Operating Stresses on S. S. "Michigan" Propeller

Extensive laboratory development program is carried out to test gaging attachments and protection systems in salt-water environments

by Given A. Brewer

ABSTRACT—The failure at sea of a number of merchant-ship propeller blades initiated a research program supported by the Maritime Administration, the American Bureau of Shipping, the American President Lines and the States Steamship Company. Although extensive theoretical work has been carried out in the past (predicting propeller operating stresses), very little full-scale experimental stress measurements have been obtained.

Brewer Engineering Laboratories, Inc. was retained by the sponsors to attach and protect fifty strain gages on the test propeller at the locations chosen by Littleton Research and Engineering Corporation.

The propeller strain gages were protected from the severe environment by means of synthetic rubber and metallic overlays. The gage wiring was led down each blade and into a specially machined, hollow tailshaft through a waterproof bulkhead.

The S.S. "Michigan," with the test group aboard, sailed from Long Beach, Calif., in April 1970. The ship called at the ports of San Francisco, Yokohama, Naha, Manila, Bogo and Hong Kong during the test period. Throughout the trip across the Pacific, the gaging and instrumentation functioned without degradation or evidence of distress after one-million propeller shaft turns. At the end of five weeks at sea and two-million turns, all but eight gages exhibited low ground resistance.

The S.S. "Michigan" went into dry dock at the Bethlehem Steel Shipyard at Terminal Island, Calif., on June 15, 1970. The propeller was removed and all strain gages checked for continuity and ground resistance. Forty-two of the original 50 strain gages were found to be intact. No sign of deterioration due to cavitation or electrolysis attack existed on the metallic overlays or spot welding. Wire chafing within the fairwater adapter cap accounted for the low ground resistance values observed at the end of the voyage to Hong Kong.

Introduction

Several failures of aluminum-bronze-alloy propeller blades in maritime service have been experienced in recent years. A number of these failures occurred on the Colorado class ships of which the S.S. "Michigan" is one. Blade fractures initiated at the 0.25 radius and resulted in the complete loss of the blade in a comparatively short period of time. Other inspections of apparently intact propellers at dry docking, revealed cracks in the region of the 0.25 radius. In some cases, the blades fractured in less than a year of operational service.

In an effort to advance the state of the art in marine propeller design and to concomitantly determine the cause of premature blade failure, several agencies combined to support a program of research.[1] These agencies were: the States Steamship Company, San Francisco, Calif.; the Maritime Administration, Washington, D.C.; the American Bureau of Shipping, New York, N. Y.; and the American President Lines, San Francisco, Calif.

The test ship was chosen by the program sponsors to be the S.S. "Michigan." This merchant ship was designed by the George C. Sharp Company. She is 579 ft in length, has a full displacement of 22,692 tons, a design draft of 27 ft, and cruises at 23 knots. It is a single-screw vessel absorbing 21,000 SHP from its turbines at cruise. The five-bladed propeller is cast from Superston 40 (ABS Type 5), weighs 51,700 lb, and is 22 ft 6 in. in diameter.

Experimenters, both here[2] and abroad,[3,4] have succeeded in measuring operating stresses on large marine propellers. The work in Europe (on a 42,000-ton tanker) involved 10 three-element rosettes switched sequentially into a single-channel slip ring. The installation survived submergence for about a month. In this country, some 18 gages were attached to the starboard outboard propeller of the "Franklin D. Roosevelt." The gage circuitry failed after about 47 hours of submergence. Despite this early circuitry mortality, considerable data were recorded by the test crews.

The S.S. "Michigan" tests were planned to comprise 50 points of stress measurement on the propeller and, additionally, to include propeller shaft thrust and torque recordings. Vibration measurements of the ship hull at various speeds and draft were included in the test program.[1] The strain-gage installations and recordings, together with the thrust and torque-measurement responsibility, were assigned to Brewer Engineering Laboratories, Inc. (BEL). The location of the transducers and recording of the vibration data were assigned to Littleton

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Research and Engineering Corporation, together with the responsibility for writing the final test report.

Installation

A number of Superston 40 (ABS Type 5) constant-stress plate bending specimens were machined and strain gages cemented to their surfaces. Protective overlays were then spot welded to the surfaces of these test plates. A technique involving synthetic rubber overlays with a final shingling of stainless steel had previously been developed for the protection of gages attached to Francis turbine and steam-turbine blading.[5] This technique was modified, both in stainless composition and in welding discharge energy, for the propeller installation. Completed installations were vibrated in a seawater environment at zero mean stress, and at mean and cyclic stresses estimated to be the maximum encountered in service ($10{,}000 \pm 5{,}000$ psi to 10^8 cycles and one month of exposure). The final gage installation was also spun on a Superston disk in a tub of seawater for a month at the erosion sweep speeds occurring at the 0.25 propeller radius (31 fps) at 106 rpm.

The strain gages employed were BLH Type FAB-12-35S11. Fourteen 3-element rosettes and eight linear gages were cemented to the propeller blades, as shown in Fig. 1. The gage areas were cleaned mechanically and chemically; and then the strain gages were cemented down, utilizing Micro-Measurements certified Eastman 910 cement. Armstrong C-3 with flexibilizer was painted over each gage for mechanical protection and then followed by two coats of Products Research & Chemical synthetic rubber. The final protection comprised shingling over the areas and wiring, utilizing 0.003-in.-thick stainless-steel shimming. The shingling was spot welded to the propeller blading, utilizing condenser-discharge portable welders.

Instrumentation of the propeller was accomplished

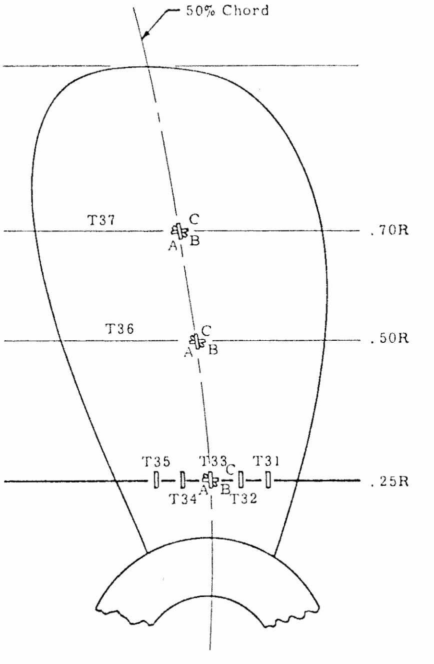


Fig. 2—Master-blade (No. 3) tension side

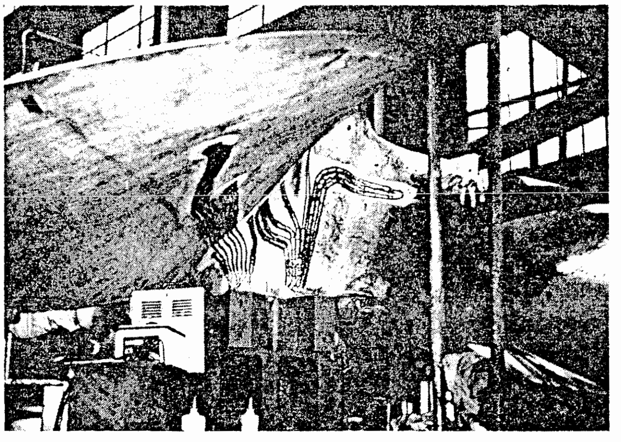


Fig. 3—S.S. "Michigan" propeller showing strain gaging

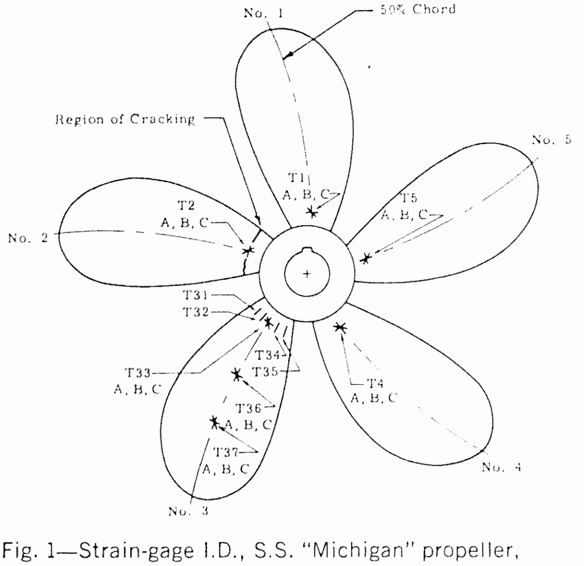


Fig. 1—Strain-gage I.D., S.S. "Michigan" propeller, tension side shown

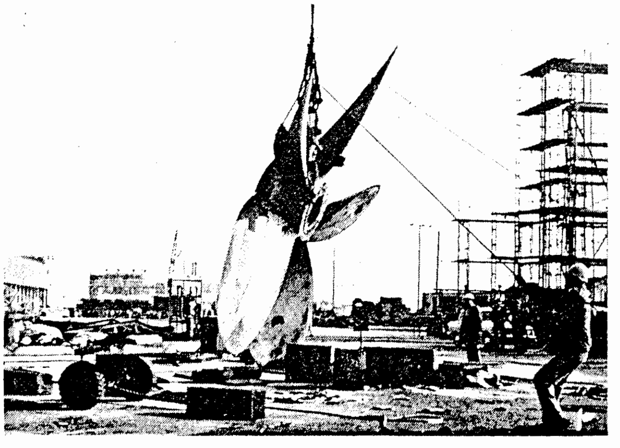


Fig. 4—Propeller being lifted into position

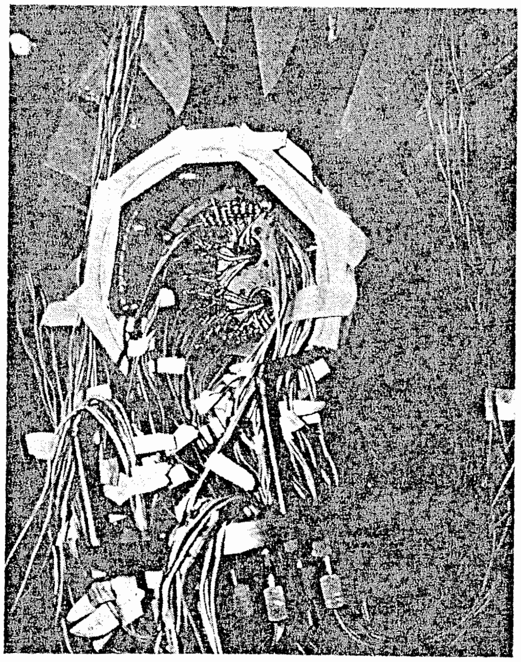
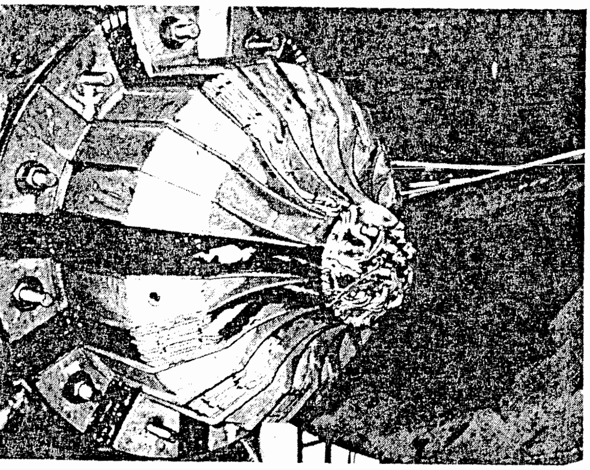


Fig. 5—Tailshaft terminals before closure

in Long Beach, Calif. Modulus of elasticity of the blades at a number of points was determined by sonic means before fastening to the tailshaft.[6] Fifty strain gages were attached to the five blades of the test propeller furnished by the American President Lines (see Figs. 1, 2, 3 and 4). The location of previous failures is sketched on Fig. 1. A special steel adapter cone was fabricated and fastened to the end of the tailshaft (Fig. 5). Strain-gage wiring was led down the blades over the adapter into Conax fittings penetrating a bulkhead sealing off the end of the hollow tailshaft (see Fig. 6). Terminals were provided inside the tailshaft for termination of the strain-gage wires penetrating the Conax fittings. Four multiconductor cables were led through the tailshaft and fastened to Bakelite terminal strips secured to the flange of the end of the tailshaft within the hull. Three-wire circuitry from the gages to the terminal strips was employed. The cables were secured against vibration within the tailshaft by the introduction of a foaming plastic. The streamlined fairwater cap was then fastened over the adapter cone and the inner space foamed, completing the submerged portion of the instrumentation (see Figs. 7, 8 and 9).



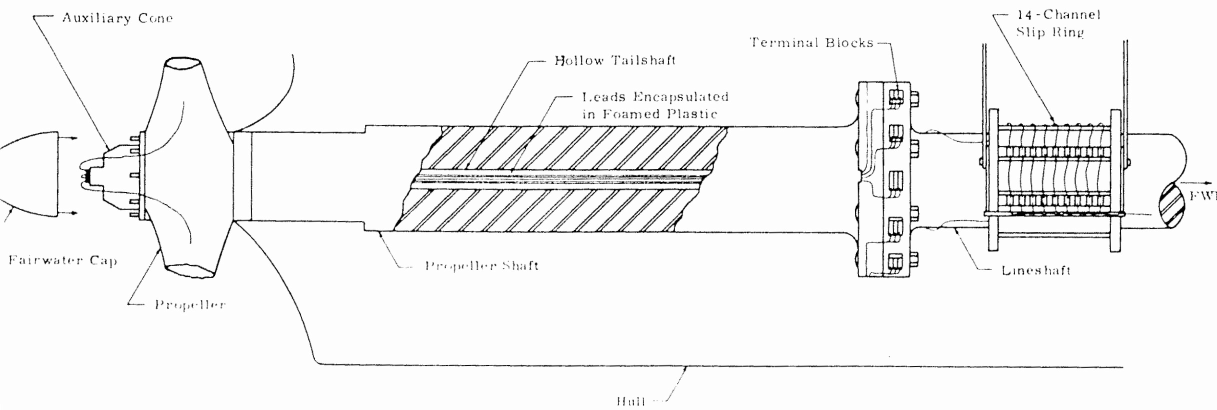
g. 6—Strain-gage wiring and fairwater adapter cap

Instrumentation

A 20-element split slip ring (14 channels), Figure 10, was fastened to the drive shaft in the propeller-shaft tunnel. Connections from the gage terminals to the slip ring were made in groups of 14 in accordance with the test schedule. Cabling from the slip ring was run up 300 ft to the assistant chief engineer's office where the recording instrumentation was installed (see Fig. 11). Two separate power rings and common rings were utilized so that, in the event of failure of a single gage, only 50 percent of the system would be lost.

A BLH-Waukesha thrust meter and a radio torquemeter were appropriately installed on the single drive shaft with their signal outputs routed into the instrument room.

A digital tachometer was fastened to the line shaft. Seven magnet blocks were clamped to the shaft. These blocks passed a proximity sensor with the



7—Sectional view of tailshaft wiring

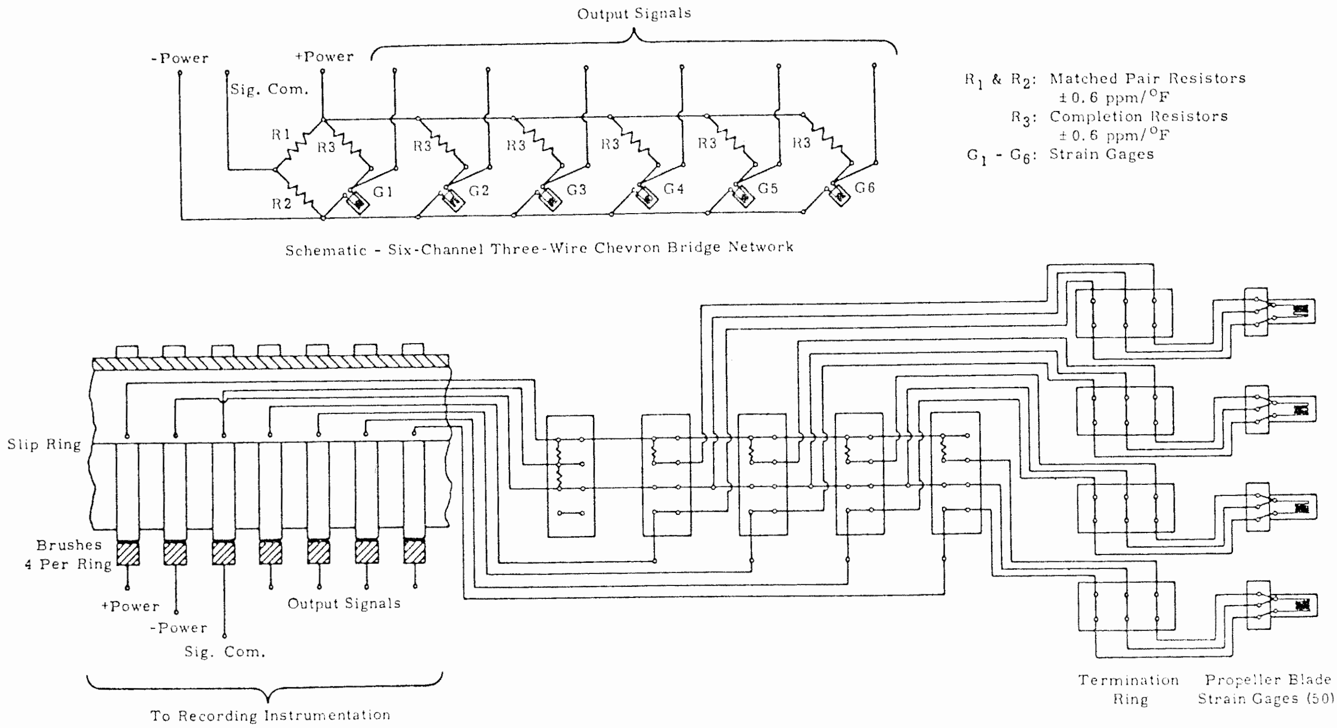


Fig. 8—Rotating three-wire chevron bridge networks

resulting impulse recorded on one channel of the tape recorder. The impulse signals were simultaneously displayed on a crystal-controlled digital frequency meter (DFM). Two blocks were spaced closer together than normal. The two-block passage impulse occurred when the No. 1 blade was straight up; i.e., at 0 deg.

All instrumentation channels were recorded on a wide-band FM magnetic-tape recorder. Signals were simultaneously recorded oscillographically (Fig. 12).

The instrument arrangement permitted direct record and simultaneous playback into the oscillograph for review of the recorded test data.

Sea Tests

The blade first-mode bending frequency was determined by pluck tests both in air and submerged. The S.S. "Michigan" then put to sea on April 14, 1970, sailing from Long Beach to San Francisco. Propeller stresses, thrust and torque were recorded for all gage sets during the night of April 14-15. The Littleton group concomitantly measured vibration displacements of the hull at several predetermined locations.

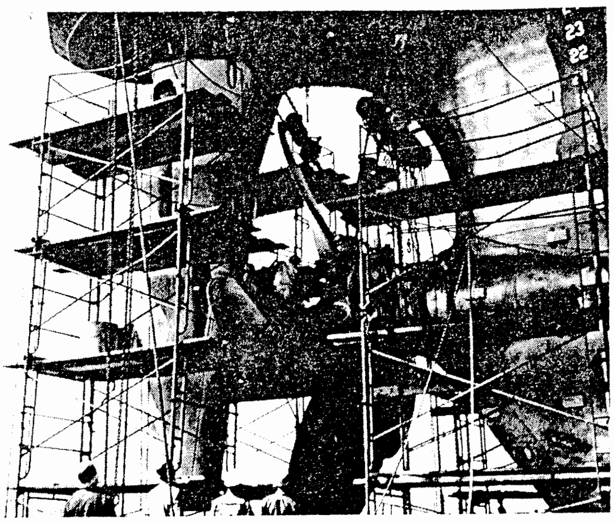


Fig. 9—Propeller installed on tailshaft

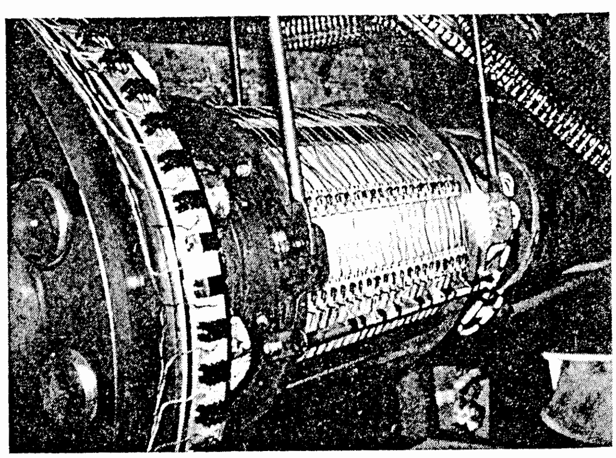


Fig. 10—Fourteen-channel slip ring

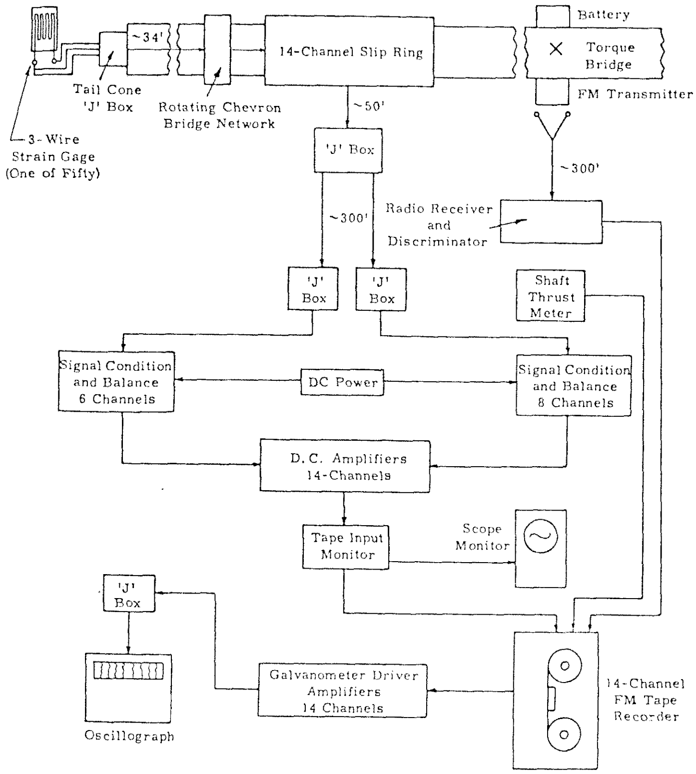


Fig. 11—Instrumentation block diagram

Upon arrival at San Francisco, a preliminary analysis of the propeller stress data was made.[7]

On April 20, 1970, the S.S. "Michigan" sailed from San Francisco for Japan and Hong Kong with the test crew and witnesses aboard. Periodically, data were recorded for different draft conditions and sea states. The tape recorder was operated at 1-7/8 ips; and 7,200 ft of 14-track data were recorded. The most significant records have been played back oscillographically, hand analyzed by and reported to the project sponsors.[6] The tests of main importance for future analysis are listed in Table 1. The test log is carried in Ref. 16. A paper,[8] based upon the final

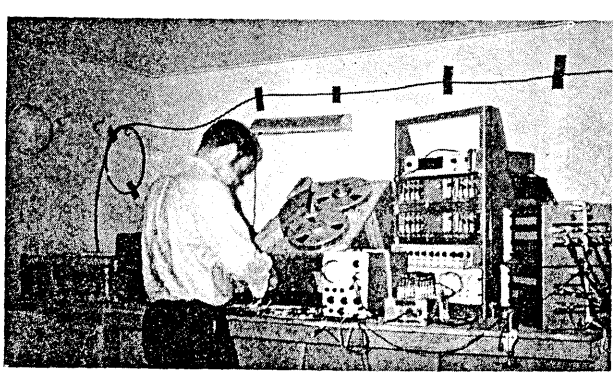


Fig. 12—Recording instruments aboard S.S. "Michigan"

TABLE 1—DATA TO BE DIGITIZED

Date	Leg of Voyage	No. of Tests	Gage Sets	Remarks
4-12-70	Los Angeles	2	A&D	Pluck Tests
4-14-70	Los Angeles to San Francisco	4	A, B, C, D	21' 11" Draft
4-20-70	San Francisco to Japan	4	A, B, C, D	28' 01" Draft
4-24-70	San Francisco to Japan	3	D, E, E	Even Keel
4-30-70	San Francisco to Japan	1	E	Even Keel, Rough Seas
5-06-70	Japan to Okinawa	1	E	Dyn. Only, Rough Seas
5-11-70	Okinawa to Subic Bay	1	D	Dyn. Only, 23' 05" Draft

TABLE 2—TENSION-SIDE MEAN-STRESS SUMMARY

RPM	T-1A	T-2A	T-33A	T-4A	T-5A	Average	$\sigma = f(N)^2$
52.3	2,520	-3,620*	2,540	4,330	4,970	3,590	2,530
61.4	3,300	-2,960*	3,180	4,720	4,970	4,040	3,490
65.9	4,090	-2,150*	3,180	5,100	5,440	4,450	4,020
72.5	4,660	-1,320*	4,440	6,290	7,080	5,620	4,870
81.5	6,220	+340*	5,930	7,860	7,790	6,940	6,150
87.8	6,810	+5,610	6,150	8,250	8,750	7,110	7,130
88.3	7,180	+6,080	5,710	8,250	8,970	7,240	7,220
92.1	7,570	+6,590	6,760	9,040	9,460	7,880	7,852
95.4	7,960	+7,420	7,400	9,430	9,940	8,430	8,420
96.6	8,740	+8,080	7,840	9,820	10,160	8,930	8,640
103.1	8,920	+8,400	8,260	10,410	11,100	9,420	9,840
104.9	9,720	+9,060	8,690	11,000	11,580	10,010	10,190
106.4	10,290	+8,890	9,110	11,800	12,290	10,480	10,480

NOTE:
*1. Not included in Average Values.
2. The modulus of elasticity used in calibrating the above stress values was 16.9×10^6 psi. $\mu = 0.30$.
3. 4-15-70; 0107-0319.
4. Displacement 9940 Tons; 21' 11" Mean Draft.
5. Mean stress $= \frac{\sigma_{min} + \sigma_{max}}{2}$

TABLE 3—COMPRESSION-SIDE MEAN-STRESS SUMMARY

RPM	C-1A	C-2A	C-33A	C-4A	C-5A	Average	$\sigma = f(N)^2$
52.3	-2,910	-1,390	-2,940	-1,990	-1,660	2,180	2,260
61.4	-3,790	-2,180	-3,380	-2,670	-3,040	3,010	3,110
65.9	-4,240	-2,990	-4,120	-3,500	-3,580	3,690	3,580
72.5	-4,920	-2,380	-4,850	-3,990	-3,870	4,000	4,340
81.5	-6,270	-4,380	-5,880	-5,150	-5,260	5,390	5,480
87.8	-6,910	-4,770	-6,470	-5,980	-6,640	6,150	6,360
88.3	-7,370	-5,370	-6,470	-6,830	-6,910	6,590	6,430
92.1	-7,810	-5,580	-7,490	-6,830	-6,910	6,920	7,000
95.4	-7,810	-6,560	-7,640	-7,820	-8,030	7,570	7,510
96.6	-8,260	-6,370	-7,350	-8,150	-8,570	7,740	7,700
103.1	-9,380	-6,960	-8,670	-9,140	-9,680	8,770	8,770
104.9	-10,060	-7,990	-9,550	-10,160	-10,220	9,600	9,080
106.4	-10,280	-8,550	-11,320	-10,490	-10,050	10,140	9,340

TABLE 4—TENSION-SIDE (PEAK ½ AMPLITUDE) DYNAMIC-STRESS SUMMARY

RPM	T-1A	T-2A	T-33A	T-4A	T-5A	Average
52.3	±960	±1000	±1070	±980	±830	±970
61.4	±1350	±1490	±1270	±1770	±1540	±1480
65.9	±1830	±1490	±1270	±1770	±1890	±1650
72.5	±1830	±1980	±1690	±2160	±1660	±1660
81.5	±3110	±3060	±3180	±3350	±3080	±3160
87.8	±3500	±3460	±3280	±3450	±3430	±3420
88.3	±3300	±3300	±3380	±3630	±3790	±3480
92.1	±3500	±3620	±3380	±3730	±3550	±3560
95.4	±3790	±3950	±4230	±4120	±3900	±4000
96.6	±4770	±4610	±4660	±4920	±4730	±4740
103.1	±5240	±5100	±5090	±5100	±4730	±5050
104.9	±5050	±4930	±5090	±4920	±4730	±4940
106.4	±5830	±5590	±6030	±6080	±5680	±5840

report, was presented by MARAD and Littleton in January of 1971, before the S.N.A.M.E. in Washington, D.C.

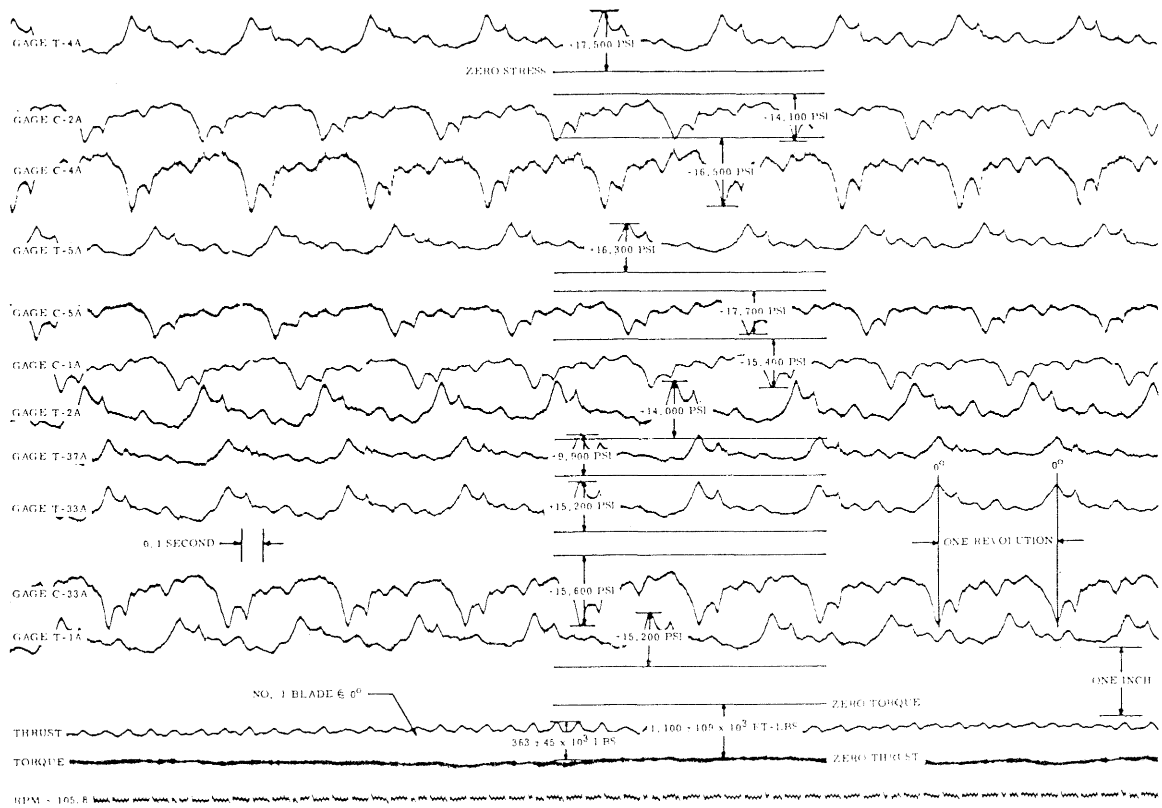


Fig. 13—Seven-second playback record 0148; set 'D' 0256 on 4-15-70, S.S. "Michigan"

TABLE 5—COMPRESSION-SIDE (PEAK ½ AMPLITUDE) DYNAMIC-STRESS SUMMARY

RPM	C-1A	C-2A	C-33A	C-4A	C-5A	Average
52.3	±900	±1100	±730	±1660	±1660	±1210
61.4	±1340	±1790	±1320	±1420	±1660	±1510
65.9	±1670	±1790	±1320	±1830	±1940	±2010
72.5	±1890	±2100	±2280	±2670	±2770	±2340
81.5	±3130	±3380	±3380	±4240	±4010	±3630
87.8	±3580	±3990	±3950	±4830	±4700	±4210
88.3	±3580	±3790	±3950	±5000	±4970	±4260
92.1	±4120	±4170	±3950	±4900	±5100	±4450
95.4	±4240	±4480	±4480	±3970	±5390	±4510
96.6	±4800	±4970	±4560	±6150	±6350	±5370
103.1	±4920	±5170	±5440	±6490	±6350	±5670
104.9	±5580	±5270	±5580	±7250	±6640	±6060
106.4	±6200	±5970	±6390	±8150	±7860	±6910

TABLE 6—VIBRATION ORDERS OBSERVED IN HOLDS 3 AND 4

RPM	Order Referenced to Shaft RPM
48	5
72	5
82	1
86	5
92	5 & 10
*96-108	5
102	2

NOTE: *Relatively high 5th order vibration noted in both holds during 'Z' maneuver.

Results

A brief analysis of the oscillographic data taken during the early morning of April 15, 1970, has been performed. During these tests, displacement was 9,940 tons with an aft draft of 21 ft 11 in. A sample oscillographic playback appears in Fig. 13. Propeller stresses at several locations are tabulated against rpm in Tables 2, 3, 4 and 5. Sample vibration data are carried in Table 6. The significant vibratory information obtained on the propulsion system is summarized below:

Propeller Pluck Tests

As determined from strain gages:

In air: fn = 26.7 Hz — Both independent of blade position;

In water: fn = 16.2 Hz — i.e., vertical or horizontal.

Shaft Torsional

Only torsional shaft oscillation was observed at ~33 rpm (as predicted by the General Electric Company).

Shafting Longitudinal

Tenth-order longitudinal vibrations (thrust) of ±40,000 lb were observed at 85 to 86 rpm. Fifth-order longitudinal vibrations were observed at 107 to 108 rpm of ±65,000 lb. During 'Z' maneuver,

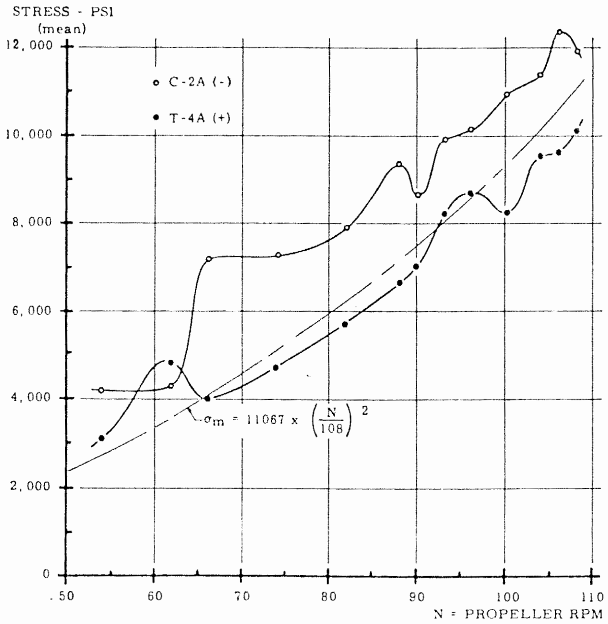


Fig. 14—Radial propeller mean stress 0.25 R, April 15, 1970

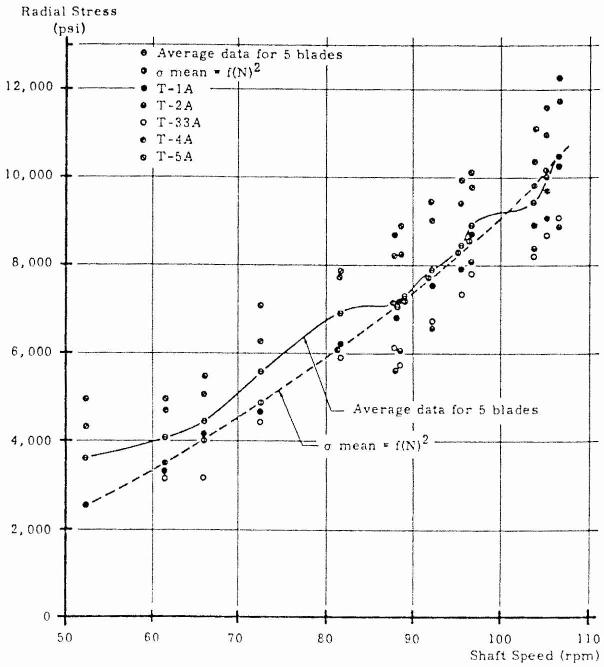


Fig. 16—Driving face steady-state component, 0.25 radius, 4-15-70; 0107-0319 hrs

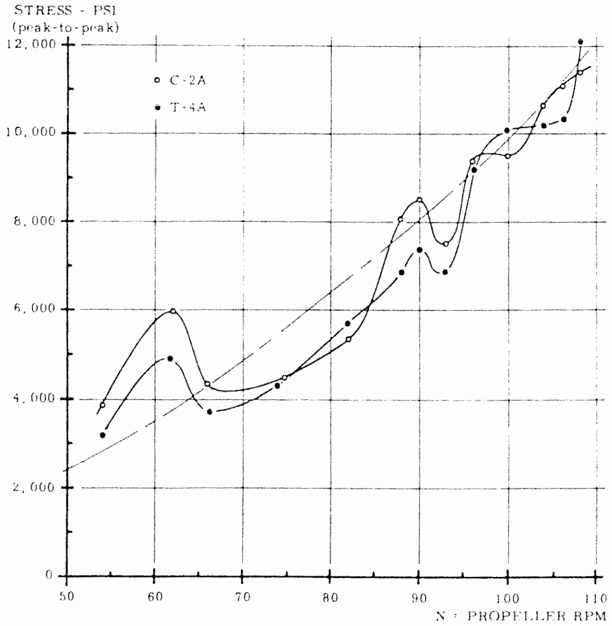


Fig. 15—Radial propeller alternating stress 0.25 R, April 15, 1970

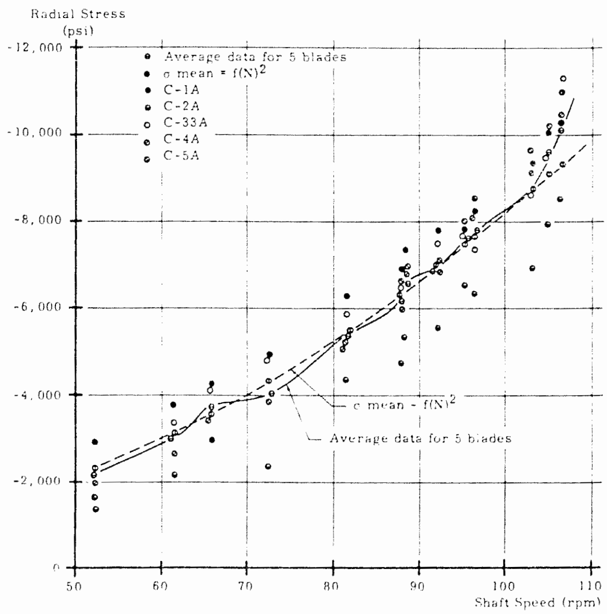


Fig. 17—Suction face steady-state component, 0.25

longitudinal thrust oscillations of ±90,000 lb were observed (fifth order) on a mean thrust of 302,000 lb.

Hold Vibration

Using accelerometers at the 1/3 points in Holds No. 3 and No. 4 on the ship's center line, records indicate significant vibration levels occurred at the speed frequencies tabulated in Table 6.

Plots of the tabulated data, both steady-state and vibratory, are carried in Figs. 14 and 15 from the preliminary data and in 16, 17, 18 and 19 from later analyses. The distribution of maximum stress on the driving surface of Blade No. 3 is shown on Figs. 20 and 21.

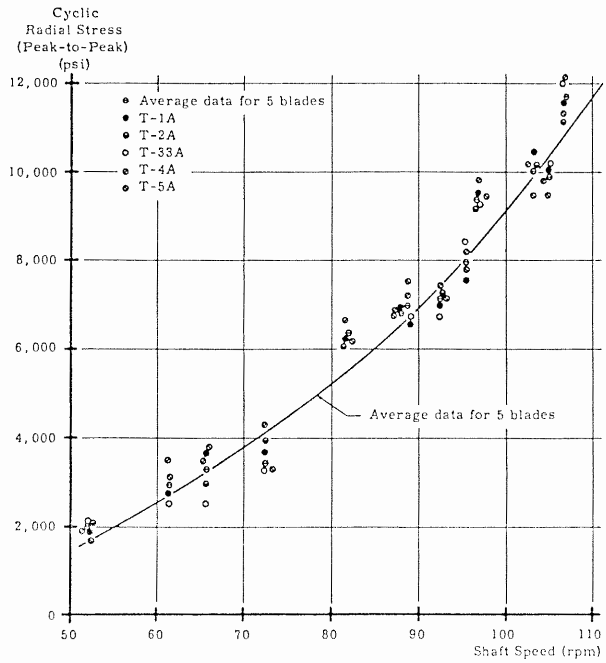


Fig. 18—Driving face cyclic-stress component, 0.25 radius

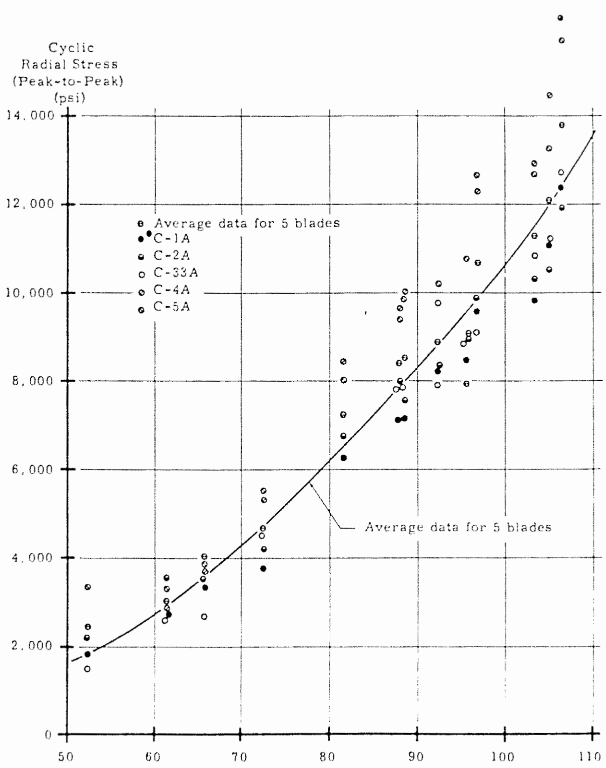


Fig. 19—Suction face cyclic-stress component, 0.25 radius

During one test on April 14, T-2A exhibited drift at the rate of 6μin./in./hr. However, the overall system drift in general was less than 6μin./in./hr. Gages T-2A and C-2A exhibited an unexpected shift in zero at low rpm; and, consequently, the mean data for T-2A below 82 rpm have not been used in the average-mean-stress plot of Fig. 16. Concomitantly, the C-2A plot in Fig. 14 may be somewhat high.

It should be noted that the chordwise and spanwise data presented in Figs. 20 and 21 were taken on different runs. The stress magnitudes coincident at the intersection of the 0.25 radius and the 50 percent chord planes are, therefore, not the same in magnitude.

The Littleton Research and Engineering Corporation and the Davidson Laboratory of the Stevens Institute of Technology have calculated the blade bending stress at the 0.3 radius for the "Michigan's" five-bladed Sharp propeller operating at 106 rpm with a ship speed of 23.4 knots and at a hull displacement of 18,050 tons. The theoretical plot of blade stress vs. position is shown in Fig. 22. The experimental blade stresses measured at Gage Position T-4A at 108 rpm, with a mean draft of 21 ft 11 in. and a displacement of 9,940 tons, have also been plotted on Fig. 22 for purposes of comparison.

Propeller-life Forecasts

From the data, it may be seen that the double-amplitude cyclic stresses are essentially the same magnitude as the steady-stress component at each gage location. Using the fatigue-strength values from Stone Manganese,[9] a modified Goodman diagram can be drawn for the Superston material. The fatigue

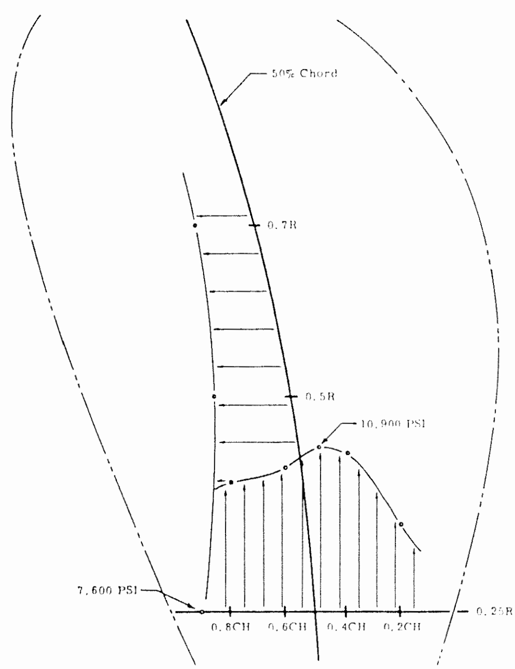


Fig. 20—Spanwise and chordwise radial mean stress at 102 rpm, driving face, blade No. 3

diagram is presented in Fig. 23 and is based upon 10^8 cycles of reversed stress and an exposure to 24 days of a salt-spray environment. The endurance limit used is reported to be characteristic of material removed for test from the root sections of propeller castings. The stress ranges calculated by Littleton Research and Engineering Corporation and the statistical average of those measured during the S.S. "Michigan" sea tests are carried in this diagram. Additionally, the BEL laboratory tests conducted on a virgin sample with the S.S. "Michigan" strain-gage overlay are also shown.

The Metal Properties Council of New York is sponsoring a research program[11] that will investigate the long-term fatigue strength of a number of propeller materials exposed to seawater at erosion rates com-

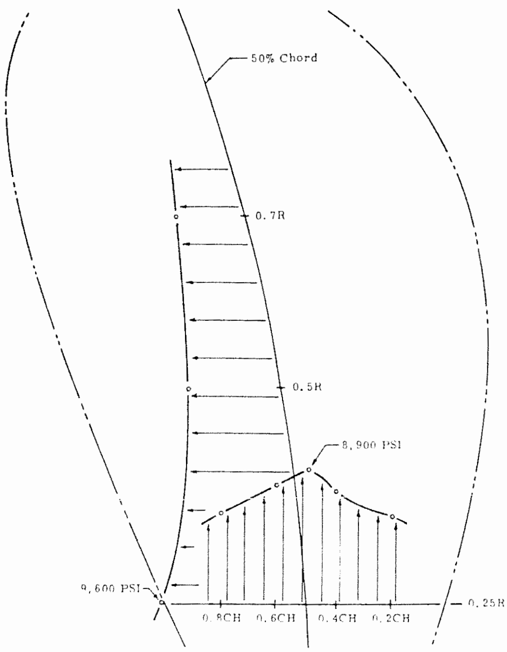


Fig. 21—Spanwise and chordwise radial peak-to-peak stress at 102 rpm, driving face, blade No. 3

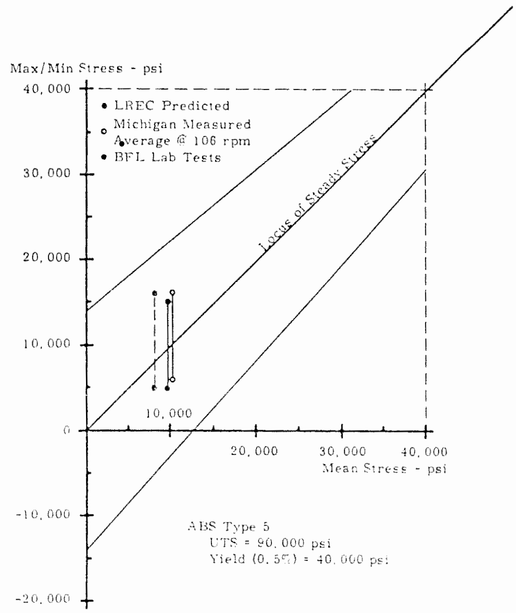


Fig. 23—Modified Goodman diagram for Superston 40 based upon Ref. 9 at 10^8 cycles, 24 days' salt spray

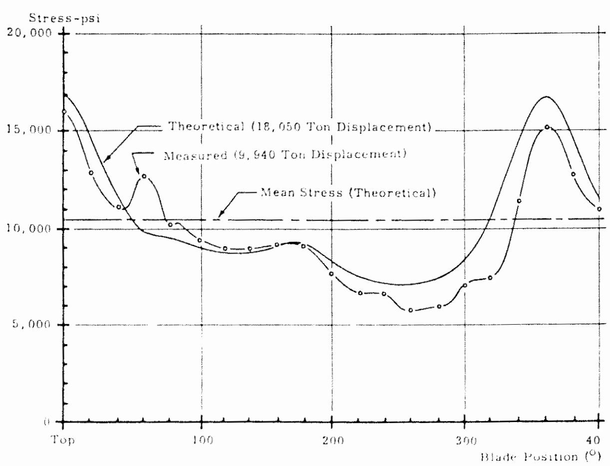


Fig. 22—Stress at gage T-4A measured 4-15-70 vs. theory

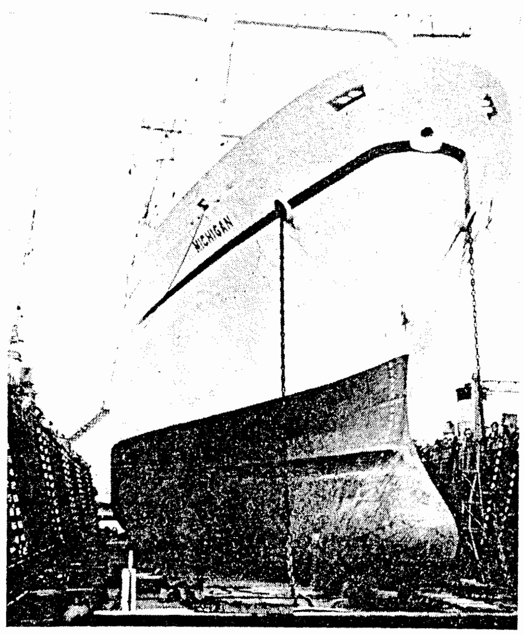


Fig. 24—S.S. "Michigan" in drydock, June 16, 1970

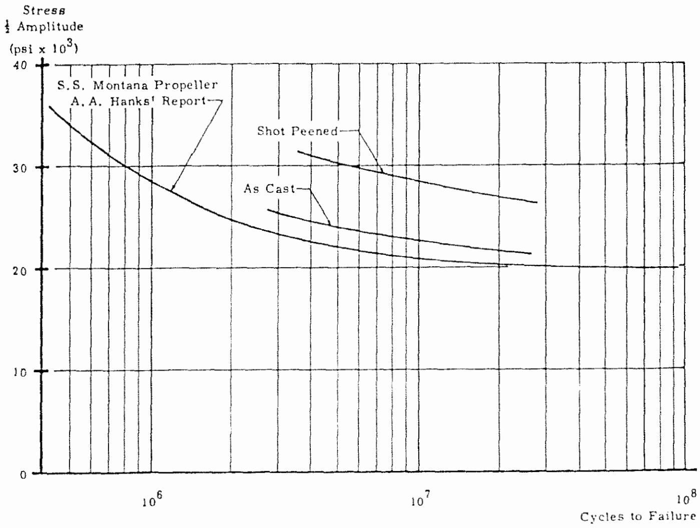


Fig. 25—S-N curves, "as cast" and shot-peened specimens in air

parable to those encountered in maritime operations. The effect of mean stress upon fatigue characteristics will also be investigated in this research program.

Conclusions

It may be concluded that the "Michigan" strain-gage attachment and overlay procedures may be utilized to measure propeller stresses for periods of several-million cycles and several months of operational exposure to seawater. The failure of the undersea system on the final leg of the trip, between the Philippines and Hong Kong, was due to wire erosion within the fairwater cap. Erosion resulted from the plating away of the ferrous surface on the adapter cap, just under the wiring, as it left the blades and entered the fairwater cavity. On future tests, depositing a surface of bronze on the adapter cap as it mates with the propeller should eliminate attack at this area.

Recommendations

It is recommended that the magnetic tapes carrying the sea-test strain-gage signals be digitized so that all of the data may be analyzed by computer techniques. Variations in torque[8] during the recording periods are responsible for appreciable scatter in the stress vs. rpm record. Computer techniques can be programmed to rationalize these variables and provide more-consistent data.

Deep cold rolling of the oval footprint on the driving surface of the blades, in the vicinity of the 0.25 radius, should be investigated. Experience with steel shafts and axles indicates that surface rolling can greatly extend the life of these members when subjected to repeated stress.[12,13] Brief experiments conducted by BEL,[14] shot peening Superston 40 plate bending specimens, indicate a considerable increase in endurance strength (see Fig. 25). The superficial depth of shot peening, unfortunately, eliminates the process from consideration as applied to propeller blades. The surface compression depth is quite shallow, 0.012 in. to 0.024 in., and would erode away or be penetrated by scratches, thereby exposing the tension layer beneath. To be effective, the compressed surface layer should extend 1/4 in. beneath the surface It is recommended[15] that surface prestressing be explored both in application and in technique for its final effect upon fatigue strength.

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