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Testing Propulsion Systems for Performance in Ice

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Abstract

With an increasing interest in shipping in ice-covered waters, propulsion systems on these vessels are exposed to ice loads, which can be significantly larger than the open water loads. Severe damages can occur if the propulsion system is under-designed. At present rule formulae based on ice torque, which is related to the ice class of the vessel, is used in the design of these systems. The fact that failures continue to happen shows the need to study ice loads on these systems. Hence, at National Research Council (NRC) - Institute for Marine Dynamics (IMD) several types of propulsion systems have been tested in ice conditions over the years. Among them are highly skewed propellers and azimuthing podded propellers. This paper describes the capabilities developed at NRC-IMD to test these two different propulsion systems.

For the tests for the performance of highly skewed propellers a dynamometer to measure ice loads encountered by an individual blade was designed and built. It is mounted inside the hub and the blade attached to it. It can measure six component loads encountered by the blade. Some sample test results are given in the paper.

In a different study, azimuthing podded propellers are being modelled. The experimental model is designed so that ice loads on different locations on the podded system can be measured: blade loads, shaft loads, shaft bearing loads and the global loads on the whole system. A brief description of the system is presented in the paper.

* Contact person

Introduction

Navigation of ships in ice-covered waters around the world is increasing and changing. For example, a number of cruise ships and ferries are making regular trips in ice-covered waters. Furthermore, resource exploration and development is moving into Arctic regions, bringing with them various support and transportation vessels. Propulsion systems on these vessels are exposed to loading from ice contact. The ice loads can be significantly larger than open water loads and constitute a greater demand on the propeller and propulsion system. An under designed system will be susceptible to severe damage. Presently, these systems are designed using rule formulae based on ice torque, which is linked to the ice-class of the particular vessel. The efficacy of these rules needs to be kept constantly under review since propeller designs based on them still experience damage and failure. Bose et al. [1]have proposed a new method for ice class propeller design and analysis, as have Katsman and Andruishin [2] and Koskinen et al. [3].

Various model and full-scale tests have been completed to help determine the extra loading caused by propeller interaction with ice. To date, however, the majority of this work has been completed on the more conventional propeller blade geometries, both with and without ducts, fitted to icebreakers. While the results of these investigations may be applicable to other conventional propellers, their extension to the highly skewed propellers used on passenger ferries and other vessels navigating in ice is questionable and requires additional investigation.

With the intention of addressing this gap in knowledge, Searle et al. [4]completed a series of ice milling tests with both a highly skewed and a conventional propeller model. During the tests in model ice, the shaft loads resulting from propeller-ice interaction were recorded. Perhaps the most interesting result from these experiments was that the ice loading was more highly dependent on operating condition, in terms of advance coefficient and hence blade angle of attack, than the hydrodynamic loads. Furthermore, this sensitivity to blade angle of attack was more apparent for the highly skewed propeller than for the conventional ice class propeller.

In a parallel development, Transport Canada's initiative in the Harmonization of Polar Shipping Rules is leading to a Polar Code being accepted by the International Maritime Organization (IMO) based on proposed Unified Requirements of the International Association of Classification Societies (IACS). Amongst other things, these requirements give guidance for the design of machinery for operation in the polar regions. Amongst them are podded propulsion systems. While these proposed regulations do discuss load requirements in podded systems, they do not include the loads on the pods themselves. In addition, the loads on podded propellers are simply taken as loads on open propellers, no account is given to operation in off centreline pod orientations. Hence, an experimental study has been planned to investigate the loads on a podded propulsion system operating in ice conditions. In the last part of the paper, a description of the experimental setup developed at IMD is presented.

Ice tank

The ice tank [5] in the National Research Council of Canada's Institute for Marine Dynamics has a usable area of 76 m long and 12 m wide. It is 3 m deep. A 15 m long setup area is separated from the ice sheet by an insulated door allowing for the preparation of the equipment while the ice sheet is growing.

The carriage on this tank is capable of velocities from 0.1 to 4.0 m/s. The carriage is designed with a central testing area where a test frame, mounted to the carriage frame, allows the experimental setup to move transversely across the entire width of the tank.

The model ice used in the tests is EG/AD/S ice. This is a mixture of water, (e)thylene (g)lycol, (a)liphatic (d)etergent and (s)ugar, EG/AD/S, specifically designed to provide the scaled flexural failure strengths of real sea ice [6]. The ice sheet is started with a seeding process and continued to grow at approximately -20°C until it reaches the target thickness. The temperature of the room is then raised to above freezing and the ice is warmed and softened in a tempering process until the target ice strength is obtained. EG/AD/S ice models the flexural and shear strengths of sea ice fairly accurately. The tempering process mentioned above causes the disintegration of model ice along the grain boundaries. Therefore, the failure behavior of the model ice would be different from that of the sea ice. This may result in some scaling problems. Nevertheless, the failure envelopes of EG/AD/S ice are reported to be as good as the other model ices' failure envelopes, even superior in certain aspects [6].

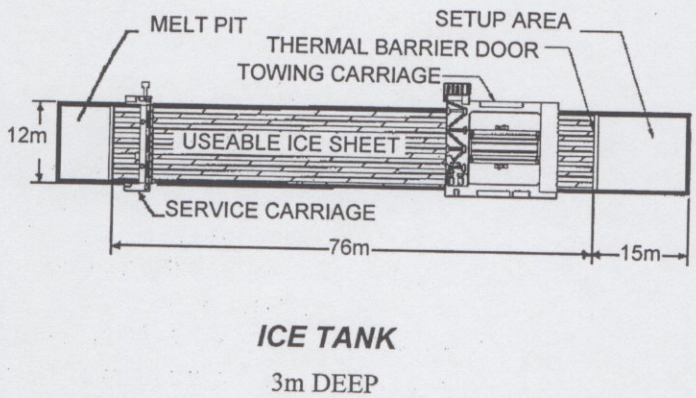


Figure 1. Schematic of IMD's ice tank facility

Ice performance tests of highly skewed propellers

Experimental model

Experimental model of highly skewed propellers consisted of a propeller boat, an ice plow forward of the housing, stern tube and the hub as illustrated in Figure 2. The propeller was powered by a 3 kW electric motor at 3000 rpm. A 90° 3:1 ratio gearbox was used between the motor and the propeller shaft to reduce the rotational speed. The output shaft of this gearbox was attached to a shaft dynamometer through a flexible coupling. In order to prevent damage to either the shaft dynamometer or the gearbox, a weak link is fitted on the propeller side of the shaft as shown in the figure. A water-cooled brass bearing supported the drive shaft at the end close to the hub. The other end was supported by the in-line shaft dynamometer.

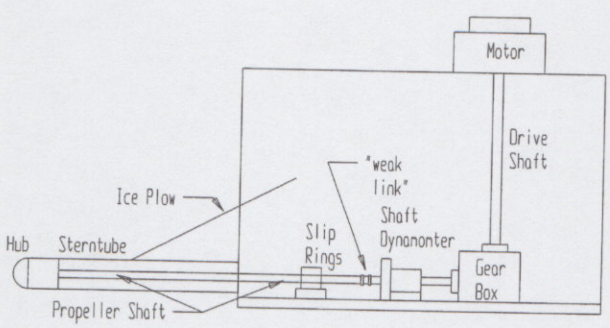


Figure 2. Propeller boat

During open water tests, wave deflectors were fitted forward and aft to avoid swamping of the boat due to overtopping bow or following waves. The top cover was installed to keep water spray and waves out. For operation in the ice sheet, the ice itself prevented the formation of a significant bow wave and the bow wave deflector was removed to allow the installation of the ice plow.

Propeller Blades

Two sets of blades were provided by Lloyd's Register identified with 6602 and 6603 model numbers (Figure 3). They were 1:19.259 scale models of a controllable pitch propeller design. The main characteristics of the propellers are given in Table 1. The diameter of the model propeller was 270mm. This is a similar design to the ones used on passenger ferries in Canada and Europe.

Table 1. Propeller characteristics

Model no	P/D @ 0.7R	Skew	A_E	A_P
6602	1.460	50°	0.542	0.444
6603	1.337	50°	0.542	0.449

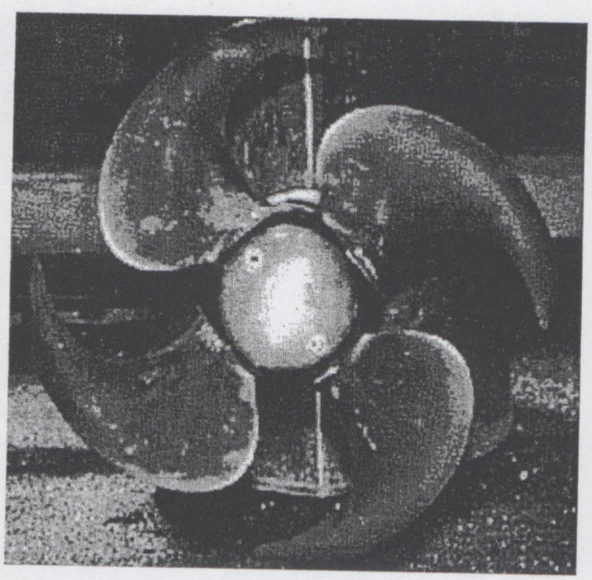


Figure 3. Assembled model propeller (Model number 6603)

As the propellers being modeled were controllable pitch, it would have been ideal to change the pitch while the model propeller was moving as in the full-scale system. This was not practical and instead a series of tapped holes were milled into the hub in the three blade mounting locations, corresponding to the design pitch, a reduced pitch 7.23° below the design, a reduced pitch 15.27° below the design, and a reverse pitch 52.23° below the design pitch.

Blade Dynamometer

One of the propeller blades was mounted to a special in-house designed and built six-component dynamometer. It has a cylindrical shape, with the outer wall varying in thickness (Figure 4). It is made of stainless steel. It is located inside the hub and capable of measuring six-component of blade loads. All mating of the dynamometer parts was conducted with stainless steel machine screws or grade eight machine screws sealed with waterproofing to prevent rusting. Before calibration, the dynamometer was waterproofed and mounted inside the hub to eliminate mounting variation as a source of error.

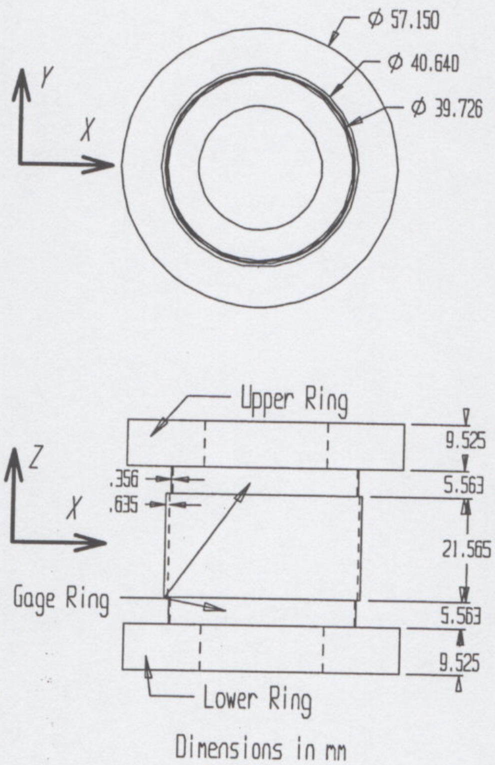


Figure 4. Blade dynamometer

In an attempt to reduce the amount of noise picked up by the strain gauges, they were outfitted with an electrical pre-amplification of approximately 100 times. This enabled the transmission of higher voltages through the slip ring assembly reducing the noise to signal ratio. The signals were further amplified to approximately 75% of the voltage required for full-scale deflection in a signal conditioner. Due to the high rotational speed and the requirement of a large number of data points over each blade-ice contact, each of these channels was sampled at 5000 Hz.

The design load limits of the blade dynamometer were as follows: maximum forces in the *x* and *y* directions were both 800 N, maximum force in the *z* direction was 600 N. Maximum moments were 85 Nm about both *x* and *y* axes and 50 Nm about *z* axis. Figure 5 shows the definitions of *x, y,* and *z* directions with respect to the dynamometer. It also shows the orientation of these directions with respect to the propeller. Note that in the figure the propeller would be rotating counter clockwise and progressing forward out of the page.

Calibration of the blade dynamometer was conducted by securely mounting the hub, containing the dynamometer inside, in a known position and applying known loads and moments to the blade end of the dynamometer. These loads were applied in such a way that the dynamometer was excited in the positive and negative directions of all six components. To determine interaction effects, the relative magnitude of forces and moments were varied for different calibration setups. The data was then analyzed to produce a calibration matrix, which was subsequently confirmed using 45 linearly independent loading conditions designed to simulate the expected operating range of the dynamometer during ice interactions. Once the first set of blades was tested, the dynamometer was re-calibrated before being used on the second set of blades.

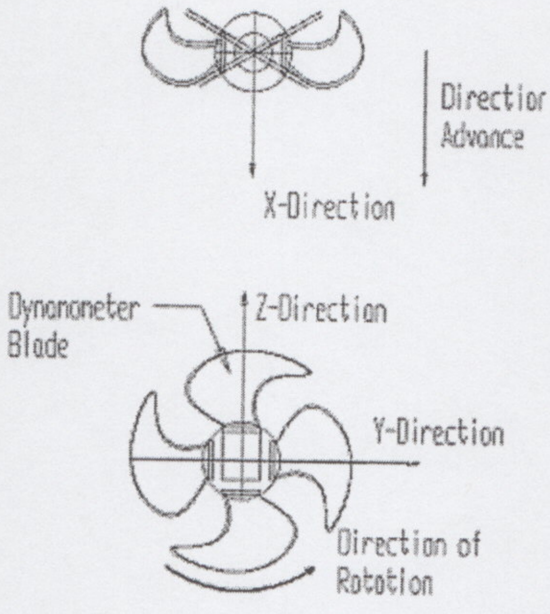


Figure 5. Directions x, y, and z on the Propeller

Other Sensors

Other sensors were used to measure the shaft load, shaft speed, propeller submergence, and carriage speed.

Shaft loads were measured using an inline dynamometer manufactured by Sensor Development Inc. that had a maximum thrust load rating of 890 N and maximum torque load rating of 110 Nm. The dynamometer had 100% overload capacity and maximum rotational speed of 20 rps. These two channels were also sampled at 5000 Hz.

Shaft speed was measured by a tachometer built into the 3 kW motor. The motor tachometer was calibrated using a laser tachometer. The motor was run through a series of settings and the rotational speed on the propeller side of the gearbox was measured using the laser tachometer. The settings for the correct motor speeds were then determined. As the rotational speed was not expected to vary much in the ice it was sampled at 100 Hz to conserve memory space.

Propeller submergence was measured using a potentiometer, in which the base unit was mounted to the carriage and the end of the wire was attached to the test frame holding the propeller boat. The zero value for this potentiometer was set at the point where the upper edge of the propeller circle just failed to break the surface of the water. Once set, the vertical position of the test frame did not change and it was sampled at only 50 Hz.

Carriage speed was recorded from a tachometer built into the drive motors of the carriage. Carriage speed did not vary significantly during the propeller interaction event. It was sampled at 100 Hz as well.

In addition to the sensors, three above water cameras were used to record the ice milling event. These recorded each test looking from the bow, starboard side and directly above the propeller.

Model ice properties

Four ice sheets were used for each of the two highly skewed propeller models. The first sheet was used to examine the testing procedure; it had a relatively weak target flexural strength of 30 kPa and a target thickness of 60 mm. The remaining sheets were stronger with flexural strengths of about 50 kPa. The second and third sheets were 60 mm thick, and the last sheet was 80 mm thick (see Table 2).

The initial testing procedure involved placing longitudinal cuts approximately 0.75 m apart and 10mm deep along the length of the ice. Transverse cuts were then made at approximately 0.5 m spacing and 10 mm depths, as illustrated in Figure 6. These were provided to allow the ice to break when the propeller boat plow contacted it, thus preventing the ice from rising off the propeller. The cut pattern also caused the ice to break easily, preventing cracks from proceeding across the remaining ice sheet. The next test pass was then run alongside the previous one, thereby requiring only one longitudinal cut (the opposite edge being open water).

Table 2. Ice Sheet Strengths

Sheet (Blade#-Sheet#)	Target (Strength/Thickness) [kPa / mm]	Actual (Start)	Actual (End)
1-1	30 / 60	20 / 59	15 / 60
1-2	50 / 60	40 / 52	20 / 61
1-3	50 / 60	54 / 51	30 / 60
1-4	50 / 80	65 / 74	25 / 81
2-1	20 / 45	34 / 48	20 / 48
2-2	45 / 60	42 / 60	29 / 62
2-3	45 / 60	35 / 59	33 / 60
2-4	60 / 80	53 / 82	50 / 82

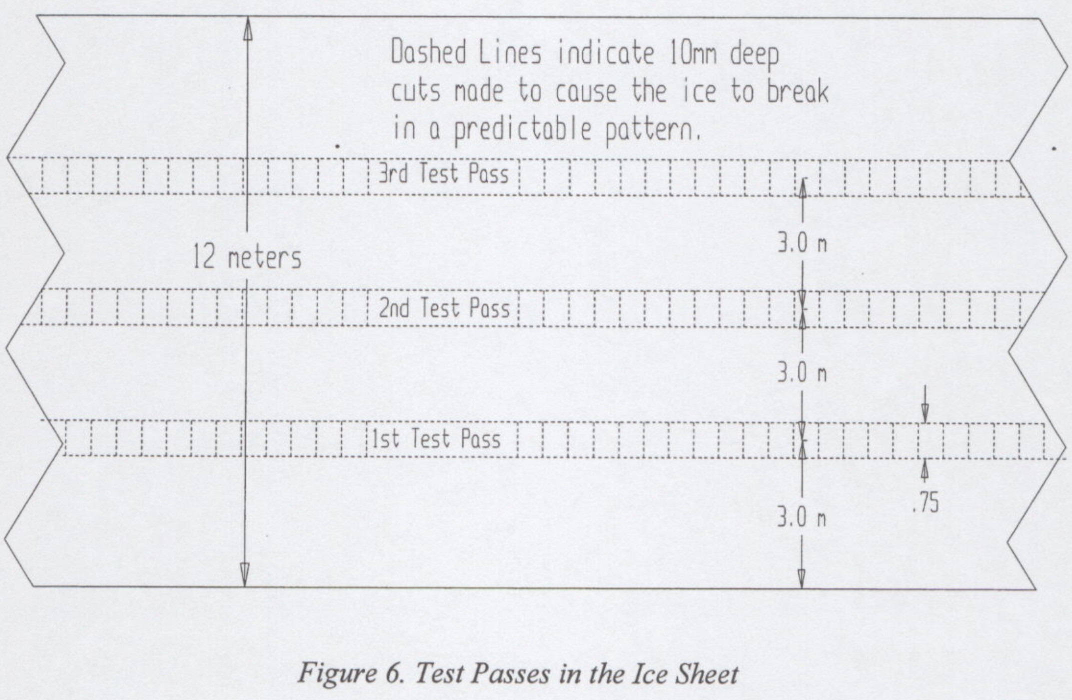


Figure 6. Test Passes in the Ice Sheet

Results

In the following are the some samples of the data obtained from the experiments. A more detailed description of the results are given in [8] and [9].

Figure 7 shows the variation of X-blade moment for two consequent cycles. This is given for a pitch setting of –7.23°, ice thickness of 34 mm and the advance coefficient of 0.2. The loads include the hydrodynamic effects on the blade, but have the centrifugal loads removed through the taring process. Mean open water blade load for the same pitch setting and advance coefficient is also shown. The orientations of the blade loads are defined using the convention presented in Figure 5. While there was no method provided in these experiments to determine precisely when the instrumented blade entered the ice sheet, it is assumed that it is the point where the loads begin to increase, that is, at times of 0.0 and 0.1 seconds.

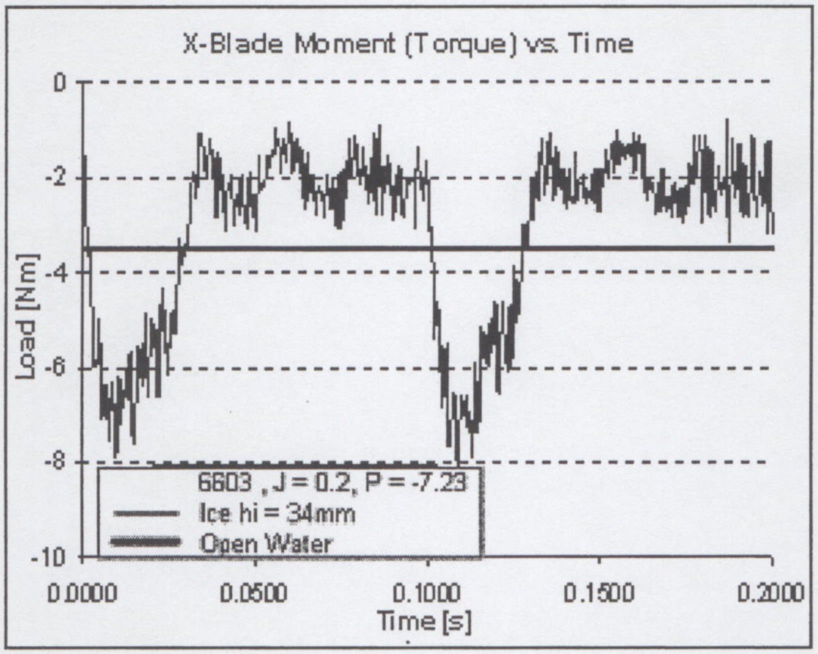


Figure 7. Time series of X-blade moment

Kt and Kq were also calculated for different test cases. A sample graph is given in Figure 8. In this case, ice thickness was 7 mm for the Model 6603 at the design pitch setting.

Total shaft thrust and torque were measured directly using a shaft dynamometer. It was found that the mean measured blade load was, after allowing for shaft losses, ¼ of the total shaft load, which was expected. Shaft loads for both propellers showed similar patterns.

Based on these experiments, the patterns of loading experienced as a result of changes in advance coefficient are generally similar for both propellers for most of the load components, although there are significant differences in magnitudes. There were differences at the maximum loads encountered between the two models. However, it was difficult to ascertain the causes of these differences as both the radial pitch distribution and ice strength differed in this group of tests.

It was also found that the increase in shaft loads occurred at progressively lower cut depths as the pitch was reduced from the design pitch. That is, conditions farther from the design conditions increased the amount of ice loading relative to the hydrodynamic load.

Both thrust and torque coefficients increase with increases in depth of cut, although dependence of thrust coefficient on ice strength appears to be relatively weak. Further, the effect of ice strength appears to be relatively more pronounced for torque coefficient than the thrust.

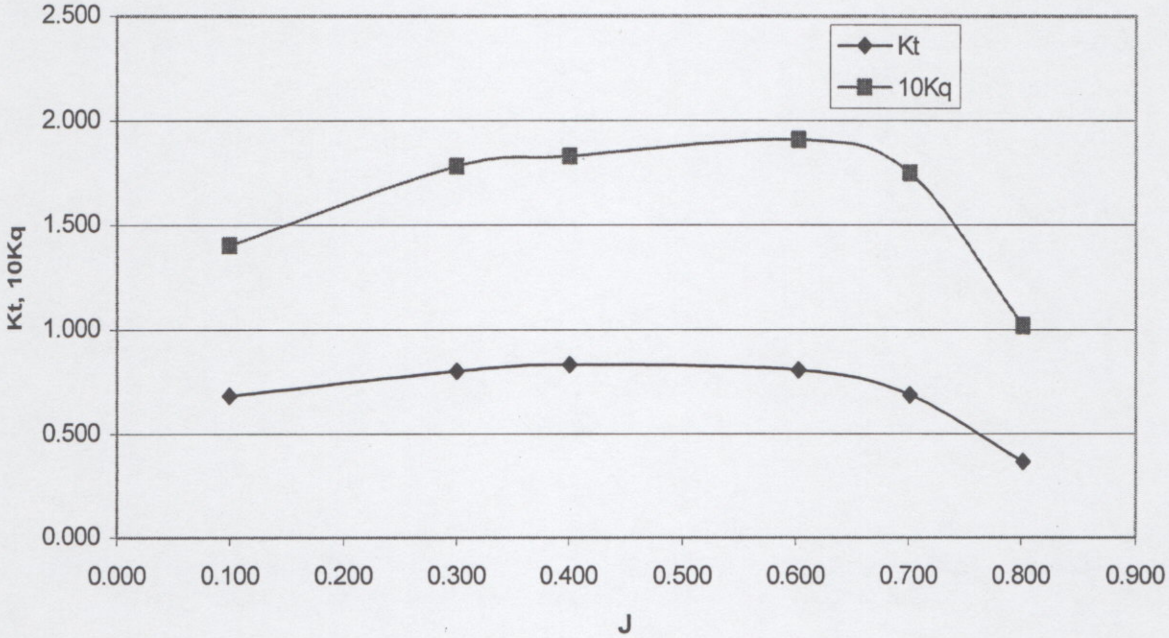


Figure 8. Kt, Kq curves for the Model propeller 6603 with 7mm ice thickness at the design pitch setting.

Ice performance tests of podded propulsion systems

The research being conducted here will give information on the loads on a podded propulsion system operating in ice. It involves the design and testing of scale models of a podded propeller in ice using the facilities of the IMD. The propellers will be instrumented to record contact and "milling" loads exerted on the propulsion system as it is passed through an ice sheet. This will include measuring shaft and strut loads, the loads on one individual blade while operating in ice, and the in-plane loads exerted on the stern bearing of the pod.

The following section describes the experimental model. The tests are scheduled for sometime later this year.

Experimental model

A model stern was constructed for the tests (Figure 9). It houses the dynamometer that measures global forces and moments acting on the whole podded system. It will also simulate the stern of a hull to a certain extend. Use of a complete model hull would be necessary to model the effects of hull ice interactions more accurately, however, this is not included in the current scope of the work.

The main body of the podded propeller system is made of stainless steel, with some parts being aluminum. The propeller, the pod, the strut and the drive motor are a stand alone system, which is connected to the main frame through loads cells, which form the global dynamometer (see also Figure 10). The cross section of the strut is formed by two intersecting circular arcs. Its width and thickness are 0.3m and 0.15m, respectively. Its span length is approximately 0.5m from pod body to the false stern.

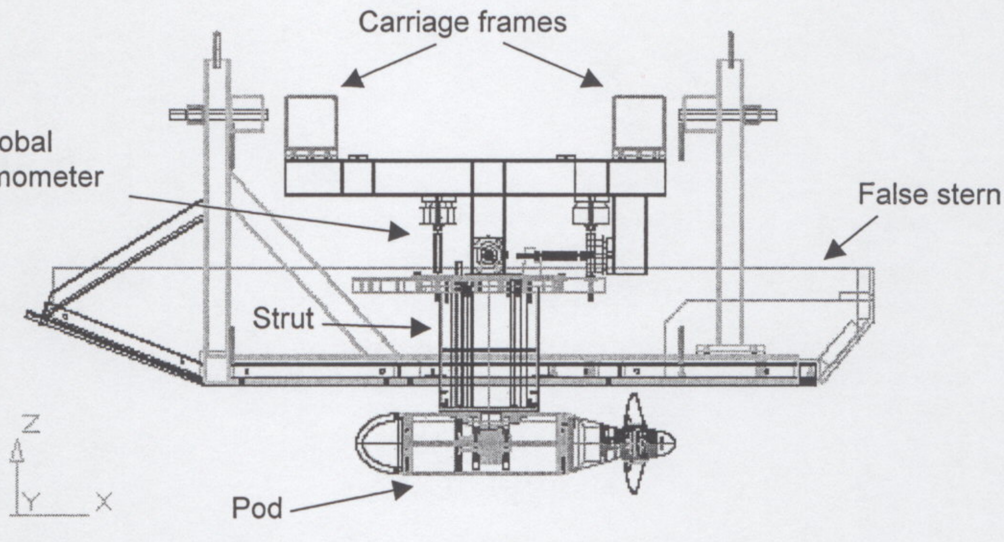


Figure 9. Schematic of the podded propeller test setup

Figure 11 shows the assembly of the pod and the propeller. The pod is of circular cross section with a diameter of 0.17m and a length of 0.95m. Propeller is designed to be similar to a general icebreaker propeller. It has a radius of 0.3m and four blades. Mean Pitch/Diameter ratio is 0.76. Only one of the blades has been instrumented to measure the loads (forces and moments in three orthogonal directions). The diameter of the hub at the propeller is 0.11m.

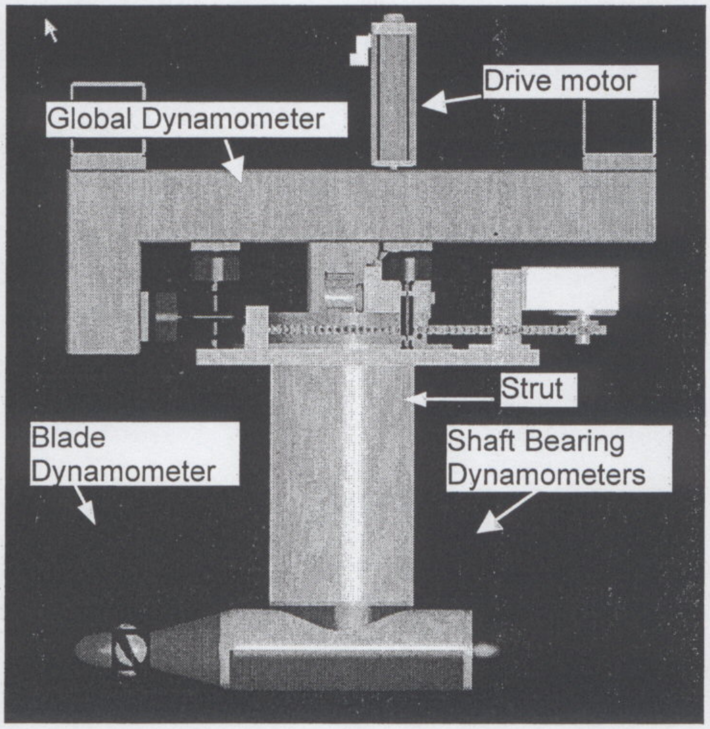


Figure 10. Complete system

The propeller shaft is driven by a 4.5HP electric drive motor at 7000 rpm.The motor is attached to the upper part of the housing (see Figure 10). It drives a vertical shaft into a 5:1 ratio reduction gearbox. There is a bevel gearbox coupling between this 5:1 ratio gearbox and another 2:1 reduction gearbox (not clearly shown in the figure). The output shaft of the second gearbox is connected to the propeller shaft through a drive belt system.

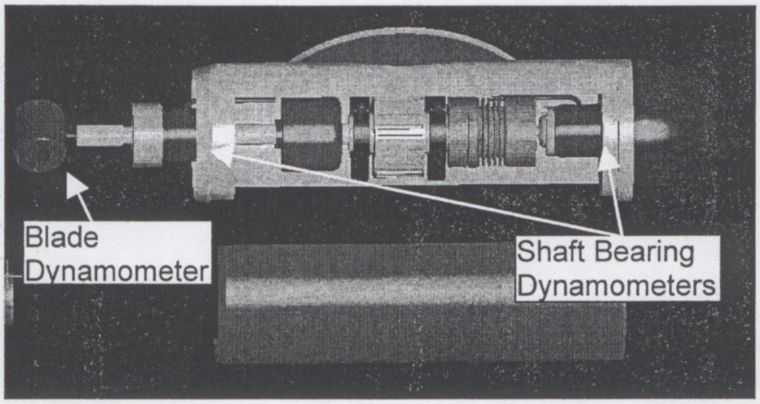


Figure 11. The pod and the propeller of the experimental model.

Sensors

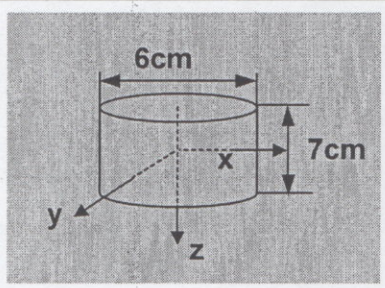
In the design of the podded system, special attention has been given to obtain load information on three levels: on an individual blade, on the shaft and on the whole podded system. Figure 10 gives locations of the different load sensors.

Blade and Shaft Bearing Dynamometers

The three dynamometers shown in Figure 11 are identical. They are special type for underwater use and capable of measuring six components – forces and moments in three orthogonal directions. Their capacity for singly loading is given in Table 3. x, y and z reference frame attached to the blade dynamometer is shown with respect to the propeller.

Table 3. Dimensions of blade dynamometers and maximum rated loads

F_x	2227 N
F_y	2227 N
F_z	4454 N
M_x	57 Nm
M_y	57 Nm
M_z	57 Nm



Loads on the drive shaft will be measured by the bearing dynamometers. Additionally, a strain gage is mounted on the drive shaft for shaft torque measurement. Shaft thrust will be computed using the information obtained from the two bearing dynamometers. The bearing dynamometer on the opposite end of the shaft to the propeller is mounted after a thrust de-coupler, therefore, is not expected to encounter as large forces or moments as the other one.

Blade dynamometer is housed inside the hub. It is attached to one of the blades. The angular position of this blade will be recorded during the tests with a resolution of half of a degree. This will enable an accurate matching of the forces and moments due to the contact of the blade with ice.

The wiring for the strain gage and the blade dynamometer is run through the drive shaft and a set of slip rings to a signal conditioner. Due to the high rotational speed and the requirement of a large number of data points over each blade-ice contact, each of these channels will be sampled at 5000 Hz.

Global Dynamometer

In order to measure the forces and the moments acting on the whole podded system, including the strut, a global dynamometer was designed and fabricated at NRC-IMD. It consists of six high precision 8907 N load cells oriented as shown in Figure 12. The global dynamometer is fixed in a reference frame attached to the model stern. The strut and the pod can be rotated 360 degrees in vertical direction using the azimuthing gear, while the model stern, hence the global dynamometer, remains fixed.

Other Sensor Equipment

Other sensors were used to measure shaft speed, propeller submergence, and carriage speed.

Shaft speed will be measured by a tachometer built into the 4.5 HP motor. The motor tachometer will be calibrated using a laser tachometer. As the rotational speed is not expected to vary much in the ice it will be sampled at 100 Hz to conserve memory space.

Propeller submergence will be measured using a potentiometer, in which the base unit is mounted to the carriage and the end of the wire is attached to the test frame holding the propeller boat. The zero value for this potentiometer is set at the point where the tip of the blades would just fail to break the surface of the water. The vertical position of the test frame will be sampled at only 50 Hz.

Carriage speed is recorded from a tachometer built into the drive motors of the carriage. Carriage speed is not expected to vary significantly during the propeller ice interaction event. It will be sampled at 100 Hz as well.

In addition to the sensors, three above water cameras will be used to record the icemilling event. These will record each test looking from the bow, starboard side and directly above the propeller.

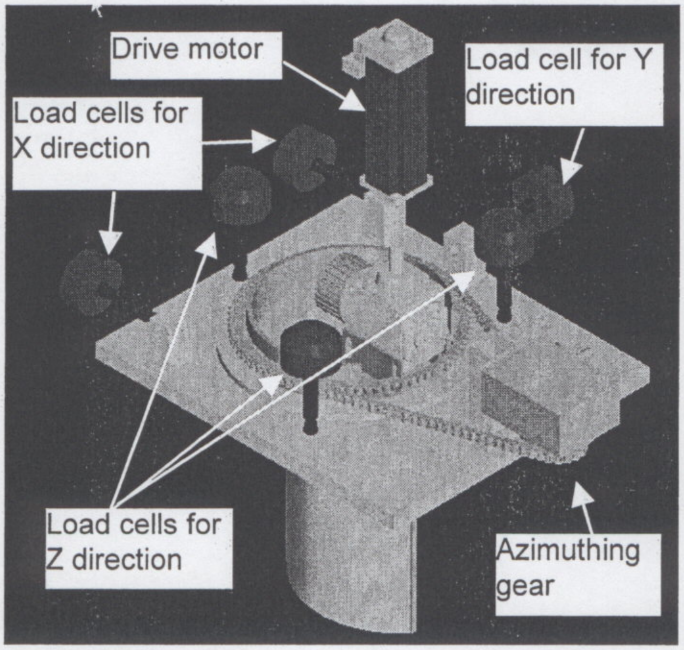


Figure 12. Loads cells which form the global dynamometer.

Conclusions

The descriptions of two experimental setups- one for highly skewed propellers and the other for podded propulsion systems, designed and built at IMD are given. For the former, a dynamometer, which is capable of measuring six component loads encountered by one of the blades, was designed and built in house. The tests for these highly skewed propellers have been completed successfully. Some sample results are presented here. In general, the tests confirmed the opinion that highly skewed propellers are susceptible to ice load damage both because their geometry attracts high loads and because they tend to be relatively weak, especially near the tip. Furthermore, tip unloading can result in even higher ice loads than unloaded tips. In general, highly skewed designs should be avoided for ice class propellers [9].

The experimental model for podded propulsion systems is designed and built in such a way that it will measure six component loads on an individual blade, shaft loads (torque and thrust), six component shaft bearing loads and the global loads occurring on the whole podded system.

Acknowledgements

The research on the highly skewed propellers was funded primarily through a joint project between Transport Canada, Lloyd's Register, and the National Research Council Canada. Also, the Natural Sciences and Engineering Research Council (Canada) and the Government of Newfoundland & Labrador supported this project financially. Thanks are due to the people from Oceanic Consulting Corporation and Art Bowker Inc.

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Nomenclature

A_E	Expanded area ratio
A_P	Projected area ratio
D	Propeller diameter
P	Propeller pitch
R	Propeller radius
rps	Revolutions per second

References

[1] Bose, N., Veitch, B., and Doucet, M. 1998. "A design approach for ice class propellers." *Transactions, Society of Naval Architects and Marine Engineers,* Vol. 106.

[2] Katsman, F.M., and Andruishin, A.V. 1997, "*Strength Rates for Blades as Intended for the Propellers of Ice Breakers and Ice Ships,"* Russian Maritime Register of Shipping.

[3] Koskinen, P., Jussila, M., and Soininen, H., 1996, "Propeller Ice Loads Models", Technical Research Centre of Finland, *Research Notes 1739.*

[4] Searle, S., Veitch, B., and Bose, N., 1999, "Ice-Class Propeller Performance in Extreme Conditions," *Transactions, Society of Naval Architects and Marine Engineers Vol 107.*

[5] Jones, S. J., 1987, "Ice tank Test Procedures at the Institute for Marine Dynamics," Institute for Marine Dynamics Report No. LM-AVR-20.

[6] Timco, G.W., 1986, "EG/AD/S: A New Type of Model Ice for Refrigerated Towing Tanks," Cold Regions Science and Technology, Vol. 12, pp. 175-195.

[7] Jones, S. J., Timco, G. W., and Frederking, R., 1989, "A Current View on Sea Ice Modelling," Proceedings of the 22nd American Towing Tank Conference, St John's, NF, Canada, pp114-120.

[8] Moores, C., Veitch, B., Bose, N., Jones, S., Carlton, J., 2002. Multi-Component Blade Load Measurements on a Propeller in Ice. *SNAME Transactions*

[9] Moores, C., Veitch, B., Bose, N., Jones, S. 2002. Experimental results of two highly skewed propellers in ice: blade and shaft load measurements. *TR-2002-06*, Institute for Marine Dynamics, St John's, NL, Canada.