Effect of lateral ice pressure on ship resistance

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SUMMARY:

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ABSTRACT

The effect of lateral ice pressure on the level ice resistance of a 1/30 scale model of an OBO is examined experimentally. The model was equipped with an instrumented side panel. The apparatus for producing lateral pressure is described, as well as the model tests themselves. Ice pressures were recorded using extensimeters mounted in the ice. The results indicate that the compressive pre-stress in the ice had little or no effect on the breaking resistance. However, as the model moved well into the pressurized region ice was forced against the hull and very high lateral loads were generated giving rise to frictional resistance. The test results indicate that bow reamers may be beneficial when operating in pressurized conditions.

1.0 INTRODUCTION

Compressive pack ice is a severe impediment to safe and effective navigation. Pack ice forces can hinder ability of a ship to effectively transit and manoeuvre in the ice field [1]. In extreme conditions, pack ice forces have caused structural damage and loss. Ice pressure in pack ice appears to be often caused by wind acting upon the ice surface although in one reported case water current appeared to be the driving force. Observations by Bradford [2,3] aboard CCGS *John A. MacDonald* and *Louis St. Laurent* show a strong correlation between wind speed and the occurrence of pressured ice and the probability was greatest with onshore winds. He also observed that the ice concentration was always >9/10$^{\text{ths}}$ and the pressure appeared to intensify as the vessels moved closer to shore.

Analytical studies [4,5] that account for the effect of ice pressure on ship resistance assume that there are two possible effects. First, the pressure may act as a compressive pre-stress which must be over come before tensile flexural failure can occur. The second effect may be the closing of the channel around the hull and the frictional resistance resulting from ice forces along the sides. During the second voyage of the *S.S. Manhattan*, Bradford reported that the channel behind the ship could close within one mile behind[6]. He estimated channel closure rates up to 0.1 m/s.

Physical modelling of this phenomena has had limited success partially due to the limited size of the ice basins. In one of the more recent studies reported by Kujala et al. the relatively large ice basin at Helsinki University of Technology was utilized [7]. The main towing carriage was used to laterally push a large floe against one side of the model while it was towed at constant speed in a direction perpendicular to the pressure force. These tests simulated the passage of a ship in a lead that was closing under pressure, similar to a vessel following an escort. A significant increase in resistance was seen when the pressure was applied and a further increase was observed as the lead closed ahead of the bow. In this current study the model is inside a floe which is subject to external pressure, thus simulating passage of an independent ship.

In another test program[8], a model of a LNG carrier was towed through an ice floe under pressure. Pressure was applied on one side by six 0.9 m long pusher bars. Lateral pressure was generated by deadweights acting through a pulley system. As the model passed each segment a shear line formed in the ice sheet and a piece of ice was driven into the model at relatively high speed. When this occurred over an extended area of the hull the resistance almost doubled over that in unpressured ice.

2.0 DESCRIPTION of APPARATUS

The apparatus was designed to model the passage of a ship through a large floe which is under external pressure [9], as depicted in Figure 1. Pressure was applied to the ice sheet via two pneumatic hoses installed in each side of the tank, 4 metres from the centreline. The hoses were 10 cm in diameter and were 12 m long. Aluminum guides restricted vertical movement of the hoses as they filled with air and each pressure tube could extend 70 mm. The apparatus, Figure 2, was in essence a pressure amplifier with the pressure in the ice related to the tube pressure by;

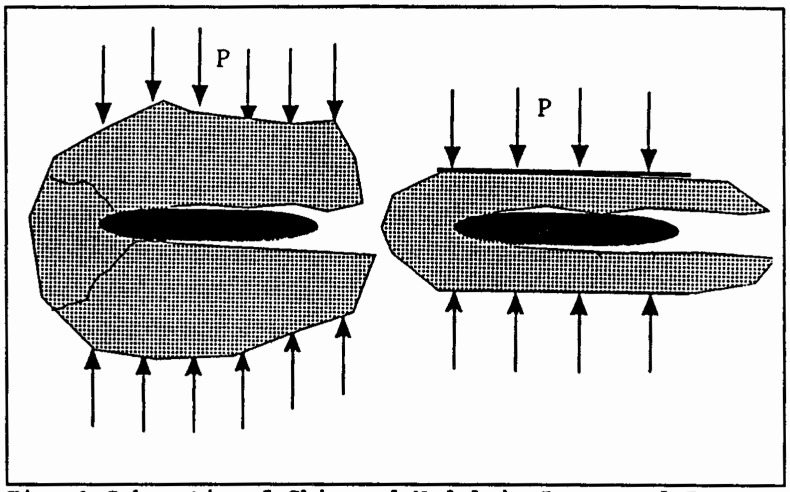

Fig. 1 Schematic of Ship and Model in Pressured Ice.

$$P_{ice} = P_{tube}\left[\frac{D}{h_i}\right] \qquad (1)$$

where P_{ice} and P_{tube} are the pressures in the ice and tubes respectively. D is the height of the pressure face (114 mm) and h_i is the ice thickness. At the fullest extension full pressure could not be maintained because the tube could not keep in full contact with the pressure face.

The tubes were installed immediately after the ice sheet was seeded so the growing ice sheet would bond well to the apparatus. Pressure was controlled by manually operating a ball valve while observing the tube pressure via a manometer. Tube pressure was recorded using an electrical pressure transducer. Both tubes were connected together by a hose so pressure was equalized between them. A displacement transducer was installed to record the displacement of the pressure bar on the port side of the model.

Two sensitive extensimeters were frozen into the ice sheet to measure in-plane compression when the pressure was applied. Attempts were made to calibrate these by mounting them on a 0.6 m wide by 0.4 m long in-situ beam which was compressively loaded using a hydraulic actuator. The load and strain from the extensimeter was then used to calculate an effective elastic modulus

$$E_{eff} = \frac{P\,L}{A\,\Delta L} \qquad (2)$$

where P was the applied force, A was the area (0.6 x h_i), L is the gage length (0.34 m) and ΔL was the strain. The results indicate that the extensimeters gave reasonable indications of ice pressure but were not sufficiently accurate. The mean of the measured moduli were about 40% higher than that determined by loading the ice plate in its centre. However, the mean deviation of the results was almost 50% making the technique quantitatively unreliable.

The line load required to cause the buckling of a semi-infinite beam [10] on an elastic foundation is given by

$$q = \sqrt{\frac{\rho_w\, g\, E\, h^3}{12\ (1 - \nu^2)}}\left[1 + \frac{3.32}{2(b/l) + (b/l)^2}\right] \qquad (3)$$

where ρ_w is the density of water(1000 kg/m^{-3}), g is the gravitational constant (9.806 m/s^2), E is the elastic modulus of the ice and ν is Poisson ratio (0.3).

The ratio of pressure length to ice sheet characteristic length, b/l, is approximately 20 so the term inside the square brackets is essentially one. For the EGADS model ice used at IMD the characteristic length is about 11 times the ice thickness so that;

$$E = 12\ (1 - \nu^2)\ \rho_w\, g\, 11^4\, h_i \qquad (4)$$

Substituting (4) into (3) we get an in-plane compressive stress, σ_b, above which the ice sheet will buckle

$$\sigma_b = \frac{q}{h_i} = 11^2\, \rho_w\, g\, h_i \qquad (5)$$

For the 50 mm thick ice sheets used in these experiments the buckling stress is about 60 kPa, and our target pressure was approximately 50% of this. From (1) the required hose pressure was about 12 kPa. An ice pressure of 30 kPa was a substantial pre-stress considering the target flexural strength was only 40 kPa.

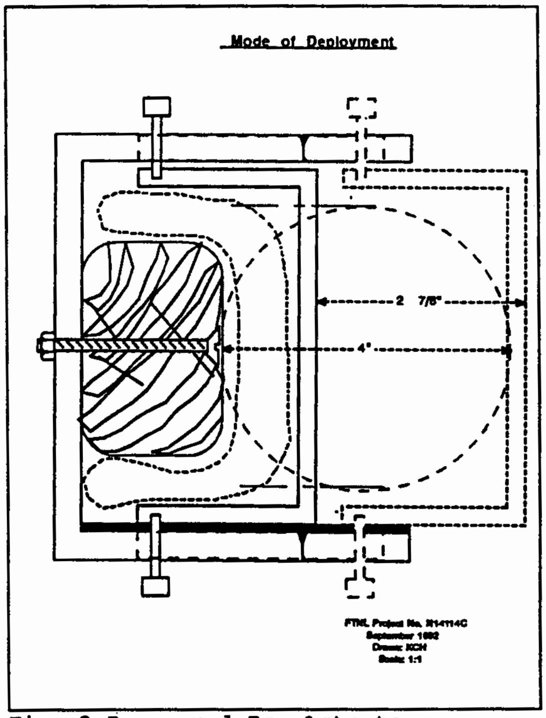

Fig. 2 Pressured Ice Actuator.

The model used in this program was 1/30 scale model of the *M.V. Arctic* with its original bow form, see Figure 3. This vessel has a high length/beam ratio of 8.6 and over 50% of its length is vertical parallel middle body (from station 4 to 16.5). The hull had an ice-hull dynamic friction coefficient of 0.1. The model was outfitted to measure side loads via a panel located on the starboard side, Figure 4. The panel was 750 mm long and 330 mm deep and was fitted so that its centre 3.55 m fore of the aft perpendicular and 0.370 m above the keel.

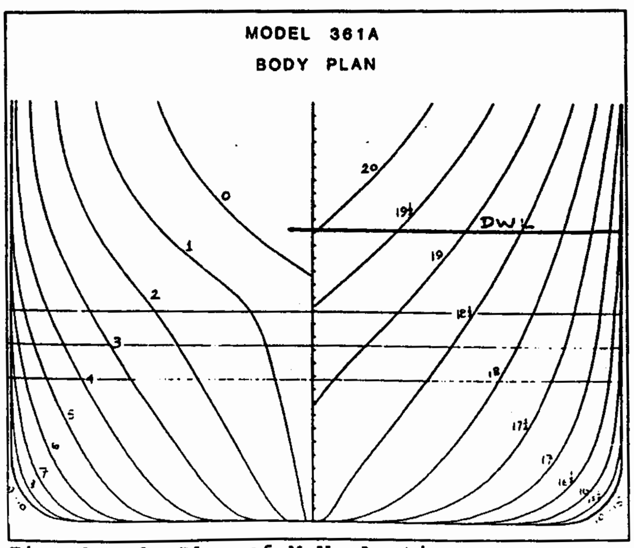

Fig. 3 Body Plan of M.V. Arctic

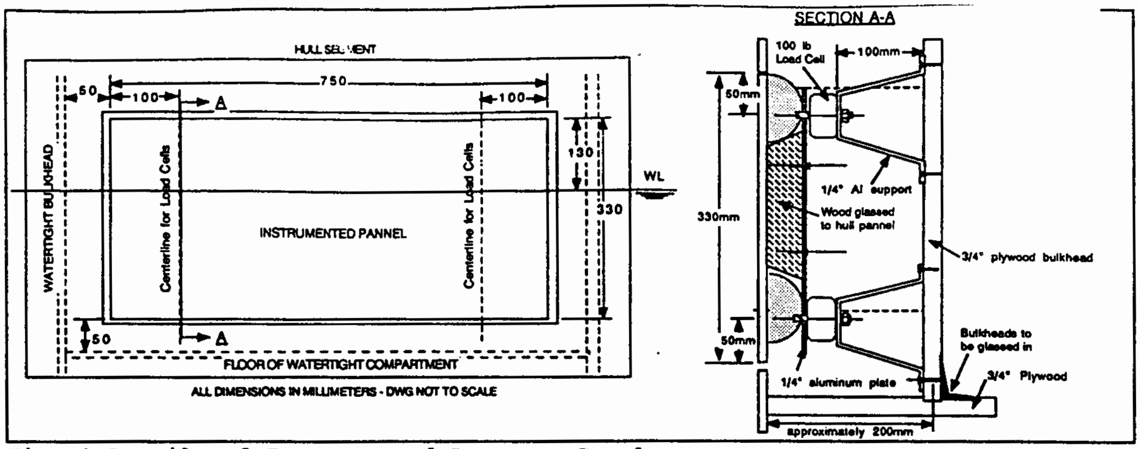

Fig. 4 Details of Instrumented Pressure Panel.

Four submersible load cell were located in each corner of the panel 550mm apart horizontally and 230 mm apart vertically. These were used to measure the normal force acting on the panel. The model was free to pitch, roll and heave. A rigid towing system restrained surge and measured resistance.

3.0 DESCRIPTION of TESTS

Typically, in each ice sheet, four different tests were conducted, see Figure 5. The first 15 metres was pre-swan to remove the strength dependent resistance, leaving only an ice clearing component. In the next 15 m section, 0.5 m wide strips of ice were removed from along each tank wall. This was to see if the confinement of the ice sheet by the walls had an effect on resistance. The next section was normal level ice to provide a base case for the other tests. The final section consisted of pressurized ice. The pressure apparatus was located between 50 metres and 62 metres. Saw cuts were made from the ends of the tubes towards the centre of the tank simulating a large ice floe. The model was towed up the tank at a constant speed of 0.282 m/s corresponding to a ship speed of 3 knots. When the model approached the 40 m mark the apparatus was slowly pressurized so that it reached target pressure when the bow of the model approached the 50 m mark. When the bow penetrated about 6 m into the pressure zone the pressurized ice began to move against the hull sides and the system pressure would begin to fall. The air supply valve was then opened to maintain the pressure.

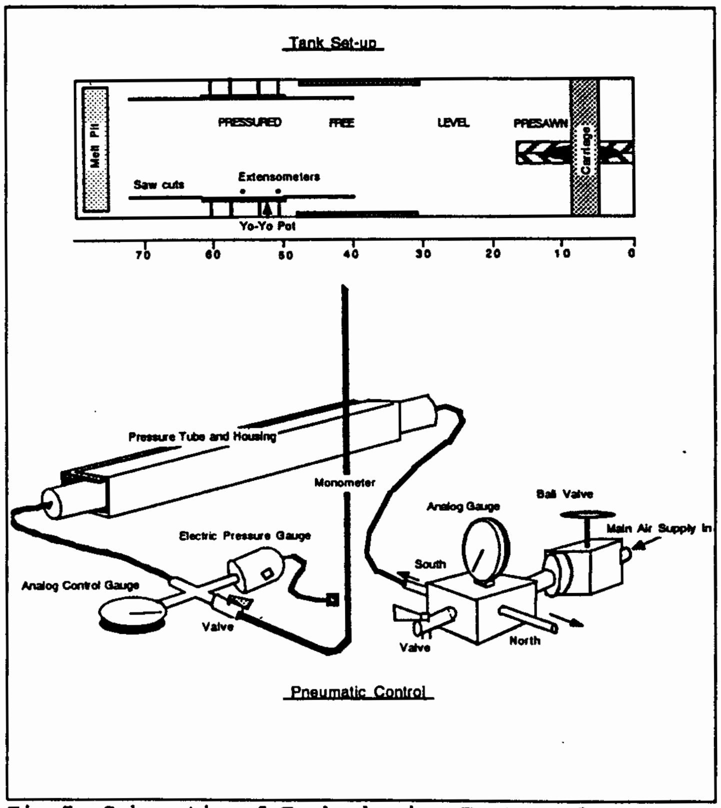

Fig 5. Schematic of Tank showing Tests and Pressure Control System.

4.0 RESULTS

The first step in the analysis was to remove the measured pre-sawn resistance from the total resistance measured in the remainder of the tank. To do this, the pre-sawn resistance from several tests was plotted against a term representing the buoyancy of the ice, and a least squares line was fitted, Figure 6. Thus, the pre-sawn resistance can be represented as;

$$R_{ps} = 4.0\,\Delta\rho_i\, g\, h_i\, B\, T + 21 \quad (6)$$

where $\Delta\rho$ is the density difference between water and ice, B is the model beam, and T is its draft. Using (6) the presawn resistance was subtracted from the total resistance during the remainder of the run. What remains is assumed to be a component of resistance that is primarily due to the flexural failure of the ice sheet and linearly dependent on ice strength. Figure 7 shows the mean resistance in level ice plotted against that where the ice had been sawn away from the walls to remove any possible confinement. It appears that the tank walls do not confine the ice in any way.

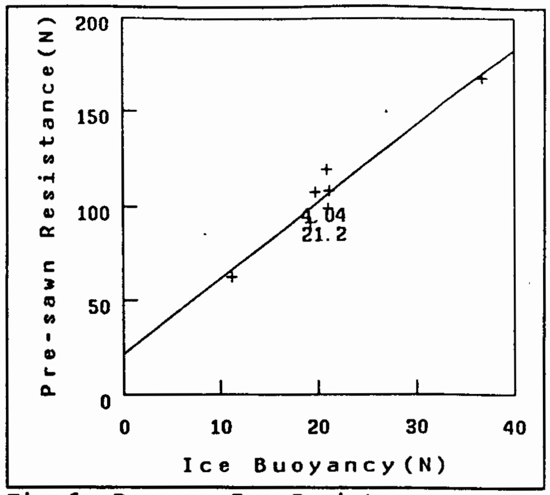

Fig 6. Presawn Ice Resistance vs Ice Buoyancy.

In total, eight tests were conducted with pressure, however, it was only during the last couple when it became clear what the actual process was. The final test was by far the most successful qualitatively as well as quantitatively due to more extensive instrumentation and better test procedures. We will focus our attention on the results of this test, but all of the tests except one revealed a substantial increase in resistance as the model passed through the pressured region. Typically, the increase in resistance was about 50%. Interestingly, in one test where no increase in resistance was recorded it was found subsequently that ice had formed in the main air supply line restricting flow to the apparatus. This meant the ice sheet, although initially pressurized, lost pressure during expansion due to the restricted air flow. This was significant in that it demonstrated that the compressive pre-stress had little or no effect on the breaking resistance. The initial lateral pressure in this ice sheet was about 15 kPa while the flexural strength of the ice was 60 kPa. Thus, if the pre-stress had an effect we should have seen about a 25% (15/60) increase in the breaking resistance, but none was observed. Since the ice had not moved into the channel during this experiment it was possible to back up the model in its own channel and test it moving in a existing channel under pressure similar to the tests of Kujala. As the ice moved against the model a substantial increase in resistance was observed.

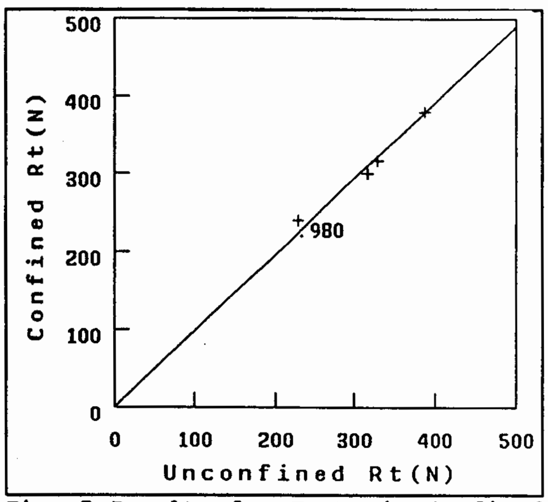

Fig. 7 Results for Tests in Confined and Unconfined Level Ice.

In Figure 8, the breaking component of resistance is plotted against bow position. On this same graph the sum of the four side panel load cells is also plotted, along with apparatus pressure and apparatus displacement. During tests in unpressurized ice the model appears to provide itself a channel of sufficient width so that there is little ice

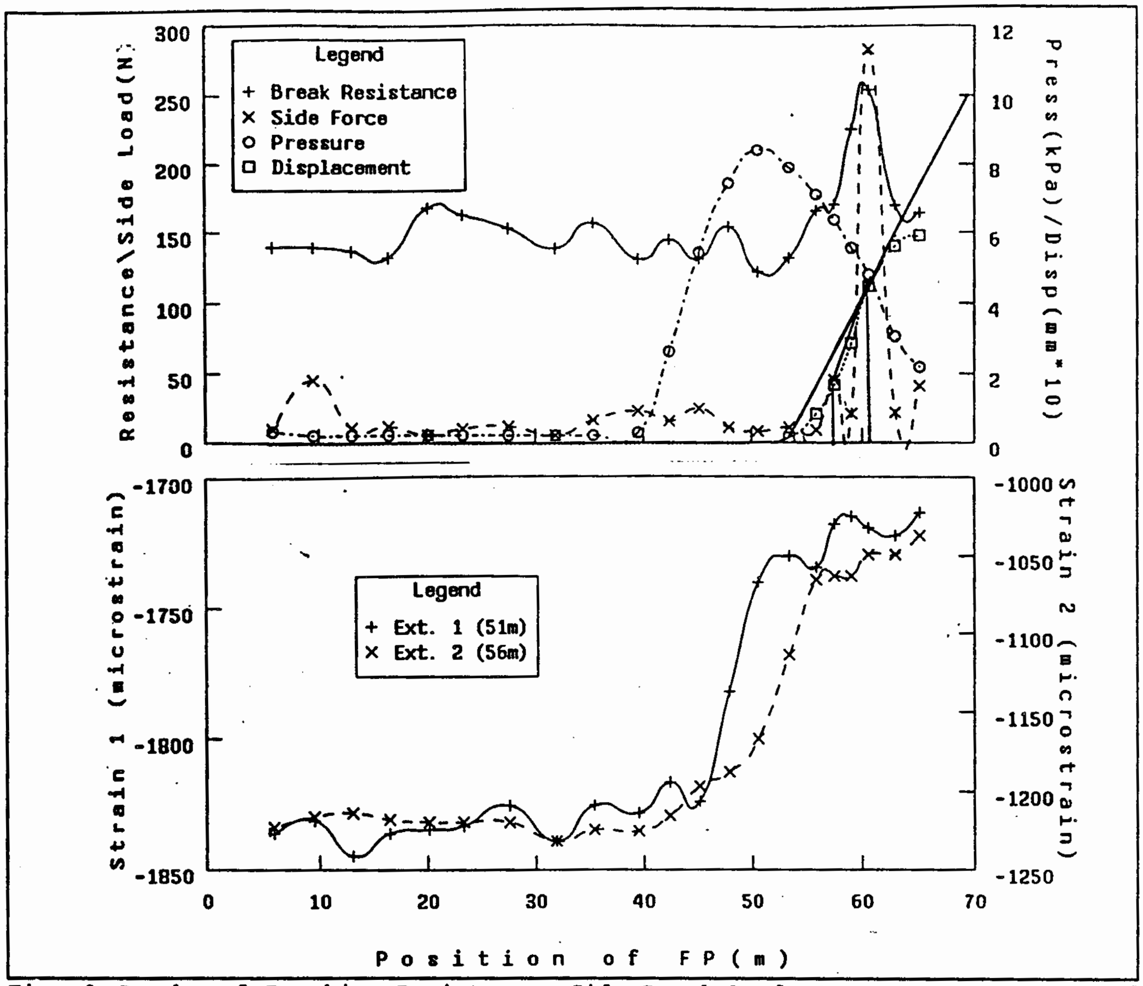

Fig. 8 Graphs of Breaking Resistance, Side Panel Load, Apparatus Pressure and Displacement, and Ice Strains with respect to Bow Position

pressure on the sides of the hull. This can be seen by examining the side panel load in Figure 8. The load is essentially zero for the unpressured part of the test. However, when the model penetrates deeply into the floe some cracks form in the floe and the ice begins to close on the model sides. When the bow reached 56 m, or 6 metres into the pressurized area, we can see from the apparatus displacement that the ice is beginning to move into the channel. When this happens the side loads rise sharply and the resistance correspondingly so. The maximum apparatus pressure during this test was about 8.4 kPa, corresponding to 19 kPa of lateral pressure, from (1). Even though this pressure represents about 50% of the 40 kPa flexural strength no increase in resistance was observed while the bow initially penetrated into the pressure region. Certainly, the ice was under pressure because both extensimeters recorded strains between 100 and 150 $\mu\varepsilon$. Using an effective elastic modulus of 100 MPa these would translate into normal stresses of 10-15 KPa which are reasonable. The resistance begins to increase when the bow is at 56 m. Also at this point the ice apparatus pressure begins to drop as the ice sheet begins to constrict the channel. When this happens the side panel force rises from near zero when the bow was at 56 m to 280 N at 61 m. The 100 N jump in resistance exactly corresponds to the increase in side panel load. The maximum in pressure on the panel is 280N/(0.75m x 0.05m) or about 7.5 KPa which is approximately equal to the ice pressure at this point. The side panel only represents about one-quarter of the parallel middle body on one side or one-eight of the total. If the 7.5 KPa pressure was seen uniformly over the entire middle body, and the friction coefficient was 0.1, then the expected increase in resistance would be about 225 N (ie. 7.5KPa x .05m x 0.75m x 8 x 0.1). In fact, the actual resistance increase, or load transmission factor, was only 45% of this. In the other tests load transmission factor varied from 60-100%. The rate of channel closure was found to be 3.8 mm/s(one side) which corresponds to .04 m/s full-scale. This value is quite reasonable when compared with the reported value of 0.1 m/s.

Using the concept of a transmission factor, C_l, the incremental resistance due to an inplane pressure, σ, is

$$\Delta R = 2\ C_l\ \sigma\ \mu\ L_e\ h_i \qquad (7)$$

where L_e is the effective length under pressure, h_i is the ice thickness and μ is hull-ice friction coefficient. If the in-plane pressure is taken as some percentage of the buckling pressure, say $C_\sigma \sigma_b$, then from (5)

$$\Delta R = 2\ C_\sigma\ C_E^2\ C_l\ \mu\ \rho_w\ L_e\ h_i^2 \qquad (8)$$

note C_E equals 11 the constant found in (5). (8) has the same basic form as the formulation presented in [8]. Putting in typical values from these experiments $C_E = 11$, $C_\sigma = 0.30$, and $C_l = 0.5$ then $2\ C_E^2\ C_\sigma\ C_l = 36$ compares favourably with the value of 45 given in [8].

5.0 CONCLUSIONS

In summary, in pressure free ice the hull provides for itself a channel of sufficient width so that lateral ice loads acting on the hull are generally small. The ice tank walls do not appear to confine the ice sheet sufficiently to affect the resistance. The primary source of resistance in pressurized pack ice appears to be due to the closing of the channel, leading to frictional resistance on the sides of the hull.

This study saw no evidence that the compressive pre-stress in the ice increased the force required to fail the ice sheet. This is probably because at the stem the tensile failure stress is more-or-less orthogonal to the compressive pre-stress. Once the channel begins to form the compressive stress are relieved because of the proximity of the free edge. Thus, in general, it maybe possible to mitigate the effect of lateral ice pressure on vessel resistance by application and maintenance of a low friction coating on the sides. Also, use of bow reamers would create a wider channel allowing the vessel to proceed before the ice pressure could fully develop.

It appears that the entire floe pressure can be concentrated on a relatively small area of the hull resulting in high local loads which may damage the hull. This problem seems to be most acute when the hull is exiting from a floe and there is no ice ahead of the model to withstand the pressure. This gives the resistance a cyclic nature; it is relatively low as the model enters the floe and reaches its maximum as the model exits.

If a vessel is experiencing difficulty and required to stop in pressured ice it may be prudent for it to just enter a large floe before stopping. The floe may be able to shield the vessel from pressure while stopped and facilitate subsequent breakout.

ACKNOWLEDGEMENT

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