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M. V. Kalvik Propulsion System Full Scale Tests: Bearing Instrumentation and Data

No. 19

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ABSTRACT

Full scale propulsion tests were conducted on the Class 4 Icebreaker M.V. Kalvik. The program, executed by Fleet Technology Limited on behalf of Transport Canada, BeauDril Limited, and The Johnson Rubber Company included complete dynamic measurements of the starboard shafting system response, hull bending stresses, and structural accelerations, during icebreaking operations. Unique instrumentation was developed and employed to measure the reactions in each of the two salt water lubricated rubber stern tube bearings.

The instrumentation was installed when the M.V. Kalvik was drydocked from August 6, 1986, to August 21, 1986. Following the drydocking, the M.V. Kalvik proceeded to Resolute Bay, N.W.T., through the Prince of Wales Strait and Viscount Melville Sound. Extensive high speed digital recordings of the total shafting system response to severe ice milling were made during transiting and maneuvering in heavy second year and multi-year ice, from August 27, 1986 to September 13, 1986.

The successful execution of the project resulted in a detailed measurement of the forces on the intermediate shaft bearings and the stern tube bearings due to dynamic ice loads generated at the propeller.

INTRODUCTION

During the past ten years major advances have been made in improving the understanding of underlying phenomena important in the design of ice-going shipping. In Canada, much of this work has been motivated by the recognition of a growing need for much larger and more ice-capable ships than have ever before been constructed, if the goal of year-round navigation in the Canadian Arctic is to be achieved. It is understood that it will be difficult to extrapolate the design criteria for such vessels from the experience of existing, much smaller and less powerful shipping, without a sound appreciation of the basic physical processes involved.

One of the least understood aspects of the design of ice-going shipping is the problem of dynamic loading of propellers, shafting and shaft bearings due to ice blocks passing through the propellers. Current design practice has been largely heuristic: propellers, shafting and bearings have been progressively strengthened in response to the occurrence of damages encountered in operational icebreakers and ice-going merchant ships. Nonetheless, damage to propellers, shaft seals and shaft bearings continues to be a major problem, even in the newest and most powerful ice-going ships [5].

A program of full scale measurement was defined by the Transportation Development Centre to provide a base of engineering data that could be used to address the problems of shafting design for dynamic response to ice impact loads at the propellers. The project plan called for the detailed instrumentation of one of the two shaft lines of BeauDril's Class 4 Icebreaker the M.V. Kalvik, including the measurement of dynamic reactions in each of the two rubber stave stern tube bearings. The Johnson Rubber Company (manufacturer of the M.V. Kalvik's bearings) provided a new set of bearings and engineering assistance in modifying them for instrumentation and in installing them in the ship. BeauDril Limited provided logistic and engineering support for the execution of the project.

Of importance to this project was the fact that the M.V. Kalvik's propellers are unducted. As data had previously been obtained for a ducted propeller (M.V. Robert Lemeur) [3,4], the opportunity to observe the differences in ice loading occurring in an unducted installation was considered to be important. The M.V. Kalvik's history of successful operation for three years in the Beaufort Sea provided the assurance that she could be safely operated in the severe multi-year ice conditions which would yield the most important data from the point of view of ongoing development of propulsion design criteria.

This paper provides a description of the unique bearing instrumentation system installed on the M.V. Kalvik. Data collected on the stern tube bearings and the intermediate shaft bearings are presented along with some interpretation. A description of the data acquisition system used and the roundtrip voyage of the M.V. Kalvik from Tuktoyaktuk to Resolute Bay are also included.

THE M.V. KALVIK

Figure 1 illustrates the M.V. Kalvik and Table 1 lists the principal particulars of this vessel. The M.V. Kalvik is an icebreaking, anchor handling tug and supply vessel designed and built specifically to support BeauDril's Arctic oil exploration program in the Beaufort Sea. It was delivered from its builder, Burrard Yarrows Corporation of Vancouver, B.C. in July of 1983. At 6820 tons displacement and 23,200 BHP, the M.V. Kalvik is one of the largest and most powerful privately owned icebreakers in the world. The M.V. Kalvik incorporates a number of design features that have been pioneered by oil exploration companies in the Canadian Beaufort for their support fleet icebreakers.

The hull is largely constructed of flat or single curvature plate to reduce the time and cost of construction. The propulsion system consists of two LIPS ice strengthened controllable pitch propellers each capable of absorbing 11,000 HP. The propellers are set well out from the hull on long projecting bosses stiffened to the hull by a vertical skeg. The ship has a single rudder at the aft end of a deep centerline skeg which extends from the ship's keel line up to the underside of the stern between the two propellers. Each propeller is directly driven by a pair of Stork Werkspoor Medium Speed Diesel engines through a Lohman & Stohlterfoht 600:130 main reduction gear. Each engine engages its gear pinion through an air clutch that can be programmed to automatically open if a preset torque level is exceeded for a specified time interval (1- 3 seconds typically).

Table I Characterists of the Icebreaker Kalvik

Class:	ASPPR Ice Class 4
Length O.A.:	88.0 m
Length B.P.:	75.0 m
Breadth Extreme:	17.8 m
Draft Extreme:	8.3 m
Displacement:	6821 tonnes
BHP:	23,200
Propellers:	2 LIPS, N.V.C.P. 4 blades
Bollard Pull:	Over 220 tonnes
Service Speed:	
(two engines):	13.2 knots
(four engines):	15.5 knots

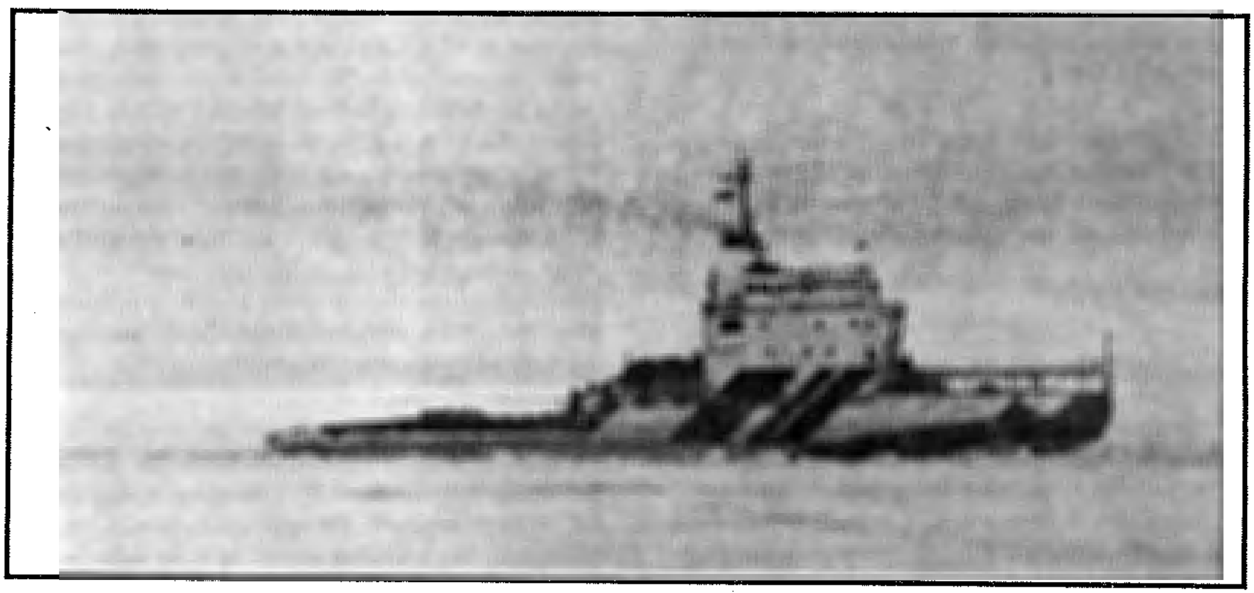

Fig. 1 The M.V. Kalvik

INSTRUMENTATION SYSTEM

The instrumentation system employed on the M.V. Kalvik starboard propulsion train was designed to provide complete engineering values on the response of the system to ice forces induced on the propeller blades and the hull during icebreaking operations. Figure 2 illustrates the location and the type of measurements recorded on the starboard propulsion system during the full scale tests on the M.V. Kalvik. Standard strain gauge techniques were used for all strain measurements. Rotating machinery signals (shaft thrust, shaft bending, etc.) were transmitted via standard FM telemetry systems. Other machinery measurements were recorded from the existing outputs on the vessel. The instrumentation system designed and installed to measure the direct loads on the stern tube bearing and the reactions of the three intermediate shaft bearings is described below.

Intermediate Bearing Measurement

There are three intermediate shaft bearings as illustrated in Figure 2, and their particulars are the following:

Length	0.62 m	(2.04 ft.)
Diameter	0.47 m	(1.54 ft.)
Max Continuous Rating	320 kN	(32.6 tonnes)

Measured Static Loads:		
Aft. Int. Bearing	110. kN	(11.2 tonnes)
Aft. Flywheel Bearing	67. kN	(6.87 tonnes)
Fwd. Flywheel Bearing	124. kN	(12.65 tonnes)

A standard temperature compensated half bridge strain gauge configuration was mounted on both sides of the supporting structure of each bearing. The output of these gauges were recorded simultaneously with a known applied load, which provided a calibration factor for calculating the vertical bearing load from the strain gauge readings. Figure 3 illustrates the location of the gauges and the calibration procedure used. The transverse load on the bearing was calculated from the difference in the output of the two gauges, and equilibrium criteria. The resultant load and the direction in which it acts was computed from the vector sum of the vertical and transverse loads.

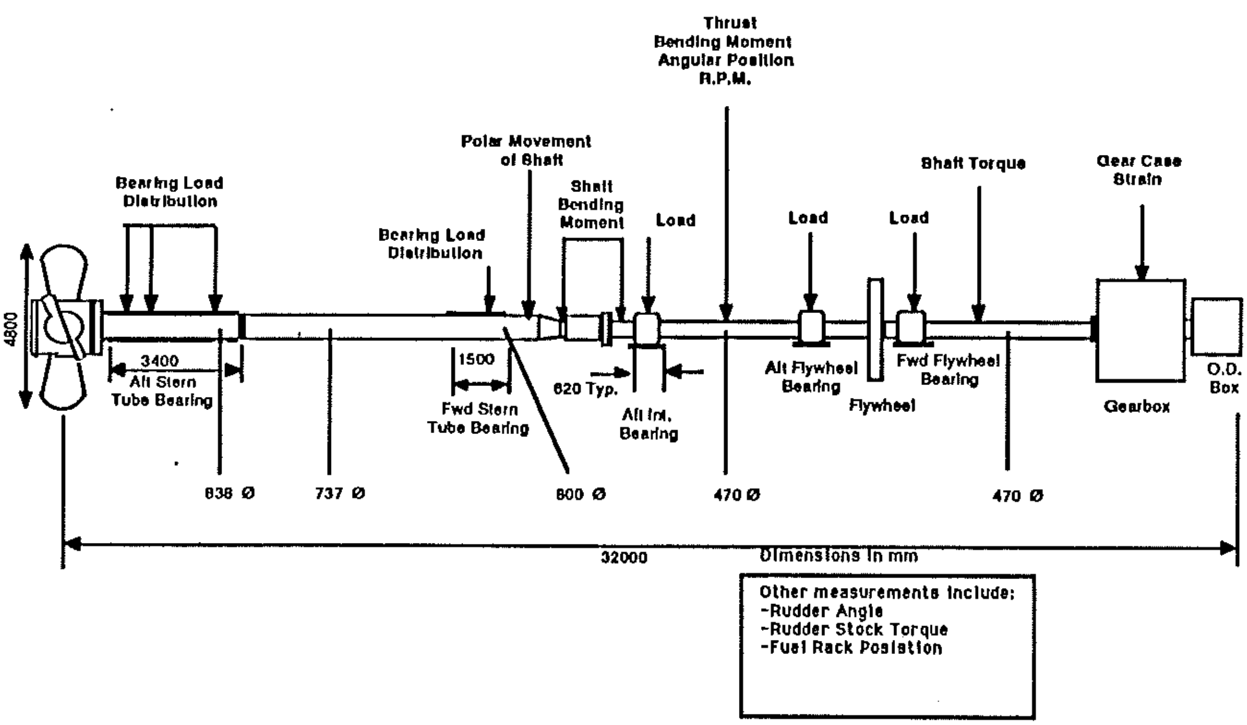

Fig. 2 Starboard Shafting Instrumentation

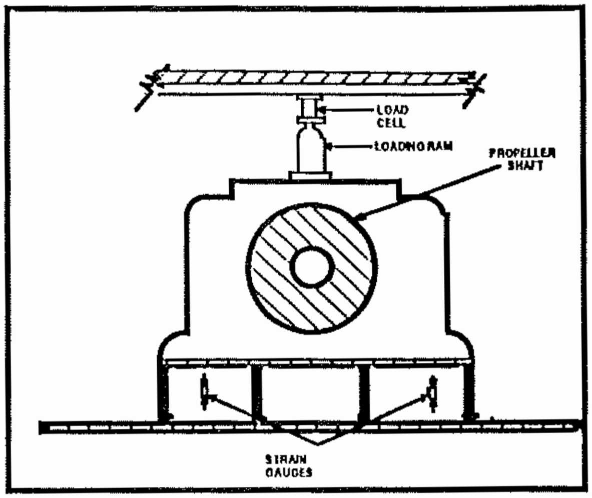

Fig. 3 Shaft Intermediate Bearing Instrumentation and Calibration

Stern Tube Bearing Instrumentation

To design safer and more reliable stern tube bearings it is essential to determine the magnitude and distribution of dynamic loads on the bearings while icebreaking. To the authors knowledge, due to the difficulty of instrumenting stern tube bearings, the direct measurements of full scale bearing forces have not been undertaken before. This section describes the unique instrumentation employed to determine the dynamic loads on the starboard stern tube bearings of the M.V. Kalvik.

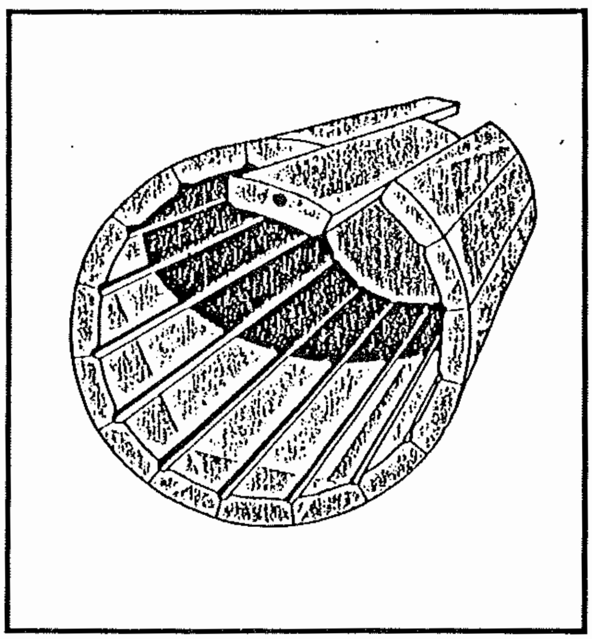

Fig. 4 Three Dimensional View of a Stern Tube Bearing

A three dimensional representation of a stern tube bearing is illustrated in Figure 4. The stern tube bearings of the M.V. Kalvik are manufactured by The Johnson Rubber Company of Middlefield, Ohio. These bearings have the following particulars:

Type of Material	Rubber
Forward Bearing:	
Length (1 section)	1524 mm
Diameter (12 staves spaced equally)	802 mm
Aft Bearing:	
Length (2 sections each 1.7 m long)	3400 mm
Diameter (12 staves spaced equally)	840 mm
Thickness	50 mm
Water Groove Depth (12 grooves)	12 mm
Water Flow Rate	80 gpm
Static Load	276 kPa (40 psi)
Radial Clearance	0.75 - 1.1 mm

Strain Gauge Installation in Rubber Bearing Staves

Strain gauges were installed in the rubber bearing staves to measure compressive loads as shown in Figure 5. A recess was machined in the back side of the rubber stave, and a groove was machined from this recess to the stave end. In the recess, a vertical wall parallel to the end of the stave was prepared for a strain gauge. The gauge was mounted on to the wall and clamped until dry. Wires were attached to the strain gauge and run out along the machined groove. The recess and a portion of the groove was then filled and sealed with special waterproofing material. The remainder of the groove was filled with a rubber compound.

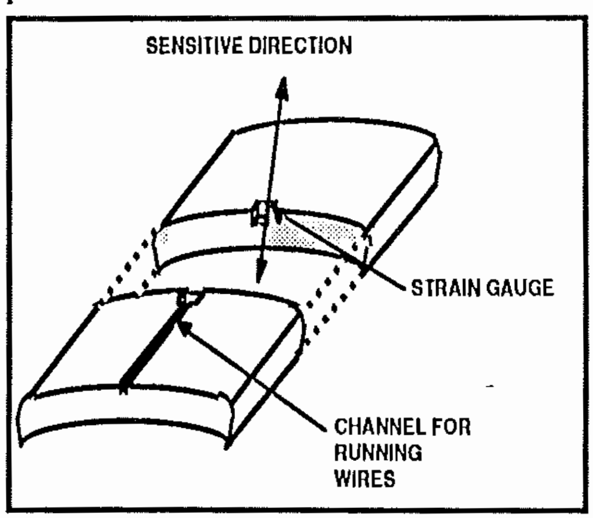

Fig. 5 Strain Gauge in Rubber Bearing Stave

Installation of Instrumented Rubber Bearing Staves in Stern Tube

The installation of the instrumented bearings was performed while the ship was in drydock, after the propeller shaft and the old bearings had been removed. Figure 8 shows a section of the stern tube with the cabling arrangement for the stern tube bearings. Holes were drilled in the brass retaining ring of the forward and aft bearings to permit the wires to pass from the backside of the rubber staves into the stern tube. To run the wires out of the stern tube and into the ship, a hole was cut through the stern tube at the 12 o'clock position inside a watertight compartment. A collar was welded into the hole to provide stress relief and a smooth surface to pass wires over. A steel tube was welded on the outside around the hole in the stern tube to form a watertight coffer dam. A steel channel was welded longitudinally to the top of the stern tube, extending from the forward stern tube bearing to the aft bearing and intersecting the stern tube penetration, to act as a collecting duct for wires leading out of the bearings.

The stern tube was cleaned in preparation for the installation of the instrumented rubber bearings. The forward stern tube bearing was installed first beginning with the bottom stave, and adding staves circumferentially on either side of the bottom one. The strain gauge wires were fed through the holes in the compression ring as each stave was installed.
A plywood cylinder was placed in the tube to act as temporary support for the upper half of the bearing. The last of the twelve staves to be installed was the top dead centre one. This stave had to be physically hammered in with a large sledge hammer to provide the circumferential compression required in the bearing. The compression ring was installed and provided a longitudinal compression of 1.25%.

The installation of the aft stern tube bearing was more difficult. The aft stern tube bearing is composed of twenty-four staves, two longitudinal sets of twelve staves circumferentially spaced. The wires from the aft half of the bearing were set in the groove in the forward half of the bearing, the groove was filled with a rubber compound and the forward and aft staves were slid into place in unison. The wires coming from each bearing were bundled and fed through the steel channel and out through the coffer dam. The coffer dam was filled with a potting compound and sealed. The individual strain gauges lead wires were then connected to the data acquisition system.

The system proved reliable with only a few of the highly loaded gauges failing from fatigue. The area around the penetration of the stern tube was inspected after the tests and no leakage or any sign of damage could be found.

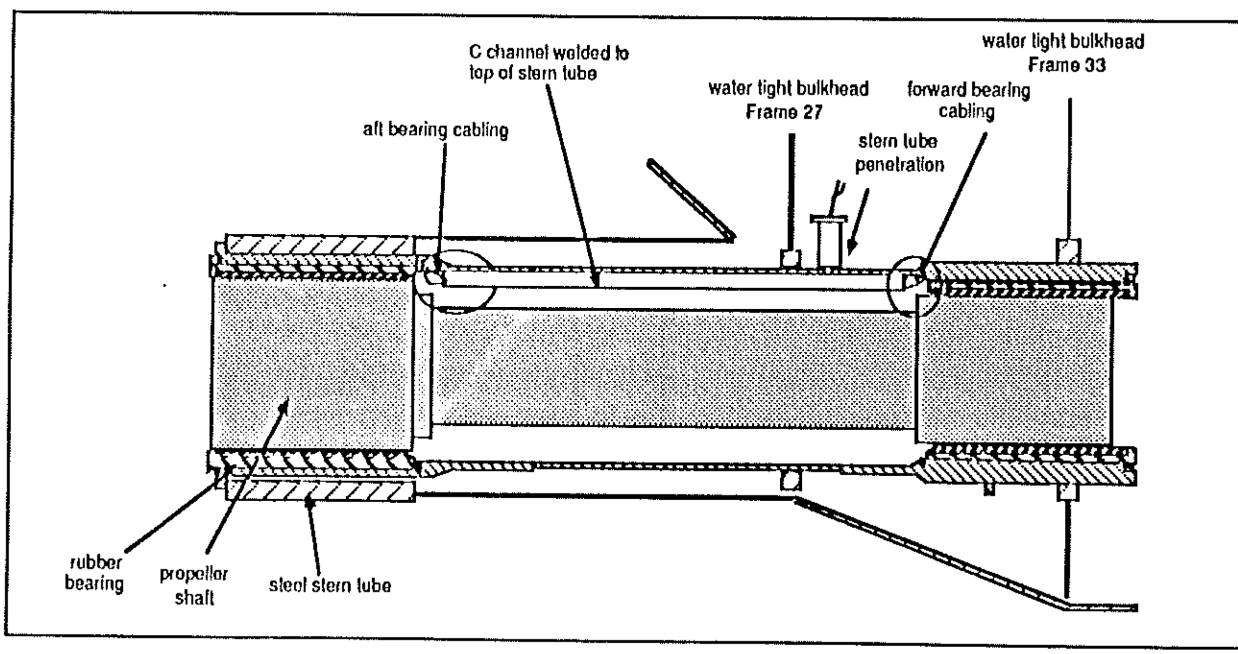

Fig. 8 Longitudinal Cross Section Through Stern Tube

Calibration

In the laboratory it was found that the strain measured by the strain gauge was less than that of the actual strain occurring. This can be attributed to the effect of milling a recess for the strain gauge and then filling the recess with a potting compound of reduced stiffness which changes the stress field in the sample. Despite the fact the strain measured by the bearing strain gauge does not correlate directly with actual strain, the pressure vs. strain curves does show constant, repeatable characteristics for the same loading rate and longitudinal compression.

Calibration tests were also performed in the installed instrumented stern tube bearing. The bearing was compressed using a hydraulic ram with a load cell in series with it, as illustrated in Figure 9. The applied load and the output of the strain gauge were measured simultaneously. Figure 10 shows a plot of applied pressure vs measured strain.

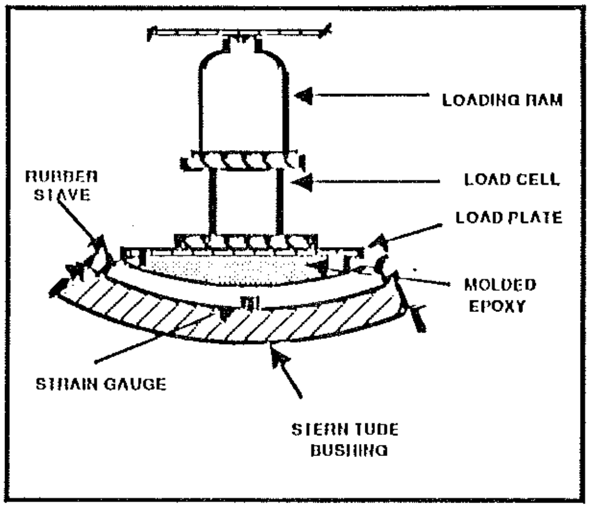

Fig. 9 Calibration of Instrumented Stern Tube Bearing Stave

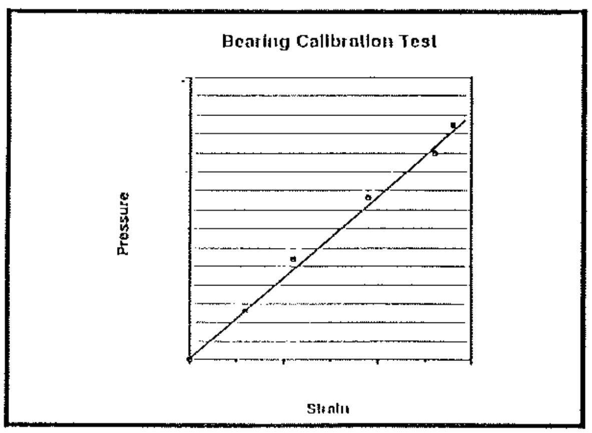

Fig. 10 Pressure vs Strain for Stern Tube Bearing Instrumentation

DATA ACQUISITION

The digital data acquisition system used is illustrated in block form in Figure 11. This system has been developed over the last four years from previous full scale tests conducted by Fleet Technology Limited personnel and uses the most advanced and reliable field equipment available. Each signal was continuously sampled at 500 readings/second. An ice/propeller interaction event duration can vary from 0.5 to 6.0 seconds but is typically 3 to 4 seconds long. An event was detected by the excursion of selected readings by a specified amount (e.g., shaft torque greater than 125% of MCR torque). Data from each event consisting of a time window 6.25 seconds long, 3 seconds before the event was detected and 3.25 seconds after, was stored on a magnetic floppy disk (greater than 190,000 readings per event). Software routines were written for automatic event detection and data storage. Table II lists the signals recorded during the full scale tests.

TABLE II SIGNALS RECORDED

NO.	DESCRIPTION
33	Strain Gauges on Rubber Staves
4	Hull Bending Gauges
2	Gear Case Bending Gauges
6	Intermediate Bearing Gauges
3	Shaft Bending Gauges
1	Shaft Torque Gauge
1	Shaft Thrust Gauge
1	Shaft Speed and Orientation
1	Propeller Pitch
1	Rudder Angle
1	Rudder Torque
3	Hull Accelerations
1	Ship Speed
2	Shaft Movement Probes
1	Engine Fuel Rack
61	Total

Laboratory Testing

Two periods of laboratory testing on an instrumented sample of the rubber bearing were conducted. The first session was conducted prior to the field tests and the second after returning from the full scale tests. The following were the objectives of the first session of laboratory testing:

1) Establish the durability of the installed-strain gauge under expected radial and longitudinal compression.
2) Ensure that the gauge could withstand high dynamic loads and remain water-proof under a prolonged pressurized water environment (as would be the case in the actual bearing to be installed).
3) Examine the relationship between the measured response of the gauge and applied load on the bearing surface.

The gauge responded well to dynamic loads with good repeatability in results. It was not damaged when exposed to water or under severe longitudinal compression. Therefore, the strain gauge installation technique was approved and employed on a new set of bearings, provided by The Johnson Rubber Company. These were installed in the vessel during its drydock in August 1986.

After the full scale tests were conducted a second set of laboratory tests were performed on the test piece of rubber bearing, over the full range of loading as was observed in the full scale tests, to determine the following:

1) The effect of longitudinal compression (confinement) and loading rate on the material properties of the bearing.
2) Evaluate the calibration factors.

A test apparatus for simulating the condition of the bearing in the vessel was constructed. A bearing test piece was confined circumferentially and compressed longitudinally as was the bearing in the vessel. Loading rates and the maximum measured strain were determined from the full scale data collected. Similar loads were applied to the test piece of bearing material and the corresponding response from the gauge was measured. It was determined that the bearing material had an elastic modulus that is highly dependent on longitudinal compression and weakly dependent on loading rate. Under the same loading rate the elastic modulus of the bearing increased with longitudinal compression. The modulus changes from 74 MPa at 0% longitudinal compression to 98 MPa at 2.2% longitudinal compression for a loading rate of 6.8 MPa/sec., as shown in Figure 6. The relationship between elastic modulus and loading rate is also indicated on this figure. At a longitudinal compression of 2.2%, the elastic modulus increases from 98 MPa to 115 MPa (15%) when the loading rate is increased from 6.8 MPa/sec. to 136 MPa/sec (2000%). For the loading rates measured and the existing longitudinal compression in the bearings during the full scale tests the following was determined:

Fwd bearing Elastic Modulus=105 MPa. (15,200 psi)
Aft bearing Elastic Modulus= 100 MPa. (14,500 psi)

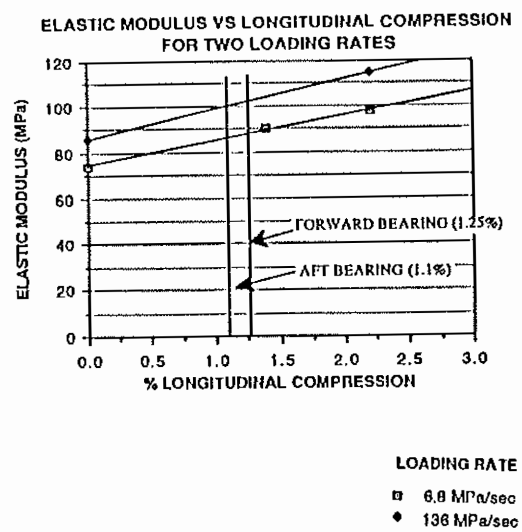

Fig. 6 Elastic Modulus vs Longitudinal Compression for Rubber Bearing Material

A total of 33 strain gauges were installed in the back side of a new set of rubber stave bearings. The gauges were distributed in the bearings such that the circumferential and longitudinal distribution of load could be estimated. They were arranged in three rings, 0.47 m and 2.17 m from the aft end in the aftermost bearing, and 0.43 m from the forward end of the forward bearing. Additional gauges were also placed at the 5 o'clock position along the bearing to provide further information on the distribution of loading in the longitudinal direction. Gauges were not placed in the 11, 12, or 1 o'clock staves due to installation restrictions. The strain gauge layout is shown in Figure 7.

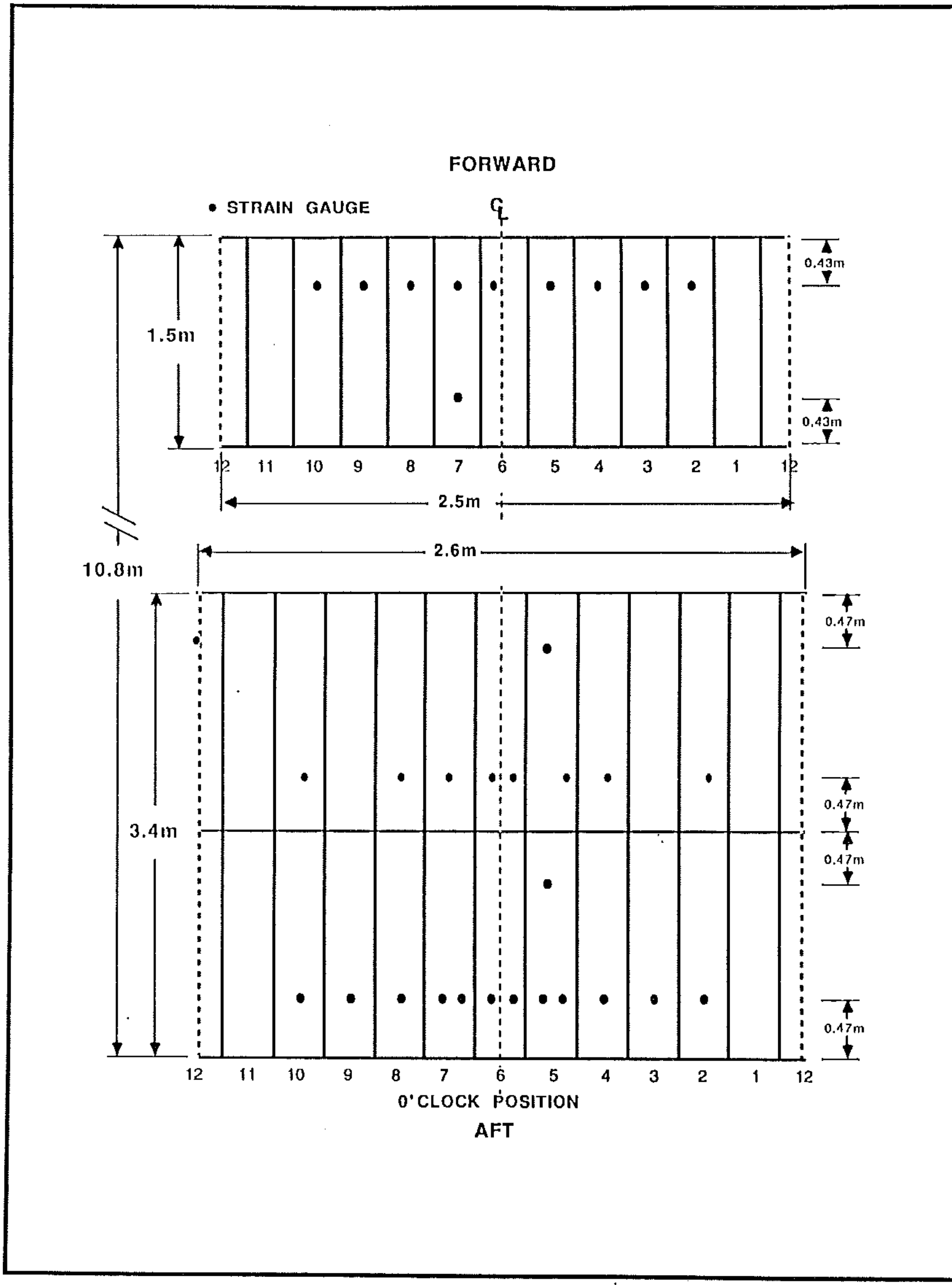

Fig. 7 Projection of Stern Tube Bearing Showing Strain Gauge Locations

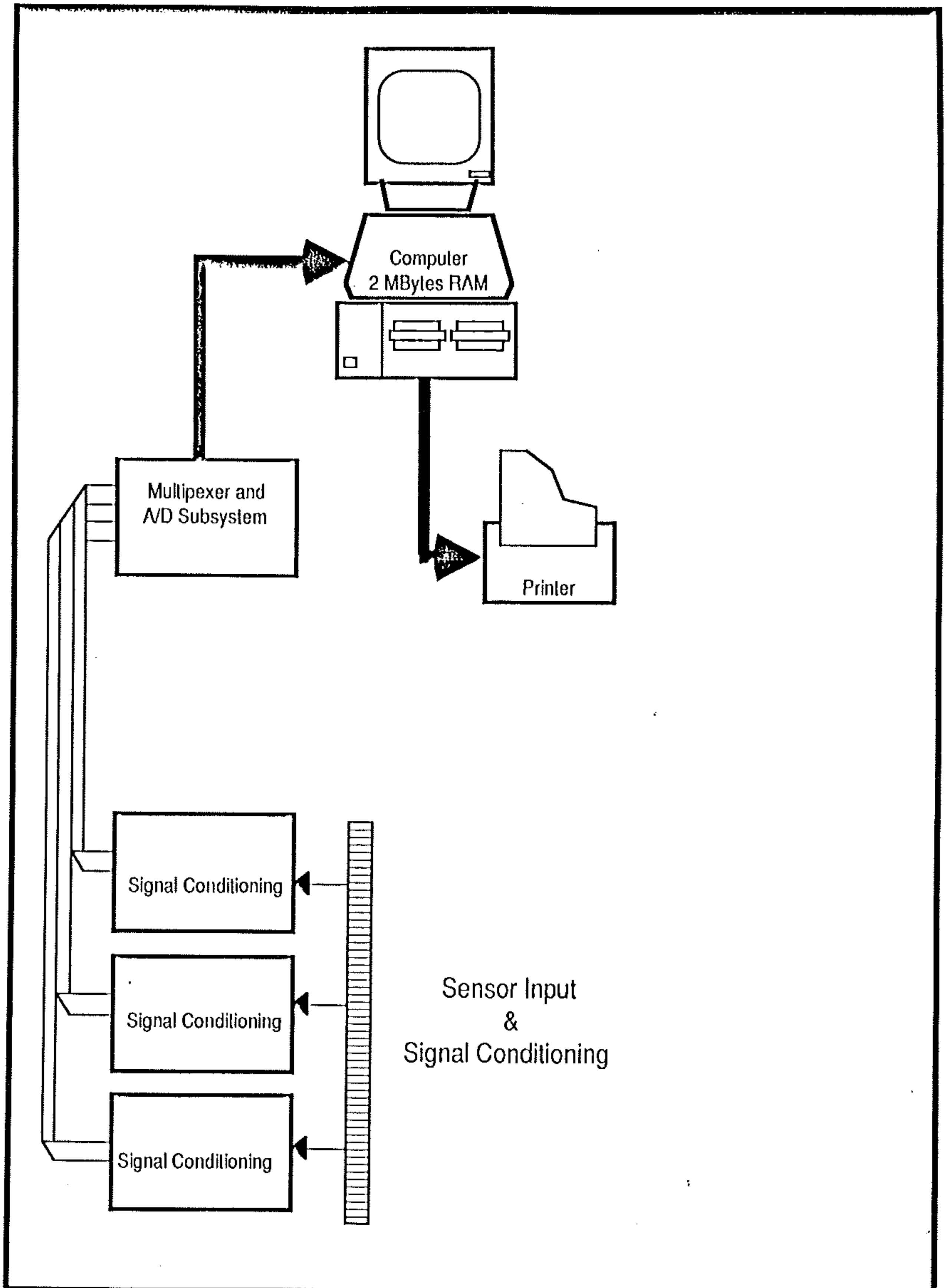

Fig. 11 Data Acquisition System Block Diagram

VOYAGE OF THE M.V. KALVIK

The M.V. Kalvik was drydocked in Arctic Transportation Limited's floating dock, at Tuktoyaktuk, N.W.T. on August 6, 1986. The ship maintenance was completed by August 21, 1986, and the vessel was chartered by the Canadian Coast Guard on August 26, 1986. The M.V. Kalvik proceeded to Resolute Bay, N.W.T., through the Prince of Wales Strait as shown in Figure 12. Heavy ice conditions were encountered as the vessel entered Viscount Melville Sound. Repeated ramming through multi-year ice and heavy second year ice was required. Figure 13 illustrates a milled ice block.

Upon arriving in Resolute Bay (August 31, 1986), the M.V. Kalvik joined the CCGS John A. MacDonald. Performance tests on the two vessels were conducted from September 2, 1986 to September 13, 1986 [10]. The vessels operated in second year and multi-year ice floes. Open water trials were conducted on September 13, 1986. The M.V. Kalvik returned to the Beaufort Sea near Tuktoyaktuk, N.W.T. on September 15, 1986. The John A. MacDonald returned to Resolute Bay, N.W.T. to continue its duties for the Canadian Coast Guard. Figure 14 shows the voyage and trial locations from September 3 to September 13, 1986 and the return route to Tuktoyaktuk, N.W.T.

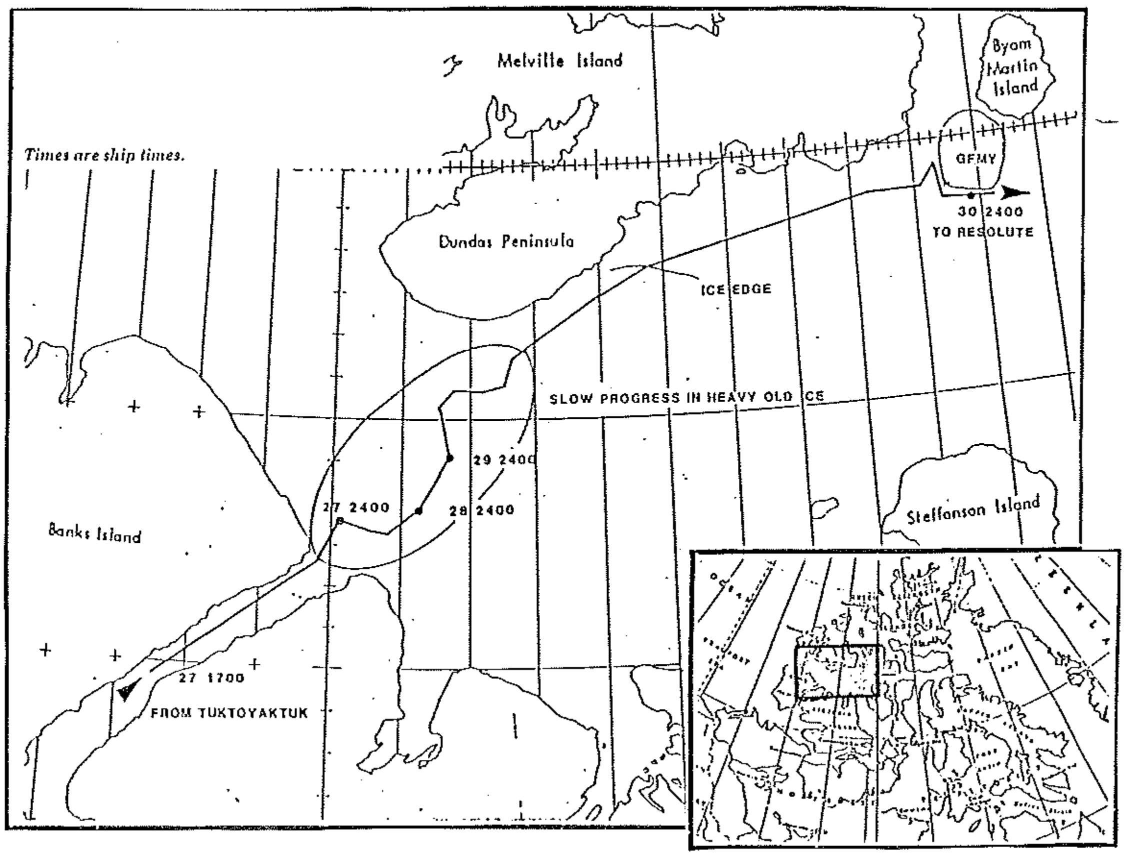

Fig. 12 Voyage of the M.V.. Kalvik in Test Area and Southbound to Tuktoyaktuk

Fig. 13 Milled Ice Block

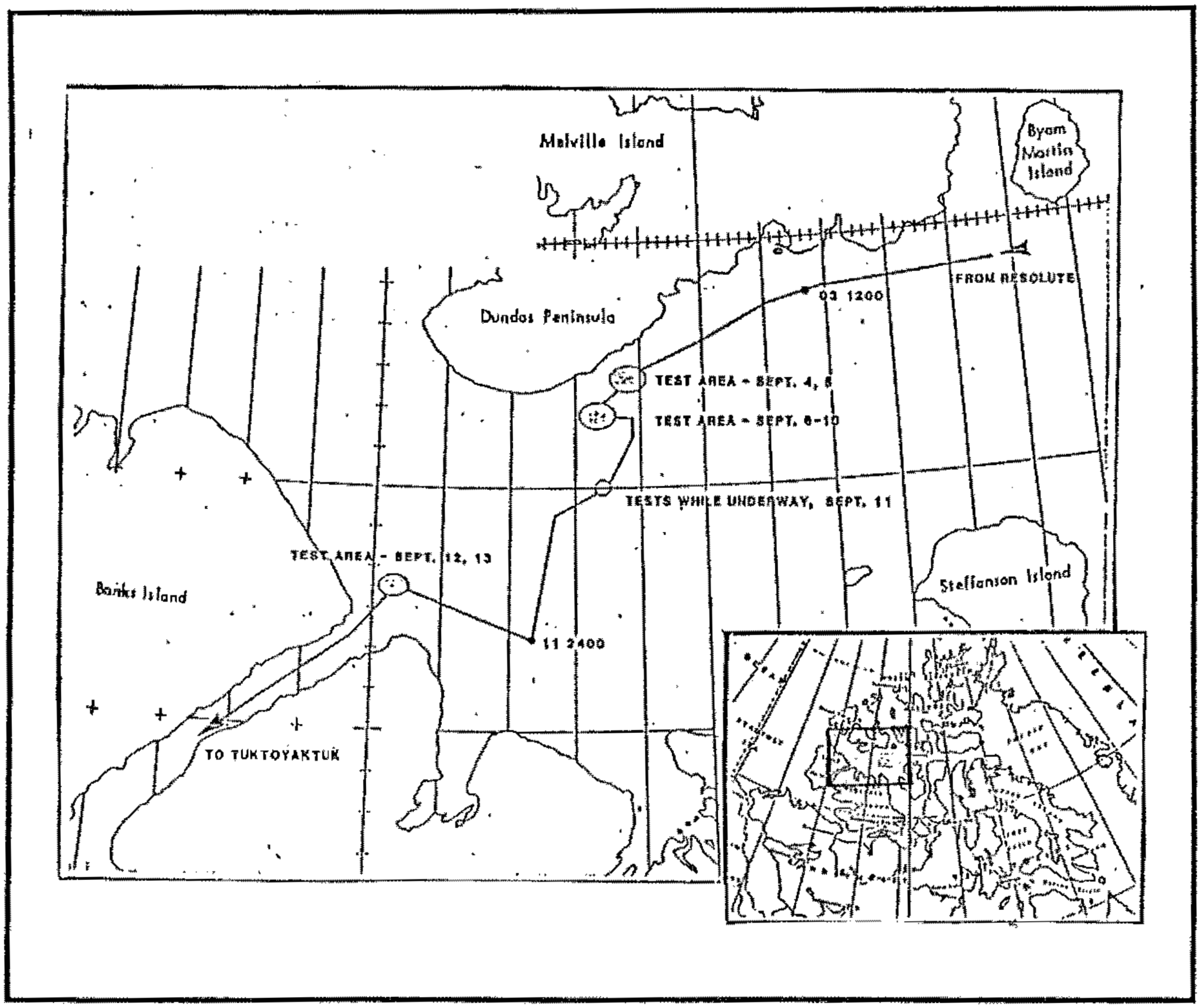

Fig. 14 Voyage of the M.V. Kalvik in Test Area and Southbound to Tuktoyaktuk

DATA COLLECTION AND INTERPRETATION

Stern Tube Bearing Data

One of the main objectives of this project was to obtain direct measurement of the magnitude and distribution of the reactions of the the stern tube bearings during ice propeller interactions. The following steps outline the procedure and assumptions made in calculating the total reaction of the bearing from strains measured at the 33 locations.

1) BASE LINE CALCULATION - A base line value was determined to calculate the absolute compressive strain in the bearing (static plus the dynamic load). During an ice milling event the shaft moves around the bearing such that each gauge location is unloaded at some point in time during the event. This strain gauge output at this time was defined as being the base line value. This value was subtracted from each reading during an ice milling event to give the absolute compressive strain.

2) LOCAL PRESSURE - The strain gauges placed in the bearing stave measure the local compressive strain occurring instantaneously at the gauge location. The strain reading is converted to local pressure on the surface of the stave using the calibration factor determined from work in the lab and field.

3) LOCAL REACTION - The local pressure on a single stave is converted to a reaction per unit length at that gauge location by multiplying the pressure measured by the circumferential arc length of the individual stave that is capable of carrying a hydrodynamic wedge (Figure 15). For staves with one gauge the arc length assumed is 0.16 m, and for staves with two gauges side by side an arc length of 0.8 m is assumed. This reaction acts radially through the stave. By knowing the measured reaction magnitude and the staves angular location, the components of the reaction in the vertical and horizontal direction can be calculated (Figure 16).

4) TOTAL VERTICAL AND HORIZONTAL REACTION OF THE FORWARD BEARING - As the reaction at only one longitudinal location is known, a constant reaction per unit length is assumed longitudinally. The total reaction is calculated by multiplying the reaction at the one known location by the length of the forward bearing (1.5 m).

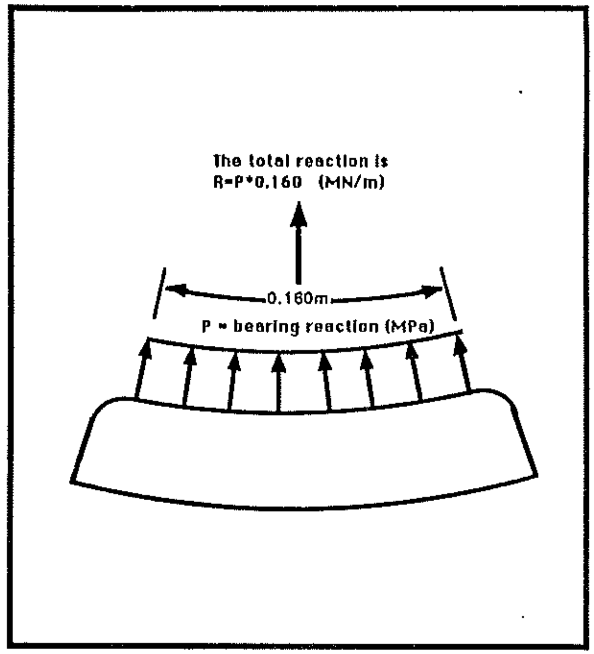

Fig. 15 Calculation of Reaction at One Gauge Location

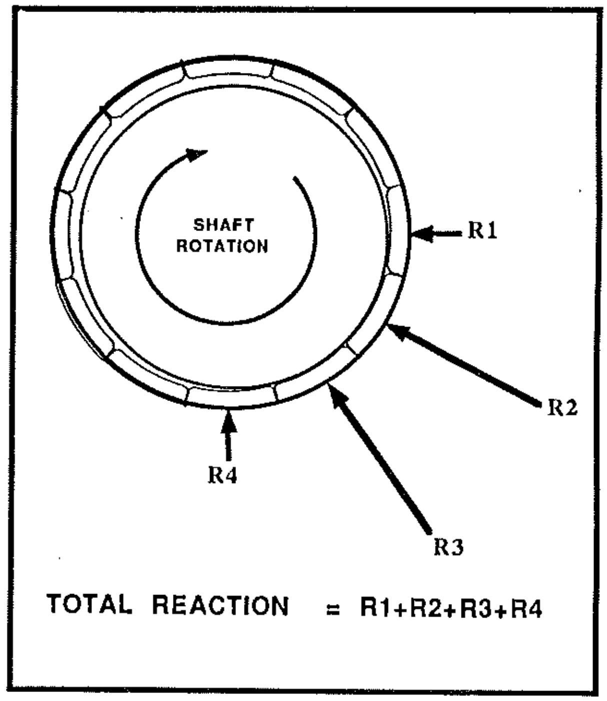

Fig. 16 Illustration of Calculation of Reaction at a Transverse Location

5) TOTAL VERTICAL REACTIONS OF AFT BEARING - In the aft bearing the reaction per unit length of the bearing is known at two longitudinal locations. A linear distribution through these two points is assumed (Figure 17). In cases when the extrapolated distribution predicts a negative reaction, a zero reaction per unit length is specified over that portion of the bearing, as the shaft does not significantly contact the top section of the bearing due to high clearances. The total transverse reaction is equal to the integration of this distribution over the defined contact length.

6) TOTAL TRANSVERSE REACTION OF THE AFT BEARING - The transverse reaction per unit length is known at two locations and a linear distribution is assumed through these two points. If the shaft is not in contact with the bearing along the full length, as indicated by a zero vertical reaction, then the transverse force along this section is defined to be zero, due to bearing and shaft geometric constraints. The total transverse reaction is equal to the integration of this distribution over the defined contact length (Figure 18).

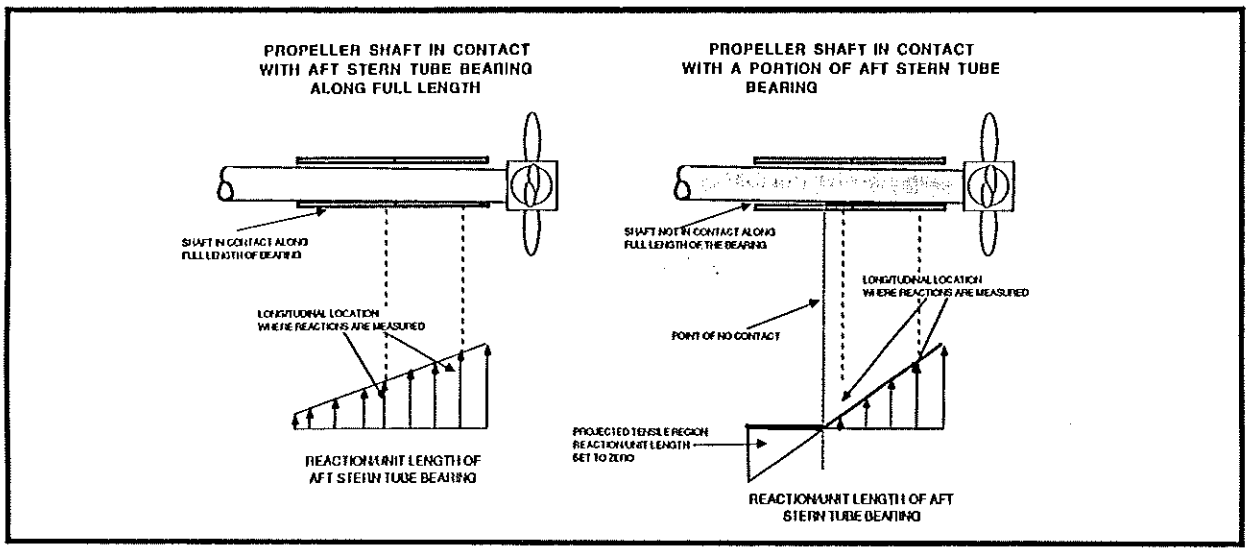

Fig. 17 Longitudinal Distribution of Vertical Reaction Along Aft Stern Tube Bearing

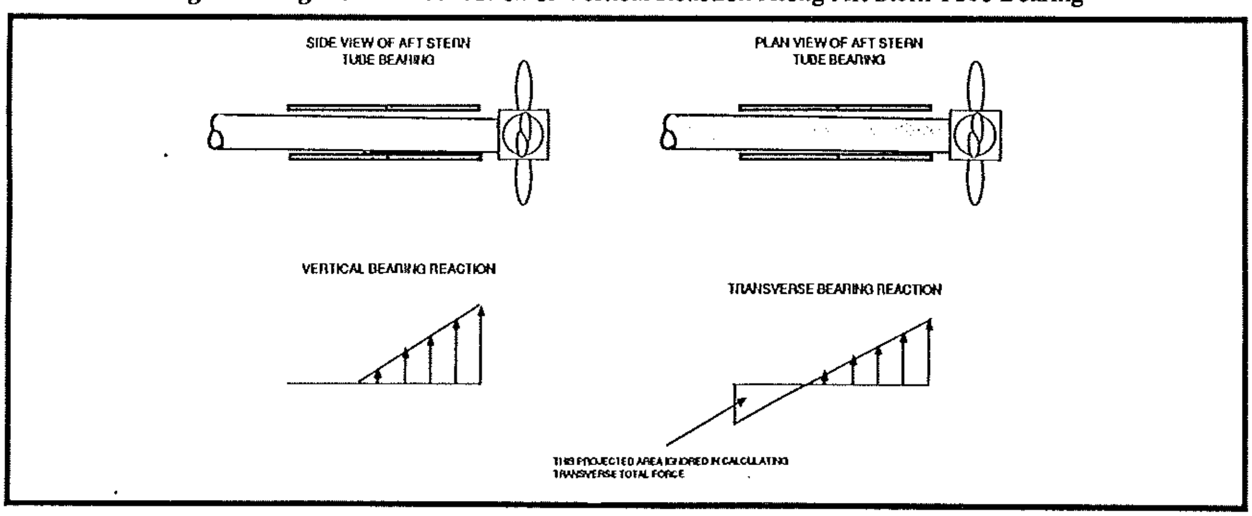

Fig. 18 Transverse Reaction Distribution when Shaft is not in Contact Along its Full Length

Measured Local Pressure on the Staves

The strain gauge placed in the bearing measures the local strain, which is converted to a pressure (using the calibration factor) at that gauge location. Figure 19 shows a typical time history of the pressure measured at one gauge location. For the first 2 seconds the propeller blades are ice free and the pressure on the stave oscillates at the frequency of the blade rate (8.7 Hz). Between 2 and 6 seconds heavy ice milling occurs. The pressure continues to oscillate at the blade rate, but with an amplitude 7 to 10 times higher than in the open water condition.

A scaled portion of Figure 19 is shown in Figure 20. Each time a blade strikes the piece of ice the pressure on the stave is increased. The magnitude of the bearing pressure decreases to approximately the open water pressure prior to the next pressure peak (next blade impacting the ice).

Bearing Reactions During Ice Milling

The transverse and vertical reaction of the forward stern tube bearing during an ice milling event is illustrated in Figure 21 and Figure 22 respectively. The corresponding resultant reaction and its angle are illustrated in Figures 23 and 24 respectively. The reaction plots for the aft stern tube bearing are illustrated in Figures 25 to 28. The units used for these plots have been removed because the data is classified as confidential. A positive transverse reaction indicates the shaft is impacting on the starboard side (outboard) of the bearing. A positive vertical reaction indicates that the shaft is pressing on the bottom of the bearing. Figure 33 shows the sign convention used for the bearing reaction plots. Both shafts rotate in the outboard direction over the top. The dynamic reactions have a frequency equal to the blade rate, each time the blade strikes a block of ice the reaction is increased.

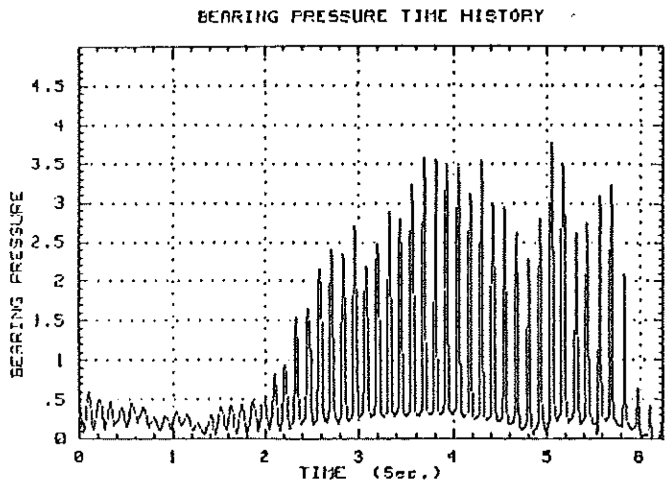

Figure 19: Time History Plot of Pressure on Rubber Stave (0-6.25 sec.)

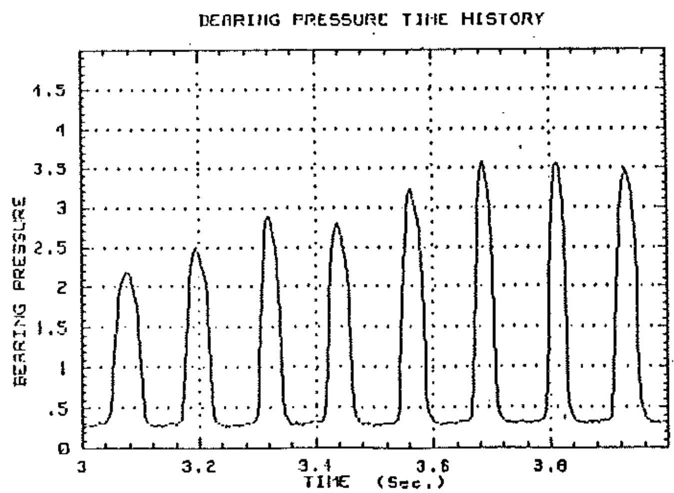

Figure 20: Time History Plot of Pressure on Rubber Stave (3-4 sec.)

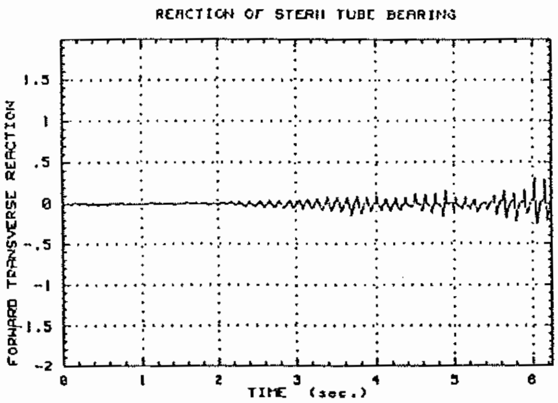

Figure 21: Forward Stern Tube Bearing Transverse Reaction: Time History

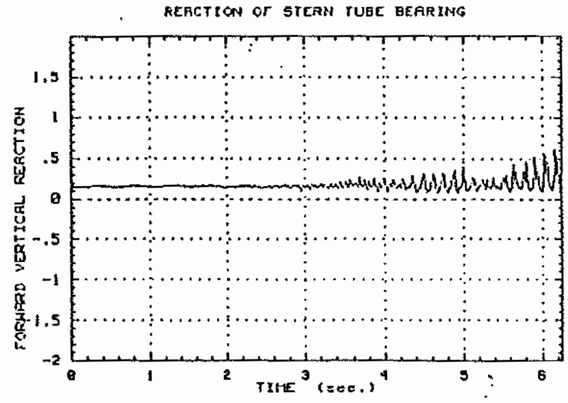

Figure 22: Forward Stern Tube Bearing Vertical Reaction: Time History

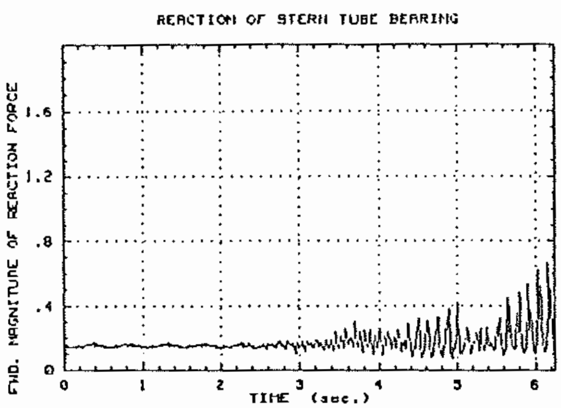

Figure 23: Forward Stern Tube Bearing Resultant Reaction: Time History

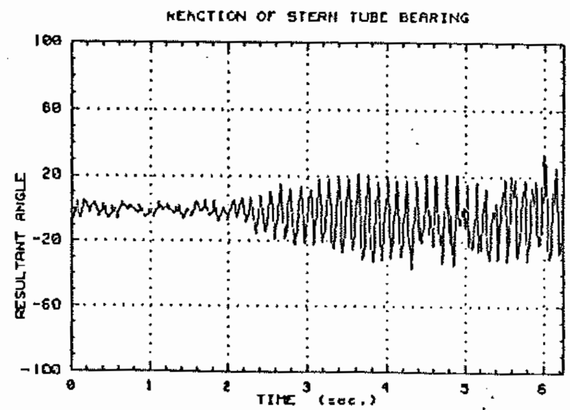

Figure 24: Angle of Forward Stern Tube Bearing Resultant Reaction: Time History

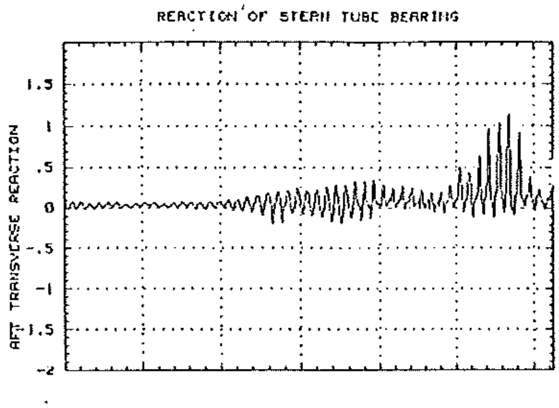

Figure 25: Aft Stem Tube Bearing Transverse Reaction: Time History

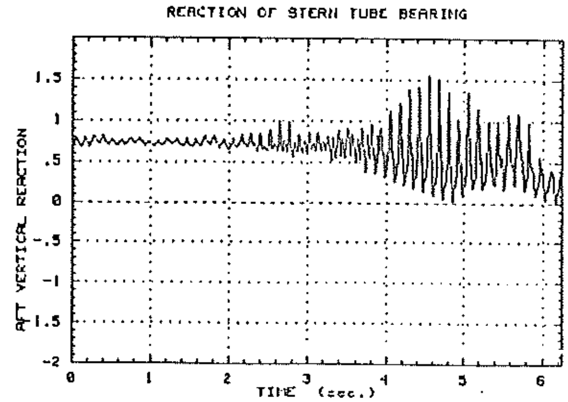

Figure 26: Aft Stem Tube Bearing Vertical Reaction: Time History

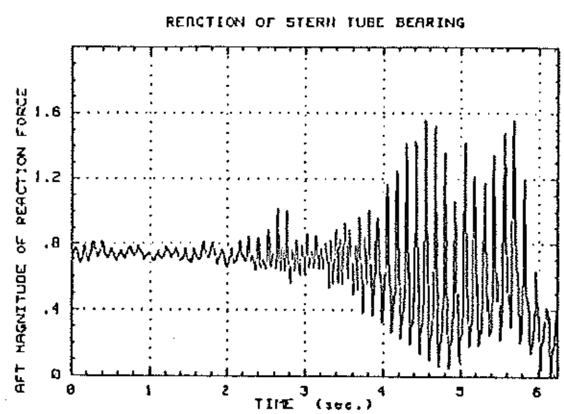

Figure 27: Aft Stem Tube Bearing Resultant Reaction: Time History

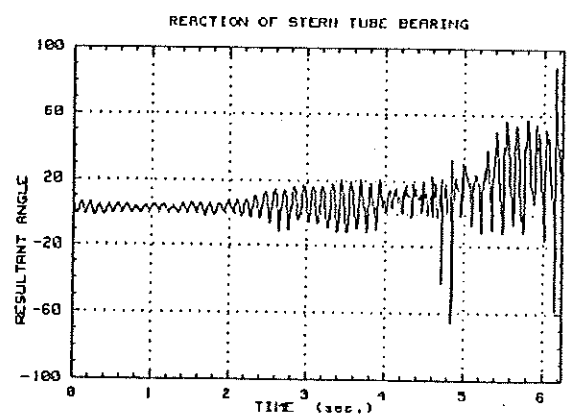

Figure 28: Angle of Aft Stem Tube Bearing Resultant Reaction: Time History

During ice milling (2 to 6 seconds) several observations can be made. The transverse reaction is predominantly in the positive direction for the aft stern tube bearing. When a blade strikes the ice, the shaft is pushed outboard and impacts on the starboard side of the bearing indicating that the ice block is located inboard of the propeller. The amplitude of the dynamic transverse reaction of the forward stern tube bearing is much less than that of the aft stern tube bearing. The mean vertical reaction of the aft bearing decreases during the ice milling part of the event, indicating that the shaft is lifted by an ice block located under the propeller hub. This change in the alignment condition results in a lower reaction on the aft stern tube bearing. The amplitude of the dynamic transverse and vertical reactions of the aft and forward bearings are approximately the same magnitude. The mean resultant reaction force increases on the forward bearing and decreases on the aft bearing, however, the amplitude of the dynamic reaction of the aft bearing is much higher than the forward bearing. The angle of the resultant reaction is biased towards the positive direction for the aft bearing and oscillates around the zero line for the forward bearing.

These observations conclude that the ice block is below and inboard of the propeller hub. The ice block is impacted by the blades as the shaft rotates in the outboard direction. The shaft is lifted up and thrown outboard each time a blade impacts the ice block. This type of interaction was commonly observed in several data sets analysed.

Data from the gauges also provided information regarding the distribution of loading throughout the length of each bearing, and the compression of the bearing material at each gauge location. However, this information is confidential and not discussed in this report.

Intermediate Shaft Bearing Data

Time history plots of the vertical, transverse and resultant load, as well as the direction of the applied load on the aft intermediate bearing are illustrated in Figures 29 to 32. Open water conditions exist at the propeller during the first 2.2 seconds and ice-propeller interaction occurs from 2.2 to 3.8 seconds. The ice block releases from the propeller and open water conditions are evident from 3.8 to 6.25 seconds.

Several observations can be made by analyzing these Figures. For the first 2.2 seconds, the vertical reaction oscillates about its static load at a frequency equal to the shaft rate (2.2 Hz). During ice-propeller interaction the mean vertical load decreases slightly and oscillates at the blade rate (8.7 Hz). The transverse load, shown in Figure 30, oscillates with a mean static load of zero during the open water conditions. When ice engages the propeller, the transverse load oscillates at the blade rate with approximately the same magnitude as the vertical reaction in open water conditions. The resultant reaction and the corresponding angle at which it acts are illustrated in Figure 31 and Figure 32 respectively. The amplitude of the dynamic resultant reaction is as high as 50% of the open water load on the bearing. The angle through which the resultant reaction oscillates is approximately five times more in the ice-milling condition than in the open water condition.

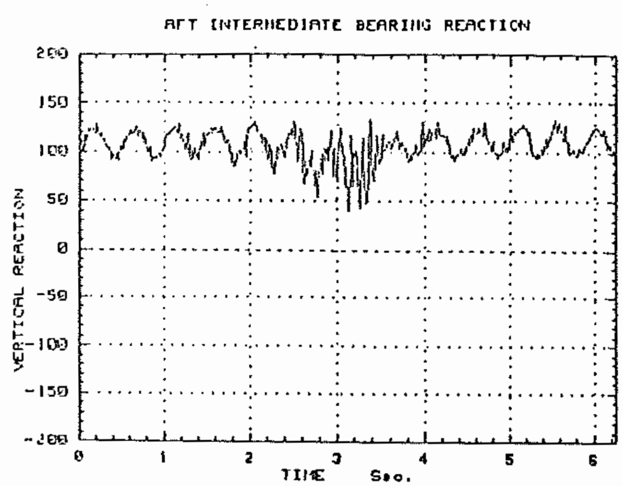

Figure 29: Aft Intermediate Bearing Vertical Reaction: Time History

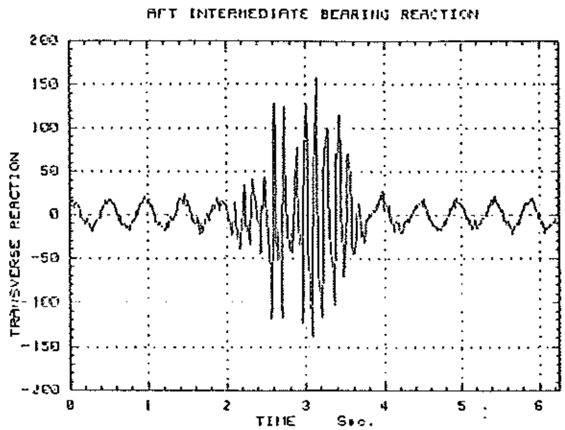

Figure 30: Aft Intermediate Bearing Transverse Reaction: Time History

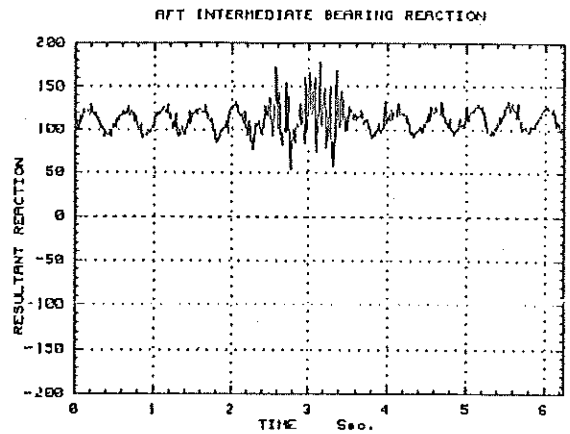

Figure 31: Aft Intermediate Bearing Resultant Reaction: Time History

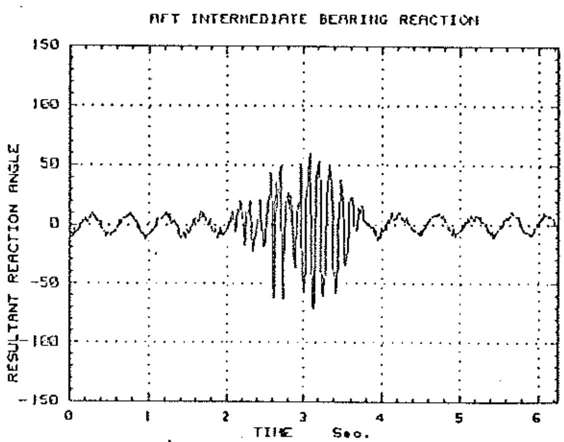

Figure 32: Angle of Aft Intermediate Bearing Resultant Reaction: Time History

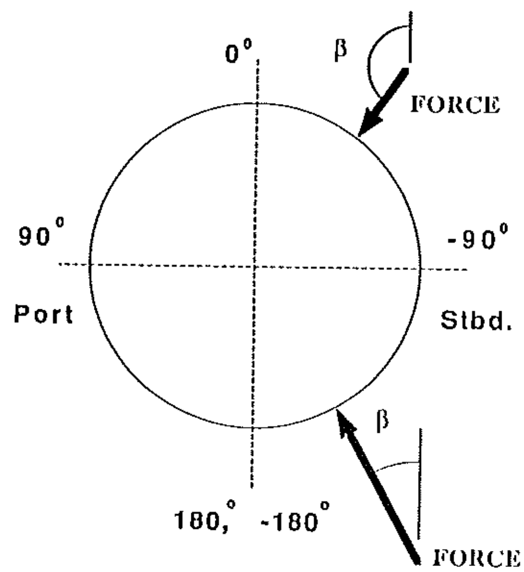

Figure 33: Sign Conventions for Bearing Reactions

CONCLUDING REMARKS

To the authors' knowledge, this is the first time that direct full scale measurements of the loading on the stern tube bearings of an icebreaker have been collected. The instrumentation performed exceptionally well under very severe conditions. The data presented here is only a portion of the data collected from the full scale tests; however this analysis has produced a number of observations. The transverse (port to starboard) reaction during ice milling is of the same magnitude as the vertical reactions in the stern tube bearings and the aft most shaft intermediate bearing. The location of the ice block can be determined, and an increased understanding of the ice-propeller interaction process can be obtained, by analysing the relative transverse and vertical reaction time histories of the bearings. The frequency of the dynamic response of the bearings correspond to the blade rate, such that each time a blade strikes a piece of ice a portion of the bearing is loaded.

The set of data collected during these full scale tests is an important addition to the base of knowledge used to develop propulsion system design criteria for ice class shafting systems. In addition, the improved understanding of the ice-propeller interaction process, and the qualitative picture obtained of the shafting system response to ice loads, will support the development of a modelling methodology for ice class shafting systems

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