Bulker and Tanker in Open Water and in Ice

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1. Introduction

During the last years a high number of newly constructed large bulk carriers and tankers were put into service and this boom is likely to remain unchanged throughout the next few years. Due to the worldwide increasing demand for oil and oil products on the one hand and the steadily rising oil price on the other hand oilfields located in – at least during winter time – ice covered regions are explored. Thus, the growth of the fleet of ice going tankers is higher than the growth of total tanker fleet. The most vessels with ice class notification are tankers for the transport of crude oil and refined products from the Baltic region.

For the crude oil transport in the Baltic Sea Aframax tanker with about 110,000 DWT are found to be most suitable. For transport of refined products in ice the most often ordered tankers are of Handymax size with dead-weights between 40,000 DWT and 50,000 DWT.

Since the new Finnish-Swedish Ice-Class Rules became effective in year 2002 new build ships, intended to sail in the Baltic Sea during the winter season, have to install a main engine power much higher than required for service in open water. In combination with slow speed two-stroke diesel engines fixed pitch propellers can not be designed to meet the requirements of these two very different operation conditions.

In this paper some hydrodynamic aspects of the operation of tankers and bulkers in open water and ice are discussed. Concepts to solve problems like yaw instability in open water and propulsion arrangements, insufficient to fulfil new regulations introduced by Scandinavian authorities, are presented.

2. Bulkers and Tankers Today

In the shadow of the spectacular development of very large containerships with capacities exceeding 8000 TEU the market for tanker and bulker newbuildings was stable at a high level throughout the last three years. In total oil tankers, product tankers, chemical tankers, gas tankers and bulk carriers share about 60% of the world wide new ship production (in cGT). About 40% of the vessels are constructed in Korea, about 30% are build in Japan and about 10% of the vessels are produced in China, with an increasing tendency.

Placed orders for new ships are shared as follows: Korea about 35%, Japan about 40% and about 15% in China (mid of 2004).

In total more than 90% of the tankers and bulk carriers are currently constructed by Asian shipyards. European maritime industry concentrates on the production of small and special purpose tankers, ship equipment and propulsion systems as well as on consulting and design services offered by consulting engineers and ship model basins. It is common practice that the hydrodynamic design for a newly developed tanker or bulker is validated and optimised by a European ship model basin on behalf of Asian shipyards.

At HSVA the number of investigations for tankers and bulkers increased throughout the last years, although only very few ships of these types are build by German shipyards nowadays. This tendency was influenced positively by the world-wide well known competence of HSVA regarding the performance and evaluation of model tests in the ice basin, which became more important since the Finish and Swedish Maritime Authorities changed they regulations in 2002.

Today tankers are built with capacities up to about 300,000 DWT. The largest bulk carrier are built with capacities up to about 200,000 DWT.

Tankers

For tankers currently the following classes are distinguished in the shipbuilding and shipping business:

Panamax	Able to pass through the Panama Canal Capacity about 60,000 to 80,000 DWT
Aframax	"Afra" means "average freight rate assessment" Best ship size for ice-going tankers in the Baltic Sea Capacity about 80,000 to 120,000 DWT
Suezmax	Able to pass through the Suez Canal Capacity about 120,000 to 200,000 DWT
VLCC	Very Large Crude Oil Carrier Loading capacity about 200,000 to 300,000 DWT The draught of these ships is limited by the water depth of the Malacca Straight. Consequently they are also called Mallacca-type.
ULCC	Ultra Large Cruide Oil Carrier Loading capacity above 300,000 DWT

Special Purpose Tankers

The fleet of the special purpose tankers is distinguished from the crude oil tankers by the cargo they carry:

Product Tankers:	Transport of refined oil products	
	Handysize	about 25,000 to 40,000 DWT
	Medium Range (MR)	from 40,000 to 50,000 DWT Best ship size for ice-going tankers in the Baltic Sea
	Large Range (LR)	from 50,000 to 90,000 DWT
Chemical Carriers:	Transport of chemical goods	
	Seldom larger than 20,000 DWT	

Liquefied Natural Gas Carrier (LNG):
Up to 150,000 m^3 ships are in operation

Liquefied Petroleum Gas Carrier (LPG):
Up to 80,000 m^3 ships are in operation

Dry Bulk Carrier

Large dry bulk carriers are distinguished into three classes:

Handysize	Capacity about 15,000 to 50,000 DWT
Handymax	Capacity about 35,000 to 50,000 DWT
Panamax type	Capacity about 70,000 DWT Limited in beam and draught
Cape size type	Capacity up to above 150,000 DWT Unable to pass the Suez Canal, have to sail around the **Cape** of Good Hope

3. Open Water Hydrodynamics

Tankers and bulk carriers are mainly characterised by they high block-coefficient often close to 85% (see Fig. 1).

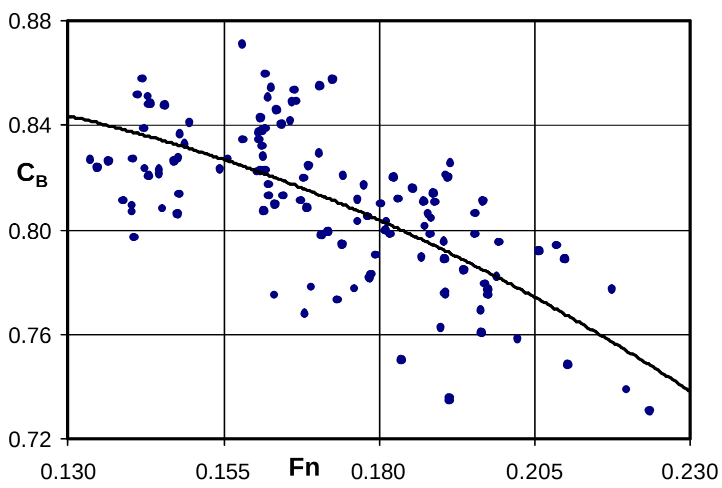

Fig. 1: Block-coefficient vs. Froude-number; Source: HSVA-Database

In order to ensure a sufficient water flow to the propeller an extremely forward position of the longitudinal centre of buoyancy (LCB) is required (see Fig. 2).

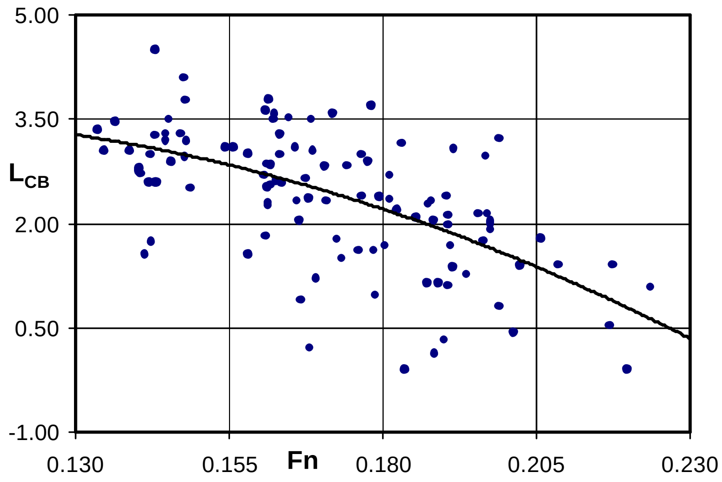

Fig. 2: Position of LCB vs. Froude-number; Source: HSVA-Database

The fore bodies of modern tanker and bulk carriers are characterised by a bulbous bow offering good performance for the fully loaded and the ballast condition. Fig. 3 shows a typical example for such a bulbous bow design.

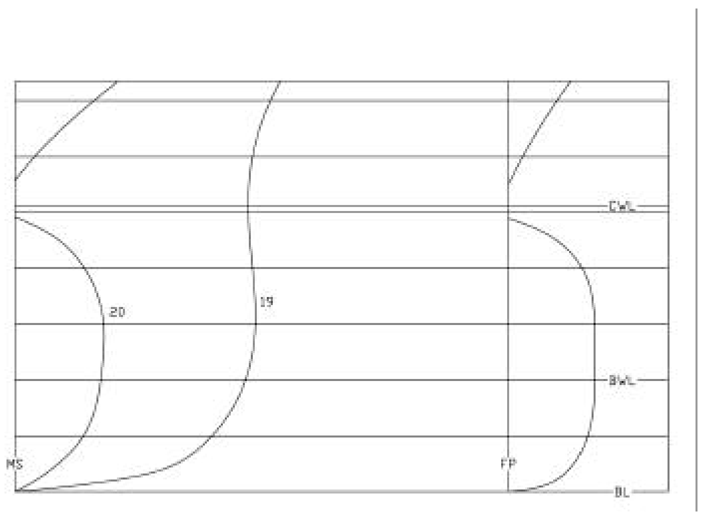

Fig. 3: Bow design of a modern VLCC

The aft body must be designed thoroughly to meet two requirements: Firstly the water flow to the propeller must be sufficient, and secondly the vessel must comply with the IMO regulation regarding yaw stability. Fig. 4 shows a typical example for such an aft body design.

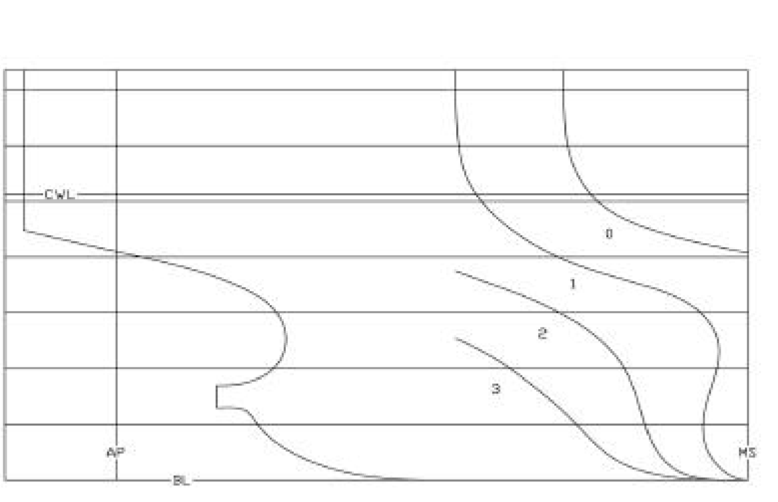

Fig. 4: Aft body design of a modern VLCC

Since tankers and bulk carriers sail at relatively low speeds and since, due to the high draught, the optimum propeller diameter can be realised they have only in a few cases problems with propeller cavitation. Rudder cavitation almost never occurs.

The following table presents some exemplary principal particulars of three different tankers and bulk carriers:

Ship Type			**VLCC**	**170K B/C**	**Aframax**
Capacity			**300,000 DWT**	**170,000 DWT**	**110,000 DWT**
Ship	Lpp	[m]	325.00	280.00	235.00
	B	[m]	60.00	45.00	43.00
	T	[m]	20.50	16.00	12.00
	C_B abt.	[-]	0.80	0.84	0.82
	LCB abt.	[% of Lpp]	+3.0	+2.5	+3.0
Propeller	D_P	[m]	10.30	8.00	7.10
	z	[-]	4	4	4
	A_E/A_O	[-]	0.45	0.60	0.50
Main Engine	NOR	[kW]	23,000	12,300	11,000
	n	[rpm]	70	86	100
Service speed	V	[kts]	15.7	15.2	15.2
	Fn	[-]	0.143	0.149	0.163

4. Yaw Stability of Full Blocked Ships

One of the few hydrodynamic problems of full-block ships in open water conditions can be the yaw stability. Especially ships with L/B-ratios lower than 6 often lack sufficient yaw stability. Smaller vessel are endangered more than larger ones. Already in 1977 a simple method was published (Clarke, 1977), still well suited for a quick check of the yaw stability in the early project phase (see Fig. 5).

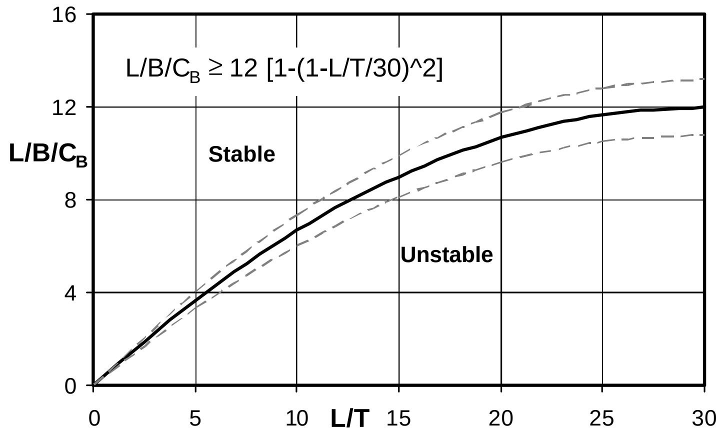

Fig. 5: Diagram for the estimation of the yaw stability of ships, Source: Clarke et al., 1977

If the point defined by the $L/B/C_B$-value and the L/T-value is above the boundary curve shown in Fig. 5 the ship is stable in yaw. If the point is below the boundary curve a conventional ship will be instable in yaw. In such a case special measures in the aft body must be taken in order to ensure sufficient yaw stability. These measures in order of decreasing effectiveness are:

1. Lengthening of the waterlines below the propeller shaft with sharp edges at the end and the bottom.
2. Modification of the bilge design far forward of the propeller aiming the generation of bilge vortices transporting water to the region above the propeller shaft (see Fig. 6).
3. Increasing the rudder blade area.

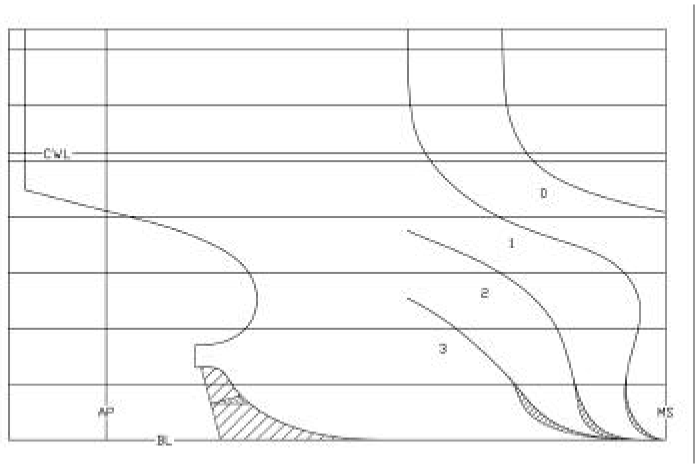

Fig. 6: Aft body with integrated yaw stability fin and bilge vortex generating ship form

5. Tankers Operating in Ice

Due to the increasing amount of crude oil exported from Russia a quickly growing fleet of tankers is employed in seasonally ice-infested waters like the Baltic Sea, the southern Barents Sea (Petchora), and the waters around Sakhalin (Far East). Many new crude oil tankers with ice classes are presently under construction mainly in Korea, but also on European ship yards. Except for the tankers which will serve the terminals in the Petchora area in the winter month, the majority of these newbuildings are of conventional hull form design without particular ice breaking capability (qualities). The steel structure of these vessels is strengthened and the main engine power is increased according to the Baltic Ice Class Rules 1B or even 1A. Presently the typical size of these tankers is the so-called Aframax-Tanker. The following part of the present paper will focus on the main engine power requirements and problems with the selection of an adequate propulsion system.

5.1 Power Requirement according to the new Finnish Swedish Ice Rules.

The background for the power requirements stated by the Finnish/Swedish Authorities is based on the presumption that sufficient icebreaker assistance can be provided so that the commercial vessels don't need to *break* ice by their own. Rather, a good ice transit performance in old ice clogged channels is demanded by the Finnish and Swedish Authorities. This is reflected by the latest issue of the Baltic Ice Class Rules, where a minimum speed of 5 knots in a brash ice channel of defined thickness is considered as satisfactory, whereby the thickness of the brash ice in the channel increases with the Ice Class. The new rules imply that a satisfactory brash ice performance identifies the vessel also for facile operation in convoy with icebreakers

Based on model test series and full scale surveys, theoretical formulas have been developed in Finland (by the Helsinki University of Technology). These semi-empirical formulas determine the brash ice resistance of a candidate hull design, based on the main dimensions and a few hull form parameters: These are:

- waterline angle at B/4
- buttock angle at B/4
- bow length
- parallel midbody length
- L, B and T

The power requirement is derived from the calculated brash ice resistance taking into account the diameter and the number of propellers. A allowance of about 10% is granted by the FSIR for a controllable pitch propeller compared to a fixed pitch propeller. The compliance with the Ice Class Rules has to be proven for both fully loaded (summer fresh water load line) and ballast conditions. For simplicity reasons this paper will only focus on the loaded condition.

When the new ice class rules are applied to the typical size of vessels operated in winter in the northern Baltic, in most of the cases the required power does not much differ from the power calculated according to the former Ice Class Rules. However, when the new Baltic Ice Class Rules are applied to large tankers with their typical full bow forms, U-shaped frames and fixed pitch propeller, the main engine power which is needed to fulfil e.g. Ice Class 1A turns out to be about twice as high as the power needed for a reasonable open water speed.

Fig. 7 shows the calculated brash ice resistance and Fig. 8 the calculated power requirement for a number of the ice classed tankers of different size, which are presently under construction. Except for tanker No. 3 they can be classified into the three size categories: Panmax, Aframax and Suezmax.

For those tankers, where the resistance and the power requirement in calm water at a speed of 15.5 kts are available, these values have also been plotted for comparsion in Fig. 7 and Fig. 8, respectively. In Fig. 8also the MCR of designated main engines is shown. While for the smaller tankers (Panmax) the MCR covers at least the power requirement of the Ice Class 1 B, for most of the larger tankers (Aframax & Suezmax) the MCR does only match with the power requirement of the lowest Ice Class, 1 C.

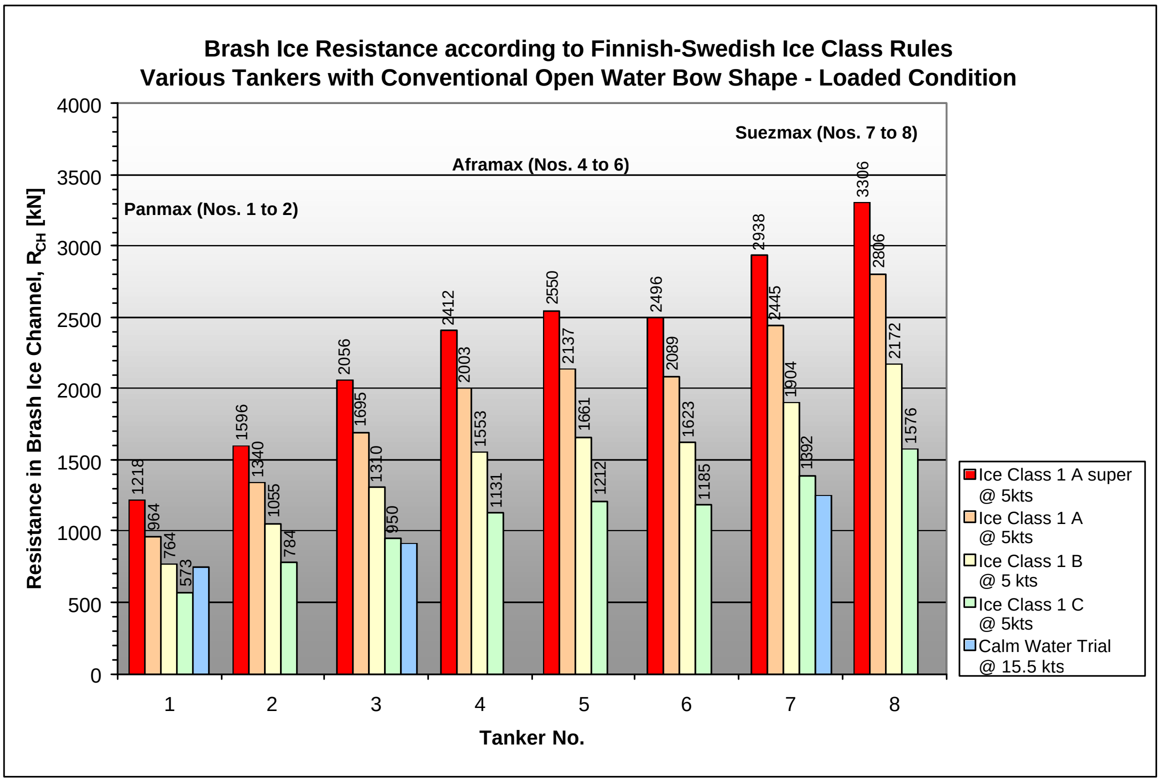

Fig. 7: Brash Ice Resistance according to FSIR

Although tanker nos. 7 and 8 have rather similar main dimensions, the resistance in the brash ice channel as calculated according the FSIR rules differs significantly for these two vessels. Only one part of the difference is caused by 2 m more breadth in the case of tanker no. 8. However, the major part of the resistance difference results from waterline and buttock angles at B/4 , which are assessed by the rule's formulas as more or less favourable regarding the resistance in brash ice channels.

How sensitive the calculated brash ice resistance (according to FSIR) and the corresponding power requirement react on these two parameters, shall be demonstrated by a parameter. For this purpose the waterline angle at B/4 was varied from 30 to 40 deg, which covers the typically range of up-to-date tanker bow forms. The buttock angle was varied from 90 deg (vertical) to 50 deg, which is deemed a feasible range for tankers. In order to simplify the variation, it was assumed that the waterline and buttock angles can be varied within certain limits without changing the bow waterline area and the length of the parallel midbody, which are the two other hull parameters effecting the brash ice resistance calculation. The influence of the waterline and buttock angles on the calculated resistance is shown for an Ice Class 1 A Suezmax tanker in Fig. 9.

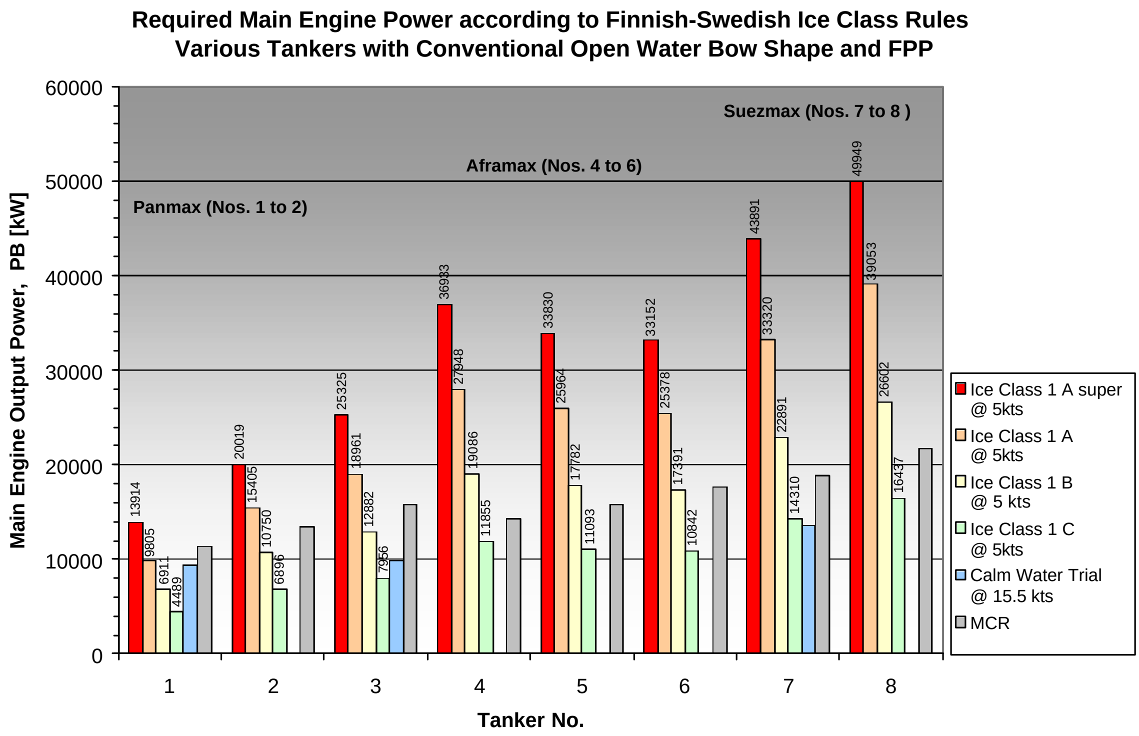

Fig. 8: Required Main Engine Power according to FSIR

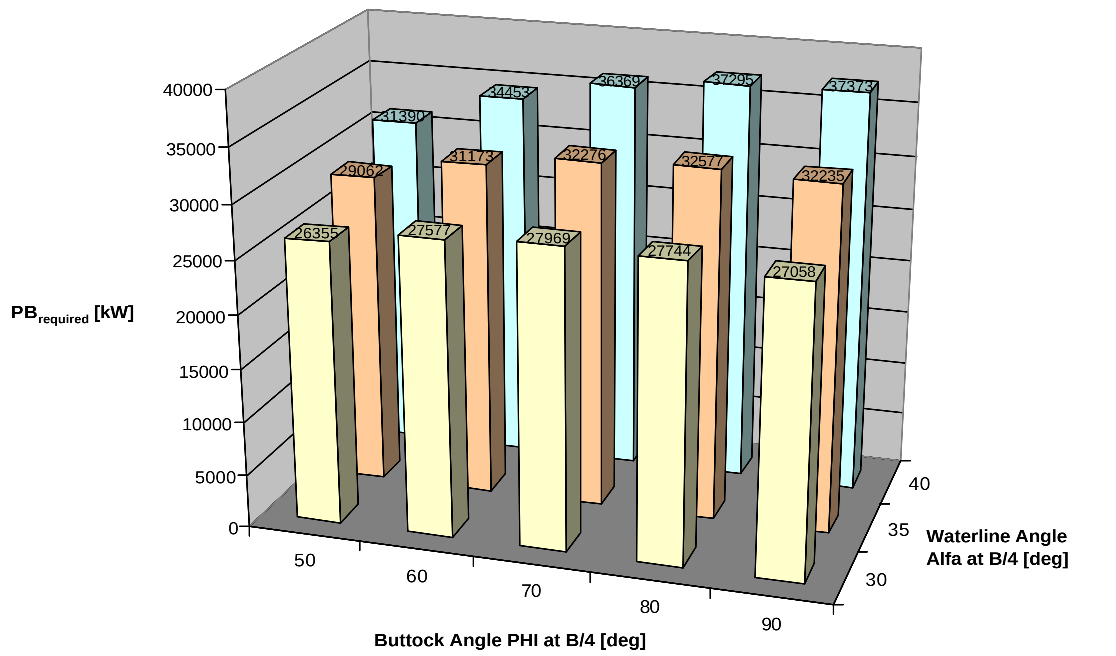

Fig. 9: Influence of the Buttock and Waterline Angle on the FSIR Power Requirement

From Fig. 9 the following tendencies can be read: The brash ice resistance calculated according to the Ice Class Rules clearly decreases with decreasing waterline angle. For variable buttock angles the resistance (power) calculation shows various maxima depending on the waterline angle. This means for the hull form designer that by means of local hull form modifications and the selection of a certain buttock and waterline angle combination the calculated resistance and correspondingly the power, which is needed to fulfil the rules, can be minimized.

5.2 Assessment of the Ice Resistance and Power Requirement in Brash Ice by Model Tests

Very soon, ship designers involved in Ice Class tanker projects found out that a hull form optimisation as mentioned above is insufficient to meet the power requirement of the Ice Classes 1 B with a main engine power which is needed to obtain a speed of 15.5 to16 knots in calm water. Thus, the option (stated in the FSIR) to prove the transit performance in brash ice channels by model tests was chosen for a number tanker projects.

In Fig. 10 and Fig. 11 the brash ice resistance and the engine power of the sample tankers determined in model tests are compared with the resistance and the power calculated by the FSIR formulas. In the cases where for a single tanker different Ice Classes were investigated, these data are connected by a thin black line.

The brash ice resistance found in the model test is unexceptional lower than the resistance calculated by the FSIR formulas. The lowest model test results (which were obtained in the latest tests) are even 50% lower than the calculated values.

It has to be noted here that the guidelines for the performance of model tests in brash ice were issued by the Finnish Maritime Authorities only after the first tanker model had been tested in brash ice. Since then, the guidelines were revised and supplemented by the Authorities twice. In spite of the fact that the first model tests performed with large tankers yielded a lower resistance and a significantly lower required power than calculated by the FSIR formulas, the model test parameters defining the ice and hull conditions (e.g., friction coefficient) were established on a lower level than those used by HSVA in the early tests. Thus, rather the model test conditions stipulated by the guidelines than the quality of the individual bow shape are the main reason for lower resistance and power values of the tanker models tested more recently.

A significantly larger relative difference between model test results and FSIR formulas was found for the propulsion power required for the transit in defined brash ice channels. As the plot in Fig. 11 shows, the power derived from the model test was in no case higher than 60% of the power calculated by the FSIR formulas. In latest tests (see above) the tanker models were transiting through the brash ice channel with only 25% of the power calculated by the FSIR formulas. The only explanation for this fact is that the large tanker propellers have a much better efficiency or, with other words, develop a significantly higher specific thrust (thrust per power) than the FSIR power approximation assumes.

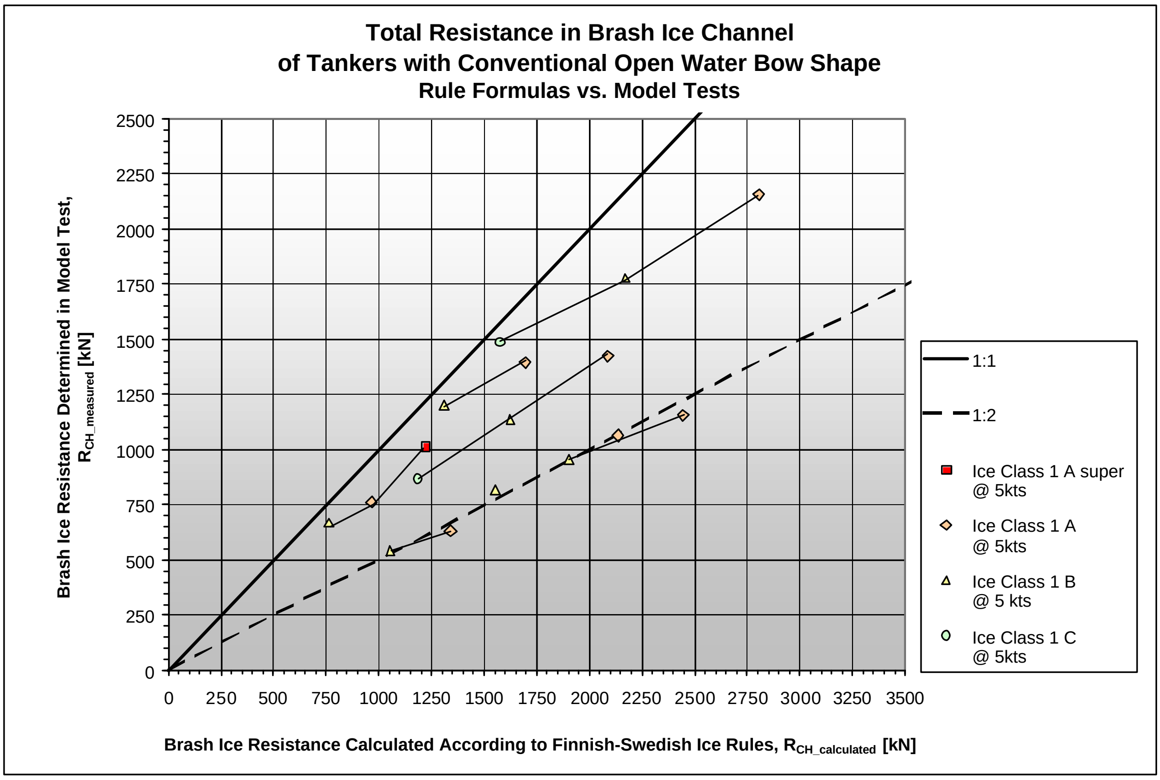

Fig. 10: Resistance in Brash Ice Channel - Model Tests vs. FSIR formulas

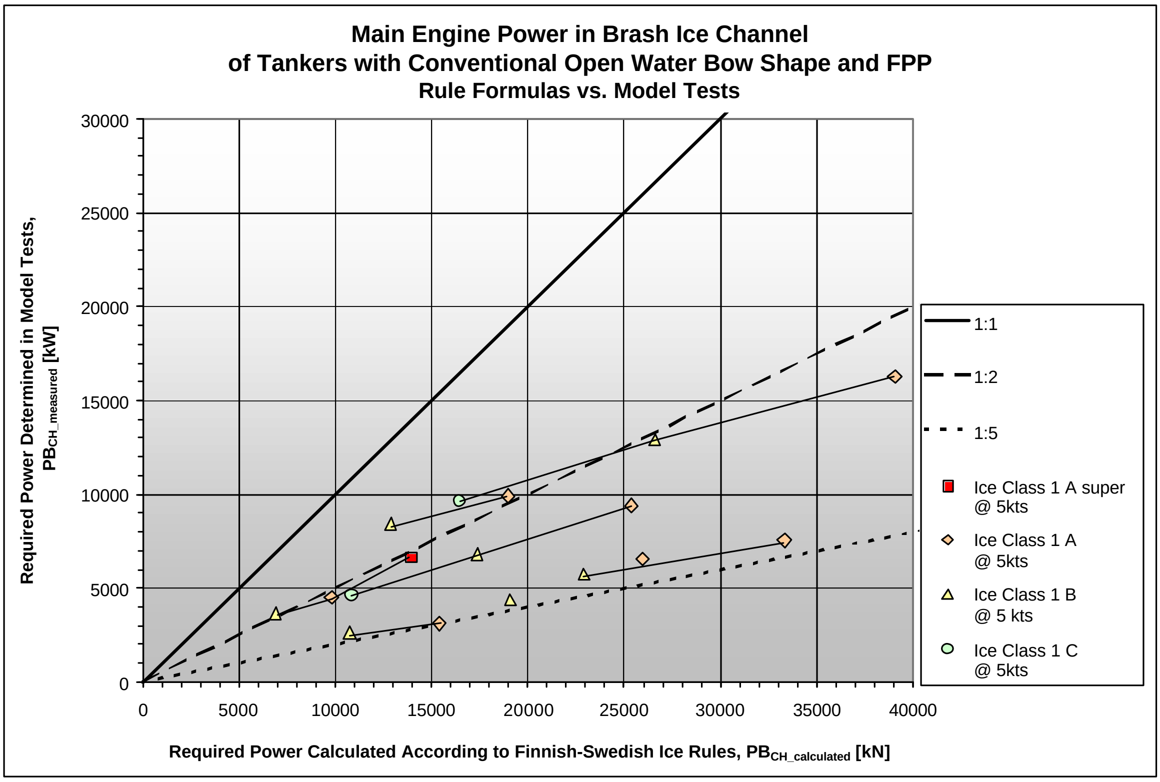

Fig. 11: Required Engine Power in Brash Ice Channel - Model Tests vs. FSIR formulas

5.3 Propulsion in Brash Ice under Consideration of Engine Load Limits

Even if model tests have shown that a large tanker can transit through brash ice channels with only the power needed for open water performance (i.e., with substantially less power than the FSIR formulas require), it is not sure that the engine(s) of the real ship can do so. In the case of a fixed pitch propeller which is usually designed for compromise operation in open water (i.e., loaded – ballast, trial-service conditions), the engine will not be able to deliver the full power to the propeller, when the vessel is sailing at 5 kts and the propeller is running correspondingly heavy. This shall be demonstrated by a real example:

The upper diagram in Fig. 12 shows the power curve vs. propeller rpm of an engine with the normal load range (bold black curve) and the propeller curve for open water free-running condition for the fully loaded vessel (bold blue curve). This diagram also shows the propeller curves at bollard pull (bold green curve) and at a constant ship's speed of 5 kts (= minimum speed in brash ice channel required by FSIR; bold red curve). From this diagram one can read that with a ship's speed of 5 kts the engine reaches the load limit at about 9400 kW, which is less than 60% of the nominal power. One can further read that under bollard pull condition the engine is only able to deliver 6000 kW (38% MCR) without leaving the permitted operation range.

In the lower diagram of Fig. 12 the <u>effective</u> propeller thrust curve at a speed of 5 kts is plotted vs. rpm. Furthermore, the brash ice resistance as calculated by the FSIR rules and as obtained from the model tests is plotted. The intersections of the thrust curve with the resistance curves lead to the rpm which the engine has to run so that the propeller can develop the required thrust.

Back in the upper diagram, we can see that in order to cope with resistance according to FSIR the engine has to be operated clearly out of the permitted range. Even for the significantly lower model test resistance the engine would still operate out of the limits.

Based on the information received from a major engine manufacturer it is possible to modified the turbo charger characteristic (higher charging pressure at reduced rpm) of the main engine so that the engine load range can be significantly extended. Such an extended engine load curve is shown in the upper diagram as dotted bold black line. With this engine modification the tanker would be able to overcome the brash ice resistance for Ice Class 1A as determined in the model tests but not the resistance for Ice Class 1 A as calculated according to the FSIR.

Another way to solve the propulsion problem at low ship speeds would be the installation of a CP-propeller. However, the additional investment cost for a CP propeller of the required size is not insignificant.

Investigation being done for twin screw tankers to be classed with the RP sign (Redundant Propulsion) have shown that in the emergency case (i.e., one propeller operation under bad weather and sea conditions) the propeller operates under very similar conditions as discussed above. This means that the same engine load problem has to be considered for vessels intended for RP certification.

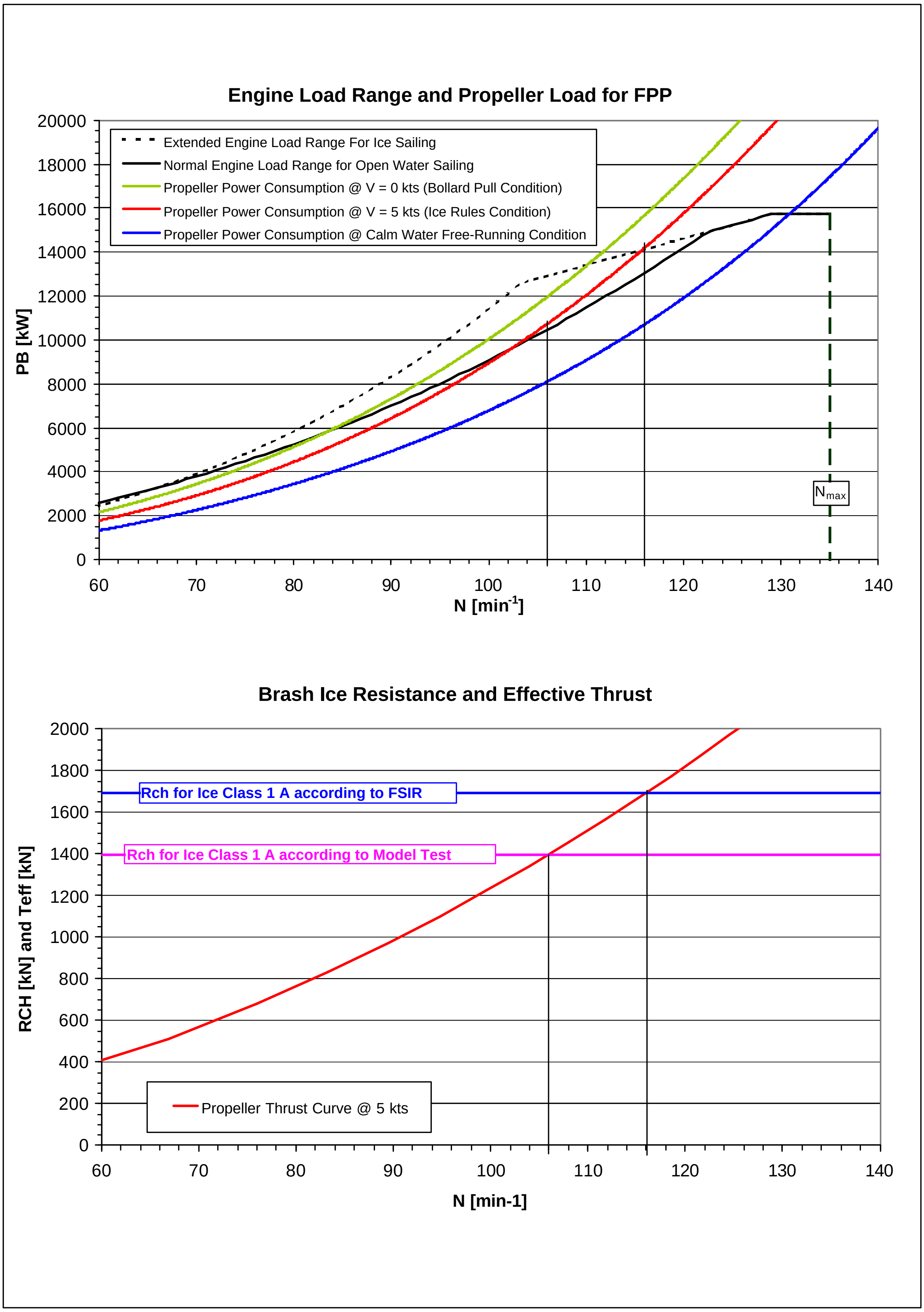

Fig. 12: Engine Load during Transit in Brash Ice Channel

6. Conclusion

There are clear indications that the sea transport of solid and liquid bulk cargo will further grow in the near future. Due to the transport of crude oil out of arctic and sub-arctic regions there is an increasing demand of tanker tonnage suitable for sailing in ice.

The operation of bulk carriers and tankers are characterized by two very different loading conditions, fully loaded and on ballast. For both loading conditions the hydrodynamic problems have to solved and the performance of the vessel has to be optimised.

Due to the large block coefficient of these vessels, special attention is required for the course stability problem.

For large tankers and bulkers (>100 TDWT) the main engine power installed for economic open water speed is insufficient to fulfil the power requirement of Ice Class 1B of the of Finnish Swedish Ice Class Rules (FSIR).

By means of ice model tests it has been proven that large tankers can cope with the brash ice resistance specified in the FSIR with significantly less propulsion power than stipulated in the rules.

To achieve the transit performance required by the Ice class 1 B with a fixed pitch propeller the normal engine load range is often insufficient. By using engines with extended load range or CP propellers the power output can be significantly increased so that at least the ice performance required for Ice Class 1 B can be achieved with the same main engine size needed for open water performance.

7. References

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