser Vibration

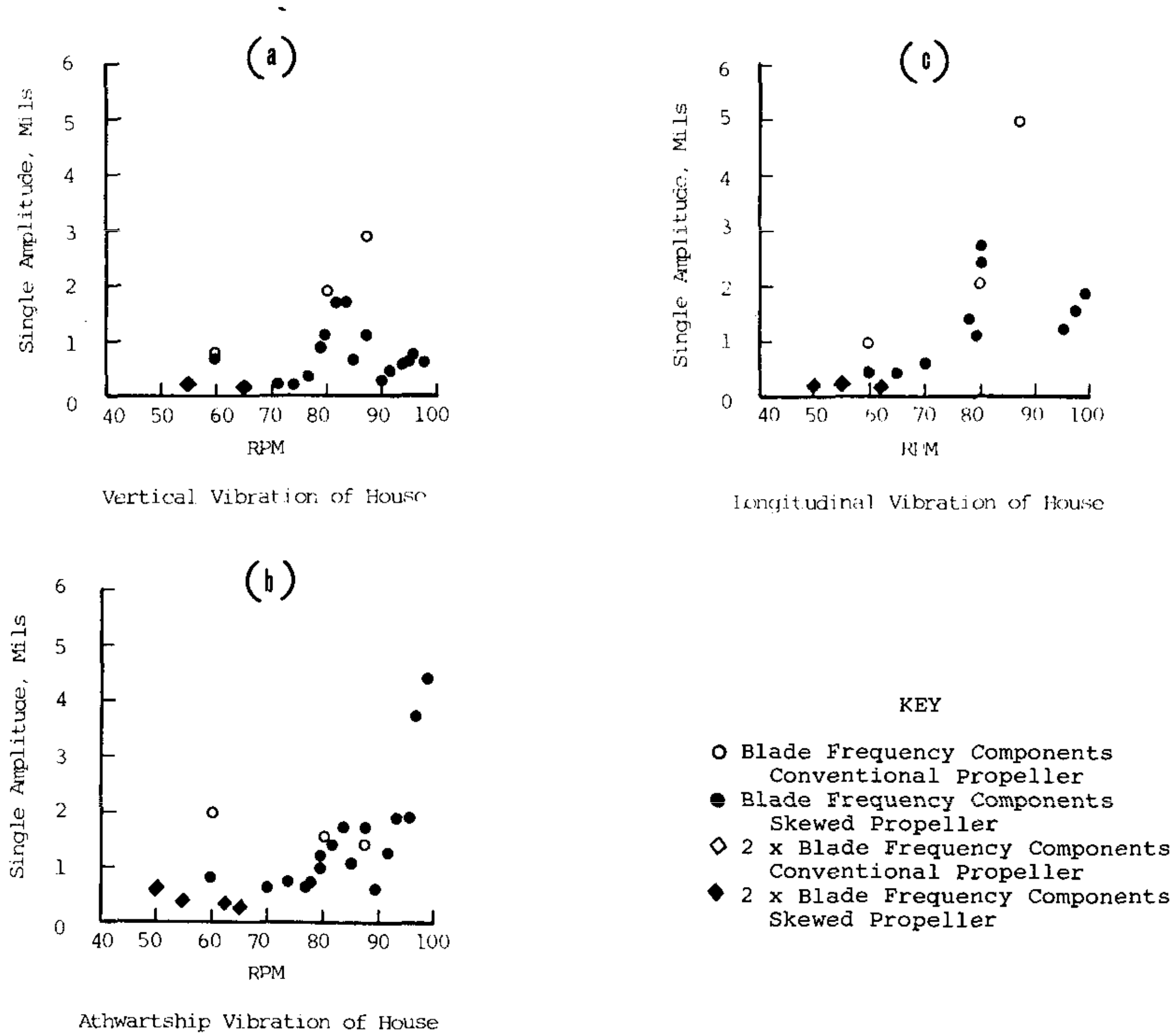


Figure 19 - House Vibration

peak stern structure because of the very fine after body hull lines.

Extensive vibration analyses were undertaken and the results of this work indicated the strong possibility that excessive lateral vibration would be encountered in the stern area. Estimates were made of the performance of both 5-bladed and 6-bladed propellers with the decision made to move forward with a 6-bladed conventional design propeller. Overall arrangements of this 37,000 shaft horsepower RO/RO vessel are shown on Figure 20.

A proposal was later made to consider the use of highly skewed propellers on the ships to reduce the excitation forces in the stern area and subsequently a change under contract was issued for the design, test and manufacture of four DTNSRDC designed highly skewed propellers. Because of lead times, launch dates, etc. the plan was that in the event the conventional propeller proved satisfactory on the trials of the first ship, that propeller would be retained aboard as the service propeller. Three of the highly skewed propellers would, however, be installed on the following three ships, with the fourth highly skewed propeller becoming the spare.

Propeller model testing work for this design took place in the Summer of 1974, approximately the same time as the Sea Bridge model work was taking place. While the propeller design and construction of a model propeller were carried out at DTNSRDC, the actual model test program work was carried out at the Netherlands Ship Model Basin. It should be noted that construction of the model propeller was completed about the same time as the model of the second Sea Bridge highly skewed propeller

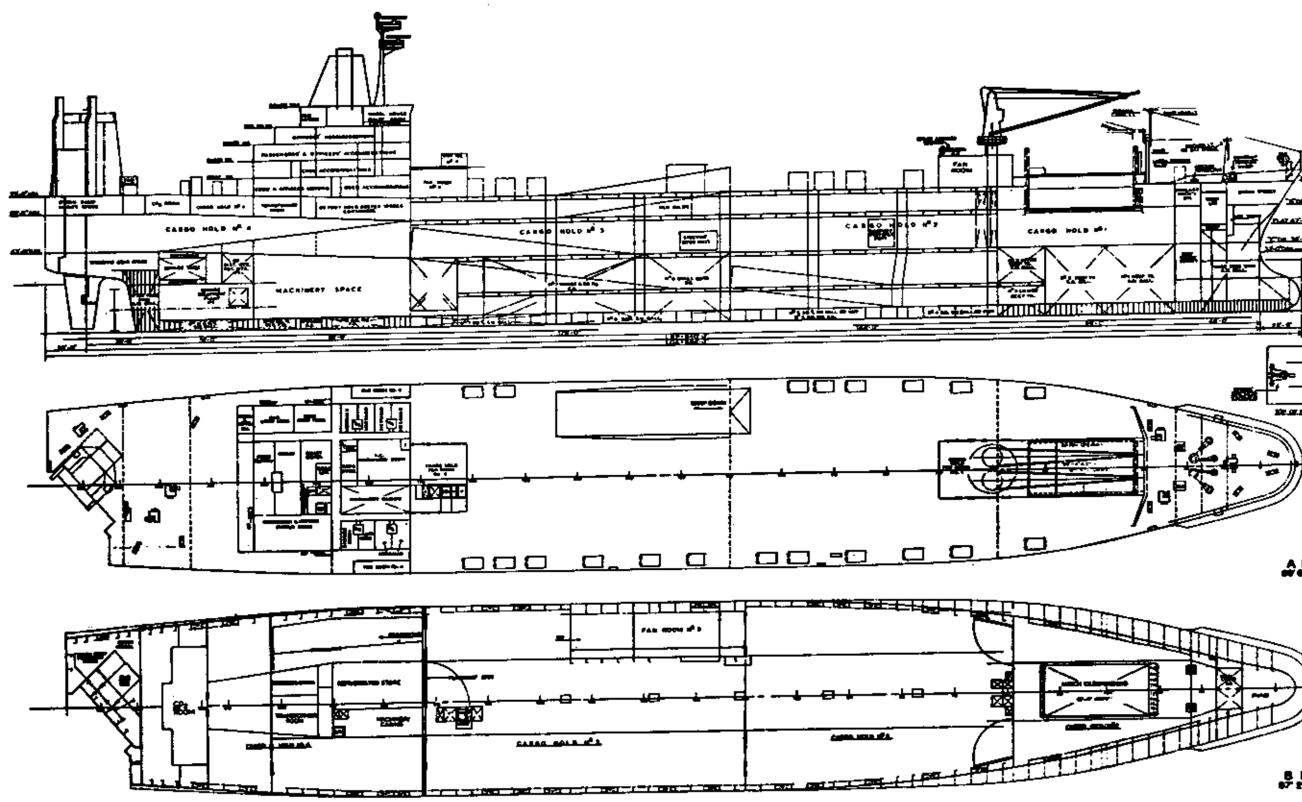


Figure 20 - General Arrangements, Maine Class

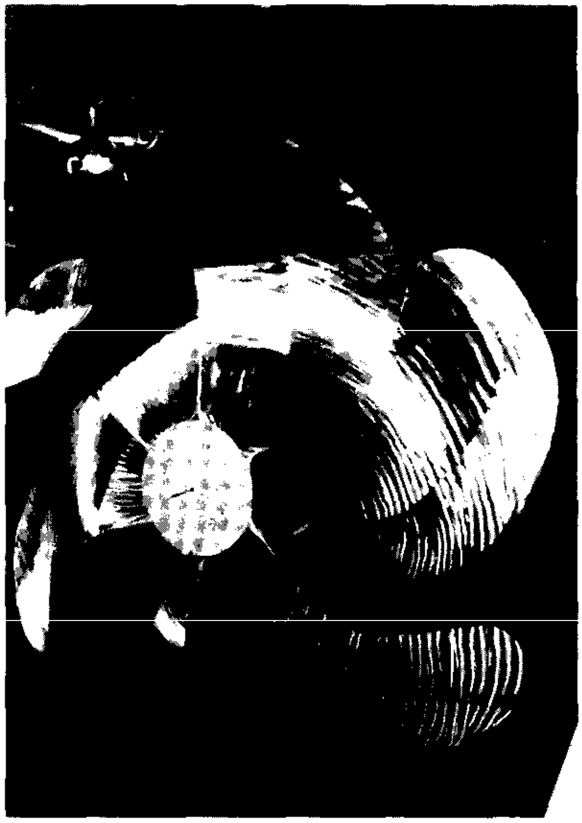


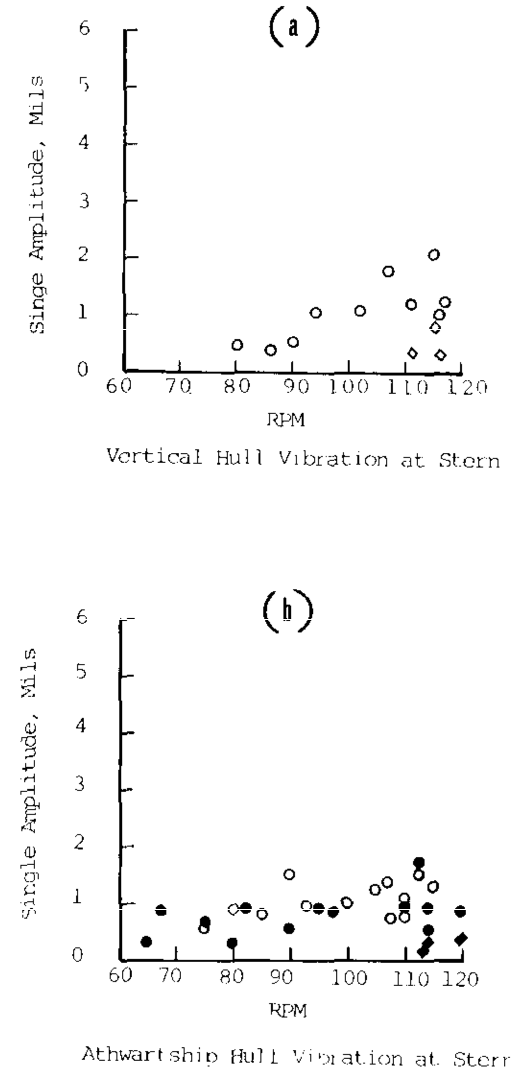
Figure 21 - Maine Class
Highly Skewed Propeller

design and in fact both propeller models were in the model shop at the same time. Differences between the two designs were most striking. A photograph of the Maine class highly skewed propeller is shown in Figure 21 and the reader should compare Figure 21 with Figure 6 from the DEFIANCE.

Due to the vessel design, propeller weights and other factors, considerable attention had to be given to shaft alignment on the Maine class vessels. While severe stern tube bearing problems were encountered on the MAINE (first ship) fitted with the conventional propeller, these bearing problems are beyond the scope of this paper addressing the effectiveness of highly skewed propellers and will not be discussed.

<u>Vibration Measurement Results.</u> Results from the baseline vibration measurements obtained on the MAINE are shown on Figures 22a through 24e. Superimposed on these figures are the results of measurements obtained on the NEVADA (second ship) fitted with the highly skewed propeller.

Since the amount of skew on the Maine class highly skewed propellers, is much less than that for the Sea Bridge and San Clemente projects, one would expect that the amount of improvement on the NEVADA vibration levels would be less than that measured on the other two projects. The vibration measurements at some locations such as the hull at stern, Figure 22b, and the house structure, Figure 24e, show only a modest improvements. Surprising, however longitudinal vibration measurements on main machinery components such as the high pressure turbine, low pressure turbine and condenser show significant improvements in the order of



KEY

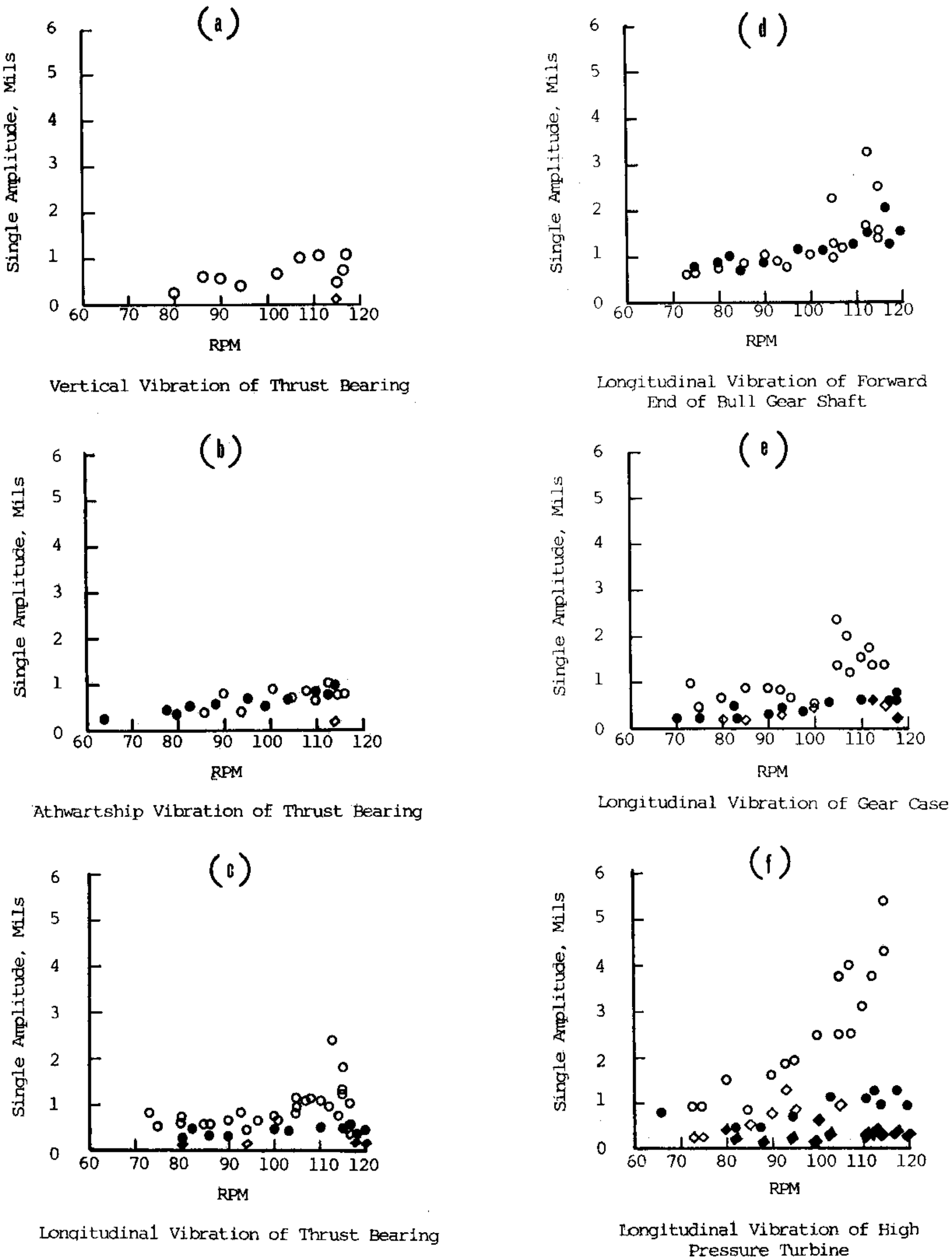
○ Blade Frequency Components
Conventional Propeller

● Blade Frequency Components
Skewed Propeller

◇ 2 x Blade Frequency Components
Conventional Propeller

◆ 2 x Blade Frequency Components
Skewed Propeller

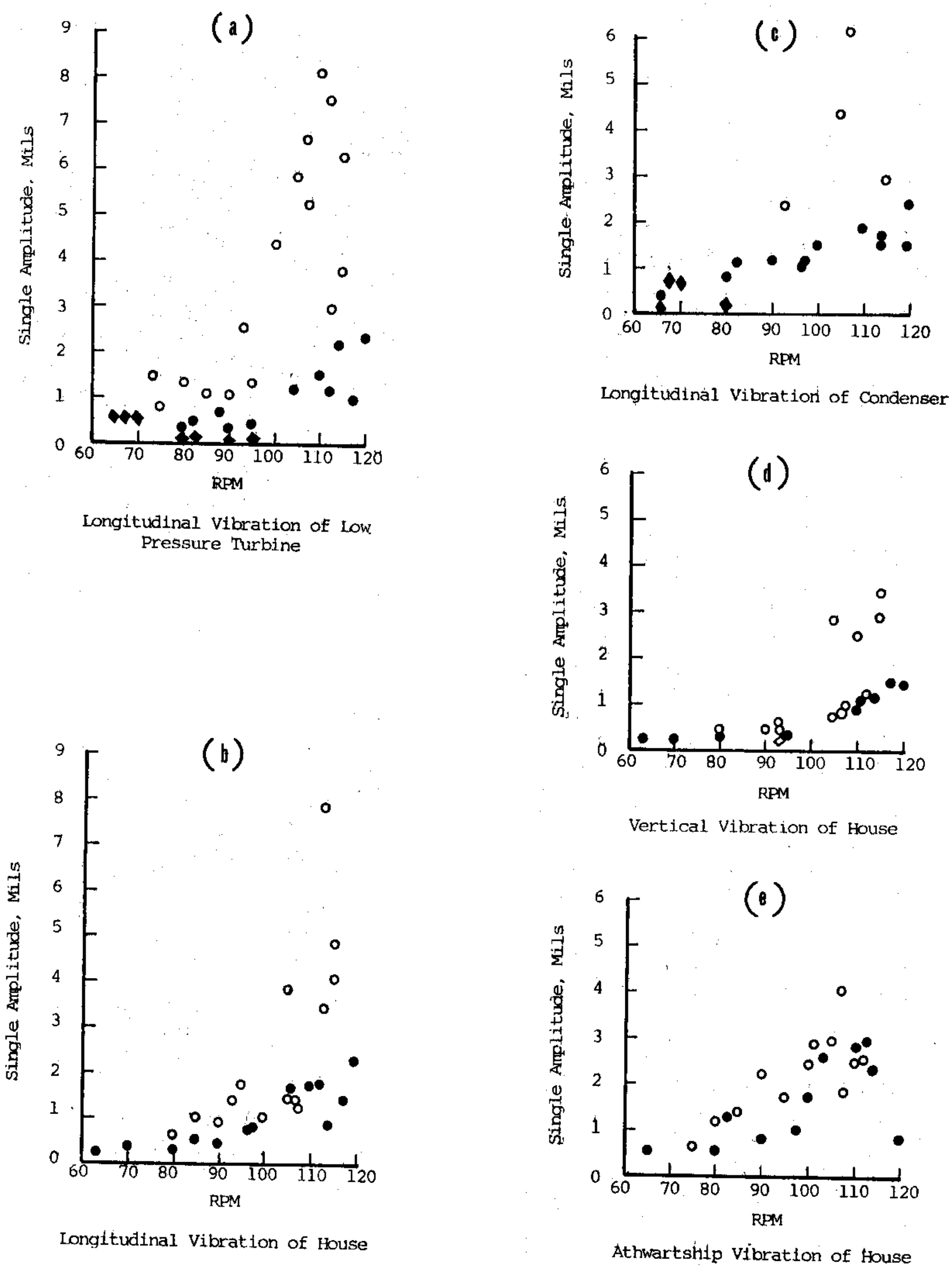
Figure 22 - Hull Vibration



KEY

- ○ Blade Frequency Components Conventional Propeller
- ● Blade Frequency Components Skewed Propeller
- ◇ 2 x Blade Frequency Components Conventional Propeller
- ◆ 2 x Blade Frequency Components Skewed Propeller

Figure 23 - Thrust Bearing, Bull Gear, Gear Case and High Pressure Turbine Vibration



KEY

- o Blade Frequency Components Conventional Propeller
- ● Blade Frequency Components Skewed Propeller
- ◇ 2 x Blade Frequency Components Conventional Propeller
- ◆ 2 x Blade Frequency Components Skewed Propeller

Figure 24 - Low Pressure Turbine, Condenser and House Vibration

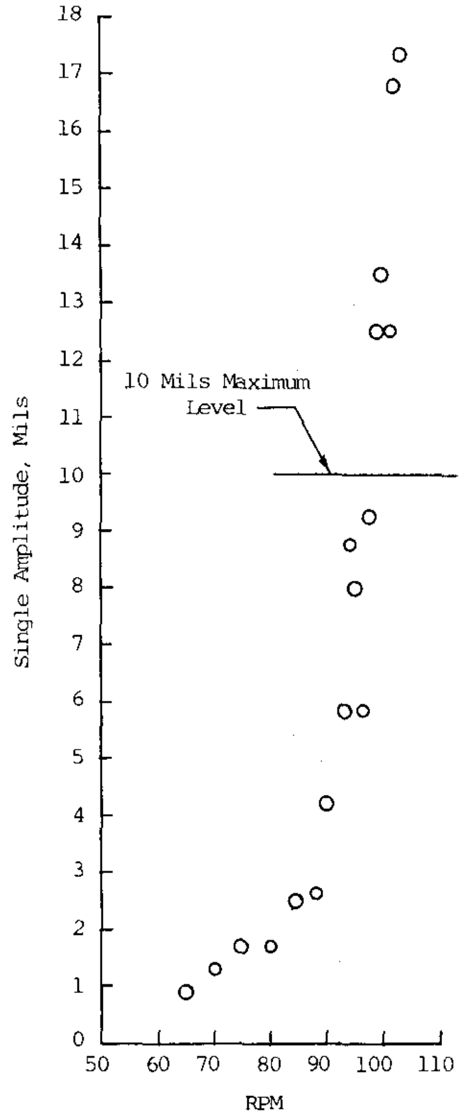
60 percent or greater. For example, examination of Figures 23f through 24c, reveals that movement of the high-pressure turbine was reduced from 5.5 mils to 1.5 mils for a 63 percent reduction, movement of the low-pressure turbine was reduced from a peak value of 8.0 mils to 2.5 mils for a 70 percent reduction, and movement of the condenser reduced from 6.5 mils to 2.5 mils for a 62 percent reduction.

Improvement in vibration levels on the structural measurement points tended to range from virtually no improvement in the athwartship vibration at the stern, Figure 22b, to only a 25 percent reduction in the athwartship vibration of the house as shown in Figure 24e. Considering that the Maine class propellers were designed to reduce the lateral movement of the after peak stern structure, review of the measurements could lead one to conclude that the highly skewed propeller failed to meet its objective. Conversely, one could also say that the highly skewed propeller performed exactly as expected since vibration levels of 2.0 mils and less are very small indeed and that the conventional propeller performed much better than any of the detailed vibration analyses predicted. Regardless of which view point the reader may take the evidence is clear that the Maine class highly skewed propeller from a vibration standpoint performed much better than the conventional propeller. The characteristic of highly skewed propellers to show greatest improvement at resonant conditions of vibration was again demonstrated.

STRUCTURAL REINFORCEMENT VIS A VIS HIGHLY SKEWED PROPELLER

The purpose of this section is to address some of the factors involved in situations where it is found necessary to correct excessive vibration by adding structural reinforcement. Clearly it is impossible to address all types of problems, e.g. excessive hull girder, local vibration of large substructures, machinery vibration, etc. and possible cures. To a limited degree the authors' hope that the full-scale performance data and discussion provided thus far on the three highly skewed propeller projects can give the reader some insight into hull girder, house structure and machinery behavior relative to this matter.

For illustrative purposes, one actual example of a severe longitudinal machinery vibration will be presented wherein a problem was corrected by structural reinforcement. "Before" and "after" vibration measurements will be presented with information on the extent of steel reinforcement added to correct the vibration. Following this

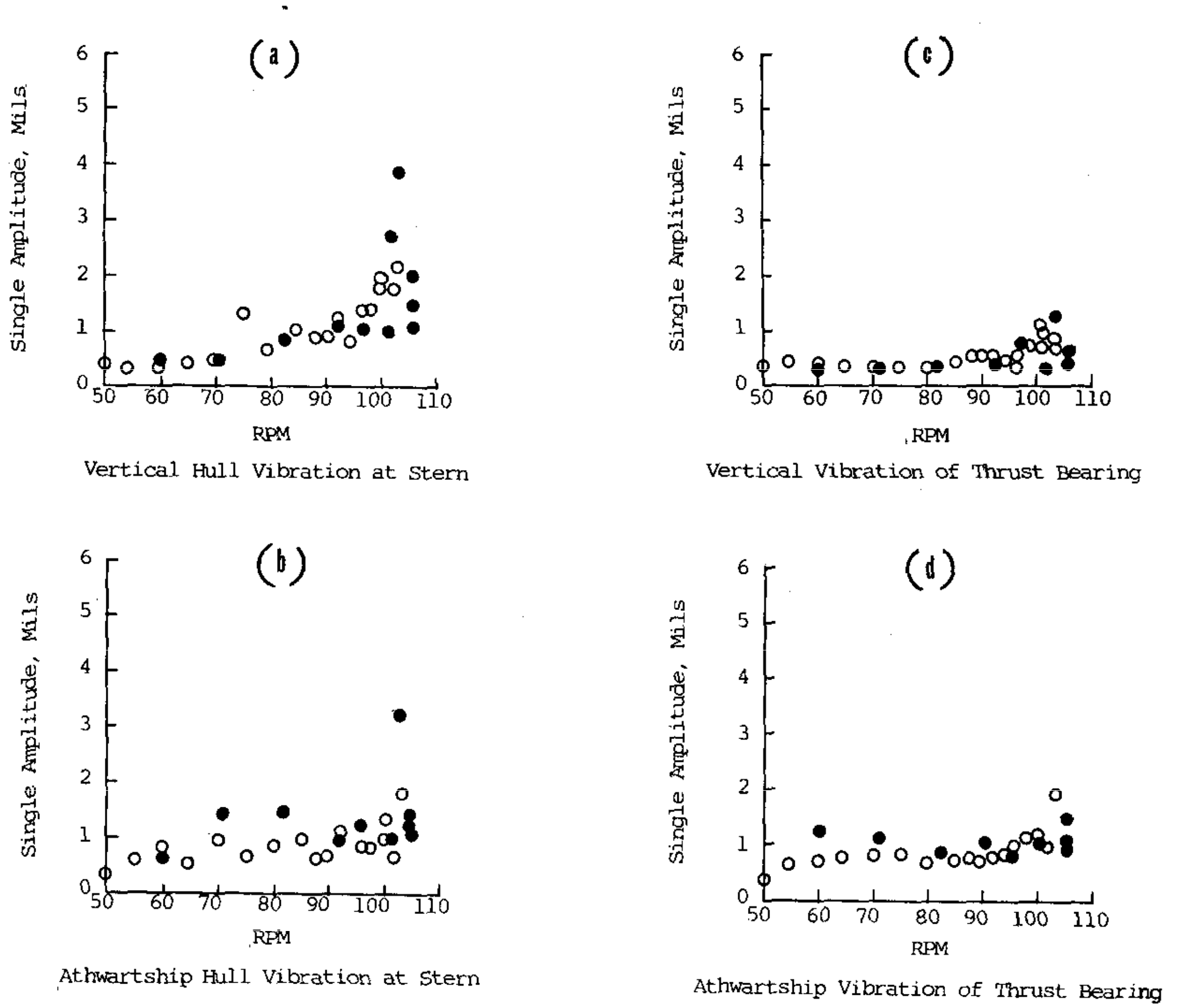


KEY: o Blade Frequency Components

Figure 25 - Longitudinal Vibration of Forward End of Bull Gear Shaft

some estimates will be presented on the range of possible improvement that might have been achieved if a highly skewed propeller alternative been followed in lieu of the structural stiffening approach.

Example - Vessel with Severe Longitudinal Machinery Vibration. Figure 25 displays the results of baseline measurements on the bull gear shaft of a 28,500 SHP container ship that experienced severe longitudinal machinery vibration. Upon discovering the problem, the shipowner was requested to limit the operation of the vessel to 95 RPM and below. After consultation with the turbine and gear manufacturer, 10 mils single



KEY

o Blade Frequency Components Before Structural Modifications

● Blade Frequency Components After Structural Modifications

Figure 26 - Hull and Thrust Bearing Vibration

amplitude was established as the maximum vibration level at which the main propulsion machinery could tolerate without speed restrictions. The objective then became to find a cure that would reduce the vibration level to the 10 mils target level.

In situations such as cited by this example, there are many potential remedial measures that can lead to a technical solution. Unfortunately there always remains substantial uncertainty about the degree of success one might expect with each alternative. Also, a key element affecting the decision process is the reluctance on the part of either the shipowner or the shipbuilder to take the lead in choosing the specific corrective action that should be taken. This situation always exists because of the possibility of failure and the fact that the cost of the failure must be paid be someone. While ultimately a structural cure was found, many alternatives were considered by the participants associated with the subject containership.

To illustrate the range of alternatives, Table 7 has been prepared to outline some of the advantages and disadvantages associated with eight alternative courses of action that were considered to eliminate the excessive vibration encountered on the containership. These alternatives included such concepts as structural reinforcement, relocation of the thrust bearing foundation, propeller modifications, and even consideration of a speed restriction.

The overall effort including discussions, meetings, analytic studies, exchanges of correspondence, etc. consumed slightly more than two months.

Table 7 - Alternative Courses of Action to Correct Excessive Longitudinal Machinery Vibration

Alternatives	Advantages	Disadvantages
1. Cut and decrease thrust bearing foundation stiffness.	Simple modification very little delay in ship delivery. Slight cost in testing modification.	Accepts critical operating condition in range of propeller RPM. Increased static movement of propeller shafting and gear train between ahead and astern conditions.
2. Cut propeller tips and reduce diameter (increase RPM)	Can be accomplished with little delay in ship delivery. Slight cost increase.	Accepts critical frequency in range of propeller RPM. Increased rotational speed of turbines and gears may void warranty. May increase propeller blade cavitation.
3. Twist propeller blades (increase pitch/decrease RPM).	Reduction in shaft speed will avoid steep part of response curves.	Increased shaft torque and gear loading. May void warranty for gears and turbines. All blades may not have equal pitch after modification.
4. Add additional thrust bearing foundation reinforcement.	Will raise critical frequency of shafting system. Little delay in ship delivery.	Difficult to add new structure in existing engine room. Relocation of some machinery components, piping, etc. required.
5. Relocate thrust bearing foundation.	Enables thrust bearing to be keyed into strongest support structure.	May require new line shaft sections. Substantial ripout of existing structure. High costs involved.
6. New propeller	Maximum blade frequency can be moved below critical frequency. Depending on whether conventional or highly skewed propeller followed, excitation forces may be reduced.	New propeller requires 12-18 months lead time. Ship forced to operate at reduced power until replacement. Cost of new propeller plus other potential cost items.
7. Restrict service speed	No changes to vessel needed. May decrease fuel oil cost.	Ship has more power than can be used. Speed less than upon which design and economics based. Money invested in excess power plant.
8. Combinations of above alternatives.	Selected items from above.	Selected items from above.

However, once the decision point had been reached to move forward with structural reinforcement of the thrust bearing foundation, the actual ship modifications were completed within only approximately one and one-half weeks. The reinforcement work included the addition of two new sloping bulkheads in the machinery space and relocation of piping and wiring in the affected area. New bulkheads, approximately 26 feet long and 10 feet high, effectively extended the existing shaft alley longitudinal bulkheads into the machinery space, terminating at the thrust bearing foundation. The total structural reinforcement required about 6½ tons of 3/4 inch plate and associated stiffeners.

Figures 26a through 27b display the "before" and "after" results of

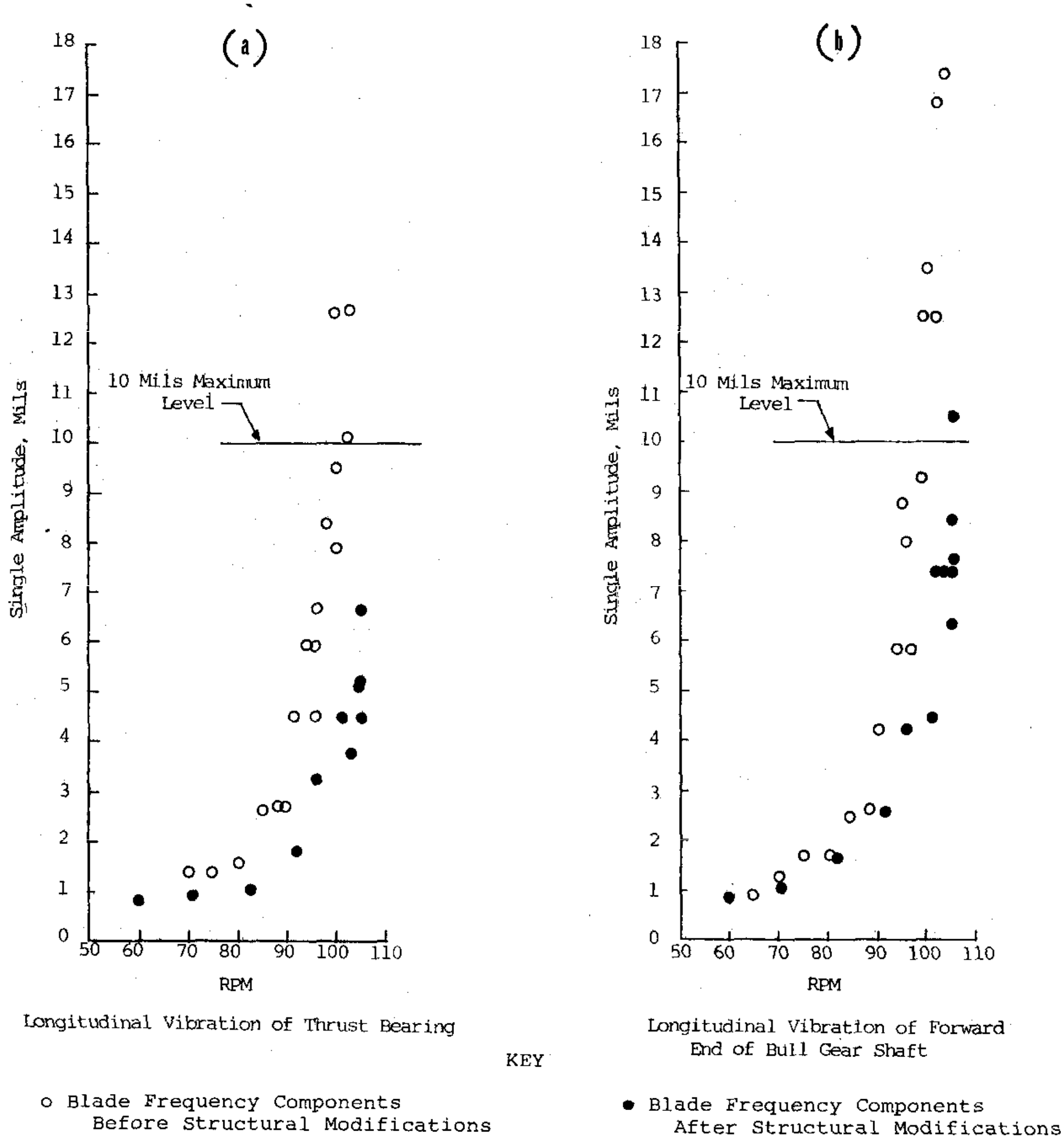


Figure 27 - Thrust Bearing and Bull Gear Vibration

measurements at selected locations on the vessel. Figures 26a and 26b indicate vertical and athwartship vibratory amplitudes of the hull girder as measured at the stern of the vessel, and Figures 26c through 27a indicate the thrust bearing foundation movement in the three principal directions. Figure 27b presents the measurements taken on the forward end of the bull gear shaft. Superimposed on these curves are the results of measurements made after extensive structural reinforcement had been made to the thrust bearing foundation.

From inspection of Figure 27b it can be seen that the initial maximum longitudinal propeller shaft movement was reduced from a maximum amplitude of 17.5 mils to 10.5 mils or a 40 percent reduction at the maximum 28,500 horsepower of the vessel. Looking at the longitudinal movement of the thrust bearing foundation, as shown on Figure 27a, maximum movement was reduced from 12.5 mils to approximately 6.5 mils or a 48 percent reduction. Both figures indicate the extent of test scatter one can encounter when operating close to a critical vibration frequency.

Inspection of Figures 26a and 26b displaying respectively vertical and athwartship vibration of the hull girder, shows that at some propeller speeds vibration was not reduced, but rather increased significantly over the baseline values.

One key observation to be made from the measurements is that it is extremely difficult to achieve a 50 percent reduction in longitudinal machinery vibration by rather massive structural stiffening. Also, while the stiffening may reduce the movement at the location of concern, the additional structural members may redistribute the excitation forces throughout the vessel with some locations actually increasing in the amount of movement.

Based on test results presented for the Sea Bridge vessels, the San Clemente vessels and the Maine class vessels a substantially different result would be expected if a skewed propeller solution had been followed. Bull gear shaft motion would probably have been reduced 60 to 65 percent within the range of the critical speed band (95-105 RPM) and perhap 50 percent at speeds below 95 RPM. Elsewhere, vibration levels would have decreased 50 percent instead of rather increased in level.

If one were given the option today to correct the original condition, which path should be followed? From a time and cost standpoint the structural cure is the better choice. If one had installed a highly skewed propeller at the outset, would there have been a problem in the first place? Probably not.

ECONOMIC ASPECTS AND RISKS

While the discussion on highly skewed propellers up to this point has concentrated primarily on technial aspects, the matter of economics and risks will now be discussed. There are several intangible factors for which it is difficult to place a dollar amount with great certainty. For example, if vibration levels are reduced through use of a highly skewed propeller, how much money will indeed be saved on an annual basis? Since it is not possible to quantify with great precision the "savings" that a shipowner might expect, the discussion presented herein will concentrate primarily on estimating the magnitude of possible cost increases that may be expected for a given situation. By following this path, it is hoped that sufficient baseline information will be presented to enable readers to prepare a cost/benefit analysis tailored to the specific situation in which they may be considering the use of highly skewed propellers.

Cost Considerations

Figure 28 outlines some of the cost factors associated with the design, test and manufacture of conventional and highly skewed nickel aluminum bronze propellers for a range of shaft horsepower levels. It should be noted that these cost figures represent approximate cost levels as of January 1, 1978, for propellers corresponding to those already installed on the merchant ships discussed in this paper. Cost amounts in Figure 28a represent the situation where only 1 propeller is designed, manufactured and installed aboard a vessel whereas cost amounts presented in figure 28b represent the situation

Table 8 - Ship Construction Situation Scenarios

		Alternatives	
Case No.	Situation	Conventional	H. S. Propeller
1	No problems expected, and none encountered.	Baseline Costs	Cost increase. Lower vibration levels.
2	No problems expected, however problem encountered.	Unexpected corrective work Cost increases.	Problem may be avoided.
3	Problems expected but not encountered.	Some structural corrective action may be required.	Similar to Case No. 1.
4	Problems expected and encountered.	Corrective actions needed. Perhaps problem not cureable by conventional methods.	Minimizes extent of problem. Further corrective action required.

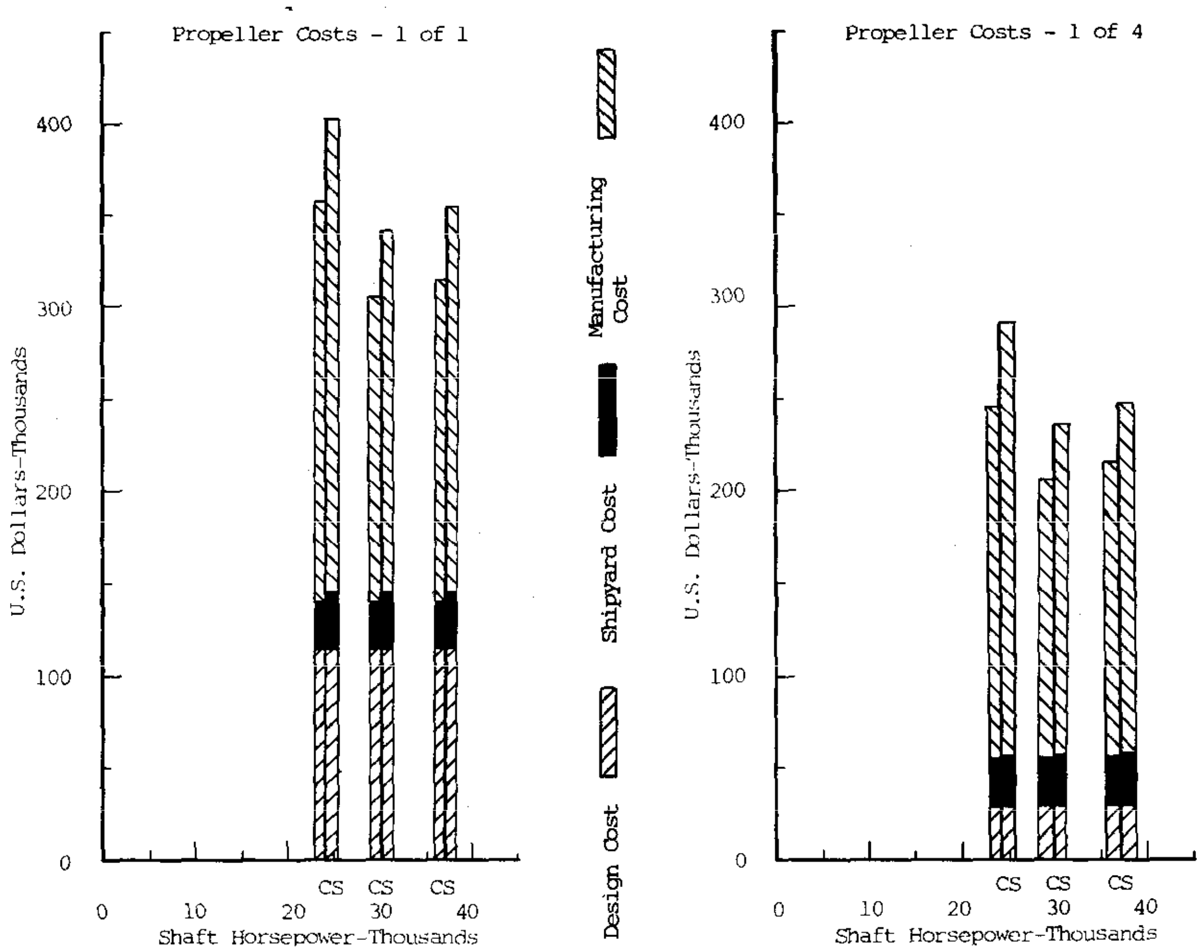


Figure 28 - Cost Factors for Conventional and Highly Skewed Propellers

where a series of four propellers are designed, manufactured and installed aboard a series of vessels. Because of the history associated with the background of these designs, where primary concern was to minimize risk of failure, they therefore do not represent designs optimized on the basis of both performance and cost. In other words, the cost differential for follow-on designs could be less in the future because of the experience gained from the propellers outlined in Table 5.

Figures presented as design costs are based upon government and industry estimates and the authors found that there is little or no difference in design costs for skewed propellers versus conventional propellers. If, however, the skewed propeller is being designed with special consideration towards vibrational problems it is possible some additional costs could be incurred.

Figures presented as shipyard costs are based on the cost of installation of the propeller plus the freight cost of delivering the propeller to the shipyard. Some cost differential is found for skewed versus conventional, and this is a result of some additional cost for installation for the skewed propeller.

The figures presented as propeller manufacturing costs are the costs of materials and construction for the propeller. It should be noted that inasmuch highly skewed propellers tend towards greater weight there is a corresponding greater cost.

Nevertheless, using the cost information developed for the existing propeller designs, it can be noted that a highly skewed propeller will cost somewhere between 11 and 13 percent more than the baseline conventional propeller. However, this clearly may not be the complete story, because several potential cost items have not been considered. These cost items are: cost for larger tail shaft, cost for larger stern tube bearing, cost for larger stern tube, realignment of shafting system. All of these items are affected by the magnitude of the increase in weight of the highly skewed propeller over that of the conventional propeller. Fortunately for all of the projects described in the paper, physical changes to the tail shaft, stern tube bearing and stern tube were not required based on results of the engineering analysis. Adjustments in shafting alignment were needed for only one of the three ship projects.

Risk Scenarios

Moving forward, one must consider the possible choices facing a shipowner or shipbuilder when contemplating propeller selection for a new ship design. Basically, the tendency in the past decade has been that ship specification requirements with regard to vibration have become more specific and detailed. This situation has arisen because of numerous unpleasant experiences in the past with new ship designs. Again focusing on Figure 4 pertaining to the horsepower growth curve it is clear that ship designs breaking new ground as plotted on this figure face the risk of encountering unacceptable vibration levels. Indeed, experience has shown that even if the design does not break new ground, the state-of-the-art is such that one cannot guarantee problems will not be encountered.

Consider then four possible situation scenarios outlined in Table 8. Case Number 1 portrays the situation where "no problem was expected and none was encountered". This situation corresponds to that described in the San Clemente project where the ULTRAMAR and ULTRASEA could ultimately operate successfully from a vibration standpoint without restrictions. Vibration levels were clearly lower on the vessel fitted with the highly skewed propeller, and both propellers were off the design RPM.

Case Number 2 represents the situation where "no problems were expected but severe problems were encountered". This situation coincides with that described concerning the Sea Bridge class. A cure could not be found following the path of structural modifications because a complete redesign of an existing structure would have been necessary. Installation of highly skewed propellers at the outset would have avoided the problem completely. Given the identical circumstances today, investment of $35,000 or 11.5 percent more than the baseline cost with the conventional propeller would have avoided the problem.

The situation described concerning the containership longitudinal machinery vibration becomes somewhat of a toss-up. Investment of $40,000 or 11-13 percent more than the baseline cost initially would have avoided the problem. However once the problem had been encountered, a structural cure was found within a matter of weeks as opposed to 12-18 months if it had been decided to follow the path of designing and constructing a highly skewed propeller.

If on the other hand the structural modifications had not solved the problem and one then was forced to proceed with the design and manufacture of a highly skewed propeller at that point in the project, an additional outlay of $350,000 would have been required. Also a situation would have been created wherein there would have resulted at least one unusable conventional propeller posing a disposal problem and further increasing costs.

Case Number 3 represents the situation where "problems were expected but none encountered". This was the situation relative to the Maine class RO/RO vessels. Although the vessels did not exceed previously established horsepower levels, elements of the ship design were significantly advanced to pose real concern about possible excessive vibration. Again, recognition must be given to the state-of-the-art of vibration prediction technology wherein it is still not possible to place total confidence in prediction results. If the situation anticipated had been encountered, namely excessive vibration, the cure had already been developed and was on hand for prompt corrective action.

Case Number 4 represents to a degree an unusual situation. Here, "problems are expected and indeed encountered". This could have been the situation with the Maine class vessels, but it was not. Discussion on this situation is very difficult to present, because it must be tailored to the specifics of the problem. In other words, the first step of resolution must be to identify the possible courses of action that could be taken to rectify the problem. The alternatives could include rearrangement of structural members, movement of machinery components, all the conventional remedies such as change of propeller RPM by cutting blade tips, etc. If the problems were anticipated, those participating in the engineering decisions would most likely have some idea of possible cures. In fact in some instances where a problem is expected (although minor in nature) it is routine practice of some shipyards to delay the cure until there is positive confirmation of the existance of the problem. If the magnitude of the projected problem is great, however, there may not be any inexpensive cures available or on the horizon. Thus far, the performance of highly skewed propellers has been so remarkable that if this choice of propeller type had been

Table 9 - Shipowner Experience with Highly Skewed Propellers

	Questions	Shipowner No. 1	Shipowner No. 2	Shipowner No. 3
Q1.	Is there any noticeable speed difference between ship(s) fitted with highly skewed propeller(s) and conventional propeller(s)?	No.	There is no speed difference between ships fitted with highly skewed as compared to conventional propellers.	No.
Q2.	Is there any noticeable fuel consumption difference between ship(s) fitted with highly skewed propellers(s) and conventional propeller(s)?	No.	There is no noticeable fuel consumption difference on between ships fitted with highly skewed as compared to conventional propellers.	No.
Q3.	Has the crew ever commented on reduced noise/vibration levels of ship(s) fitted with highly skewed propellers?	Yes. Appreciable reduction noticed at maximum power and operation in shallow water.	All crews have commented very favorably on reduced noise/vibration levels on ships fitted with highly skewed propellers as compared to the conventional propellers.	No.
Q4.	Have highly skewed propellers shown a tendency for greater or reduced propeller blade cavitation erosion relative to conventional propellers?	No. To early to judge.	Believe that the highly skewed propeller has a reduced tendency to propeller blade cavitation erosion as compared to the conventional propellers.	No.
Q5.	Has any highly skewed propeller shown more susceptibility for damage from floating debris relative to a conventional propeller?	No.	Had only one occurrence, i.e. blade damage and discount the feeling that a highly skewed propeller would be more susceptible to floating damage as compared to a conventional propeller.	No.
Q6.	Has there been any reduction in equipment failures such as radars, controllers, etc. on ships fitted with highly skewed propellers?	No.	There has been a drastic marked reduction in navigational communication, bridge equipment failures on vessels fitted with highly skewed propellers as compared to conventional propellers.	No.
Q7.	Have ship masters commented on any noticeable differences is astern backing power on ships fitted with highly skewed propellers relative to conventional propellers?	No.	Ship masters have commented that highly skewed propellers in a light ballast condition there is a noticeable difference in back power by approximately 25 percent as compared to a conventional propeller. In laden condition, i.e. with full propeller immersion there is no difference.	No.
Q8.	How many ship months of service operation has each highly skewed propeller seen?	22 months	1. 12 months 2. 24 months 3. 16 months 52 months	1. 15 months 2. 13 months 3. 7 months 35 months

Table 9 (Cont'd) - *Shipowner Experience with Highly Skewed Propellers*

Questions	Shipowner No. 1	Shipowner No. 2	Shipowner No. 3
Q9. If you were to build additonal new vessels would you prefer a highly skewed propeller over a conventional propeller?	Yes	Yes, without reservation.	Highly skewed propeller.
Q10. As conventional propellers near the end of their service life would you replace them with highly skew propellers?	Yes, for ships with vibration problems.	Yes, without reservation.	Yes

selected and the propeller installed without achieving a complete cure to the projected problem, it would indeed be a difficult problem. While this situation is conceivable, it is not likely. However, the investment in the more expensive highly skewed propeller would already have substantially reduced the number of shipboard locations experiencing the excessive vibration, thereby diminishing the magnitude of the corrective effort. Generally however situations such as this are never resolved from a technical standpoint. The vessel is placed in service with a speed restriction with which it must operate for the remainder of its useful life. Surely a skewed propeller would "payoff" in this situation.

CONCLUSIONS

Thus far considerable discussion has been outlined in preceding pages on full-scale vibration test results, alternative solutions to solving vibration problems, and lastly some economic and risk considerations regarding the choice of propeller type. The purpose of this section is to present an overall assessment of the performance of highly skewed propellers and outline some general conclusions that the authors' have reached.

It may be recalled in the initial portion of the paper, specifically Table 1, a number of merits and disadvantages were cited regarding highly skewed propellers. Some of the disadvantages cited were: higher costs, greater fuel bills, more susceptibility to damage, shorter propeller life, etc. The merits for the highly skewed propellers were: reduced vibration levels, improved crew comfort, reduced repairs to navigation equipment, etc.

In order to examine these alleged merits and disadvantages more fully a questionaire outlining ten fundamental questions was prepared and distributed to each of the three shipowners[9] presently operating ships fitted with the skewed propellers. The questions and shipowner responses are summarized in Table 9.

Based upon examination of operating experience ranging from 22 to 55 months, it is apparent that each shipowner is please with the overall performance of the highly skewed propellers and when faced with the decision in the future will most likely choose a highly skewed propeller over the conventional type. Also, based upon the experience accumulated thus far there does not appear to be any speed or fuel penalty associated with the highly skewed propellers.

Therefore based upon the vibration measurement results, shipowner experience, and lastly economic cost and risk considerations the authors have reached the following conclusions:

- Highly skewed propellers reduce overall ship vibration levels approximately 50 percent. Greatest improvement appears to coincide with resonant conditions where highly skewed propellers may reduce vibration levels 65 percent or more.
- Crew comfort has improved on all ships fitted with highly skewed propellers, however the crew is only aware of this on only two of the three projects discussed in the paper.
- Highly skewed propellers can be installed on vessels for about 40,000-$50,000 increased over

[9] As of April 1978, Farrell Lines became owner of all American Export Lines vessels including the Sea Bridge class ships.

conventional designs. This diffential should narrow as greater effort is made to design for desired performance at minimum cost.

- Operating results indicate there is no noticeable speed penalty on vessels fitted with highly skewed propellers.

- Operating results indicate there is no noticeable fuel cost increase on ships fitted with highly skewed propellers.

- Operating results indicate that highly skewed (H.S.) propellers do not appear to be more susceptible to damage than conventional propellers. However once damaged, repair costs will be greater on highly skewed propellers.

- Highly skewed propellers have not yet clearly demonstrated less cavitation erosion than conventional designs, however the blade erosion is about the same as conventional.

- Prediction of propeller RPM is improving, however model testing methods may miss target propeller RPM up to 5 percent.

- Full potential of H.S. benefits at horsepower levels greater than 40,000 SHP remains to be demonstrated.

- Maximum horsepower for single screw ships fitted with H.S. or conventional propellers is not known. However it appears to be in excess of 50,000 SHP.

- Service life of H.S. propellers from a fatigue standpoint has not been demonstrated to be greater or less than conventional designs.

- Structural cures of vibration problems may increase levels of vibration elsewhere in a vessel whereas there is no evidence this occurs using highly skewed propeller cures.

- Propeller design technology has not yet developed analytical or model techniques for determining optimum skew angles and/or distributions primarily because of unknown contribution of pressure forces.

- In regard to effectiveness of the three highly skewed propellers designs in reducing ship vibration, the San Clemente class propeller probably had excessive skew, the Sea Bridge class propeller close but slightly more than optimum skew, and the Maine class propellers too little skew.

- Stern tube bearings and tail shafts need not be larger than those for conventional propellers provided the detailed engineering phase contemplates heavier propellers at the outset.

In regard to a final assessment or overall conclusion the authors' must state that the successful development of highly skewed propellers for merchant ships represents the single most important advance in propeller-induced ship vibration reduction technology within the past decade and perhaps within the last century. While the ultimate potential of these novel propeller designs to operate with less cavitation erosion has not yet been demonstrated, this aspect may from an economic standpoint ultimately prove to be the greatest benefit resultant from the highly skewed propeller concept. Therefore propeller designers are encouraged to continue with their work and shipowners encouraged to install such propellers on their vessels.

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APPENDIX A

HIGHLY SKEWED PROPELLER DESIGN CONSIDERATIONS

Propeller blade skew has been defined as the displacement at the mid-chord point of the blade section from the blade reference line along the pitch helix. Generally, the value of the angular displacement in the projected view at the tip of the blade is used as a measure of the skew (13). Skew can, perhaps, be most clearly defined by Figure A1.

The amount of skew for any given propeller is calculated by measuring the angular displacement of all the blades and dividing that displacement by the number of blades. For example, if the angular displacement of the blades of a certain six-bladed propeller is 180° then; 1) the skew angle is 30°, and 2) the skew is 50 percent. Skews of 50 percent or greater are considered to be highly skewed by propeller designers. Skews greater than normal amounts are generally considered highly skewed by shipowners and shipyards.

Skewed propellers were illustrated by Bourne over one-hundred years ago. Bourne presented two concepts, one by Beadon proposed in 1851, Figure A2, and another by Hirsch proposed in 1860, Figure A3. Beadon proposed essentially a two-bladed propeller with 100 percent skew or 180°. The object of this concept was that "inasmuch as the cutting edge, by not coming into such direct and rapid contact with the water, will not

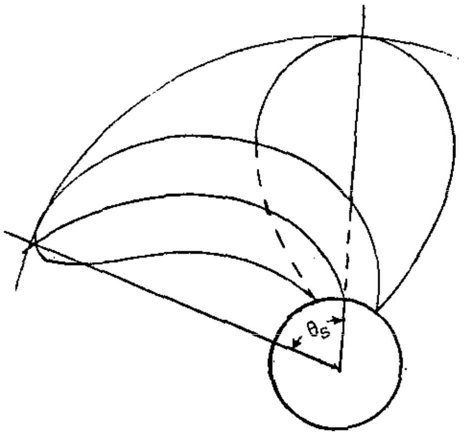

Figure A1 - Definition of Skew

experience so much resistance", appears to be a first attempt at reducing propeller induced forces. Bourne stated that Beadon's propeller did not appear to be as strong as the common form of screw propeller. Hirsch proposed a left handed propeller that used skew distributed reversely to that used today. In his concept, the blade tip section would come into contact with the water first with successively lower blade sections coming into contact gradually (14).

The hydrodynamic force (lift) generated by a screw propeller which propels a ship through the water also causes propeller induced vibration. The lift, developed by the propeller blades is periodic and when the blades rotate in the non uniform velocity field behind the vessel the lift becomes unsteady. This action causes the generation of unsteady forces and moments. The above forces and moments

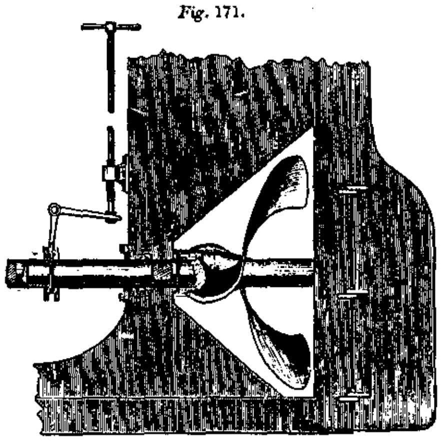

Figure A2 - Beadon's Screw Propeller

are the source of the hull vibration.

The unsteady forces and moments can be divided into two groups, pressure-forces and bearing-forces. Pressure-forces are transmitted to the hull by the action of the water against the shell and appendages. Bearing-forces are transmitted to the hull through the shafting to the bearing and then to the hull. Efforts to avoid propeller induced vibration in the past, have generally consisted of selecting the blade number and RPM to avoid critical hull frequencies and providing adequate clearances in the propeller aperture. Studies conducted at DTNSRDC have indicated that substantial propeller blade skew may significantly reduce propeller-induced vibration, both hull pressure-forces and bearing-forces (15). Skewing a propeller blade allows each section of the blade to enter the wake at a different instant, thus reducing the peak forces. That is the skewed propeller blade, which is rotating in the wake, is more gradually introduced into the water flow thereby affecting decreased forces compared to a conventional propeller.

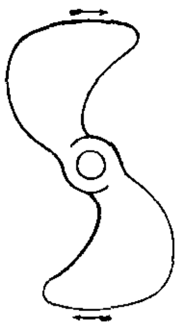

Figure A3 - Hirsch's Screw Propeller

In addition model highly skewed propellers have been found to have a decreased susceptibility of the propeller to cavitation when operating in a wake. To properly evaluate the reduction in propeller-induced vibration obtained by applying skew angle, the total vibration excitation force, which is a vector sum of the pressure-force and bearing-force components, must be considered (15).

With regard to bearing forces it has been found that skew effectiveness is different for propellers with odd or even number of blades. For vessels of a certain design high skew tends to be most effective in reducing the forces that are the largest for unskewed propellers. Thrust and torque are reduced for even-bladed skewed propellers but vertical and transverse horizontal side forces are reduced for odd-bladed skewed propellers.

At this time it is beyond the state-of-the-art to determine precisely the magnitude and distribution of skew angle that will minimize the total, pressure plus bearing, vibration excitation force in a specified direction. A skew angle can be designed though, that will reduce the magnitude of the bearing forces. Boswell and Cox (1) found that the data they studied indicated that skew at the blade tip of 100 percent is generally desirable. But, they also found little guidance as to the proper tip skew and radial distribution of skew that should be applied.

APPENDIX B

HISTORICAL BACKGROUND

Historical records citing instances of excessive ship vibration date back to the days of sail and wooden vessels. For purposes of this Appendix however the starting point will be approximately the year 1850 when steam engines and screw propellers started to come into popular usage.

Figure B1 outlines the growth of installed horsepower for single screw merchant ships for the period 1850 to 1978. Also shown on this figure are approximate dates for major technological advances affecting ship vibration either from a excitation or response standpoint. Introduction of new engine types, greater horsepower levels, would be considered excitation factors; whereas changes of hull materials, construction methods, etc. would represent typical vibration response factors.

While most naval architects have read in the literature some facts about the remarkable steam vessel the GREAT EASTERN propelled by both paddles and a screw propeller, it is interesting to note that this 680 foot vessel built in 1858 had paddle engines of 1000 horsepower and screw engines of 1700 horsepower (14). The propeller construction was years ahead of its time with the four bladed propeller being 24 feet in diameter coinciding roughly with the propeller diameters for the three highly

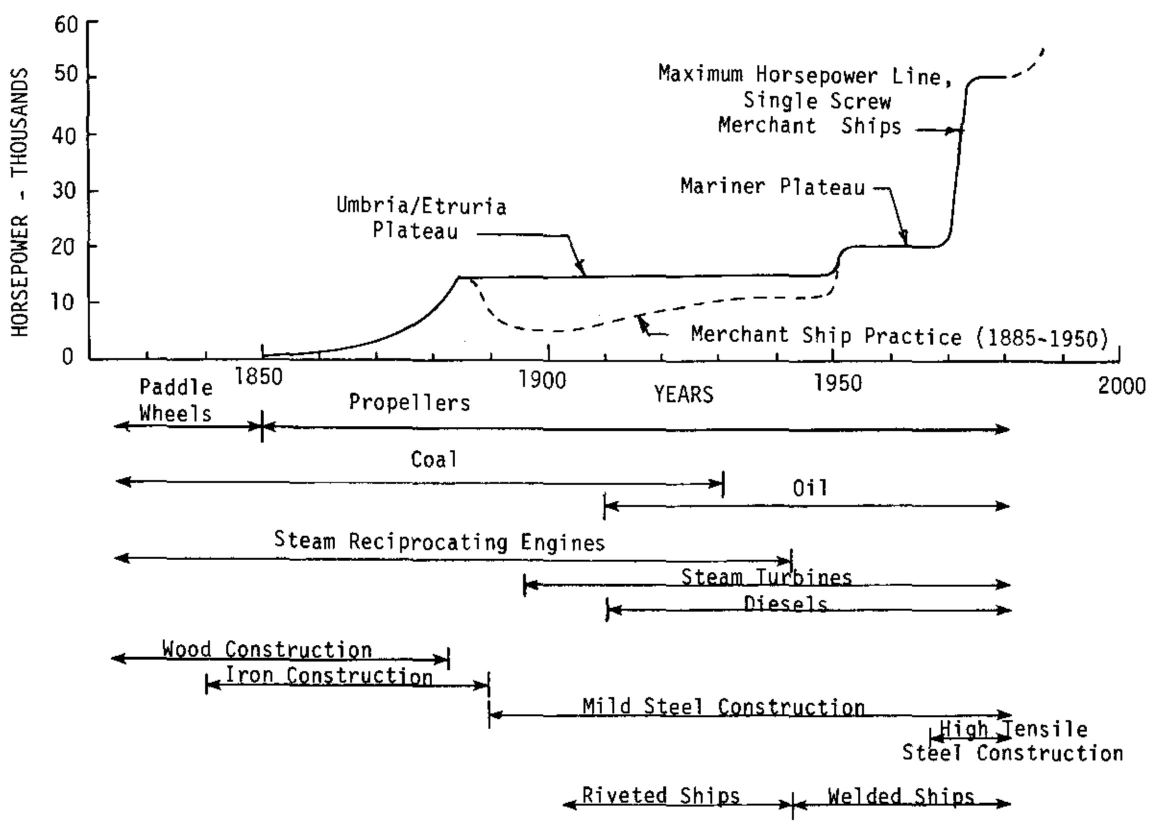

Figure B1 - Growth of Single Screw Merchant Ship Horsepower and Vibration Excitation Response Factors skewed propeller projects discussed in this paper. Figure B2 displays the propeller/hull stern configuration used on the GREAT EASTERN and also shows some propeller construction details.

According to Corthell (16) in the year 1848 there were 128 steamers in existence with only 36 of these built of iron. For comparison purposes, at the same period there were 10,579 sailing vessels with but 79 of these vessels built of iron. By the year 1883, the construction of new vessels had reached the point where 83 percent were constructed of iron with 15 percent being constructed of steel. Only six years later hull material had changed to where 92 percent of the new vessels were constructed of steel and but 8 percent of iron.

By the year 1884 the installed horsepower of single screw ships reached a record high of 14,700 when the Cunard Line single screw passenger vessel UMBRIA commenced trans-Atlantic service. The steam vessels UMBRIA and ETRURIA (her sistership) operated successfully almost 25 years until 1910 and 1909, respectively when they were laid up and ultimately scrapped. Although the authors' have not located any records citing specific vibration or propeller blade erosion problems these vessels may have had, the UMBRIA and ETRURIA are known to have operated with but two broken propeller shaft failures during their years of service. As an interesting item, reportedly there were 11 engineers, one electrician and 109 firemen needed to operate these vessels and in March 1887 the ETRURIA completed an eastbound crossing at an average speed of 19.45 knots.

Thus after only approximately 35 years following introduction of the screw propeller, steam propelled vessels had developed rapidly to the limits of technology for single screw vessels and the UMBRIA and her sistership the ETRURIA appear to have held a record horsepower level that was not surpassed until the Mariner class ships were introduced in 1951, some 66 years later.

The UMBRIA and ETRURIA were the last of the high-powered single-screw passenger vessels and in the period 1885-1890 twin screw vessels such as the CITY OF PARIS and OCEANIC appeared further extending the horsepower growth curve. Although such famous vessels are not plotted on Figure B1 which is limited to single-screw vessels it should be noted that the 68,000 horsepower level was reached by the quadruple-screw steamers MAURITANIA and LUSITIANA in 1907.

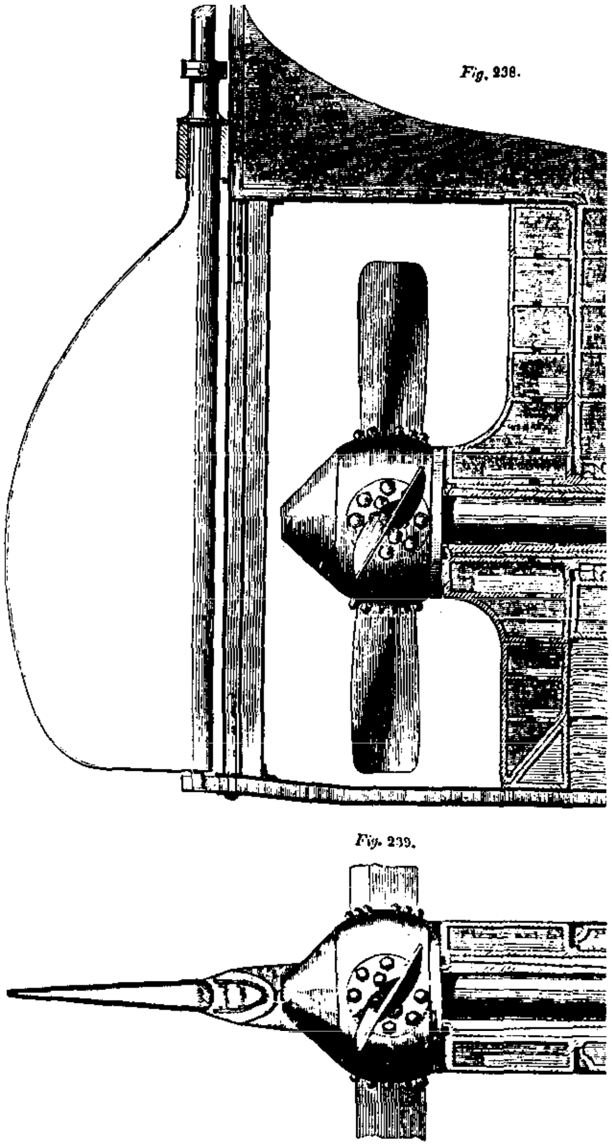

Figure B2 - GREAT EASTERN Propeller/Hull Stern Configuration

While the historical records tend to discuss propeller and vibration problems in only descriptive general terms, it is clear that some of the problems however were very real in that many instances had been encountered where vibration had loosened rivets in the sternpost of vessels and shipowners were concerned about the possibility of a calimity.

In 1884, Otto Schlick presented a landmark paper "On the Vibration of Steam Vessels" to the Insitution of Naval Architects (17). He opened the paper with the comments:

"All steamers, without exceptions, shake to a more or less degree when the engines are in motion. This phenomenon is usually considered as so natural, that in most cases little or no attention is paid to it, and when ships with comparatively powerful engines show an unusually strong vibration, it is looked upon as quite natural, or the phenomenon is simply accounted for by saying that the ship is of too weak construction."

Schlick continued and outlined his ideas on hull natural frequencies, elasticity, and comment on the forces produced by a working engine, citing the following: Forward thrust, turning couple of the engine, sideward pressure of the propeller, pressure or reciprocating masses, and the pressure of rotating masses. He even commented on the aspect of what should be done in order to avoid or to diminish violent vibration, citing a change in propeller, alteration of number of revolutions of the engine, redistribution of cargo, as possible alternatives.

It should be noted that while the undesirability of vibration was known, instruments for measurement of actual full scale vibration had not been developed. Thus comments about excessive vibration during and before the 1884 time period were subjective in nature.

In 1893 Schlick presented another significant paper (18) describing an apparatus for measuring ship vibration levels. Thus until Schlick's invention there was no instrument available that could be used to measure or analyze ship vibration.

While earlier reports frequently cited stern movements in the order of 1 or 2 feet or more, actual measurements of full scale vibration using Schlick's device confirmed that previous subjective estimates were vastly overstated and that true values were but a small fraction of earlier reported amounts.

In 1893, C. H. Cramp, (19) indicated "you cannot put more than 12,000 IHP through one screw". Further more, "whenever you require more than 12,000 IHP you must have 2 screws and if you find it necessary to exceed 24,000 IHP three screw will be required".

In light of the existence of the UMBRIA/ETRURIA at this time period one can only believe that these vessels may have had substantial shipboard vibration and/or propeller problems or that Cramp's proposed rule of thumb was unfounded.

By the year 1910 motorships started to appear and a new source of vibration excitation appeared. The seagoing motorship-oil tanker, the VULCANUS having 480 horsepower and speed just over 6 knots appeared. At approximately this time period major emphasis was concentrated on reducing excitation forces created by reciprocating steam engines and diesel engines.

By 1925, in response to numerous shafting failures that had been experienced, F. M. Lewis presented his first paper on torsional vibration of diesel engines (20). Only two years later Lewis presented another paper (21) further discussing vibration and engine balance in diesel ships.

Probably by the beginning of 1930 all major types of propeller induced or machinery induced vibration problems had been encountered on merchant ships. While the literature since then contains numerous articles on ship vibration, the growth of horsepower, ship size and major technical advances that took place in the period of 1885 to 1920 is truely remarkable.