Model-scale/full scale correlation in open water and ice for Canadian Coast Guard "R-Class" icebreakers

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ABSTRACT

Model scale data from the National Research Council of Canada, Institute for Marine Dynamics for the Canadian Coast Guard's R-Class icebreaker are compared with previous model tests and, more importantly, with three sets of full-scale ice trials data collected in 1978,1979 and 1991. In open water, good agreement between model and full-scale was found for bollard tests, and for self propulsion tests provided a roughness allowance of 0.0008 was used. In ice, good correlation was found with the 1978 tests when the ship was new and there was little snow cover, using a model hull/ice friction coefficient of 0.05. Good agreement with the later tests, 1979 and 1991, was also obtained with somewhat higher model/ice friction coefficients of 0.055 and 0.065. This is attributed to a deteriorating, and hence rougher, full-scale ship hull surface. The model tests showed that a change in friction coefficient from 0.03 to 0.09 causes a doubling of the delivered power. For the full-scale ship, it is suggested that relatively inexpensive localised hull maintenance in the shoulder area, where ice jamming occurs and hence hull/ice friction is important, could improve performance and reduce the chance of structural damage.

INTRODUCTION

The Canadian Coast Guard operates three R-Class icebreakers. These vessels have a power of 10,000 kW and carry an Arctic Class 3 rating. This paper examines the correlation between R-Class ship trials in open water and in ice, with measurements on a 1/20-scale model tested at the Institute for Marine Dynamics, St. John's, Canada. While the full-scale tests have been published in the literature (Williams et al., 1992; Edwards et al., 1979), the model tests and the correlation have not, except for internal IMD reports. This ship is probably the most extensively tested icebreaker, having been the subject of a round-robin test program organised by the International Towing Tank Conference (Tatinclaux et al., 1989). Open water model and full-scale tests were correlated. Level ice resistance and power was correlated using the concept of "correlation friction" analogous to the correlation allowance used in clearwater testing. In this paper, a correlation between full-scale and model scale tests in ice is shown in the following manner. First, full-scale open water bollard and speed-power trials are compared with corresponding model tests, from which it is shown that the model tests accurately predict the ship performance in open water, and a thrust deduction factor is determined from open water, overload, model tests. In general, a ship moving through an ice sheet is in an overload condition. Thus, the thrust deduction factor used to estimate level ice resistance should be derived from overload tests. While it would be preferable to measure the thrust deduction factor at full-scale, these tests are expensive to conduct because they require a second vessel to provide a tow load. An alternative is to use thrust deduction factors derived from model overload tests. Second, the full-scale trials in ice are discussed and the results summarised. Third, the model scale resistance tests in ice are described and equations for the resistance of the full-scale ship derived from the model test results. Fourth, the full-scale resistance in ice for the ship is then determined from the measured trials data and the thrust deduction factor mentioned above, and it is shown that the equation from the model tests is a good fit to the full-scale data when the friction of the ship's hull is considered.

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A paper describing this research has been accepted by the *Journal of Ship Research*. This version is, therefore, an outline of the work and a summary of the results.

FULL-SCALE OPEN WATER TRIALS

The performance of the *CCGS Sir John Franklin* was measured in open water during the winter of 1990, and additional speed-power trial data were collected during acceleration trials conducted off St. John's in May 1991. A hull roughness survey was also conducted on the Sir John Franklin in December 1989.

Bollard Tests

Bollard pull tests were conducted in Marystown by connecting a 100 m long, 3" diameter, wire rope from the bollard to the vessel aft towing winch.

Open Water Propulsion Tests

The open water tests in Conception Bay were run over a one-mile course with, and against, the wind. The wind was initially 25 knots but fell during the trials to 13 knots. The resulting ship speed was corrected for wind. No correction for current was made. For the later test the ship had to run abeam of the wind to avoid icebergs and a 0.25 to 0.5 knot current was present. There were only five tests with shaft speeds of about 180 rpm. Corrections were made for current but not wind.

MODEL OPEN WATER TESTS

Bollard Pull

A 1:20 scale model was outfitted with scale propellers. Two bollard pull experiments were conducted with the towline connected to the stern at an elevation corresponding to the aft towing winch. Fig. 1 shows good agreement between measured bollard pull and that predicted by the model tests.

Resistance and Self Propulsion

Resistance and self-propulsion tests were conducted on the model and a form factor, *(1+k)*, was determined by the method of Prohaska (1966).

$$C_{TS} - C_{TM} = (1+k)(C_{FS} - C_{FM}) - C_A \qquad \ldots \quad 1$$

where C_A is the ship hull roughness correlation allowance (assumed zero for the model) and the ship hull roughness correlation allowance, C_A, was varied from 0.0004 to 0.0008.

Figure 2 shows the comparison of total shaft power predicted from the model tests and those measured on the ship. The agreement is good when a correlation allowance of 0.0008 is used. The acceleration tests conducted in May 1991 contained scatter in the speed measurements. This may have been due to the tests having been conducted across the prevailing wind, to avoid some icebergs, which may have led to excessive rudder angles being used to maintain course. Similar good correlation was found for thrust and shaft speed when a 0.0008 correlation allowance was used.

Model Overload Experiments

The model was outfitted as for the self-propulsion tests except the drafts were altered to correspond to the full-scale speed-power trials in ice. The model was tested at four speeds corresponding to ship speeds of 4, 7, 10, and 13 knots.

Model overload thrust, torque and towing force were reduced to non-dimensional coefficients and plotted against advance coefficient. These curves were then smoothed by fitting second order polynomials. The thrust-deduction factor, *(1-t)*, was calculated for the model using equation 2.

$$(1-t) = \frac{F_D + R_{TM}}{T_M} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad 2$$

where F_D is the tow force, R_{TM} is the open water resistance of the model, and T_M is the total thrust of the model (port plus starboard). The total model resistance, R_{TM}, was calculated by interpolating the model resistance curve at the appropriate model speed. The model thrust, T_M, and towing force, F_D, were calculated from K_T and K_{FD} curves respectively.

FULL-SCALE LEVEL ICE TRIALS

Several full-scale speed-power icebreaking trials have been conducted with this ship class. The *CCGS Pierre Radisson* was tested soon after delivery in 1978 during transit in the Central Arctic, and in February of 1979 further tests were conducted in thinner but stronger ice in the St. Lawrence River (Edwards et al., 1979). Two probes were made by *CCGS Sir John Franklin* into Lake Melville in 1980 (Michailidis, 1980). Finally, in February 1991, further tests were conducted on this ship in Notre Dame Bay, Newfoundland (Williams et al., 1991; Williams et al., 1992). Together these trials constitute a large body of data describing the performance in ice for this hull form.

MODEL LEVEL ICE RESISTANCE

The object of this section is to derive equations for the level ice resistance from the model tests, which can later be used to predict the resistance of the full-scale ship. Model level ice resistance tests cover a wide range of speed, ice thickness and strength as well as two hull-ice friction coefficients. We follow the IMD standard method of analysis (IMD Standard Test Method, 2000) which involves dividing the total ice resistance, R_T, into four components; open water, R_{OW}, ice buoyancy, R_B, ice clearing, R_C, and ice breaking, R_{BR}

$$R_T = R_{OW} + R_B + R_C + R_{BR} \qquad \ldots\ldots\ldots\ldots \quad 3$$

Details will be outlined in the presentation and are given in full in the paper in press. The end result is two equations:-

$$R_I = 0.90\,F_h^{-0.739}\rho_i B h_i V_M^2 + 1.08\,S_N^{-1.672}\rho_i B h_i V_M^2 + 1.31\Delta\rho g h_i BT \quad \ldots\ldots \quad 4$$

$$R_I = 2.03\,F_h^{-0.971}\rho_i B h_i V_M^2 + 2.19\,S_N^{-1.579}\rho_i B h_i V_M^2 + 2.67\Delta\rho g h_i BT \quad \ldots\ldots \quad 5$$

These equations 4 and 5 give the total level ice resistance for the low (0.03) and high (0.09) friction model respectively. These equations were derived entirely from model tests, can now be used to give the resistance of the full-scale ship at the conditions of the full-scale trials. Fig. 3 shows that these equations describe the model test data well.

CORRELATION BETWEEN MODEL AND SHIP IN LEVEL ICE

The correlation was made by comparing measured ship resistance, and predicted ship resistance from equations 4 and 5, for the three trials, and also by a comparison of measured and predicted delivered power for the 1979 trials.

Ship Resistance

First, we calculated the resistance in ice for the full-scale ship from the trials data. The thrust required in ice, T_I , was first calculated by subtracting the open water thrust from the total ship thrust measured in ice, both of which were measured on the trials. The resistance in ice, R_{IS} , for the ship was estimated by multiplying the ice thrust by a thrust deduction factor derived from the above model overload tests.

$$R_{IS} = (1 - t)T_I \qquad \ldots\ldots\ldots\ldots\ldots \quad 6$$

This was then compared to the ice resistance from the model tests given by equations 4 and 5 above.

Correlation Results

Figures 4, 5 and 6 show the ice resistance for the full-scale ship as calculated from the model tests by equations 4 and 5, against that estimated from the full-scale ship trials. Fig. 4 is for the Arctic 1978 trials of the *Pierre Radisson*, Fig. 5 for the February 1979 trials of the *Radisson* and Fig. 6 is for the 1991 trials of the *Franklin*. For these figures, the line of perfect correlation clearly lies between the two model friction coefficients of 0.03 and 0.09. If it is assumed that the influence of friction on resistance is linear over this range, an estimate of the "perfect correlation friction" may be obtained from these figures. From Fig. 4, it is estimated that the perfect correlation friction is 0.05 for the 1978 Arctic *CCGS Pierre Radisson* trials. Figure 5 shows a perfect correlation of 0.055 for the St. Lawrence trials of the *Radisson* in February 1979.

Finally, in Fig. 6 results are presented for the February 1991 trials of the Franklin in Notre Dame Bay. In this case, it appears that the 0.09 friction model results are only about 15% high, resulting in a perfect correlation friction of 0.065.

CONCLUSIONS AND DISCUSSION

1. For the bollard pull tests, a good correlation existed between model and ship. The model tests predicted accurately the bollard pull, thrust and power.

2. The model self-propulsion open water tests also correlated well with the ship if a roughness allowance of approximately 0.0008 was used. This is close to the value of 0.0009 which is obtained by Bowden and Davison's (1974) method, for a hull with a surface roughness of 316 µm, but somewhat higher than the 0.0005 given by Holtrop and Mennen's (1978) equation.

3. The ice resistance model tests agreed reasonably well with the previous ITTC round robin series of tests. The icebreaking component of the resistance was shown to be 67% of the total resistance at 0.5 m/s.

4. Models with a hull-ice dynamic friction of 0.05 gave good correlation for a new R-Class hull in snow free ice, as was the case for the 1978 trials. The winter 1979 St. Lawrence trials required a slightly higher coefficient, between 0.055 and 0.06, but this may be due at least partially to a deteriorating hull condition. This relatively small change in correlation friction between the 1978 and

1979 trials demonstrated that the model tests predicted ship performance over a wide range of ice conditions. For the trials in 1991, a higher model hull-ice friction coefficient of about 0.065 was required. We believe that this is because the hull, although coated with INERTA, was rough due to poor paint application and the condition of the underlying hull. These results point to a gradual decline in hull condition over the 13-year period. It is worth noting that Enkvist and Mustamaki (1996) also found good model to full-scale correlation for the *Protector* with a friction coefficient of 0.05.

5. The model tests demonstrated that there was almost a doubling of delivered power when going from a hull-ice friction of 0.03 to 0.09. This is shown by the slope of the correlation lines in Figures 4-6 that double between the two friction values. Even if a hull-ice friction of 0.05 represents the best finish that can be achieved on a new hull, there appears to be almost a 50% increase in required power if it is allowed to deteriorate to a 0.09 finish. This represents a 10% increase in power for every 0.01 change in friction coefficient.

6. The R-Class hull form may be sensitive to friction because of the jamming of ice near the shoulders. As the broken ice cusps rotate normal to the hull they become jammed between the hull and the intact ice sheet. This jamming occurs only near the shoulders where the beam is approaching its maximum. Since the flare angles can be steep, in excess of 80 degrees, hull ice friction can play a significant role in inducing jamming. Consider Figure 7 which shows an ice floe trapped between the hull and intact ice sheet. The hull is exerting a normal force, N on the floe, giving rise to a frictional force μN opposing motion. The hull then exerts a downward or clearing force on the ice piece of:-

$$F = N(\sin\alpha - \mu\cos\alpha) \quad \ldots\ldots\ldots \quad 7$$

where α is the angle of the hull segment measured relative to the vertical. It can be seen from equation 7 that if μ becomes equal to tan α then there is no net clearing force and failure must occur by means other than flexure, typically crushing. The result would be very high local loads, leading to increased resistance and the possibility of structural damage. For example, if the hull-ice friction is 0.1, then a local flare angle of 84^{o} or greater results in the ice becoming jammed regardless of normal load. For a hull-ice friction of 0.2, the critical flare angle is reduced to 79^{o}. It is possible, therefore, that inexpensive localised hull maintenance in this shoulder region would greatly improve the performance of the ship and reduce the chance of structural damage.

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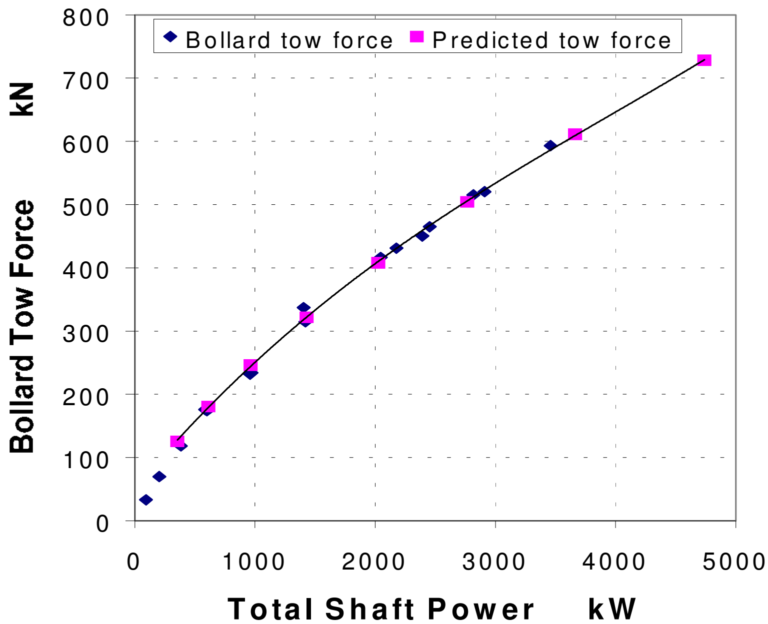

Fig. 1 The results of the Marystown bollard tests showing excellent correlation between the full-scale data (diamonds) and the predicted data from the model tests (squares). The line is a third order polynomial fit to the model data only.

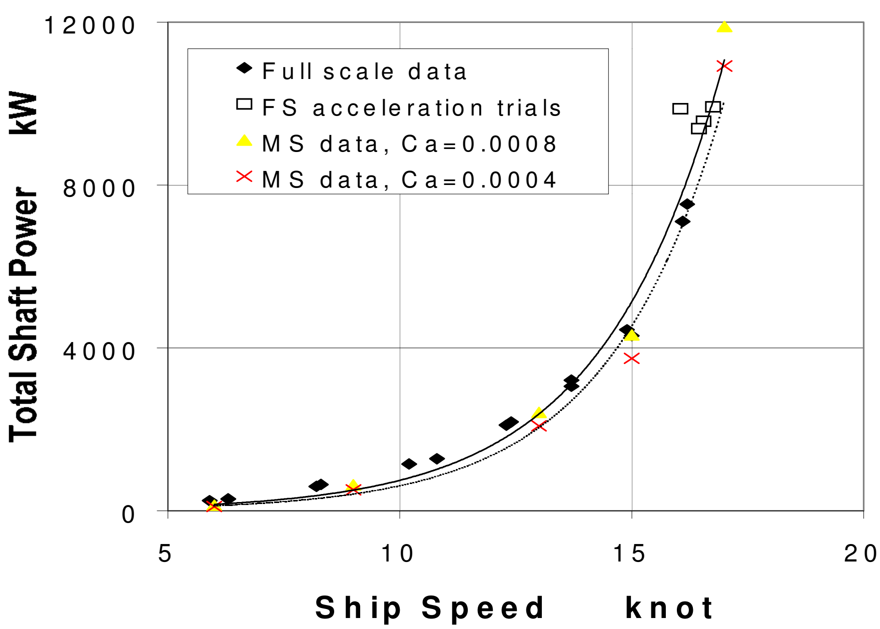

Fig. 2 Open water total shaft power for the ship and model data for two correlation allowances, showing a good exponential fit with an allowance of 0.0008 (sea water 35 ppt, temp. 2°C, form factor 1+k=1.4).

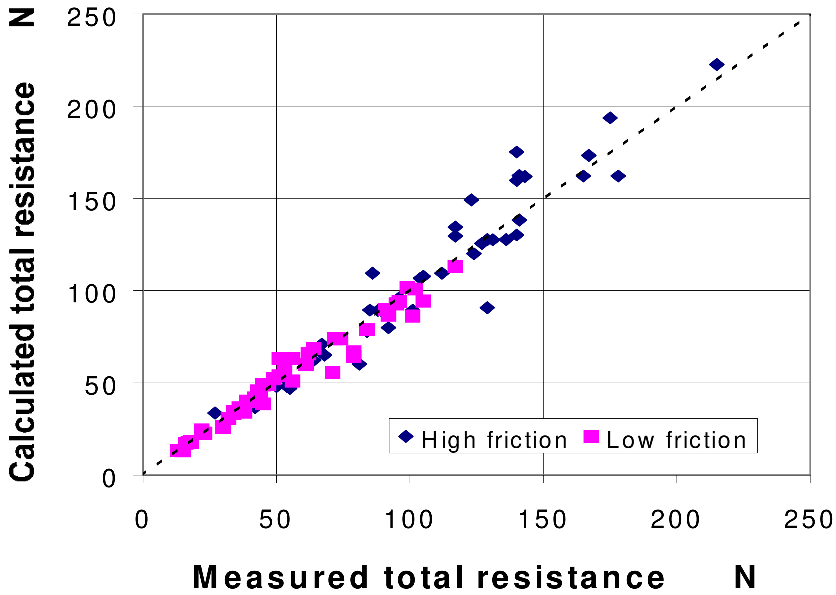

Fig. 3 Comparison of calculated total resistance from equations 4 and 5 and the experimentally measured values. The dotted line is be a 1:1 fit and shows that these equations do indeed describe the data well.

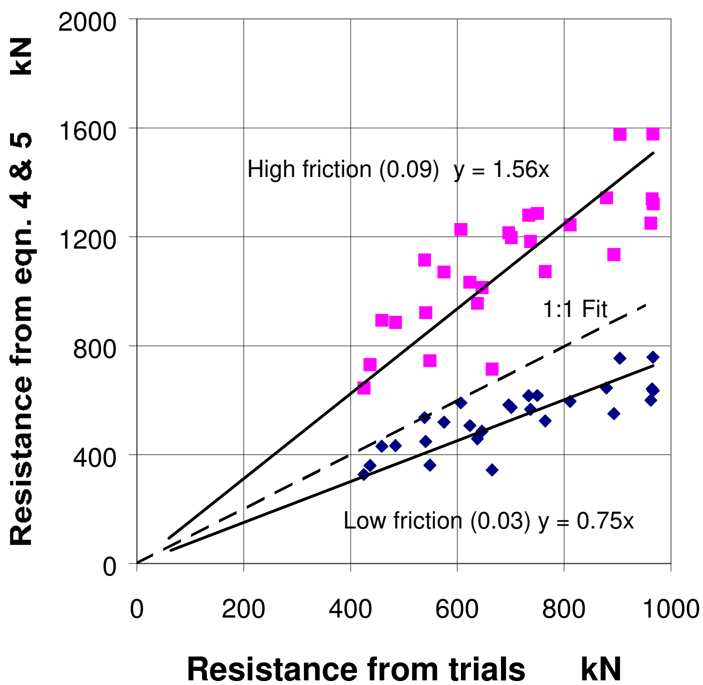

Fig. 4 Comparison for the CCGS *Pierre Radisson* during the August 1978 trials. The data points are calculated from equations 4 and 5 for conditions corresponding to the full-scale trials. The line of perfect correlation (1:1) lies approximately midway between the high (0.09) and low (0.03) friction model test results, implying a perfect correlation of 0.05.

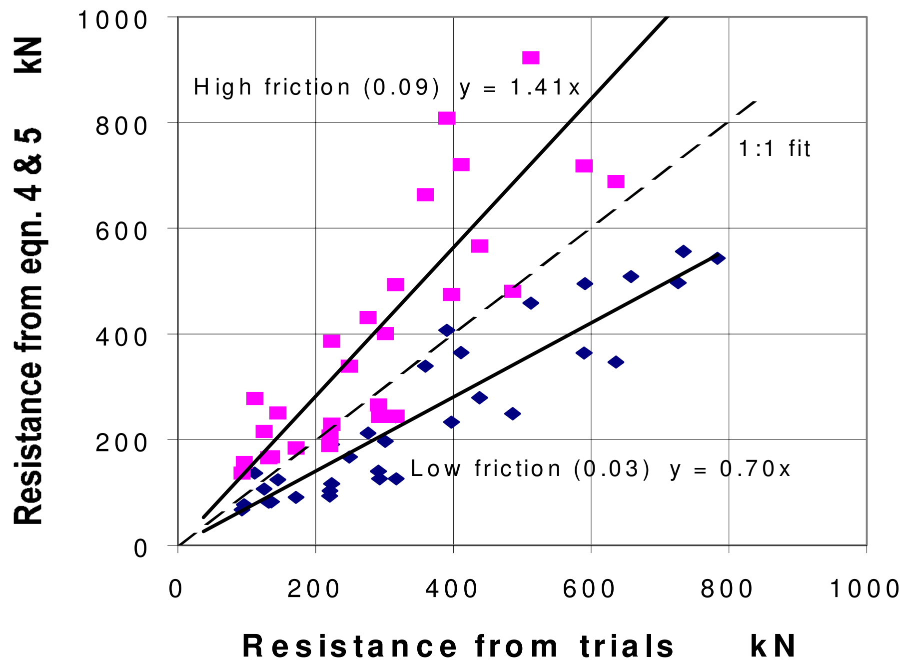

Fig. 5 Comparison of ship resistance for the *CCGS Pierre Radisson* during the February 1979 trials, with resistance calculated from the model equations 4 and 5, implying a perfect correlation friction of 0.055.

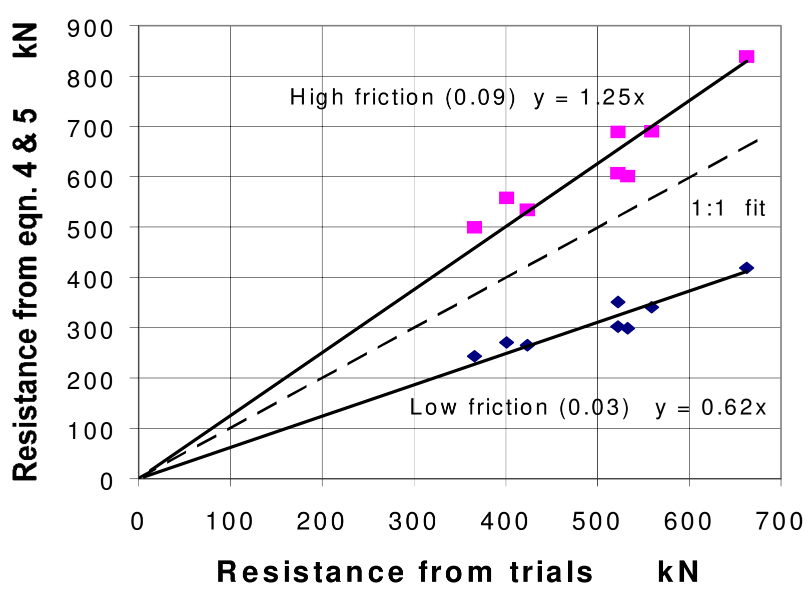

Fig. 6 Results for the *Sir John Franklin* trials in Notre Dame Bay in February 1991. The 0.09 friction model results are only about 15% high, implying a perfect correlation friction of 0.065.

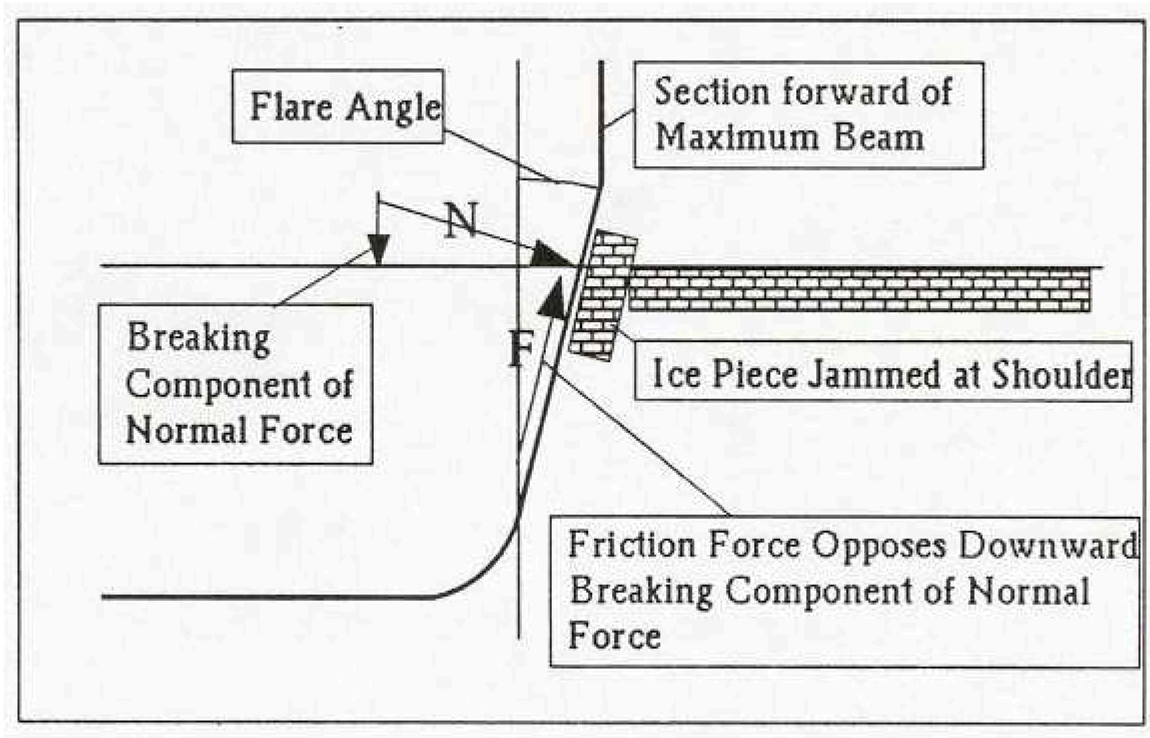

Fig. 7 Diagram to illustrate an ice floe trapped between the hull and intact ice sheet.