Ice classed ships

MAN Energy Solutions
Future in the making

Main engines, propulsion & EEDI

Future in the making

Contents

Navigation in the Arctic and in icy waters sets specific requirements for the capabilities of the hull and propulsion plant of the ship. The hull must be reinforced and the propulsion plant must deliver sufficient thrust to advance through these icy waters to transport and deliver goods safely. These requirements for both capabilities of the hull and the propulsion plant have been put into regulation.

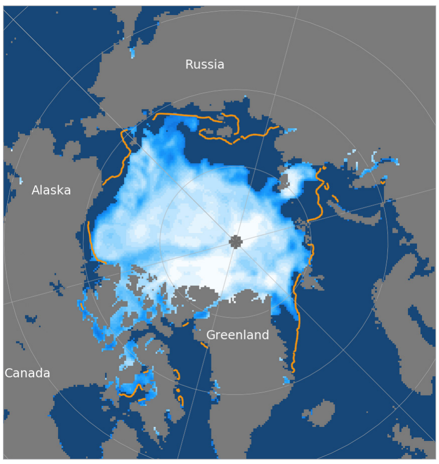
This paper focuses on the challenges and specific aspects to consider for a propulsion plant for an ice-classed ship, especially for the new and stricter EEDI phases.

Introduction

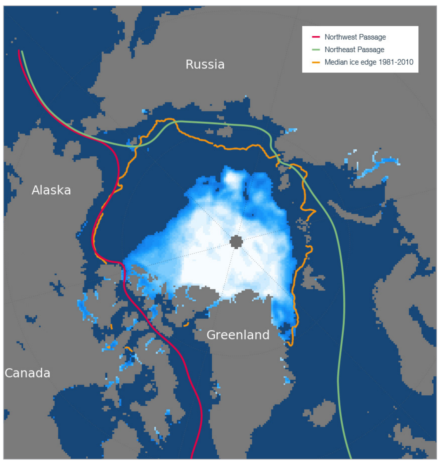
Ships with an ice class for winterisation are built with thicker hulls with stronger girders, beams, and bulkheads. How thick and how strong depends on the different ice class levels, which also specify a minimum power output.

In general, for a normal ship, the installed propulsion power of the main engine needed to obtain the required ship speed in service may be sufficient for propulsion conditions during wintertime.

Sea ice concentration, 3 August 2021



Sea ice concentration, 28 August 2012



(Image courtesy of the National Snow and Ice Data Center, University of Colorado, Boulder)

However, depending on the form and thickness of the ice, and thereby the ice class required, combined with the lowest ambient air temperature appearing during winter operation, some increased capabilities of the main engine may be required. This paper describes these higher demands.

Considerations on how to meet EEDI requirements have led to an increase in the number of enquiries on ice-classed newbuilding's in recent years. The reason is the fact that ice class ships have correction factors that allow a lower EEDI obtained by the ship. To comply with the ice class, the restrictions not only increases the weight of the ship due to the thicker hull, but also, depending on the ice class level selected, the complexity of the propulsion system regarding the propeller and the engine output.

With global warming shrinking the Arctic ice pack, the application of ice-classed ships may intensify as new sea routes open in the future, for example the Northwest and Northeast passages, see Fig. 1, Ref. [1]. The first transits of container ships through the Northeast passage have already been performed on an experimental basis during summer, and the route is regularly used by LNG carriers.

This paper describes ice class requirements for merchant ships only, and not for special ice going ships like icebreakers. Details on hull design, hull thickness, etc., are not included as well. The Finnish-Swedish ice classes have been used as the reference for the description of the ice-classed ships. Ice classes for other classification societies are described as well.

Sea ice concentration, 3 August 2021 | Sea ice concentration, 26 October 2021 | Sea ice concentration, 15 December 2021

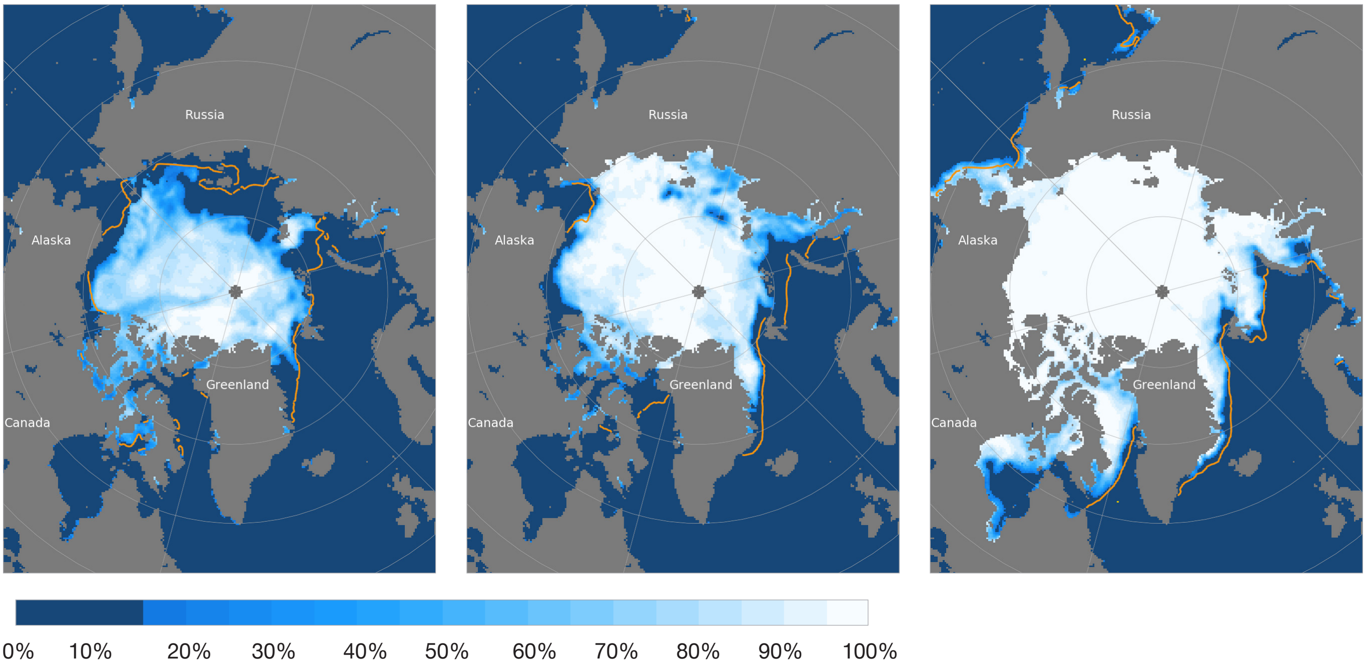


Fig. 1: A view of how the ice develops through the fall. (Image courtesy of the National Snow and Ice Data Center, University of Colorado, Boulder.)

Ice classes and requirements

Ships with an ice class have a strengthened hull to enable them to navigate through sea ice. Most of the higher classes require several forms of rudder and propeller protection, and strengthened propeller tips are often required as well. Depending on the class, sea chests, i.e. the openings in the hull for seawater intake, have to be properly arranged to avoid blocking up with ice.

Finnish-Swedish ice class designation

The Finnish-Swedish ice class rules are divided into four types of ice classes based on the ice strengthening hull design required by ships for navigating in ice, see Ref. [2]. The “Finnish safety transport agency” (Trafi) is responsible for the Finnish-Swedish ice classes.

The design requirements for ice classes are a minimum speed of 5 knots in brash ice channels, i.e. channels of already broken ice, see Table 1.

Table 1: Finnish-Swedish ice classes

Ice class	Brash ice thickness
IA Super	1.0 m
IA	0.8 m
IB	0.6 m
IC	0.4 m
II	Light ice conditions
III	Ice-free open water conditions

The most common ice class notation for ships is the IC. Classes IA, IB, and IC are assumed to rely on ice breaker assistance to keep an advance speed of 5 knots. For IA super ice breaker assistance is normally not required.

It is anticipated that ice ramming does not take place for any of these ice classes. Ramming is if the ship comes to a hold and it is able to move backwards and then ram into the ice again, and in that way advance through the ice. Ice ramming is further described in the section "Ice classes with ramming".

Other ice classes

International classification societies have incorporated the Finnish-Swedish ice class rules to their own rule books and offer equivalent ice class notations recognised by the Finnish-Swedish authorities. However, the ice class designations and requirements applied by the different classification societies may sometimes differ from each other.

For example, the authorities accept DNV ice class Ice-IA as being equivalent to the Finnish-Swedish ice class IA and apply a similar naming for the notations.

Another notation is the Polar ice class as set out in the polar code of the IMO [3]. It defines the capability of the ship for operation in icy and polar waters. These definitions are shown in Table 2.

For the polar ice classes, the ships are assumed to maintain continuous movement through thin ice. Ships in polar classes category B, category A, and ICE IA* may proceed by ramming through the ice. But note that ICE IA*, PC6 and PC7 ships are mostly designed so that they are restricted from intentionally ramming.
Ships in lower ice classes are not assumed to do any ramming at all.

If the icy conditions are too severe, ships incapable of ramming must go astern to proceed backwards and seek an alternative course unless they are under the notation DAV.

DAV, or double-acting vessels, are ships capable of operating in polar waters without any ice breaker assistance. Such ships are double-acting in the manner that they are capable of moving astern in ice.

They are not assumed to do any ramming through the ice when moving astern, but use the propeller to mill through the ice. This paper does not cover this ship type.

More information about DAV can be found in Ref. [4].

Table 2: Polar ice classes by DNV [3]

Ship category	Polar ice Class	Operating capability
A	PC1	Year-round operation in all polar waters
A	PC2	Year-round operation in moderate multi-year ice
A	PC3	Year-round operation in second-year ice, which may include multi-year inclusions
A	PC4	Year-round operation in thick first-year ice, which may include old ice inclusions
A	PC5	Year-round operation in medium first-year ice, which may include old ice inclusions
B	PC6	Summer/autumn operation in medium first-year ice, which may include old ice inclusions
B	PC7	Summer/autumn operation in thin first-year ice, which may include old ice inclusions
C	ICE IA* /E4	First year ice to 1.0 m
C	ICE IA /E3	First year ice to 0.8 m
C	ICE IB /E2	First year ice to 0.6 m
C	ICE IC /E1	First year ice to 0.4 m
C	ICE C	Light ice conditions
C	none	Ice free/ open water conditions

Another ice class notation is from the Russian Maritime Register, see Table 3.

For all of these notations, the engine output has to be evaluated. This is because the Russian ice class system has different requirements for engine output compared to the Finnish-Swedish ice class.

For example, the Russian Maritime Register of Shipping describes arc5 in Table 3 as equivalent to IA Super in Table 1, but with the restriction that the ship must meet the specified minimum engine output requirement for the Finnish-Swedish ice class to comply with the IA Super.

Ships operating in very tough arctic ice conditions with operation in multi-year ice may also be constructed to meet the ice classes of the Russian Maritime Register of Shipping. For example, when sailing to the new/coming oil and gas areas in the arctic region in northwest Siberia. As one can see in Table 1, this corresponds to the bottom five rows of Table 3. Moving above Arc5, you can see that the ships must be able to navigate through brash ice, which is slightly thicker. Even though Arc6 and Arc7 have stricter requirements in regard of the ice thickness, it is not necessarily equivalent to being IA Super as mentioned earlier. This is due to the difference in engine output requirements set by the Finnish-Swedish ice class and the Russian Maritime Register. Therefore, the compatibility depends on whether the ship meets the engine output requirements for the Finnish-Swedish ice class. which means that Arc5, Arc6, and Arc7 all are comparable to IA Super.

For comparison of the ice classes in Tables 1, 2, and 3, Table 4 shows how these correspond to other classification societies' notations of the ice class, equivalent to the Finnish-Swedish ice classes and polar classes.

Table 4 shows how the different ice classes are compatible. Arc6 and Arc7 from the Russian Maritime Register are not mentioned in this table even though they could be directly compatible. This is because they actually belong to a thicker ice class, but note that they should be considered. The same applies for PC6 and PC7 in the polar ice class.

Table 3: Russian Maritime Register of Shipping ice classifications [5]

Arc9/LU9	In summer/autumn navigation – voyage in all areas of the ocean. In winter/spring navigation in Arctic – voyage in very close floating ice and in compact multi-year ice of up to 3.5-m thickness and in freezing non-arctic seas without restrictions
Arc8/LU8	In summer/autumn navigation – voyage in all areas of the ocean. In winter/spring navigation in Arctic – voyage in close floating second-year ice up to 2.1-m thickness and in freezing non-arctic seas without restrictions
Arc7/LU7	In summer/autumn navigation – voyage in all areas of the ocean. In winter/spring navigation in Arctic – voyage in close floating first-year ice up to 1.4-m thickness and in freezing non-arctic seas without restrictions
Arc6/LU6	In summer/autumn navigation in Arctic – voyage in open floating first-year ice up to 1.3-m thickness. In winter/spring navigation in Arctic – voyage in open floating first-year ice up to 1.1-m thickness. Year-round voyage in freezing non-arctic seas
Arc5/LU5	In summer/autumn navigation in Arctic – voyage in open floating first-year ice up to 1.0-m thickness. In winter/spring navigation in Arctic – voyage in open floating first-year ice up to 0.8-m thickness. Year-round voyage in freezing non-arctic seas
Arc4/LU4	In summer/autumn navigation in Arctic – voyage in open floating first-year ice up to 0.8-m thickness. In winter/spring navigation in Arctic – voyage in open floating first-year ice up to 0.6-m thickness. Year-round voyage in freezing non-arctic seas in light ice conditions
Ice3/LU3	Regular voyage in open floating ice cake ice of non-arctic seas up to 0.7-m thickness
Ice2/LU2	Regular voyage in open floating ice cake ice of non-arctic seas up to 0.5-m thickness
Ice1/LU1	Episodical voyage in open floating ice cake ice of non-arctic seas up to 0.4-m thickness

Propulsion power requirements

About propulsive machinery power, a few requirements are set up to make sure that the ship can navigate through the icy waters and the resulting added resistance. To be able to comply with the requirements, a few standards have been set up. Some of the examples

Table 4: Finnish ice classes equivalent to class notations of recognised classification societies and the determination of t

Finnish – Swedish	ABS	Bureau Veritas	DNV GL
IA Super	**ICE Class I AA**	**ICE CLASS IA Super**	**ICE(1A*F) / ICE(1A*)**
IA	**ICE Class I A**	**ICE CLASS IA**	**ICE(1A)**
IB	**ICE Class I B**	**ICE CLASS IB**	**ICE(1B)**
IC	**ICE Class I C**	**ICE CLASS IC**	**ICE(1C)**
II			
III			

[1] evaluation of engine output required.

describing how to achieve compliance are given by DNV and The Finnish Transport Agency [2, 7].

In addition to machinery and engine power, there are requirements for the propeller in terms of thickness, radius, chord length, and span of blades to be certified for navigation in the icy waters. These will shortly be mentioned in this paper, but can be found with further details in, for example, DNV regulations for cold climate [4].

Finnish transport agency (TRAFI)

In this section, the rules for what is required to comply with the different ice classes are given from the Finnish transport agency. This paper only states the requirements for the propulsive system, but for further details see Ref. [2].

New ships

For new ships to qualify for ice classes IC, IB, IA and Super IA, two calculations of minimum power output are needed at different draughts to estimate the required minimum power output of the engine. One for the maximum allowable draught (UIWL) amidships, and one for the minimum allowable draught (LIWL). The calculation in equation (1) gives the minimum power output for an ice-classed ship to navigate at 5 knots depending on the category of ice-class. For both calculations, the parameter values used, such as length and beam, must be taken at the maximum allowable draught.

EQ. (1)

$$P_{min} = \frac{K_e\left(\frac{R_{CH}}{1000}\right)^{\frac{3}{2}}}{D_p}\ kW$$

In equation (1), D_p is the propeller diameter. As the propeller diameter is part of the denominator, it is recommended to consider how and if the diameter can be increased, as it would lower the minimum propulsion power for any ice class. The reason is that the propeller efficiency increases with an increasing diameter. However, it is important to make sure that the top of the propeller is not too close to the water surface due to the brash ice.

K_e depends on the propeller configuration and the number of propellers, see Table 5.
It is clear from Table 5 that the minimum power required for an ice class can be lowered by 10%, simply by changing from an fixed pitch, FP to a controlable pitch, CP propeller. CP propellers also have other advantages for operation in ice, which will be discussed later.

When calculating the resistance R_{CH}, the minimum power output has to be the larger of the two calculations for P_{min} using (UIWL) and (LIWL) when calculating the resistance.

R_{CH} is the resistance in newton for a ship sailing in brash ice and consolidated surface layer. There are two ways to determine R_{CH}. One is by analytical formulas, where the thickness of the ice along with the length, beam, draught, and the shape of the ship front is used to estimate the resistance. For specified calculations of this, see Ref. [2]

Another way is by model testing. Sometimes this can be accepted, but the test must demonstrate knowledge about navigation in ice. This can also allow for a calculation of K_e by model testing instead of using values from Table 5.

DNV

DNV follows the rules from the Finnish-Swedish ice class, but has set up some formulas for evaluation.
In the DNV regulation about cold climate, there are a few statements on

Table 5: K_e values for different propeller configurations [2].

Number of propellers	CP propeller, electric or hydraulic propulsion machinery	FP propeller
One propeller	2.03	2.26
Two propellers	1.44	1.60
Three propellers	1.18	1.31

he ice classes of ships [6]

Korean Register of Shipping	Lloyds Register	Russian Maritime Register of Shipping	IMO POLAR Class
IA super	100 A1 ICE Class 1AS	Arc5/LU5[1]	PC7[1] - ICE IA* /E4
IA	100 A1 ICE Class 1A	Arc4/LU4[1]	PC6[1] - ICE IA /E3
IB	100 A1 ICE Class 1B	Ice3/LU3[1]	ICE IB /E2
IC	100 A1 ICE Class 1C	Ice2/LU2[1]	ICE IC /E1
ID	100 A1 ICE Class 1D	Ice1/LU1[1]	ICE C
			None

the minimum continuous output for the main engine. In general, the minimum power output for ice class C is not to be less than the result from equation 2. The power output is in kW, and inputs for length, L, and breadth, B, are given in metres. Note that this is only valid for ice class C, and it is often a slight underestimation of the actual calculations for the Finnish-Swedish ice class, but it is simple way to get a fast estimate.

EQ. (2)

$$P_s = 0.73 \times L \times B$$

If a ship is designed with a special bow for navigation in ice, less power is required, and equation 3 is valid for minimum propulsion power for ice class C.

EQ. (3)

$$P_s = 0.59 \times L \times B$$

If the ship is fitted with a controllable pitch propeller (CP), the output can be reduced by 25% according to DNV.

Propeller configuration

As the ship progresses through ice, it is inevitable that the propeller will be exposed to broken ice that must be “cut” through to advance. Therefore, there are design requirements regarding geometry and thickness of the blades.

Propeller clearance and transom stern

The propeller clearance, the stern frame clearance, and the ice clearance must be sufficient to avoid clashes between propeller and ice. Definitions of these can be seen in fig.2. Otherwise the incoming ice will cause problems, especially when the ship moves astern. If there is a significant risk of the propeller hitting large ice floes, a propeller clearance of at least 0.5 m should be accounted for to avoid high loads on the propeller blades. This is illustrated in Fig. 2 [7].

The transom stern should be narrow and not exceed below the UIWL as the transom stern will crush the ice against it while going astern, thereby increasing the resistance significantly if the transom stern exceeded into the water.

The capability of going astern is mostly relevant for higher ice class ships or DAV. If it is unavoidable that the transom stern exceeds the UIWL, extra strengthening of the stern is necessary as the ice adds a massive pressure on the hull when the ship moves astern.

Thickness of the propeller

DNV has given some guidelines on how to calculate the thickness of the propeller for ice-classed ships. Most likely, the propeller needs to be thicker than a regular propeller as it is expected to cut through ice.

Restrictions on calculations for the thickness at $t_{0.25}$, $t_{0.35}$ and $t_{0.6}$ are therefore given, where the relations in Table 6 are valid.

$t_{0.25}$	=	thickness at 0.25 R
$t_{0.35}$	=	thickness at 0.35 R
$t_{0.60}$	=	thickness at 0.60 R

Table 6: Thickness parameters defined for three blade positions

Material and geometry factors are included in the calculation. Further information on how to calculate these can be found in Ref. [4].

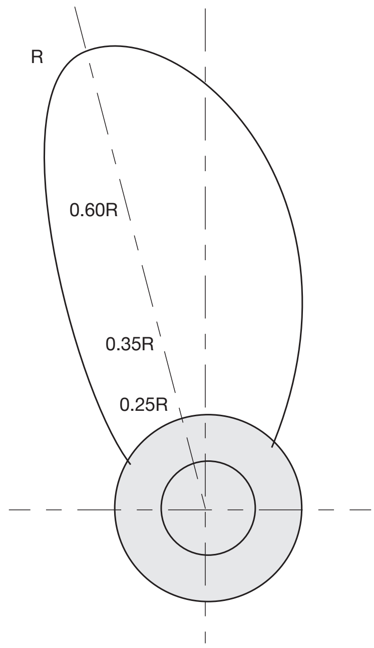


Fig 3 propeller blade including length for where the thickness requirements are based.

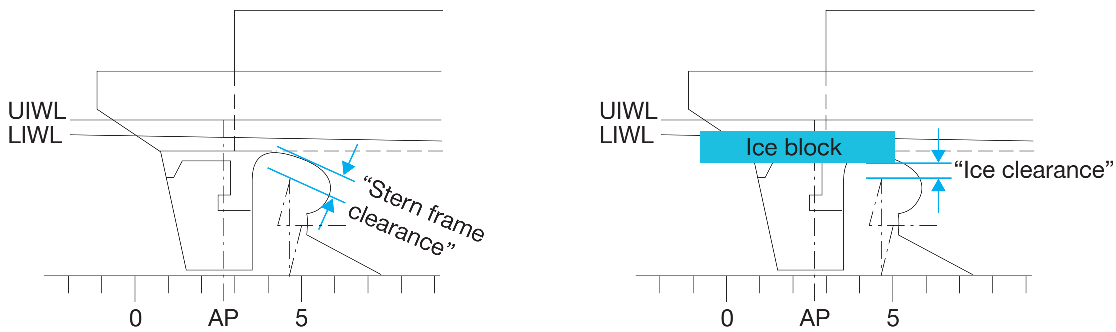


Fig. 2: The clearance between the stern frame and the propeller (left), and the ice sheet and the propeller when the ship is at LIWL (right) [7]

Propulsion advantages with CP propeller

The simplicity and high efficiency of FP propellers makes them standard in merchant ships, which also apply for low ice classes. The propellers are cast in one block, and the position of the blades, and hence the propeller pitch, is fixed and cannot be changed after installation in the ship. This means that when operating in, for example, heavy weather and/or ice, the propeller performance curve will be very heavy (reduced speed for the same power, i.e. demanding a higher torque).

Compared to the FP propeller, the position of the blades for the CP propeller, and thereby the propeller pitch, can be controlled to avoid heavy running and overloading of the main engine. Therefore, it can be advantageous to install CP propellers on ships that see a lot of manoeuvring, or where it is feasible to avoid a high lightrunning margin. This even applies to ships not sailing in ice but which instead often encounter harsh weather, slow steaming or high load running. The CP propeller is especially relevant for both moderate ice classes and for very strong ice classes.

Ships designed for high ice classes with ice ramming are typically designed with CP propellers due to the advantages of frequent changes between going ahead and astern.

For calculations of minimum propulsive power for ice class ships, the K_e factor is mainly defined by the choice of propeller . When deciding between CP and FP propellers, the minimum power for an ice class ship could be lowered by 10% simply by choosing a CP propeller. This is due to the calculation in equation (1), and table values for K_e in Table 5.

Since most ships are for commercial use without any need for ramming, then a simple propulsion system consisting of a single low speed two-stroke main engine directly coupled to a ducted CP propeller would be an extremely reliable and cost-efficient propulsion system for high ice classes.

EEDI restrictions

The EEDI guideline is a mandatory instrument adopted by IMO to ensure compliance with international restrictions on CO_2 emissions from new ships. The EEDI represents the amount in grams of CO_2 emitted when transporting one deadweight tonnage of cargo for one nautical mile. Equation 4 is the EEDI calculation in simplified form.

EQ. (4)

$$EEDI \approx \frac{CO_2}{Transport\ work} \approx \frac{P_{ME} \times C_F \times SFC}{capacity \times V_{ref}}$$

The EEDI calculation is based on cargo capacity, propulsion power, ship speed, specific fuel consumption, and fuel type. However, by installing, for example, waste heat recovery systems (WHRS) or making the ship compliant with an ice class, certain correction factors apply, and reductions of EEDI are obtainable.

A reference index calculation for a specific ship type is based on data from ships built in the period from 2000 to 2010. The required EEDI value for new ships is reduced in three phases. As an example, tankers built after 2025, have a mandatory EEDI reduction of 30% (phase 3) compared to the reference value. For an example regarding the EEDI as a function of dwt, Fig. 4 shows the three phases for tankers. Bulk carriers have the same EEDI restrictions, whereas container ships may have up to 50% EEDI reductions as of April 2022 depending on the dwt carried.

Further phases have not yet been implemented, as the discussion on a potential EEDI phase 4 is ongoing.

For all cargo ships, except container ships, the reference, and the actual EEDI values calculated are based on 100% utilisation of capacity (in dwt). The reference speed must be consistent with this loading of the ship, at 75% of SMCR, and with the hull in a condition as on sea trial. The calculated actual EEDI may not exceed the required EEDI.

The reduction depends on ship type, where phase and size should also be taken into account. Phase 2 is in force from 1 January 2020 for all ship types. For more information on reduction factors for all ship types and further investigation of restrictions, see Ref. [9].

There are several methods for lowering the EEDI, but this section focuses narrowly on the reductions caused by implementing an ice class due to the correction factors for ice-classed ships. This is followed by an evaluation of the dilemma between minimum propulsive power output for an ice-classed ship and maximum allowed SMCR to comply with the EEDI. These may intersect for the different ice classes to be compatible for both the EEDI and the minimum power required for the specific ice classes.

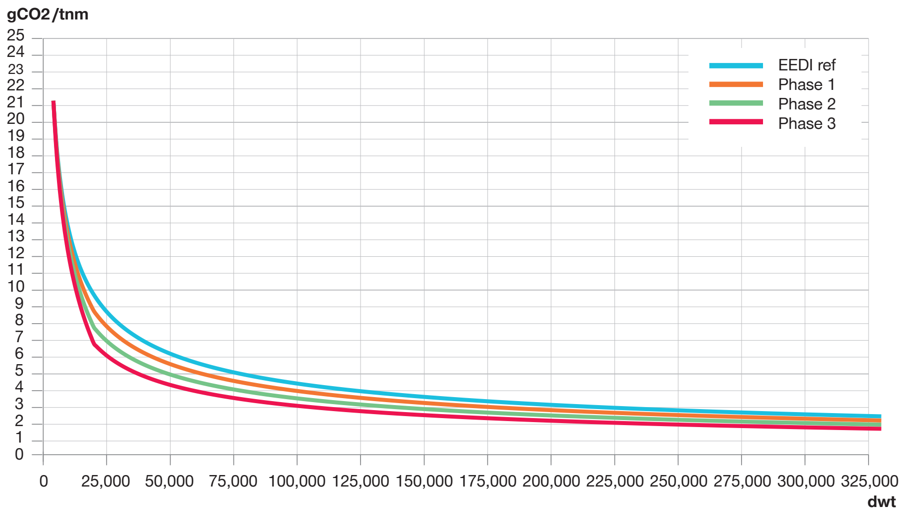


Fig. 4: EEDI reference line and requirements for tankers [8]

Correction factors for ice-classed ships

Ice-classed ships have correction factors because they must meet requirements set by the classification societies to be able to navigate in icy waters.

Therefore, to maintain compliance with EEDI regulations even with the extra power or capacity, the EEDI regulations include three correction factors for ice-classed ships. For all categories of ice-classed ships, these are the capacity correction factor (f_i) and the power correction factor(f_j). For the ice classes IA and IA Super, an additional third factor is added to the equation, the so-called (f_m), which corrects for the higher power addition of the higher ice classes [10].

The implementation of these correction factors in the EEDI equation is given in equation 5.

EQ. (5)

$$EEDI = \frac{f_j \cdot (P_{ME} \cdot CF_{ME} \cdot SFC_{ME}) + (P_{AE} \cdot CF_{AE} \cdot SFC_{AE})}{f_i \cdot Capacity \cdot V_{ref} \cdot f_m}$$

- P_{ME} is the propulsion power of the main engine at 75%
- P_{AE} is the auxiliary engine power
- C_F is the fuel-specific carbon conversion factor for CO_2 emissions
- SFC is the engine and fuel dependent specific fuel consumption
- Capacity depends on the deadweight
- V_{ref} is the ship speed.

It must be noted that additional correction factors may apply for individual ship types such as chemical tankers, etc., but these will not be evaluated in this paper.

Capacity correction factor

The capacity correction factor (f_i) is calculated by using the two contributions in equation 6.

EQ. (6)

$$f_i = f_{i(ice\ class)} f_{i,CB}$$

The first contribution, $f_{i(ice\ class)}$, is the capacity correction factor for the ice strengthening of the ship. As the hull thickness will be larger, the deadweight must be reduced due to the added weight for strengthening of the hull. f_i is calculated by using one of the equations in Table 7.

Table 7: Calculation of the capacity correction factor

Ice class	fi(ice class)
IC	**fi(IC) = 1.0041+58.5/DWT**
IB	**fi(IB) = 1.0067+62.7/DWT**
IA	**fi(IA) = 1.0099+95.1/DWT**
IA Super	**fi(IAS) = 1.0151+228.7/DWT**

However, the EEDI guideline offers an alternative way to calculate the correction factor $f_{i(ice\ class)}$. The alternative correction factor in equation 7 is named (f_{ivse}) and it can substitute $f_{i(ice\ class)}$. It can be used if the ship uses another ice class notation than the Finnish-Swedish ice classes.

EQ. (7)

$$f_{iVSE} = \frac{DWT_{refference\ design}}{DWT_{enhanced\ design}}$$

Where:

EQ. (8.1)

$$DWT_{enhanced\ design} = \Delta_{ship} - LWT_{enhanced\ design}$$

EQ. (8.2)

$$DWT_{reffernce\ design} = \Delta_{ship} - LWT_{reffence\ design}$$

Δ_{ship} is the displacement of the ship, and the same value is used for equation 8.1 and 8.2. There is also the restriction, that changes in alloy or in the material grade are not allowed between the calculations for enhanced design and reference design.

Both calculations give estimates of how much extra weight the added steel for ice strengthening implies for the ship dwt. These factors are never to be taken less than 1.0, since it accounts for the ice strengthening contribution towards lowering the EEDI. The second factor, $f_{i,\ CB}$ is the capacity correction factor for improved ice going capability, as the fullness of the hull is normally lowered for ice going ships. As previous, this factor is never taken to be less than 1.0.

Use equation 9 to calculate this factor.

EQ. (9)

$$f_{i,C_B} = \frac{C_{B\ reference\ design}}{C_B}$$

Here, C_B is the actual block coefcient of the ship, and C_B reference design is from Table 8, depending on the ship type and size.

For all other ships types than the three mentioned in Table 8, the value $f_{i,CB}$=1.0 must be used.

This factor was implemented to make up for differences in block coefficients as the front is not as voluminous as on a normal tanker, bulk carrier, or general cargo ship. Ice-classed ships must be able to cut through the ice easily when sailing in icy waters and, therefore, have a more streamlined shape at the bow. Note that for Finnish-Swedish ice classes, this is valid only for brash ice or open water ice.

Power correction factor

This factor cannot be larger than f_{jmax}=1.0. This is to be taken as the greater value of either of two calculations of (f_{jo}) and (f_{jmin}). These calculations are shown in Table 9.

Table 8: Reference block coefficients for calculation of the correction factor

	Size category				
Ship type	**Below 10,000 dwt**	**10,000–25,000 dwt**	**25,000–55,000 dwt**	**55,000–75,000 dwt**	**Above**
Bulk carrier	0.78	0.80	0.82	0.86	0.86
Tanker	0.78	0.78	0.80	0.83	0.83
General cargo			0.80		

Table 9: Correction factors for power output calculations

		f_{jmin} depending on the ice class			
Ship type	f_{jo}	**IA Super**	**IA**	**IB**	**IC**
Tanker	$\frac{17.444 \cdot DWT^{0,5766}}{\sum_{i=1}^{nME} MCR_{ME(i)}}$	$0.2488 \cdot DWT^{0.0903}$	$0.4541 \cdot DWT^{0.0524}$	$0.7783 \cdot DWT^{0.0145}$	$0.8741 \cdot DWT^{0.0079}$
Bulk carrier	$\frac{17.207 \cdot DWT^{0,5705}}{\sum_{i=1}^{nME} MCR_{ME(i)}}$	$0.2515 \cdot DWT^{0.0851}$	$0.3918 \cdot DWT^{0.09556}$	$0.8075 \cdot DWT^{0.0071}$	$0.8573 \cdot DWT^{0.0087}$
General cargo	$\frac{1.974 \cdot DWT^{0,7987}}{\sum_{i=1}^{nME} MCR_{ME(i)}}$	$0.1381 \cdot DWT^{0.1435}$	$0.1574 \cdot DWT^{0.144}$	$0.3256 \cdot DWT^{0.0922}$	$0.4966 \cdot DWT^{0.0583}$
Refrigerated cargo ship	$\frac{5.598 \cdot DWT^{0,696}}{\sum_{i=1}^{nME} MCR_{ME(i)}}$	$0.5254 \cdot DWT^{0.0357}$	$0.6325 \cdot DWT^{0.0278}$	$0.7670 \cdot DWT^{0.0159}$	$0.8918 \cdot DWT^{0.0079}$

An alternative to this calculation exists if the ice-classed ship design is based on a design of an open water ship with same shapes and sizes. Then it would be possible to calculate the correction factor by using the power of the current open water ship (P_{OW}) and the new ice-classed ship ($P_{ice\ class}$) in compliance with the power requirements.

Correction factor for ice-classed ships – IA Super and IA

The last factor corrects for the higher ice class. The factor is noted as (f_m) and will be either 1.05 or 1.00 depending on the ice class. If the ship is classed IA or IA Super the factor is f_m =1.05, and if the ice class is either IC or IB then f_m =1.0 [11].

For further information on the calculation of EEDI and other environmental regulations, see Chapter 4 of the paper: Basic principles of ship propulsion [12].

Minimum propulsion power for ice-classed ships

Existing ships with conventional main engines

In regards of EEDI, the engine power is the limiting factor. Usually, the EEDI is lowered by reducing the speed of the ship, as the power P is proportional to the speed by an exponent of 3–4, $P \propto V^{3\ \text{to}\ 4}$. This means that when the design speed is reduced by 4%, the propulsive power will be lowered by approximately 12–14% depending on the ship. This will lower the EEDI, as the power is in the numerator and the speed is in the denominator, as equation (4) describes. If the ship speed is reduced by 4%, the EEDI is expected to decrease by around 10%, depending on the relation between power and speed, corresponding to the reduction between phase 2 and 3.

As there is a minimum requirement to the engine output of ice-classed ships, this needs to be considered along with the EEDI. For both tankers and bulk carriers, the statistical data for SMCR is taken directly from the current fleet before complying with EEDI phases. This data is then used to calculate the requirements to achieve compliance with phases 2 and 3.

Along with the minimum power requirements for the various ice classes, this is illustrated in Fig. 5 (for bulk carriers) and in Fig. 6 (for tankers). In both cases, the calculations of EEDI requirements are from calculations for regular bulk or tankers without ice class notations. The ice-classed minimum propulsion power is calculated by assuming that an FP propeller is installed on the different ship sizes. The dimensions used can be found in the papers: Propulsion trends for bulk [13] and Propulsion trends for tankers [8].

The statistical data in Figs. 5 and 6 shows that the ships do not necessarily comply with the EEDI. But the data also shows that, with no

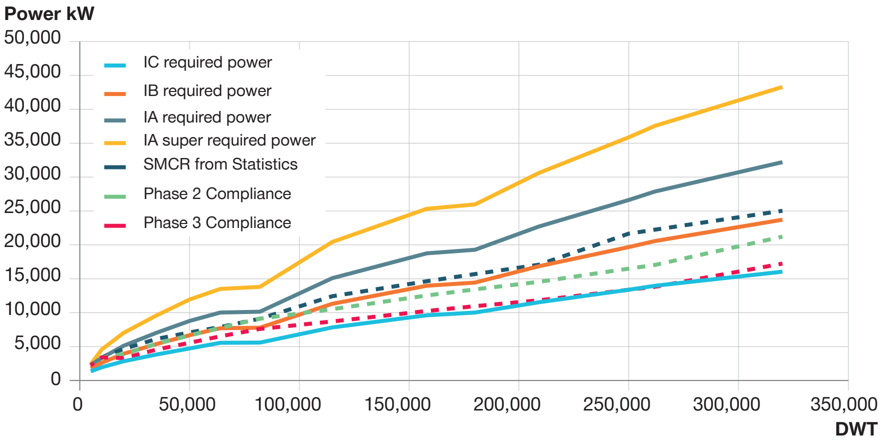


Fig. 5: Required power for bulk carriers in the different ice classes, depending on ship class and statistical data for ships in operation. Phase 2, and phase 3 compliant ships without ice class.

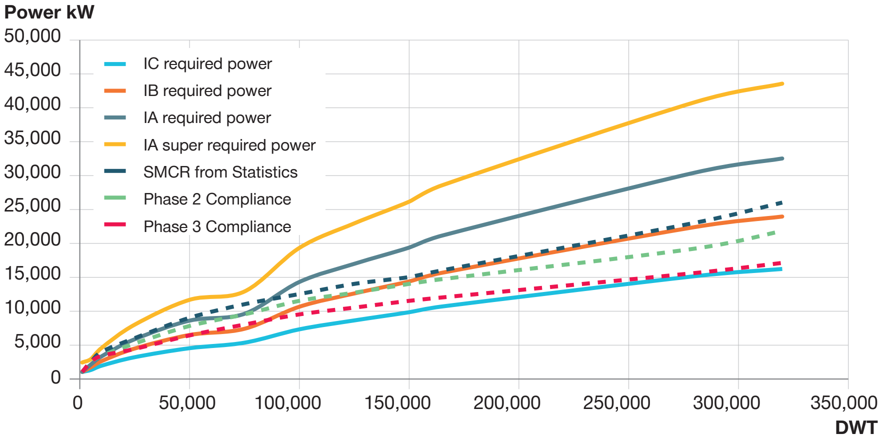


Fig. 6: Required power for tankers in the different ice classes depending on ship class and statistical data for ships in operation. Phase 2, and phase 3 compliant ships without ice class.

addition to engine power, the SMCR design point values of the current fleet would meet the minimum power requirements for ice class IB. It seems that tankers up to 80,000 dwt with respect to propulsion power would actually comply with IA as well. This does not apply for the bulk carriers. The SMCRs of these vessels are well below the IA requirements for minimum power for ice-classed ships. Moreover, there seems to be a plateau for ships in the range between 50,000 dwt and 80,000 dwt for the minimum power for the different ice classes. This is due to the restrictions set by the Panama canal, as the width is the limiting factor of the canal.

Looking further into the figures, the requirement of lowering the engine power really is easy to see when looking at the requirements for phase 2 and 3.

Further to this, it is worth considering a ship design that is compliant with an ice class, as the current engine output would already fit the minimum output requirements from an ice class and the EEDI requirements. The reduction factors implied by the EEDI would then make it possible to add some extra capacity and engine power and, thereby, increase the speed. The ship would still fit within the limits of these boundaries. This would come with an extra cost due to the added thickness of the propeller blades and hull to satisfy the requirements for ice operation, but could end up being cost beneficial.

As can be seen in Figs. 5 and 6, a higher engine power output is required for the higher ice classes. The difference in power requirement and the resulting load diagram of the main engine is shown in Fig. 7, relative to an EEDI phase 3 compliant Kamsarmax bulk carrier.

All plants have been designed with a light running margin of 9%, and the same absolute SMCR speed. The x's are the resulting operational points as calculated for the added resistance as in ice conditions specified for the ice class, when sailing ahead at a speed of 5 knots, as determined by equation (1).

Fig. 7 illustrates the massive increase of torque required by the propulsion plant to move through increasingly thick layers of brash ice, or even solid ice as for IA Super. Besides the power requirement, Fig. 7 illustrates the importance of also considering the light running margin for ice-classed ships equipped with an FP propeller. If the light running margin is not sufficient, the operational point when moving though ice will fall to the left of the engine load diagram, i.e. beyond the capability of the main engine. The propeller will be so heavy running that the engine cannot attain full speed and thereby not full power. With a CP propeller, the pitch can be reduced to ensure that the engine can be fully loaded.

The absolute same engine speed has been applied for the calculations. For this example, the same engine bore is used, but the number of cylinders needed for the required SMCR spans from 5 to 8 cylinders for the various configurations in Fig. 7.

For the higher ice classes, problems with passing the barred speed range might be a problem when the ship sails in open water at lower speeds, and thereby a lower rpm. For this reason, it is recommended to lower the barred speed range (BSR) as much as possible to avoid the risk of damaging the shaft when operating in these areas. This applies especially to engines with a low number of cylinders.

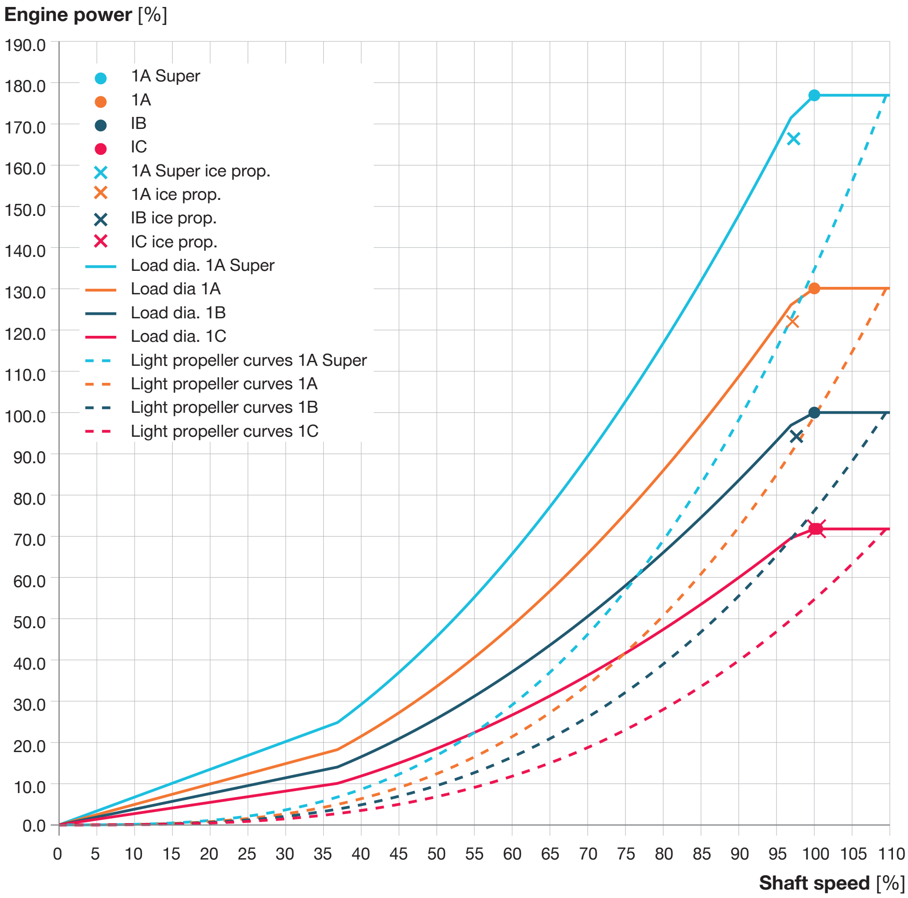


Fig. 7: Engine load diagram comparison for different ice classes of ships.

The BSR is illustrated in Fig. 8, which shows an example of the BSR a bit high. Usually, the BSR is in the span between 45-60% rpm.

Extended main engine load diagram

As described earlier, applying a controllable pitch propeller may be an advantage for higher ice-classed ships.

However, because of the high efficiency and simplicity, a fixed pitch propeller (FP propeller) may often be preferred for lower ice-classed ships.

When a ship with a fixed pitch propeller is operating in normal sea service, it will generally be operating around the design propeller curve 6, as shown in the standard load diagram in Fig. 9.

Sometimes, when operating in heavy weather, the fixed pitch propeller performance will be more towards heavy running, that is, for equal power absorption of the propeller, the propeller speed will be lower and the propeller curve will move to the left.

As the two-stroke main engines are directly coupled to the propeller, the engine has to follow the propeller performance, i.e. also in heavy running propeller situations. For this type of operation, there is normally enough margin in the load area between line 6 and the normal torque/speed limitation line 4, see Fig. 9. To the left of line 4 in torque-rich operation, the engine will lack air from the turbocharger to the combustion process, i.e. the heat load limits may be exceeded and bearing loads might also become too high.

Measurements show that the propeller curve at bollard pull (zero ship speed) will be approximately 15–20% heavy running, but it depends on the propeller arrangement and ship type. This indicates the maximum size of heavy running operation when sailing in thick ice involving a very high torque on the propeller.

With an FP propeller and no ice ramming for ships with special operating conditions, like occasional

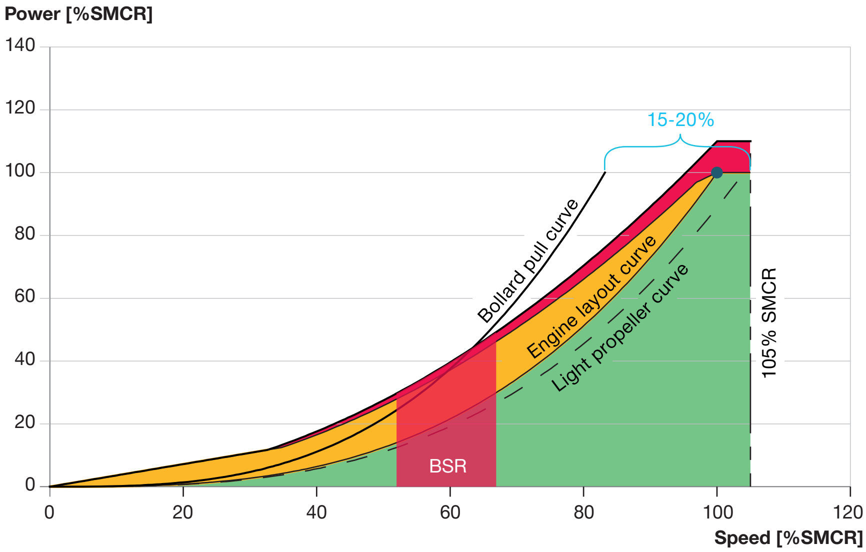


Fig. 8: Bollard pull curve. The two-stroke engine can always accelerate the propeller to about 50% rpm quickly. The BSR in the figure is placed high up in the rpm range, and BSR passage may not be quick [12].

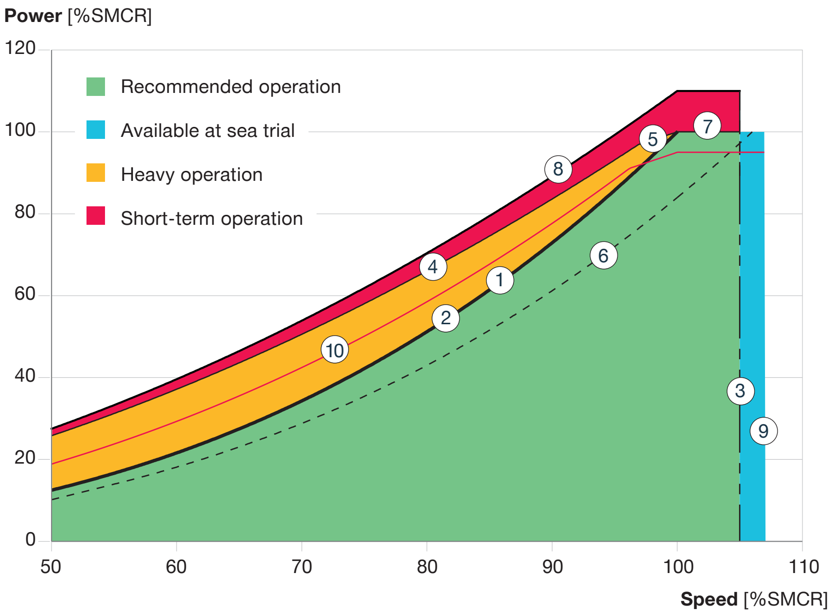


Fig. 9: Engine load diagram with the engine layout diagram applied. The numbers represent the different lines in the diagram. The load diagram is independent of the SMCR point. For further notes, see chapter 3 of the paper: basic principles for ship propulsion.

operation in thick ice, it would be an advantage during normal operating conditions to operate the propeller/ main engine close to line 6. But in ice situations with heavy running propeller, it should be inside the torque/speed limit, line 4.

For ships occasionally operating in heavy ice, an increase of the operating speed range between line 6 and line 4 of the standard load diagram may be carried out, as shown in Fig. 10 for the extended load diagram for a speed-derated engine with increased light running.

The maximum speed limit (line 3) of the engines is normally 105% of the SMCR speed, as shown in Fig. 9.

However, for speed and, thereby, power derated engines, it is possible to extend the maximum speed limit to 105% of the engine's nominal speed, as Fig. 10 shows. For this example, the load diagram can be extended up to 122.5% rpm of the design point for the derated engine, but only if permitted by the torsional vibration conditions. The design point of LP would correspond to 105% of the maximum rpm for, in this case, a 7S50ME-C engine as shown in the figure. Thus, the shafting must be approved regarding torsional vibrations by the relevant classification society, based on the extended maximum speed limit.

When choosing to apply an increased light running margin for the design of the propeller, the load diagram area may be extended from 105% to 122.5%, as shown in Fig. 10. The propeller/main engine operating curve shown in Fig. 10 may have a correspondingly increased heavy running margin before exceeding the torque/speed limit. This is a very normal procedure for ships sailing in ice or heavy weather conditions.

A corresponding reduction of the propeller efficiency may be the result, due to the higher propeller design speed used.

CP propeller and ice ramming

When a ship fitted with a CP propeller operates under ice ramming conditions, the running point on the combinatory curve of the CP propeller (could be on the dotted line of Fig. 10) will suddenly change and move to the left in the load diagram because of the ice ramming. The reason is the reaction time when changing the CP propeller pitch.

For such running conditions, the extended load diagram shown in Fig. 10 may also be useful for the main engine operation. More information can be found in the paper: Basic principles of ship propulsion [12]

Ice classes with ramming

In short, the ramming procedure consists of sailing with a specified speed through the ice until the ice resistance stops the ship as specified in the POLAR code. The ramming on ice may involve occasional high torque on the propulsion system and, therefore, the diesel-electric system with CP propeller may traditionally be preferred, as the electric motor is suitable for high torque deviations, see Fig. 12.

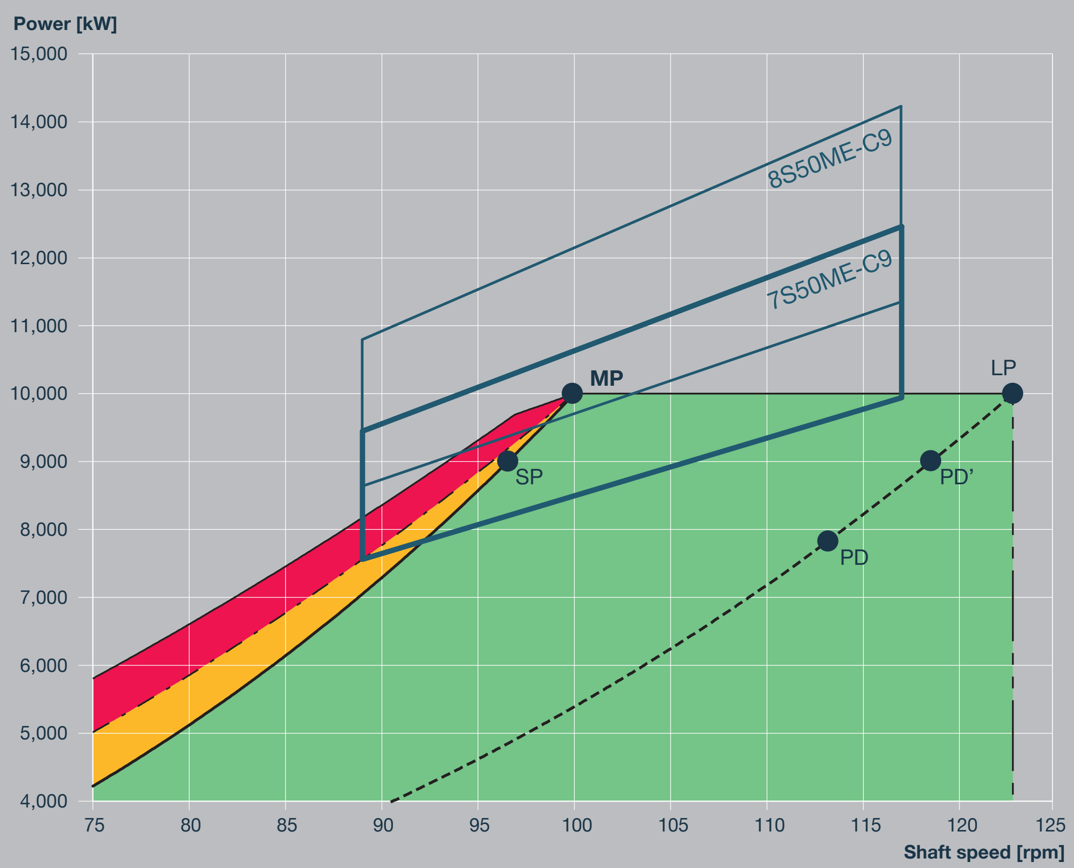


Fig. 10: RPM extended load diagram for an extreme case [12]

However, such a propulsion system has a lower efficiency compared to a propulsion system with a CP propeller directly coupled to an MAN B&W two-stroke engine as seen in Fig. 11. Therefore, as the major part of the ship operation is often in normal sea service without ice, the two-stroke alternative to the conventional diesel-electric propulsion system might be preferred.

During ice ramming, the ship goes astern to come free of the packed ice, and then full ahead into the ice, to break through the ice until the ship is stopped by the ice resistance. The procedure is used for thick ice and ice ridges, which put some very unusual demands on the main engine.

To be able to go ahead and astern within a short time cycle, the ship is equipped with a CP propeller. Then it is not necessary to reverse the main engine, which can be a time-consuming task, only the propeller pitch is reversed. The CP propeller is furthermore enclosed in a nozzle, both for protection of the propeller against blocks of ice and for increased thrust while running heavy.

While the ship is used for ramming ice, the load of the main engine will cycle up and down. The engine load will be high when breaking the ice, and low during pitch reversal of the propeller. The engine load will also be high when the ship goes astern and ahead to ram the ice. The engine, therefore, has an extended load diagram to allow for large torque variations.

The procedure for ramming the ice may be repeated up to 10 times per hour, which increases the demands on turbochargers and auxiliary blowers. Because the load will drop below 25% SMCR every time the sailing direction of the ship is changed, the auxiliary blowers will start up to make a sufficient scavenge air pressure. The auxiliary blowers must be designed to cope with two starts per ramming procedure.

When ramming the ice with some speed, the load of the main engine increases rapidly. This results in a decrease in engine rotational speed. To make sure that the engine will not stop, it is equipped with a wide extended load diagram, and the CP propeller control will reduce the pitch to avoid engine overload.

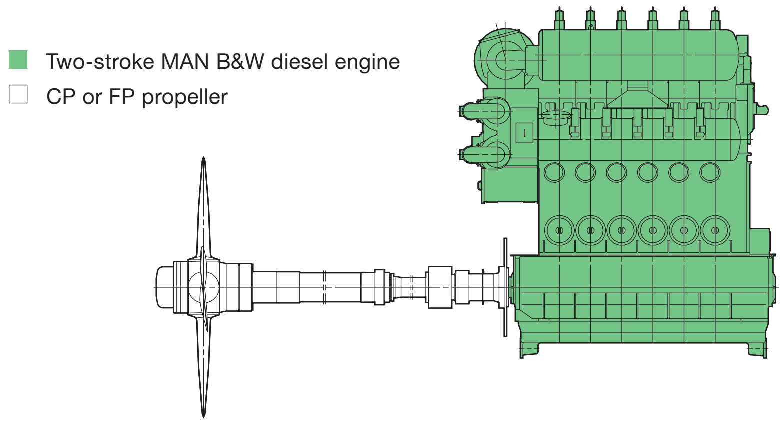


Fig. 11: Example of a diesel-electric propulsion system with CP (or FP) propeller

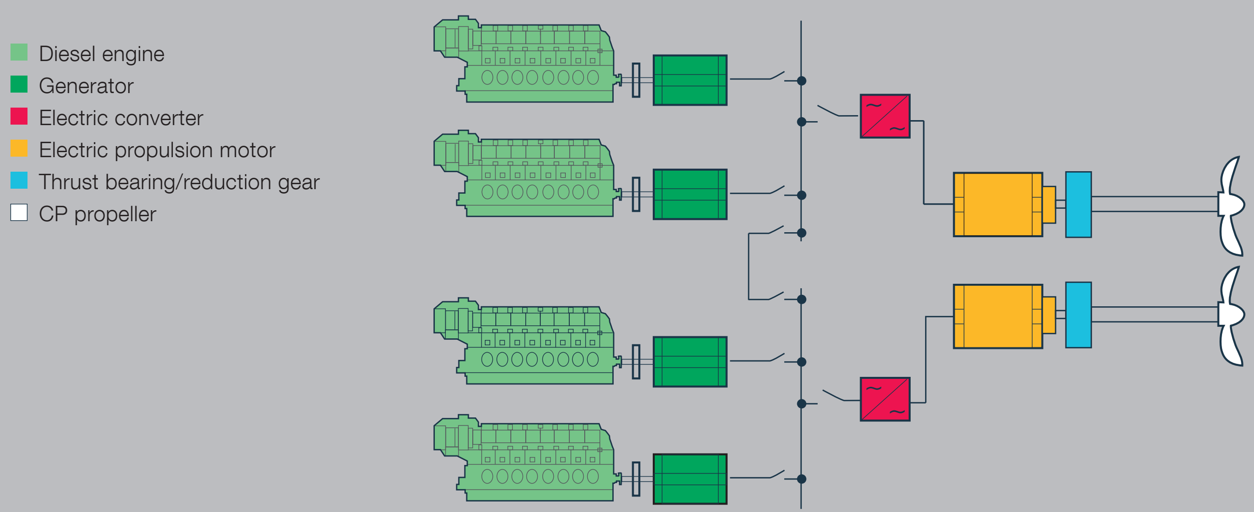


Fig 12: Example of diesel-electric propulsion system with CP propeller

The engine is equipped with a standard load-dependent arctic exhaust gas bypass system which works differently during the ice ramming procedure. Normally, the exhaust gas bypass will open under very cold ambient conditions to avoid too high pressures in the engine. During the ramming cycle, the exhaust gas bypass will close when the scavenge air pressure is low, thereby improving the dynamic performance of the engine.

Ice ramming also poses some other stresses on the propulsion system as the dynamic loading will be different from the normal propulsion mode. The thrust bearing on the main engine must handle the propeller thrust, including the dynamic loading from the ice operation. This may require a modification of the thrust bearing, including the engine structure around the thrust bearing, the journal bearing support, and the thrust collar.

One of the additional challenges with ice-ramming is that, most of the time, a ship with ice ramming capabilities will operate under normal open water conditions without ice. It means that the average main engine load can be very low, when comparing with the high engine load during ice ramming, a holistic approach to engine optimisation is required to suit the vessel operation.

Temperature restrictions and load-up procedures

To protect the engine against cold corrosion attacks on the cylinder liners, minimum temperature restrictions and load-up procedures have to be considered before starting the engine. It is important for cold regions where special load-up procedures may be necessary.

Starting up a warm engine – normal load-up procedures

In summary, the load-up procedures recommended for normal start-up of an engine are shown in Table 10.

The load-up procedures described are valid for MAN B&W two-stroke Mk. 9 engines and later with a cylinder bore greater than or equal to 80 cm. The procedures may with benefit also be applied on engines with a smaller bore size. However, the load-up program recommendation (from 80% to 90% of SMCR speed in 30 minutes and from 90% to 100% in 60 minutes) is still valid for engines with bore sizes of 70 cm and lower.

Note that the following recommendations are based on the assumption that the engine has already been well run in.

Fixed pitch propellers
Normally, a minimum engine jacket water temperature of 50ºC is recommended before the engine may be started and gradually run up to 80%. The speed is to be increased slowly from 80% to 90% of the specified MCR speed (SMCR rpm) during 60 minutes.

For running-up to between 90% and 100% of SMCR rpm, it is recommended that the speed is increased slowly over a period of 90 minutes.

Controllable pitch propellers
Normally, a minimum engine jacket water temperature of 50°C is recommended before the engine may be started and gradually run up to 50% pf SMCR power, and slowly from 50% to 75% of specified MCR load (SMCR power) during 60 minutes.

For running-up to between 75% and 100% of SMCR power, it is recommended that the load is increased slowly over a period of 90 minutes.

These requirements are summed up in Table 10.

Table 10: Load-up requirements for 80-bore engines and larger. Also recommendable for engines of lower bore

Requirements	
Mk. 9+ engines >= 80 bore with fixed pitch propeller, 150-minute load-up program	
Speed intervals, % SMCR speed (rpm)	**Time (minutes)**
80 → 90	60
90 → 100	90
Mk. 8 and older engines >= 80 bore with fixed pitch propeller, 90-minute load-up program	
Speed intervals, % SMCR speed (rpm)	**Time (minutes)**
80 → 90	30
90 → 100	60
Mk. 9+ engines >= 80 bore with controllable pitch propeller, 150-minute load up program	
Speed intervals, % SMCR speed (rpm)	**Time (minutes)**
50 → 75	60
75 → 100	90

Starting up a cold engine – exceptional load-up procedures

Fixed pitch propellers

In exceptional circumstances where it is not possible to comply with the normal recommendations regarding engine jacket water temperature, a minimum temperature of 20°C can be accepted before the engine is started and slowly run up to 80% of SMCR rpm.

Before exceeding 80% SMCR rpm, a minimum jacket water temperature of 50°C should be obtained before the normal start load-up procedure may be continued, see Table 11.

Controllable pitch propellers

In exceptional circumstances where it is not possible to comply with normal recommendations regarding engine jacket water temperature, a minimum of 20°C can be accepted before the engine is started and slowly run up slowly to 50% of SMCR power.

Before exceeding 50% SMCR power, a minimum jacket water temperature of 50°C should be obtained before the normal start load-up procedure may be continued, see Table 11.

The time required for increasing the jacket water temperature from 20°C to 50°C depends on the amount of water in the jacket cooling water system and the engine load, and cannot be stated explicitly.

Preheating during standstill periods

During short port stays (less than 4-5 days), it is recommended keeping the engine preheated. The purpose is to prevent temperature variations in the engine structure and the corresponding thermal expansion variations, and, thereby, lower the risk of leakage and fatigue.

The jacket cooling water outlet temperature should be kept as high as possible (approximately 85°C). Before start-up of the engine, the freshwater temperature should be increased to at least 50°C, either by the auxiliary engine cooling water, or by a built-in preheater in the jacket cooling water system, or a combination of both.

A standard preheater system with a built-in preheater is shown in Fig. 13.

The preheater is placed in parallel with the jacket water pumps. As the arrows indicate, the directions of the preheater water flow and the jacket water flow are the same. The system makes it possible to circulate the water through the engine and thereby back to the jacket water pumps and the preheater, even during standstill periods. The water temperature can then be adjusted to a specified high temperature before start-up.

Preheater capacity

When a preheater is installed in the jacket cooling water system as shown in Fig. 13, the preheater pump capacity should be about 10% of the jacket water main pump capacity. Based on experience, it is recommended that the pressure drop across the preheater should be approximately 0.2 bar. The preheater

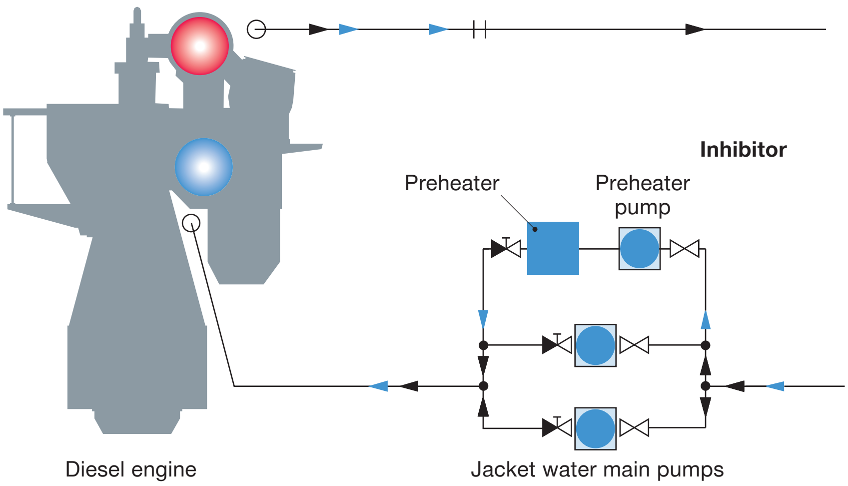


Fig. 13: A temperature sensor after the preheater controls the preheater operation.

Table 11: Temperature restrictions and load-up procedures for starting a cold engine in exceptional cases

Start of a cold engine (exceptional load-up procedures) Required jacket water temperature: minimum 20°C	FPP: fixed pitch propeller CPP: controllable pitch propeller
Recommended start of an engine for normal engine load operation	
A. run up slowly Minimum temperature: 20°C	FPP – from 0% up to 80% SMCR speed CPP – from 0% up to 50% SMCR power
B. run up slowly (minimum 60 min.) Minimum temperature: 50°C	FPP – from 80% up to 90% SMCR speed CPP – from 50% up to 75% SMCR power
C. run up slowly (minimum 90 min.)	FPP – from 90% up to 100% SMCR speed CPP – from 75% up to 100% SMCR power

pump and the jacket water main pump should be electrically interlocked to avoid the risk of simultaneous operation.

The preheater capacity depends on the required preheating time, and the required temperature increase of the engine jacket water. Fig. 14 shows the relationship between jacket water temperature and preheating time. The relationship is almost the same for all engine types.

In general, a temperature increase of about 35°C (from 15°C to 50°C) is required, and a preheating time of 12 hours requires a preheater capacity of about 1% of the engine`s NMCR power.

When sailing in arctic areas, the required temperature increase may be higher, possibly 45°C or even higher, and therefore a larger preheater capacity is required, as shown with dotted lines in Fig. 14.

The curves are based on the assumption that, at the start of preheating, the engine, and engine room are of equal temperatures. It is assumed that the temperature of the engine structure will increase uniformly during preheating, therefore steel masses, and engine surfaces in the lower part of the engine are also included in the calculation.

The results of the preheating calculations may therefore be somewhat conservative.

Engine room ventilation

In addition to providing sufficient air for combustion purposes in the main engine, auxiliary diesel engines, fuel fired boiler, etc., the engine room ventilation system should be designed to remove the radiation and convection heat from the main engine, auxiliary engines, boilers, and other components.

Enough ventilation air should be supplied and exhausted through suitably protected openings arranged in such a way that these can be used in all weather conditions. Care should be taken to ensure that no seawater can be drawn into the ventilation air intakes.

Furthermore, the ventilation air inlet should be placed at an appropriate distance from the exhaust gas funnel to avoid the suction of exhaust gas into the engine room.

For arctic operation with very low ambient air temperatures, the application of a direct inlet air suction system for the combustion air to the main engine itself, could be an advantage to avoid a too low air temperature in the engine room. It may be necessary to heat the air for the main engine.

Having direct air suction for the engine implies that the engine room itself is fitted with its own separate air ventilation system and fans. For further information on engine room ventilation and efficiency improvements, see the separate section and Fig. 15 in the paper: Efficiency improvements [14].

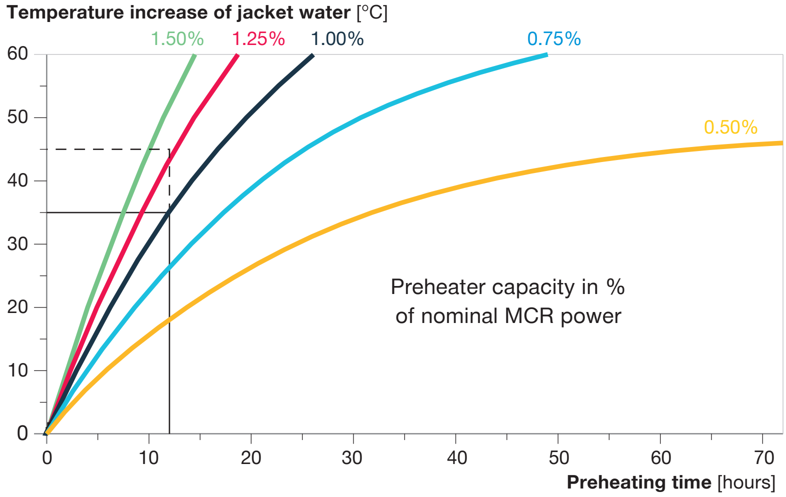


Fig. 14: Preheating of MAN B&W two-stroke combustion engine. Curves showing the temperature increase as function of preheating time are shown for different preheater sizes (% of nominal MCR power).

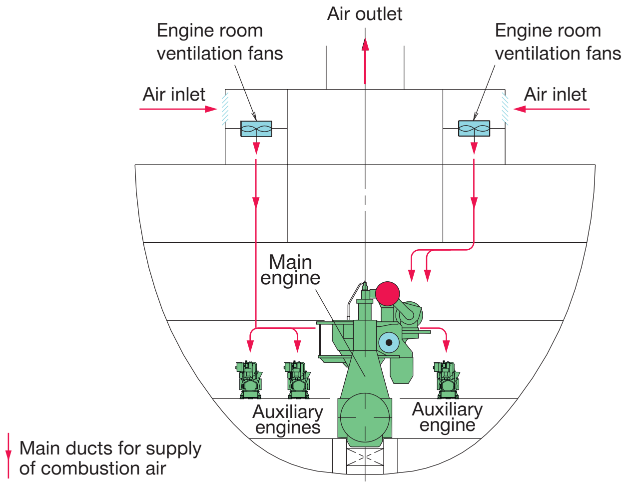


Fig. 15: An example of an engine room ventilation system, where ventilation fans blow air into the engine room via air ducts

Design recommendations for extremely low air temperature operation

When a standard ambient temperature-matched main engine on a ship operates under arctic conditions with low turbocharger air intake temperatures, the density of the air will be higher than at ISO conditions. As a result, the scavenge air pressure will also be higher. In order to prevent such excessive pressures under low ambient air temperature conditions, the turbocharger air inlet temperature should be kept as high as possible (by heating, if possible).

In general, a turbocharger with a normal layout can be used in combination with an exhaust gas bypass. If not sufficient, an arctic exhaust gas bypass system should be considered.

Furthermore, the scavenge air cooling water temperature should be kept as low as possible. However, for an inlet air temperature below approx. –10ºC, some engine design precautions must be taken.

Arctic exhaust gas bypass

With a load-dependent arctic exhaust gas bypass system, part of the exhaust gas bypasses the turbocharger turbine. It results in less energy to the compressor and a reduction of the air supply and scavenge air pressure to the engine.

The exhaust gas bypass system ensures that when the engine is running at part load at low ambient air temperatures, the load-dependent scavenge air pressure is close to the corresponding pressure on the scavenge air pressure curve valid for ISO ambient conditions. When the scavenge air pressure exceeds the read-in ISO-based scavenge air pressure curve, the bypass valve will variably open and, irrespective of the ambient conditions, ensure that the engine is not overloaded.

For considerations on extreme ambient conditions, the influence hereof, and further information on the application of an arctic exhaust gas bypass, please see the separate paper “Influence of ambient temperature conditions”

Other low-temperature precautions

Low ambient air temperatures and low seawater temperature conditions come together. The cooling water inlet temperature to the lube oil cooler should not be lower than 10°C, as the viscosity of the oil in the cooler will otherwise be too high, and the heat transfer inadequate.

Furthermore, to keep the lube oil viscosity low enough to ensure proper suction conditions in the lube oil pump, it may be advisable to install heating coils near the suction pipe in the lube oil bottom tank.

The following additional modifications of the standard design practice should be considered as well:

- Larger electric heaters for the cylinder lubricators or other cylinder oil ancillary equipment
- Cylinder oil pipes to be further heat traced/insulated
- Upgraded steam tracing of fuel oil pipes
- Increased preheater capacity for jacket water during standstill
- Different grades of lubricating oil for turbochargers
- Space heaters for electric motors
- Sea chests must be arranged so that blocking with ice is avoided.

Engine components and start air amounts

For ships with the Finnish-Swedish ice class notation IC, IB, IA, and even IA super or similar, most MAN B&W two-stroke diesel engines meet the ice class demands, i.e. there will normally be no changes to the main engines. This again means that the standard thrust bearings for most of the MAN B&W engine types are sufficient, but it must be evaluated on a case-by-case basis, considering the maximum thrust of the propeller.

If the ship has ice class notation IA super, and the main engine must reverse to go astern (FP propeller), the starting air compressors must be able to charge the starting air receivers within half an hour, instead of one hour, i.e. the compressors must be double in size compared to normal.

For other special ice class notations, the engine’s compliance with specific requirements must be checked individually.

Increased steam production in wintertime

When the air intake temperature drops, so does the exhaust gas temperature after the turbine and the service steam production from the ecnomiser reduces.

In order to cover the lack of service steam an auxiliary burner is applied but on the cost of additional fuel being burnt.

In order to decrease the total fuel consumption of a vessel MAN ES has developed the Economicer energy control (EEC) feature. The EEC feature minimizes the overall fuel consumption by allowing more exhaust gas energy to be extracted from the ME

The paper " Economicer energy control for increased service steam production" provides detailed descriptions of the EEC feature available for MAN B&W two-stroke engines. In addition, an example showing how the EEC feature can reduce total fuel consumption is also provided.

Examples of EEDI calculations for ice-classed ships

In this section, different ships with an ice class have been evaluated to figure out how much change in engine power is required to meet the EEDI and minimum power requirements for the ice class of the ship. The calculations are based on the following assumptions:

- The hull form is a standard design of a bulk carrier or a tanker
- A margin of 5 or 4 percentage points is assumed to ensure EEDI compliance
- The first ship has an engine running on LNG, but calculations for MDO are also included for comparison
- The following ship calculations are only done for MDO.

Aframax
The current configuration for the Aframax example is ice class IB. If the ship uses MDO, it will not be able to keep the IB ice class notation because the minimum propulsion power is higher than the limit for maximum propulsive power required to attain the EEDI in phase 3 for any propeller configuration.

It would still be able to stay within the limits in phase two, but only if the ship has a CP propeller reducing the power required to attain the ice class notation.

It would be possible for the ship to comply with the lower ice class IC for any of the configurations. But assuming that the ship's main fuel is LNG, any of the ice classes can be met

Table 12: Calculations for ice class compliance for an Aframax tanker

Ship type	Aframax tanker
Fuel	HFO – MDO – LNG
Length, L_{pp}	241 m
Breadth, B	44 m
Draught, D	14 m
Block coeficient, C_B	0.83
dwt, scantling	113,000 tonnes
Ice class	IB
SMCR power	13,800 kW
SMCR speed	87.3 rpm
Assumed SFC for MDO at 75% load	158.7 g/kWh
Assumed SGC for LNG	132.8 g/kWh / 3.14 g/kwh SPOC
Gross tonnage, GT	65,000 gt
Lightweight, lwt	23,600 tonnes
Cargo capacity	130,000 m^3

Propeller	FP	CP
Minimum power to comply with IC	8,100 kW	7,290 kW
Minimum power to comply with IB	12,000 kW	10,800 kW
Minimum power to comply with IA	16,200 kW	14,580 kW
EEDI reference line	4.36	100%
EEDI requirement phase 2	3.49	80%
EEDI requirement phase 3	3.04	70%

Current config		MDO		LNG
Without ice class applied	3.83	88%	2.98	68.4%
EEDI if IC applies	3.75	85.4%	2.92	66.9%
EEDI if IB applies	3.74	85.1%	2.91	66.8%
EEDI if IA applies	3.55	80.8%	2.76	63.4%

Maximum SMCR to for compliance with EEDI	MDO	EEDI	LNG	EEDI
EEDI Phase 2 without ice class	11,100 kW	76%	16,800 kW	77.4%
EEDI Phase 2 for ice class IC	11,200 kW	76%	17,300 kW	77.2%
EEDI Phase 2 for ice class IB	11,200 kW	75.8%	17,400 kW	77.3%
EEDI Phase 2 for ice class IA	12,200 kW	76%	19,000 kW	77.6%
EEDI Phase 3 without ice class	8,900 kW	65.8%	13,500 kW	67.4%
EEDI Phase 3 for ice class IC	9,000 kW	66%	14,000 kW	67.5%
EEDI Phase 3 for ice class IB	9,000 kW	65.8%	14,000 kW	67.4%
EEDI Phase 3 for ice class IA	9,800 kW	66%	15,000 kW	66.8%

for any configuration of the propeller and comply with the limit for EEDI phase 3. This shows how much can actually be gained by changing to a lower carbon intensive fuel, which is a possible way of complying with future restrictions.

Looking again at the maximum propulsion power allowed to attain the required EEDI, it can be seen that for MDO, the gain in power is not significant. This is probably due to the already high block coefficient, and also that the capacity is quite high in relation to the engine output for the ship. Therefore, this ship does not benefit much from the ice class correction factors, but benefits a lot from the choice of fuel.

Kamsarmax bulk

The next example for evaluation is a Kamsarmax bulk carrier of ice class IC, see Table 13.

Table 13 shows that, without any reduction of power, the ship would be able to comply with the EEDI restrictions for phase 2 and the minimum power required to attain the ice class notation desired with FP propeller and CP propeller, even if the ship was of ice class IB.

If the ship belonged to ice class IB equipped with an FP propeller, there would be a slight problem for EEDI phase 3. But if a CP propeller was installed, the ship would actually be able to meet the requirements.

This calculation clearly shows that by implying an ice class, the maximum SMCR allowed for the engine can be increased, and for the EEDI you can lower the EEDI by 6% just by implying an ice class. But the calculations also show that the correction factors do not have a significant influence as regards choosing between IB and IC as the maximum power to comply with EEDI is almost the same for ice class IC as IB, which was also shown in the previous example in Table 12.

Table 13: Calculations for ice class compliance for a Kamsarmax bulk carrier

Ship type	Kamsarmax bulk carrier
Fuel	HFO - MDO
Length, L_{pp}	225.5 m
Breadth, B	32.2 m
Draught, D	14.5 m
Block coeficient, C_B	0.80
dwt	82,000 tonnes
Ice class	IC
SMCR power	10,000 kW
SMCR speed	92.2 rpm
Assumed SFC	156.1 g/kWh
Gross tonnage, GT	44,000 gt
Lightweight, lwt	14,000 tonnes

Propeller	FP	CP
Minimum propulsive power to comply with IC	5,300 kW	4,770 kW
Minimum propulsive power to comply with IB	7,400 kW	6,660 kW
EEDI reference line	4.36	100%
EEDI requirement phase 2	3.49	80%
EEDI requirement phase 3	-	-
Current configuration	MDO	
Without Ice class applied	3.67	84%
EEDI if IC applies	3.40	78%
EEDI if IB applies	3.40	78%

Maximum SMCR to for compliance with EEDI	MDO	EEDI
EEDI phase 2 without ice class	8,500 kW	77.9%
EEDI phase 2 for ice class IC	10,000 kW	78%
EEDI phase 2 for ice class IB	10,000 kW	77.8%
EEDI phase 3 without ice class	6,900 kW	65.8%
EEDI phase 3 for ice class IC	7,800 kW	66.1%
EEDI phase 3 for ice class IB	7,800 kW	65.9%

Handymax bulk carrier
The next example illustrates a case of applying an ice class to a ship which is without ice class in its original design. Here, a standard Handymax bulk carrier is evaluated, applying the same assumptions as in the previous cases, see Table 14.

This Handymax bulk carrier is from a standard Handymax bulk carrier design without any ice class implementation. The EEDI can be decreased by 4% by implementing an ice class. This requires some restructuring of the hull and the propeller, etc., but would allow the ship to remain compliant if a slight derating of the SMCR would be initiated.

Looking at the numbers, it can be seen that this design would actually be able to comply with the ice class and EEDI for phase 2, though only up to ice class IC.

Looking at phase 3, it shows that this ship design would not be able to comply with the minimum power for any ice class and EEDI. This indicates that a change in the design would be necessary if this standard ship design needs to comply with the requirements for an ice class and the EEDI. Even during this, it would probably also require derating of the engine anyway, for compliance with the future EEDI phases.

Table 14 - Calculations for ice class compliance for a Handymax bulk carrier

Ship type	Handymax bulk carrier
Fuel	HFO, MDO, LNG
Length, L_{pp}	184 m
Breadth, B	32.2 m
Draught, D	12 m
Block coeficient, C_B	0.79
dwt	50,000 tonne
Ice class	-
SMCR power	7,250 kW
SMCR speed	88.7 rpm
Assumed SFC	158.7 g/kWh
Gross tonnage, GT	-
Lightweight, lwt	-
Cargo capacity	64,000 m^3

Propeller	FP	CP
Minimum propulsive power to comply with IC	4,800 kW	4,320
Minimum propulsive power to comply with IB	7,000 kW	6,300
EEDI reference line	5.52	100%
EEDI requirement phase 2	4.41	80%
EEDI requirement phase 3	3.86	70%
Current configuration	MDO	-
Without ice class applied	4.63	84.0%
EEDI if IC applies	4.44	80.5%
EEDI if IB applies	4.43	80.3%

Maximum SMCR to for compliance with EEDI	MDO	EEDI
EEDI phase 2 without ice class	6,200 kW	76.5%
EEDI phase 2 for ice class IC	6,800 kW	77.1%
EEDI phase 2 for ice class IB	6,700 kW	76.9%
EEDI phase 3 without ice class	5,000 kW	65.6%
EEDI phase 3 for ice class IC	5,400 kW	66.1%
EEDI phase 3 for ice class IB	5,400 kW	66.0%

Closing remarks

As demonstrated in this paper, MAN B&W two-stroke main engines on ocean-going ships are capable of operating smoothly in very cold and even arctic ice conditions, as long as special low-temperature and heavy running precautions have been taken.

The EEDI index is an important factor for future restrictions on CO_2 emissions. This applies especially to new designs, which aim at lowering emissions. However, because ice-classed ships meet stricter design requirements to the hull, propeller, power, etc., correction factors are set up to lower the requirements to enable compliance with EEDI. These factors include capacity, power and if the ship has a higher ice class such as IA or IA super.

MAN B&W two-stroke combustion engines can be derated to ensure compliance with the EEDI restrictions. However, it is important to bear in mind that a minimum power output is required to comply with the ice class. This means that the selected engine must fit within the limits of maximum power relative to EEDI and minimum power relative to ice class. Sometimes it requires a substantial increase in engine power to navigate in the icy conditions, but MAN B&W two-stroke engines can easily comply with the added power.

Even in partly ice ramming conditions valid for the high ice classes, an MAN B&W two-stroke engine directly coupled to a CP propeller can be applied as prime mover.

Ice-classed ships may come into focus in the near future, following the exploration of the passage way around the arctic going northeast and northwest. This could lead to new routes, especially during the summertime, but as the ice might melt even further in the years to come, it may open new opportunities for the ice-classed ships.

References

[1] National Snow and Ice Data Center, 28 August 2021 (new: https://nsidc.org/arcticseaicenews/)

[2] Finnish-Swedish Ice Class Rules, TRAFI, 22 December 2021 https://www.finlex.fi/data/normit/43682TRAFI_494_131_03_04_01_00_2016_EN_Jaaluokkamaarays_2017.pdf

[3] Maritime polar code, DNV-GL, 22 December 2021 https://issuu.com/dnvgl/docs/dnv_gl_imo_polar_code_2017-05_web

[4] RULES FOR CLASSIFICATION - Part 6 Additional class notations Chapter 6 Cold climate, DNV-GL, 22 December 2021 https://rules.dnv.com/docs/pdf/dnv/ru-ship/2021-07 dnv-ru-ship-pt6ch6.pdf

[5] Rules for the classification and construction of sea-going ships part i, Russian Maritime Register of Shipping, 22 December 2021 https://rs-class.org/upload/iblock/a2b/73513281_E73C_417F_BB15_0EF448302439.pdf

[6] Finnish ice classes equivalent to class notations of recognized classification societies and the determination of the ice classes of ships, TRAFI, 22 December 2021 https://www.finlex.fi/data/normit/43683/TRAFI_383804_03_04_01_00_2016_Vastaavuusluettelo_EN.pdf

[7] Guidelines for the application of the 2017 finnish-swedish ice class rules, trafi, 22 December 2021 https://www.traficom.fi/sites/default/files/media/regulation/FSICR%20Guidelines%202019.pdf

[8] Propulsion trends in tankers, MAN Energy Solutions, Denmark, December 2021

[9] Draft amendments to MARPOL Annex VI, IMO, December 2021

[10] 2018 guidelines on the method of calculation of the attained energy efficiency design index (EEDI) for new ships, IMO, December 2021

[11] Amendments to the 2018 guidelines on the method of calculation of the attained energy efficiency design index (EEDI) for new ships (resolution mepc.308 (73)), IMO, December 2021

[12] Basic principles of ship propulsion, MAN Energy Solutions, Denmark, December 2021

[13] Propulsion trends in bulk carriers, MAN Energy Solutions, Denmark, December 2021

[14] Efficiency improvements, MAN Energy Solutions, Denmark, December 2021

MAN Energy Solutions
2450 Copenhagen SV, Denmark
P +45 33 85 11 00
F +45 33 85 10 30
info-cph@man-es.com
www.man-es.com

All data provided in this document is non-binding. This data serves informational purposes only and is not guaranteed in any way. Depending on the subsequent specific individual projects, the relevant data may be subject to changes and will be assessed and determined individually for each project. This will depend on the particular characteristics of each individual project, especially specific site and operational conditions.

Copyright © MAN Energy Solutions. 5510-0140-01
Jun 2022 Printed in Denmark