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Research Report No 53

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ON THE POWER REQUIREMENT IN THE FINNISH-SWEDISH ICE CLASS RULES

Sjöfartsverket
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Espoo, September 2002

0 ABSTRACT

The purpose of this study is to present the power requirement in the Finnish-Swedish Ice Class Rules and to give some background on the factors influencing this requirement and the constants included therein.

The structure of the power requirement, the background of the performance requirement, the rule channel resistance equations and the power requirement equation are all presented in this report in detail. The results on the performance requirement and the channel resistance are collected from other reports. These results are presented in this report to give perspective and understanding on the basis of the power requirement and the factors influencing it. A special attention is given to the derivation of the power requirement formula from the bollard pull equation and to derivation of the constants included in the present rules.

The main focus of this report is an analysis of the K_e coefficient describing the relationship between the propulsion power and propeller thrust in the power requirement equation which is taken as a constant in the present rules. The purpose is to investigate the validity of this approach and the assumptions influencing it. Also the effect of the different factors like the propeller diameter on the power requirement were studied.

It is shown in this report that the coefficient K_e is not constant but it depends on several factors. The calculations show that the coefficient is a function of propeller diameter and required thrust i.e. the resistance to be overcome and the propulsion characteristics. The effect of thrust deduction on the coefficient is also notable.

A databank of vessels built was collected in order to validate the theoretical calculations. For these vessels the actual K_e coefficient can be calculated and compared with the theoretical values. These results support the theoretical calculations as the values from the databank show similar trend versus the propeller diameter as the theoretical results.

Finally a few examples using existing ships are calculated which present how different factors influence the power requirement. Also the theoretical calculations were compared with different rule power requirements (2002 rule, 1985 rule) and installed power. This comparison was conducted for a group of cargo vessels which operate in the Baltic Sea.

The most important results are presented in figures where the rule power requirement is compared with the theoretical calculations. The theoretical calculations give slightly smaller values for power for all studied vessels. Especially for smaller vessels the difference becomes larger. In general it was shown that the present rules give similar but slightly conservative results as the more thorough calculations. For smaller vessels the rule requirement becomes about 10 percent more stringent compared with the theoretical calculations. The rule requirement has, according to the calculations, adequate bases. Further, the level of the required rule power is correct when the factors influencing the winter navigation system (traffic restrictions, icebreaker escort) are taken into account.

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List of symbols

R_i	Ice resistance (kN)
L	Length (m)
L_{par}	Length of the parallel midbody at waterline (m)
L_{bow}	Length of the foreship at waterline (m)
A_{WF}	Waterplane area of the foreship (m^2)
B	Breadth (m)
T	Draught (m)
λ	Stem angle (deg)
…	Flare angle (deg)
ζ	Waterplane entrance angle (deg)
h_i	Level ice thickness (m)
H_M	Brash ice thickness in channel (m)
P_s	Shaft power (kW)
D_p	Propeller diameter (m)
A_0	Propeller disc area (m^2)
CP	Controlled Pitch propeller
FP	Fixed Pitch propeller
R_{ch}	Channel resistance (kN)
R_T	Total resistance (kN)
P_E	Towing power (kN)
v	Ship speed (m/s or kn)
v_{ow}	Ships maximum speed in open water (m/s or kn)
K_T	Thrust coefficient
K_Q	Torque coefficient
T	Thrust (kN)
Q	Torque (kNm)
ψ	Water density (ton/m^3)
n	Propeller revolutions per second
K	Quality criterion for bollard pull

T_{pull}	Bollard pull (kN)
T_{net}	Net thrust (kN)
t	Thrust deduction fraction
P	Pitch
P/D	Pitch to diameter ratio

1 INTRODUCTION

The aim of this report is to present the background and development of the power requirement for merchant vessels included in the Finnish-Swedish Ice Class Rules. Especially it is intended to show how the formulae and the coefficients are derived and to give background information on the factors influencing this requirement. The power requirement discussed in this report was added into the existing Finnish-Swedish Ice Class Rules (1985) on 1.10.1999 as a new chapter replacing the old chapter 3.3 in appendix I. The requirement is based on the proposal presented in Riska et al (1997). This report includes also some discussion on the proposed modifications to the present (1999) requirements.

This report includes two quite separate parts. In the first part (chapters 2-6) the development of the power requirement in the Ice Class Rules is presented in detail. The structure of the power requirement, the performance requirement, the resistance equation and especially the derivation of the power requirement formula are presented. The specific aim of the first part is to give background information, understanding and insight on the power requirement and the physical basis and restrictions related to it. The results on the performance requirement and the channel resistance are collected from other reports.

In the second part of the report (chapters 7 and 8), the constants and certain factors influencing the power requirement are studied in more detail. The focus is on the validity of the quality constant in the bollard pull equation for reasons presented later. The aim is to study and discuss the assumptions and restrictions in the bollard pull equation and their effect on the power requirement. Also a databank of bollard pull results is collected and used as a validation for the developed relationships.

2 THE BALTIC WINTER NAVIGATION SYSTEM

A common approach to winter navigation in the Baltic has been developed in Finland and Sweden. This approach has evolved during the decades since the mid 60's when it was decided to keep also the northern Baltic harbours open year-round. The general principle of the system is that the flow of export and import must be as regular as possible while maintaining an adequate safety level of the traffic. This objective of regularity and safety is reached by giving all ships coming to Finnish or Swedish ports icebreaker assistance when the ice conditions require it. This assistance is restricted to traffic to and from the winter ports and to ships complying with the traffic restrictions. These winter ports are distributed along the whole Finnish and Swedish coast. The traffic restrictions, placed by the maritime administrations, specify which vessels are entitled to icebreaker assistance based on their ice class and deadweight. The reason for the traffic restrictions is that there is only a limited number of icebreakers and thus not all vessels can be assisted.

The winter navigation system can be seen as an organization having three main factors as depicted in figure 1. These main factors are the merchant fleet, the icebreakers and the activities of the maritime authorities. The merchant fleet that is trading regularly in the Baltic is usually ice strengthened. The shipowners have found a balance between the quite stringent requirements stemming from the ice conditions and the relatively short period these features are necessary. This is a design compromise between the short time – typically less than 50 days annually - spent in ice and the excess weight and power of ships in open water.

The icebreakers are provided and maintained by the maritime administrations of Finland and Sweden. The principle of the icebreaker assistance is that all eligible vessels arriving from the open sea are given assistance from the ice edge to the beginning of the fairway leading to the port in question. Thus the vessels need not to navigate independently in the drift ice field where extensive ridging occurs. The number of the icebreakers is based on a balance between the ice-going capability of merchant fleet and the requirements on regularity of the traffic. At present even some hours waiting time for icebreaker assistance is quite rare. Finally, the maritime authorities take care of the proper infrastructure for a continuous and smooth trade. For this purpose, special Finnish-Swedish ice class rules are given. These rules are used to maintain adequate safety level of shipping and also as a basis for setting the traffic restrictions and fairway dues. Before describing the traffic infrastructure in more detail, a short description of the Baltic ice conditions is given.

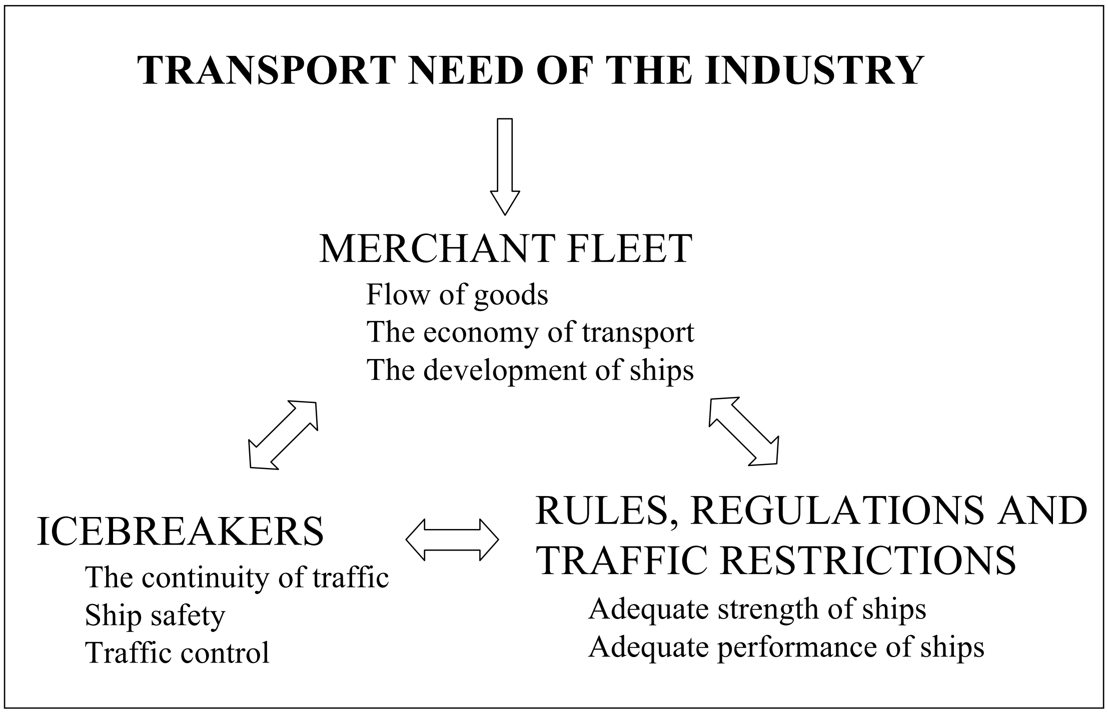


Figure 1. The basic structure of the Baltic (Finnish and Swedish) winter navigation system.

2.1 Baltic ice conditions

Ice conditions in the Baltic are controlled by two factors; the amount of degree days of frost (the cumulative average temperature) and the prevailing westerly winds. During the winter months the temperature contours i.e. isotherms are in north-south direction in the region of the Gulf of Finland and in the normal east-west direction in the Bothnian Sea and Bay. Therefore the climate gets colder when moving eastwards in the Gulf of Finland and northwards in the Bothnian Bay. This leads to the fact that the Baltic Sea starts to freeze first from the northern parts of the Bothnian Bay and from the eastern parts of the Gulf of Finland. In winter 2000-2001 the ice cover was formed close to St. Petersburg even earlier than in the northern Bothnian Bay (see e.g. the ice charts published by the Finnish Ice Service).

The prevailing westerly winds push the ice cover east against the west coast of Finland and against the bottom of the Gulf of Finland. This wind action results in a very dynamic ice field where extensive ice ridging occurs. The result of this is that the average total thickness of the ice cover increases when moving eastwards. Also the probability of encountering large ridges increase and the distance between ridges decrease. Only during hard winters, when the ice cover is frozen immobile, the active zone where the ridging takes place is shifted first west and then south towards the central Baltic. The westerly winds cause compression in the ice field in east-west direction and this is thus more significant for the traffic up to the Bothnian Bay and also for the traffic between Finland and Estonia.

The average thickness of all ice (level and deformed) can be used as an index of the severity of the winter. In an average winter this quantity is almost 60 cm in the eastern Gulf of Finland while it is about 70 cm in the Bothnian Bay (Leppäranta et al, 1988). The greatest thickness of the level ice at sea during an average winter is about 50 cm in the eastern Gulf of Finland and 70 cm in the Bothnian Bay. The corresponding average maximum thicknesses of deformed ice are 10 cm and 30 cm, respectively. During a severe winter the greatest level ice thicknesses are 70 cm and 100 cm and the deformed ice thicknesses are 70 cm for these sea areas.

A conclusion of the ice conditions is that the design ice conditions for ships intended to operate in the eastern Gulf of Finland are only slightly less severe than for those to operate in the Bothnian Bay. This is the result of severe ridging in the eastern Gulf of Finland – where the conditions get quickly less severe moving westwards. During a severe winter the same design basis can be used for ship performance in both sea areas. The ice conditions during a severe winter are more stringent for ships intended to navigate in the Bothnian Bay. Experience has shown that the ice class IA is adequate for the Gulf of Finland while sometimes even a higher class has been required in the Bothnian Bay. A recent incident when several ships bound to the Saimaa Canal were beset in ice in the eastern Gulf of Finland in April show that even on an average winter the ice class IB is not necessarily adequate in the Gulf of Finland.

2.2 Traffic infrastructure

2.2.1 Traffic restrictions

The less ice capable merchant vessels consume more icebreaker assistance time than the more capable vessels as the assistance speed depends also on the merchant vessel performance. As there is only a restricted number of icebreakers available, the number of vessels to be assisted must be restricted in some manner. The selection of vessels to be assisted by the icebreakers is done by the traffic restrictions stating the minimum ice class and cargo carrying capacity of ships eligible for icebreaker assistance. The traffic restrictions decrease the number of less ice capable ships and thus the icebreaker time is not used unduly much in escorting these ships. The traffic restrictions evolve throughout the winter as the ice conditions get more difficult. An example of this is given in figure 2. A typical most severe traffic restriction to the port of Kemi in the northernmost Baltic is ice class IA and 4000 dwt. The idea of the highest ice class IA Super is that the traffic restrictions never apply to it.

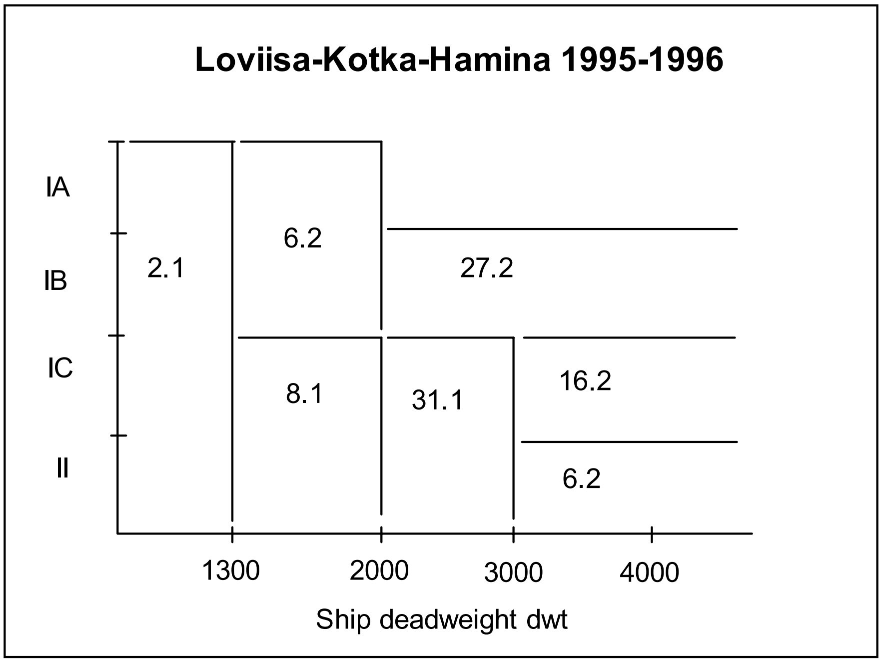


Figure 2. The evolution of the traffic restrictions during the winter to the ports of Loviisa, Kotka and Hamina in winter 1995-1996.

2.2.2 Traffic information and Ice Service

The task of the Ice Service is to provide information about the ice cover to those sailing in ice. Today this information is available in electronic format containing a chart of ice conditions and often also radar and visual wavelength satellite images. This information may be relayed to ships in order to facilitate the progress in ice, even if the usual wintertime traffic control is exercised by the Finnish and Swedish icebreakers in the Bothnian Sea and Bay. A similar system is now being considered for the Gulf of Finland to be operated jointly by Estonian, Russian and Finnish maritime administrations. The ice charts are given in quite large scale and are thus suitable for general voyage planning. More detailed satellite images may be used for detailed voyage planning even if here often occur ambiguities based on interpretation of data. A forecast of the ice motion is available from most of the national ice services even if this is not used much in guiding the shipping. The need for ice information is emphasized for ships navigating independently as it helps them to avoid the worst ice conditions.

The Finnish Maritime Administration operates also a database mainly intended for icebreakers, IBNet. IBNet is a distributed Traffic Information System for Icebreakers. It contains data about ships arriving and departing and also about the icebreaker activities. IBNet works over mobile telephone connections and is used as a Decision Support System on the icebreakers and at the coordinating centers in Finland and Sweden. Every node in IBNet has a database of its own and the changes stored in this local database are automatically distributed to the other IBNet users over mobile telephone connections. IBNet acts as an important tool for the co-operation between the Finnish and Swedish icebreaker fleets. The statistics reports enable follow-up of how the icebreaker fleet has operated and provide tools for improvement of the icebreaker activities. IBNet has been in service since March 1999 and it is in operational use on Finnish and Swedish icebreakers, in the Finnish and Swedish centers and in several ports/pilot stations in Sweden (see VTT web page www.vtt.fi).

The joint presentation of ice cover images and the traffic data results in a general insight of the traffic situation. This is provided by the IBPlott system. IBPlott is a graphical map based presentation tool for displaying the traffic situation on top of satellite images, ice charts and coastlines. IBPlott is a part of the IBNet system. IBPlott is mainly used on the Finnish and Swedish icebreakers operating in the Baltic Sea by the Finnish Maritime Administration and the Swedish Maritime Administration. An example of IBPlott image is given in figure 3.

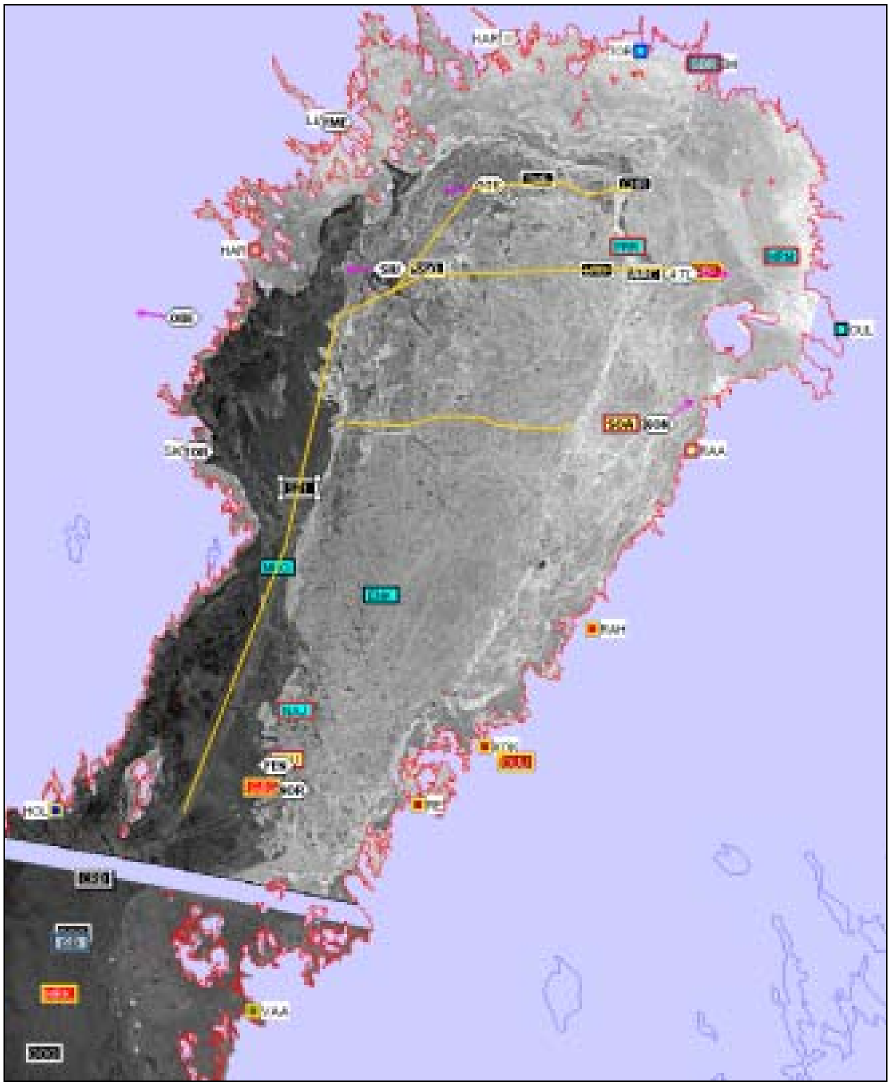


Figure 3. An example of the IBPlott image showing the Bothnian Bay, suggested ship routes and location of icebreakers and merchant vessels (source: VTT Information Technology Web page).

2.2.3 Ice class rules

The purpose of the Finnish-Swedish ice class rules is to guarantee an adequate strength of the ship hull and propulsion thus reducing the risk of ice damage to an acceptable level. The other aspect in the ice class rules is to ensure an adequate ship performance in conjunction with the traffic restrictions. The adequate performance in ice makes it possible to follow an icebreaker in the ridged ice field with a proper speed and also make the ship able to proceed along the archipelago fairway unassisted. Because of these principles, the Finnish-Swedish ice class rules define the required strength of the hull and propulsion by defining the scantlings and also the minimum propulsion power. The latter requirement is the topic of this report. The ice class rules are not intended to apply to vessels navigating independently through the pack ice zone in the middle of the sea basins. These vessels require more hull strength and power than stipulated in the rules.

The ice damages can be avoided to a large extent if the ships have a proper ice class according to the traffic restrictions. The required ice class increases during the winter, as the ice conditions become more severe. Also the icebreaker assistance influences the required ice class. The Finnish and Swedish winter traffic is based on icebreaker support and thus the ice class definition includes this. Also the requirements on the strength are balanced with the requirements for the performance i.e. the propulsion power. If no escort was provided, it would influence the required ice class and also the rule contents.

The Finnish-Swedish ice class rules include in one specific case an explicit assumption of icebreaker support. This is the strength requirement for the midbody. It is now set so that the midbody is not strong enough to withstand the loads from ice compression in normal operating conditions.

Usually the ships should have only adequate performance in ice because too powerful ships could get in too heavy ice conditions which cause high loads. There is, however, one important situation where the better ship performance could reduce the risk of ice damage. It is the situation where an escorted vessel is following an icebreaker in a compressive ice field. Because of the pressure, the track opened by the icebreaker may start to close after the icebreaker and then the vessel is likely to get stuck in the closing channel. High midbody ice loads are induced on the merchant vessel temporarily while the channel is closing. The ice class rules assume that in these situations the icebreaker releases the pressure before damage occurs. If the icebreaker is slow or the ship was proceeding independently, extensive damage may occur.

3 THE STRUCTURE OF THE POWER REQUIREMENT

The power requirement is based on an explicit performance design point which is different for different ice classes. The requirement is stated as the environmental conditions in which the vessels of different ice classes must be capable to operate. These conditions are set based on the most common operation modes and ice conditions encountered in the Northern Baltic. The limit of operability is stated as a minimum speed in commonly encountered navigation channels which are formed by brash ice. How the environmental conditions are determined is presented in the next chapter.

The power requirement in the Finnish-Swedish Ice Class Rules is set in the following manner. First the design point is set in terms of ice conditions and ship speed. The environmental conditions (channel thickness) and the speed requirement (minimum 5 kn) define the rule channel resistance which is a function of ship geometry and the ship size. The resistance calculation is presented in chapter 5. The required propulsion power is calculated from the channel resistance as the power to give the thrust to overcome the specified resistance. In chapter 6 it is shown how the power formulation is derived from the bollard pull equation. Here it should be noted that the basic rule requirement is at least 5 knots speed in channels of a given thickness. This capability may also be demonstrated by other means than calculations using the given rule equations.

Performance Requirement

Class	Speed	Channel thickness H_m	Consolidated layer h_i
IAS	5 kn	1 m	0.1 m
IA	5 kn	1 m	0
IB	5 kn	0.8 m	0
IC	5 kn	0.6 m	0

Channel Resistance R_{ch}

$$R_{ch} = R_{ch}\,(H_m,\ h_i,\ \text{ship geometry, ship size})$$

Propulsion Power P_s

$$P_s = P_s\,(R_{ch},\ D_p)$$

Figure 4. The structure of the power requirement

4 BACKGROUND OF THE PERFORMANCE REQUIREMENT

The transport system in the Northern Baltic is based on ice strengthened cargo vessels and icebreaker assistance. The system must allow an economical balance to be found in designing ships for the long open water season and fairly short winter season.

The basic performance requirement for different ice classes is defined explicitly as a design point for different ice classes. These requirements are based on the normal operation conditions encountered in the Northern Baltic. The ice classes IA Super and IA are intended for year-around operation everywhere in the Baltic and therefore the vessels have to perform well also in ice conditions. Ice class IA Super is designed for independent navigation in old navigation channels and therefore a consolidated layer of 10 cm is added to the channel brash ice thickness. The purpose is that icebreakers escort these vessels to the beginning of the archipelago fairway and from there the vessels are able to proceed independently to the harbour.

Ice class IA is designed for independent operation in newly broken channels and therefore the consolidated layer thickness is not added to the requirement. The rationale here is that an icebreaker or another vessel has previously broken the consolidated layer. The lower classes are required to proceed independently in thinner broken channels. These are to be found in early or late winter season or in more southerly areas.

These performance requirements are based on a long-term research project on the efficiency of winter navigation, which was initiated in 1990 by the Ship Laboratory at Helsinki University of Technology in co-operation with the Finnish Board of Navigation. The goal was to collect information on the factors influencing the performance of an ice-going vessel. This data-collecting campaign lasted for five years (1990-1994) and during that period an extensive databank about encountered ice conditions, operation modes and merchant vessel performance was collected.

The data were collected by observing the ship navigation in the Northern Baltic, especially vessels bound to and from Kemi and Oulu were used. The onboard observers collected information mostly on the encountered ice conditions and on the operation modes used. During the five-year period the observers were onboard on several voyages. In addition to this some cross-sections of the navigation channels were determined by drilling every winter. The results of the observation programme are described in detail in reports Kujala & Sundell (1992), Pöntynen (1992), Lehtinen (1993) and Lehtinen (1994). The vessels used in the data collection campaign are listed in appendix 1.

The most important results of the observation campaign include knowledge about the most frequently used operation modes of merchant vessels and the most commonly encountered ice conditions. The results from years 1991, 1993 and 1994 are collected in figures 5-9.

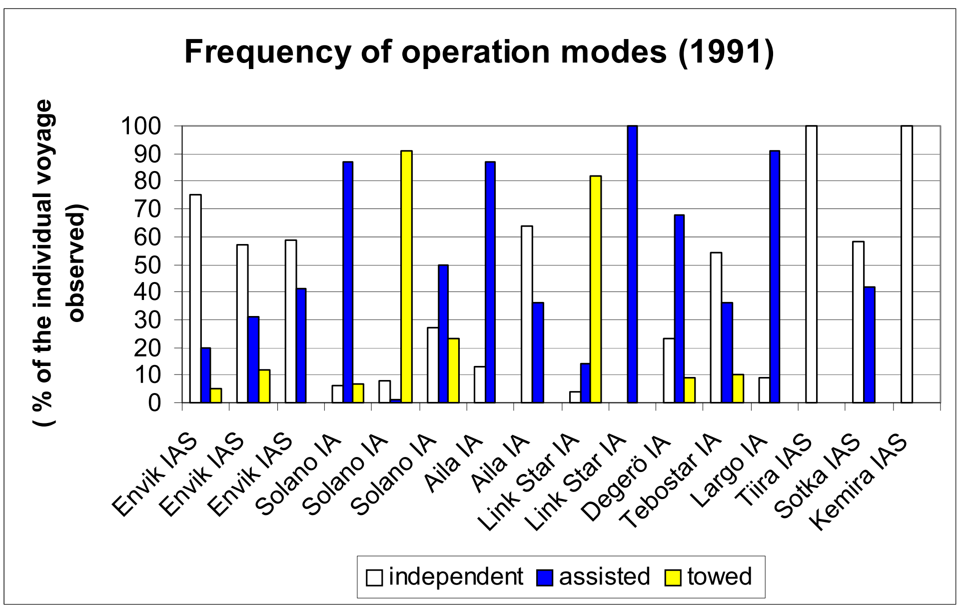


Figure 5. Frequency of operation modes in winter 1991 for all individual voyages observed.

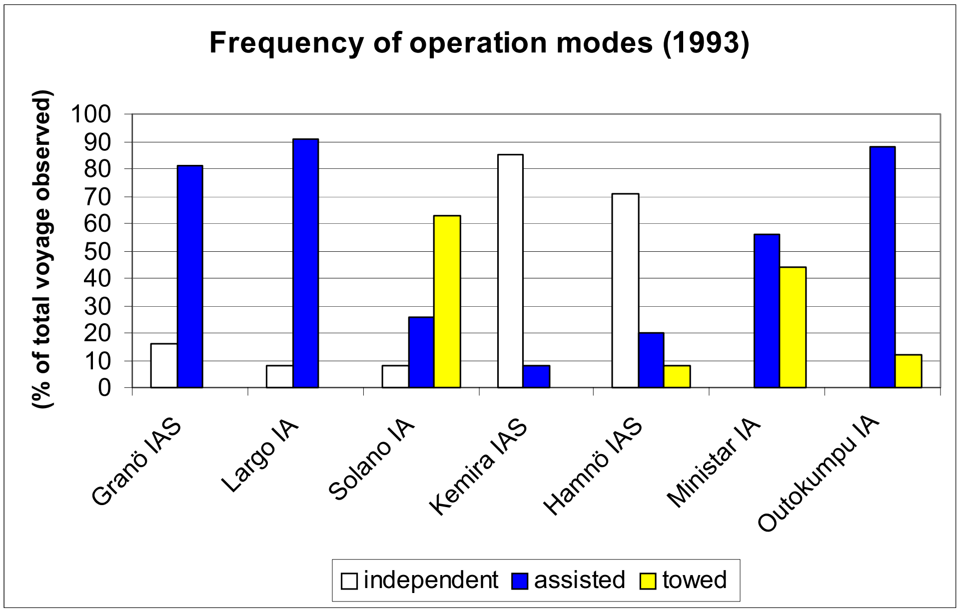


Figure 6. Frequency of operation modes in winter 1993 for total voyage observed.

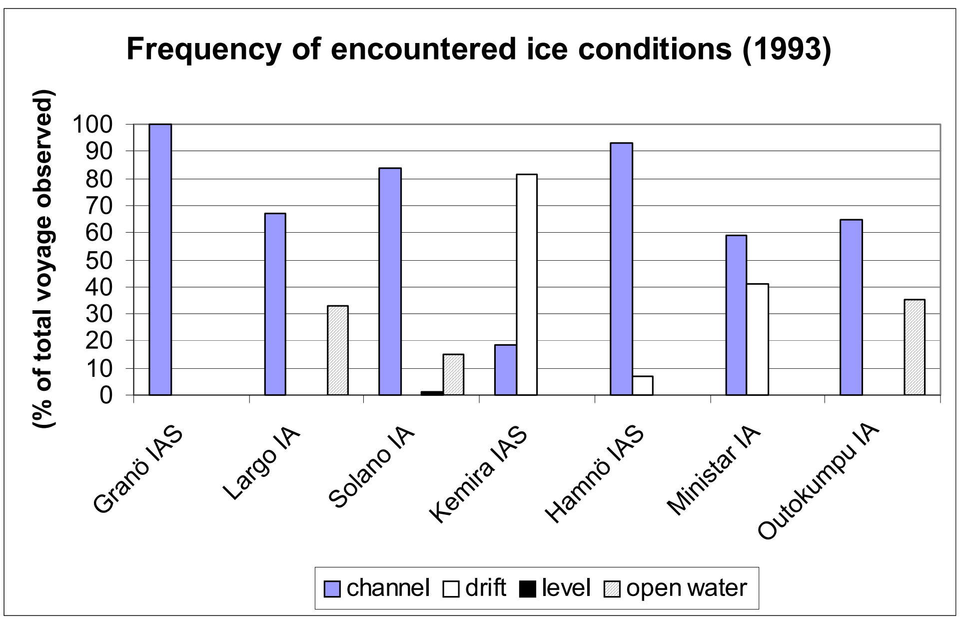


Figure 7. Frequency of encountered ice conditions in winter 1993 for total voyage observed.

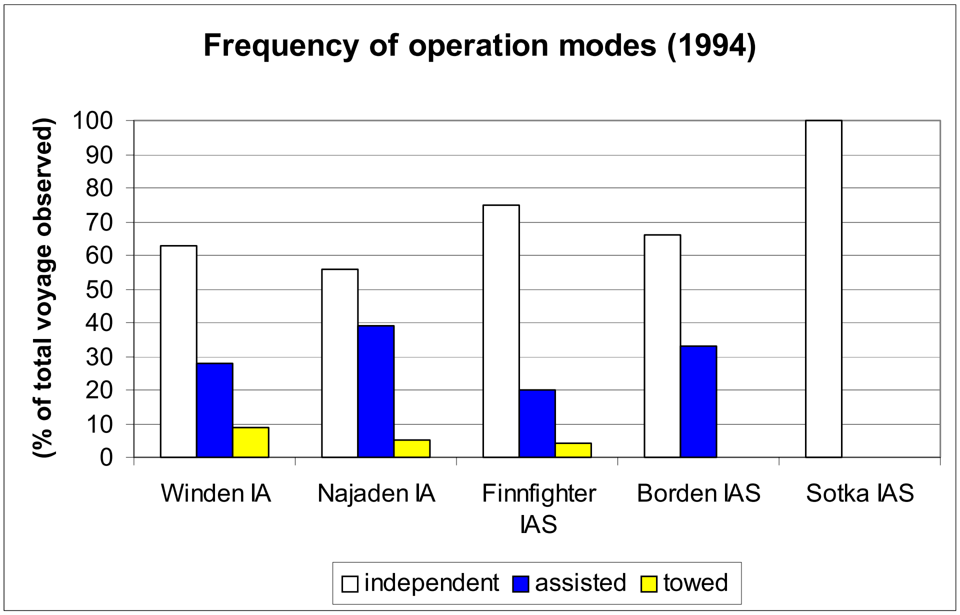


Figure 8. Frequency of operation modes in winter 1994 for total voyage observed.

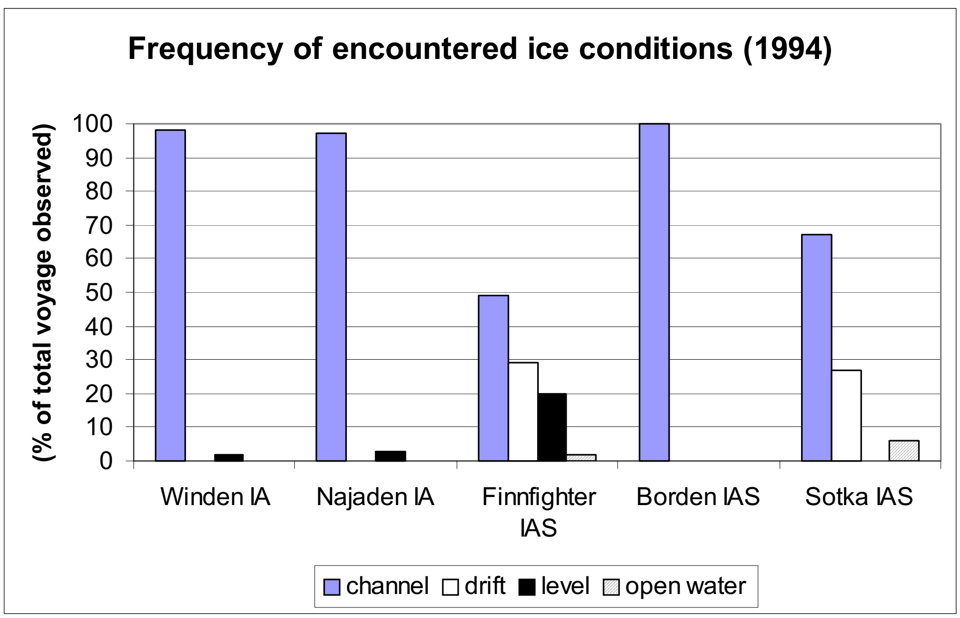


Figure 9. Frequency of encountered ice conditions in winter 1994 for total voyage observed.

The figures clearly show that the most common ice condition the vessels encounter is an old navigation channel covered with brash ice. This is due to the fact that the Finnish coasts are shallow and the vessels must use the existing fairways to the harbours. When ships repeatedly break a channel in the shorefast ice, the brash ice thickness quickly increases beyond the thickness of level ice. The brash ice thickness can be in excess of 1 m in the middle of the channel.

The need for icebreaker assistance for different classes can also be seen from the figures. Vessels of ice class IA require icebreaker assistance most of the time whereas IA Super-vessels can operate independently more often requiring assistance only in most severe conditions.

The large proportion of independent navigation in winter 1994 depends on the extent of the observations. In several cases the observations started well before the ice edge or went on after the ice edge, see for details Lehtinen (1994).

The other ice conditions and operation modes clearly do not form the design scenario for the Baltic ice-strengthened vessels. Even though the ridged ice fields are far more difficult to navigate than the channels, the vessels operate in these conditions only escorted by icebreakers. Therefore designing the vessels for independent operation in these conditions would lead to highly uneconomical hull shape and engine power for navigation in open water.

5 DETERMINATION OF THE ICE RESISTANCE

The total resistance in level ice is normally divided into open water and ice resistance components. The open water resistance at ice breaking speeds is commonly assumed to be small and therefore the cross coupling between ice forces and hydrodynamic forces does not lead to a significant error.

The ice resistance in brash ice present in navigation channels is normally divided further into two components, one due to breaking the brash ice and displacing it down and sideways and the other due to the friction along the parallel midbody. Here also a speed dependant component is added to the brash ice resistance. The consolidated layer is treated here as level ice of the same thickness and the resistance due to it is superimposed to the brash ice resistance. This is done despite of the fact that the consolidated layer does not necessarily have the same properties as level ice and the consolidated layer does not lie on water but on brash ice. No data about these factors are available and thus these assumptions have to be contended with. Ice resistance equations are presented with more details and background information in Riska et al (1997). Here the most important results are presented.

5.1 Level ice resistance (consolidated layer resistance)

Based on model and full-scale test results, the level ice resistance is here assumed to be linear with speed. Thus the ice resistance R_i contains two constants C_1 and C_2 which are dependent on ship parameters;

$$R_i = C_1 + C_2 v \tag{1}$$

The constants' C_1 and C_2 dependency on ship particulars is derived by modifying the formulations of Ionov (1988) and Lindqvist (1989). The equations for C_1 and C_2 are

$$C_1 = f_1 \frac{1}{2\frac{T}{B}+1} B L_{par} h_i + (1 + 0.021\lambda)\left(f_2 B h_i^2 + f_3 L_{bow} h_i^2 + f_4 B L_{bow} h_i\right)$$

$$C_2 = (1 + 0.063\lambda)\left(g_1 h_i^{1.5} + g_2 B h_i\right) + g_3 h_i \left(1 + 1.2\frac{T}{B}\right)\frac{B^2}{\sqrt{L}} \tag{2}$$

where h_i is level ice thickness, B is ship breadth, T is ship draught, L is ship length (between perpendiculars), L_{par} is the length of the parallel midbody at waterline, L_{bow} is the length of the foreship at waterline and λ is the stem angle at CL.

The values for constants based on model and full-scale data are

$f_1 = 0.23$ kN/m^3	$g_1 = 18.9$ kN/(m/s*m$^{1.5}$)
$f_2 = 4.58$ kN/m^3	$g_2 = 0.67$ kN/(m/s*m^2)
$f_3 = 1.47$ kN/m^3	$g_3 = 1.55$ kN/(m/s*m$^{2.5}$)
$f_4 = 0.29$ kN/m^3	

Resistance predictions for two ships, MT Sotka and MV Finnmerchant (ex: Arcturus) are presented in figure 10 together with the measured full scale values. The calculated resistance is slightly higher than the measured values. This is partly due to the use of somewhat high ice bending strength and friction values in resistance equation (σ_f=500kPa and μ=0.15 respectively).

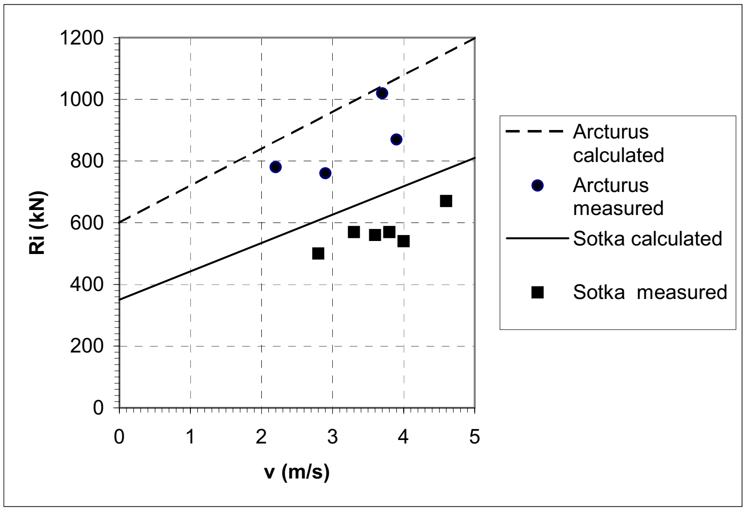


Figure 10. Calculated ice resistance versus ship speed for two vessels; MT Sotka (h_i=0.54m) and MV Arcturus (h_i=0.45m) compared with some full scale values. (Riska et al, 1997)

The resistance formulation was validated on the performance of the ships shown in table 1. All the vessels are ice-strengthened cargo vessels and therefore the resistance equations should be suited for vessels of this type. The performance curves i.e. the speed the vessels may attain in different level ice thickness are presented in figure 11, where the speed is scaled by the maximum open water speed of the vessel.

Table 1. Vessels used in validation of the level ice resistance formula.

Ship	Class	L	B	T	Ps (MW)	Dp	÷ (t)
Envik	IA Super	96	16.2	5.2	2.74	3.05	5583
Kemira	IA Super	105	17	6.6	4.12	4.15	8565
Link Star	IA	98	17	5.8	2.96	3.6	6877
Solano	IA	116.3	21	6.2	5.52	3.8	10458
Atserot (ex: Tebostar)	IA	105.3	17.6	6.6	3.68	3.7	7810
Sotka	IA Super	150	21.5	9.5	11.47	5.45	22033
Finnoak (ex: Ahtela)	IA Super	112	19	6.1	5.92	3.7	9200
Aila	IA	97.4	16	5.8	2.96	3.6	6320
Arcturus	IA Super	146	25	7.3	13.2	5.7	18000
Tervi	IA	193.7	30.2	12	10.8	7.4	57300

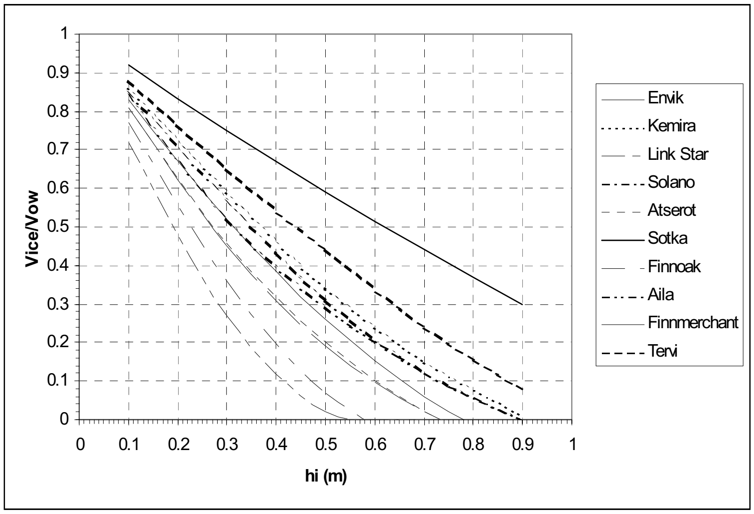


Figure 11. The performance in level ice of the ten vessels in table 1 (Riska et al, 1997)

The good performance of MT Sotka is clear, but the other vessels show only some variation, the limit ice thickness being from 40 cm to about 85 cm.

5.2 Channel resistance (brash ice resistance)

The brash ice resistance arises from displacing the brash ice present in the channel both down and sideways. The sideways motion is limited because of the side ridges, always present in old navigation channels. Also the channel is assumed to thicken slightly from the centerline toward the sides. The slope angle is assumed to be 2° (see figure 12). The brash ice resistance is formulated using soil mechanics. Here a speed dependant formula for brash ice resistance R_{ch} is derived from the equations by Englund (1996) and Wilhelmson (1996),

$$R_{ch} = \frac{1}{2}\sigma_B \psi_{\div} g H_F^2 K_P \left(\frac{1}{2} + \frac{H_M}{2H_F}\right)^2 \left(B + 2H_F\left(\cos\iota - \frac{1}{\tan\ldots}\right)\right)\left(\sigma_h \cos\lambda + \sin\ldots\sin\zeta\right) + \sigma_B \psi_{\div} g K_0 \sigma_h L_{par} H_F^2 + \psi_{\div} g \left(\frac{LT}{B^2}\right)^3 H_M A_{WF} Fn^2 \quad (3)$$

where σ_B=1-p and p is porosity (σ_B=0.8…0.9), ψ the difference between the densities of water and ice, g the gravity constant, K_P the constant of passive stress (soil mechanics), H_M the thickness of the brash ice in the middle of the channel, ι the slope angle of the side wall of the brash ice (22.6°), σ_H the coefficient of friction between the ice and the hull, λ the angle between the waterline and the vertical at B/2, K_0 the coefficient of lateral stress at rest, L_{par} the length of the parallel midbody at the waterline, A_{WF} the waterline area of the foreship and Fn the Froude number. H_F describes the thickness of the brash ice layer which is displaced by the bow and is moved to the side against the parallel midbody (see figure 13). This is a function of ship breadth, channel thickness and two slope angles which are dependent of the inner properties of brash ice (ν=2° and ι=22.6° are used) in the following manner

$$H_F = H_M + \frac{B}{2}\tan\nu + (\tan\nu + \tan\iota)\sqrt{\frac{B\left(H_M + \frac{B}{4}\tan\nu\right)}{\tan\nu + \tan\iota}} \tag{4}$$

This formula has been simplified by an approximation which is valid when B>10m and H_M>0.4m

$$H_F = 0.26 + (BH_M)^{0.5} \tag{5}$$

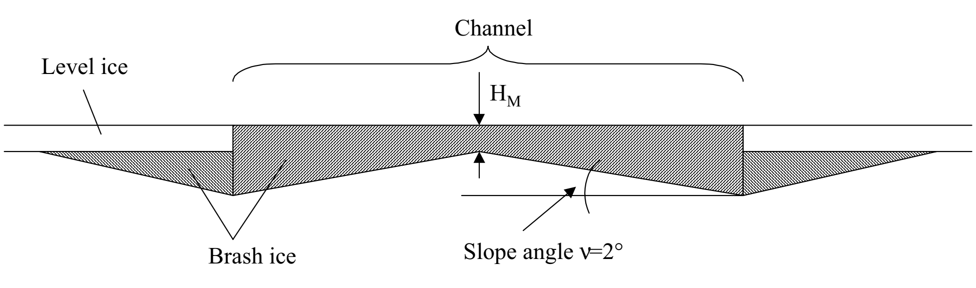


Figure 12. Definitions for H_M and ν

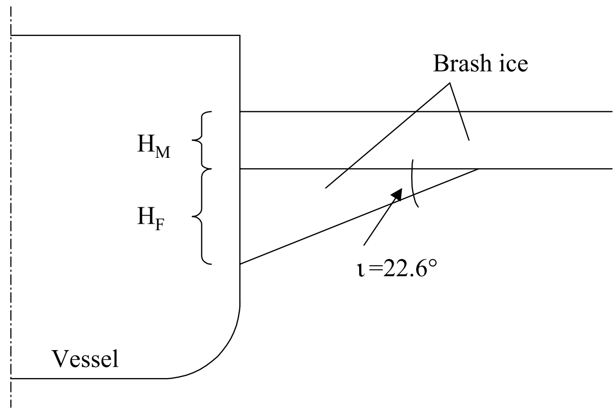


Figure 13. Definitions for H_M, H_F and ι

The flare angle … may be eliminated from the equations using the following trigonometric identities

$$\sin\ldots| \ \frac{\tan\lambda}{\sqrt{\sin^2\zeta\ 2\ \tan^2\lambda}}$$

$$\ldots| \ \arctan\left(\frac{\tan\lambda}{\sin\zeta}\right\}$$

In figure 14 the channel resistance is shown as a function of the channel thickness calculated for two vessels. For these vessels the full scale values have also been measured. As there was much scatter in the full scale results, only the range of the full scale values is presented in the figure.

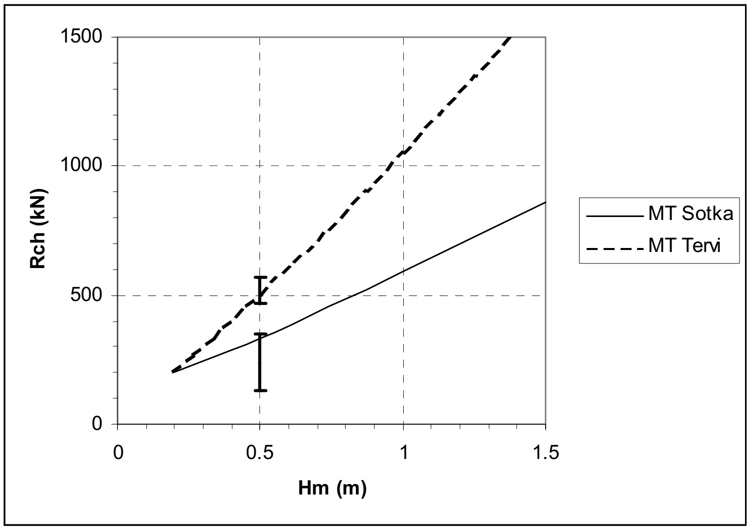


Figure 14. Channel resistance versus the channel thickness for MT Sotka and MT Tervi compared with full scale values. There was much scatter in the full scale values so only the range of the values is shown (Riska et al, 1997).

5.3 Rule channel resistance

When the design point in the performance requirement presented earlier is taken into account in the rule formulation and values for different material properties are selected, the rule channel resistance for different ice classes can be presented in the following form. The resistance equations contain the bow angles λ and ζ. Usually these are measured at the bow on CL from where most of the icebreaking forces come. This is not correct for brash ice where the whole bow is displacing ice. Thus an average value for angles ζ and λ would be suitable. The use of the average values is, however, not practical and thus, as a representative value, these angles are to be measured at waterline at distance of B/4 from centerline (this is valid for ζ and λ_2, λ_1 is to be measured at centerline).

The rule resistance equation is thus the following: (Note that this equation is slightly modified from the existing 1999 Ice Class Rules. This formula will replace in the autumn 2002 version of the rules the existing formula)

$$R_{ch} = C_1 + C_2 + C_3 (H_F + H_M)^2 (B + C_\psi H_F) C_\sigma + C_4 L_{par} H_F^2 + C_5 \left(\frac{LT}{B^2}\right)^3 \frac{A_{WF}}{L} \qquad (6)$$

where

$$H_F = 0.26 + (H_M B)^{0.5} \qquad (7)$$

$$C_\sigma = 0.15 \cos\lambda_2 + \sin\psi \sin\zeta, \quad \min 0.45 \qquad (8)$$

$$C_\psi = 0.047\psi - 2.115, \quad \min 0.0 \qquad (9)$$

$$\psi = \arctan\left(\frac{\tan\lambda_2}{\sin\zeta}\right) \qquad (10)$$

$$C_1 = f_1 \frac{BL_{par}}{\frac{2T}{B} + 1} + (1 + 0.021\lambda_1)(f_2 B + f_3 L_{bow} + f_4 B L_{bow}) \qquad (11)$$

$$C_2 = (1 + 0.063\lambda_1)(g_1 + g_2 B) + g_3 \left(1 + 1.2\frac{T}{B}\right) \frac{B^2}{\sqrt{L}} \qquad (12)$$

The constants C_1 and C_2 apply only for ice class IA Super. For lower classes they are to be taken as zero. For ships of ice class IA Super with a bulb, the stem angle λ_1 is to be taken as 90°.

$C_3 = 845.576$ kg/(m^2s^2)
$C_4 = 41.74$ kg/(m^2s^2)

C_5 as in table 2.

$f_1 = 23$ N/m^2 $\quad$ $g_1 = 1537.3$ N

$f_2 = 45.8$ N/m $\quad$ $g_2 = 172.3$ N/m

$f_3 = 14.7$ N/m $\quad$ $g_3 = 398.7$ N/m$^{1.5}$

$f_4 = 29$ N/m^2

The term $\left(\frac{LT}{B^2}\right)^3$ is to be taken as 20 if $\left(\frac{LT}{B^2}\right)^3 > 20$ or 5 if $\left(\frac{LT}{B^2}\right)^3 < 5$.

Table 2. H_M, h_I and C_5.

Ice class:	IA Super	IA	IB	IC
H_M [m]	1.0	1.0	0.8	0.6
h_i [m]	0.1	0	0	0
C_5 [kg/s]	825.6	825.6	660.5	495.4

For ships where the determination of certain parameters is difficult due to e.g. lack of linesdrawings, a simplified equation may in certain cases be used for calculating the rule channel resistance R_{ch}. This equation is based on average or slightly conservative values of the bow angles and lengths ($L_{par}/L=0.45$, $L_{bow}/L=0.2$, $A_{wf}=\frac{1}{4}LB$, $\zeta=30^o$, $\lambda_1=40^o$ (for ships without a bulb) or 90^o (for ships with a bulb) and $\lambda_2=40^o$).

$$R_{ch} = C_1 + C_2 + C_3 (H_F + H_M)^2 (B + 0.658 H_F) + C_4 L H_F^2 + C_5 \left(\frac{LT}{B^2}\right)^3 \frac{B}{4} \tag{13}$$

where

$$H_F = 0.26 + (H_M B)^{0.5} \tag{14}$$

$$C_1 = f_1 \frac{BL}{\frac{2T}{B}+1} + 1.84\left(f_2 B + f_3 L + f_4 BL\right) \tag{15}$$

$$C_2 = 3.52\left(g_1 + g_2 B\right) + g_3\left(1 + 1.2\frac{T}{B}\right)\frac{B^2}{\sqrt{L}} \tag{16}$$

for ships <u>without</u> a bulb or

$$C_1 = f_1 \frac{BL}{\frac{2T}{B}+1} + 2.89\left(f_2 B + f_3 L + f_4 BL\right) \tag{17}$$

$$C_2 = 6.67\left(g_1 + g_2 B\right) + g_3\left(1 + 1.2\frac{T}{B}\right)\frac{B^2}{\sqrt{L}} \tag{18}$$

for ships <u>with</u> a bulb.

The constants C_1 and C_2 apply only for ice class IA Super. For lower classes they are to be taken as zero.

$C_3 = 459.993$ kg/(m^2s^2)
$C_4 = 18.783$ kg/(m^2s^2)

C_5 as in table 2.

$f_1 = 10.35$ N/m^2 $g_1 = 1537.3$ N
$f_2 = 45.8$ N/m $g_2 = 172.3$ N/m
$f_3 = 2.94$ N/m $g_3 = 398.7$ N/$m^{1.5}$
$f_4 = 5.8$ N/m^2

The term $\left(\frac{LT}{B^2}\right)^3$ is to be taken as 20 if $\left(\frac{LT}{B^2}\right)^3$ >20 or 5 if $\left(\frac{LT}{B^2}\right)^3$ <5.

6 DETERMINATION OF THE PROPULSION POWER

The relationship between the thrust required to overcome the ice resistance and the propulsion power is presented in this chapter. The derivation of this relationship and its constants is presented in detail.

The required propulsion power is the power that gives high enough thrust to exceed the ice resistance in the design ice conditions at the design speed. The ship speeds are low in the design ice conditions and therefore the requirement in the Finnish-Swedish Ice Class Rules is derived from the bollard pull situation. The resulting formulation in the Ice Class Rules for the required propulsion power P_s (kW) is

$$P_s = K_p \frac{(R_{ch})^{3/2}}{D_p} \tag{19}$$

where K_p is given in the table 3, D_p is propeller diameter (m) and R_{ch} (kN) is the rule channel resistance (as in chapter 5).

Table 3. Values of the constant K_p in the Ice Class Rules.

Propeller type or machinery	CP or electric or hydraulic propulsion machinery	FP propeller
1 propeller	2.03	2.26
2 propeller	1.44	1.6
3 propeller	1.18	1.31

Note that in this paper K_p is used in the power requirement formula instead of K_e to avoid confusion between the K_e coefficient normally used with the bollard pull. Calculation of the coefficients K_e and K_p as well as derivation of equation 19 is described in chapters 6.1 and 6.3. The power requirement equation itself is the same for all ice classes. The difference between the classes is included in the channel resistance as presented in previous chapters.

6.1 Bollard pull of a single screw ship

The required propulsion power in the design ice conditions can not be determined directly from the effective power ($P_E=R_{T*}v$) or the thrust power ($P_T=T*v_A$) because the ship speeds are low and the efficiency of the propulsion must be addressed. Thus the power requirement is derived from the power needed to generate the bollard pull. The derivation of the bollard pull and the power requirement is presented below.

A relationship must be developed between the channel resistance R_{ch} to be overcome at the design speed and the power to produce this thrust. This can be done by first considering an ideal propulsor (see e.g. Matusiak 1993). The power delivered at the bollard pull situation (speed of advance zero) of an ideal propulsor is (Leiviskä 2001)

$$P_D = \sqrt{\frac{T^3}{2\psi A_0}} = \sqrt{\frac{2T^3}{\phi \psi D_p^2}}\,, \tag{20}$$

where A_0 is the propeller disc area.

From this expression the bollard thrust of an ideal propulsor is obtained as

$$T = \sqrt[3]{\frac{1}{2}\psi\phi}\,(P_D D_p)^{2/3} \tag{21}$$

Similar expression for actual propellers instead of ideal propellers can be obtained using the propeller K_T-K_Q-curves as

$$T = \frac{K_T}{K_Q^{2/3}} \sqrt[3]{\frac{\psi}{4\phi^2}}\,(P_D D_p)^{2/3} \tag{22}$$

where $K_T / K_Q^{2/3}$ is an empirical definition for the quality criterion for bollard pull (Tornblad 1987) and is normally denoted as K i.e.

$$K = \frac{K_T}{K_Q^{\frac{2}{3}}} \tag{23}$$

to be calculated at J=0.

The factor K describes the goodness of the propeller open water characteristics in view of the bollard pull. Larger value of K means more thrust for a given power.

The above expressions suggest the definition of a bollard thrust coefficient being the ratio of the actual and ideal thrust as

$$\xi_{BP} = \frac{K_T}{K_Q^{2/3}} \frac{1}{\sqrt[3]{2\phi}} - 0.253 \frac{K_T}{K_Q^{2/3}} = 0.253K \tag{24}$$

Usually, however, a dimensional coefficient is used to characterize the bollard pull, see e.g. Tornblad (1987). This dimensional coefficient is called the quality constant of the bollard pull, K_e, and is defined at the bollard pull situation as

$$T_{pull} = K_e (P_s D_p)^{2/3} \tag{25}$$

where the units of the variables are kN, kW and m, respectively. Note that the K_e-coefficient is dimensional. Therefore the units of thrust, power and diameter must be as above. The relationship between the dimensionless factors and the quality coefficient is

$$K_e = K \left(\frac{\psi}{4\phi^2} \right)^{\frac{1}{3}} (1 - t) \tag{26}$$

where t is the thrust deduction fraction.

The value of K_e describes the goodness of the vessel in view of the bollard pull. Higher value means better designed vessel and propellers. K_e takes into account the vessel in form of the thrust deduction fraction whereas K describes only the goodness of the propeller. Equations 20-24 are for open water propeller but equations 25 and 26 are for a situation where the ship and propeller-hull -interaction is taken into account. Therefore t is added to equation 26 i.e. $T_{pull} = T(1-t)$.

The quality factor K_e describes the ability of the propeller to convert delivered power into bollard pull. Thus there is a target, optimum value for K_e for good bollard pull design. Normally the factor K_e is taken as a constant and the optimum value for K_e is mentioned in several technical papers (see e.g. Tsoy 1983). The normally used value is K_e=0.78 for single screw vessels with CP propellers and for FP propellers the value for K_e should be multiplied by 0.9. The origin of this reduction is at least partly empirical.

6.2 Bollard pull of double or triple screw ships

When changing the design of the vessel from a single to double screw installation, the resistance of the ship stays almost unchanged and thus also the thrust required. In bollard pull situation the wake is presumed to be small and therefore the difference in hull efficiency ξ_H between single and double screw installations is not taken into account here. In a single screw ship the required thrust is

$$T_{pull,1} = K_{e,1}(D_{p,1}P_{s,1})^{2/3}$$

where the subscript 1 refers to the number of propellers.

The K_e-coefficients for double and triple propeller installations are determined so that the power density i.e. the propulsion power divided by the propeller disc area is the same in all cases. Thus, if the same thrust with the same power is obtained from the double screw ship, i.e. $T_{pull,1}=T_{pull,2}$ and $P_{s,1}=P_{s,2}$, then the propeller diameter required in the double screw ship can be calculated. This gives

$$T_{pull,1} = K_{e,1}\left(D_{p,1}P_{s,1}\right)^{2/3} = T_{pull,2} = 2*K_{e,1}\left(D_{p,2}\frac{P_{s,1}}{2}\right)^{2/3} \quad \text{i.e.} \tag{27}$$

$$D_{p,2} = \frac{D_{p,1}}{\sqrt{2}} \tag{28}$$

In the same way the propeller diameter for triple screw vessel is obtained

$$D_{p,3} = \frac{D_{p,1}}{\sqrt{3}} \tag{29}$$

The validity of the assumption of constant power density is investigated later in this report. This assumption means that only one K_e-coefficient is needed. This is clear as the bollard pull for the double screw ship can be written as two single propellers:

$$T_{pull,2} = 2*K_{e,1}\left(\frac{P_{s,tot}}{2}D_{p,2}\right)^{2/3} \tag{30}$$

where $P_{s,tot}$ is the total shaft power. From this expression, assuming the diameters to be calculated by (28), the K_e-coefficient for double screw vessel is obtained as

$$K_{e,2} = \sqrt[3]{2}K_{e,1} = 0.98 \tag{31}$$

The same can be done for triple screw vessel with the result

$$K_{e,3} = \sqrt[3]{3}K_{e,1} = 1.12 \tag{32}$$

This way the values for K_e (K_e=0.78 for single, K_e=0.98 for double and K_e=1.12 for triple propeller installations) are obtained for CP propellers (see e.g. Tsoy 1983). For FP propellers the values for K_e should be multiplied by 0.9.

The relations presented above are valid only if the K_e coefficients for the individual propellers in multi-screw vessels are assumed to be the same as for single screw vessels. As shown later, the K_e coefficient actually varies as a function of the propeller diameter and the required propeller thrust. In that case the required power can be calculated with the rule power requirement formula but the K_e coefficient has to be calculated separately for individual cases as shown later in chapter 7. A more simple approach is, however, selected in the rules with the statement that if more thorough calculations are made, lower power values than given by the formulation above can be accepted.

6.3 Rule power requirement

The design speed in the Finnish-Swedish ice class rules is 5 knots, so the bollard pull value should be complemented with some knowledge about open water resistance. This is done here by introducing the concept of net thrust T_{net}. This is the thrust available to overcome the ice resistance after the thrust used to overcome the open water resistance is taken into account. Thus the net thrust is defined as

$$T_{net}(v) = T_{tot}(v)(1 - t) - R_{ow}(v) \qquad (33)$$

The thrust deduction is taken into account with thrust deduction fraction t. Even though t is not the same in open water conditions and in ice conditions, but it varies in a complicated manner with speed and resistance, it is here simply estimated as a reduction from the total thrust with factor 1-t.

The effect of speed v is approximated here by a quadratic factor f(v) (Riska et al, 1997) depending on the maximum open water speed v_{ow} as

$$T_{net}(v) = f(v) \cdot T_{pull} = \left(1 - \frac{1}{3}\frac{v}{v_{ow}} - \frac{2}{3}\left(\frac{v}{v_{ow}}\right)^2\right) \cdot T_{pull} \tag{34}$$

i.e. when the ship speed is zero, the net thrust is equal to bollard pull. When the ship speed is the maximum open water speed, the net thrust is zero.

The power requirement can now be calculated directly from the net thrust. The normal open water speed of a merchant vessel can be taken as 15 knots and the design speed is, as mentioned, 5 knots. Equation 34 gives approximately

$$f(v) \approx 0.8 \tag{35}$$

This thrust is available to overcome ice resistance i.e. equality $T_{net}=R_{ch}$ gives the power requirement directly from the bollard pull equation (25) as:

$$P_s = K_p \frac{(R_{ch})^{3/2}}{D_p} \tag{19}$$

where the constant K_p is

$$K_p = \frac{1}{(0.8K_e)^{3/2}} \tag{36}$$

These equations (19) and (36) are valid in all cases and for all K_e. Thus, by using the values for K_e mentioned previously, we obtain K_p=2.03 for single, K_p=1.44 for double and K_p=1.18 for triple screw vessels with CP propellers. For FP propellers the values for K_p are divided by 0.9 and are K_p=2.26 for single, K_p=1.6 for double and K_p=1.31 for triple screw vessels. These are the values used in the Ice Class Rules. If the value for K_e is determined by other means, the required power can still be calculated with these equations.

The accuracy of equations 34 and 35 can be assessed by comparing the net thrust from equation 34 with values obtained directly from measurements. The results from model tests with MT Uikku are used here. The measured open water resistance and the delivered thrust (including the thrust deduction coefficient) are given for MT Uikku in figure 15. The open water speed can be estimated from these results and it is 8.71 m/s. The net thrust is the difference between the curves in figure 15. This measured net thrust is given point by point in figure 16, where also the calculated (equation 34) net thrust is given as a continuous curve. The difference of the two curves in figure 16 is small suggesting the validity of using the formulation presented here. The slight difference in higher speeds between the curves would be corrected using coefficients ¼ and ¾ in the linear and quadratic term, respectively. There is, however, not enough data to make this adjustment.

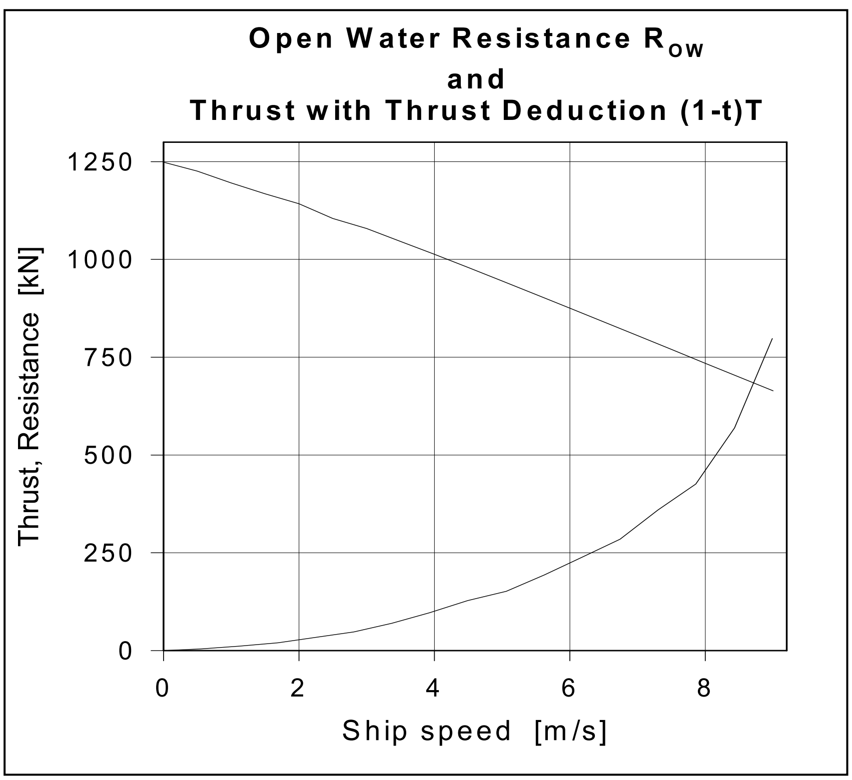


Figure 15. The open water resistance and the propeller thrust (thrust deduction taken into account) from model tests for MT Uikku.

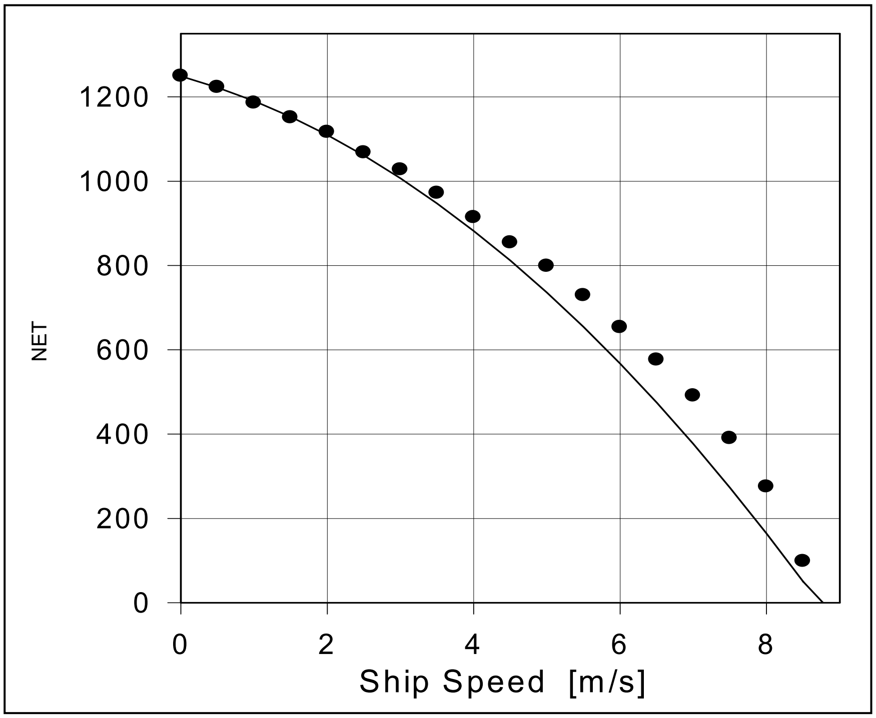


Figure 16. The estimated (using the quadratic polynomial, continuous curve) and measured (dotted curve) net thrust.

7 EVALUATION OF THE K_E-COEFFICIENT

The different factors influencing the power requirement are studied in more detail in this chapter. The most important results are presented in form of examples. The primary issue in this chapter is to investigate the power requirement and especially its applicability for different propeller diameters in single and double screw ships. The focus is in the K_e-coefficient in the bollard pull equation.

7.1 Effect of the pitch on the K_e-coefficient

As mentioned previously, the K_e-coefficient can be calculated theoretically from the K_T-K_Q-curves and the thrust deduction fraction as

$$K_e = K\left(\frac{\psi}{4\phi^2}\right)^{\frac{1}{3}}(1-t)$$

where K is

$$K = \frac{K_T}{K_Q^{\frac{2}{3}}},\ J=0.$$

The thrust deduction fraction in forward motion is normally between 0.2 and 0.3 in open water speeds for typical single screw merchant vessels. In the bollard pull situation the thrust deduction fraction is estimated to be about 0.02 or 0.03 (Isin 1987). As described in the previous chapter, the difference in the thrust deduction fraction in open water and in ice is neglected here and the same thrust deduction fraction is used in both cases in bollard pull situation. The difference in the thrust deduction between single and double screw vessels is discussed later in chapter 7.5.

The value of the bollard pull quality criterion K is known to be affected by the pitch to diameter ratio P/D of the propeller. An example of this influence may be derived using the Wageningen Propeller Series. From the K_T-K_Q-curves for different P/D ratios at zero speed the value of K as a function of P/D is obtained. Using the B4-85-series (B-series, 4 blades, area ratio 85%, FP propeller) a linear approximation for K(P/D) is obtained as

$$K\left(\frac{P}{D}\right) = -0.5848\left(\frac{P}{D}\right) + 3.349 \tag{37}$$

calculated at J=0.

The same calculation was made for five propeller series, with blade area ratios varying from 40% to 100%, and the results of K(P/D) are collected into figure 17.

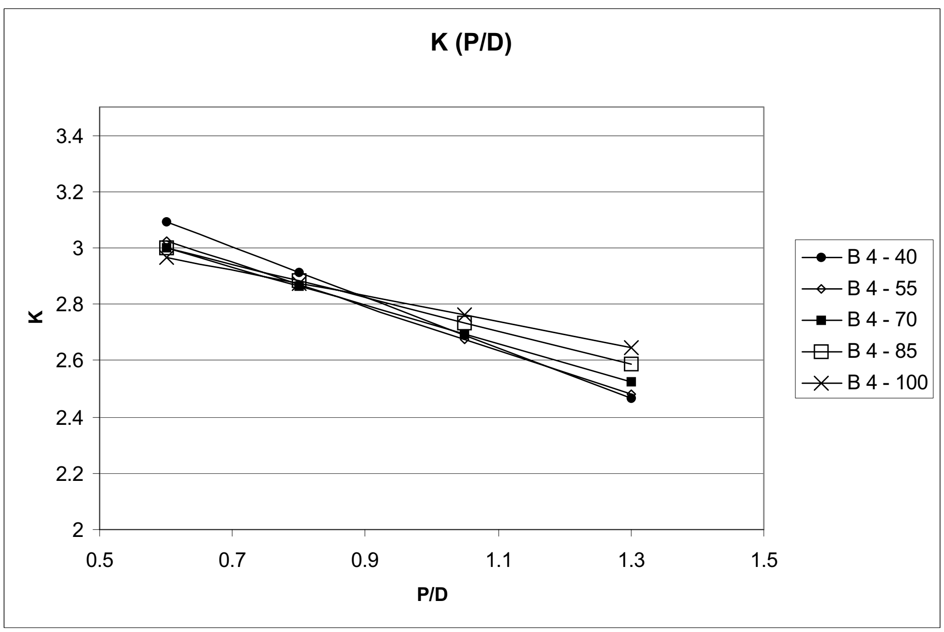


Figure 17. Values of K for different propeller series as a function of P/D. Calculated at J=0

It is clear that the value of K and therefore also the value of K_e is affected by the pitch to diameter ratio of the propeller. It can be expected that the propeller parameters, including the P/D-ratio, vary when the propeller diameter is varied (keeping the delivered thrust as constant). Now the question is how much the value of K_e is affected by the propeller diameter, i.e. is K_e a function of D_p?

7.2 *Effect of D_p on the K_e-coefficient*

In order to find the possible dependence of K_e on D_p, the relationship between P/D and the propeller diameter has to be studied. This was done by calculating the *optimum* pitch to diameter ratio for different propeller diameters while keeping the required thrust as constant. This optimisation was done for four different ships for which a rough estimate of the open water resistance and thus the necessary thrust could be calculated. At this point the optimisation of the P/D is done for *open water conditions* (v_{ow}=15kn). The ships used in these calculations were Kemira, Envik, Sotka and Tervi. The optimisation was done also for half of the Sotka's open water thrust for later use (chapter 7.4).

The required propeller thrusts varied from 250 kN to 1046 kN. Using these values and varying the propeller diameter from 3 m to 8 m for each required thrust, an optimum P/D ratio for each D_p was calculated using the Wageningen B4-85 Propeller Series. The optimisation was done according to the procedures presented in Matusiak (1993). The speed and thrust are kept constant all the time and for each propeller diameter the parabolic function

$$K_T \mid \frac{T}{\psi D^2 V_A^2} J^2 \tag{38}$$

was calculated and drawn on top of the propeller series. Now the propeller efficiency ξ_0 could be read from the figures as a function of P/D and then the optimum pitch to diameter ratio and the optimum rotation speed could be found at the P/D where the efficiency ξ_0 reaches it's maximum. As a result, linear approximations of optimum P/D as a function of D_p were obtained for all studied thrusts. The effective wake fraction was estimated to be 0.3. The results are collected into figure 18.

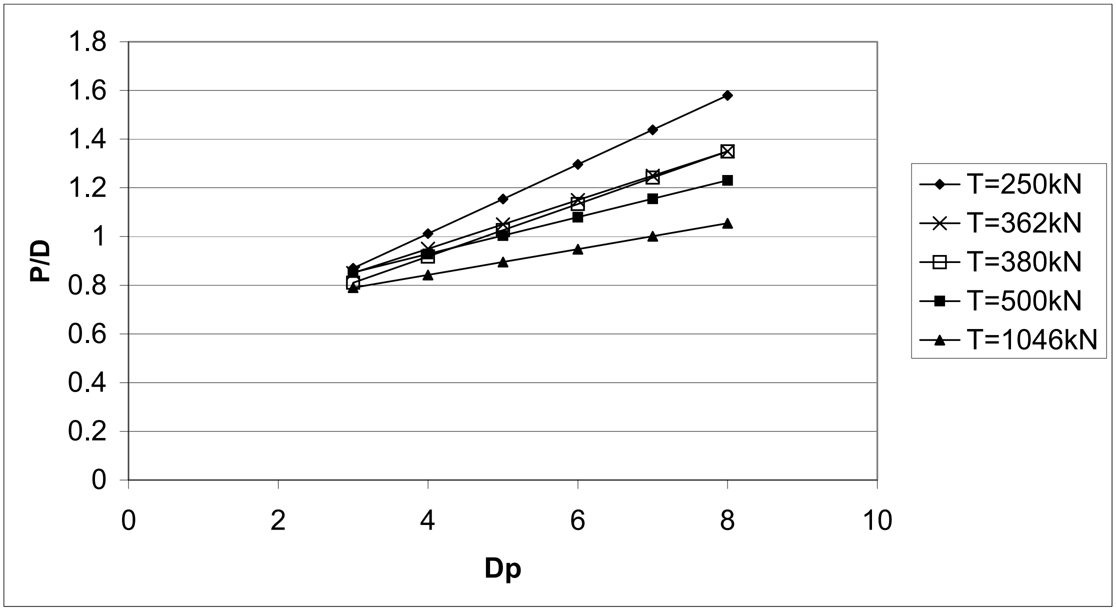


Figure 18. Optimum pitch to diameter ratio as a function of D_p for all studied cases (open water conditions, v=15kn). Note that rotation speed n is not constant.

Note that in this optimisation the rpm is not kept constant but it decreases as the propeller diameter increases. In all studied cases the optimum revolutions dropped below 70 rpm when the propeller diameter was above 6 m. Therefore these results are not practical with propeller diameters above six meters.

It is now possible to calculate K_e as a function of D_p (in bollard pull situation). Using still the B4-85-series (K(P/D) presented in previous chapter), the coefficient K_e was calculated for each example ship. Thrust deduction fraction is estimated to be 0.05. The results are collected into figure 19.

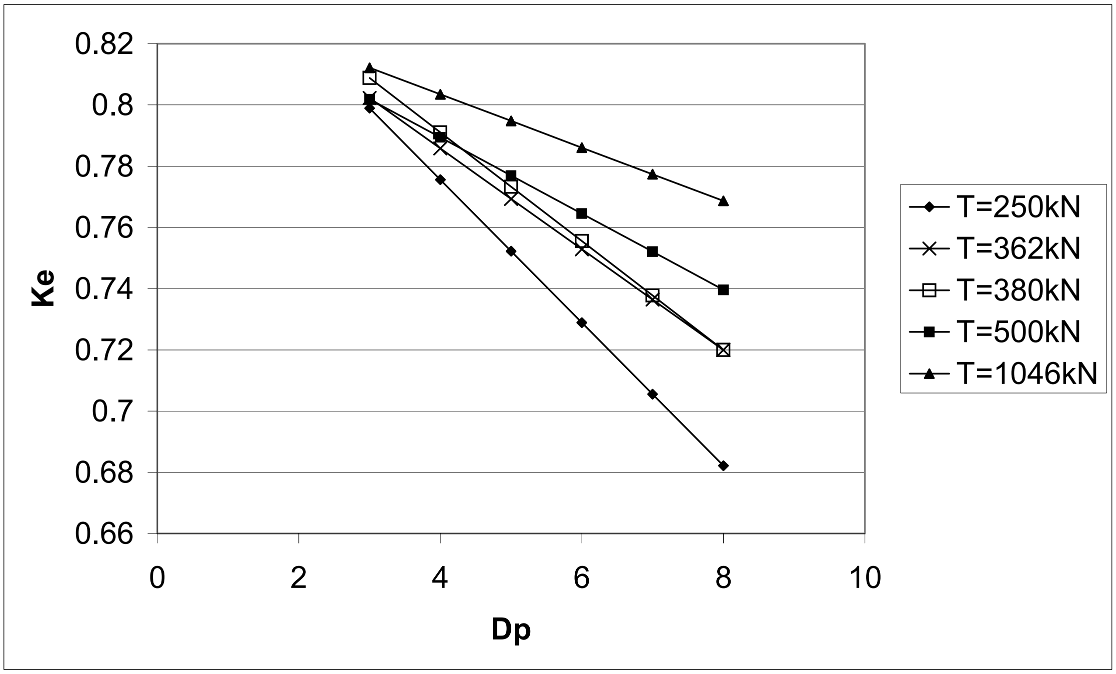


Figure 19. K_e as a function of D_p for all studied cases. (FP propeller)

The linear equations for the relationship $K_e(D_p)$ in figure 19 are:

T=250 kN	$K_e = -0.0234D_p+0.8691$	(Sotka 50 % thrust)
T=362 kN	$K_e = -0.0165D_p+0.8516$	(Kemira)
T=380 kN	$K_e = -0.0178D_p+0.8621$	(Envik)
T=500 kN	$K_e = -0.0125D_p+0.8392$	(Sotka)
T=1046 kN	$K_e = -0.0087D_p+0.8383$	(Tervi)

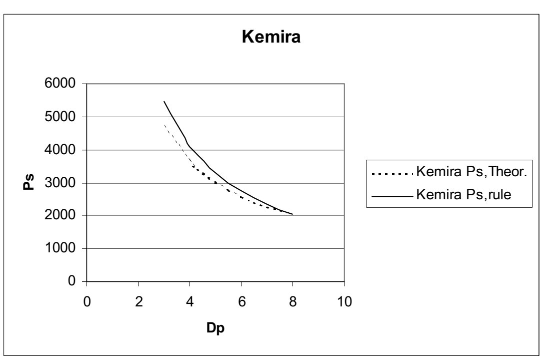
These values are calculated for FP propellers, even if the ships named here are actually CPP-ships. Also the resistance is not accurate, more like a rough estimate. Therefore the values are not accurate for these ships. They are only used as an example of ships of their size.

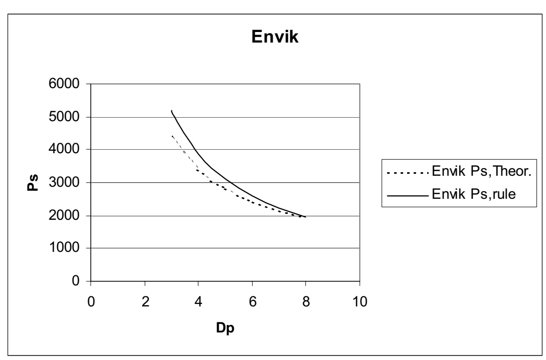
In this optimisation the propeller revolutions, propeller diameter and the P/D-ratio were free variables. Normally there might be physical limitations for these factors, especially the propeller revolutions are in many cases determined from the engine and gearbox characteristics and the propeller diameter is limited by the draft and stern lines. These restrictions must be considered with the results presented here.

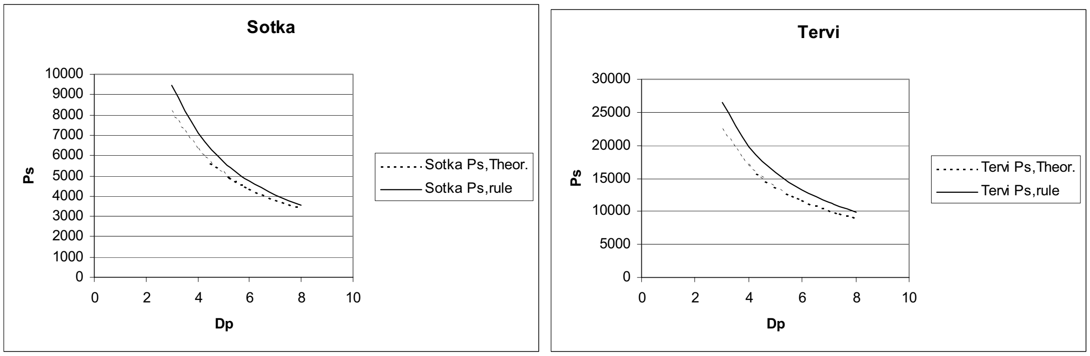
From the figure 19 can be seen that the propeller diameter actually does have an effect on the K_e-coefficient. Also the required propeller thrust seems to affect the coefficient. These conclusions are valid only for the performance of single screws. When changing the basic design of the vessel from single to double screw, the hull shape and all propulsion characteristics change and therefore these calculations can not be applied directly for double screw vessels.

It is notable that the calculated results give higher values for K_e than the normally used constant value (0.702 for single FP-propeller), especially with smaller propeller diameters. Therefore using the constant value for K_e gives conservative estimates for bollard pull and power.

Now, as an example, the bollard pull and the power requirement can be calculated using the new, D_p-dependant K_e values. For the ships named above (all single screw ships), the channel resistance and the power requirement can be calculated and the results compared with the original rule requirement. The comparison between the power required by the 2002 rule and the power calculated with K_e-values in figure 19 is done in tabular form. The table of calculation is given in appendix 2. Here the results are presented in figures 20-23 where the required power is plotted versus the propeller diameter.







Figures 20-23. Power requirement for example ships calculated with the 1999 rule and the D_p-dependant K_e-values (FP-propellers).

The influence of Dp on the K_e and P_s can be seen from the figures (the channel resistance R_{ch} is the same in both cases). The dependence of K_e from D_p changes the steepness of the curves but the difference is not large. It should be reminded here that the propellers have been optimised for open water speed. It should also be pointed out that the influence of the thrust deduction t is quite significant on K_e. Here constant value t=0.05 is used. Also the influence of propeller diameter on stern lines, t and other parameters is not included in this brief study.

7.3 Effect of D_p and T on the K_e-coefficient

As the K_e-coefficient is a function of both the propeller diameter D_p and the required thrust T, an equation which take into account both factors (K_e=K_e(D_p,T)) is fitted to the linear equations in figure 19 (units for D_p and T are m and kN respectively):

$$K_e(D_p,T) = (1.53\cdot 10^{-5}T - 0.0234)D_p + (43.2\cdot 10^{-5}T + 0.868) \qquad (39)$$

Note that equation 39 is calculated for single FP-propellers. K_e(D_p,T)-graph is presented in figure 24. The difference between the equation 39 and the calculated results in figure 19 are presented in figure 25.

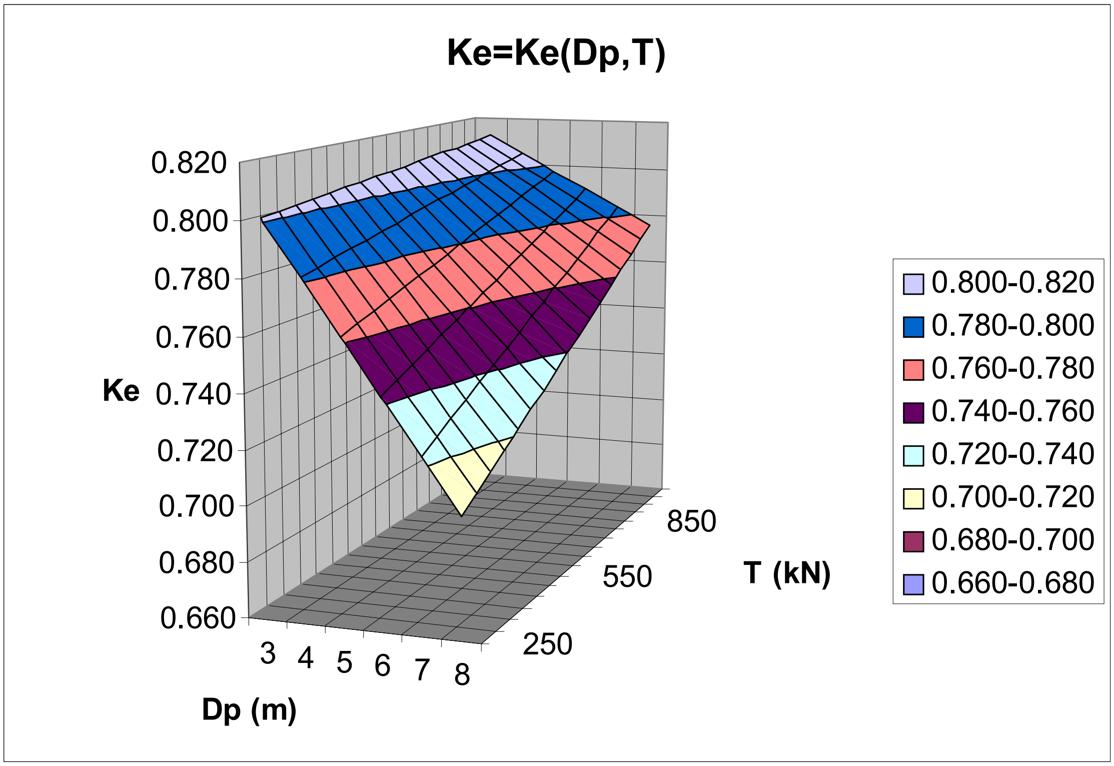


Figure 24. $K_e(D_p,T)$-graph.

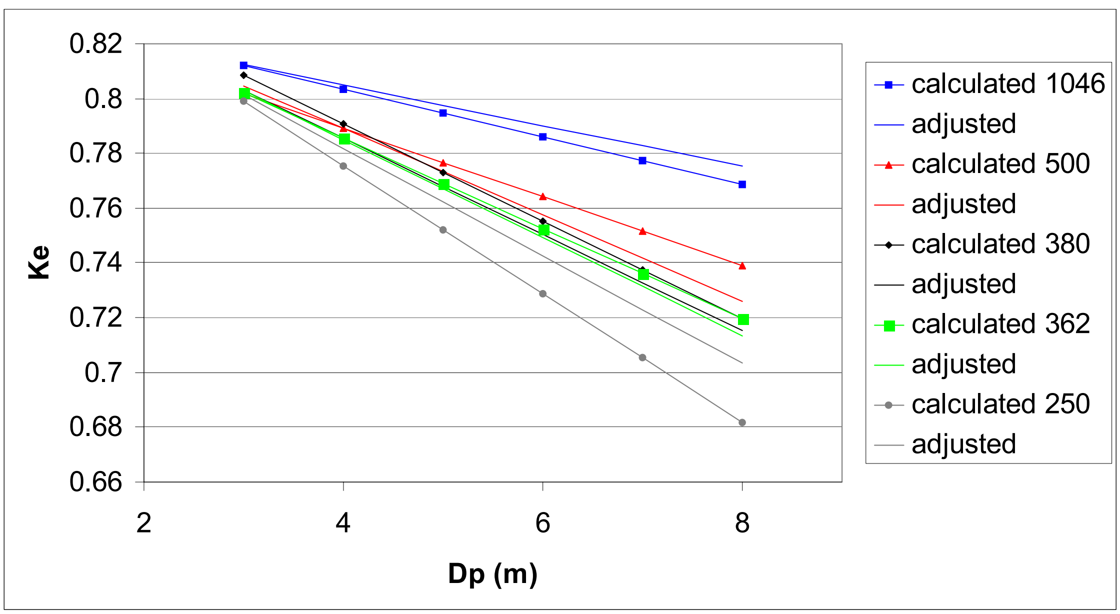


Figure 25. Calculated results in figure 19 compared with equation 39 (fitted)

7.4 Effect of propeller number

The first case considered is a comparison of a single propeller and twin propeller alternatives for a given ship. The basic design is kept the same and in this example the wake and the hull efficiency are presumed to be the same for single and double screw solutions. Therefore the required open water thrust and the channel resistance are constant. The idea is to calculate the power requirement for single screw and double screw solutions with different propeller diameters. This example presents how the number of propellers and propeller diameter affects the power requirement.

The open water resistance (v=15 kn) is taken as 500 kN (MT Sotka) and the channel resistance at 5 kn as 540 kN. The propeller characteristics are taken from the Wageningen B4-85-series for FP propeller (figure 19). The draught of the ship is taken as 9.5 m and the breadth of the ship is big enough to allow a double screw solution with all reasonable propeller diameters. The power is calculated using both the power requirement equations and constants and the new, D_p-dependant K_e values.

The power is calculated with four different propeller diameters. The propeller diameter is usually taken as big as possible, D_p/T is normally between 0.6-0.7. (5.7-6.65 m in this example). The smaller propeller diameters are attained by using equation 28 (4.03-4.7 m). Therefore the rule power requirement gives the same power for a single screw vessel with bigger propeller diameters and for a double screw vessel with smaller propeller diameters.

7.4.1 Rule power requirement, K_e=constant

The propulsion power is calculated directly with equation 19. The K_p-coefficients are 2.26 for a single and 1.6 for a double screw vessel (FP-propeller). The results are collected into table 4 and shown in figure 26.

7.4.2 New formula, $K_e=K_e(D_p)$

In this case the power is calculated also with the equation 19, but the K_p-coefficient is calculated with equation 36. The K_e-coefficient depends on the propeller diameter as shown in chapter 7.2. For the single screw vessel $K_e = -0.0125D_p+0.839$ (see figure 19).

For the double screw vessel the required open water thrust for each shaft is 250 kN and therefore K_e must be taken as $K_e(D_p)=-0.0234D_p+0.869$ (see figure 19). Then the power is calculated for each shaft (using R_{ch}=270 kN for each shaft) with equations 19 and 36. The total propulsion power is obtained adding both shafts together. The wake and the thrust deduction fractions are kept constant in all cases. In reality these factors change and therefore also the results change slightly. The results are collected into table 4 and shown in figure 26.

Table 4. Power requirement for different propeller installations

		Rule requirement				New formula						
		Single screw		Double screw		Single screw			Double screw			
Dp	Dp/T	Kp	**Ps (kW)**	Kp	**Ps (kW)**	Ke	Kp	**Ps (kW)**	Ke (1 prop.)	Kp (1 prop.)	Ps (1 prop.)	**Ps tot (kW)**
4.03	0.42	2.26	**7073**	1.6	**4982**	0.798	1.994	**6209**	0.775	2.048	2255	**4510**
4.7	0.49	2.26	**6034**	1.6	**4271**	0.78	2.029	**5417**	0.759	2.114	1996	**3992**
5.7	0.60	2.26	**4975**	1.6	**3522**	0.768	2.07	**4557**	0.736	2.213	1722	**3444**
6.65	0.70	2.26	**4265**	1.6	**3065**	0.756	2.12	**4000**	0.713	2.321	1548	**3096**

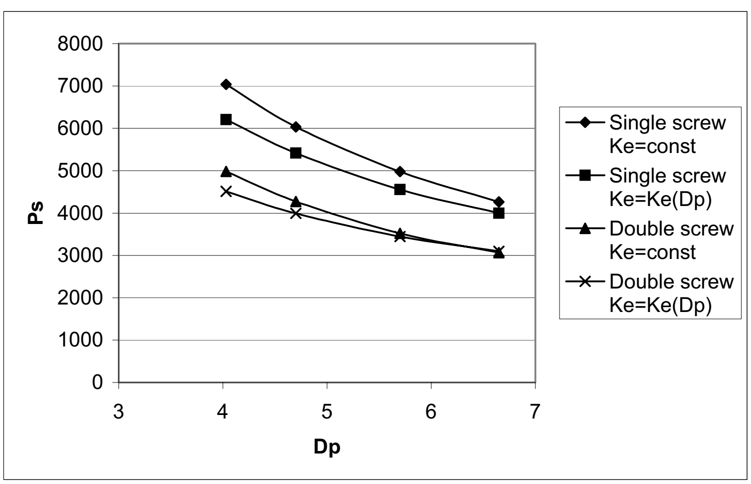


Figure 26. Power requirement for different propeller installations

As this example shows, it is reasonable to install as large propeller as possible because the efficiency of the propeller increases as the propeller diameter increases. Using constant or D_p-dependent K_e values gives slightly different results. The biggest difference is in the steepness of the curves as can be seen in figure 26. The D_p-dependant K_e values give slightly smaller power values over the whole range of diameters but it must be kept in mind that the absolute values for power calculated with the D_p-dependant K_e values could include some inaccuracy caused by the factors mentioned earlier.

7.5 Effect of thrust deduction and wake

In this second case the effect of thrust deduction and wake to the power requirement in the single and double screw vessels is studied. The power requirement is calculated for two vessels (Sotka and Tervi, originally 1CPP-vessels) with different propulsion configurations. In this example the thrust deduction fraction and the wake fraction are calculated separately for single screw and double screw options and the effect of the difference in hull efficiency (ξ_H=(1-t)/(1-w)) to the power requirement is considered.

For these vessels the open water resistance (towing resistance) and the rule channel resistance can be calculated. The hull geometry does not change when changing the design from single to double screw (in view of the resistance) and therefore the towing resistance and channel resistance are the same in both cases. The difference is in the hull efficiency which affects to the required open water thrust and therefore also to the propeller characteristics. The propeller optimisation and calculation of the K_e-coefficient is done according to the procedures presented in chapters 6 and 7.2 using again the Wageningen B4-85 propeller series (FP-propellers).

The propeller diameters for single screw options are the normal diameters for these vessels (5.45 m (Sotka) and 7.4 m (Tervi)). For double screw options the propeller diameters are scaled down with equation 28 (3.854 m (Sotka) and 5.23 m (Tervi)) i.e. the rule power requirement is the same than for single screw option. The power requirements are also calculated for double screw options with bigger propeller diameters (5.45 m (Sotka) and 7.4 m (Tervi)). The results of all calculations are collected into table 5.

The thrust deduction fraction and the wake fraction are estimated with simple Taylor equations (Tornblad 1987):

$w = 0.5C_B - 0.05$ (for 1 propeller) and $w = 0.55C_B - 0.20$ (for 2 propellers)
$t = 0.6w$ (for 1 propeller) and $t = 1.25w$ (for 2 propellers)

For single screw options the thrust deduction fraction can be estimated with more accurate Holtrop method (Holtrop 1984). The open water resistance (v=15 kn) was also calculated with Holtrop method. The open water resistance and the thrust deduction fraction give the required total thrust in each case as $T_{req}=R_{ow}/(1-t)$. In double screw options the thrust for one shaft is naturally half of that.

When T (per shaft), D_p, V_A ($V_A=V(1-w)$) and ψ are known, the optimum pitch to diameter ratio and propeller revolutions can be determined for all cases using the Wageningen B4-85 propeller series (FP-propellers). The optimisation procedure is the same than used in chapter 7.2. When the propeller is optimised to open water conditions, K_T and K_Q can be determined from the propeller curves and K, K_e and K_p can be calculated (as presented earlier in chapter 6). At this point the thrust deduction fraction in bollard pull situation is estimated to be 0.05 and the same in all cases. Finally the engine power can be calculated with equation 19. The rule power requirement and the difference to the calculated results are also presented in table 5.

Table 5. Calculation results in example 2.

Ship	propellers	D_p (m)	C_B	t_{ow}	w_{ow}	ξ_H	R_{ow} (kN)	T_{ow} (kN)	T_{ow}/shaft	P/D_{opt}	n_{opt} (1/min)
Sotka	1	5.45	0.737	0.189	0.3185	1.19	406	500.6	500.6	1	95
	2	3.854	0.737	0.257	0.205	0.93	406	546.4	273.2	1.05	143
	2	5.45	0.737	0.257	0.205	0.93	406	546.4	273.2	1.3	72
Tervi	1	7.4	0.837	0.194	0.3685	1.28	843	1045.9	1045.9	0.95	72
	2	5.23	0.837	0.325	0.26	0.91	843	1248.9	624.4	0.95	116
	2	7.4	0.837	0.325	0.26	0.91	843	1248.9	624.4	1.1	61

Ship	propellers	D_p (m)	K_T	K_Q	K	t_{bp}	K_e	K_p	R_{ch} (kN)	**P_s (kW)**	**$P_{s, rule}$ (kW)**	diff.
Sotka	1	5.45	0.48	0.072	2.773	0.05	0.780	2.028	539.4	**4662**	**5195**	533
	2	3.854	0.505	0.08	2.720	0.05	0.765	2.088	539.4	**4799**	**5201**	401
	2	5.45	0.635	0.122	2.581	0.05	0.726	2.258	539.4	**3671**	**3678**	7
Tervi	1	7.4	0.4525	0.0653	2.792	0.05	0.785	2.008	1055	**9298**	**10465**	1167
	2	5.23	0.4525	0.0653	2.792	0.05	0.785	2.008	1055	**9303**	**10483**	1181
	2	7.4	0.535	0.088	2.704	0.05	0.761	2.106	1055	**6897**	**7409**	512

If the power was calculated for Sotka's double screw solution as in example one (keeping t and w as constant), the required power would have been 4670 kW (see figure 26) while here in example two the result is 4799 kW.

There is only a small difference in the required power between single and double screw solutions when the propeller diameter is scaled down with equation 28 (i.e. rule requirement results in the same power). Even though the required open water thrust is bigger in double screw solution, the required power is almost the same as in single screw solution. This is due to the fact that with smaller propeller diameters the K_e –coefficient is higher than with bigger diameters and this compensates the difference caused by the required open water thrust. These results are understandable when they are considered together with figure 24.

The difference in the hull efficiency ξ_H between single and double screw options can be seen only as an increase in the required open water thrust and therefore it affects only to the propeller optimisation. Smaller P/D gives higher values for K_e. The hull efficiency does not affect to the rule channel resistance and it is constant for all propeller configurations.

In double screw solutions the same effect can be seen as in example one. With bigger propeller diameters the required power is lower than with smaller diameters but with the new calculations the difference is smaller than with the rule power requirement i.e. the new calculations favour smaller vessels compared to the 1999/2002 rule requirements.

7.6 Validation of the calculations

For validation purposes, a databank from existing ships was collected. This databank should include vessel characteristics, D_p and bollard pull. The problem is that for most commercial cargo vessels the bollard pull has not been measured. Usually the bollard pull is important only for tugs, supply vessels and icebreakers. Because of this, most of the vessels in the databank are of these types and only few of them are cargo vessels. The databank consists of total 71 vessels for which the engine power, propeller diameter and bollard pull are given and therefore K_e can be calculated. The databank is given in appendix 3 (with references).

The other problem with the databank is that the vessels are equipped with different types of propellers. There are both CP and FP propellers, both of them with and without a nozzle. For some vessels the information about the nozzle is missing. Also single, double and triple screw installations are included in the databank. Only a few vessels are installed with exactly the same type of propulsion system. One possibility to go around this problem is to scale the calculated K_e values by assuming that $K_{e,FP}=0.9*K_{e,CP}$ and $K_{e,nozzle}=1.3*K_{e,no\ nozzle}$. The number of the propellers is taken into account with equations 31 and 32. This of course reduces the accuracy of the results but gives a possibility to compare the theoretical results against a fairly large group of existing vessels. The measured bollard pull values, calculated K_e-values and the scaled values are presented in appendix 3.

From the data given in the databank, K_e is calculated from equation 25 and is plotted against the propeller diameter. The results are presented below in figures 27-30. The results are divided into categories based on the type of the propulsion system. In these figures the normally used constant values for K_e are given in the legend in parenthesis.

The scaled (adjusted) values which take into account the number of propellers, the propeller type and nozzle are presented against the propeller diameter in figure 31. The K_e values are scaled to match a single CP-propeller without a nozzle. The difference in the thrust deduction in single, double and triple screw vessels is not taken into account.

As presented earlier, K_e is a function of both the propeller diameter and the required open water thrust. Therefore in figure 32 the scaled K_e values are plotted against the propeller diameter and the bollard pull. In this figure the diameter of the circles represents the scaled value of K_e and therefore figure 32 is best viewed together with figure 31. Even though the bollard pull does not correlate directly with the required open water thrust, especially in vessels which are designed for good bollard pull, it gives a rough estimate on the combined effect of thrust and propeller diameter on K_e.

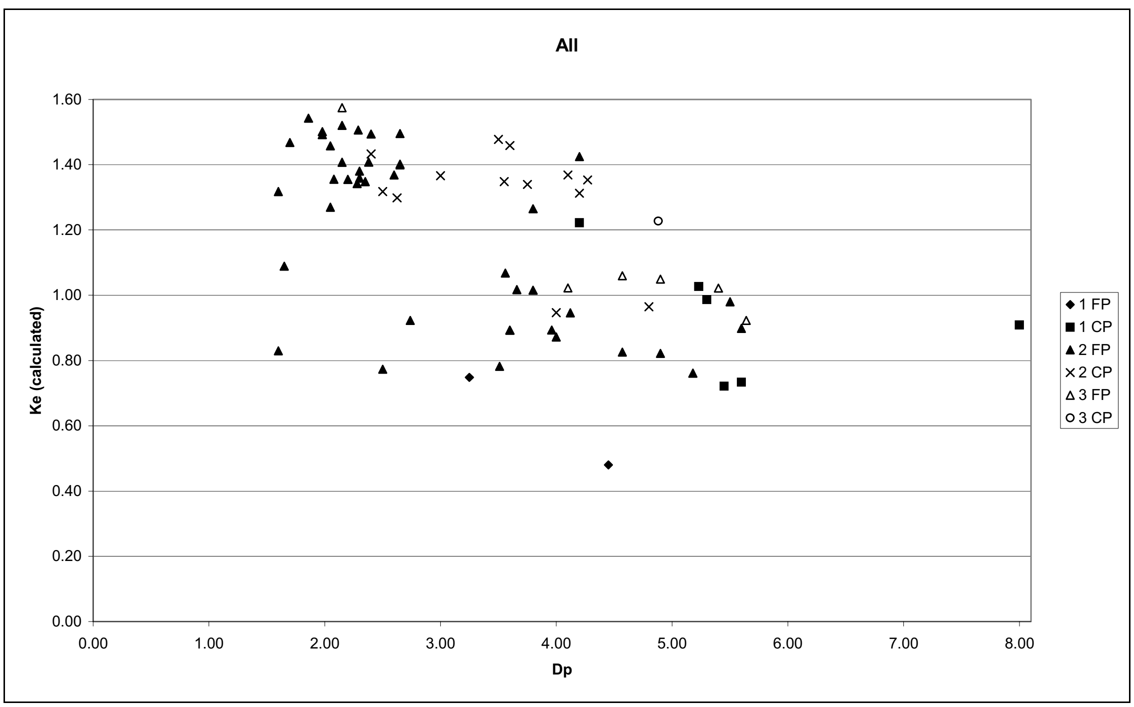


Figure 27. K_e values calculated from the databank.

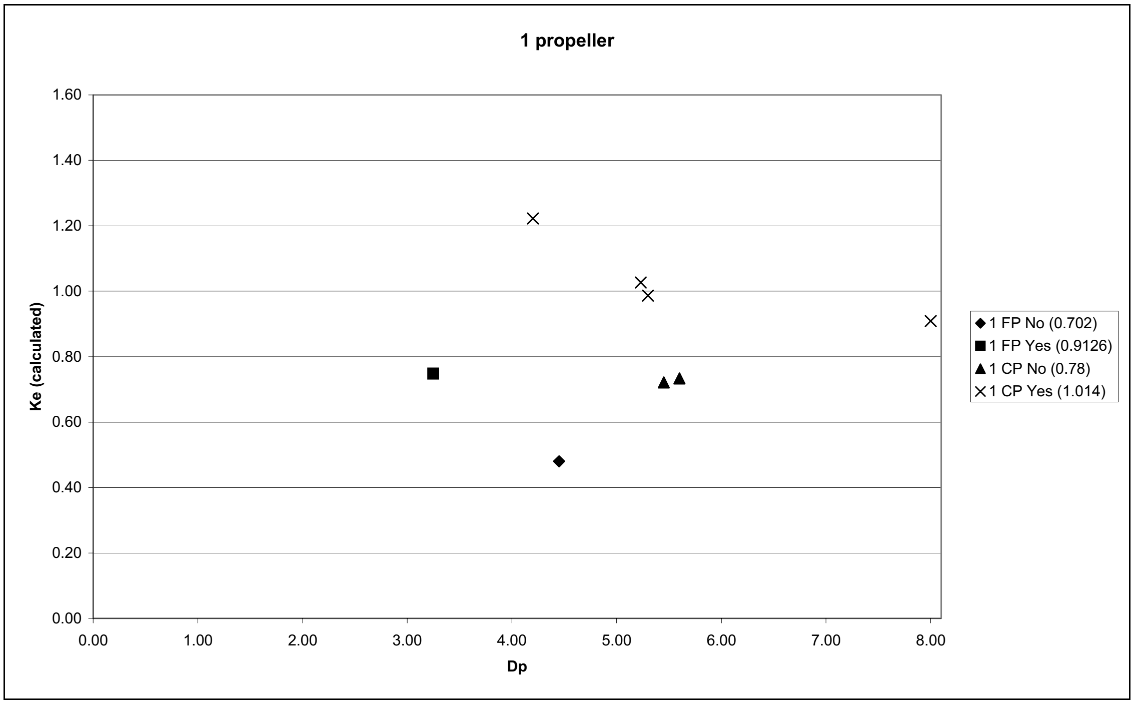


Figure 28. Calculated K_e values for single screw ships (Yes or No refers to a nozzle)

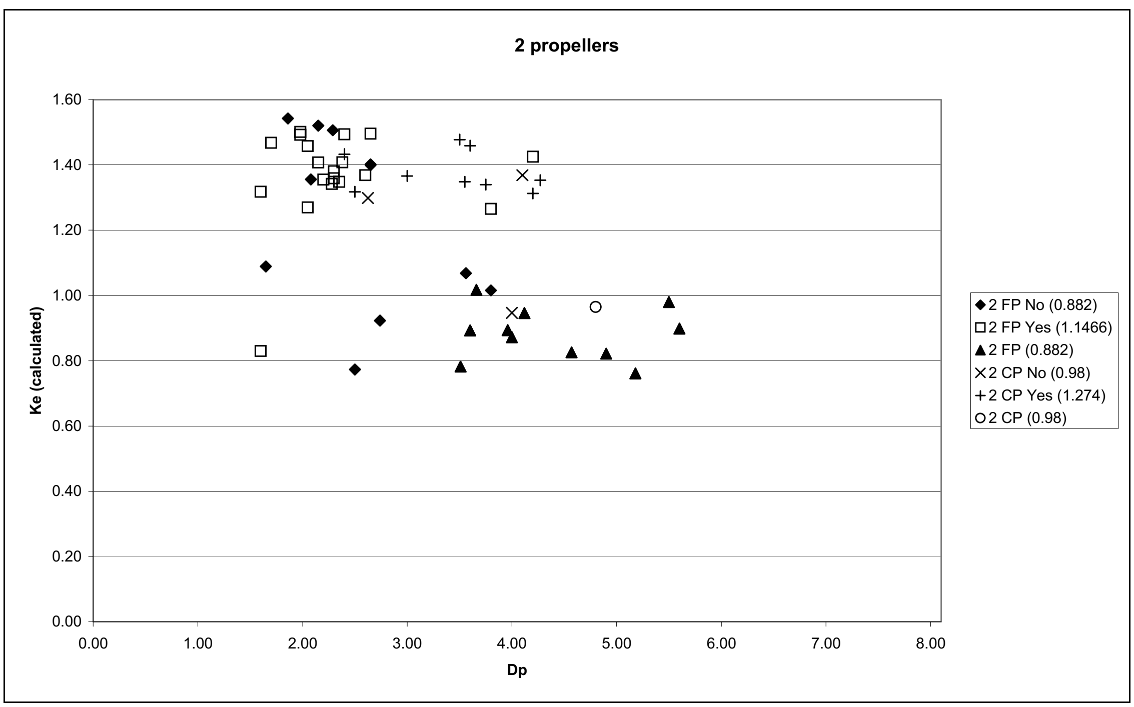


Figure 29. Calculated K_e values for double screw ships

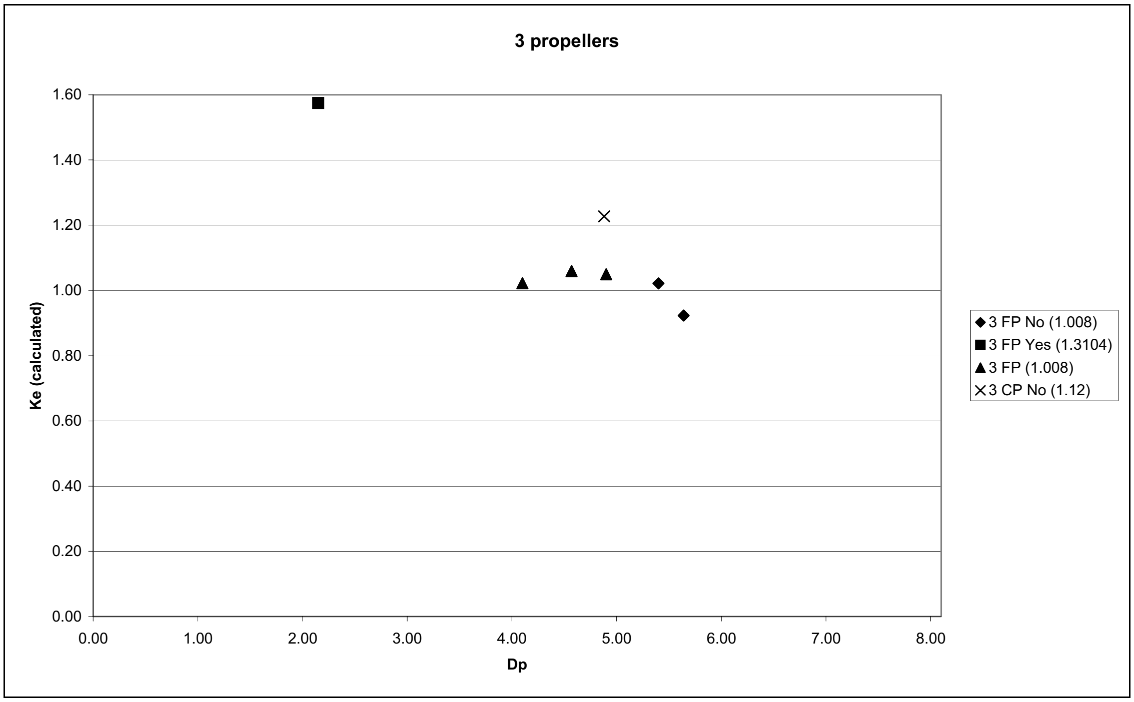


Figure 30. Calculated K_e values for triple screw ships

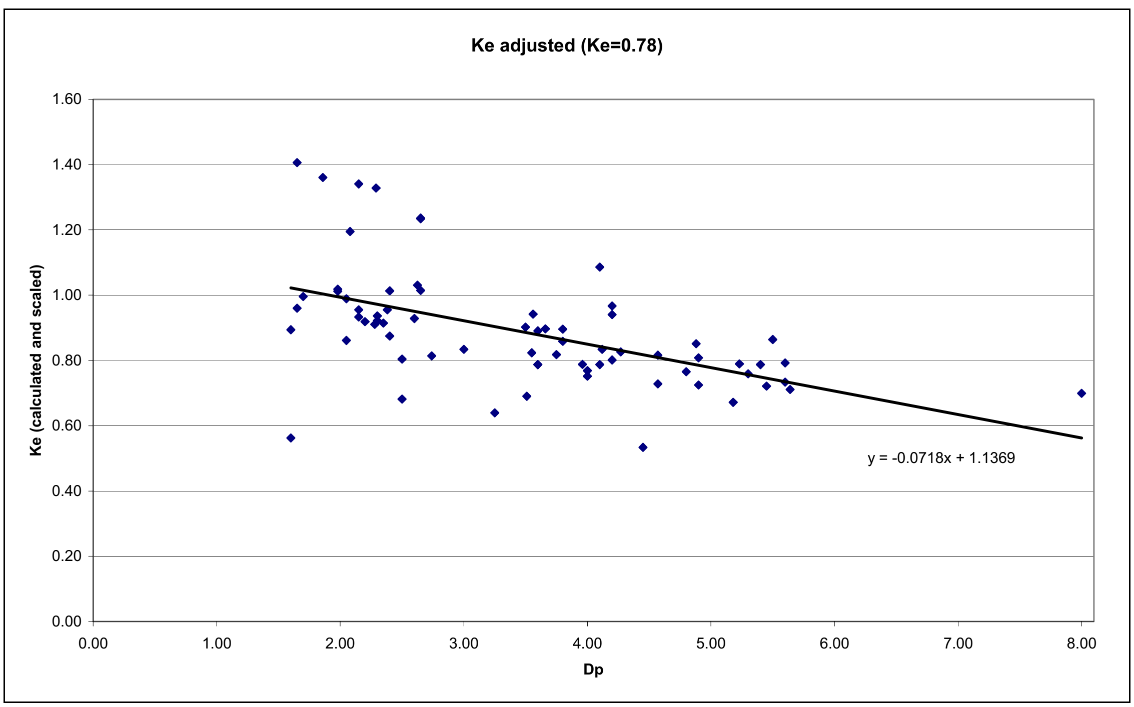


Figure 31. Adjusted K_e values. Values are scaled to match a single CP propeller without a nozzle. The trendline is the least squares fit.

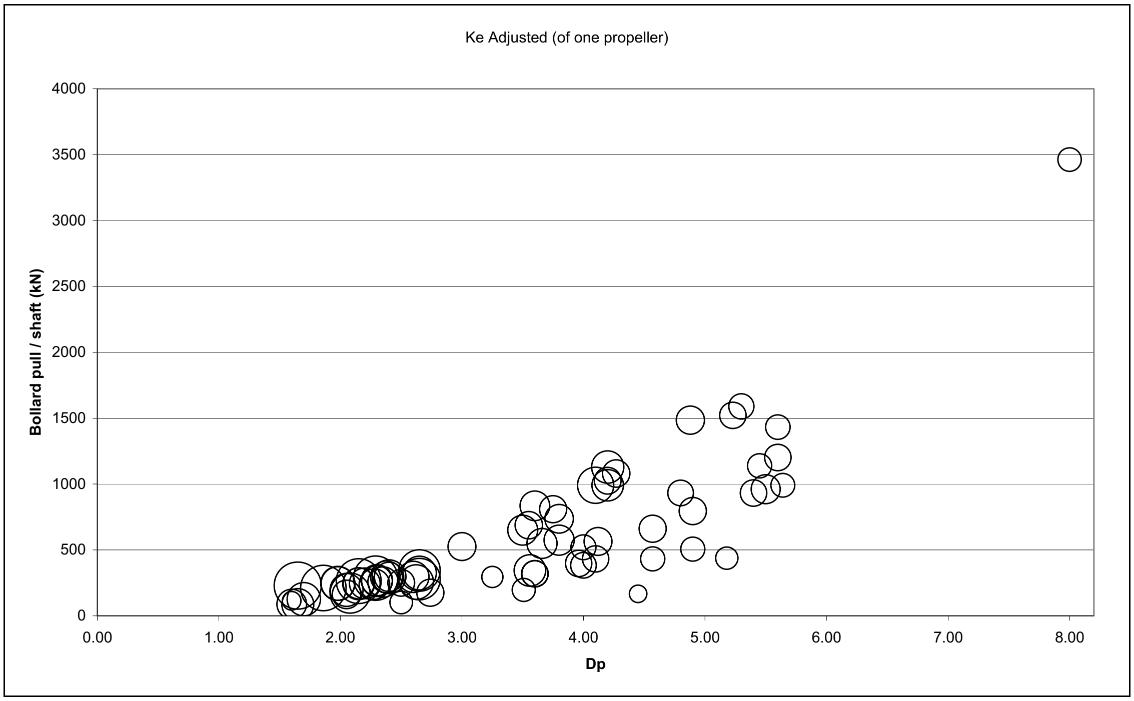


Figure 32. Adjusted K_e values versus the propeller diameter and the bollard pull. The diameter of the circles represents the value of K_e. Values are scaled to match a single CP propeller without a nozzle. Bollard pull is for one shaft.

The calculated and scaled K_e values imply that the value of K_e decreases when the propeller diameter increases. The change is not big and the effect on the value of the bollard pull is fairly small. The scaled values have to be considered with some caution because the scaling factors (FP/CP, Nozzle/No nozzle) are only estimates. Also the scaling between the different number of propellers include the same assumptions as the original power requirement formula (t, w, ξ_H are constant). Therefore the scaling includes several factors of uncertainty. Also there is a lot of scatter in the calculated results.

The values calculated and scaled to match the single CP propeller without a nozzle are close to the normally used constant value (0.78). The scaled Ke values show the tendency to decrease as the propeller diameter increases. The values related to smaller propeller diameters seem to be higher than the constant value but with propeller diameters from 5 to 6 meters the values are very close to the nominal value 0.78 assumed. Most of the vessels in the databank are designed especially for good bollard pull and this explains why quite many of the vessels, especially smaller ones, have K_e-values higher than 0.78.

The calculated and scaled results support the theoretical calculations conducted in previous chapters 7.2 and 7.3. The theoretical calculations were made for cargo vessels and the optimisation was done for open water conditions and speeds. Despite of the inaccuracies in calculations and difference in design bases, the theoretical calculations (figure 24) and the calculated K_e-values (figure 32) show similar tendencies to decrease as the propeller diameter increases and the required thrust decreases.

8 COMPARISON OF BUILT SHIPS

The values for the power requirement are calculated for the vessels in table 6. The purpose is to investigate the different rule values and compare these with the theoretical approach on ships from which there exists information about ice performance. For these vessels there is also adequate information on the hull form and the rule channel resistance can be calculated using the equations in chapter 5.3. The table presents the installed power, the new power requirement (2002 rule), the old power requirement (1985 rule) and the power requirement calculated with the D_p-dependant K_e values (this is referred in the table as the Theoretical). The theoretical value is calculated using the K_e values in figure 19. These design curves were calculated only for five vessels and therefore for most of the vessels the value for K_e had to be estimated from that figure. The power requirement was calculated for FP-propellers and then it was scaled for CP-propeller by multiplying by 0.9.

The results are collected into figures 33-42, where the calculated power requirements are plotted against the installed power or the ship length.

Table 6. Vessels used in comparison of different rule power requirements

Laiva	Luokka	Bulb	DWT ton	ζ deg	λ1 deg	λ2 deg	L m	B m	T m	Lbow m	Lpar m	Awf m^2	Dp m	Propeller no	 type	Rch rule (2002) kN	Installed power kW	New rule (2002) **kW**	Old rule (1985) kW	Theory (scaled for correct prop. type) kW
Tervi IA	IA	n	45000	53	41	41	193,7	30,2	12	32,9	136	680	7,40	1	CP	1071,1	10800	**9616**	10081	8750
Shuttle Göteborg IA	IA	y		50	90	25	82,5	13	3,6	6,6	74,2	46,7	3,20	1	CP	187,8	2000	**1633**	1625	1441
Link Star IA	IA	y	4017	23	90	41	102,7	17	5,8	29,4	44	340	3,60	1	CP	304,1	2960	**2991**	2685	2639
Solano IA	IA	n	7769	23	31	29	116,3	21	6,2	33,4	51	440	3,80	1	CP	367,7	5520	**3767**	3543	3311
Aila IA	IA	n	4402	23	31	60	97,4	16	5,8	25	33	280	3,60	1	CP	290,2	2960	**2788**	2450	2470
Atserot IA	IA	n	6060	19	47	35	105,3	17,6	6,6	33,5	37	400	3,70	1	CP	312,9	3680	**3037**	2507	2680
Envik IAS	IAS	n	3683	38	39	39	92	16,2	5,2	24,4	38	310	3,05	1	CP	362,0	2740	**4584**	3453	3911
Sotka IAS	IAS	n	15954	24	29	29	150	21,5	9,5	32,4	77	490	5,45	1	CP	541,3	11470	**4691**	6927	4293
Kemira IAS	IAS	n	5583	22	35	35	105	17	6,6	32,3	41	420	4,15	1	CP	375,4	4120	**3557**	3836	3181
Finnoak IAS	IAS	y	5399	19	90	35	116,2	19	6,1	43,1	50	490	3,70	1	CP	488,3	5920	**5921**	5046	5195
Finnmerchant IAS	IAS	y	13025	17	90	24	146	25	8,3	52,9	73	710	5,70	1	CP	739,0	13200	**7154**	8423	6459
Nossan IB	IB	y		35	90	73	84,9	13,2	5,5	14,8	54,9	148	3,20	1	CP	231,9	1470	**2240**	1467	1962
Sirius IB	IB	n		19	61	47	79	12	4,6	24,9	38,4	138	2,20	1	FP	144,8	1300	**1789**	1134	1490
Futura IC	IC	y	95195	34	90	90	231,2	40	14	47,6	131,7	1480	7,80	1	CP	971,0	10860	**7874**	10791	7263
Arkadia IC	IC	y	47442	37	90	67	184,5	32,2	11,7	27,1	126,4	585	5,40	1	CP	711,7	9267	**7137**	7746	6419
Windia IC	IC	n		23	66	39	68,3	11,9	4,1	20,4	25,2	161	2,10	1	FP	90,0	1100	**919**	505	765

The results show in general that for smaller ships the required rule power gets in relative terms larger (figure 35). This is clear as the requirement is based on a set ice thickness. Usually, however, the installed power exceeds the new requirement (figure 36). This shows that usually the ice class ships are designed for some ice going capability. If the more complicated bollard pull quality factor is taken into account, the effect on smaller vessels decreases.

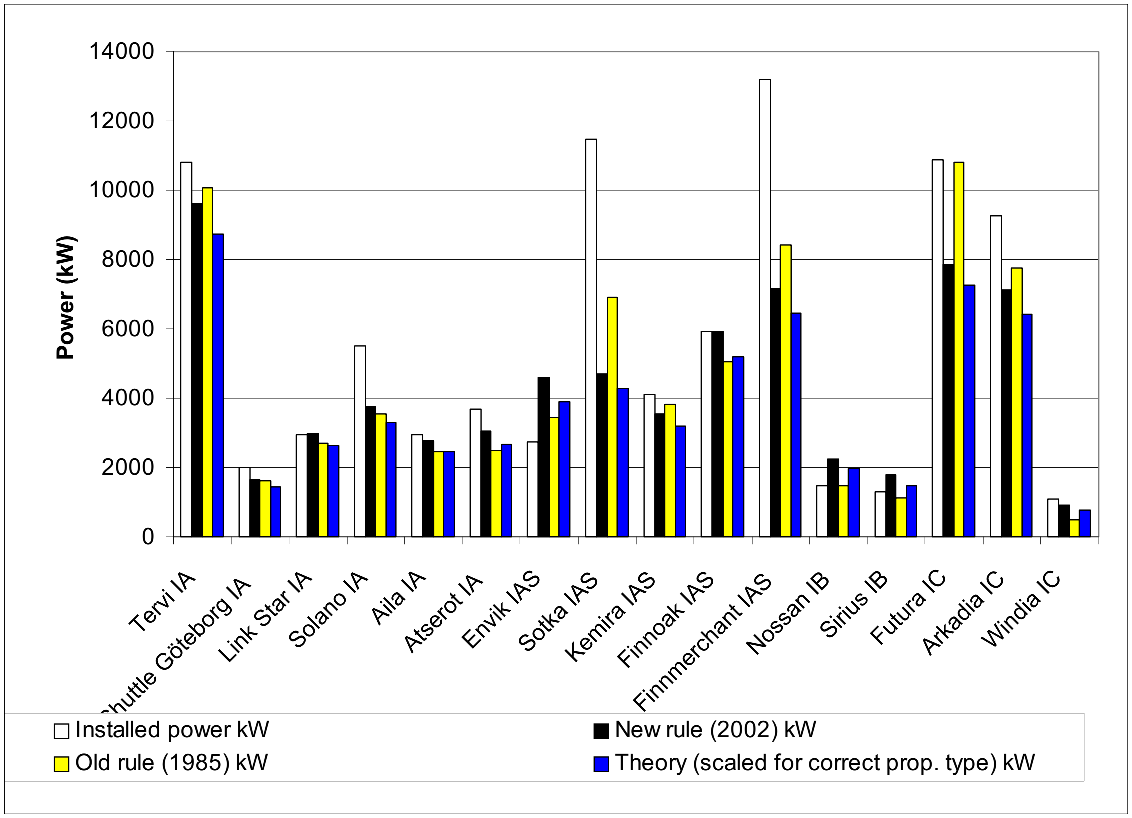


Figure 33. The installed power and the different power requirements for the vessels in table 6.

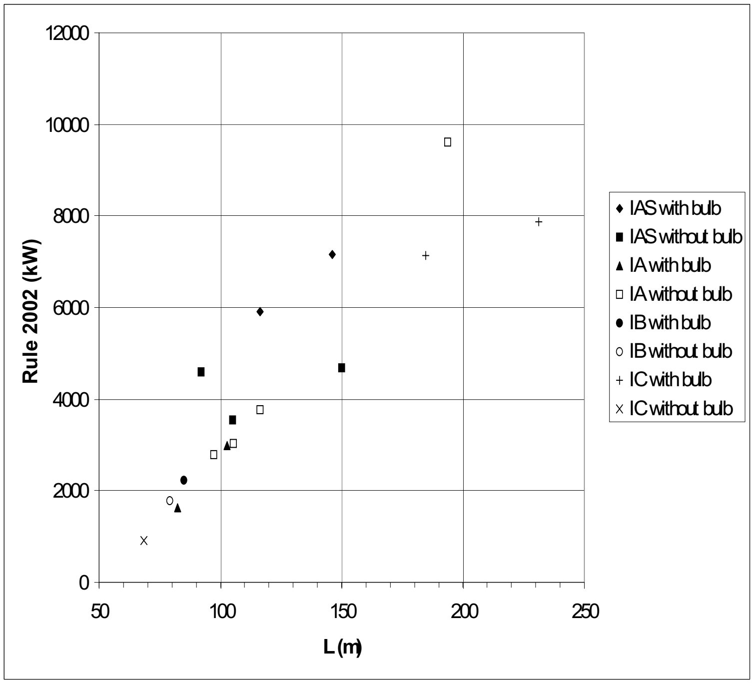


Figure 34. The value of the new power requirement (2002) versus the ship length.

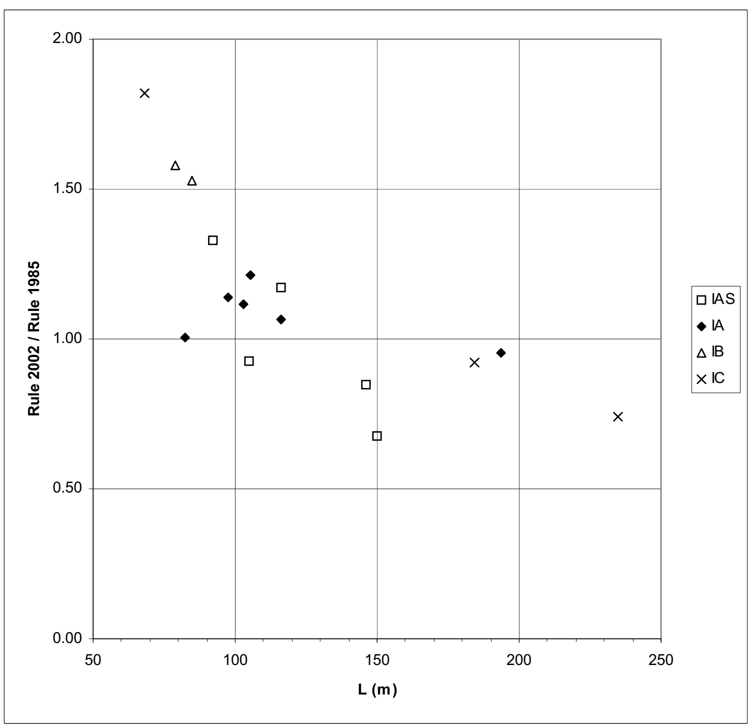


Figure 35. The new power requirement (2002) versus the old requirement (1985) plotted against the ship length.

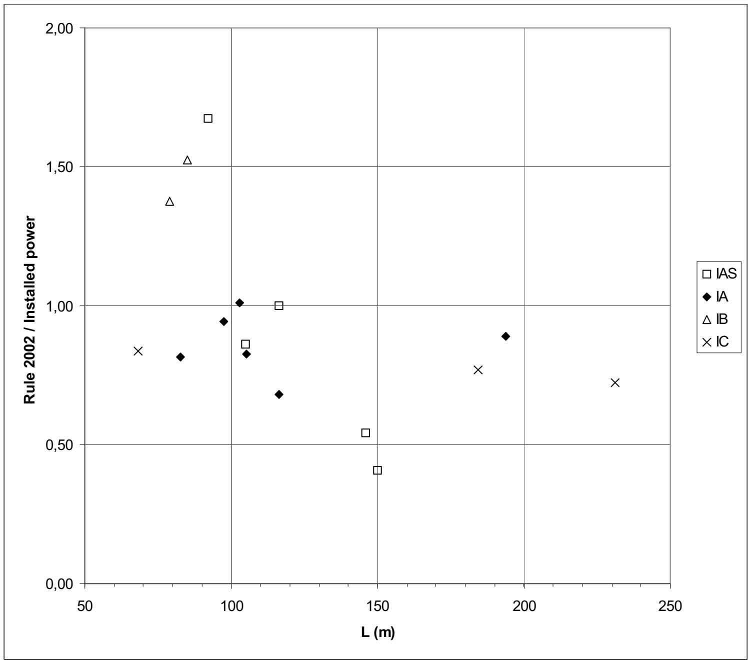


Figure 36. The new power requirement (2002) versus the installed power plotted against the ship length.

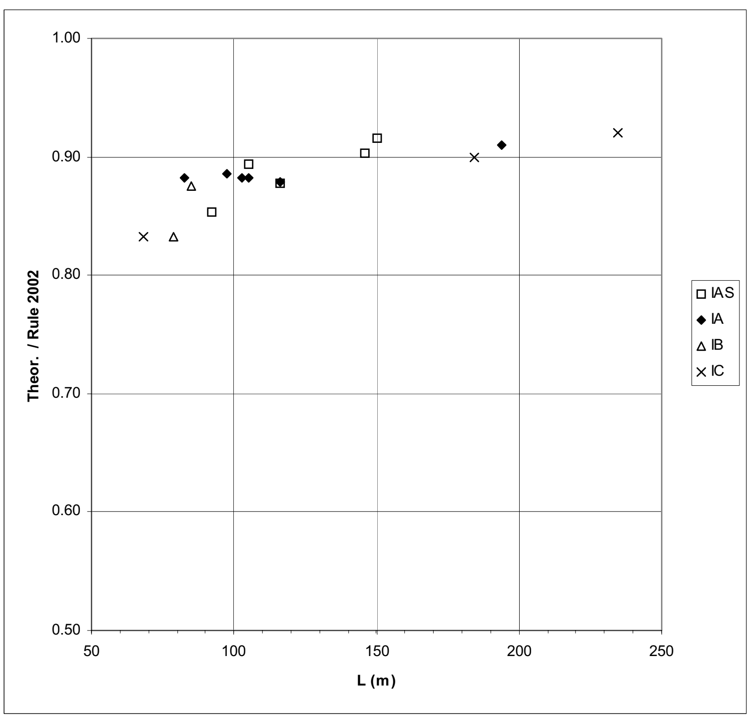


Figure 37. The theoretical power requirement versus the new power requirement (2002) plotted against the ship length.

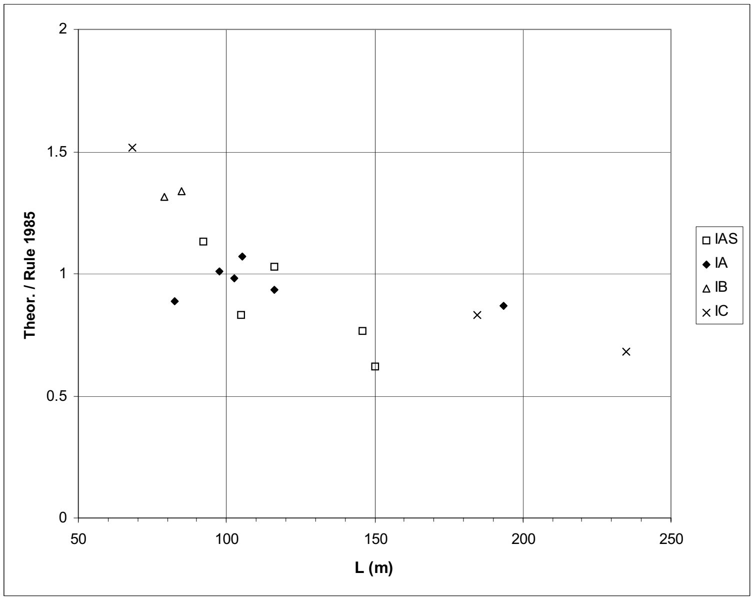


Figure 38. The theoretical power requirement versus the old requirement (1985) plotted against the ship length.

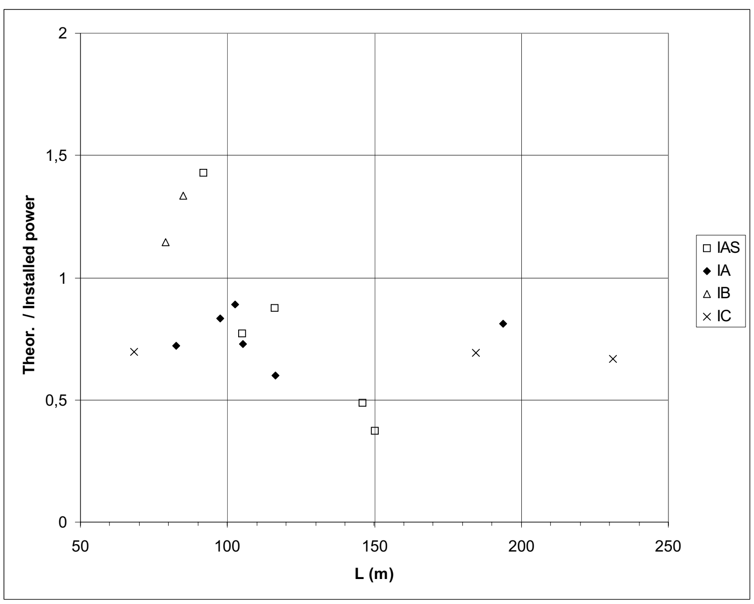


Figure 39. The theoretical power requirement versus the installed power plotted against the ship length.

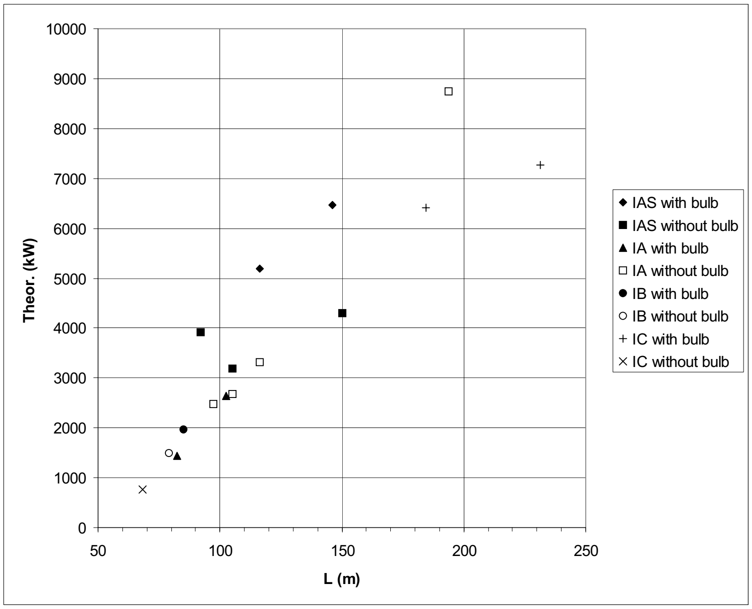


Figure 40. The value of the theoretical power requirement versus the ship length.

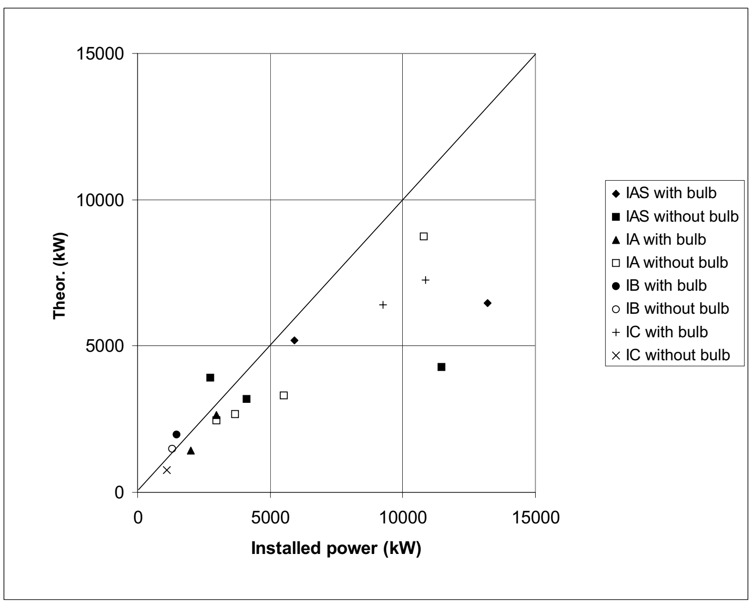


Figure 41. The theoretical power requirement versus the installed power.

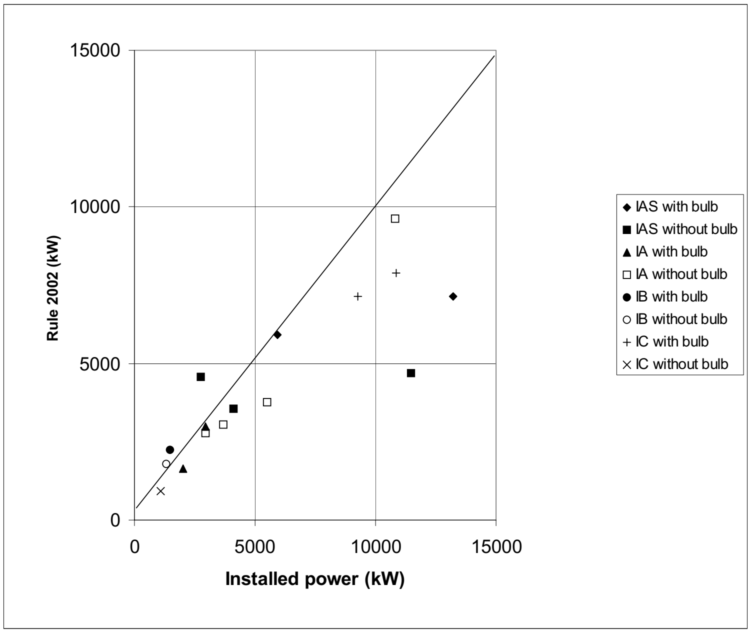


Figure 42. The new power requirement (2002) against the installed power.

9 DISCUSSION AND CONCLUSIONS

The purpose of this study is to present the power requirement in the Finnish-Swedish Ice Class Rules and to give background information on the factors influencing this requirement and the constants included therein.

The power requirement in the Ice Class Rules is based on an explicit performance requirement, which is different for different ice classes. The requirement is stated as minimum speed of five knots in the design ice conditions. The environmental conditions in the Northern Baltic during wintertime, the performance requirement for different ice classes and the background of the performance requirement are briefly described in chapters 2-4. These factors have been presented previously in detail in an earlier report (Riska et al, 1997) and therefore these are not discussed more in this report.

The equations by which the required power can be determined are presented in the Ice Class Rules. The rule channel resistance formula is presented here in chapter 5 and the rule power requirement formula is presented in chapter 6. This power requirement formula and the factors influencing it are the focus of this report. Here it should be remembered that the basic requirement is at least five knots speed in channels of a given thickness. This capability may be demonstrated also by other means than calculations using the given equations.

The rule channel resistance formula and its main components are presented in chapter 5. The derivation of the formula is presented in earlier reports and therefore it is not discussed in more detail in this report. In the resistance formula is notable that it does not make any difference between single and double screw vessels. The formula does not include any geometric information on the aftship and therefore the rule channel resistance is the same for single and double screw vessels. Further it is notable that the formula does not take into account the thrust deduction or the wake, the formula gives only the towing resistance. Propulsion characteristics are taken into account in the bollard pull equation and that way also in the power requirement formula.

The resistance equation is validated with a group of built ships for which the ice going capability is known. These vessels are all single screw cargo vessels with quite conventional hull shapes. Therefore it can be presumed that the resistance equation is well suited for vessels of this type. For other types of vessels, for example icebreakers and tugs, the resistance equation could give inaccurate results. It should be remembered that in the Ice Class Rules the resistance can also be determined by other means than using the given equation.

The power requirement formula is derived from propulsion characteristics in bollard pull situation. In chapter 6 it is shown how the relationship between thrust and propulsion power at bollard pull is derived and how the bollard pull quality factor K_e can be calculated theoretically. Also the derivation of the power requirement formula and the K_p coefficient for different number of propellers is presented in detail. The relationship between thrust and propulsion power at bollard pull, and therefore also the power requirement, is based on a simple empirical formula for bollard pull quality criterion. The K_e coefficient in the bollard pull equation was studied in more detailed fashion. Normally the coefficient is taken as a constant and the values normally used are presented in several technical papers. The K_e-value for a single CP-propeller (0.78) presented in the technical papers is based at least partly on empirical results and is probably originally calculated for tugs. The other K_e-values can be directly calculated from this value as presented in chapter 6. The bollard pull equation includes the thrust deduction and it is a part of the K_e-value.

For double and triple screw vessels the K_e-values and therefore also the K_p-values in the Ice Class Rules are calculated directly from the value of the single CP-propeller by assuming that power density i.e. the propulsion power divided by the propeller disc area is the same in all cases. This assumption requires that the thrust deduction fraction, the wake fraction and therefore the hull efficiency are constant despite of the propeller type or number. Equations 28, 29, 31, 32 and the values in table 3 are based on these assumptions. These relations are valid only if the K_e coefficient for the individual propellers in multi-screw vessels is assumed to be the same as for single screw vessels. The validity of this assumption is discussed in more detail in chapter 7 and that is the main focus of this report. The K_e-values for FP-propellers are calculated directly from the respective values for CP-propellers by multiplying them by 0.9. This factor is also at least partly empirical.

The equations for the K_p-coefficient and the propulsion power are always valid despite of how the K_e-coefficient is achieved. The required power can always be calculated with equations 19 and 36 after K_e is calculated or determined by other means. The factor 0.8 in equation 36 results from the effect of speed (5 kn) on the bollard pull and this factor, even if it is based on an approximative equation (34), seems to be quite correct according to the example calculated in chapter 6.

It is shown in chapter 7 that the value of K_e is not constant but it depends on the propeller diameter and the required thrust. This is shown by examples calculated with the Wageningen Propeller Series. In these calculations the optimum pitch to diameter ratio and the optimum propeller revolutions were determined for four different ships. This optimisation was done for open water conditions and speed (15 kn) and after that the propeller characteristics at bollard pull situation were investigated. The calculations clearly show that K_e is not constant but it decreases as the propeller diameter increases (keeping the required thrust as constant) or as the required thrust decreases (keeping the propeller diameter as constant). The calculations were made for FP-propellers and the calculated K_e-values were significantly higher than the normally used constant value (0.702), especially with smaller propeller diameters. Naturally the optimum P/D and propeller revolutions can not be applied in all cases but they are restricted by several physical limitations. These limitations reduce the calculated K_e-values in some degree but the quantity of this reduction was not investigated further in this study.

The power requirement was calculated for four example ships using the new equations for K_e. Even if the new power requirement was calculated using optimum propellers, the effect of D_p on P_s is shown quite well in these figures (20-23). The principle of installing as big propeller as possible is clear. The calculations only suggest that the steepness of the design curves ($P_s(D_p)$) is slightly smaller than using the constant coefficients.

Also a general equation which takes into account both the propeller diameter and the required thrust was fitted to the calculated results. This equation is also for single FP-propellers and it is calculated with optimum P/D and revolutions. Therefore the restrictions mentioned earlier apply also to this equation. Nevertheless it is an estimate of the general formula for K_e, which can be used if more accurate calculations are not necessary. Note that the thrust T in the equation is the required thrust in open water conditions and speed (15 kn). It would be reasonable to use the rule channel resistance R_{ch} instead of T in the general equation but the relationship between these two was not investigated in this study. If this kind of equation is added to the Ice Class Rules, this relationship between T and R_{ch} should be studied more and the equation should be given as a function of D_p and R_{ch}.

One of the biggest factors of uncertainty in the theoretical calculations is the value of the thrust deduction fraction in the bollard pull situation. The thrust deduction is taken into account in the power requirement as a part of the K_e-coefficient. The thrust deduction fraction and the wake fraction vary in a complicated manner with speed and bollard pull. In some references the value of the thrust deduction in the bollard pull situation is taken as 2 or 3 percent but this is only an estimate. Normally, in open water conditions and speeds, the value of the thrust deduction is larger for double and triple screw vessels than for single screw vessels. It is reasonable to presume the same to hold for bollard pull conditions but the true values of t in open water conditions and speeds, in bollard pull conditions and in ice conditions should be determined in order to achieve accurate results with direct calculations. The effect of the propeller number or diameter on the stern lines and therefore also on the thrust deduction fraction was not studied here. Because these influence the thrust deduction and thus the bollard pull, these factors should be studied more.

Two examples are calculated in chapter 7 which present how the number of propellers, propeller diameter, required thrust, thrust deduction fraction and the wake fraction affects the power requirement compared to the rule requirement. The examples are calculated using the equations derived in chapter 7.

A databank of vessels built was collected in order to validate the theoretically calculated K_e values. The databank (total 71 vessels) consists mainly of tugs, icebreakers and supply vessels. Only few cargo vessels were included in the databank because for most cargo vessels the bollard pull has not been measured. For the vessels in the databank the power, propeller diameter and bollard pull are known and K_e can be calculated. There is quite a lot of scattering in the results mainly because there are several different types of propulsion configurations included in the databank. There are FP and CP propellers, with and without a nozzle and different number of propellers in all possible combinations. The calculated K_e values from the databank were plotted against the propeller diameter but here the dependence of K_e on D_p is not clear because there is always only a few vessels installed with exactly the same type of propulsion.

To compare the theoretical calculations with the databank the calculated K_e-values were scaled to match a single CP-propeller without a nozzle. The scaling includes some factors of uncertainty because the scaling factors are only estimates and they also include the same assumptions as the original power requirement formula (t, w, ξ_H are constant). The calculated and scaled values were plotted against the propeller diameter and also against the propeller diameter and bollard pull. These results support the theoretical calculations conducted in chapter 7. The theoretical calculations were made for cargo vessels and the optimisation was done for open water conditions and speeds. Despite of the inaccuracies in scaling and difference in design bases, the theoretical calculations (figure 24) and the calculated and scaled K_e-values (figures 31 and 32) show clearly similar tendencies to decrease as the propeller diameter increases and the required thrust decreases.

The scaled databank values are roughly at the same level as the normally used constant values or theoretically calculated values. Some vessels have better K_e values than the nominal 0.78 but the reason for this might be that most of the vessels in the databank are of type designed especially for a good bollard pull. Also it is likely that some kind of reserve is included in the constant coefficients or they are some kind of average values.

In chapter 8 the different rule values and theoretical calculations were compared. This comparison was conducted for 16 different single propeller cargo vessels of different sizes and ice classes. For these vessels adequate information on the hull form is available. The most important results are collected into figures 35-38. These figures clearly show how the new 2002 rule increases the power requirement for smaller vessels (L<120 m) compared to the old 1985 rule. The theoretical calculations also show the same tendency to increase the power requirement for smaller vessels but difference is a bit smaller. The theoretical rule requirement is compared with the 2002 rule requirement in figure 37. In this figure the difference between these two is clearly seen. The theoretical calculations give slightly smaller values for required power for smaller vessels (L<120 m). Note that the theoretical values are smaller than the 2002 rule values over the whole range but this is due to the fact that the theoretical values are calculated for optimum propellers and therefore they are about 10 percent smaller than the rule values. Even if the theoretical equations were scaled to give the same power for bigger vessels than the 2002 rule, the theoretical rule requirement would still give about 5-10 percent smaller power requirements for smaller vessels.

All the calculations and information imply that the theoretical calculations conducted in chapter 7 are correct in general terms. The K_e-coefficient in the bollard pull equation is not constant but it depends on the propeller diameter and the required open water thrust. The collected databank supports these calculations and the influence of them on the power requirement is presented with examples in chapter 8. The theoretical calculations give slightly smaller values for power than the constant coefficients but the difference is due to several different factors. One of the most important one is that the theoretical calculations are made for optimum propellers and it is not always possible to apply them to actual ships due to several constraining factors.

The results presented above are valid for a single propeller. For double screw (or triple screw) vessels the calculations are slightly more complicated as presented in the examples in chapter 7. In double and triple screw vessels the hull shape is normally quite different than in single screw vessels and therefore the hull efficiency is smaller. Also the cross coupling of the propellers and the forces induced by them cause even more uncertainty to the calculations. Therefore the results for double and triple screw vessels must be considered with some caution.

The trust deduction fraction affects K_e quite significantly but regardless of that the effect of it on the power requirement is not large. For single screw vessels the constant K_e-values give roughly correct values for power and the results are more on the conservative side. The rule requirement is bigger than the theoretical requirement for all ship sizes and propeller diameters. Especially for smaller vessels the rule requirement is quite stringent. For double screw vessels the theoretical results are closer to results calculated with constant coefficients than for single screw vessels i.e. there is not similar reserve between the rule requirement and theoretical requirement which are calculated with optimum propellers. This margin reduces when propeller diameters increase.

10 REFERENCES

Armstrong T. On the Actual Performance of Tugs. The tenth International Tug Convention, 1988.

Dick R.A, Laframboise J.E, 1989. An Empirical Review of the Design and Performance of Icebreakers. Marine Tehnology, Vol. 26, No. 2, April 1989, pp 145-159.

Guiper G, 1992. The Wageningen Propeller Series. MARIN Publication 92-001, 1992.

Holtrop J, 1984. A statistical re-analysis of resistance and propulsion data. ISP Vol. 31, No 363, 1984.

Isin Y.A, 1987 Practical Bollard-Pull Estimation. Marine Tehnology, Vol. 24, No. 3, July 1987, pp 220-225.

Kujala P, 1991. Damage Statistics of Ice-Strengthened Ships in the Baltic Sea 1984-1987. Winter Navigation Research Board, Report 50, Helsinki, 66 p, 1991.

Kujala P, Sundell T, 1992. Performance of Ice-strengthened Ships in the Northern Baltic Sea in Winter 1991. Helsinki University of Technology Ship Laboratory, Report M-117, 1992.

Lancaster G.H, Haines J.R, 1979. Towing Vessel Screw Propulsion. The Sixth International Tug Convention London, Thomas Reed Industrial Press, 1979.

Lehtinen P, 1993. Performance of Ice-Navigating Ships in the Northern Baltic in Winter 1993. Helsinki University of Technology Ship Laboratory, Report M-182, 1993.

Lehtinen P, 1994. Observations on Ice Navigation Performance of Ships in the Baltic in Winter 1994. Helsinki University of Technology Ship Laboratory, Report M-187, 1994.

Leiviskä T, 2001. Esitutkimus laivan propulsiosta jäissä, unpublished manuscript, 2001.

Leppäranta M, et al, 1988. The Phases of the Baltic Winter. Finnish Institute of Marine Research, Report 254, Appendix 2, Helsinki, 83 p, 1988.

Matusiak J, 1993. Laivan Propulsio. Helsinki University of Technology Ship Laboratory, Report M-176, 1993.

Pöntynen H, 1992. Performance of Ice-Navigating Vessels in the Northern Baltic in Winter 1992. Helsinki University of Technology Ship Laboratory, Report M-123, 1992.

Riska K, Wilhelmson M, Englund K, Leiviskä T, 1997. Performance of Merchant vessels in the Baltic. Research report no 52, Winter Navigation Research Board, 1997.

Tornblad J, 1987. Marine Propellers and Propulsion of Ships. Marine Laboratory KaMeWa AB, 1987.

Tsoy L.G, 1983. Modelling of vessel’s movement in a channel broken up by icebreaker. POAC’83, Proceedings vol. 2, Helsinki, 5-9 April. VTT symposium 28. 1983.

Appendix 1. The vessels used in the data collection campaign during winters 1991-1994.

Ship	Class	Type	L	B	T	Ps (kW)	Dp	÷ (t)	1991	1992	1993	1994
Aila	IA	Dry cargo	97.4	16	5.8	2960	3.6	6320	x			
Degerö	IA	Ro-Ro	130.7	21	6.2	5520	3.8	12341	x			
Envik	IA Super	Cement carrier	96	16.2	5.2	2740	3.05	5583	x			
Kemira	IA Super	Chemic. Tanker	105	17	6.6	4120	4.15	8565	x		x	
Largo	IA	Dry cargo	90.8	15.8	6.4	4500			x		x	
Link Star	IA	Dry cargo	98	17	5.8	2960	3.6	6877	x			
Solano	IA	Ro-Ro	116.3	21	6.2	5520	3.8	10458	x		x	
Atserot (ex: Tebostar)	IA	Tanker	105.3	17.6	6.6	3680	3.7	7810	x			
Sotka/Tiira	IA Super	Tanker	150	21.5	9.5	11470	5.45	22033	x	x		x
Bremön	IB	Gen. Cargo	111.4	16.62	7.65	3119				x		
Haväng				14.4		3180				x		
Klenoden	IA	Dry cargo	96.9	16	6.07	2990	3.5			x		
Madzy	IA	Bulk carrier	134.8	18.6	7.56	4413				x		
Norden	IA	Bulk carrier	134.8	18.6	7.56	4413				x		
Rautaruukki+barge	IA Super	Tug+barge	154.6	27.2	6.7	7680				x		
Sambre		Gen. Cargo	83	12.6		1500				x		
Star Skoganger		Bulk carrier	155.3	22.9	8.65	9300				x		
Stella Atlantic										x		
Toftön		Gen. Cargo	91.5	13.95	6.07	2060				x		
Dimitry Pozkorskiy										x		
Kapitan Nazarev		Bulk carrier	154.9	22.9	9.88	8238				x		
Kuzma Minin		Bulk carrier	154.9	22.9	9.88	8238				x		
Viktor Tkattchyov		Bulk carrier	154.9	22.9	9.88	8238				x		
Finnoak (ex: Ahtela)	IA Super	Ro-Ro	112	19	6.1	5920	3.7	9200		x		
Christina	IA	Dry cargo	96.9	16	6.07	2990	3.5			x		
Gunilla	IA	Ro-Ro	108.7	16	5.95	6620				x		
Klosterfelde	IB	Gen. Cargo		14.6	7.2	2360				x		
Lovisa Gorthon	IA Super	Ro-Ro		21	6	4200				x		
Granö/Hamnö	IA Super	Ro-Ro	122	19	6.1	5920					x	
Ministar	IA	Dry cargo	107.5	17	5.8	2960					x	
Outokumpu	IA	Bulk carrier	99	16	6	2600					x	
Winden/Najaden	IA	Dry cargo	104.8	16	5.8	2960						x
Finnfighter	IA Super	Dry cargo	159.2	21.1	9.13	7281						x
Borden	IA Super	Ro-Ro	142.3	19.2	7.02	8828						x

Appendix 2. Power requirement for example ships in chapter 7.2.

ENVIK		Rch	362	(rule 02)				Envik	Envik	Difference
D	P/D	K	ψ	t	Ke	Kp,Theor.	Kp,rule	Ps,Theor.	Ps,rule	%
3	0.81	2.87531	1.025	0.05	0.80884	1.92	2.26	4411	5189	-15.0
3.05	0.8154	2.87215	1.025	0.05	0.80795	1.92	2.26	4346	5104	**-14.9**
4	0.918	2.81215	1.025	0.05	0.79107	1.99	2.26	3420	3891	-12.1
5	1.026	2.749	1.025	0.05	0.77331	2.06	2.26	2831	3113	-9.1
6	1.134	2.68584	1.025	0.05	0.75554	2.13	2.26	2443	2594	-5.8
7	1.242	2.62268	1.025	0.05	0.73777	2.21	2.26	2170	2224	-2.4
8	1.35	2.55952	1.025	0.05	0.72001	2.29	2.26	1969	1946	1.2

KEMIRA		Rch	375	(rule 02)				Kemira	Kemira	
D	P/D	K	ψ	t	Ke	Kp,Theor.	Kp,rule	Ps,Theor.	Ps,rule	
3	0.85	2.85192	1.025	0.05	0.80226	1.94	2.26	4708	5471	-13.9
4	0.95	2.79344	1.025	0.05	0.78581	2.01	2.26	3642	4103	-11.2
4.15	0.965	2.78467	1.025	0.05	0.78334	2.02	2.26	3527	3955	**-10.8**
5	1.05	2.73496	1.025	0.05	0.76936	2.07	2.26	3008	3282	-8.4
6	1.15	2.67648	1.025	0.05	0.75291	2.14	2.26	2589	2735	-5.3
7	1.25	2.618	1.025	0.05	0.73646	2.21	2.26	2294	2345	-2.2
8	1.35	2.55952	1.025	0.05	0.72001	2.29	2.26	2076	2051	1.2

TERVI		Rch	1071	(rule 02)				Tervi	Tervi	
D	P/D	K	ψ	t	Ke	Kp,Theor.	Kp,rule	Ps,Theor.	Ps,rule	
3	0.7897	2.88718	1.025	0.05	0.81218	1.91	2.26	22307	26404	-15.5
4	0.8426	2.85625	1.025	0.05	0.80348	1.94	2.26	17003	19803	-14.1
5	0.8955	2.82531	1.025	0.05	0.79478	1.97	2.26	13826	15842	-12.7
6	0.9484	2.79438	1.025	0.05	0.78607	2.01	2.26	11714	13202	-11.3
7	1.0013	2.76344	1.025	0.05	0.77737	2.04	2.26	10210	11316	-9.8
7.4	1.02246	2.75107	1.025	0.05	0.77389	2.05	2.26	9723	10704	**-9.2**
8	1.0542	2.7325	1.025	0.05	0.76867	2.07	2.26	9086	9902	-8.2

SOTKA		Rch	541	(rule 02)				Sotka	Sotka	
D	P/D	K	ψ	t	Ke	Kp,Theor.	Kp,rule	Ps,Theor.	Ps,rule	
3	0.8523	2.85057	1.025	0.05	0.80188	1.95	2.26	8163	9479	-13.9
4	0.928	2.80631	1.025	0.05	0.78943	1.99	2.26	6268	7110	-11.8
5	1.0037	2.76204	1.025	0.05	0.77698	2.04	2.26	5135	5688	-9.7
5.45	1.037765	2.74212	1.025	0.05	0.77137	2.06	2.26	4763	5218	**-8.7**
6	1.0794	2.71777	1.025	0.05	0.76452	2.09	2.26	4385	4740	-7.5
7	1.1551	2.6735	1.025	0.05	0.75207	2.14	2.26	3852	4063	-5.2
8	1.2308	2.62923	1.025	0.05	0.73962	2.20	2.26	3456	3555	-2.8

Appendix 3. Measured K_e values for vessels included in the databank.

	Shaft Power	Propeller				Bollard			
Ship Name	**Total (kW)**	**Dia. (m)**	**No.**	**CP/FP**	**Nozzle?**	**Pull (kN)**	**Ke**	**Ke scaled**	Reference
William C. Daldy	1454.12	4.45	1	FP	False	166.77	0.480	0.534	http://daldy.q.co.nz
Max Waldeck	2400.00	3.25	1	FP	True	294.30	0.748	0.640	Marine Technology, Vol. 26, No. 2, April 1989
Arctic	10900.00	5.23	1	CP	True	1520.55	1.027	0.790	Marine Technology, Vol. 26, No. 2, April 1989
Kigoriak	12200.00	5.30	1	CP	True	1589.22	0.986	0.759	Marine Technology, Vol. 26, No. 2, April 1989
Lunni-luokka	11500.00	5.45	1	CP	True	1137.96	0.721	0.555	Marine Technology, Vol. 26, No. 2, April 1989
SA-15	15400.00	5.60	1	CP	True	1432.26	0.734	0.564	Marine Technology, Vol. 26, No. 2, April 1989
Sevmorput	29400.00	8.00	1	CP	True	3462.93	0.909	0.699	Marine Technology, Vol. 26, No. 2, April 1989
Zeus	5495.81	4.20	1	CP	True	990.81	1.222	0.940	Ship & Boat Int'l April 1996
Fairplay 21 &24	4101.35	2.65	2	FP	False	686.70	1.400	1.234	Ship & Boat Int'l March 1999
Alaska Mariner	2699.43	2.74	2	FP	False	350.36	0.923	0.814	http://vaporcontrols.com/vessels/alaskamariner.html
Boa Master	2982.80	2.15	2	FP	False	524.84	1.520	1.341	http://www.boa.no/fleet/boamaster.html
Botnica	10000.00	3.80	2	FP	False	1147.77	1.015	0.896	
Cecilia B Statten	3206.51	2.29	2	FP	False	568.98	1.506	1.328	Ship & Boat Int'l September 1998
Irving Juniper	1677.83	2.50	2	FP	False	201.11	0.773	0.682	http://www.atlantictow.nb.ca/atl12.html
J George Betz	4578.60	3.56	2	FP	False	686.70	1.068	0.942	Ship & Boat Int'l March 1996
Lam Tong	2430.98	1.86	2	FP	False	421.83	1.543	1.360	Ship & Boat Int'l March 1996
Sung Kong	2982.80	2.65	2	FP	False	556.23	1.402	1.236	Ship & Boat Int'l April 1995
Tai Tam	1938.82	2.08	2	FP	False	343.35	1.355	1.195	Ship & Boat Int'l April 1995 and March 1996
Fuji	8800.00	4.90	2	FP	False	1010.43	0.822	0.725	Marine Technology, Vol. 26, No. 2, April 1989
Labrador	7400.00	4.57	2	FP	False	863.28	0.826	0.728	Marine Technology, Vol. 26, No. 2, April 1989
Humphrey Gilbert	3200.00	3.51	2	FP	False	392.40	0.782	0.690	Marine Technology, Vol. 26, No. 2, April 1989
John Cabot	6700.00	3.96	2	FP	False	794.61	0.893	0.788	Marine Technology, Vol. 26, No. 2, April 1989
Norman MacLeod	9700.00	3.66	2	FP	False	1098.72	1.017	0.897	Marine Technology, Vol. 26, No. 2, April 1989
Sir John Franklin	10000.00	4.12	2	FP	False	1128.15	0.946	0.834	Marine Technology, Vol. 26, No. 2, April 1989
Geo. R Pearkes	5300.00	3.60	2	FP	False	637.65	0.893	0.788	Marine Technology, Vol. 26, No. 2, April 1989
Edward Cornwallis	5300.00	3.60	2	FP	False	637.65	0.893	0.788	Marine Technology, Vol. 26, No. 2, April 1989
Earl Grey	6500.00	4.00	2	FP	False	765.18	0.872	0.769	Marine Technology, Vol. 26, No. 2, April 1989
West Wind	7500.00	5.18	2	FP	False	873.09	0.761	0.671	Marine Technology, Vol. 26, No. 2, April 1989
Glacier	15800.00	5.50	2	FP	False	1922.76	0.980	0.864	Marine Technology, Vol. 26, No. 2, April 1989
Aleksei Kosygin	24700.00	5.60	2	FP	False	2403.45	0.899	0.792	Marine Technology, Vol. 26, No. 2, April 1989
Escort Eagle	2982.80	1.98	2	FP	True	490.50	1.501	1.018	Ship & Boat Int'l June 1995 and March 1996
Sea Salvor	3360.00	2.30	2	FP	True	539.55	1.380	0.936	Ship & Boat Int'l March 1999
Fennica	15000.00	4.20	2	FP	True	2256.30	1.425	0.967	
0	10439.80	3.80	2	FP	True	1471.50	1.265	0.858	http://www.irvingshipbuilding.com/products/supply.htm
Adulis	2804.00	1.60	2	FP	True	225.63	0.829	0.563	Ship & Boat Int'l September 1995
Atlantic Spruce	2982.80	1.98	2	FP	True	487.56	1.492	1.012	Ship & Boat Int'l April 1995
Bentley and Melton	3600.00	2.38	2	FP	True	589.58	1.408	0.955	Ship & Boat Int'l July/August 1996
Delta Carey	3281.08	2.40	2	FP	True	591.23	1.494	1.013	Ship & Boat Int'l April 1998
Domancay	2155.07	2.05	2	FP	True	392.40	1.457	0.989	Ship & Boat Int'l October 1998
Don Carlos	3131.94	2.35	2	FP	True	510.12	1.348	0.915	Ship & Boat Int'l October 1998
El Oriental	3400.00	2.65	2	FP	True	647.46	1.495	1.014	Ship & Boat Int'l October 1998
Herbert Ballam	969.41	1.60	2	FP	True	176.58	1.318	0.894	Ship & Boat Int'l March 1998
Nana Kobina-Nketsia IV	1986.00	2.05	2	FP	True	323.73	1.270	0.861	Ship & Boat Int'l March 1996
Neeltje P	3430.22	2.60	2	FP	True	588.60	1.369	0.928	Ship & Boat Int'l June 1999

Portgarth	2980.00	2.15	2	FP	True	485.60	1.408	0.955	Ship & Boat Int'l June 1995
Pyrrhos	3300.00	2.30	2	FP	True	524.84	1.359	0.922	Ship & Boat Int'l December 1998
Regain	1200.00	1.65	2	FP	True	171.68	1.089	0.739	Ship & Boat Int'l April 1998
Seabulk Carolyn	3131.94	2.20	2	FP	True	490.50	1.355	0.919	Ship & Boat Int'l October 1998
TUGZ	2982.80	2.28	2	FP	True	481.74	1.342	0.910	Ship & Boat Int'l March 1996
Watergeu	1308.70	1.70	2	FP	True	250.16	1.468	0.996	Ship & Boat Int'l September 1995
Tai Koo	2982.80	2.63	2	CP	False	512.08	1.299	1.031	Ship & Boat Int'l April 1995
Tor Viking	13440.00	4.10	2	CP	False	1981.62	1.368	1.086	
Terry Fox	17700.00	4.80	2	CP	False	1863.90	0.964	0.765	Marine Technology, Vol. 26, No. 2, April 1989
Robert Lemeur	7100.00	3.00	2	CP	True	1049.67	1.366	0.834	Marine Technology, Vol. 26, No. 2, April 1989
Ikaluk	11200.00	3.75	2	CP	True	1618.65	1.340	0.818	Marine Technology, Vol. 26, No. 2, April 1989
Polar Stern	14700.00	4.20	2	CP	True	2050.29	1.313	0.801	Marine Technology, Vol. 26, No. 2, April 1989
Mudyug	9100.00	4.00	2	CP	True	1039.86	0.947	0.578	Marine Technology, Vol. 26, No. 2, April 1989
Asmara	3468.25	2.40	2	CP	True	588.60	1.433	0.875	Ship & Boat Int'l April 1998
Koyo Maru	7457.00	3.50	2	CP	True	1299.83	1.477	0.902	Ship & Boat Int'l December 1998
Normand Neptun	14914.00	4.27	2	CP	True	2158.20	1.353	0.826	Ship & Boat Int'l April 1996
Port Maria	2960.00	2.50	2	CP	True	500.31	1.318	0.804	Ship & Boat Int'l April 1996
Trinity/Seacor	9157.20	3.55	2	CP	True	1373.40	1.348	0.823	http://marinelink.com/aug96/mr0.8305.html
Ulysse	10738.08	3.60	2	CP	True	1667.70	1.459	0.891	Ship & Boat Int'l March 1999
Shirage	22100.00	4.90	3	FP	False	2383.83	1.049	0.808	Marine Technology, Vol. 26, No. 2, April 1989
John A. MacDonald	11000.00	4.10	3	FP	False	1294.92	1.022	0.787	Marine Technology, Vol. 26, No. 2, April 1989
Louis S. St Laurent	17700.00	4.57	3	FP	False	1981.62	1.059	0.816	Marine Technology, Vol. 26, No. 2, April 1989
Lenin	32400.00	5.64	3	FP	True	2972.43	0.923	0.547	Marine Technology, Vol. 26, No. 2, April 1989
Ermak	26500.00	5.40	3	FP	True	2795.85	1.022	0.606	Marine Technology, Vol. 26, No. 2, April 1989
Kotug Rotor Tug	2982.80	1.65	3	FP	True	686.70	2.373	1.406	Ship & Boat Int'l March 1996
RT Innovation	4697.91	2.15	3	FP	True	735.75	1.575	0.933	Ship & Boat Int'l March 1999
Polar Star	44800.00	4.88	3	CP	True	4453.74	1.227	0.655	Marine Technology, Vol. 26, No. 2, April 1989