



Basic principles of ship propulsion

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Future in the making

Optimisation of hull, propeller, and engine interactions for maximum efficiency

Future in the making



This paper will explain the most elementary terms used regarding ship types, dimensions and hull forms, and clarify some of the parameters pertaining to hull resistance, propeller conditions, and main engine capabilities. The interdependencies between these illustrate the complexity of optimising the hull as well as the interaction with the propulsion plant. Based on the explanations given, the reader will be able to work through the engine selection spiral introduced in this paper. This will facilitate selection of the right main engine(s) for the right hull in order to support the ambition of creating a sustainable future.

This paper is divided into five chapters which, with advantage, may be read in close connection to each other, providing the reader with an understanding of how the hull and propeller affect the engine running conditions and vice versa. The chapters can also be read independently. Therefore, some important information mentioned in one chapter may appear in another chapter as well.

The present edition of the paper has been substantially revised compared to previous editions. Among other topics, a description has been included of the environmental regulations implemented over the past years as well as their effect on modern ship propulsion plants. This is reflected in the new chapters 4 and 5, as well as in numerous updates to chapters 1 to 3.

Chapter 1 describes the most elementary terms used to define ship sizes and hull forms such as, for example, the ship's displacement, deadweight, draught, length between perpendiculars, block coefficient, etc. Other terms described include the calm water resistance, which consist of frictional, residual and air resistance, along with the influence of these resistances in service.

Chapter 2 deals with ship propulsion and the flow conditions around the propeller(s), and describes the main parameters related hereto.

The operating conditions of a propeller according to the propeller law are described for free sailing in calm weather. The influence of the propeller size and speed is considered along with different philosophies for optimising hull and propeller interactions.

Chapter 3 explains the basic principles related to diesel engines. Two engine selection spirals for, respectively, fixed and controllable pitch propellers are introduced. Also, the principles of the engine layout diagram are explained, along with the link to the propeller curve and a description of the principles behind optimum matching of engine and propeller.

The engine load diagram, the effect of heavy running, and the necessity of a light running margin are explained. Special considerations are given to the possibilities for including a shaft generator for both propeller types.

Chapter 4 is added in this revised edition. The chapter describes some of the environmental regulations governing shipping. An overview of the possible measures allowing for fulfilment of various rules are given.

An introduction to the EEDI regulations are given, explaining the main principles and how to calculate the required and attained index. Measures that can be

implemented to reduce the attained index are discussed. Finally, requirements for minimum propulsion power are considered.

Chapter 5 is an addition to this revised edition as well. Here, examples are given of the application of the engine selection spirals. The examples underline the importance of optimum matching of engine and hull in order to fulfil environmental regulations.

In general, the interdependencies between different hull, propeller, and engine related parameters described throughout the chapters are complex, and several different paths for optimising the ship can be taken by the ship designer, all depending on the priorities of the project. This also explains why two tender designs for the same ship never look the same.

It is considered beyond the scope of this publication to explain how propulsion calculations, i.e. power predictions as such are carried out, as the calculation procedure is complex. The reader is referred to the specialised literature on this subject, for example as stated in the final section "References".

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Chapter 1

Ship definitions and hull resistance

Hull dimensions and load lines

This chapter starts by giving the definitions of the expressions used in various situations for length, draught and breadth, followed by the definitions of load lines, which describe how much of the hull that is submerged.

Lengths

The overall length of the ship L_{OA} is normally of no consequence when calculating the hull's water resistance. The determining factors used are the length of the waterline L_{WL} and the so-called length between perpendiculars L_{PP} . The dimensions referred to are shown in Fig. 1.01.

The length between perpendiculars is the length between the foremost perpendicular, i.e. usually a vertical line through the stem's intersection with the waterline, and the aftmost perpendicular which, normally, coincides with the rudder stock. Generally, this length is slightly shorter than the waterline length, and it can often be expressed as

$$L_{PP} = 0.96 - 0.98 \times L_{WL}$$

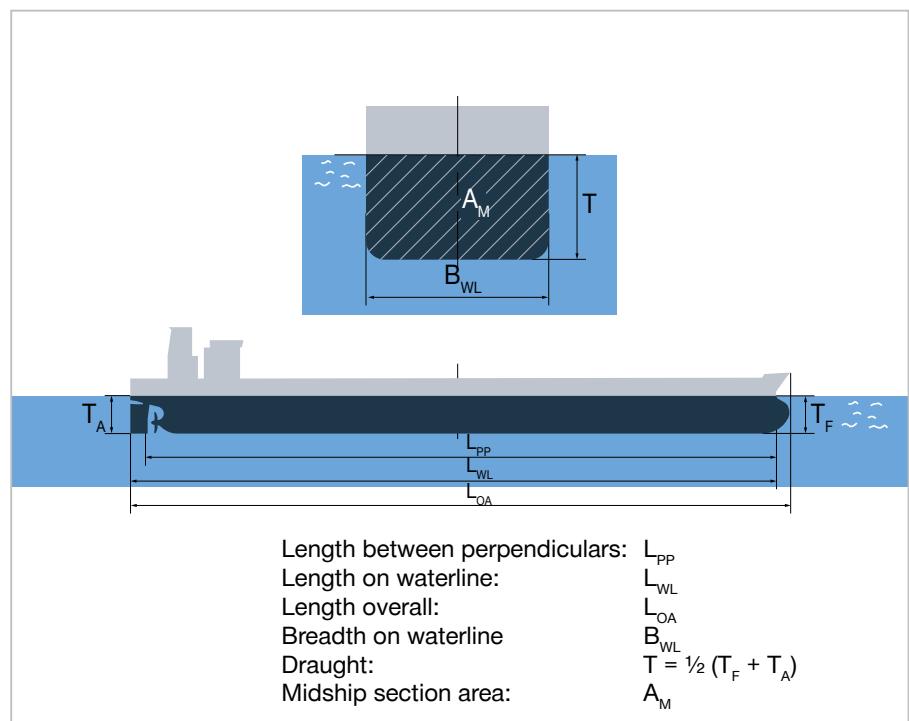


Fig. 1.01: Hull dimensions

Draught

The ship's draught, typically denoted T in literature, is defined as the vertical distance from the waterline to the point of the hull which is deepest in the water, see Figs. 1.01 and 1.02. The foremost draught T_F and aftmost draught T_A are normally the same when the ship is in the loaded condition, i.e. no trim.

The “scantling draught” is the distance from the keel to the summer load line, see the section “Load lines”.

Ballast draught is the draught of the ship with no cargo but adequate ballast water to ensure the stability of the ship.

Generally, the most frequently occurring draught between the scantling and the ballast draught is used as the “design draught”.

Breadth

Another important factor is the hull's largest breadth on the waterline B_{WL} , see Fig. 1.02.

Depth

The depth describes the distance from the keel to the underside of a given deck of the ship and is typically denoted D. The moulded depth describes the distance from the keel to the freeboard deck measured at the ship side.

Load lines

Painted halfway along the ship's side is the load line mark (also known as the Plimsoll Mark after its inventor), see Fig. 1.03. The lines and letters of the load line mark, which conform to the freeboard rules laid down by the IMO and local authorities, indicates the draught to which the ship may be loaded. The term freeboard refers to the minimum distance from the lowest watertight deck of the ship to the waterline, i.e. when fully loaded, the load line.

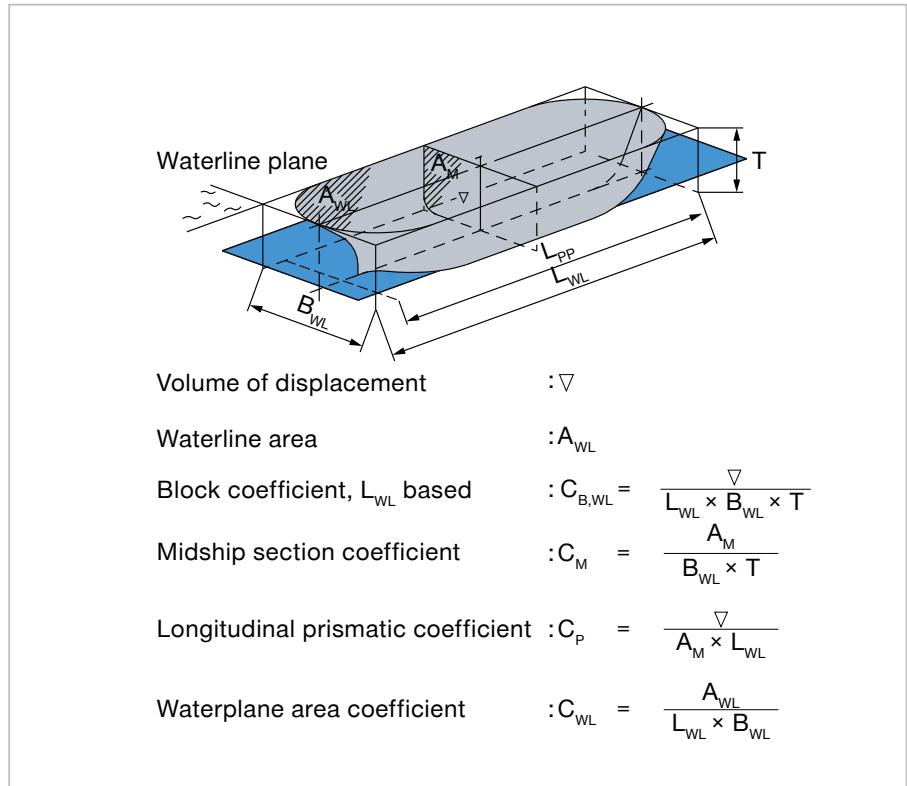


Fig. 1.02: Hull coefficients

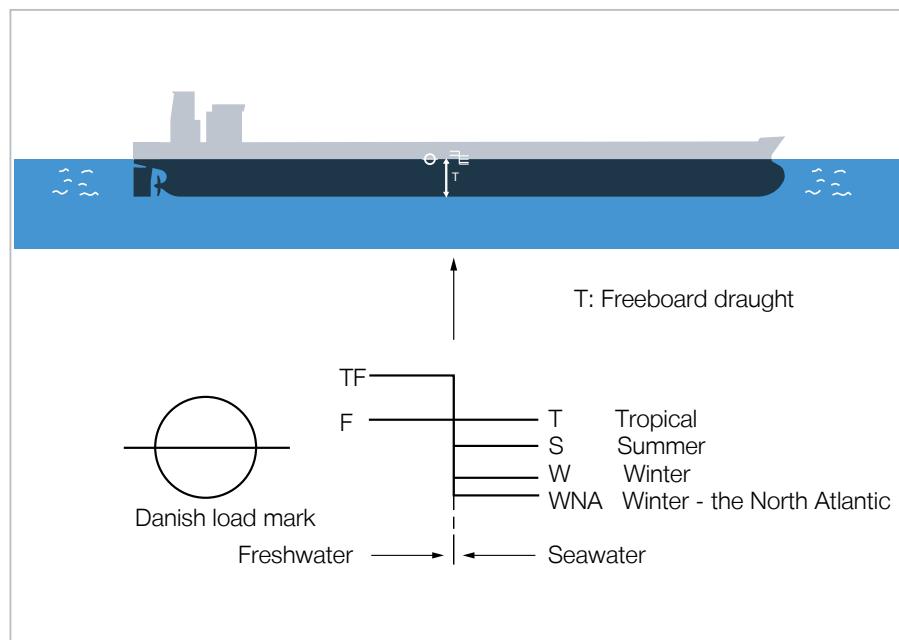


Fig. 1.03: Load lines

There are, e.g. load lines for sailing in freshwater and seawater, respectively, accounting for the difference in the mass density of water and the corresponding difference in buoyancy depending on the salinity. Further divisions for tropical conditions and summer and winter sailing are given. According to the international freeboard rules, the summer freeboard draught for seawater is equal to the "scantling draught".

The winter freeboard draught is less than that valid for summer because of the risk of bad weather. Correspondingly, a lower freeboard can be allowed in tropical waters. A further description of a ship's stability is outside the scope of this paper.

Size determining factors

Displacement, deadweight and lightweight

When a ship in loaded condition floats at an arbitrary water line, its displacement is equal to the relevant mass of water displaced by the ship. Displacement is thus equal to the total weight, of the relevant loaded ship, normally in seawater with a mass density of 1.025 t/m³.

Displacement comprises the ship's lightweight and its deadweight, where the deadweight is equal to the ship's loading capacity, including bunkers and other supplies necessary for the ship's propulsion. The deadweight at any time thus represents the difference between the actual displacement and the ship's lightweight, all given in tons.

$$\text{deadweight} = \text{displacement} - \text{lightweight}$$

The word "ton" does not always express the same amount of weight. Besides the metric ton (1,000 kg), there is the English ton (1,016 kg), which is also called the "long ton". A "short ton" is approx. 907 kg.

The deadweight tonnage (dwt) is based on the ship's loading capacity including the fuel and lube oils, etc. required for the operation of the ship. It is often used as an indication of size, referring to the loading capacity when the ship is loaded to its scantling draught, i.e. to the summer load line.

Sometimes, the deadweight tonnage may also refer to the ship's loading capacity at the design draught, if so, this will be mentioned.

The lightweight of a ship describes the weight of the ship itself in light condition, i.e. without load on the ship. The lightweight tonnage (lwt) is normally not used to indicate the size of a ship.

Table 1.01 indicates the rule-of-thumb relationship between the ship's, deadweight tonnage (summer freeboard/scantling draught) and lightweight tonnage.

The designation "payload" describes the weight that can be loaded onto the ship, excluding bunkers and supplies necessary for the ship's propulsion.

Gross and net tonnage

Without going into detail, it should be mentioned that there are also such measurements as the dimensionless gross tonnage (gt) and net tonnage (nt), which are calculated on the basis of the ship's volume and a multiplier defined by the IMO in the International Convention on Tonnage Measurement of Ships. Gross tonnage describes the entire enclosed ship space, whereas net tonnage covers only the cargo spaces. These measurements often form the basis for different regulatory requirements from the IMO.

Gross and net tonnage should not be confused with the older pre-convention terms gross register tons (grt), and net register tons (nrt) where 1 register ton corresponds to 100 English cubic feet, or 2.83 m³.

Coefficients related to the hull

The measures of length, breadth, draught and depth as well of weights can with some additional measures be used for calculating some dimensionless coefficients.

Block coefficient

Various form coefficients are used to express the shape of the hull. The most important of these coefficients is the block coefficient C_B , which is defined as the ratio between the displacement volume ∇ and the volume of a box with dimensions $L_{WL} \times B_{WL} \times T$, see Fig. 1.02.

$$C_{B,PP} = \frac{\nabla}{L_{PP} \times B_{WL} \times T}$$

In the case cited above, the block coefficient refers to the length on waterline L_{WL} . However, shipbuilders often use block coefficient $C_{B,PP}$ based on the length between perpendiculars, L_{PP} , in which case the block coefficient will, as a rule, be slightly larger because L_{PP} is normally slightly shorter than L_{WL} .

$$C_{B,PP} = \frac{\nabla}{L_{PP} \times B_{WL} \times T}$$

Table 1.01 shows typical block coefficients for different categories of ships, indicating a strong correlation between ship speed and block coefficient.

Water plane area coefficient

The water plane area coefficient C_{WL} expresses the ratio between the ship's waterline area A_{WL} and the product of the length L_{WL} and the breadth B_{WL} of the ship on the waterline, see Fig. 1.02.

$$C_{WL} = \frac{A_{WL}}{L_{WL} \times B_{WL}}$$

Generally, the water plane area coefficient is some 0.10 higher than the block coefficient.

$$C_{WL} \approx C_B + 0.1$$

This difference will be slightly larger on fast ships with small block coefficients where the stern is also partly immersed in the water and thus becomes part of the "water plane" area.

Midship section coefficient

A further description of the hull form is provided by the midship section coefficient C_M , which expresses the ratio between the immersed midship section area A_M (midway between the foremost and the aftmost perpendiculars) and the product of the ship's breadth B_{WL} and draught T , see Fig. 1.01.

$$C_M = \frac{A_M}{B_{WL} \times T}$$

In general, the midship section coefficient for merchant (displacement) ships is high, in the order of 0.95–0.99. Low values are found for fast ships such as container ships, whereas high values are found for the slower bulkers and tankers. Fast ferries and naval ships can have significantly lower values.

Prismatic coefficient

The longitudinal prismatic coefficient C_P expresses the ratio between displacement volume ∇ and the product of the midship frame section area A_M and the length of the waterline L_{WL} , see also Fig. 1.02.

$$C_P = \frac{\nabla}{A_M \times L_{WL}} = \frac{\nabla}{C_M \times B_{WL} \times T \times L_{WL}} = \frac{C_{B,WL}}{C_M}$$

The prismatic coefficient describes how voluminous the fore and aft parts of the ship are. A low prismatic coefficient indicates a voluminous midship section compared to the fore and aft parts.

Fineness ratio alias length/displacement ratio

The length/displacement ratio or fineness ratio, C_{LD} , is defined as the ratio between the ship's waterline length L_{WL} , and the length of a cube with a volume equal to the displacement volume.

$$C_{LD} = \frac{L_{WL}}{\sqrt[3]{\nabla}}$$

Longitudinal centre of buoyancy

LCB expresses the position of the centre of buoyancy and is defined as the distance between the centre of buoyancy and the midpoint between the ship's foremost and aftmost perpendiculars, termed "midships". The distance is normally stated as a percentage of the length between the perpendiculars.

Whether LCB is positive fore of the midpoint between perpendiculars, midships, or abaft midships is not stringent. The Holtrop Mennen, Ref. [1.3], power prediction method defines LCB positive fore of midships, whereas the method of Ref. [1.2] defines LCB positive abaft midships.

Relations between LCB and speed can be formulated for LCB. When measured positive fore midships: For a ship designed for high speeds, e.g. container ships, the LCB will, normally, be negative, whereas for slow-speed ships, such as tankers and bulk carriers, it will normally be positive. The LCB is generally between -3% and +3%.

Longitudinal centre of flotation

LCF is the geometrical centre of the waterplane area, which does not necessarily coincide with the midship position, due to the ship usually not being symmetrical around this point in the longitudinal direction. The ship trims about this point and not LCB.

Ship types with engine applications in overview

Depending on the nature of their cargo, and also the way the cargo is handled, ships can be divided into different categories, some of which are mentioned in Table 1.01 along with typical number of propeller(s) as well as engine type.

The three largest categories of ships are container ships, bulk carriers (for bulk goods such as grain, coal, ores, etc.) and tankers, which again can be divided into more precisely defined sub-classes and types.

Furthermore, Table 1.01 includes general cargo ships, where different kinds of cargo is brought onboard in pieces, and roll-on/roll-off (ro-ro) ships carrying lorry trailers and other cargoes that can be rolled on board. Rolling cargo in the form of cars and lorries are also combined with passenger transport in ro-pax ships.

Table 1.01 provides only a rough outline. In reality, there are many sub-types of ships, as well as combinations of these sub-types, i.e. ships carrying ro-ro cargo in the hull and containers stacked on deck.

As seen in Table 1.01, two-stroke engines are broadly used as main propulsion engines. Two-stroke diesel cycle engines have the highest efficiencies amongst mechanical means of propulsion, see Chapter 3.

This is reflected in the fact that two-stroke engines practically are the only engine of choice for ship types where fuel prices represent the main running costs, for example tankers, bulk carriers and container ships engaged in liner traffic between the continents.

For specialised ships, such as ferries and cruise ships, other parameters can constitute the main economic factors. The higher power density and lower height of four-stroke engines can make them more attractive in such cases.

For large ocean-going ships, fixed pitch propellers (FPP) are the most common choice, whereas controllable pitch propellers (CPP) are often utilised for smaller ships calling on smaller ports which demand increased manoeuvrability. In the case of a two-stroke main engine, it is possible to couple the propeller directly to the main engine, as an engine with an rpm matching the propeller optimum can be found. For

four-stroke engines, the higher rpm of the engine requires a reduction gearbox. Propeller and engine matching is elaborated in Chapter 3.

Efficiencies affecting the total fuel consumption

The total fuel power, P_{fuel} (power delivered through the fuel), required for propelling a ship through water is governed by the fuel equation. This section describes the parameters included in the equation, and references are made to other parts of this paper where these are explained in detail.

$$P_{fuel} = \frac{R_T \times V}{\eta_H \times \eta_O \times \eta_R \times \eta_S \times \eta_E} \text{ "Fuel equation"}$$

The resistance of the hull, R_T , is influenced by multiple parameters as described in the section “Resistance and influencing parameters”. In general, the resistance is proportional to the speed to the power of $i = 2$ to 3 , $R \propto k \times V^i$. This means that when increasing the speed with the power of 1 (linearly), the power required will increase with the speed to a higher power, $P \propto k \times V^{i+1}$.

Category	Type	Propeller	Main engine type	Size factor	C_B	V, kn	Iwt/dwt
Tanker	Crude oil carrier	1 FP	2-stroke	dwt	0.78-0.83	13-17	0.13-0.20
	Gas tanker / LNG carrier	1 FP	2-stroke, steam turbine	dwt / cubic meter (cbm)	0.65-0.75	16-20	0.30-0.50
	Product	1 FP	2-stroke	dwt	0.75-0.80	13-16	0.15-0.30
	Chemical	1 FP	2-stroke	dwt	0.70-0.78	15-18	0.30-0.50
Bulk carrier	Ore carrier	1 FP	2-stroke	dwt	0.80-0.85	14-15	0.11-0.15
	Regular	1 FP	2-stroke	dwt	0.75-0.85	12-15	0.13-0.30
Container ship	Liner carrier	1 FP or 2 FP	2-stroke	teu	0.62-0.72	20-23	0.28-0.34
	Feeder	1 FP or 1 CP	2 or 4-stroke	teu	0.60-0.70	18-21	0.34-0.41
General cargo ships	General cargo	1 FP	2 or 4 stroke	dwt / nt	0.70-0.85	14-20	
	Coaster	1 FP or 1 CP	2 or 4 stroke	dwt / nt	0.70-0.85	13-16	
Roll-on/roll-off cargo ship (ro-ro)		1 CP or 2 CP	2 or 4 stroke	Lane meters (lm)	0.55-0.70	18-23	0.6-1.4
Passenger-cargo ship (ro-pax)		2 CP	2 or 4-stroke	Passengers / lm	0.50-0.70	18-23	
Passenger ship	Cruise ship	2 CP	4-stroke	Passengers / gt	0.60-0.70	20-23	
	Ferry	2 CP	4-stroke	Passengers / gt	0.50-0.70	16-23	

Table 1.01: Typical characteristics of different ship types

These relations are further discussed in Chapter 2, section "Propeller law and power/speed curves".

The design speed of the ship is typically set according to the desire of the owner, if not limited due to regulations. In the fuel equation, the product of the resistance and speed is divided by the efficiencies of all of the components involved in propelling the ship - remembering all the efficiencies to consider is easy, as the indices combined spells HORSE.

η_H is the hull efficiency. It takes into account the difference in the effective (towing) power and the thrust power that the propeller must deliver to the water because the hull accelerates the water and the propeller sucks water past the hull. This is further elaborated in Chapter 2, section "Flow conditions".

η_O is the open water efficiency of the propeller (not behind the hull), giving the efficiency of the propeller working in a homogeneous wake field with flow perpendicular to the propeller and is further investigated in Chapter 2, section "Efficiencies and influencing parameters".

η_R is the relative rotative efficiency of the propeller, accounting for the changes in water flow to the propeller behind the ship relative to the open water flow conditions, as further described in Chapter 2, section "Rotative efficiency".

η_s accounts for the efficiency of the shaft connecting the main engine and propeller. It depends on the length of the shaft, whether gearing is necessary, etc. This is further described in Chapter 2, section "Shaft efficiency".

η_E describes the efficiency of the engine, depending on type of design and a variety of other parameters described in Chapter 3.

Resistance and influencing parameters

The magnitude of resistance on the ship's hull, R_T , is naturally paramount for the power required to move the ship (as seen in the fuel power equation) and hereby the fuel consumption.

The following section will explain the physical phenomena giving rise to the different components of the total calm water resistance, also termed the source resistances, and quantify their contribution to the total resistance.

Components of resistance

A ship's calm water resistance is particularly influenced by its speed, displacement, and hull form. The total resistance R_T consists of many source-resistances R , which can be divided into three main groups, viz.:

- 1) Frictional resistance, R_F
- 2) Residual resistance, R_R
- 3) Air resistance, R_A

The influence of frictional resistance depends on the wetted surface of the hull, whereas the magnitude of residual resistance describes the energy lost by the ship setting up waves, eddies and by the viscous pressure resistance, which all depend on the hull lines. For slow moving ships such as tankers and bulkers, the frictional resistance is often of the greatest influence (70-90%) whereas for fast going ships, such as panamax container carriers, the frictional resistance may account for as little as half of the combined resistance.

Air resistance normally represents about 2% of the total resistance, however, with a significant increase up to approx. 10% for ships with large superstructures such as container ships with containers stacked on deck. If wind resistance is considered, the figures may increase.

Water with a speed of V and a density of ρ has a dynamic pressure of:

$$p_{dyn} = \frac{1}{2} \times \rho \times V^2 \text{ "Bernoulli's law"}$$

Thus, if water is being completely stopped by a body, the water will react on the surface of the body with the dynamic pressure, resulting in a dynamic force on the body.

This relationship is used as a basis when calculating or measuring the source-resistances R of a ship's hull, by means of dimensionless resistance coefficients C . Thus, C is related to the reference force K , defined as the force that the dynamic pressure of water with the ship's speed V exerts on a surface which is equal to the hull's wetted area A_s . The rudder surface is also included in the wetted area. The general data for resistance calculations are thus based on the following two important relations:

$$\text{Reference force: } K = \frac{1}{2} \times \rho \times V^2 \times A_s$$

$$\text{Source resistances: } R = C \times K$$

On the basis of many experimental tank tests, and with the help of pertaining dimensionless hull parameters, methods have been established for calculating all the necessary resistance coefficients C and, thus, the pertaining source-resistances R at an early project stage without testing, Ref. [1.2] and [1.3].

In practice, the approximate calculation of a particular ship's resistance, which is required for the initial dimensioning, is often verified and optimised by testing a model of the ship in a towing tank.

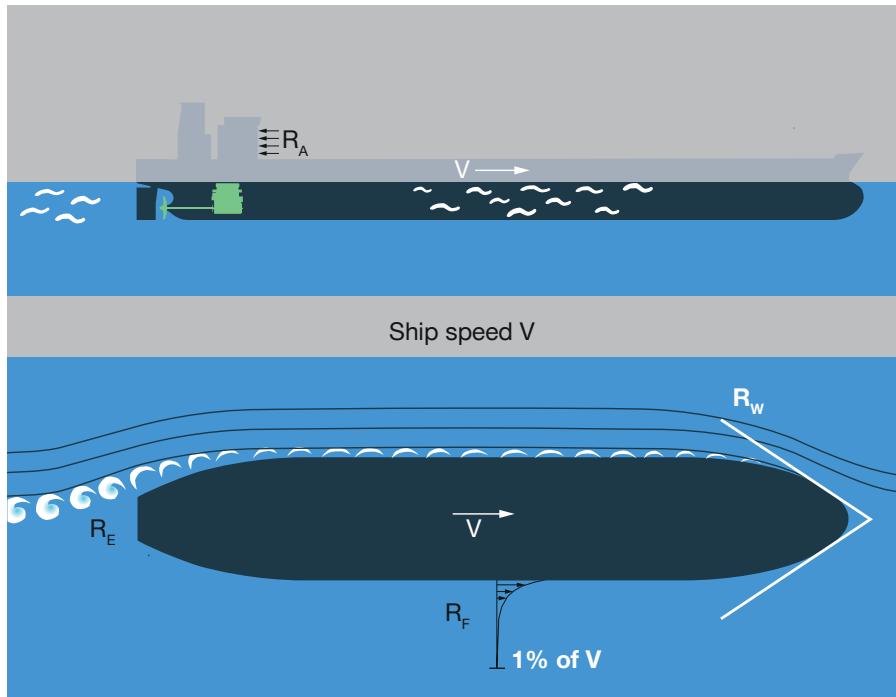


Fig. 1.04: Components of resistance on a ship

Frictional resistance, R_F

The frictional resistance R_F accounts for two effects: Firstly the friction of a flat plate of the same length as the hull and an area equivalent to the hull's wetted surface, A_s . Secondly the frictional resistance caused by the curvature of the ship's hull compared to a flat plate, see Fig. 1.05. The magnitude of the frictional resistance increases with the fouling of the hull, i.e. by the growth of algae, sea grass and barnacles.

$$R_F = C_F \times K$$

The wetted surface area A_s can be estimated based on Mumford's formula. As a rule of thumb, it can be estimated by this formula within 15% accuracy.

$$A_s = 1.025 \times L_{PP} \times (C_B \times B + 1.7 \times T) \\ = 1.025 \times \left(\frac{V}{T} + 1.7 \times L_{PP} \times T \right)$$

Residual resistance, R_R

Residual resistance R_R comprises wave making resistance and viscous pressure resistance. Sometimes the additional frictional resistance due to the curvature of the ship's hull are also included in the residual resistance.

Wave making resistance refers to the energy lost by the setting up waves during the ship's propulsion through water, and will typically, depending on the ship speed, form the greater part of the residual resistance, see Fig. 1.05.

The viscous pressure resistance is usually a small part of the total resistance. It arises from separation losses in the boundary layer and the increased thickness hereof along the ship's side. The result of increased boundary layer thickness is a lower pressure on the aft of the ship than on the fore part, which gives rise to resistance. The concept of the boundary layer is explained in Chapter 2, section "Flow conditions".

Generation of eddies when the flow separates from the hull surface is a special part of the viscous pressure resistance, which results in separation

losses, sometimes referred to as eddy resistance. Eddies are created both due to separation in the boundary layer and on the aft part of the ship, due to the abrupt changes in curvature found here.

The procedure for calculating the specific residual resistance coefficient C_R is described in specialised literature, Ref. [1.2].

$$R_R = C_R \times K$$

The residual resistance is typically very limited at low ship speeds, see the section "The Froude number", but it can be significant at elevated ship speeds.

This results in the combined resistance growing by as much as the cubic of the speed or even more. Accordingly, the required power and, hence, fuel consumption can be proportional to the speed to the power of four, $P \propto k \times V^4$ or even more, for fast ships. See also "Propeller law and speed power curves" in Chapter 2.

Air resistance, R_A

In calm weather, air resistance is, in principle, proportional to the square of the ship's speed, and the cross-sectional area of the ship above the waterline. It is important to distinguish air resistance from wind resistance, air resistance only accounts for the resistance from moving the ship through the air with no wind.

The air resistance can, similar to the foregoing resistances, be expressed as $R_A = C_A \times K_A$. Using a drag coefficient C_A and an area of the ship above the water line A_{air} , the resistance can be expressed as:

$$R_A = \frac{1}{2} \times \rho_{air} \times C_A \times A_{air} \times V^2$$

Typically, the drag coefficient can be estimated at approx. $C_A \approx 0.9$ depending on the shape of the superstructure, whereas the area above the waterline, on which the wind acts, must be calculated because the area depends on the wind direction. Further investigations on air resistance have been done in Ref. [1.4].

Parameters influencing resistance and optimisation hereof

Numerous parameters influence the total calm water resistance of a ship, and the nature of the influence can differ significantly depending on the speed of the ship. The following section provides an overview of the influencing parameters and a brief introduction to their relation. For in-depth understanding the reader is advised to consult special literature such as Ref. [1.2].

The Froude number

The Froude number is a dimensionless coefficient especially important for the resistance of ships. Ships with a similar Froude number will experience similar relative amounts of source resistance (relation between C_F , C_R , C_A) despite not being the same size and moving at different speeds.

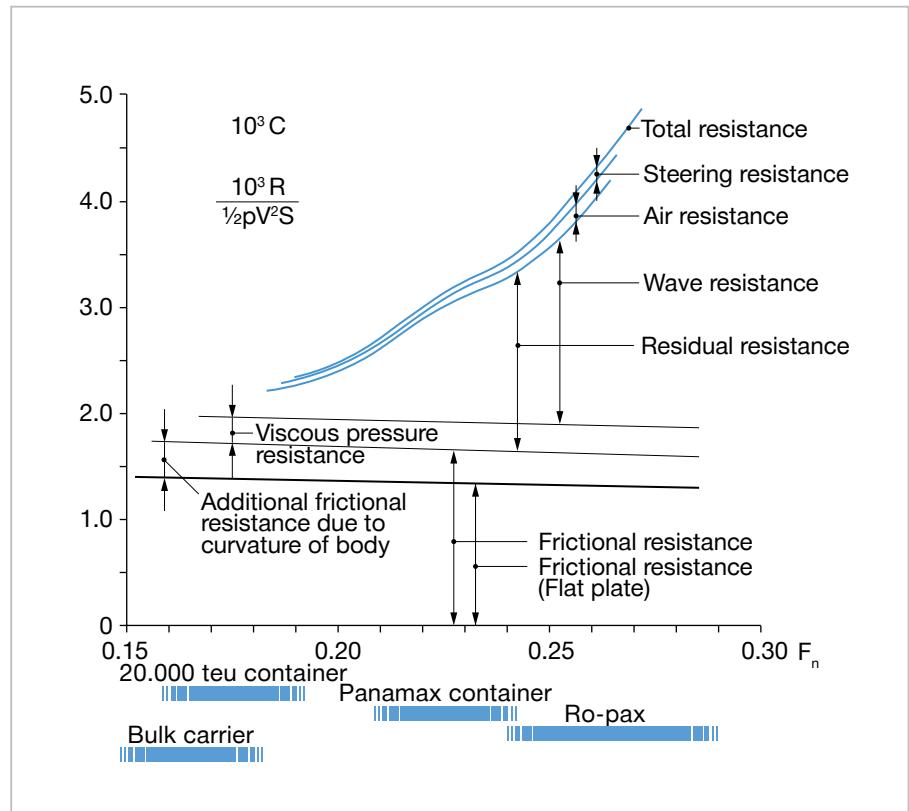


Fig. 1.05: Components of the total resistance as function of the Froude number from Ref. [1.2]

Length, m	speed, kn						
	14	16	18	20	22	24	26
100	0.23	0.26	0.30	0.33	0.36	0.39	0.43
150	0.19	0.21	0.24	0.27	0.30	0.32	0.35
200	0.16	0.19	0.21	0.23	0.26	0.28	0.30
250	0.15	0.17	0.19	0.21	0.23	0.25	0.27
300	0.13	0.15	0.17	0.19	0.21	0.23	0.25
350	0.12	0.14	0.16	0.18	0.19	0.21	0.23
400	0.11	0.13	0.15	0.16	0.18	0.20	0.21

Table 1.02: Froude number as function of the hull length and ship speed

$$Fn = \frac{V}{\sqrt{g \times L_{WL}}}$$

The total resistance, and thus fuel consumption, increases when the Froude number increases. This is because the wave making resistance increases significantly when the Froude numbers ex-

ceed approx. $Fn = 0.16$ to 0.17 , as can be seen in Fig. 1.05 from Ref. [1.2]. From this figure it is also clear that the friction is practically the only influencing parameter at $Fn < 0.15$.

In order to exemplify the influences of the different terms in the Froude number, a parameter sweep is displayed in Table 1.02.

When considering the formula for the Froude number, it becomes evident that by increasing the length of the ship, the value of the Froude number will decrease. Of course, this does not come without a price, as it increases the wetted surface area of the ship, and hereby the frictional resistance. Depending on the speed of the ship, this increase in absolute frictional resistance may in some cases be meaningful.

Typically, the length of the ship is the most costly parameter to adjust with regards to capital costs, but considering the EEDI regulations as described in Chapter 4 an increased length can be necessary if a specific speed is required for a special trade.

On the other hand, fuller container ships with larger block coefficients are seen now, as the speed of such ships has dropped significantly.

Influence of wetted surface area

The wetted surface area is paramount for the frictional resistance of the ship, which is seen to be predominant at $F_n < 0.15$ in Fig. 1.05.

By increasing the block coefficient, the wetted surface area is reduced relative to the cargo amount that can be carried, which is why slow-going ships such as tankers and bulkers have a high block coefficient.

The wetted surface area can be estimated by Mumford's formula, see section "Frictional resistance, R_F ".

Influence of block coefficient

The block coefficient describes the overall fullness of the hull. It is a good and easy-to-calculate indicator of a hull's resistance and capabilities with regard to speed. However, the other parameters described in this section may have a relatively larger influence, specifically on the resistance, than small changes to the block coefficient. In general, fast ships, for which the wave making is the primary resistance parameter, demand small block coefficients, whereas slower ships, for which the friction against the hull is the main resistance parameter, are most effective with a higher block coefficient.

Influence of the fineness ratio alias length/displacement ratio

The fineness ratio describes the length of the ship relative to the displacement, and influences the resistance, especially for faster ships. When increasing the fineness ratio for constant displacement the residuary resistance decreases. However, at the same time the wetted surface area is increased. The optimum length of the specific hull, with its specific design speed, must be found with due consideration to the limitations imposed by canal passages, harbours, production costs etc.

Influence of the prismatic coefficient

The prismatic coefficient is an important measure for describing how well the ship displaces the water while moving through it. The optimum for the prismatic coefficient changes greatly with the Froude number of the ship, and the reader is advised to consult specialised literature.

Added resistance in various conditions

Shallow water

In general, shallow water will have no influence when the water depth is more than 10 times the ship draught. Shallow water may increase the resistance of the ship significantly for a variety of rather complex reasons listed below. The reader is advised to consult special literature e.g. Ref. [1.5].

The pressure set up by the ship's motion through the water will be greater as the flow of the water is restricted, thereby retarding and increasing the size of the wave system set up by the ship, and resulting in added resistance. This effect is primarily for $F_n > 0.2$.

In addition, water will flow back around the ship due to the restriction of the waterway caused by the presence of the ship (similar to an increase in velocity due to a local obstruction in a pipe). Furthermore, the propeller will to some extent suck water faster past the ship surface than in deep water. Both these effects increase flow velocities and, therefore, also the resistance.

In addition, the so-called squat effect may occur. The squat effect means that the pressure underneath the ship is reduced as the limited gap between the seabed and the ship results in friction on the water under the hull. The result is that water will flow to the side instead of underneath the ship. The squat effect results in a larger than normal draught and hereby increased resistance. This is also primarily seen at elevated Froude numbers.

Weather, high seas

Waves created by high winds and swells can have a significant effect on the resistance experienced by the ship. Especially if the ship length and wave length are about equal.

The waves will set the ship in motion and lead to added resistance as more water is affected by the movement of the ship. Additionally the waves from the seaway will reflect on the hull, and thereby increase the resistance as well.

In combination, this is termed the wave drift force. The influence hereof can be seen if an object is left to float freely in the waves. Such an object would slowly move in the direction of the wave propagation.

Furthermore, rudder corrections will be much more frequently needed to stay on course. The use of the rudder will of course result in increased resistance, this effect being most predominant for small ships ($L_{PP} < 135 \text{ m} \approx 20,000 \text{ dwt}$) due to their lower directional stability. Additionally, the shape of the stem can greatly influence the ship's ability to "cut through" high seas.

Weather, wind

It is important to distinguish between air and wind resistance. Air resistance only accounts for the calm weather resistance from moving the ship through the atmosphere without any wind. Wind resistance is calculated in the same way as air resistance but takes into account the combined speed of the ship and the wind.

For large ocean-going ships, wind resistance will normally be significantly lower than the wave resistance, but for ships intended for trade in sheltered waters without large wave resistance, e.g. ferries, the wind resistance can be the most significant added resistance in heavy weather.

Fouling

Fouling can result in a substantial increase in frictional resistance. Extreme cases have been seen where fouling had increased the frictional resistance by as much as 100%, normally up to 20-30%.

Before IMO banned TBT (tributyl tin) for new applications from 2003, and the full ban from 2008, marine growth on the hull was slower, but TBT was banned as it is extremely toxic to the marine environment. This means that the management of the ship with regard to dry-docking, hull cleaning and propeller polishing must be optimised, and that the cost of mechanical cleaning measures and the cost of added fuel consumption due to fouling must be balanced.

When designing the ship, especially for trade in warmer seas where growth is fast, some margin must be included for fouling.

Similar to the fouling, the increased roughness of the hull over time, arising from dents from interactions with the quay, etc., will increase the resistance of the hull.

Ice

Ships sailing in ice will experience significantly higher frictional resistance. Designing for icy conditions is a specialist area. The reader is advised to consult the separate paper "Ice Classed Ships, Main Engines" and special literature on resistance in ice. An rpm extended engine *load* diagram can be of relevance to ice classed ships, see "Rpm extended load diagram" in Chapter 3.

Resistance margins in a slow steaming environment

To account for average weather, a relative resistance margin (called a sea margin) has traditionally been added to the power requirement for propelling the ship in calm waters at the design speed, as given, for example, in Ref. [1.2]. Lately the ship design speeds have been lowered as a result of the increased focus on fuel consumption and environmental impact, see Chapter 4. This has led to a lower power requirement and, therefore, smaller engines. If the traditional relative resistance margins are applied to these smaller engines, the resulting absolute power margin will also be smaller.

Nevertheless, the weather does not care whether a ship of the same size is designed for a reduced service speed or not. The absolute added resistance experienced on the hull will be the same. Therefore, it is important for the ship designer to consider resistance margins for the specific project. The designer must secure that the engine selected can ensure safe manoeuvring in all relevant conditions, and that all relevant operating points (power, rpm) for the propeller are inside the engine's *load* diagram. See the engine selection spirals in Chapter 3 for further guidance on this.

Chapter 2

Propeller propulsion

The traditional agent employed to move a ship is a propeller. It is typically applied as a single-screw plant or, sometimes, as a twin-screw plant. In rare cases more than two propellers can be found on naval ships and high speed ferries.

This chapter starts by giving a series of definitions on the geometry of the propeller, followed by considerations on the flow conditions around a propeller. Next, the parameters affecting the propeller efficiency are investigated, along with philosophies for optimising the hull and propeller interactions. In conclusion, acceleration performance and manoeuvring are discussed.

Definitions of parameters

Diameter

With a view to obtaining the highest efficiency and lowest fuel consumption, the largest possible propeller diameter, d , will normally be preferred.

In general the propeller is not allowed to extend below the baseline of the hull on merchant ships, and typically a small margin is included to ensure that the propeller is not damaged in the event of a grounding, during dry docking or similar. Interference between the propeller and seabed, or a rock, can damage not only the propeller, but also the propeller shaft, the gearbox (if installed), and the main engine itself.

Furthermore, the propeller size is limited by the distance between the propeller-tip and the bottom of the hull, as a propeller being too close to the hull can result in both high vibrations and noise.

In all operating conditions the propeller must be fully submerged in the water. Especially for bulkers and tankers, often operating in ballast condition, this sets a limit on the propeller diameter. See Table 2.01 for a comparison of a rule-of-thumb typical maximum draught T and diameter d ratios.

Container ships operate with a more constant displacement, as the mix between empty and full containers today is almost constant regardless of journey. This allows for increasing the propeller size.

Volume ships carrying low-density cargo, such as ro-pax ships, can typically have even larger d/T ratios as they have a very constant displacement. Such large ratios will require consideration to the design of the aft ship.

Pitch diameter ratio

The pitch diameter ratio p/d expresses the ratio between the propeller's pitch (angle) p and its diameter d , see Fig. 2.01. The pitch p is the distance the propeller "screws" itself forward through the water per revolution, provided that there is no slip, see "Slip".

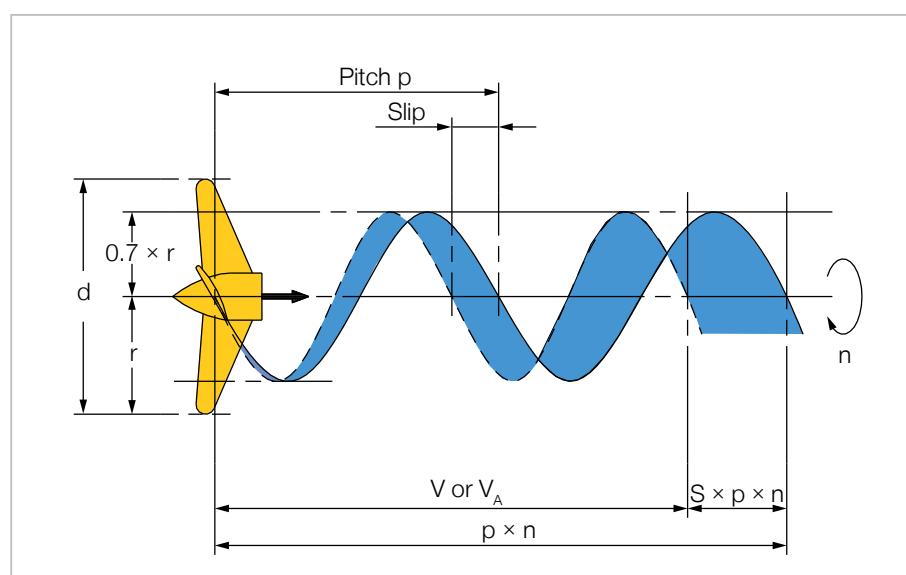


Fig. 2.01: Propeller pitch and slip

Bulkers and tankers	$d/T_d \approx 0.65$
Container ships	$d/T_d \approx 0.75$
Volume ships, high speed (i.e. ro-pax)	$d/T_d \approx 0.85$

Table 2.01: Approximate upper limit of ratio between propeller diameter and draught

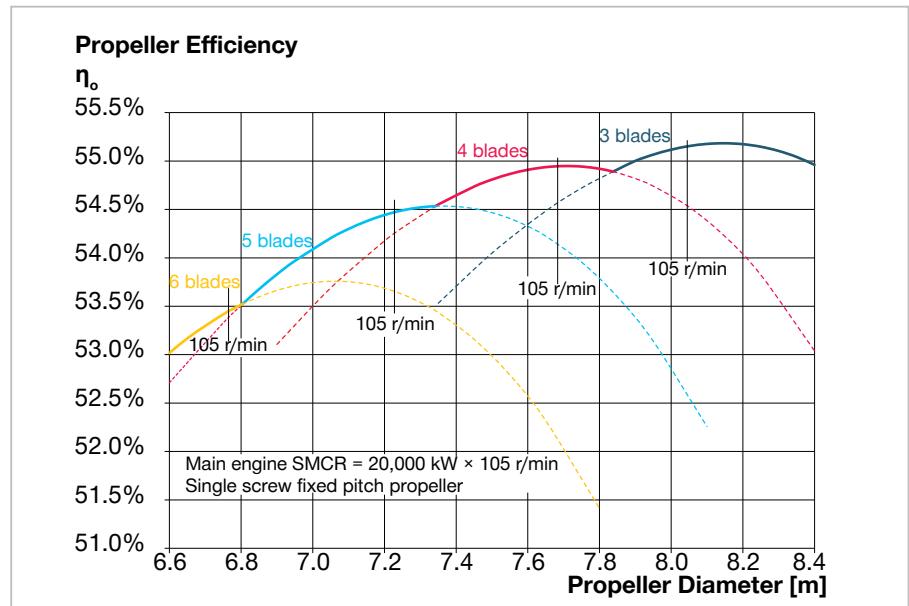


Fig. 2.02: Propeller efficiency and blade number. Notice that rpm is constant and therefore the diameter must be increased to achieve optimum

As the pitch varies along the blade's radius, the ratio is normally calculated for the pitch at $0.7 \times r$, where $r = d/2$ is the propeller's radius. The optimum pitch depends on the rate of revolution and propeller diameter.

Disk area coefficient

The disk area coefficient (sometimes referred to as expanded blade area ratio) defines the developed surface area of the propeller in relation to its disk area. This area must be sufficiently large to avoid harmful cavitation that can lead to erosion, see "Cavitation", but not too large, as this will increase the frictional drag on the propeller. A ratio of 0.40 to 0.60 is typical for normally loaded 4-bladed propellers. Highly loaded single screw container ships can see values as high as 1.0. In order to accommodate this, the blade number must be increased.

Blade number

Propellers can be manufactured with 2, 3, 4, 5, 6 or 7 blades. In general, the fewer the number of blades, the higher the propeller efficiency will be. However, for reasons of strength and vibrations, 4, 5 and 6-bladed propellers are normally used on merchant ships, with 4 blades being the most common.

The lower limit of the blade number is typically set by the varying magnitudes of the forces experienced by the blades when moving through the "shadow" of the hull when in top, and in the less restricted flow at the bottom. The flow conditions behind the hull are typically described as the "wake field", see the section "Flow conditions". A sufficient number of blades are required to smoothen the load.

Ships with a relatively large power requirement and heavily loaded propellers, e.g. fast single-screw container ships, may need 5 or 6-bladed propellers in order to have a sufficient area for transferring the load without cavitating.

The optimum propeller speed depends on the number of blades. Thus, for the same propeller diameter, a 6-bladed propeller has an about 10% lower optimum propeller speed than a 5-bladed.

For vibrational reasons, propellers with certain numbers of blades may be avoided in individual cases in order not to give rise to the excitation of natural frequencies in the ship's hull or superstructure, Ref. [2.1].

Propeller types and geometry

Fixed pitch propellers

Propellers of the fixed pitch (FP) type are cast in one block and are normally made of a copper alloy. The position of the blades is once and for all fixed, with a given pitch. This means that when operating in, for example, heavy weather conditions, the propeller performance curves, i.e. the combination of power and speed (rpm) points, will change according to the external conditions, see "Light running margin" in Chapter 3.

FP-propellers are simple and allow for the highest number of blades and blade area ratio. However, the drawback is that the engine must be stopped and reversed if reversal of the propeller is needed for deaccelerating the ship or to go astern. Note that the electronically controlled two-stroke engine mirrors the engine valve timing in astern, and is therefore equally capable of running astern as ahead.

Most ships that do not need a particularly good manoeuvrability, such as ocean-going container ships, tankers and bulk carriers are equipped with a FP-propeller, as it offers the highest efficiency. This is due to a small hub/diameter ratio of 0.15-0.20. See Fig. 2.03 for the conceptual differences to a CP-propeller.

Controllable pitch propellers

Propellers of the controllable pitch (CP) type have a relatively larger hub compared with the FP-propellers because the hub must accommodate the hydraulically activated mechanism to control the pitch. Therefore the CP-propeller is more expensive than a corresponding FP-propeller.

The great advantage of the CP-propeller is that it allows for operating the engine at any revolution or load desired, depending on the capabilities of the propeller control system. It also decouples the direction of thrust from the rotational direction of the engine, thereby enabling swift manoeuvring.

Because of the relatively larger hub, approx. a hub/diameter ratio of 0.22-0.30, the propeller efficiency is slightly lower, typically 1-2%. The deficiency of the CP-propeller can in most cases be regained as it is more adaptable to different operating conditions – ballast versus loaded, fouled hull versus clean, heavy weather, etc.

The application of a well-designed rudder bulb (see “Energy saving devices”) would reduce the difference between the efficiency of a FP and CP propeller

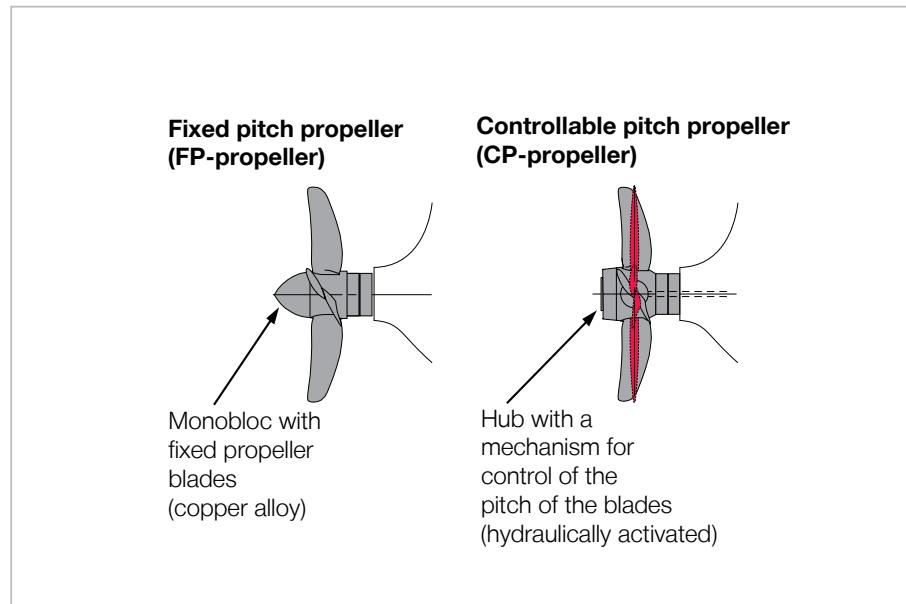


Fig. 2.03: Propeller types

significantly in most cases, as a rudder bulb eliminates the low pressure behind the hub.

The blade area ratio and blade number are limited, in order to allow for the rotation of the blades about their own axis. For a medium sized 4-bladed propeller, a disk area coefficient of approx. 0.78-0.80 can be regarded a limit.

Rake and skew

Fig. 2.04 illustrates the concept of rake and skew. For normal merchant ships the blades are typically raked aft to increase the clearance between the hull and the propeller. Skew is introduced to improve vibration characteristics, as the blades will then meet any changes in the wake field progressively, see “Flow conditions”.

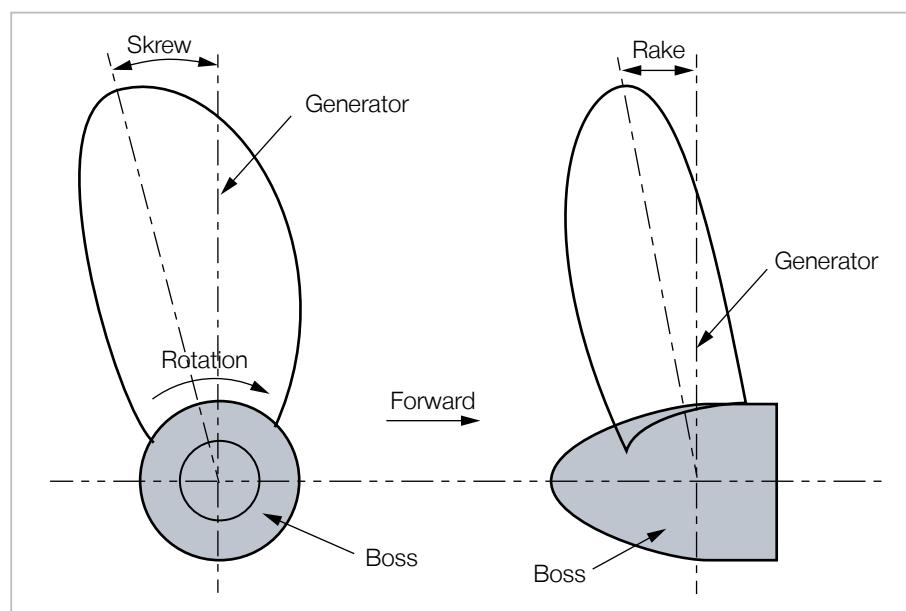


Fig. 2.04: Rake and skew

Special geometry propellers - the Kappel propeller

The Kappel propeller applies a special design where the tip of the blade is curved towards the suction side of the blade (i.e. in the forward direction), which minimises the size of the tip vortices, see "Open water efficiency". This design is able to reduce the required engine power by 3-6%, especially for rather loaded propellers, see Fig. 2.05.

Flow conditions

Boundary layer and wake

When the ship is moving, the friction of the hull will create a so-called boundary layer around the hull. In this boundary layer, the velocity of the water on the surface of the hull is equal to that of the ship, and reduces with the distance from the surface of the hull. At a certain distance from the hull, and, per definition, equal to the outer "surface" of the boundary layer, the water velocity is unaffected by the ship, see Fig. 1.04.

This "dragging" along of water on the ship's surface is what creates the frictional resistance discussed in "Resistance and influencing parameters" in Chapter 1.

The thickness of the boundary layer increases with the distance from the fore end of the hull. The boundary layer is therefore thickest at the aft end of the hull, Ref. [2.1]. This means that the pressure is lower on the aft body of the ship compared to the fore part, resulting in the viscous pressure resistance described in Chapter 1.

Additionally, the ship's displacement of water will set up wake waves both fore and aft.

The combined effects implies that the propeller will be working in a wake field.

Wake fraction coefficient, w

Mainly due to the boundary layer, the water flowing to the propeller will have an effective wake velocity V_w which has the same direction as the ship's speed V , see Fig. 1.04. This means that the velocity of arriving water V_A at the propeller (given as the average velocity over

the propeller's disk area and equal to the speed of advance of the propeller), is V_w lower than the ship's speed V .

The effective wake velocity at the propeller is therefore equal to $V_w = V - V_A$ and may be expressed in dimensionless form by means of the wake fraction coefficient w , by Taylor defined as:

$$w = \frac{V_w}{V} = \frac{V - V_A}{V} = 1 - \frac{V_A}{V}$$

The value of the wake fraction coefficient depends on the shape of the hull (related to the 3D effects of the hull's frictional resistance), and it can greatly influence the working conditions and, hereby, the efficiency of the propeller. The larger the block coefficient, the larger the wake fraction coefficient. Typical values of w are shown in Table 2.02 (p. 22), a first estimate can be given by $w = 0.5 \times C_B - 0.05$, Ref. [2.2]. For large d/T ratios (propeller diameter to draught of the ship) the size of the wake fraction will be reduced.



Fig. 2.05: Kappel propeller

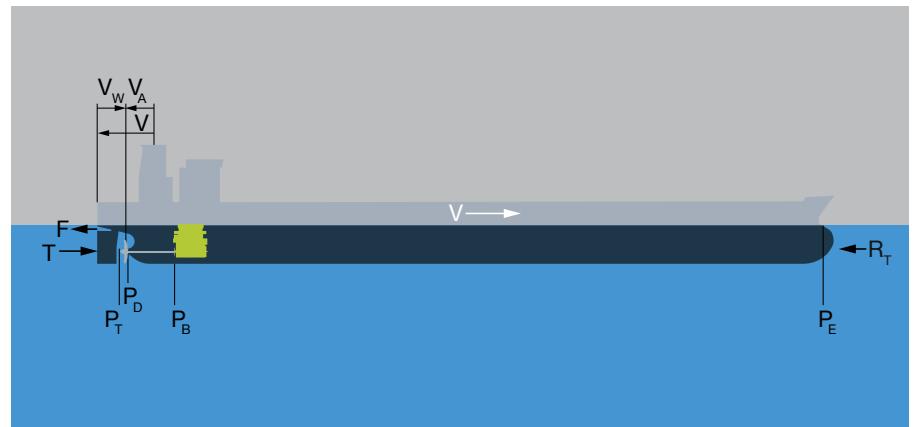


Fig. 2.06: Propulsion power

Thrust deduction coefficient, t

The action of the propeller causes the water in front of it to be “sucked” towards the propeller. This results in extra resistance on the hull normally called “augment of resistance” or, if related to the total required thrust force T on the propeller, “thrust deduction fraction” F , see Fig. 2.06. This means that the thrust force T on the propeller has to overcome both the ship’s towing resistance R_T and the extra resistance on the hull due to the sucking action of the propeller.

The thrust deduction fraction F may be expressed in dimensionless form by means of the thrust deduction coefficient t :

$$t = \frac{F}{T} = \frac{T - R_T}{T} = 1 - \frac{R_T}{T}$$

The thrust deduction coefficient t can be calculated by models set up in special literature, Ref. [1.2] and [1.3] or by CFD simulations.

Typical values of t are shown in Table 2.02 (p. 22), a first estimate can be given by $t = 0.27 \times C_B$ or by $t = 0.60 \times w$, again considering that the value of t will reduce for large d/T ratios.

Propeller coefficients

Propeller performance is described in systematic model tests, but to facilitate the practical use of these tests, certain dimensionless propeller coefficients have been introduced in relation to the diameter d , the rate of revolution n , and the water’s mass density ρ .

Advance number

The advance number is a dimensionless expression of the propeller’s speed of advance, V_A :

$$J = \frac{V_A}{n \times d}$$

For a given propeller pitch the advance number expresses the angle of the incoming water flow relative to the propeller blades, see Fig. 2.07. For a given propeller and pitch, knowledge of the advance number is sufficient to determine propeller thrust, torque and efficiency.

Thrust coefficient

Thrust force T , is expressed dimensionless, with the help of the thrust coefficient K_T :

$$K_T = \frac{T}{\rho \times n^2 \times d^4}$$

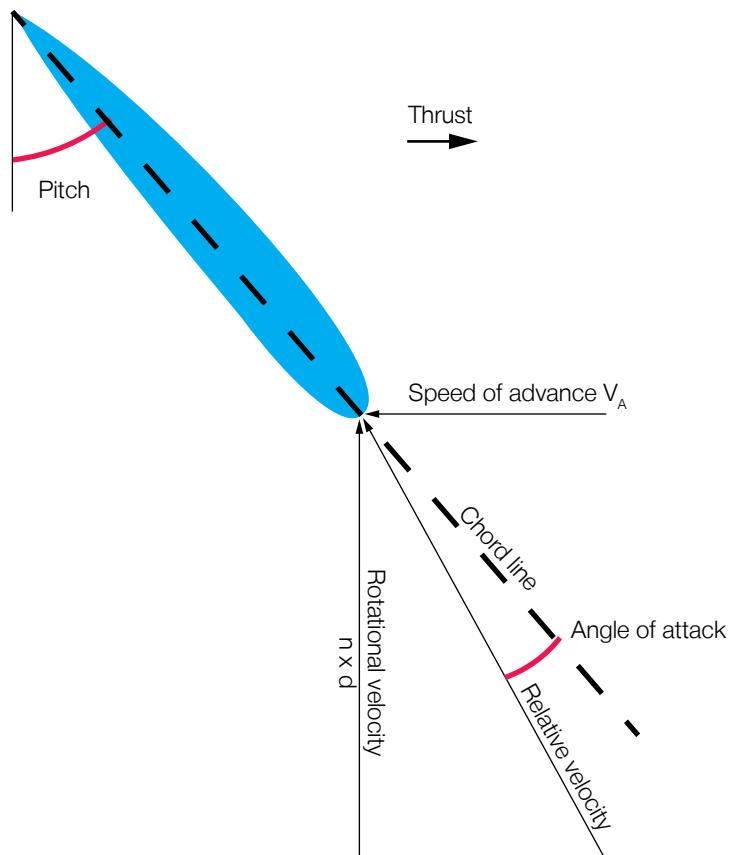


Fig. 2.07: Advance number

Torque coefficient

The propeller torque $Q = P_D / (2\pi n)$ is expressed dimensionless with the help of the torque coefficient K_Q :

$$K_Q = \frac{Q}{\rho \times n^2 \times d^5}$$

Thrust loading coefficient

The thrust loading coefficient describes the loading degree of the propeller, as the pressure of the propeller is related

to the dynamic pressure of the incoming water flow to the propeller, i.e. thrust is divided by the speed of advance and diameter squared:

$$C_{th} = \frac{p_{propeller}}{p_{dynamic}} = \frac{\frac{\pi}{4} d^2}{\frac{1}{2} \rho V_A^2} = \frac{T}{\frac{1}{2} \rho V_A \frac{\pi}{4} d^2} = \frac{8K_T}{\pi J^2}$$

Slip

Sometimes the propeller is referred to as a screw, drawing on its similarity to a screw moving though a solid material. If this was the case, the propeller would move forward at a speed of the pitch times the rate of revolution, $V = p \times n$, see Fig. 2.01.

As water is not a solid material, the water yields (accelerates aft) due to the thrust of the propeller and therefore the propellers speed forward decreases compared to the non-yielding case. This difference is termed the slip.

Two terms describe the slip, the apparent based on the speed of the ship, V , and the real based on the speed of the water arriving at the propeller V_A , see Fig. 2.01. Both in principle quantify the ratio between the apparent/real advance per rotation and advance without water yielding:

$$S_A = \frac{p \times n - V}{p \times n} = 1 - \frac{V}{p \times n}$$

$$S_R = 1 - \frac{V_A}{p \times n} = 1 - \frac{V \times (1-w)}{p \times n}$$

The apparent slip can be calculated by the crew in order to provide an indication of the propeller load. Typically, the real slip cannot be calculated by the crew, but as the speed of advance V_A is smaller than V , it will be greater. The load is proportional to the slip on the propeller, i.e. the higher the slip, the higher the load and vice versa.

This implies that under increased resistance, the rate of revolution of the propeller must be increased to keep a constant ship speed. Due to the load limits of the main engine, this is only possible to a certain extent, and in severe weather the rpm of a FP-propeller will drop, limiting the possible power output of the main engine, see "Propulsion margins, including light running margin" in Chapter 3. Depending on the capabilities of the control system, the pitch

can be lowered on a CP-propeller to enable the engine to maintain its rpm in severe conditions.

When performing bollard pull, i.e. $V = 0$ and high thrust, the slip will be equal to one. All the water will yield (accelerate aft) as all the thrust delivered by the propeller will be used to accelerate the water and not the ship.

Remembering that the advance number is $J = V_A / (J \times d)$, the slip is considered in the advance number through the inclusion of V_A .

Cavitation

Cavitation occurs when the local pressure of the fluid drops below the vapour pressure of the fluid. On marine propellers, sheet cavitation occurs near the nose of the blade section and bubble cavitation on the back. As long as cavitation is limited, it does not reduce the thrust and torque that can be delivered. Cavitation can in some cases cause erosion if the vapour phase/cavitation bubbles implode on any surface of the propeller or rudder.

Cavitation is one of the main contributors to vibration and noise on-board ships. This is a particular nuisance for the comfort of most ships and, especially, if the accommodation is located close to the propeller. A well-designed propeller will achieve a good balance between the needs for a high efficiency and an acceptable cavitation performance.

Efficiencies and influencing parameters

The fuel equation in Chapter 1 describes the combined fuel power required to propel the ship. It is strongly dependent on the individual efficiencies of the propulsion plant components, which is discussed in the following, recalling that:

$$\eta_{tot} = \eta_H \times \eta_O \times \eta_R \times \eta_S \times \eta_E$$

Open water efficiency

The propeller efficiency η_o is related to working in open water, i.e. the propeller works in a homogeneous wake field with no hull in front of it. The propeller efficiency depends, especially, on the speed of advance V_A , the thrust force T , the diameter d , and on the design of the propeller, such as the number of blades, disk area ratio, and pitch/diameter ratio, as previously explained.

The ideal, and not reachable efficiency, is given by the thrust loading coefficient:

$$\eta_{o,ideal} = \frac{2}{1 + \sqrt{1+C_{th}}}$$

$$C_{th} = \frac{T}{\frac{1}{2} \times \rho \times V_A^2 \times \frac{\pi}{4} \times d^2}$$

As seen, the lighter that the propeller is loaded (low C_{th}), the higher the ideal efficiency will be. In practice there is a lower limit of load, below which the efficiency will decrease again. This limit is typically not met for practical applications, see specialist literature on propellers.

The two principal means to minimise the load include either to increase the diameter within the limits previously mentioned, or to increase the speed of advance. When working behind the ship, the hull can be optimised to decrease the wake fraction coefficient, hereby increasing V_A , see "Different approaches for optimising the propulsive efficiency" or some devices altering the flow can be installed, see "Energy saving devices".

For a wide beam bulk carrier, with a low V_A , and a relatively small propeller, the load expressed by C_{th} will be high. On the other hand a twin-screw container carrier will have lightly loaded propellers.

The C_{th} -based approach to propeller efficiency given above does not take into account the propeller rpm. There-

fore it is often more convenient to express the propeller efficiency as a function of the advance number J as well as the non-dimensional thrust and torque coefficient explained previously. These are obtainable from various propeller-series, see Fig. 2.08 that provides an example of both the usable power/ efficiency and the losses occurring on the propeller.

$$\eta_o = \frac{P_T}{P_D} = \frac{T \times V_A}{Q \times 2\pi \times n} = \frac{K_T}{K_Q} \times \frac{J}{2\pi}$$

Depending on the propeller loading and design, an efficiency of $\eta_o = 0.55 - 0.70$ is typically attained.

The axial losses account for most of the loss at low J and frictional losses at high J . Rotational losses as well as tip and hub vortex losses (usually small compared to others) constitute the rest, see Ref. [1.2] and [2.2].

Rotative efficiency

The actual velocity of the water flowing to the propeller behind the hull is neither constant nor at right angles to the propeller's disk area, but has a kind of rotational flow described by the wake field of the hull. Therefore, compared with when the propeller is working in open water, the propeller's efficiency is affected by the propeller's relative rotative efficiency, η_R , which can be interpreted as, Ref. [2.2]:

$$\eta_R = \frac{\text{power absorbed in open water at } V_A}{\text{power absorbed in wake behind ship at } V_A}$$

On ships with a single propeller, the rotative efficiency η_R is normally around 1.0 to 1.07. The rotative efficiency η_R on a ship with a conventional hull shape and with two propellers will normally be less, approx. 0.98, whereas for a twin-skeg ship with two propellers, the rotative efficiency η_R will be almost unchanged. In the initial stage of a project, η_R can be set equal to 1.

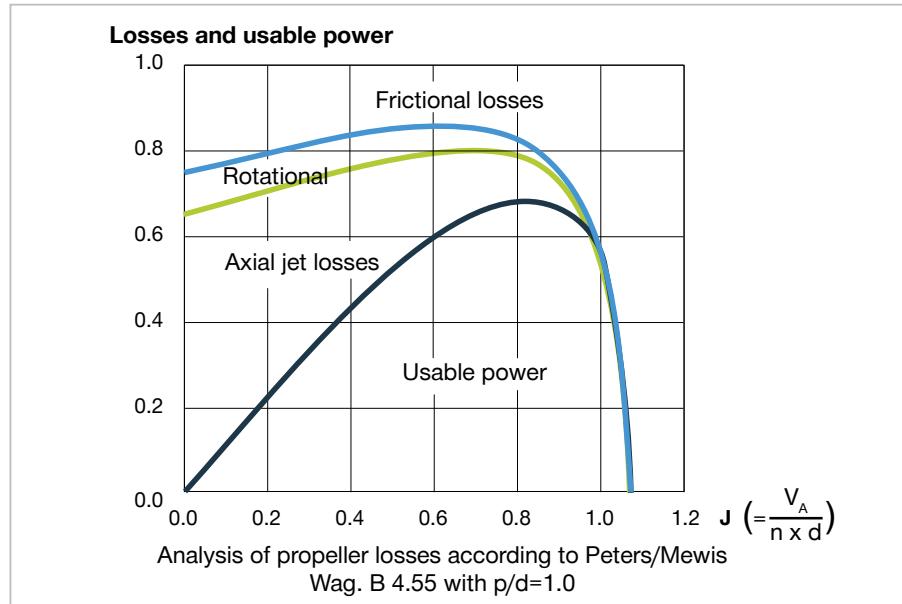


Fig. 2.08: Propeller losses. Note that this is only an example, various combinations of speed and pitch exist in different propeller-series. Courtesy of HSVA, Ref. [2.3]

Efficiency, working behind ship

The term behind efficiency describes the efficiency of the propeller when working behind the ship:

$$\eta_B = \frac{P}{P_D} = \eta_o \times \eta_R$$

Hull efficiency

The hull efficiency η_H is defined as the ratio between the effective (towing) power $P_E = R_T \times V$, and the thrust power delivered to the water by the propeller $P_T = T \times V_A$, i.e.:

$$\eta_H = \frac{P_E}{P_T} = \frac{R_T \times V}{T \times V_A} = \frac{\frac{R_T}{T}}{\frac{V_A}{V}} = \frac{1 - t}{1 - w}$$

This efficiency explains the complex relationship between the thrust deduction coefficient and the wake fraction coefficient.

If the wake fraction coefficient w is larger than the thrust deduction coefficient t , which is normally the case, the hull efficiency will become larger than one, thereby reducing the overall power required to propel the ship. However, reality is far more complex than this single formula indicates, as the parameters affecting the hull efficiency to a large extent will affect the open water efficiency of the propeller.

If the wake fraction coefficient is increased, the speed of advance, or water arriving at the propeller, will be lower (remember that $w = 1 - V_A/V$). This means that the open water efficiency of the propeller will be reduced.

Table 2.02 shows the typical values of the hull efficiency depending on the propeller arrangement. In general, the hull efficiency will be higher for ships

Propeller arrangement	Wake fraction coefficient, w	Thrust deduction coefficient, t	Hull efficiency, η_H
Single-screw	0.20-0.45	0.10-0.30	1.1-1.3
Twin-screw, single skeg	0.10-0.25	0.05-0.15	0.95-1.15
Twin-screw, twin skeg	0.15-0.35	0.05-0.25	1.05-1.25

Table 2.02: Typical approximate propulsion coefficients

with a high block coefficient, creating a large wake fraction coefficient (and large hull resistance).

Propulsive efficiency

The term “propulsive efficiency” describes the combined efficiency of the hull and propeller – excluding the engine and shaft connection:

$$\eta_D = \eta_H \times \eta_O \times \eta_R$$

Traditionally, the influence of the propulsive efficiency, η_D , on the engine efficiency η_E and vice versa has not received much attention, but it is important to stress that there are strong interdependencies, which is why an optimal ship cannot be reached by optimising these independently. The process of matching hull and engine for the overall best efficiency is described in Chapter 3.

Shaft efficiency

The shaft efficiency η_S is not related to the propeller, but is included here for completeness. It depends, among other factors, on the alignment and lubrication of the shaft bearings and on the reduction gear, if installed. It is defined as the ratio between the power delivered to the propeller P_D and the brake power of the main engine P_B :

$$\eta_S = \frac{P_D}{P_B}$$

For directly coupled two-stroke engines with a short shaft $\eta_S \approx 0.99$, for long shafts $\eta_S \approx 0.98$, and if a reduction gear is installed, $\eta_S \approx 0.95$ to 0.96.

Velocities

Ship's speed : V

Arriving water velocity to propeller..... : V_A

(Speed of advance of propeller)

Effective wake velocity : $V_w = V - V_A$

Wake fraction coefficient..... : $w = \frac{V - V_A}{V}$

Forces

Towing resistance : R_T

Thrust force : T

Thrust deduction fraction : $F = T - R_T$

Thrust deduction coefficient : $t = \frac{T - R_T}{T}$

Power

Effective (Towing) power : $P_E = R_T \times V$

Thrust power delivered

by the propeller to water : $P_T = P_E / \eta_H$

Power delivered to propeller : $P_D = P_T / \eta_B$

Brake power of main engine : $P_B = P_D / \eta_S$

Efficiencies

Hull efficiency : $\eta_H = \frac{1 - t}{1 - w}$

Propeller efficiency - open water : η_O

Relative rotative efficiency : η_R

Shaft efficiency : η_S

Engine efficiency : η_E

Propeller efficiency - behind hull : $\eta_B = \eta_O \times \eta_R$

Propulsive efficiency : $\eta_D = \eta_H \times \eta_B$

Total propulsive efficiency : η_{prop}

$$\eta_{prop} = \frac{P_E}{P_B} = \frac{P_E}{P_T} \times \frac{P_T}{P_D} \times \frac{P_D}{P_B} = \eta_H \times \eta_B \times \eta_S = \eta_H \times \eta_O \times \eta_R \times \eta_S$$

Total efficiency : η_{tot}

$$\eta_{tot} = \eta_{prop} \times \eta_E = \eta_H \times \eta_O \times \eta_R \times \eta_S \times \eta_E$$

Influence of propeller diameter and pitch/diameter ratio example

As already mentioned, the highest possible propulsive efficiency at a given ship speed is obtained with the largest possible propeller diameter d , in combination with the corresponding, optimum pitch/diameter ratio p/d .

As an example for an 80,000 dwt crude oil carrier, with a service ship speed of 14.5 knots and a propeller diameter of 7.2 m, this influence is shown in Fig. 2.09.

According to the blue curve, the maximum possible propeller diameter of 7.2 m may have the optimum pitch/diameter ratio of 0.70, and the lowest possible shaft power of 8,820 kW at 100 rpm. If the pitch for this diameter is changed, the propulsive efficiency will be reduced, i.e. the necessary shaft power will increase, see the red curve.

The blue curve shows that if a bigger propeller diameter of 7.4 m is possible, the necessary shaft power will be reduced to 8,690 kW at 94 rpm, i.e. the bigger the propeller, the lower the optimum propeller speed.

For the same engine the bigger propeller though reduce the extent to which the engine can be derated. This influence on the engine efficiency is sometimes not considered when designing the propulsion plant and, depending on the specific project, it can be well worth sacrificing 0.5% of propeller efficiency for gaining a larger increase in engine efficiency, see Chapter 3.

Depending on the hull geometry, the maximum propeller diameter is sometimes not the limiting factor. For some reason, there might be an engine limitation with regard to the lowest possible rpm instead. In such cases, as the red curve shows, it may still be worth applying the largest possible propeller, and instead decrease the pitch in order to increase engine rpm. The benefit of the larger propeller will typically be significantly larger than the small decrease in efficiency from the lower pitch.

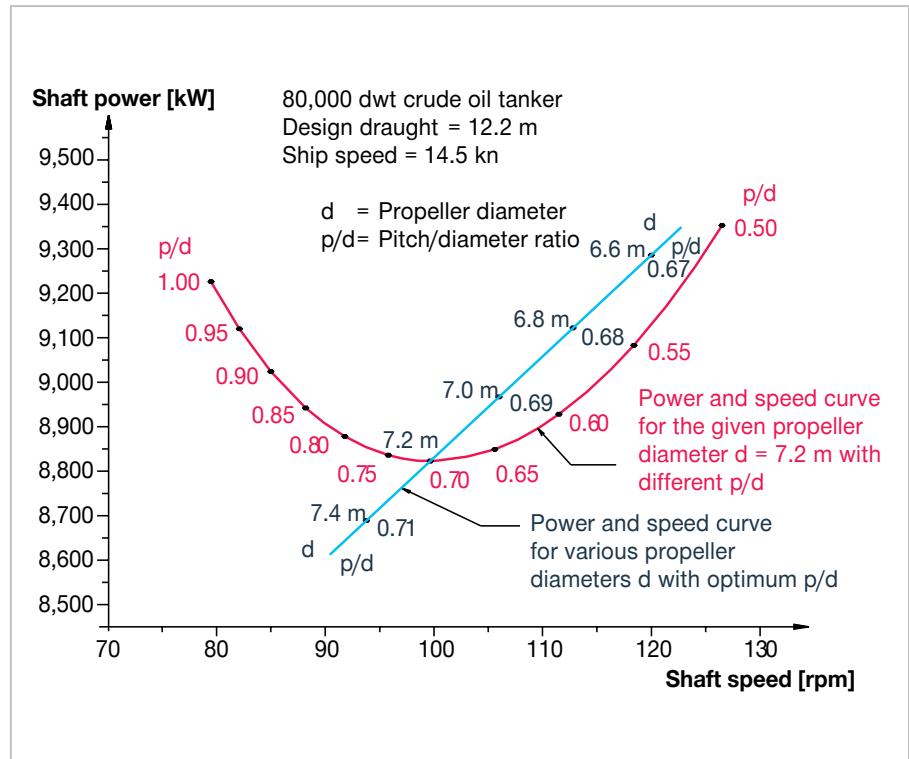


Fig. 2.09: Example of influence of diameter and pitch

Different approaches for optimising the propulsive efficiency

As described above, many different aspects affect the overall efficiency of the propulsion plant (including the engine). These aspects may also affect the hull resistance. Therefore, the designer must find a balance, and as great variety exists among ship types and designers, one size does not fit all, and different approaches can lead to the same combined efficiency.

A short example of such considerations can be given for a large $\approx 20,000$ teu container ship, where different designs exist with one and two propellers, typically twin-skeg. A single-screw ship will offer the best efficiency of the engine as the engine is larger. Furthermore, the single-screw ship will have the smallest wetted surface area, hereby reducing the frictional hull resistance.

On the other hand, the propeller of the single-screw ship will be higher loaded than on the twin-screw ship, which in itself decreases its efficiency. Furthermore a high disk area coefficient and hereby greater blade number must be employed in order to avoid cavitation, further decreasing the efficiency of the propeller for the single-screw ship.

In this case, the greatly improved propeller efficiency of a twin-skeg ship will not only level out the other losses, but usually lead to an overall increase in propulsive efficiency.

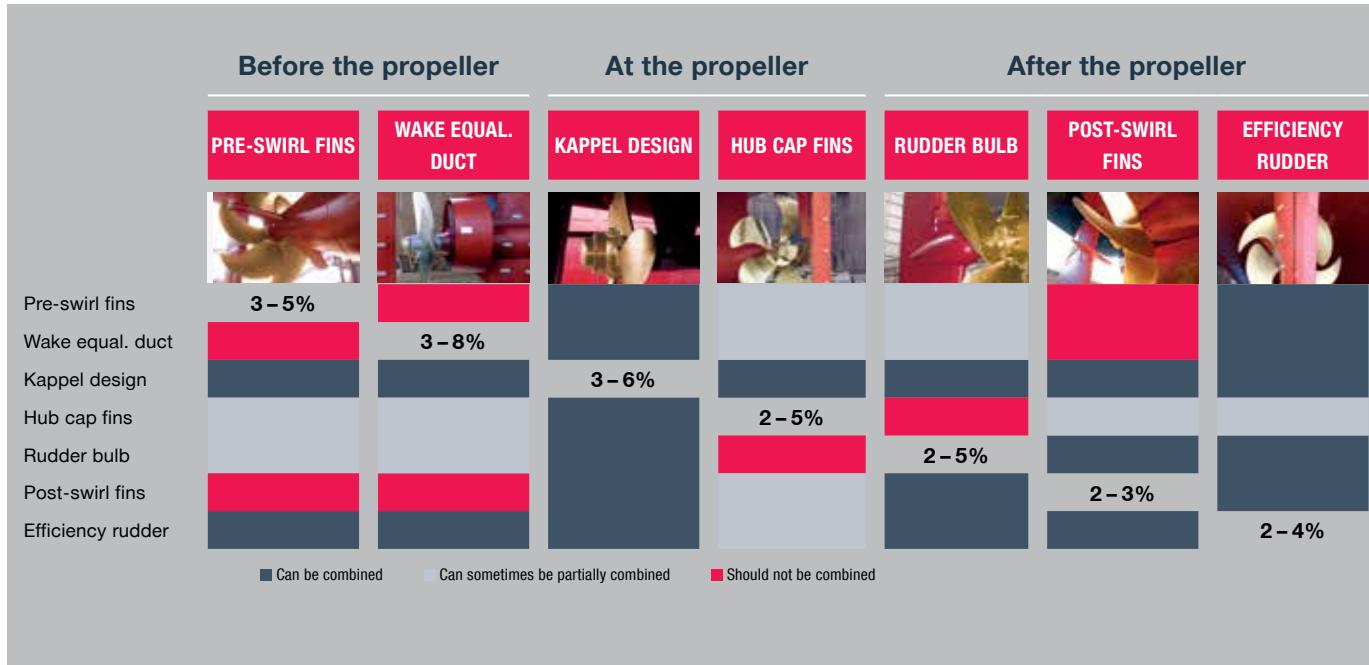


Fig. 2.10: Possibilities for combining energy saving devices, including individual saving potential

Energy saving devices

Energy saving devices is a term covering different devices designed for optimising the flow to, around or after the propeller. The devices are primarily designed to alter the wake field or eliminate the losses arising on the propeller, see "Open water efficiency". Table 2.04 categorises different devices according to their working principle.

Fig. 2.10, illustrates the savings that can be gained by employing the devices individually. It is important to note that savings cannot just be added. An individual analysis of the effect in the specific wake of the hull must be performed. Experience has shown that it is unlikely to achieve combined savings of more than 10% compared to a standard design.

When choosing the optimum energy saving device (or a combination hereof), the operating conditions of the ship must be considered.

For instance, the Kappel propeller provides the largest savings for highly loaded (high C_{th}) propellers. The wake equalising duct (and similar duct systems) works best for ships with large block coefficients, as it equalises and reduces the wake fraction coefficient.

Working principle, reducing	Device	Saving potential
Rotational losses	Pre-swirl fins	3-5%
	Twisted rudder	1-2%
	Contra rotating propellers	4-7%
Rotational losses and separations in the aft body	Wake equalising duct	3-8%
Hub vortex losses	Efficiency rudders	2-6%
	Rudder bulb	2-5%
	Hub cap fins	2-5%
Tip vortex losses	Kappel propeller	3-6%

Table 2.04: Working principles of various energy saving devices including individual saving potential, Ref. [2.4]

Category	Propeller	Length, L_{PP}	Breadth, B	Draught, T_d	C_B	V, kn (=n)	V, kn (=n)	V, kn (=n)	V, kn (=n)
Tanker, product	1 FP	174	32.2	11.0	0.78	13 (3.2)	14 (3.4)	15 (3.6)	16 (3.8)
Bulk carrier	1 FP	273	46.0	16.5	0.83	13 (3.0)	14 (3.1)	15 (3.3)	16 (3.6)
Container ship	1 FP	375	59	16.0	0.68	21 (3.2)	22 (3.3)	23 (3.3)	24 (3.4)
Container ship	2 FP, twin skeg	375	59	16.0	0.68	21 (3.1)	22 (3.2)	23 (3.2)	24 (3.3)
Ro-pax	2 CP, twin skeg	200	31.8	7.0	0.61	21 (3.4)	22 (3.7)	23 (4.1)	24 (4.8)

Table 2.05: Example ships with exponents to the power curve, $P = k \times V^n$, depending on the speed of the ship. The approximate exponent of the power equation corresponding to the speed is stated in parentheses, Ref. [2.5]

Contra-rotating propellers have high costs, not only due to the cost of two propellers, but also the complex mechanical construction required for powering both propellers. In addition, careful maintenance is required.

In general, hub vortex reducing measures (see Table 2.04) provide a rather simple, maintenance-free (other than cleaning) solution and are amongst the most popular. These factors also apply for pre-swirl fins and twisted rudders.

For all energy saving devices, their potential savings must always be evaluated against the added wetted surface and the resistance created by its presence. This is accounted for in Fig. 2.10. Additionally, when evaluating stated saving potentials for specific devices, the designer must take care whether the savings are attained at model or full scale, as scaling effects can occur.

Propeller law and power/speed curves

As explained in Chapter 1, the resistance of a ship can be expected to increase by the square of the speed, $R = \frac{1}{2} \times \rho \times A_s \times C \times V^2$ at lower Froude numbers

$$F_n = \frac{V}{\sqrt{g \times L_{WL}}}$$

where the resistance is purely frictional. As power is the force of resistance times speed, $P = V \times R$, this results in a

required power and hereby fuel consumption proportional to the cubic of the speed, $P \propto k \times V^3$.

This proportionality is termed the Propeller law, and despite it being termed a law that the required power is proportional to the cubic of the speed, it is only an assumption applicable at lower Froude numbers. At elevated Froude numbers, where the wave making resistance must be taken into account, the required power can be proportional to the speed to a power of four or even larger, $P \propto k \times V^4$. This is exemplified in Table 2.05 for different ships.

Explanations regarding the concept of a heavy running propeller and light running margin in Chapter 3 will be based on the propeller law, $P \propto V^3$.

Acceleration, barred speed range, manoeuvring speed and propeller rotation

Acceleration is an important parameter for the manoeuvring performance of the ship. In addition, there is a link between acceleration performance and ship performance under increased resistance: If the ship can produce a high thrust force for acceleration at low ship speed, it can also produce a high thrust force for overcoming high resistance at low speed in heavy weather conditions.

Two important aspects of acceleration are acceleration of the ship itself, and shaft acceleration past a barred speed range. The bollard pull propeller curve

is introduced before the discussion of shaft acceleration past a barred speed range (BSR).

Application of a CP-propeller will provide more flexibility with respect to achieving good acceleration performance. This is not treated further in this section, see the engine selection spiral for CP-propellers in Chapter 3.

Ship acceleration

Ship acceleration capability is mostly governed by the power to displacement ratio of the ship, P_B / Δ . In other words, how much power there is to push a ship with a certain displacement.

The trend towards increasingly larger ships is rooted in the fact that a large ship, compared to a smaller one, has less wetted surface relative to the cargo capacity, and thereby less resistance and fuel consumption relative to cargo capacity. Therefore, the large ship will be equipped with less engine power relative to its displacement than the small ship. As a result, the large ship will accelerate slower than the small ship.

Focus on lowering the fuel consumption and still tighter EEDI requirements may result in ships with quite low power-to-displacement ratios compared to earlier designs. It must be assured in the design phase that a ship has sufficient power for safe manoeuvring in all relevant conditions. The IMO Minimum Propulsion Power requirements address this issue, see Chapter 4.

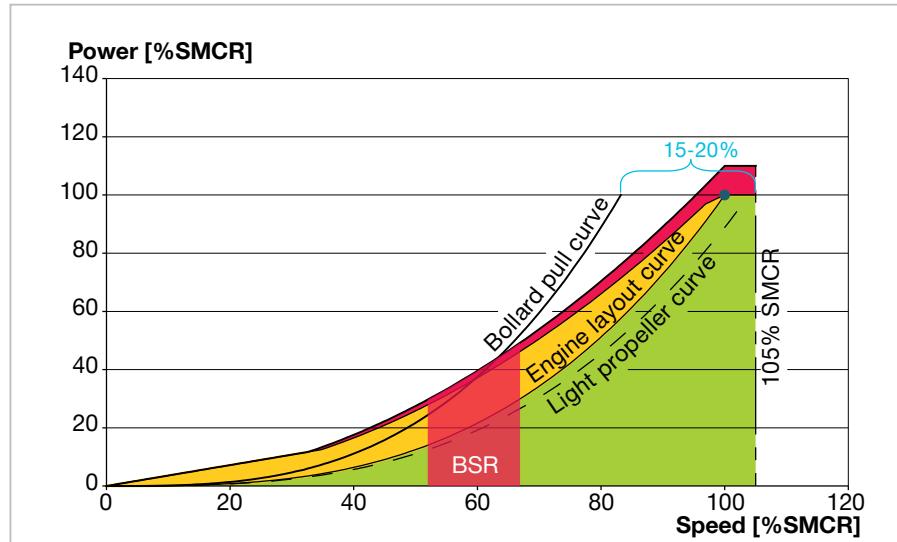


Fig. 2.11: Bollard pull curve. The two-stroke engine can always accelerate a propeller to about 50% rpm quickly. The BSR in the figure is placed high up in the rpm range and BSR passage may not be quick.

Bollard pull propeller curve for FPP

Fig. 2.11 shows the bollard pull propeller curve and the normal light propeller curve as a function of propeller rpm, which for a directly coupled propeller equals engine rpm. At the start of acceleration, from zero ship speed, the propeller will be heavy running and operating along the bollard pull curve.

The limits of the main engine is illustrated on Fig. 2.11 and is described in Chapter 3. It is clear that due to the heavy running of the propeller in bollard pull, 100% rpm cannot be reached in the bollard pull condition due to the limit of the engine power curve.

The exact value of heavy running in bollard pull condition is difficult to predict at the ship design stage. In general, a heavy running factor of 15-20% relative to the light propeller curve have been experienced, i.e. an rpm reduction of 15-20% for the same power. This is equivalent to approx. a 60-100% increase in power for the same rpm.

Considering Fig. 2.11, it is brought to the reader's attention that an engine with an FP-propeller can always accelerate to approx. 50% rpm along the bollard pull curve. At low rpm, the propeller curve is well within the limits of the engine diagram.

Shaft acceleration past a barred speed range for FPP

A barred speed range imposed by vibrations must be passed sufficiently quick in order not to damage the shafting due to vibrations resulting in excessive stresses.

What is meant by "sufficiently quick" depends on how high the stresses in the shaft are compared to the strength of the shaft material.

Furthermore, the definition of "sufficiently quick" depends on how often the barred speed range will be passed during the expected lifetime of the ship. For example, a feeder containership with many port calls will pass the barred speed range more frequently than a large crude carrier that mostly performs ocean crossings.

In general, the barred speed range must be passed within seconds, not minutes.

The acceleration of the shaft line is governed by the difference in power delivered by the main engine and the hydrodynamic power required by the propeller. As long as the power provided by the engine is higher than the power required by the propeller, the shaft line will accelerate.

During acceleration from ship standstill, the propeller curve is heavy running along the "bollard pull" propeller curve. This is comparable to a car going uphill at the same speed in the same gear as when driving on a flat highway. Thus, going uphill, the load on the engine will be greater as more work must be delivered to lift the mass of the car (in the case of the ship to accelerate the water) even if the rpm is the same.

At some point, the bollard pull propeller curve will demand more power than the engine can deliver at low rpm as full engine power can only be delivered at full engine rpm. As there is no longer an excess of engine power relative to the bollard pull curve, the propeller will accelerate at a slower rate, namely the rate at which the ship is accelerating. It is important to note that the propeller in this case will continue to accelerate with the acceleration of the ship, though not at the maximum rate along the bollard pull curve.

With this acceleration of the ship, the actual propeller curve will gradually shift towards the light propeller curve and, at some point, the actual curve will be outside the barred speed range, as seen on Fig. 2.11. Thereby, the barred speed range is passed, but over longer time, i.e. not necessarily ensuring a sufficiently quick passage of the BSR.

In other words, if the bollard pull curve crosses the power limit of the engine at an rpm below or within the BSR, the BSR may not be passed sufficiently quick.

In order to avoid issues with slow passage of the BSR, the engine power and the propeller characteristics should be matched accordingly, see the engine selection spiral for FP-propeller plants in Chapter 3. Here, the capability to pass the BSR sufficiently quick can be evaluated by means of the barred speed range power margin, BSR_{PM} .

In situations with very high resistance on the ship, such as in very rough weather, it may not be possible to accelerate the ship to speeds much high-

er than a few knots. If the ship does not have sufficient BSR_{PM} , it may not be possible to achieve a propeller speed above the BSR. Hence, the crew will be forced to reduce the rpm setting of the main engine to a value below the BSR, resulting in an even lower ship speed. A sufficient BSR_{PM} is therefore also important in other situations than acceleration.

Manoeuvring speed

Below a certain ship speed, referred to as the manoeuvring speed, the manoeuvrability of a ship will be insufficient due to the too low velocity of the water arriving at the rudder. It is rather difficult to give a general figure for an adequate manoeuvring speed of a ship, but a manoeuvring speed of 3.5 to 4.5 knots is often applied.

Direction of propeller rotation

When a ship is sailing, the characteristics of the wake field means that the propeller blades bite more in their lowermost position than in their uppermost position. The resulting side-thrust effect is larger the more shallow the water is as, for example, during harbour manoeuvres.

Therefore, a clockwise (looking from aft to fore) rotating propeller will tend to push the ship's stern in the starboard direction, i.e. pushing the ship's stem to port, during normal ahead running. This has to be counteracted by the rudder.

When reversing the propeller to astern running as, for example, when berthing alongside the quay, the side-thrust effect is also reversed and becomes further pronounced as the upper part of the propeller's slip stream, which is rotative, strikes the aftbody of the ship. Awareness of this behaviour is very important in critical situations and during harbour manoeuvres, also for the pilot.

It is therefore an unwritten law that on a ship fitted with an FP-propeller, the propeller is always designed for clockwise rotation when sailing ahead. A directly coupled main engine will of course have the same rotation.

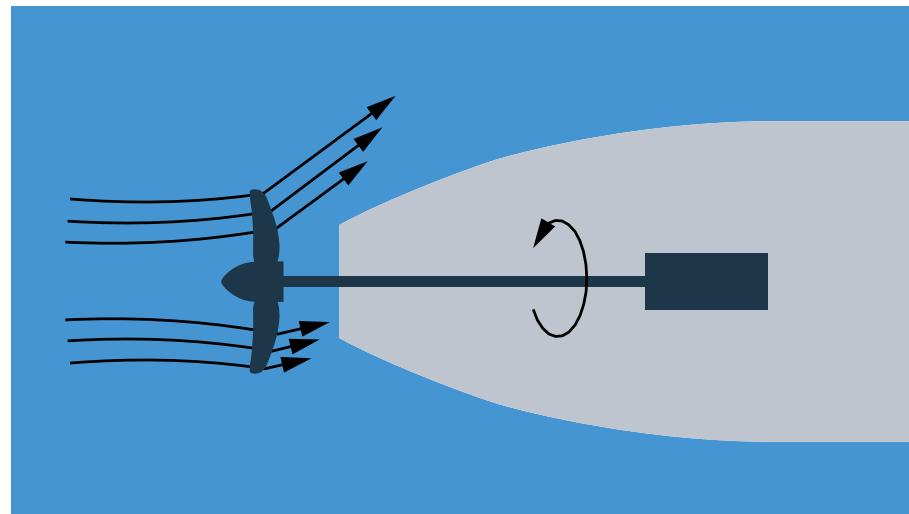


Fig. 2.12: Astern water flow

ISO 484/1 – 1981 (CE)

Class	Manufacturing	Mean pitch
S	Very high accuracy	+/- 0.5%
I	High accuracy	+/- 0.75%
II	Medium accuracy	+/- 1.00%
III	Wide tolerances	+/- 3.00%

Table 2.05: Manufacturing accuracy for propeller

In order to obtain the same side-thrust effect when reversing to astern, on ships fitted with a CP-propeller that is always rotating in the same direction, CP-propellers are set for anti-clockwise rotation when sailing ahead. As the side thrust effect is strongest when going astern, this is the most critical situation, see Fig. 2.12.

Manufacturing accuracy of the propeller

Before the manufacturing of the propeller, the desired accuracy class standard of the propeller must be chosen by the customer. Such a standard is, for example, ISO 484/1 – 1981 (CE), which has four different "Accuracy classes", see Table 2.05. Typically class S is almost only used for naval ships, whereas class I is typically used for merchant ships. Class II and III are rarely used.

The manufacturing accuracy tolerance corresponds to a propeller speed tolerance of max. +/- 1.0%. When also incorporating the influence of the tolerance on the wake field of the hull, the total propeller tolerance on the rate of revolution can be up to +/- 2.0%. This tolerance must also be borne in mind when considering the operating conditions of the propeller in heavy weather, and can explain why sister ships may have different propeller light running margins.

Chapter 3

Engines for marine propulsion plants

This chapter explains the basic working principles of a two-stroke crosshead engine, the parameters affecting its efficiency, the concepts for using alternative fuels, and how the loading of the propeller influences the engine layout and running conditions.

An engine selection spiral is introduced and provided in two editions, one for fixed pitch propellers and one for controllable pitch propellers. The engine selection spiral describes the method for selecting the optimum engine depending on the priorities of the project.

This chapter explains the basic principles. For specific calculations for two-stroke engines, we recommend the online CEAS tool available from the MAN Energy Solutions website → Two-Stroke → CEAS Engine Calculations.

Two-stroke crosshead diesel cycle engines

Diesel cycle engines are characterised by the direct injection of fuel into the combustion chamber. The fuel is ignited by the high temperatures arising from the large mechanical compression of the air prior to fuel injection. For further details, see Ref. [4.1].

The power delivered from an engine depends on the torque it can develop at a given rotational speed (angular velocity, ω) and the rotational speed itself:

$$P = T \times \omega = T \times 2 \pi \times \frac{\text{rpm}}{60}$$

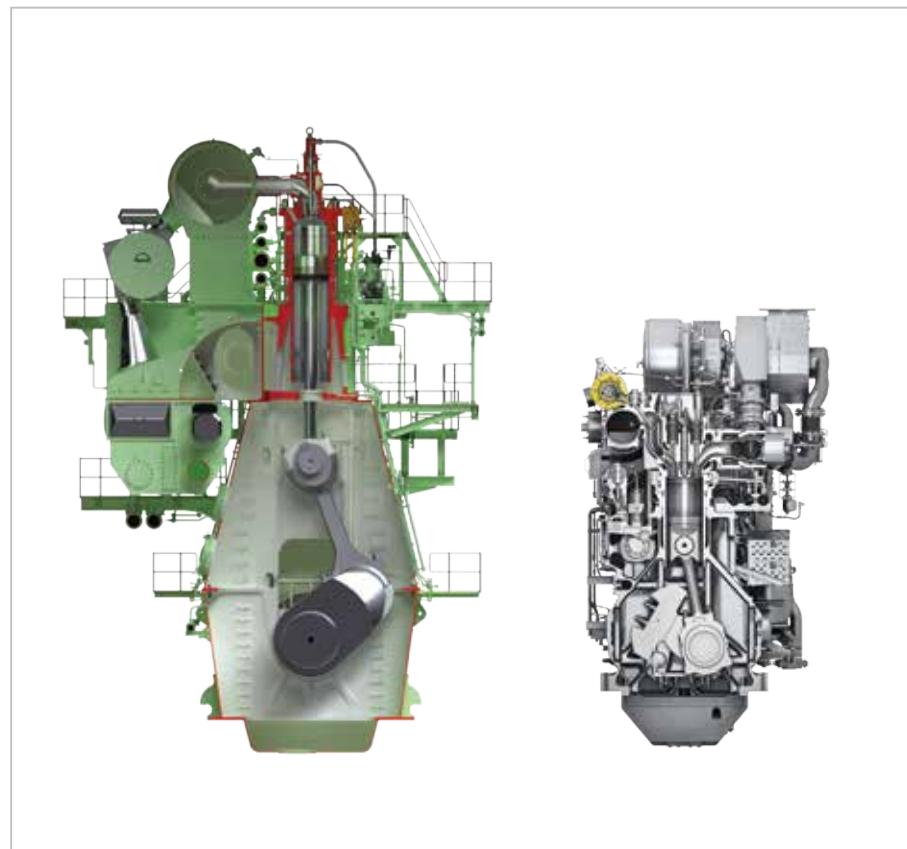


Fig. 3.01: Two-stroke crosshead engine and four-stroke trunk engine, not to scale

Diesel cycle engines may work according to the two-stroke or the four-stroke principle. Both principles are utilised for mechanical propulsion of ships, as shown in Table 1.01 (p. 9) in Chapter 1. Medium speed four-stroke engines have a higher power density than low speed two-stroke engines and requires less engine room height, but low speed two-stroke crosshead engines offers superior fuel economy.

The principle of the crosshead is illustrated in Fig. 3.01. The crosshead allows a significantly larger stroke/bore ratio compared to its counterpart, the

trunk engine, as the vertical-only movement of the piston rod allows it to be as long as required, without interfering with the cylinder liner.

The long stroke leads to both high efficiency, as well as a low rpm, making it possible to couple the engine and propeller directly. This eliminates the complexity and losses in a reduction gearbox, and ensures further savings.

Performing a combustion each time the piston is at top also increases efficiency, as the relative frictional loss is reduced.

Electronically controlled engines

Ever since the first engine was built and up until the turn of the century, marine diesel cycle engines were mechanically controlled. This meant that the timing of the fuel injection and the opening of the exhaust valves were controlled by a camshaft. Today, most new engines are electronically controlled.

The camshaft-less ME engine enables optimising of combustion parameters such as injection timing, injection pressure, and shaping of the profile hereof as well as the timing of the exhaust valve actuation. The fuel oil injection pressure is decoupled from the engine rpm which allows the pressure to be high for the whole load range, ensuring optimal efficiency and low soot formation.

On ME engines, the air amount trapped in the cylinder, and later compressed, is less dependent on the scavenge air pressure, as the closing time of the exhaust valve can be varied. This ensures a better acceleration performance (during which the scavenge air pressure is lower) compared to MC engines.

Engine efficiency parameters

The efficiency of a two-stroke diesel cycle engine is affected by many parameters, but a few main parameters important for an initial comparison of engines can be identified:

- Compression ratio:

The higher the compression ratio, or more importantly the corresponding high expansion ratio, the higher the end-temperature of the air in the cylinder when compressed. This ensures that the heat from the combustion is released at a high temperature level, and that a high degree of expansion of the hot gasses is possible before the exhaust valve needs to be opened. It is the expansion of the hot gasses that produces the power, and the high compression/expansion ratio of the diesel cycle engine is therefore a main contributor to the high efficiency of the diesel cycle engine.

- Cylinder bore:

For the same stroke/bore ratio, the larger the bore, the larger the engine. A larger bore will reduce the relative heat loss, as the surface through which heat can escape will be relatively smaller compared to the combustion volume. This improves engine efficiency.

- Stroke/bore ratio:

For a uniflow two-stroke engine, a high stroke/bore ratio improves the efficiency of the scavenging process and thereby the efficiency of the engine.

As mentioned previously, the stroke length has an influence on the rate of revolution. Typically, the biggest improvement to ship efficiency by increasing the stroke length, is not from the engine itself. Instead the increased efficiency of the larger propeller that can be applied with the resulting lower rpm brings the largest improvement, see Chapter 2.

GI and LGI dual fuel engines

Engines capable of operating on traditional bunker oil and alternative fuels are termed dual fuel engines. At present, liquefied natural gas (LNG), which primarily consists of methane, is the dominant alternative fuel, but ethane, methanol, and liquefied petroleum gas (LPG) concepts are also available. LNG and ethane are injected into the combustion chamber in gas form, giving rise to the GI (gas injection) engine type affix. LPG and methanol are injected in liquid form, giving the type affix LGI (liquid gas injection).

In general, these fuels can be characterised as low-flashpoint fuels, as they have a low carbon-to-hydrogen ratio. The low carbon content of these fuels is also what makes alternative fuels interesting climate-wise, as CO₂ emissions are also reduced hereby, see Chapter 4.

Due to its current dominance as an alternative fuel, the principle of an LNG dual fuel engine will be the base for explaining the two existing concepts for combustion of low-flashpoint fuels, see Fig. 3.02 as well:

- The diesel cycle:

Fuel is ignited and burned in a diffusion flame when injected into the combustion chamber. Therefore, this is also termed a compression ignition engine.

- The otto cycle:

Air and fuel are premixed, for marine applications either in or outside the cylinder. The mixture is then compressed and later ignited by a release of heat, e.g. a spark in a petrol car engine, or pilot fuel oil in a marine engine.

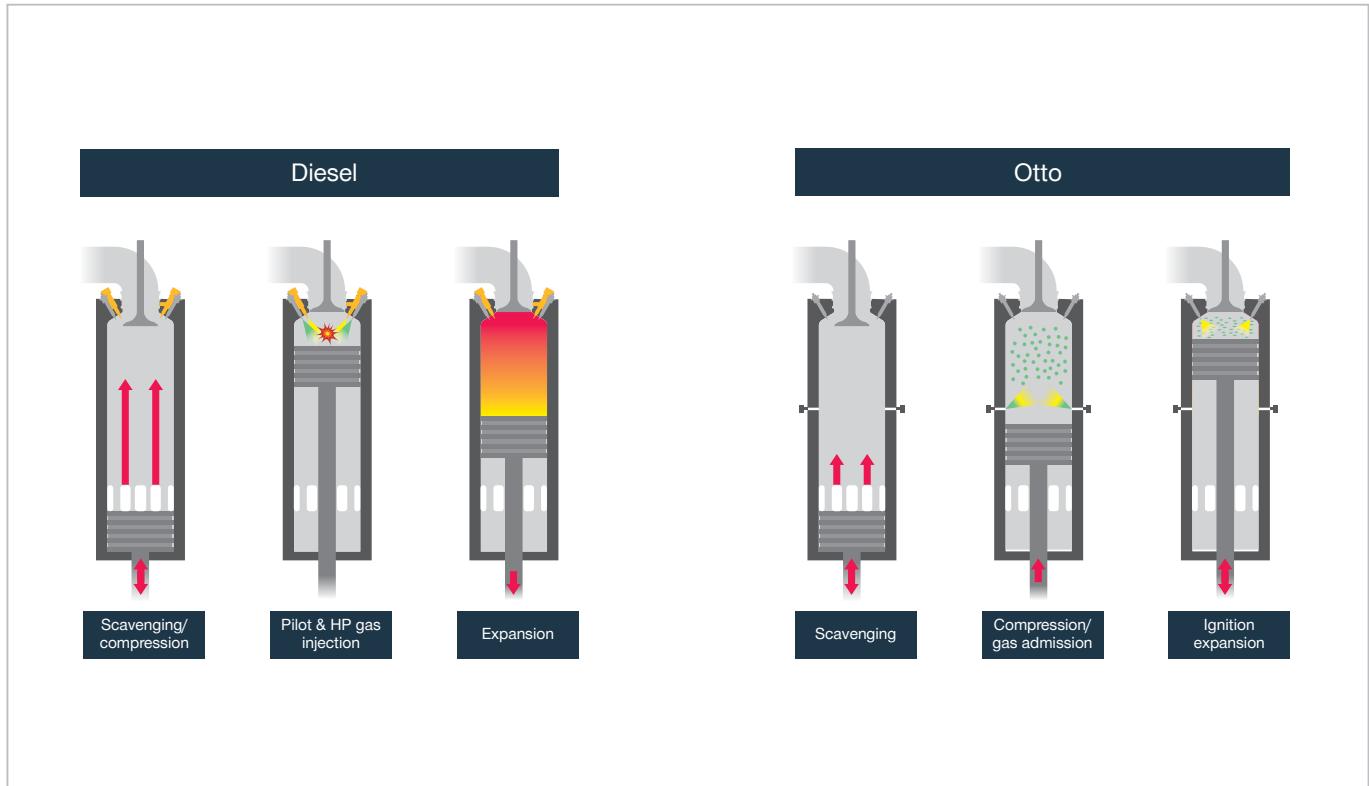


Fig. 3.02: Diesel and otto cycles for dual fuel engines

The diesel and the otto cycles are both available for two-stroke crosshead engines. The diesel cycle offers a series of advantages:

- The diesel cycle is the most efficient operating principle
- Methane slip is very limited as the injected gas is burned directly. In an otto cycle engine, the total volume of the combustion chamber is not included in the combustion. Hereby not all of the fuel/air mixture is burned, which result in significantly higher amounts of unburned methane that escape to the atmosphere. This is important as methane has a global warming potential that is 28-36 times as high as CO₂, Ref. [3.2].
- Any quality of gas can be burned in a diesel cycle engine. As the injected gas is burned directly, the methane number of the gas, indicating its resistance towards engine knocking (premature ignition), is of no signifi-

cance. For the same reason, acceleration performance and acceptance of external load variation, such as varying propeller load as a result of heavy sea are unaffected, because knocking is not an issue.

- No reduction of compression ratio. To avoid knocking, the compression ratio of an otto cycle engine is lower compared to a diesel cycle engine, resulting in a lower efficiency of an otto cycle engine when operating in traditional diesel cycle mode. The constant compression ratio allows the diesel cycle engine to operate with any mixing ratio of traditional fuel and LNG, depending on availability in the market, as well as traditional fuel only.

In otto cycle engines the gas is injected at a pressure of 6 to 20 bar depending on maker and engine load, as the gas is injected prior to the compression, where the pressure in the cylinder is low.

According to the diesel cycle, the gas is injected after the compression. In the ME-GI engine, the gas is injected at 300 bar, in order to overcome the pressure in the chamber. This is achieved by pressurising the LNG to the rated pressure and hereafter vaporise it to gas form prior to injection.

The separate paper “Cost-optimised design of ME-GI fuel gas supply systems” explains in detail how a dual fuel plant for a diesel cycle engine can be outlined.

With regard to layout of the propulsion plant, a dual fuel diesel cycle engine implies no changes to the considerations given in the later engine selection spirals. In general, engines of a larger bore or engines with an additional cylinder must be applied when operating according to the otto cycle. This is due to the lower mep (see p. 45) required to achieve satisfactory operation of this cycle.

Engine selection spiral for FP-propeller

Selecting the right engine for a ship is an important parameter for achieving the lowest possible fuel oil consumption or fulfilling whatever other priorities of the project.

This section will introduce an engine selection spiral describing the process of selecting an engine to be combined with a FP-propeller. Later a separate section will describe the process for a CP-propeller.

The steps in the engine selection spiral in Fig. 3.03 are:

1. Establish the calm water resistance, propeller diameter and propeller working conditions, Chapters 1 and 2
2. Calculate the light propeller curve giving the required power at the design speed
3. Establish the sea margin, engine margin, and the rpm “light running” margin for the project in order to specify the maximum continuous rating (SMCR)
4. Plot SMCR on engine *layout* diagrams for various possible engine types for example by using the online calculation tool CEAS
5. Based on the CEAS results, select the engine depending on the priorities of the project
6. Check for quick passage of the barred speed range (BSR)
7. Check the engine *load* diagram of the selected engine taking into consideration a possible shaft generator/PTO and PTI if desired for the project
8. Check compliance with regulations such as EEDI and Minimum Propulsion Power (MPP), see Chapter 4

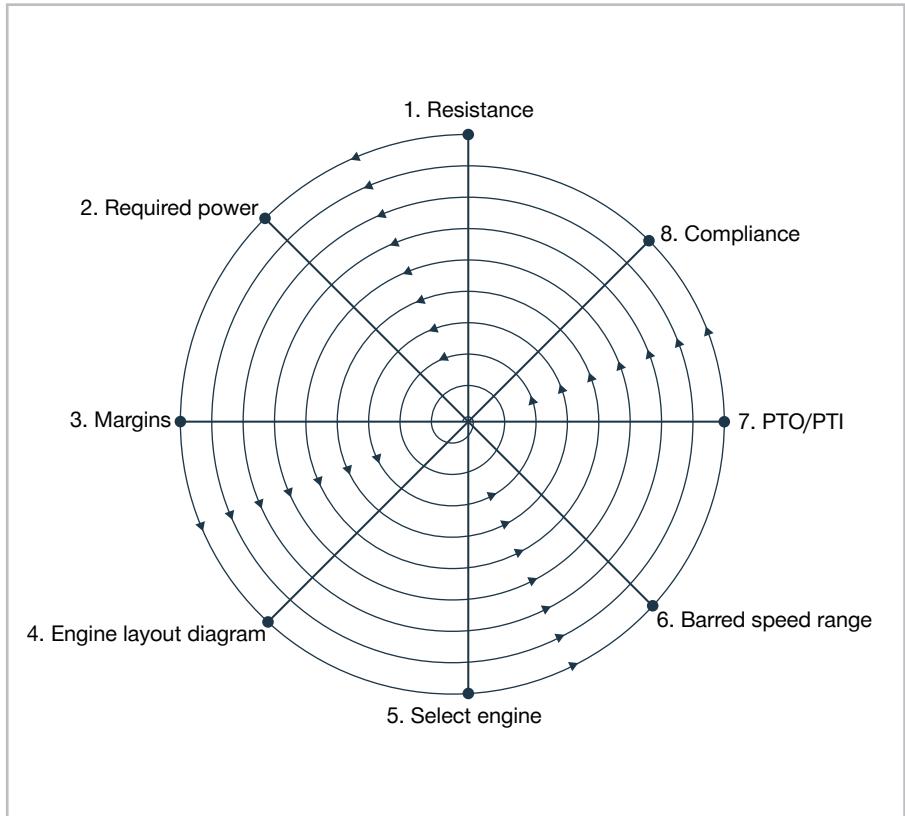


Fig. 3.03: FPP engine selection spiral

In the following sections all steps of the selection spiral will be considered except step 1, as determining the calm water resistance and propeller working conditions constitutes Chapter 1 and 2 respectively. Depending on the outcome of step 6, 7 and 8, it can be necessary to re-enter the selection spiral at either step 1, 3 or 5.

Examples on the use of the engine selection spiral are given in Chapter 5, including the important considerations on SO_x, NO_x, EEDI, and manoeuvring capabilities as described in Chapter 4.

2. Light propeller curve

In the initial project stage, estimations of the necessary propeller power and propeller rpm (for direct coupling equal to engine rpm), are based on theoretical calculations of the calm water resistance for the loaded ship and the propeller working conditions behind the hull.

Resistance calculations are typically validated, and the design is further optimised by experimental towing tank tests, ultimately giving the final propeller curve for the project.

The combination of propeller speed and power obtained on the light propeller curve at the ship's design speed can be termed the propeller design point (PD), see Fig. 3.04.

The influence of the combination of propeller diameter, ship speed and engine power is further exemplified in the later section "Constant ship speed curves".

3. Propulsion margins, including light running margin

Traditionally, first a margin termed the sea margin followed by an engine margin have been added to the propeller design point (PD) along the light propeller curve. This is then shifted left with the light running margin to achieve the engine layout curve, see Fig. 3.04. These margins are explained in the following sections.

Sea margin

The sea margin (SM) is added to account for added resistance from expected average wind and waves as explained in Chapter 1.

A sensible sea margin must be established by the designer for the specific project, operational profile, and area, typically in the range of 10 to 30%. Historically, 15% has been applied, and can be used as a first estimate.

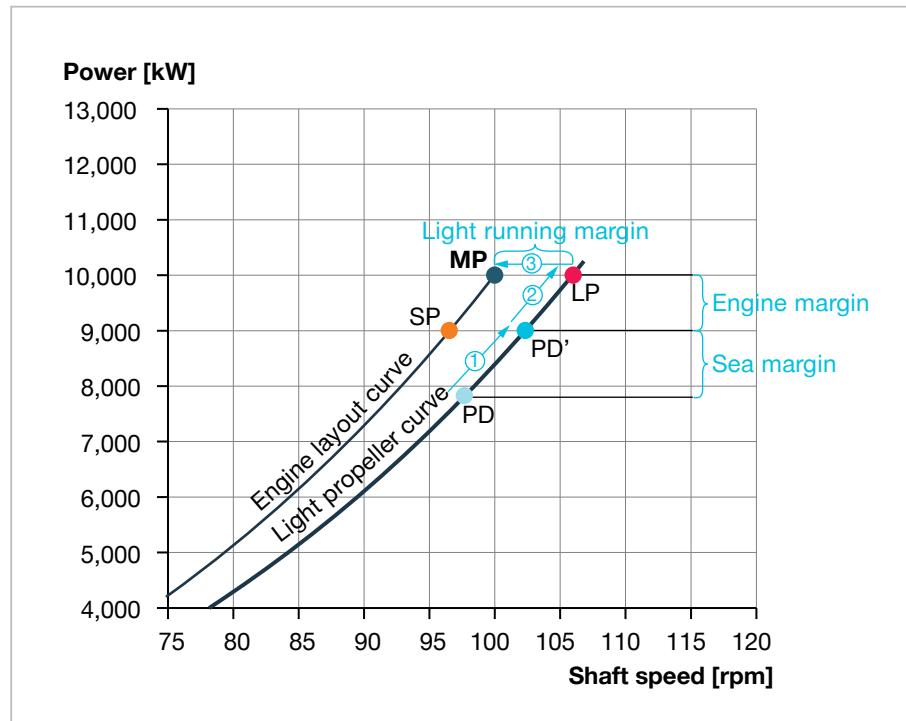


Fig. 3.04: Light propeller curve with added margins resulting in the engine layout curve

It is important to note that often the propeller designer will design the propeller to the operating point including the sea margin, why the propulsion point including sea margin can also be termed the alternative propeller design point (PD'), see Fig. 3.04.

The location on the engine layout curve after having added the sea margin and the later light running margin is termed the service propulsion point (SP).

Engine margin

Often, 100% utilisation of engine power is not desirable for normal operation due to the increased fuel consumption and a desire for a power reserve. Therefore, an engine margin is added. Historically, 10% has been applied, but the engine margin can typically vary between 10 to 30%, depending on the priorities of the project. A high engine margin (EM) is typically preferred for ships in scheduled traffic, in order for them to be able to catch up delays.

Today, the influence of IMO Minimum Propulsion Power Requirements, see Chapter 4, can also lead to higher than traditional engine margins in order to comply with these regulations.

$$SMCR_{power} = PD_{power} \times \frac{\frac{100 + SM}{100}}{\frac{100 - EM}{100}}$$

$$SMCR_{rpm} = PD_{rpm} \times \left(\frac{SMCR_{power}}{PD_{power}} \right)^{\frac{1}{3}}$$

Light running margin (rpm margin)

In the ideal world, the light propeller curve on Fig. 3.04 with its added sea and engine margins would not require further considerations, and the weather, the hull, as well as the propeller would stay in a condition as on sea trial. This is not the case.

A fouled hull will change the wake field as described in Chapter 2 section “Wake fraction coefficient”, reducing the speed of the arriving water and increasing the slip on the propeller, making it more heavy running. Heavy seas will increase the resistance for a series of reasons as described in Chapter 1, section “Added resistance in various conditions”. As a rule of thumb, 20% increase in required power from added resistance corresponds to 1% heavy running, i.e. 1% lower rpm for the same engine power.

There must be a margin to allow for this shift from the nominal, light, propeller curve, n_L , towards a heavier curve, n_H . This margin is termed the light running margin:

$$LRM = \frac{n_L - n_H}{n_H} \times 100\%$$

The curve arising after having added the light running margin to the light propeller curve is termed the engine layout curve, as it will be decisive for the final selection of the engine's specified MCR (SMCR).

Fig. 3.06 illustrates the engine *layout* diagram including sea margin, engine margin and light running margin, giving the engine layout curve with point MP which is the SMCR to be ordered.

MAN Energy Solutions recommends a light running margin of 4 to 7%, depending on the specific ship and oper-

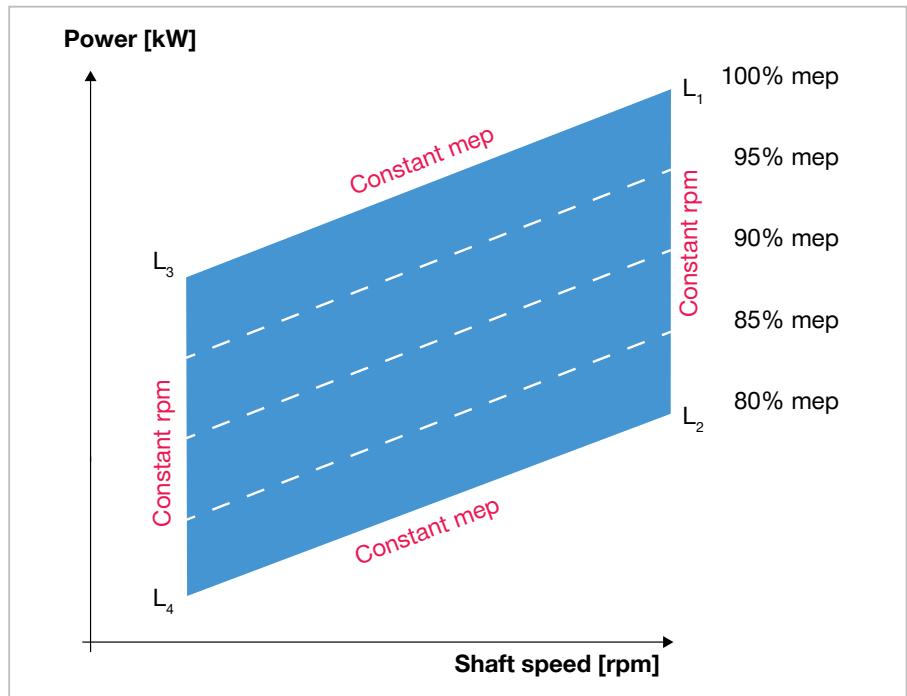


Fig. 3.05: Engine layout diagram with lines of constant mep and rpm

ational profile. In special cases up to 10%. High light running margins are relevant for ships where the expected relative increase in a ship's resistance from fouling, heavy weather, shallow water, ice, etc., is high and may also be relevant if a PTO is applied.

It is the responsibility of the ship designer to select an adequate propeller light running margin so that the desired operating points (power and rpm) in all relevant conditions, will not fall inside a barred speed range (see step 6), nor outside the engine *load* diagram (see step 7).

The recommended use of a relatively high light running margin for design of the propeller will involve that a relatively higher propeller speed will be used for the layout design of the propeller. This, in turn, may involve a minor reduction of the propeller efficiency, and may

possibly cause the propeller manufacturer to abstain from using a high light running margin.

However, this reduction of the propeller efficiency caused by the light running margin is actually relatively insignificant compared with the improved engine performance obtained when sailing in heavy weather and/or with fouled hull and propeller.

If the light running of a specific operational point is to be evaluated, this can be performed by considering the operational point (P, n) relative to the SMCR point (P_{SMCR}, n_{SMCR}):

$$LR = \left(\frac{n}{n_{SMCR} \times \sqrt[3]{\frac{P}{P_{SMCR}}}} - 1 \right) \times 100\%$$

4. Engine layout diagram with SMCR, derating

Often more than one engine can deliver the SMCR calculated for the project. The location of the SMCR within the layout diagram for these engines will provide vital information as to which engine will be the most beneficial. The basic engine layout diagram is shown in Fig. 3.05. The engine can have its SMCR within the limits of L₁-L₂-L₃-L₄.

The lines L₁-L₂ and L₃-L₄ of the layout diagram limit the speed of the engine whereas the lines L₁-L₃ and L₂-L₄ are lines of maximum and minimum mep, respectively. Depending on engine type, the line L₂-L₄ will typically be 75–80% of maximum mep.

Mean effective pressure

The mean effective pressure is an important parameter for understanding the engine layout diagram. It is an expression for how much power (P_B) the engine delivers relative to the displacement volume of the engine (V_d) and rotational speed (n). For two-stroke engines it can be expressed as:

$$\text{mep} \propto \frac{P_B}{V_d \times n}$$

The mean effective pressure can be useful for comparing engines with a different bore, and indicate how hard an engine is loaded.

Maximum continuous rating

The maximum continuous rating (MCR) is the maximum power the engine can continuously deliver.

Nominal maximum continuous rating

The L₁ point designates the engine's nominal maximum continuous rating (NMCR) and is the maximum power the engine type can deliver (depending on the number of cylinders).

Specified maximum continuous rating

Within the layout diagram there are no restrictions to the location of the SMCR, point MP in Fig. 3.06.

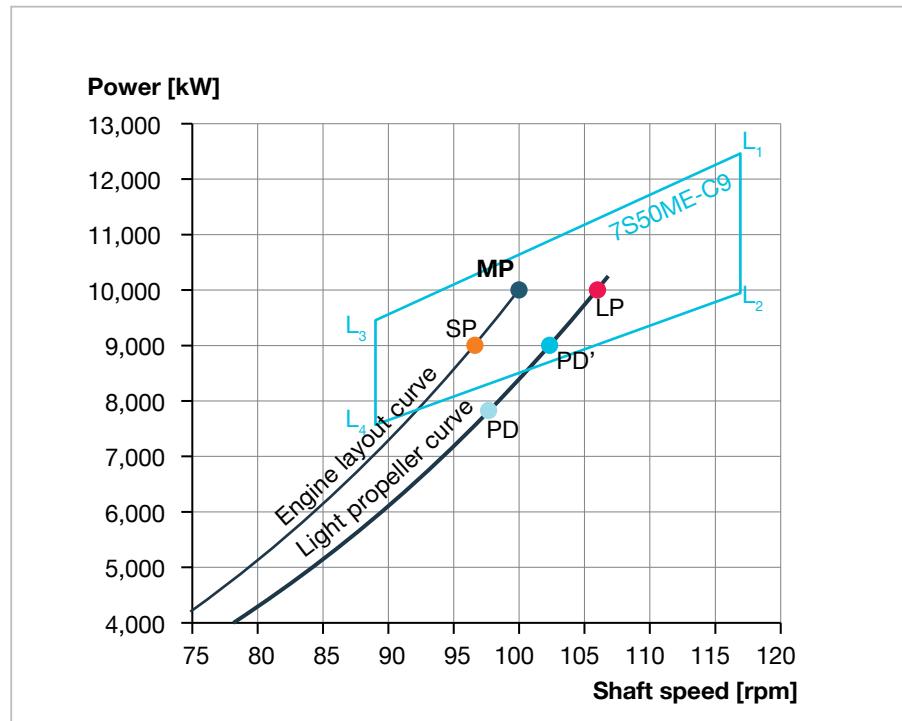


Fig. 3.06: Engine layout diagram with light propeller curve and engine layout curve

The location of SMCR within the diagram will influence the fuel oil consumption of the engine as it influences the degree of derating.

Normal continuous rating

The normal continuous rating, S, is the power needed in service – including the specified sea margin and light running margin of the propeller – at which the engine is to operate. Point S is identical to the service propulsion point (SP) unless a main engine-driven shaft generator is installed, see step 7.

As the engine margin is not included in this figure, this point is sometimes also referred to as the continuous service rating.

Engine mep-derating for reduced SFOC

A mep-derated engine, typically just termed derated, has its mep reduced relative to the maximum mep limited by the line L₁-L₃. A derated engine maintains its maximum cylinder pressure during combustion, but with an mep at SMCR, which is lower than the max. mep, fuel consumption is decreased.

It is important to note that derating is about reducing the mep, which is constant with lines parallel to L₁-L₃ and L₂-L₄. Moving the SMCR parallel with the lines L₁-L₃ and L₂-L₄ will therefore not bring any derating and associated reduction in fuel consumption.

Due to requirements of ship speed and possibly shaft generator power output, derating is often not achieved by reducing the SMCR power of an engine. Instead a larger engine is applied in order to be able to choose a lower mep rating with the same SMCR, for example an engine of the same type but with an extra cylinder, see Fig. 3.11.

Derating of an engine to a lower mep implies a series of physical changes to the engine. Shims are inserted under the piston to increase the compression ratio and, hence, the expansion ratio. The turbocharger is matched to the amount of exhaust gas generated. Fuel valve nozzles, cylinder liner cooling and lubrication, shafting system, and pump and cooler capacities are dimensioned for the SMCR.

Due to the mechanical changes implied by derating, an engine with an SMCR lower than the NMCR cannot operate at NMCR in service.

It can be an option to design the ship for a derated engine but with auxiliaries such as coolers, pumps and pipe dimensions, shafting etc., that are sufficient for a later uprating of the engine – provided that the EEDI regulations allow it. This is termed a dual rated engine. In such a case, and to avoid the more expensive in-ship testing of the engine, it is beneficial to perform the necessary testing to get the IMO technical file for the alternative SMCR, during shop testing of the engine before delivery.

CEAS

The online calculation tool CEAS, see Fig. 3.07, can be accessed from the MAN Energy Solutions website → Two-Stroke → CEAS Engine Calculations. CEAS can be used for plotting the calculated SMCR within the *layout* diagram of every engine design and cylinder number hereof present in the engine programme. Furthermore a range of options, including fuel type(s), emission reduction equipment and possible turbocharger combinations, are displayed and can be selected.

An extensive report containing output about fuel consumption, capacity of auxiliary systems, etc. is generated. The tool is easy to use, efficient compared to manual calculations, and forms a superior basis for selecting the right engine.

5. Select engine

In Fig. 3.08 several *layout* diagrams of engines that include the example SMCR point are displayed. Different priorities of the project will result in different optimum engines.

The engine design capable of delivering the largest power at SMCR-rpm can be derated the most, resulting in a reduced fuel consumption, but will typically also have the highest initial cost and largest dimensions.

CEAS Engine Calculations

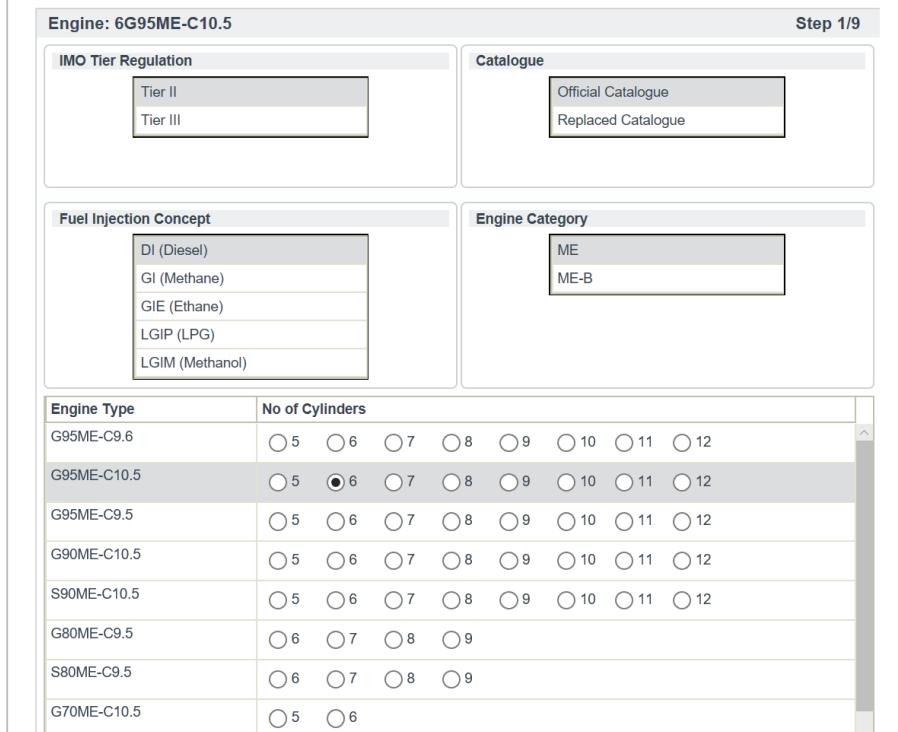


Fig. 3.07: CEAS front page

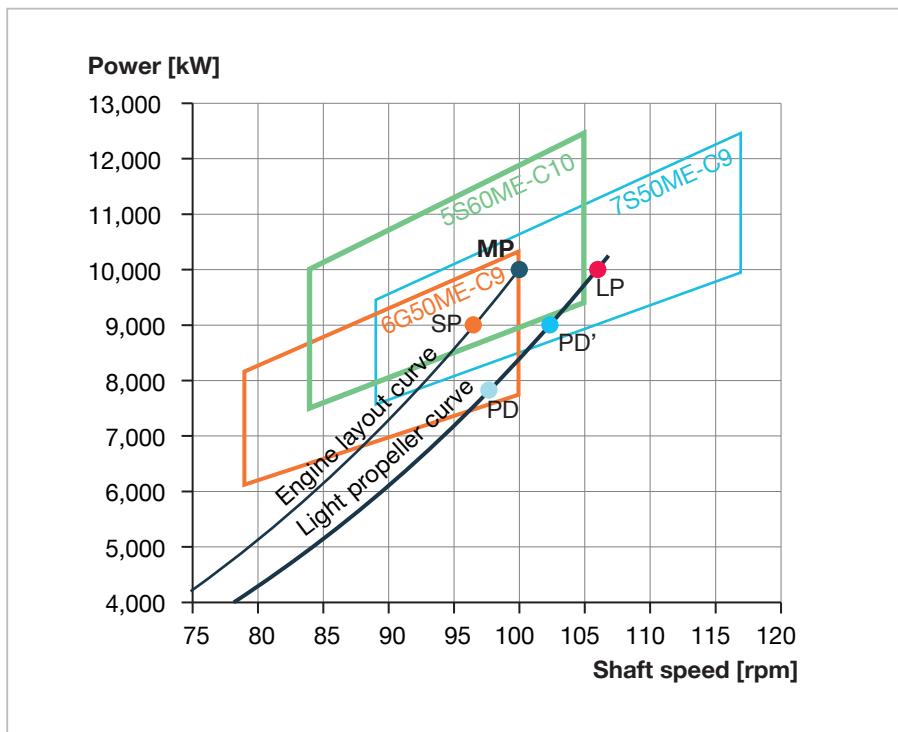


Fig. 3.08: Engine *layout* diagrams of selected possible engines, see the engine programme

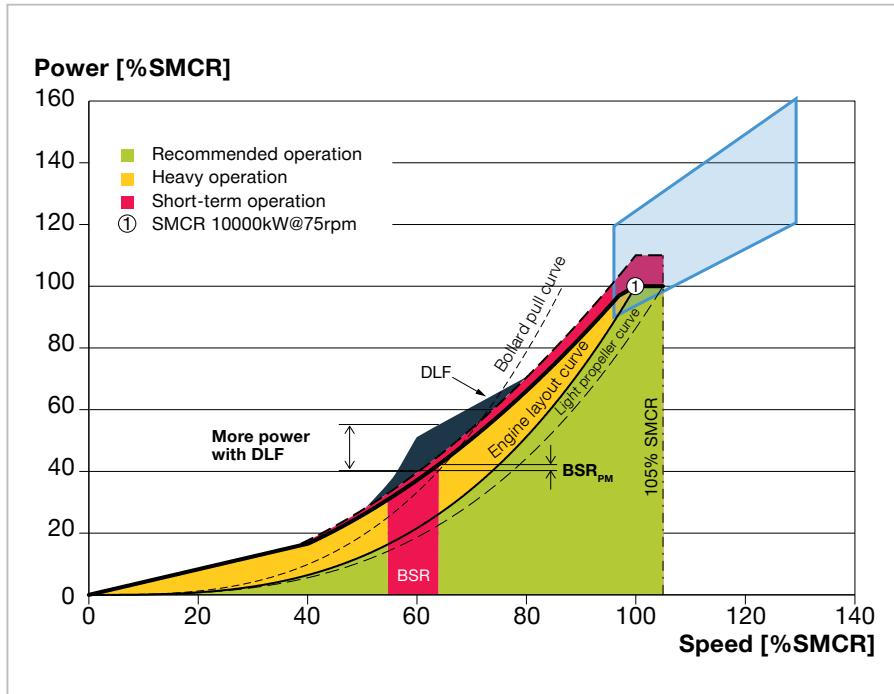


Fig. 3.09: An example of a BSR extending above 60% rpm and resulting small BSR_{PM}. In this example DLF was applied as a retrofit to reduce BSR passage time

Sometimes other parameters such as engine dimensions can have a greater priority than the fuel consumption. Engine length and width can be a challenge for smaller ships.

6. Passage of the barred speed range

As described in the section “Acceleration etc.” in Chapter 2, challenges may exist for passing the barred speed range (BSR) of the shaftline sufficiently quick to avoid shaft fatigue issues, especially for 5 and 6 cylinder engines. This situation, and the dynamic limiter function (DLF) dealing with it, is explained further in the separate paper “The dynamic limiter function”.

The most basic guidance to avoid slow passing of the barred speed range is to avoid barred speed ranges that extend higher than to 60% engine rpm.

A more detailed approach is to ensure a BSR power margin, BSR_{PM}, of at least 10% in the design.

the same rpm. As such, the BSR_{PM} expresses the excess engine power in the upper range of the barred speed range, and hereby the ship’s capability to pass it. On Fig. 3.09 the BSR and BSR_{PM} is shown along with the engine *layout* diagram and the engine *load* diagram described in the subsequent step.

When accelerating, the propeller will be heavy running to various degrees, and the bollard pull curve, see Chapter 2, is used as a reference. Experience shows that the bollard pull curve will be between 15-20% heavy running relative to the light propeller curve, 17.5% is often used when better data is not available.

$$\text{BSR}_{\text{PM}} = \frac{P_L - P_P}{P_P} \times 100$$

P_P is the power required by the bollard pull propeller curve at the upper end of the barred speed range, whereas P_L is the engine power limit without DLF at

7. Engine load diagram & considerations of PTO

The *load* diagram of the engine defines the power and speed limits of an actual engine built with a specified maximum continuous rating (SMCR) - point MP in the *layout* diagram - that conforms with the ship's power requirement.

The engine *load* diagram is important when investigating the possibility of installing a shaft generator/PTO. Additionally it provides the basis for explaining the necessity of the light running margin.

The engine *load* diagram is shown in Fig. 3.10, where different lines are numbered, setting the limits for operation of the engine. The location of the SMCR point within a *layout* diagram does not influence the appearance of the *load* diagram, see Fig. 3.11.

Line 1: The engine layout curve, per definition moving through 100% SMCR-rpm and 100% SMCR power. This curve coincides with the "heavy propeller curve", line 2. An engine without PTO will typically operate to the right of this curve about 95% of the time.

Line 2: The heavy propeller curve is the light propeller curve (line 6) shifted with the light running margin to account for heavy weather and fouled hull.

Line 3: Maximum rpm for continuous operation. For engines with SMCR on the line L₁-L₂ in the *layout* diagram up to 105% of L₁-rpm can be utilised.

If the SMCR is sufficiently speed derated, 110% of SMCR-rpm, but no more than 105% of L₁-rpm, can be utilised, if permitted by the torsional vibration conditions.

For engines where 110% SMCR-rpm does not exceed 105% of L₁-rpm, it is possible to choose an rpm extended *load* diagram. The engine and shafting can, considering the torsional vibration conditions, be constructed to run up to 105% of L₁-rpm, see the separate section "Rpm extended engine *load* dia-

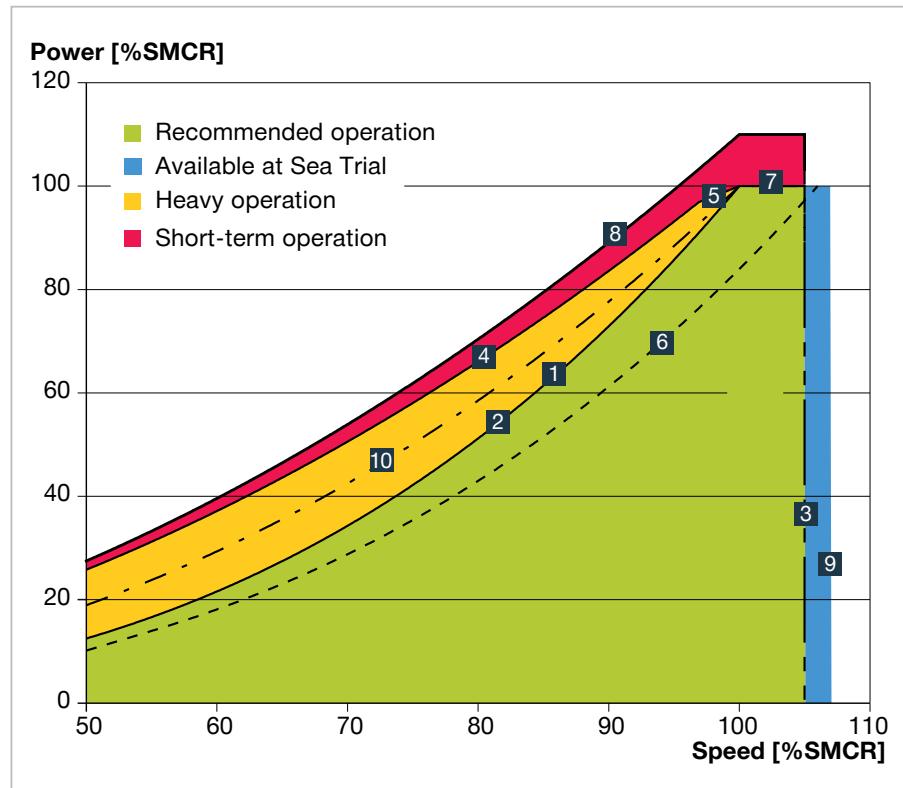


Fig. 3.10: Engine *load* diagram with marked lines

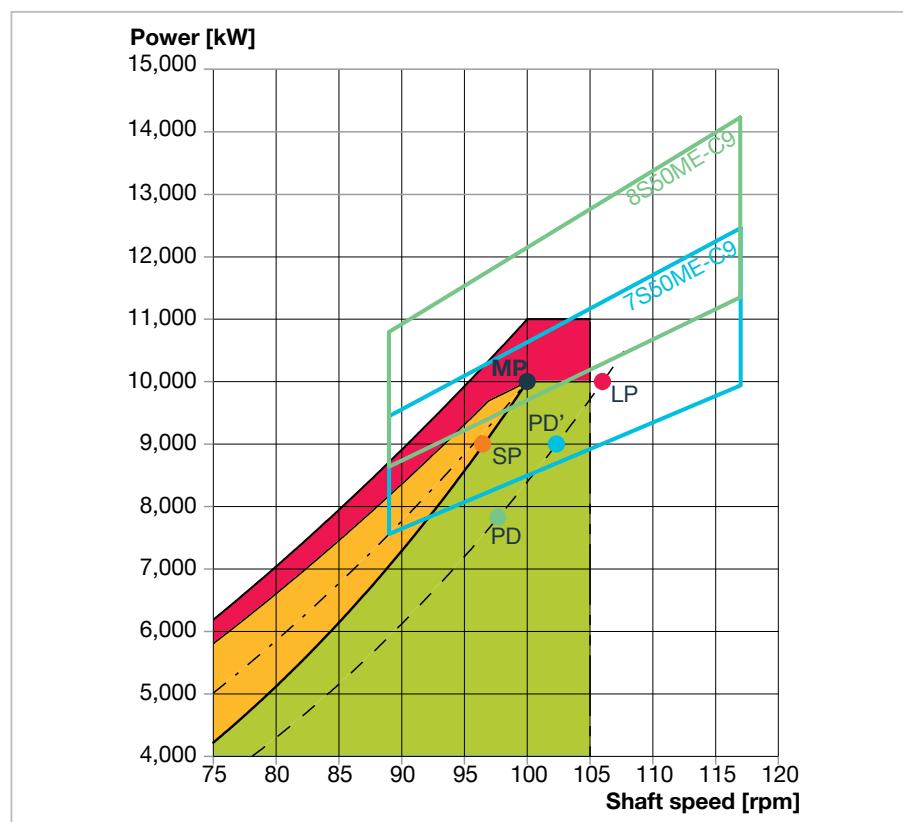


Fig 3.11: Engine *load* diagram, with different *layout* diagrams, note that the appearance of the *load* diagram is independent of the position of point MP within a *layout* diagram

gram" explaining this concept in further detail. A limit to this application is that, typically, class rules do not allow an engine to run faster than 115% of SMCR-rpm.

Line 4: Is the torque/speed limit of the engine, limited mainly by the thermal load on the engine.

Line 5: Represents the maximum mean effective pressure (mep) level acceptable for continuous operation. Note that this is only a limit at high loads and speeds. At lower speeds, line 4 is a stricter limit.

Line 6: The light propeller curve, for clean hull and calm weather. Often used for propeller layout. The rpm margin (in percent) between the engine layout curve and the light propeller curve is the light running margin, see step 3.

Line 7: Represents the maximum power for continuous operation. Note that when increasing rpm towards lines 3 and 9, the maximum power for continuous operation cannot exceed 100%.

Line 8: Represents the overload operation limit of the engine. Overload running is possible only for limited periods, 1 hour per 12 hours or when required in an emergency situation, as the resulting thermal load on the engine is high.

Line 9: Maximum acceptable engine rpm at trial conditions. 110% of SMCR-rpm, but no more than 107% of L₁-rpm if permitted by the torsional vibrations.

Line 10: PTO_{layout limit}. This curve describes the maximum combined power required by the light propeller curve and PTO at a given rpm if a shaft generator/PTO is installed. This layout limit ensures some operational margin to line 4, see the subsequent section "Shaft generator/PTO".

Recommendation for operation

The green area between lines 1, 3 and 7 is available for continuous operation without limitation.

The yellow area between lines 1, 4 and 5 is available for operation in shallow waters, in heavy weather, and during acceleration, i.e. for non-steady operation without any strict time limitation.

The red area between lines 4, 5, 7 and 8 is available for overload operation for 1 out of every 12 hours or when required in an emergency situation.

Limits for low-load running exist. An electronically controlled ME engine can operate down to around 15-20% of L₁-rpm. A mechanically controlled MC engine can operate down to 20-25% of L₁-rpm.

Shaft generator/PTO

With the SMCR and knowledge about the *load* diagram, it is possible to calculate the maximum power take out possible within the limits of the engine.

The maximum power demanded by the light propeller curve and PTO combined should not exceed a limit governed by the following equation. This equation gives line 10 on Fig. 3.10:

$$\text{PTO}_{\text{layout limit}} = P_{\text{SMCR}} \times \left(\frac{n}{n_{\text{SMCR}}} \right)^{2.4}$$

The maximum design PTO power at a given rpm is then found as the vertical difference between line 6, the light propeller curve, and line 10, the PTO_{layout limit} as marked in Fig. 3.10.

If the full SMCR power is used for propulsion, the PTO naturally cannot take out any power.

Designing the power of the PTO not for the torque/speed limit (line 4) and mep limit (line 5) but instead the PTO_{layout limit} will ensure that the PTO can be operated also in conditions not as ideal, as ideal sea trial condition.

In case of fouling and/or heavy weather, the propeller curve will shift left towards the engine layout curve, exploiting the light running margin. With increased heavy running, the electric power taken out at the PTO must gradually be decreased (and taken over by the auxiliary engines) in order not to push the operational point outside the engine limits.

In severe cases, fouling and sea conditions alone are enough to shift the propeller curve to line 4, see "Light running margin". In such cases, the PTO cannot be utilised without overloading the engine and the auxiliary engines must deliver all the electric energy. The PTO can take up load again when weather conditions have improved.

Traditionally, complicated gearing have ensured a constant electric frequency from the PTO within a range of varying engine rpm. Gearing has to a wide extend been replaced by power electronics, as electronics have dropped significantly in price over recent years.

Considering EEDI, see Chapter 4, a PTO can be an attractive solution as the EEDI reference speed is calculated at 75% engine load in trial condition, where there is a margin for PTO power.

Considerations of shaft motor/PTI

For some trades it may be attractive to use a shaft motor, also termed a power take in (PTI). In such cases, electric power from the gensets, waste heat recovery or other electric power sources are used to boost the propulsion of the ship.

When using PTI, the engine will typically deliver 100% power, but the propeller, and thus also the engine, will run at increased rpm due to the extra power delivered to the shaft by the PTI. For such type of operation, it can be necessary to consider an engine with an rpm-extended *load* diagram, see the later section.

8. Compliance with regulations

The selected engine's compliance with the environmental legislation described in Chapter 4 must be considered. Considerations on NO_x reduction measures to fulfil IMO Tier III in ECA zones does not directly influence the attained EEDI, as this is calculated for Tier II only.

Having ensured that the ship complies with SO_x and NO_x legislation as well as EEDI, the ship's capability to manoeuvre safely in all relevant conditions must be considered. This is termed minimum propulsion power (MPP) and is described in Chapter 4.

Engine selection spiral for CP-propeller

This section will shortly elaborate on the engine selection spiral for a CPP plant, referring to and explaining the difference to the FPP engine selection spiral.

The steps in the CPP engine selection spiral in Fig. 3.12 can be written as:

1. Establish calm water resistance, propeller diameter and working conditions, see Chapters 1 and 2
2. Calculate the possible CP-propeller operation and the required power
3. Consider the operating principles of the CP-propeller for inclusion of possible PTO
4. Establish the sea margin, engine margin, and the rpm "light running" margin for the project in order to specify the maximum continuous rating (SMCR)
5. Plot the SMCR on the engine layout diagrams for various possible engine types (use CEAS)
6. Based on the CEAS results, select an engine depending on the priorities of the project
7. The engine *load* diagram of the selected engine, considerations about shaft generator/PTO and PTI if desired for the project
8. Check compliance with regulations, EEDI and Minimum Propulsion Power (MPP), see Chapter 4

Depending on the outcome of considerations on a possible shaft generator in step 7 and regulations in step 8, it can be necessary to re-enter the design spiral at either step 1, 3, 4 or 6.

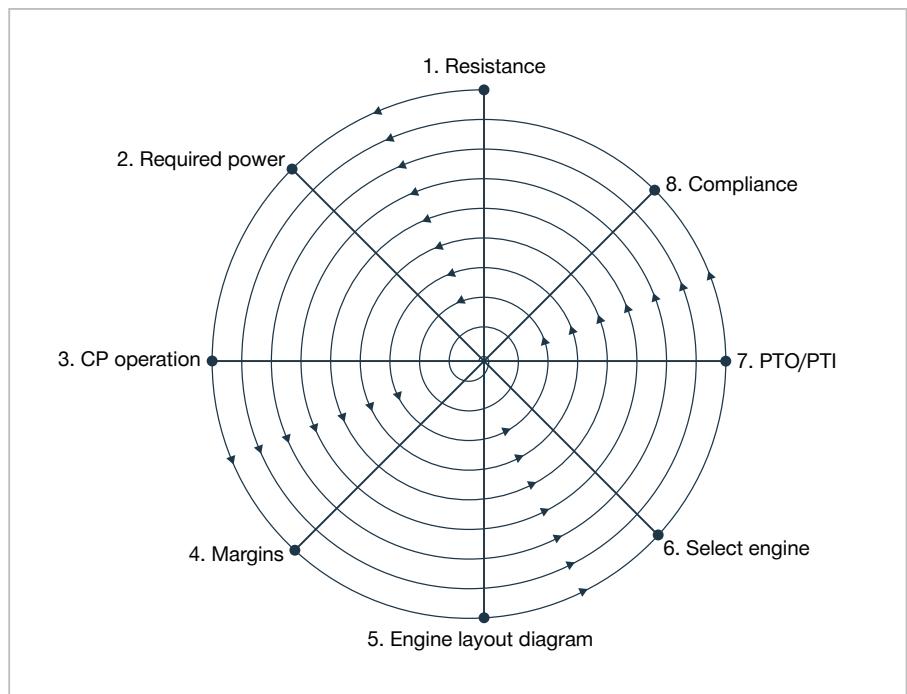


Fig. 3.12: CPP engine selection spiral

2. Possible propeller operation for CPP & required power

There are three options when operating a ship with a CP-propeller:

- Constant engine rpm
- Follow fixed combinator curve
- Control pitch and rpm individually (typically based on some combinator curve)

The required power at the design speed will be practically the same, independent of the operational method, but the required power at lower speeds will differ.

Constant rpm is only of interest if a PTO is to be included, see step 3. Constant rpm operation eliminates the need for considerations of two important principles from the FPP selection spiral: light running margin and passage of the barred speed range.

Operating on a fixed combinator curve means to follow a curve optimised for optimal setting of the propeller speed and pitch. The combinator curve is designed to take into account the cavitation patterns of the propeller, the operational profile of the ship, and to some extent the engine limits, see Fig. 3.13.

Operation along a fixed combinator curve would require considerations for a light running margin, see “Light running margin”, and sufficiently quick passage of the barred speed range, see step 6 in the FPP selection spiral.

The third option, to control the pitch and engine rpm individually (typically based on some combinator curve) is, in principle, equivalent to fitting a variable gearbox between engine and propeller.

Controlling the pitch allows for quick acceleration of the engine and shaft line, and therefore the barred speed range passage is not a problem. Additionally, the engine can be loaded at any point within the engine *load* diagram, eliminating the need for a built-in light running margin.

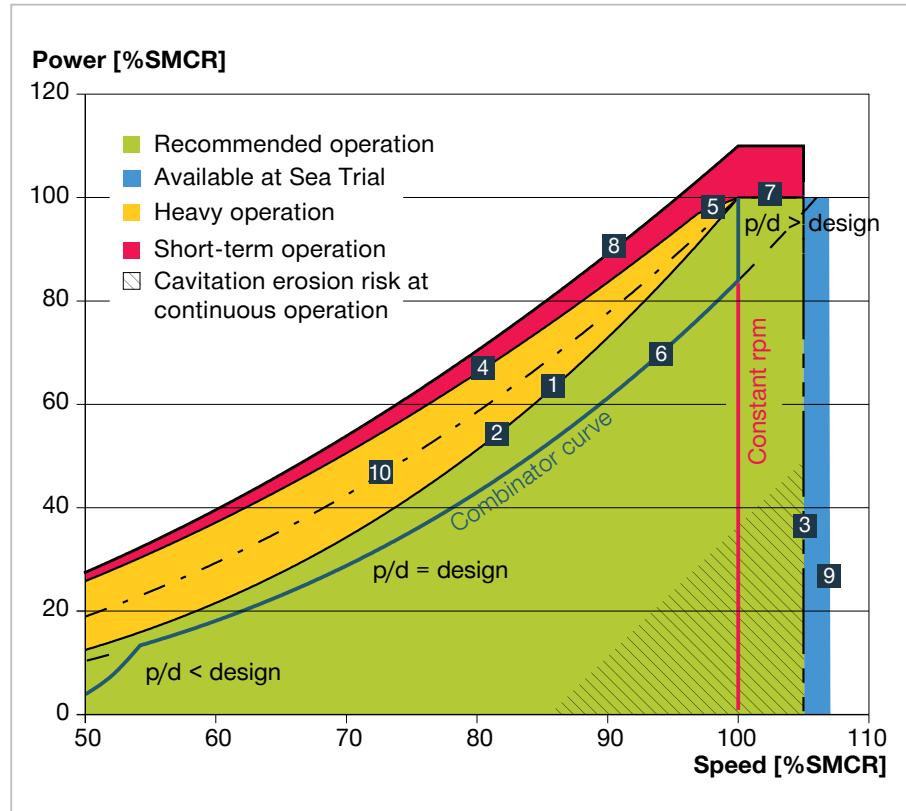


Fig. 3.13: Combinator curve with engine *load* diagram, see Ref. [3.3]. The constant rpm curve can also be referred to as the generator curve

Data on the specific fuel oil consumption of an engine along a combinator curve can be requested from MAN Energy Solutions by writing to LEE5@man-es.com.

3. CPP operating principles for inclusion of PTO

The advantage of operating at constant rpm is that it allows for easy inclusion of a synchronous PTO.

The disadvantage of operating at constant rpm is that, at low ship speeds, the frictional losses on the propeller and in the engine will be relatively higher than if a combinator curve were followed.

Even though the efficiency is low at low loads, so is the energy consumption, which results in a small penalty on the overall fuel consumption, therefore, constant rpm can still be an attractive option if a very simple system is desired.

It is of special interest that following a combinator curve, or controlling the pitch and rpm individually, means that the relative frictional losses are not increased at lower ship speeds. With the greatly reduced costs for power electronics that can ensure constant frequency from electric generators at fluctuating rpm, the previous disadvantage of operating along a combinator curve with a PTO, i.e. the expensive gearing, has been reduced.

4. Propulsion margins for CPP

The magnitude of the sea and engine margins is not changed for projects with CP-propellers. The light running margin can still be necessary for CP-propellers, all depending on the capabilities of the propeller control system.

Traditionally, the combinator curve, as illustrated in Fig. 3.13, has been calculated and set for the propeller. This means that for a given propeller speed, the combinator curve has a given propeller pitch.

Operating along a fixed combinator curve, like for a fixed pitch propeller, the propeller may be heavy running in heavy weather, requiring a light running margin. See the FPP selection spiral.

The development within fixed combinator curves has been to calculate different combinator curves from which the crew can choose, depending on the degree of fouling and sea conditions. This eliminates the need for a light running margin, but attention to the consequences of continuous heavy running is required from the crew.

Some modern CP-propeller control systems allow for continuous adaptation of the propeller pitch, exploiting that in fact installing a CP-propeller is equivalent to having free gearing between the engine and the propeller. Inclusion of a light running margin can be avoided if automatic continuous pitch adaptation is applied, as the pitch can be decreased when the propeller shifts towards a heavier curve. Control systems that continuously adapt the pitch will be able to load the engine at all points within the engine *load* diagram, and as such the engine can in principle deliver maximum power in any weather and fouling condition.

The ship designer is advised to investigate the options offered by different propeller manufacturers with regard to the capabilities of the control of the CP-propeller.

5. Engine layout diagram with SMCR for CPP

The propeller type does not affect the *layout* diagram of the engine.

6. Select engine for CPP

The application of a CP-propeller does not imply any changes with regard to the selection of engine. The optimum choice with regard to derating, installation height, first cost, etc. will be similar to the situation for a FP-propeller.

7. Engine load diagram for CPP and considerations of PTO power

The engine *load* diagram is unaffected by the combination with a CP-propeller. Instead, the position of the operational point of the engine within the *load* diagram is in principle, depending on the capabilities of the propeller control system, set free.

When running at maximum rpm with zero pitch, the power required to turn the propeller is approx. 15-20% of SMCR.

Shaft generator/PTO and PTI

As described previously, more options for operating a shaft generator with a CP-propeller exist: Constant engine rpm or (as with an FP-propeller) varying engine rpm where gearing or modern power electronics ensure constant frequency. The same considerations apply to inclusion of a PTI.

When considering the $PTO_{\text{layout limit}}$, line 10 on the engine *load* diagram in Fig. 3.13, it is evident that the most PTO power can be delivered at relative high engine rpm. Here, the distance between the light propeller curve/combinator curve and the $PTO_{\text{layout limit}}$ is largest.

CP-propellers can therefore be of interest if the electricity demand on the ship is high and the operational profile involves large variations in ship speed. If the engine rpm is independent of the ship speed (by varying the pitch), then the engine rpm can be kept high during low ship speeds, and thereby ensure a constant availability of full PTO power. This can typically be of interest for ships with many reefer units in warmer weather conditions and/or in scheduled liner traffic.

In such cases the increased efficiency of the main engine for generating electric power compared to the auxiliary engines will result in savings. Typically, these savings will be larger than the losses implied by the slightly lower efficiency of the CP-propeller compared to an FP-propeller.

8. Compliance with regulations

Considerations on NO_x and SO_x eliminating measures do not change with the selection of a CP-propeller, and neither does the EEDI requirements.

As previously briefly discussed, CP-propellers can be an attractive solution for ship designs that have difficulties towards fulfilling the requirements for minimum propulsion power, see Chapter 4.

If the CP-propeller control system can control pitch and propeller rpm individually, the engine will be capable of delivering its maximum power even at reduced ship speeds. Hereby, an increase in engine power (which would negatively influence the EEDI of the design) can be avoided.

Engine tuning

This and the following sections are intended to provide further detail on some of the engine related parameters shortly described in the process of working through the engine selection spirals.

Engine tuning applies to an engine with a specified MCR – derated or not. As of 2018 three options exist for tuning a regular ME engine in Tier II mode:

- High-load tuning
- Part-load tuning
- Low-load tuning

The tuning decides the shape of the SFOC curve as a function of engine load. High-load tuning has the lowest SFOC in the high-load range and low-load tuning has the lowest SFOC in the low-load range. Choice of tuning depends on the expectations for the operational profile of the ship. For a ship sailing a fixed schedule in a certain geographical area it is often possible to predict the required engine power quite accurately, allowing just the adequate selection of engine SMCR power. In such a case, a high-load tuning is often a good choice. For a ship engaged in tramp trade, such as many bulk carriers, the engine load can differ greatly between voyages, and low-load tuning is then sometimes selected.

Tier III engines do not offer the option for load tuning as the parameters controlling the combustion process are already fixed in order to meet both Tier II and Tier III demands.

Rpm extended load diagram

For speed derated engines with an SMCR-rpm sufficiently below L_1 -rpm, the rpm limits of the engine can be designed to extend beyond the normal maximum limit of 110% of the SMCR-rpm, but not more than 105% of engine rpm at L_1 during normal operation (107% during sea trial). This can be of special interest especially to ships often operating in conditions implying increased resistance such as:

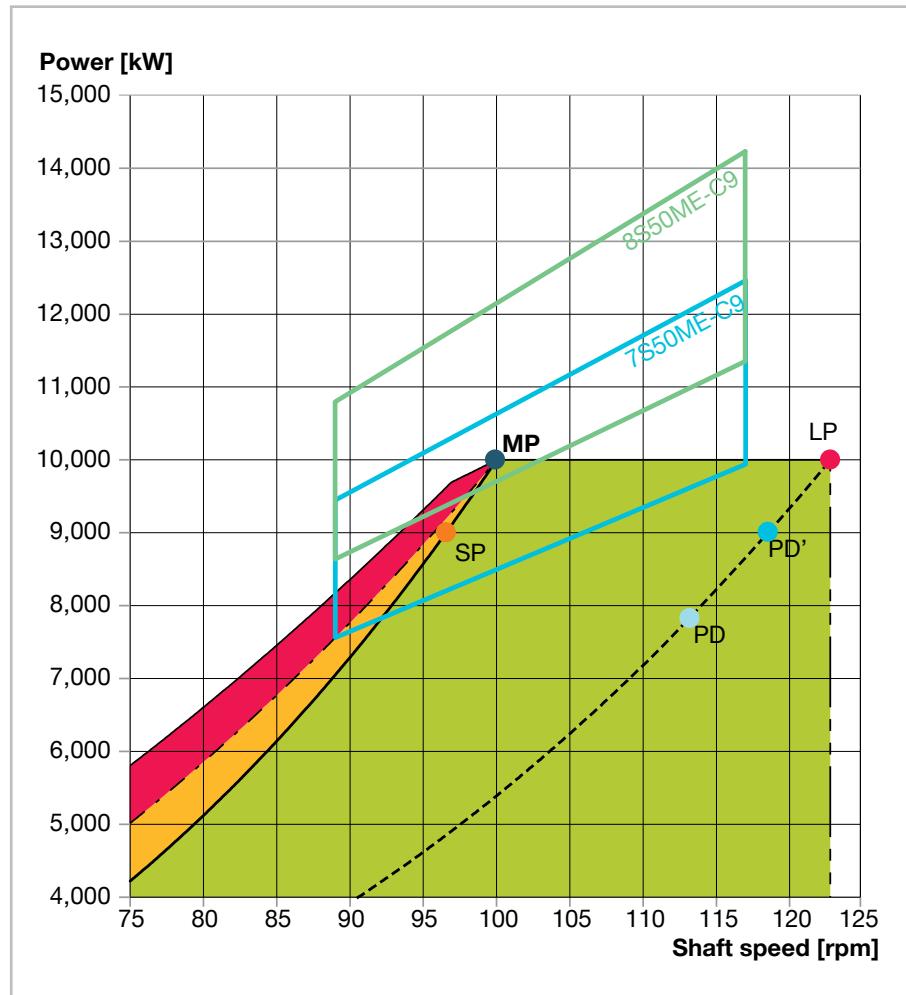


Fig. 3.14: Rpm extended load diagram, for an extreme case

- ships sailing in areas with very heavy weather
- ships operating in ice
- ships with two fixed pitch propellers/two main engines, where one propeller/engine is blocked/declutched for some reason. Measurements show an approximate 8-10% heavy running of the remaining propeller in operation for a twin-skeg ship.

This possibility for speed derated engines is described in the rpm extended load diagram in Fig. 3.14. Here, the engine layout diagram is plotted as well, showing the speed limits of the engine design.

In Fig. 3.14 it is seen that for a heavily speed derated engine with an SMCR at point MP, the rpm extended load dia-

gram permits a substantial increase to the possible light running margin, in the extreme case depicted up to 23%.

Considering that the bollard pull propeller curve is rarely more than 15-20% heavy running, a light running margin higher than this will not bring any benefit. On the contrary, there will be a significant penalty on propeller efficiency for such a high light running margin. In addition, class rules typically does not allow an engine to run faster than 115% of SMCR-rpm.

The rpm limit can only be extended if the torsional vibration conditions permit this. Thus, the shafting, with regard to torsional vibrations, has to be approved by the classification society in question.

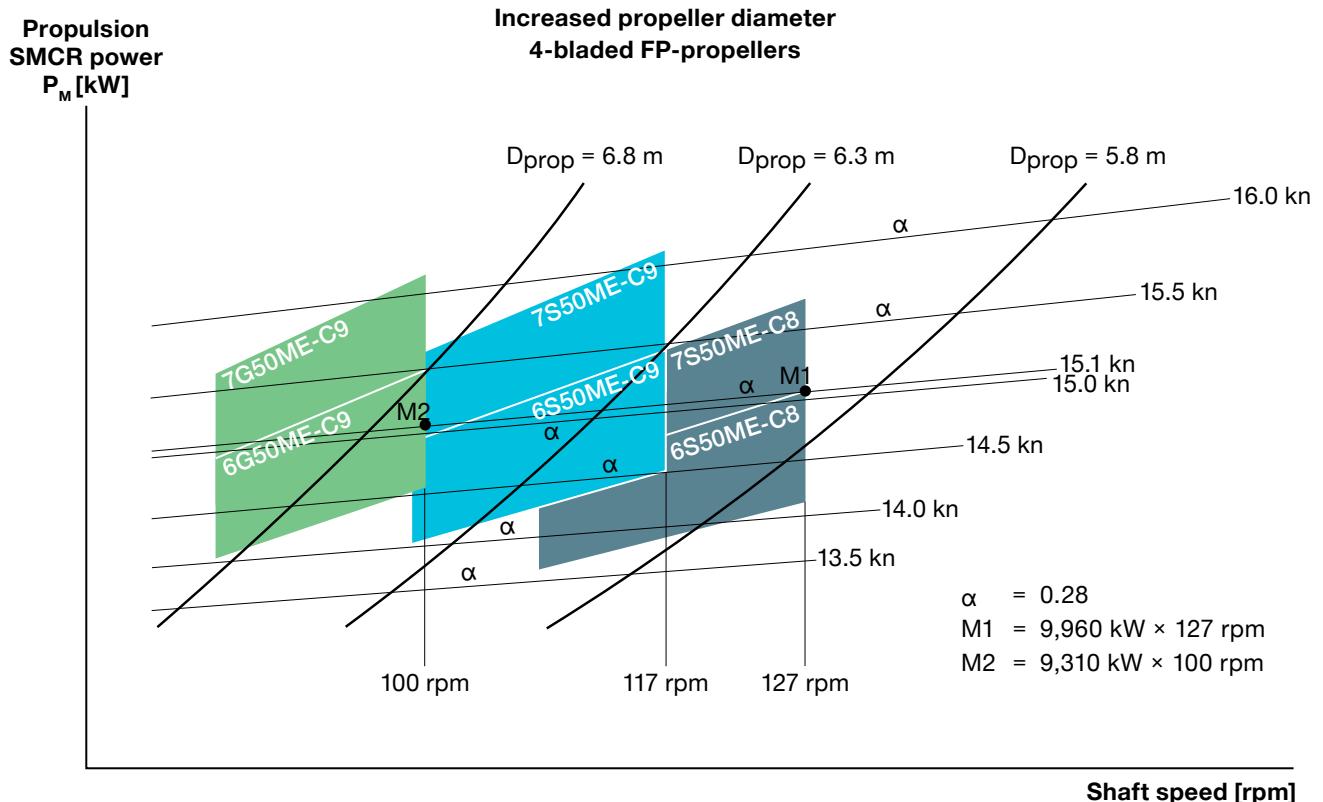


Fig. 3.15: Example of constant ship speed curves with selected engine layout diagrams

Constant ship speed curves

As earlier described in Chapter 2, the larger the propeller diameter, the higher the propeller efficiency and the lower the optimum propeller speed. There is therefore a drive to optimise the aft-body and hull lines of the ship to include a larger propeller diameter.

The constant ship speed curve, α , shown in Fig. 3.15 indicates the power required at various propeller diameters and, thereby, propeller rpm's to keep the same ship speed if at any given rpm the optimum pitch/diameter ratio is used, taking into consideration the total propulsion efficiency.

Normally, for a given ship with the same number of propeller blades but different propeller diameter, the following relation between necessary power and propeller speed can be assumed:

$$P_2 = P_1 \times \left(\frac{n_2}{n_1} \right)^\alpha$$

where:

P = propulsion power

n = propeller speed, and

α = the constant ship speed coefficient.

For any combination of power and rpm, each point on the constant ship speed curve gives the same ship speed.

When such a constant ship speed line is drawn into the engine layout diagram through a specified MCR point 'M1', selected in the layout diagram, another specified propulsion MCR point 'M2' upon this line can be chosen to give the ship the same speed for the new combination of engine power and rpm.

Provided the optimum pitch/diameter ratio is used for a given propeller diameter, the following data applies when changing the propeller diameter:

For general cargo, bulk carriers and tankers: $\alpha = 0.20 - 0.30$

Container and ro-ro: $\alpha = 0.15 - 0.25$

Fig. 3.15 shows an example of the required power speed point M1, through which a constant ship speed curve $\alpha = 0.28$ is drawn. Hereby point M2 with a lower engine power and a lower engine rpm is achieved at the same ship speed.

Thus, for a handymax tanker, if the propeller diameter is increased, and going for example from the SMCR-rpm of $n_{M1} = 127 \text{ rpm}$ to $n_{M2} = 100 \text{ rpm}$, the required power will be $P_{M2} = (100/127)^{0.28} \times P_{M1} = 0.935 \times P_{M1}$, i.e. providing a power reduction of about 6.5%.

When changing the propeller speed by changing the pitch/diameter ratio, the α constant will change.

Power functions and logarithmic scale for engine diagrams

Historically, and before the introduction of computers, it was practical to display engine *layout* and *load* diagrams in logarithmic scale. Logarithmic scale is still commonly used in the industry, and their use can be beneficial to the experienced designer, and bring additional clarification.

The effective brake power P_B of an engine is proportional to the mean effective pressure (mep) and engine rate of rate of revolution, n . When using c as a constant, P_B may then be expressed as follows:

$$P_B = c \times \text{mep} \times n^i$$

This means that for constant mep, the power is proportional to the rotational speed of the engine, $i = 1$:

$$P_B = c \times n^1 \text{ (for constant mep)}$$

When only considering frictional resistance, and as described in the section "Propeller Law and speed power curves" in Chapter 2, the power demanded for propelling a ship with a fixed pitch propeller at a speed V is proportional to the cubic of this, $i = 3$:

$$P_{\text{prop}} \propto c \times V^3$$

As described in Chapter 2 the exponent for calculating the power required for propelling the ship at a speed V can be higher than 3 when including wave making resistance etc.

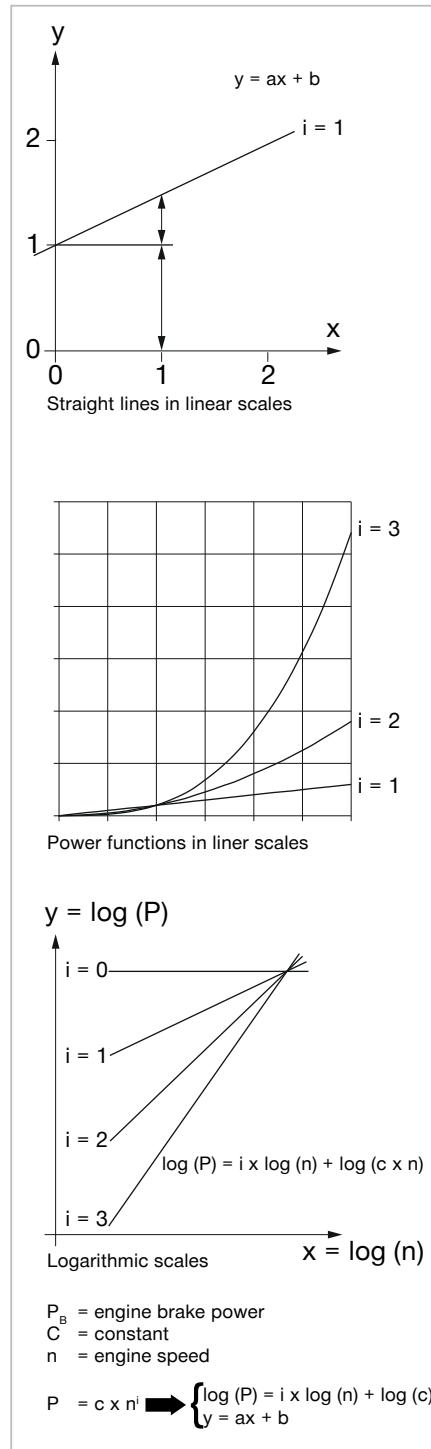


Fig. 3.16 depicts P_B and P_{prop} in a normal linearly scaled coordinate system and in logarithmic scale. The beauty of the logarithmic scale is that a power function in logarithmic scale will be a linear line, the slope of which depends on the exponent i of the power function, according to the principal formula:

$$P_B = c \times n^i$$

Resulting in the logarithmic scale

$$\log(P_B) = i \times \log(n) + \log(c)$$

This means that the P_B for constant mep will follow lines of $i = 1$ in the logarithmic scale and P_{prop} will follow lines of $i = 3$.

Logarithmic lines of $i = 1$ are equivalent to regular linear lines, and as such the shape of the engine *layout* diagram is the same regardless of applying logarithmic scales or not, only the shape of the *load* diagram will change.

Fig 3.16: Linear and power functions in linear and logarithmic scale

Chapter 4

Environmental regulations

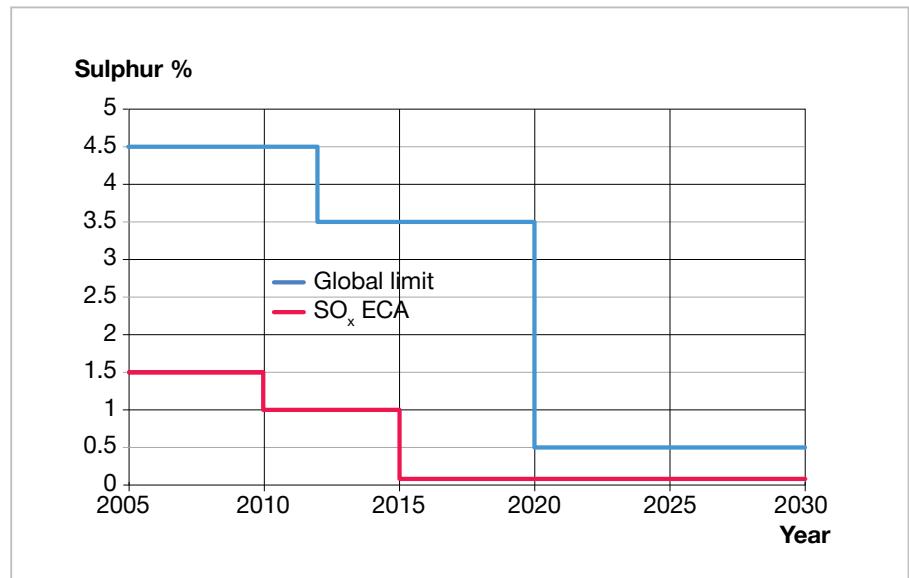


Fig. 4.01: Limits for sulphur emissions

Regulations limiting the emission of sulphur oxides (SO_x) and nitrogen oxides (NO_x) have been implemented during the latest decades. Special emission control areas (ECA) have been declared, further limiting SO_x and NO_x emissions in specific areas.

Measures towards fulfilling these regulations are briefly described in the subsequent sections on SO_x and NO_x . For in-depth knowledge, the reader is advised to consult MAN Energy Solutions' "Emission Project Guide", Ref. [4.1]. This is continuously updated and can be downloaded from our homepage → Two-Stroke → Project-Guides.

With the energy efficiency design index (EEDI) the emission of greenhouse gases is set in focus and sought limited, as described in a section of its own. As an effect of implementing EEDI, the

IMO has also made recommendations on minimum propulsion power, described at the end of this chapter.

Sulphur oxides

Sulphur in fuel leads to emission of SO_x and particulate matter (PM). The IMO has set a global limit on sulphur emissions from all ships, gradually to be lowered as seen in Fig. 4.01. From 2020, a global limit of 0.5% will apply, and a limit of 0.1% already applies locally in ECAs.

At the time of writing (2018), the sulphur content of HFO is approx. 2-3.5% and significantly lower, approx. 0.1%, for MGO, and somewhere in between for MDO. The ECA sulphur cap has led to the introduction of low-sulphur HFO (LSHFO), having a sulphur content below 1%. Future reductions are expected to meet the 0.5% cap. As low-sulphur fuels typically have different properties

with regard to the lubrication of fuel components, special care must be taken as described in the paper "Guidelines for operation on fuels with less than 0.1% sulphur".

Alternative fuels such as LNG, LPG and methanol contain an insignificant amount of sulphur.

The IMO permits the use of fuels with a higher sulphur content if technical measures can be applied to reduce the SO_x emissions to a level equivalent to using low sulphur fuel, the main option for this is SO_x scrubbers.

SO_x scrubbers

Scrubber technology has been known for decades, as inland emissions of SO_2 , e.g. from power plants, have been regulated. The primary motivation for using SO_x scrubbers in a maritime application is that HFO is significantly cheaper than low-sulphur fuels. For a

large ship, this difference can turn a scrubber investment into a good business case, naturally requiring that the space for the system can be allocated.

Wet scrubbers are typically applied for marine applications, using either seawater in an open loop or freshwater with additives in a closed loop. Typically, both options are included in a hybrid solution, see Fig. 4.02.

Open loop operation offers low running costs, as seawater is fed directly to the scrubber, neutralising the SO_x only due to the natural alkalinity of seawater. The water is then led back to the sea. The discharge criteria set by the IMO guidelines is met by the high water flow through the scrubber. Power for pumping the seawater will be the largest running cost.

A closed loop system must be used if the alkalinity of the seawater is too low, the seawater is too dirty, or where local legislation prohibits open loop operation. Typically, NaOH is added in order to form sulphate with the sulphur. Sulphate accumulates in the scrubber water amongst the particulate matters from the combustion. Water is formed in the combustion, and constitutes a constant supply of fresh water, while water is bled off from the loop to be cleaned in a special unit, before being discharged to the sea. Closed loop running costs are higher due to the constant addition of chemicals, even if the reduced flow reduces pumping costs.

With regard to fulfilling the EEDI requirements, it is important to note that a slightly increased electric load from the pollution reducing measures does not affect the index attained. The electrical consumption to use in the EEDI calculations are based on the power of the main engine.

Nitrogen oxides

IMO has regulated NO_x emissions since 2000, implementing limits in various Tiers, I, II and III. Tier II has applied globally for ships which had their keel laid after 2011. Tier III applies in

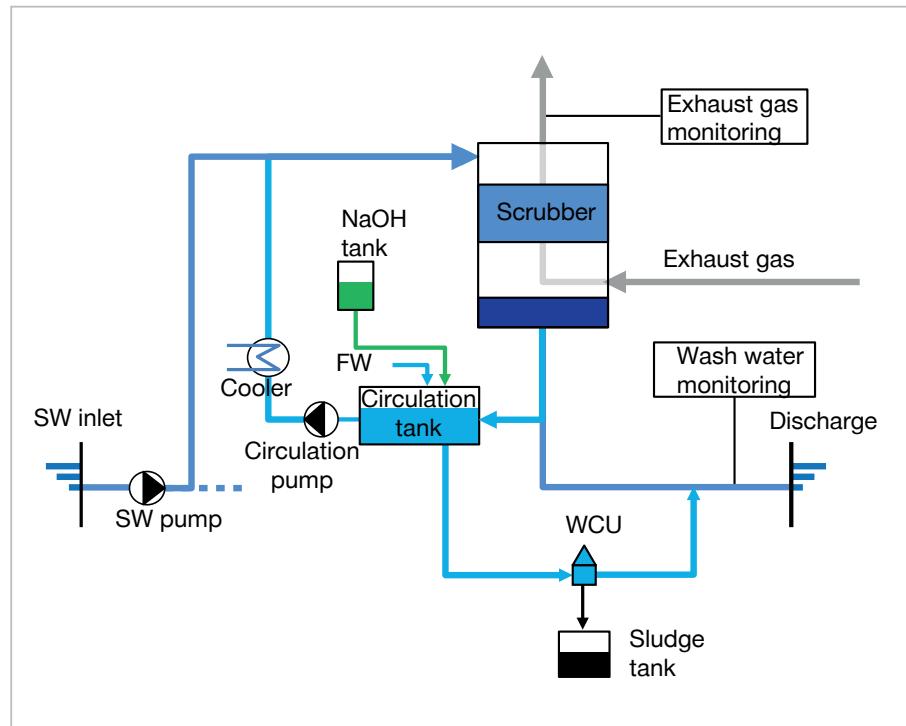


Fig. 4.02: Hybrid scrubber system

ECA-areas only, in the North American ECA from 2016 and in the North European ECA from 2021, see “Emission control areas for SO_x and NO_x”. The NO_x emission limits are a function of the rated engine speed, see Fig. 4.03.

Both NO and NO₂ is included in NO_x, and especially NO₂ is harmful. NO_x can be formed by different processes resulting in either fuel, thermal, or prompt NO_x, see Ref. [3.1]. Thermal NO_x is the main source in a diesel cycle engine.

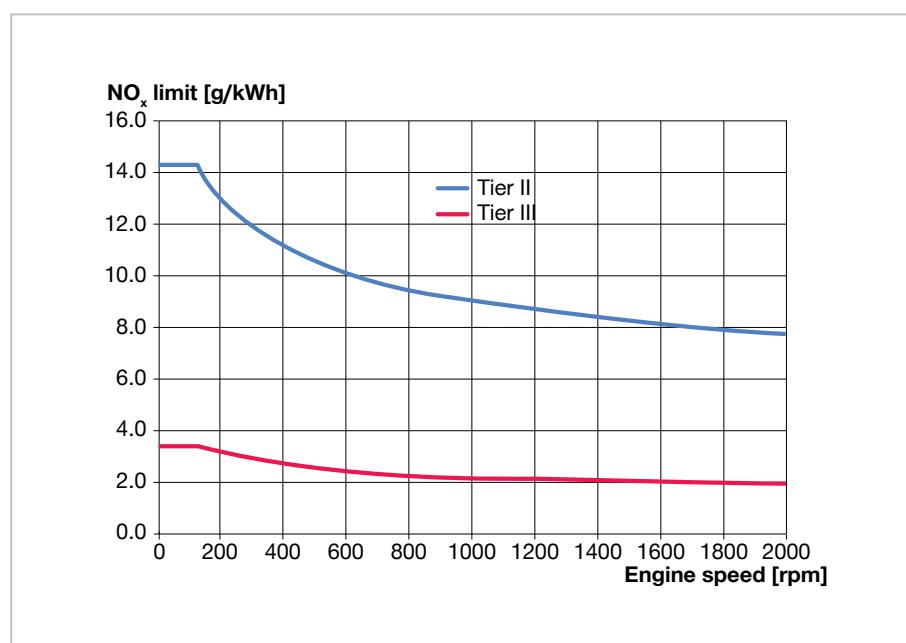


Fig. 4.03: Limits on NO_x emissions depending on engine speed



Fig. 4.04: Illustration of different NO_x reducing measures

Temperature and especially peak temperature during the combustion is the main parameter governing the amount of NO_x formed, but the timespan at peak temperature is of course also important.

Sufficient control of the temperature for fulfilling Tier II can be achieved by adjusting engine parameters such as injection timing, injection pressure, exhaust valve timing, etc.

Fulfilling Tier III limits require specific NO_x reducing technologies, described in the following sections about EGR and SCR. For certain fuels, such as methanol, WIF (water in fuel) is also a possible technology to achieve compliance with IMO Tier III NO_x limits.

Exhaust gas recirculation

Tests show that the amount of NO_x formed is related to the amount of oxygen found in the combustion chamber. The EGR-plant substitutes some of the fresh scavenge air with exhaust gasses that have a lower oxygen content. The exhaust gas is led through a cooling and cleaning cycle before re-entering the scavenge air receiver.

Due to the recirculated exhaust gas, more of the gasses in the chamber must be entrained in the combustion process, i.e. the flame must broaden to entrain sufficient amounts of oxygen to burn the injected fuel. Entrainment of more gas means that the heat capacity of the combustion zone is increased, i.e. more gas must be heated to burn the fuel, which lowers the peak temperature of the combustion. The increased mass and specific heat capacity of the CO₂ and H₂O in the recirculated gas compared to O₂ and N₂, in fresh air, also contributes to increasing the heat capacity of the combustion zone.

An EGR system placed before the turbocharger is a high-pressure (HP) system, whereas a low-pressure (LP) system is placed after the turbocharger. HP EGR is most commonly used in marine applications.

When recirculating the exhaust gasses, the amount of fresh air that must be provided from the turbocharger decreases. The turbocharger must be capable of handling this condition, either by a bypass or if multiple turbochargers are installed, by cutting out one.

Using an EGR system to achieve Tier II is an option to improve Tier II SFOC (specific fuel oil consumption). Instead of optimising the combustion parameters for fulfilment of Tier II, these can be optimised for maximum efficiency. This has been termed EcoEGR.

Selective catalytic reduction

SCR does not interfere with the combustion process, but instead treats the exhaust gas according to the chemical reactions described in the "Emission project guide", Ref. [4.1].

Since ammonia (NH₃) is poisonous in its pure form, it is stored and added in the form of urea, which is an aqueous solution of NH₃. The urea is reduced to ammonia in the system prior to the actual SCR catalyst.

An SCR system requires an elevated temperature in order to activate the catalyst elements. Furthermore, the sulphur content of the fuel sets limits for the minimum temperature, as sulphur reacts with ammonia to form ammonium bisulphate (ABS) at low temperatures. If the temperature is too low, this sticky substance can build up in the

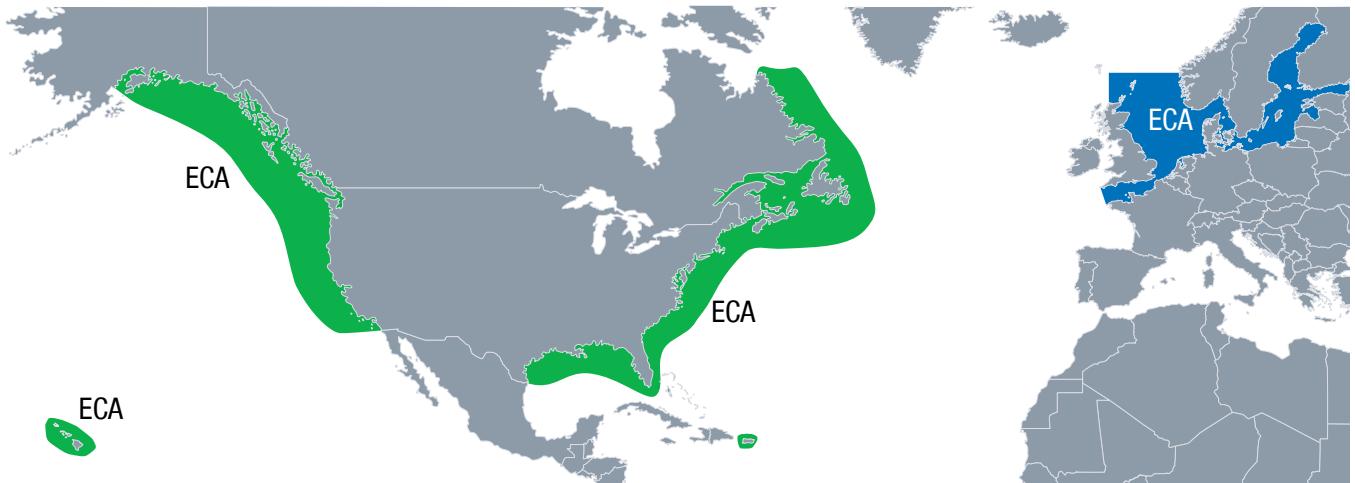


Fig. 4.05: Existing emission control areas around North America and North Europe

catalyst elements. Often, the ABS-imposed temperature limit is higher than the temperature required for the elements to perform the reduction.

SCR plants for marine applications exist both in a high and low-pressure variant. The benefit of the LP SCR is that it offers the designer better flexibility for arranging the installation, compared to the HP solution that must be positioned prior to the turbocharger. Temperatures are lower after the turbocharger than before, which means that typically only low-sulphur fuel can be combined with a low-pressure SCR system.

Especially at low load, the application of SCR increases the SFOC due to the required minimum temperature of the exhaust gas. In the HP system, a cylinder bypass is installed to decrease the mass of air in the cylinders, thereby increasing the exhaust gas temperature, and reducing the efficiency of the engine. In the LP version it is required for some of the exhaust gas to bypass the turbocharger and directly heat the SCR elements, thus also decreasing efficiency.

Emission control areas

Emission control areas (ECA) can be declared for SO_x or NO_x, or for both types of emissions. Fig. 4.05 shows the existing ECA zones. The SO_x ECA limits of 0.1% sulphur apply to all ships both in the North American and Northern Europe ECA. In the North American ECA, Tier III NO_x limitations are applicable for new ships built after 2016, and in Northern Europe after 2021.

Any abatement technology reducing the emission of SO_x and NO_x below the limits set will be accepted, given that the relevant IMO guidelines are followed.

Energy efficiency design index

In order to ensure the reduction of greenhouse gasses (GHG) from shipping, the IMO have introduced the energy efficiency design index (EEDI) as an amendment to annex VI of MARPOL, Ref. [4.2].

The principle of EEDI is to compare the emission of GHG from new ships with a baseline of existing ships built from 2000 to 2010, see Fig. 4.06.

A reduction factor of the index required for new ships is prescribed. It increases the requirement over time, as generally illustrated in Table 4.01 for large ships, Ref. [4.2]. Relaxations for small ships are made, and for general cargo ships as well as reefers, phase 2 only requires a reduction of 15%.

As the EEDI regulations are continuously evaluated, the reader is advised to consult the latest resolutions. As of 2018 these are available on:

www.imo.org → Our Work → Marine Environment → Pollution Prevention → Air Pollutant and GHG Emissions → Index of MEPC Resolutions and Guidelines related to MARPOL Annex VI → Guidelines related to Energy Efficiency of Ships

Calculation of required EEDI

A ship must attain an EEDI value lower than a required value, based on the baseline established for the ship type, corrected with the required reduction factor (X) as stated in Table 4.01.

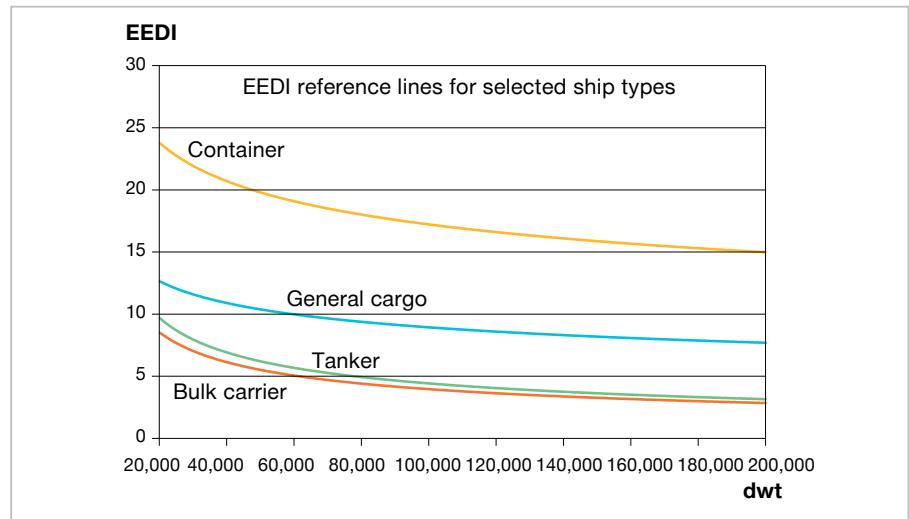


Fig. 4.06: EEDI reference lines for selected ship types

$$\text{Reference line value} = a \times b^{-c}$$

Here, a & c are baseline parameters, Ref. [4.2], and b is the maximum dead-weight of the ship given in tonnes. For container ships 100% dwt is applied as well.

$$\text{Required EEDI} = \left(1 - \frac{X}{100}\right) \times \text{Reference line value}$$

If the design of the ship falls into more than one category, the required value will be the lowest one.

Calculation of attained EEDI

The EEDI expresses the CO₂ emissions relative to the societal benefits of the transport work performed, giving a simplified version of the EEDI equation:

$$\text{EEDI} \approx \frac{\text{CO}_2}{\text{Transport work}}$$

The CO₂ emissions relative to the transport work can be expressed as the relation of total power $\sum P$ times the specific fuel consumption (SFC) times the carbon content of the fuel type, C_F, taken relative to the capacity times the speed by which this is moved:

$$\text{EEDI} \approx \frac{\sum P \times C_F \times SFC}{\text{Capacity} \times V_{ref}}$$

Phase	0, Jan 2013 -	1, Jan 2015 -	2, Jan 2020 -	3, Jan 2025 -
Reduction factor, X [%]	0	10	20	30

Table 4.01: EEDI reduction factors

$$\begin{aligned}
 &\text{MAIN ENGINE EMISSIONS} \quad \text{AUXILIARY ENGINE EMISSIONS} \quad \text{SHAFT GENERATOR / MOTOR EMISSIONS} \quad \text{EFFICIENCY TECHNOLOGIES} \\
 &\left[\left(\prod_{j=1}^M f_j \right) \left(\sum_{i=1}^{nME} P_{MEi} \cdot SFC_{MEi} \cdot C_{FMEi} \right) + (P_{AE} \cdot C_{FAE} \cdot SFC_{AE}) + \left(\left[\prod_{j=1}^M f_j \cdot \sum_{i=1}^{nPTE} P_{PTEi} - \sum_{i=1}^{noff} f_{offi} \cdot P_{AEi} \right] C_{FAE} \cdot SFC_{AE} \right) - \left(\sum_{i=1}^{noff} f_{offi} \cdot P_{offi} \cdot C_{FME} \cdot SFC_{ME} \right) \right]
 \end{aligned}$$

EEDI-equation

Type of fuel	Carbon content	$C_F, (\text{t CO}_2)/(\text{t fuel})$	Approx. LCV, kJ/kg
Diesel / Gas oil	0.8744	3.206	42,700
Light fuel oil, LFO	0.8594	3.151	41,200
Heavy fuel oil, HFO	0.8493	3.114	40,200
Liquefied petroleum gas, LPG	0.8213	3.015	46,000
Liquefied natural gas, LNG	0.7500	2.750	48,000
Methanol	0.3750	1.375	19,900
Ethanol	0.5217	1.913	26,800

Table 4.02: Carbon factor and lower calorific values for different fuels, as defined by the IMO, Ref. [4.3]. Note that LPG is a mixture of propane and butane, the values given here are for a 50/50 mixture. The primary content of LNG, pure methane, has an LCV of 50,000 kJ/kg

This equation is still a simplification of the overall equation for calculating the index, Ref. [4.3], as seen at the bottom of the left page. This equation is hereafter referred to as the EEDI-equation.

In general, the power of the main engine (P_{ME}) and the specific fuel consumption (SFC) are to be taken at 75% MCR, in g/kWh, also providing the reference ship speed, V_{ref} , at 75% MCR.

For passenger ships, for example a cruise liner, but not a ro-pax, the capacity is equal to the gross tonnage. For container ships, the capacity is set equal to 70% of the maximum deadweight. For all other ships the capacity is equal to the maximum deadweight, Ref [4.3].

The non-dimensional conversion factor (C_F) between tonnes of fuel burned and tonnes of CO_2 produced accounts for the GHG emission of different types of fuels, as shown in Table 4.02, Ref. [4.3].

In the EEDI-equation, f_{eff} refers to the availability of alternative energy generating measures. Correction factors f_f , f_i , f_c , f_l and f_w relates to specific calculations for the ship type and the systems installed onboard, such as ice classed ships, cranes, ramps, etc., and is not treated here, see Ref. [4.3].

EEDI reducing measures

Speed reduction

As explained in “Propeller law and speed power curves” in Chapter 2, the power required to propel the ship is proportional to the speed of the ship by the power of about 3 to 4, $P \propto V^{3 \text{ to } 4}$. This means that if speed is increased from, for example 20 to 21 knots, the required power (placed in the numerator of the EEDI equation) is increased by as much as 25% even if the speed (placed in the denominator) is only increased by 5%.

Accordingly, a small reduction in speed, may result in a large reduction in EEDI. When reducing the speed, and thus the power installed, the designer must make sure that the ship can still manoeuvre safely even in harsh conditions.

Reducing power required

Various steps can be taken to minimise the power required at a given speed. Details on the influence of optimum propeller size and rpm are discussed in Chapter 2. By optimising hull lines and adding energy saving devices additional reductions can be achieved.

Changing fuel type

As seen in Table 4.02, both the carbon factor (C_F) and heating values will influence the amount of CO_2 produced. Higher heating values will result in smaller masses of fuel burned and, hereby, smaller amounts of CO_2 generated, and vice versa. When heating val-

ues are included, an EEDI reduction of, e.g. 23%, is attained for LNG and 8% for methanol.

Pilot oil is injected to ignite most alternative fuels. The amount of this must be included in the final EEDI.

Decreasing specific fuel consumption

For example, a shaft generator can be installed on the main engine to reduce the SFC for producing electric energy. Application of waste heat recovery systems would have a similar effect. Additionally, EcoEGR for Tier II can be considered.

If in the early design phases a ship dimensioned with a four-stroke main engine cannot attain an EEDI lower than required, the implementation of a two-stroke engine with a lower SFC could, depending on the layout of the ship, be a way to secure EEDI compliance.

Capacity

Increasing the capacity while maintaining the total displacement of the ship can also reduce the EEDI. Reducing the lightweight is of most significance to ships with a large lwt/dwt ratio, e.g. lwt/dwt > 1.5 such as ro-pax ships, and of nearly no significance to bulk carriers and tankers, see Table 1.01 (p. 9).

For most ship types, it is generally worth considering the above-mentioned reducing measures prior to this option.

EEDI and light running margin

Due to the EEDI, there is a tendency towards installing less powerful engines. The effect of this can be illustrated by an example of a car that has been fitted with an engine with less power than usual.

The car would definitely have a lower top speed and accelerate slower, and the uphill running capability would be reduced. In those situations, the reduced power would have to be compensated for by using a lower gear ratio. However, a ship with an engine coupled directly to a fixed pitch propeller cannot shift gear. So, to cope with a reduced main engine power, the engine and propeller can be designed with a lower gear ratio, i.e. higher light running margin. In this way, improved manoeuvrability and higher ship speed in adverse conditions are achieved.

Minimum propulsion power

While lowering a ship's installed power has been acknowledged as a method to obtain a lower EEDI value, it has also raised a concern that it could result in underpowered ships with reduced manoeuvrability in heavy weather. As a result of this, the IMO has published an assessment method for determining the minimum propulsion power required to maintain the safe manoeuvrability of ships in adverse conditions, Ref. [4.4].

The evaluation of minimum propulsion power can be performed by assessment level 1 or assessment level 2.

Assessment level 1 allows calculation of the minimum power line value based on ship type and deadweight, with values a and b according to Ref. [4.4]:

$$\text{Minimum Power Line Value} = a \times \text{dwt} + b$$

However, if the propulsion power installed is below the given minimum power line value of assessment level 1, then an evaluation on the ship's design must be performed according to assessment level 2.

It should be noted that this assessment method is currently valid for phase 0 and phase 1 of EEDI. It is expected that it will also be incorporated for EEDI phase 2 which is in force from the 1st of January 2020.

If the ship cannot fulfil the criteria to any of the assessment levels, various options can be considered. Alternative fuels, that lower the EEDI, will allow for a more powerful engine. Hull lines and the bow can be refined to minimise resistance in general and from interaction with waves specifically, etc.

Chapter 5

Examples of engine selections for selected ship types

Deadweight, scantling	50,000 dwt
Deadweight, design	44,000 dwt
Length oa, L_{OA}	185 m
Length bp, L_{PP}	177 m
Breadth	32.2 m
Depth to main deck	19.1 m
Draught, scantling, T_s	12.8 m
Draught, design, T_d	11.6 m
Displacement, scantling	59,700 t
Lightweight	9,700 t
Iwt/dwt	0.19
Block coefficient, C_B scantling	0.82
Block coefficient, C_B design	0.81
Design speed	14.5 kn
Froude number at design speed	0.18
Propeller diameter, d	6.8 m

Table 5.01: Principal parameters of example 1

This chapter exemplifies the selection of main engine(s) for selected ship types based on the engine selection spirals for FP- and CP-propellers.

All calculations are performed with simple, open access tools, to underline the exemplarity of the calculations.

Three ships are considered:

- MR tanker, 50,000 dwt, FPP, with/ without PTO, HFO
- Container carrier, 19,000 teu, FPP with PTO, HFO
- Ro-ro, 6000 lm, twin screw CPP with PTO, LNG

Furthermore the reader is advised to consult the example given for a 320,000 dwt VLCC in the separate paper “Propulsion of VLCC”. Here, important considerations on EEDI and minimum propulsion power are given, which are also relevant for similar-sized bulk carriers.

Example 1 - MR tanker

This example uses a 50,000 dwt MR tanker, and includes considerations on EEDI phase 2 from 2020 and minimum propulsion power.

The tanker is designed for operation on high-sulphur HFO in North America or North Europe, fulfilling both NO_x and

SO_x ECA regulations by NO_x reducing measures and a SO_x scrubber. The design speed is set to 14.5 knots.

The principal parameters of the MR tanker are shown in Table 5.01.

1. Calm water resistance and propulsion coefficients

There are various methods for determining the calm water towing resistance. In this case, the Ship-DESMO program is used, Ref. [2.5] which is based on Ref. [1.2]. The calculated resistance is often validated and optimised by a towing tank test at a later stage of a project.

Apart from the towing resistance, it is necessary to determine the wake fraction coefficient and the thrust deduction coefficient used to establish the hull and rotative efficiency as described in Chapter 2. For a first estimate the following formulas could be used, see also Table 2.02 (p. 22).

$$w = 0.5 \times C_B - 0.05 = 0.35 \text{ and}$$

$$t = 0.27 \times C_B = 0.22 \Rightarrow \eta_H = \left(\frac{1-t}{1-w} \right) = 1.2$$

2. Light propeller curve

The light propeller curve is calculated for the design condition, based on the calm water towing resistance, propulsion coefficients, and the propeller characteristics, see Fig. 5.01. Preliminary values for propeller data could be obtained from the Wageningen B-series or another propeller-series. For more accurate values of resistance and propeller characteristics, tank test data, or CFD analysis of the actual ship design can be applied.

Considering that the propeller must be submerged when the MR tanker is sailing in ballast, the maximum propeller diameter is set to $d = 6.8 \text{ m}$, $d / T_d = 0.59$ with four blades.

The shafting efficiency is set to 99% as the engine is located aft and the shaft is short.

At design speed, the light propeller curve demands 6,570 kW at 88.2 rpm.

3. Propulsion margins

Initially, a sea margin of 15% and an engine margin of 15% is set for the ship.

The light running margin is set to 5%.

The combined inclusion of margins result in an SMCR = 8,900 kW at 93 rpm, Fig. 5.01, see p. 33 for the relevant equations.

4. SMCR on engine layout diagram

Using the CEAS calculation tool, which can be accessed from the MAN Energy Solution website → Two-stroke → CEAS Engine Calculations, the SMCR is plot-

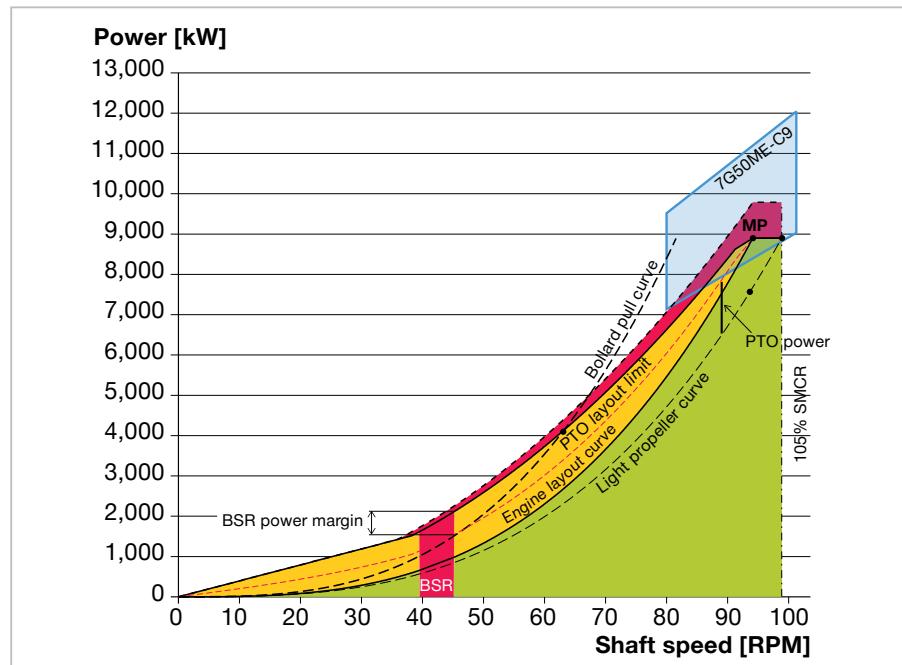


Fig. 5.01: Light propeller curve, margins and layout curve for SMCR = 8,900 kW with load diagram and layout diagram for a potential engine for a 50,000 dwt MR tanker. Maximum available PTO power at 75% MCR is approx. 1,250 kW

ted on the engine layout diagram of various possible engines, one of which is illustrated in Fig. 5.01.

This allows plotting of SFOC values as a function of engine load, see Fig. 5.02, in this case in Tier III mode.

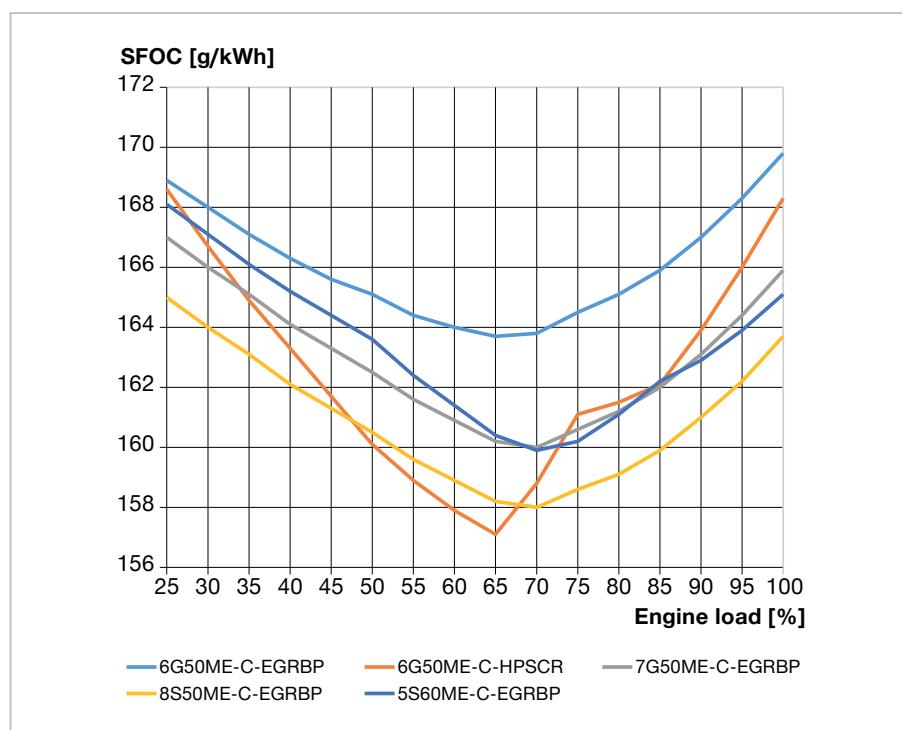


Fig. 5.02: SFOC values of possible engines for a 50,000 dwt MR tanker in Tier III mode with a scrubber installed and a SMCR of 8,900 kW at 93 rpm, as of 2018. Values may change, please use CEAS for specific values

5. Select engine

Usually, the most derated engine will result in the lowest fuel consumption over the full range but also the highest initial cost as well as the largest engine dimensions.

Additionally, it is seen in Fig. 5.02 that SCR will result in low fuel consumption while operating in Tier III mode. However, SCR may still have the highest operating costs in Tier III mode, as the SCR plant uses urea. In Tier II mode, which is of significance to EEDI calculations, EGR offers the potential of implementing EcoEGR, which reduces SFOC further.

The optimum choice of engine will depend on the priorities of the project. A sensible compromise between operational (OPEX) and capital (CAPEX) expenses could in this case be the 7G50ME-C9. This will be used for the first iteration through the engine selection spiral.

6. Passage of barred speed range

Quick passage of a barred speed range can in some cases be a challenge for 5 and 6-cylinder engines. Here, a 7-cylinder engine is employed and, therefore, the BSR is expected to be passed quickly because the BSR is placed at a relatively low rpm relative to SMCR-rpm and the required propeller power therefore is quite low.

For this example ship, the highest rpm in the BSR is estimated to be at 48% of SMCR rpm, see Fig 5.01. The location of the BSR and the ability to pass it quickly should be evaluated specifically for each project.

The barred speed range power margin (BSR_{PM}) is calculated as described in the corresponding step in the FPP selection spiral in Chapter 3, and is well above the recommendation of minimum 10%.

$$BSR_{PM} = \frac{P_L - P_P}{P_P} \times 100$$

$$= \frac{2,100 - 1,500}{1,500} \times 100 = 40\%$$

7. Standard engine load diagram

The selected SMCR and the resulting *load* diagram is included in Fig. 5.01.

8. Compliance with regulation

The operational profile of an MR tanker operating near North America or North Europe will involve operation within ECA zones. A scrubber has been selected for SO_x compliance, but an alternative option would, depending on the operational profile, be low-sulphur fuel.

An EGR solution is selected as the NO_x reducing measure in this example, one of the reasons being the saving potential offered by EcoEGR.

With regard to EEDI, the ship considered here is to be built to phase 2 standards (20% reduction from the EEDI reference line) implemented from 2020.

With a capacity of 50,000 dwt, the required EEDI can be calculated by first establishing the reference line value for a tanker, as set by Ref. [4.3]:

$$\text{Reference line value} = a \times b^{-c} = 1218.8 \times 50000^{-0.488} = 6.21$$

With the reduction from phase 2 resulting in a required EEDI of:

$$\text{Required EEDI} = \left(1 - \frac{X}{100}\right) \times \text{Reference line value} = \left(1 - \frac{20}{100}\right) \times 6.21 = 4.97$$

Through the same power prediction method as applied in step 1, applying Ref. [2.5], the reference speed at 75% engine load and 100% capacity utilisation in trial condition is established, $V_{ref} = 14.30$ knots.

The attained index calculated is based on the EEDI equation in Chapter 4, in this case for MDO. The SFOC of the main engine at 75% MCR in Tier II mode is used for the calculation. The SFOC_{ME} is obtained through CEAS, and the 6% tolerance on this number is included in the calculations. In this example the SFOC_{AE} of a smaller auxiliary engine at 50% load is estimated to 220 g/kWh including tolerance. The electric power to use in the EEDI equation is calculated to 445 kW, Ref. [4.3].

As seen, the MR tanker does not fulfil the phase 2 requirements for EEDI as the attained value of 5.38 > 4.97.

Before considering EEDI reducing options, it is sensible to evaluate the ship's capabilities with regard to minimum propulsion power at assessment level 1, considering ship type and dwt.

The MR tanker does not fulfil the MPP requirements at assessment level 1 (8,900 kW < 9,220 kW), and will instead have to be evaluated by assessment level 2. By assessing the power installed on similar MR tankers built, it is expected that compliance with assessment level 2 will be possible with a power lower than 9,220 kW.

An often relatively simple solution to reduce EEDI is to install a PTO, as this both reduces the SFOC for generating electric power, and allows for some of the main engine power to be allocated to the electric power production. This allocation of main engine power leads to a reduction in reference speed.

The required rated electric "name plate" output of the PTO to be able to cover 445 kW in the EEDI calculation is $445 / 0.75 / 0.75 = 791$ kW.

The engine selection spiral can be re-entered in step 7 to ensure that the 8,900 kW engine is capable of delivering the desired electric power, see the distance between the PTO_{layout limit} and the light propeller curve on Fig. 5.01.

With a PTO, the EEDI_{PTO} of 5.04 is now closer to the EEDI phase 2 requirement of 4.97 for this ship type and size.

$$\text{Attained EEDI} = \frac{(P_{ME} \times 0.75) \times C_{FME} \times (\text{SFOC}_{ME}) + P_{AE} \times C_{FAE} \times \text{SFOC}_{AE}}{\text{capacity} \times V_{ref}}$$

$$\text{Attained EEDI} = \frac{(8,900 \times 0.75) \times 3.206 \times (156 \times 1.06) + 445 \times 3.206 \times 220}{50,000 \times 14.30} = 5.38$$

$$\text{Minimum Power Line Value} = a \times \text{dwt} + b = 0.0652 \times 50,000 + 5,960.2 = 9,220 \text{ kW}$$

$$\text{Attained EEDI}_{\text{PTO}} = \frac{(8,900 - 445 / 0.75) \times 0.75 + 445) \times 3.206 \times (156 \times 1.06)}{50,000 \times 14.04} = 5.04$$

$$\text{Attained EEDI}_{\text{PTO+EcoEGR}} = \frac{(8,900 - 445 / 0.75) \times 0.75 + 445) \times 3.206 \times (151 \times 1.06)}{50,000 \times 14.04} = 4.90$$

To ensure compliance with the regulations, different options exist for further EEDI reductions:

EcoEGR can easily be implemented to lower the Tier II SFOC, as the ship is already dimensioned with EGR for Tier III compliance. Hereby an EEDI_{PTO+EcoEGR} of 4.90 is attained as shown above.

As an alternative, the engine margin can be reduced from e.g. 15% to 12%, leading to a SMCR of 8,600 kW, and an attained index of EEDI_{PTOredEM} = 4.90.

Other EEDI reducing measures could also be considered. For instance a Kappel propeller, rudder bulb, and/or a wake equalising duct, could also be relevant, in order to reduce required power for a given ship speed, and thereby the EEDI.

Considerations on such energy saving devices, further derating, and possibly additional speed reductions will be required to meet EEDI phase 3, if alternative fuels are not considered.

Example 2 - container carrier

This example considers a 19,000 teu container carrier, illustrating the inclusion of a PTO for ships with a high electric consumption. The principal parameters of the container carrier are shown in Table 5.02.

1. Calm water resistance

Various measures exist for determining the calm water resistance. Here, the Ship-DESMO program is used, Ref. [2.5] which is based on Ref. [1.2].

2. Light propeller curve

The light propeller curve is determined using the same methods as in example 1.

Considering limitations of propeller casting and the fact that the propeller must be submerged in all conditions, the maximum diameter of the propeller has been set to $d = 10.7$ with 5 blades. This is on the upper edge of current designs with a d/T_d -ratio of 0.74.

3. Propulsion margins

Initially a sea margin of 20% is set for the ship with an engine margin of 15%. Relative high margins are desirable in order to be able to catch up delays.

The light running margin is set to 4%, the lowest recommendable value, due to the slender hull lines of this ship and the corresponding relatively low increase in resistance in heavy weather.

The result is an SMCR point at 50,350 kW and 76.7 rpm.

4. SMCR on engine layout diagram

Using the CEAS online calculation tool, the SMCR is plotted on the engine layout diagram of various possible engines, see Fig. 5.03.

5. Select engine

Depending on the priorities of the project, the optimum engine can be selected based on the CEAS reports. For this case, the minimum fuel consumption is wanted, for which reason a 10G95ME-C10-EGRTC is selected. This is the

Deadweight, scantling	185,000 dwt
Deadweight, design	155,000 dwt
Length oa, L_{OA}	400 m
Length bp, L_{PP}	383 m
Breadth	58.50 m
Depth to main deck	30.5 m
Draught, scantling, T_s	16 m
Draught, design, T_d	14.5 m
Displacement, scantling	240,000 t
Lightweight	55,000 t
Iwt/dwt	0.30
Block coefficient, C_B scantling	0.65
Block coefficient, C_B design	0.63
Design speed	21 kn
Froude number at design speed	0.175
Propeller diameter, d	10.7 m

Table 5.02: Principal parameters of example 2

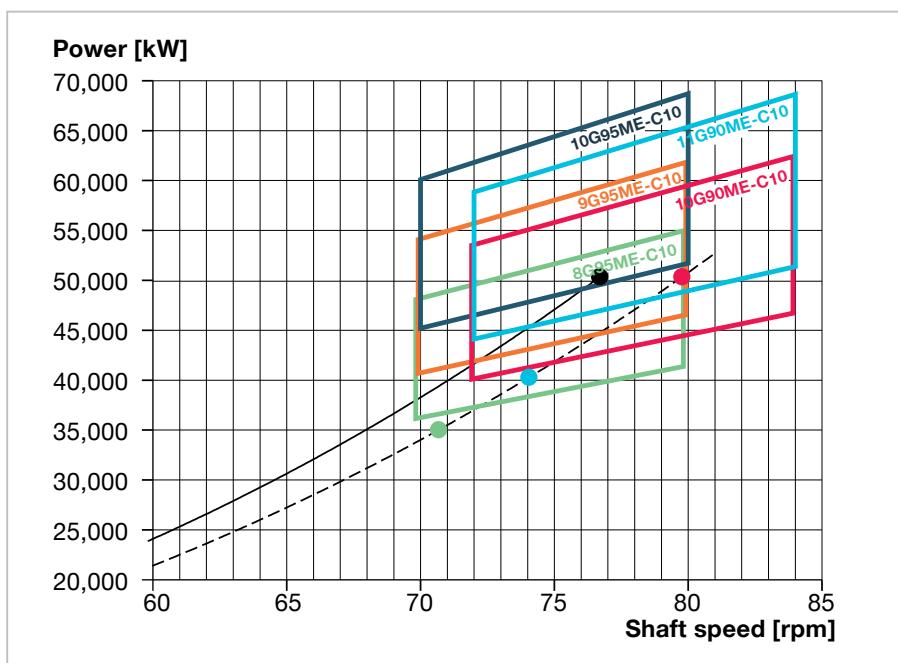


Fig. 5.03: Layout diagrams for selected engines possible for the example 19,000 teu container carrier

most derated engine. At 75% SMCR this engine has a 3% lower SFOC than an 8G95ME-C10-EGRTC.

When evaluating the lower fuel consumption of a derated engine, it is important to consider the increased lubri-

cation oil consumption from added cylinders. Even considering this, the OPEX saving will be substantial in this example.

6. Passage of barred speed range

The barred speed range will, when considering the many cylinders of the engine and the long shaft line for such a ship, be located at a low rpm. The BSR power margin will be large and the BSR passage will therefore be quick.

7. Standard engine load diagram

As a PTO is desired, the engine *load* diagram is of special interest in this case. A 6-MW PTO is sought included. The maximum power demanded by the light propeller curve and PTO combined is to be designed to the $PTO_{layout\ limit}$:

$$PTO_{layout\ limit} = P_{SMCR} \times \left(\frac{n}{n_{SMCR}} \right)^{2.4}$$

When considering the difference between the $PTO_{layout\ limit}$ and the light propeller curve (see Fig. 5.04) over the full rpm-range as it is plotted on Fig. 5.05, the availability of PTO power can be investigated. When operating along the light propeller curve and using, for example, 75% MCR for propulsion, the engine will operate at 92% rpm. Fig. 5.05 shows that 5,400 kW is available for a PTO in this case.

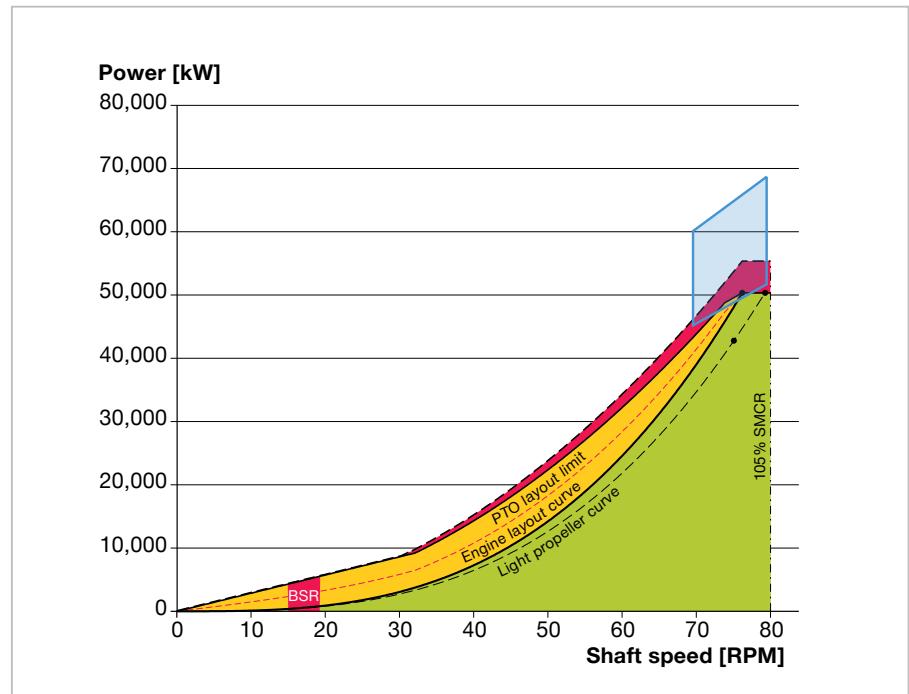


Fig. 5.04: Engine load diagram with $PTO_{layout\ limit}$ and layout diagram of the 10G95ME-C10-EGRTC

In order to increase the maximum power available for the PTO to the level desired in this example, and to minimise the ratio of heavy operation (yellow area) while the PTO is engaged, the en-

gine selection spiral is re-entered at step 4. The SMCR power is increased by 2,000 kW, SMCR-rpm is maintained, ensuring a PTO availability as shown in Fig 5.05, right.

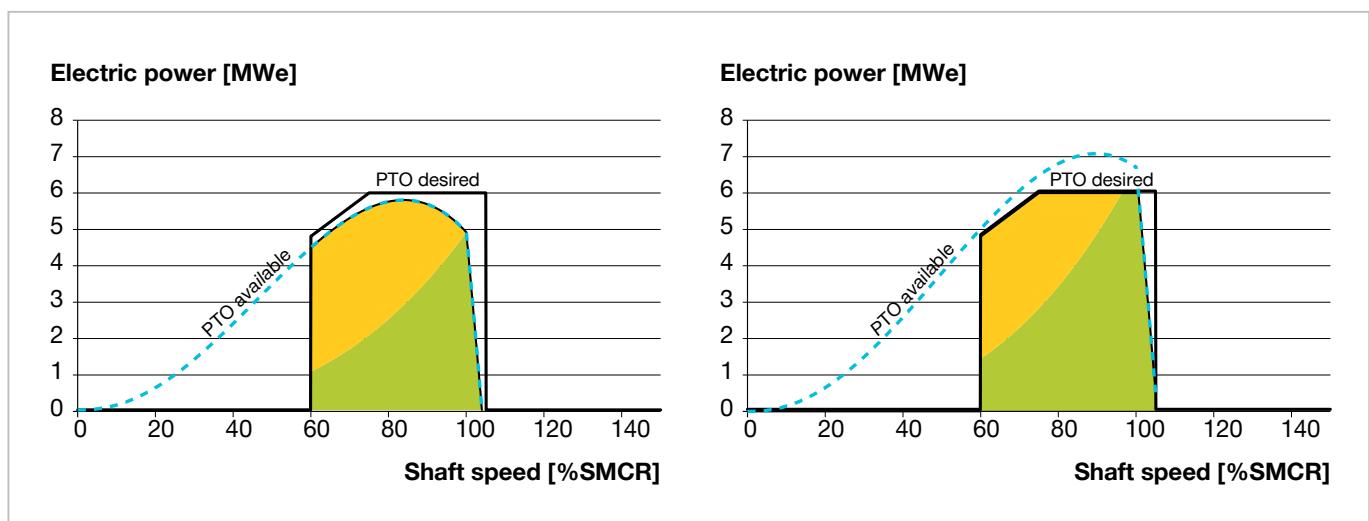


Fig. 5.05: PTO availability over the full speed range with the original (left) and increased SMCR (right)

As the SMCR point moves up, the margin to the light propeller curve (i.e. the light running margin) increases from 4 to 5.4%.

The implications of the increased SMCR to the selection of the optimal engine is investigated (through the generation of new CEAS reports) without any changes to the previous conclusion: The selection of a 10G95ME-C10-EGRTC engine as optimal is unchanged.

8. Compliance with regulation

The container carrier is considered to be built after the implementation of the NO_x ECA in the North Sea and Baltic Sea in 2021, and therefore the ship is equipped with EGR, also offering the potential of EcoEGR.

Compliance with global SO_x regulations are achieved by the application of a SO_x scrubber.

The reference EEDI is calculated, with phase 2 (20%) reduction, Ref. [4.3].

The reference speed at 70% dead-weight utilisation (for container carriers only) and 75% MCR in trial condition is calculated to 21.69 knots, Ref. [2.5] and 21.42 knots with PTO.

The attained index is calculated based on the EEDI equation in Chapter 4. The method is similar to example 1, the PTO is included along with the 6% SFOC tolerance.

As seen below, the attained EEDI is well below the required EEDI.

As of 2018, the IMO has not specified rules for container carriers with regard to minimum propulsion power. A container carrier has a significantly higher design speed than bulk carriers and tankers, and is hereby considered to have, sufficient installed power for all relevant conditions.

$$\text{Required EEDI} = a \times b^{-c} = 174,22 \times 185,000^{-0.201} \times \left(1 - \frac{20}{100}\right) = 15.22 \times \left(1 - \frac{20}{100}\right) = 12.18$$

$$\text{Attained EEDI} = \frac{\left(52,350 - \frac{1,559}{0.75} \times 0.75 + 1,559\right) \times 3.206 \times (152.1 \times 1.06)}{129,500 \times 21.42} = 7.317$$

Example 3 - ro-ro cargo

A ro-ro cargo ship with CP-propellers and PTO is considered, including the positive influence on EEDI by applying LNG as fuel. The principal parameters of the 6,000 lm ro-ro cargo ship are shown in Table 5.03.

1. Calm water resistance

Again the Ship-DESMO program is used for establishing the calm water resistance, Ref. [2.5].

2. Possible propeller operation for CPP & required power

The ship is equipped with two CP-propellers, and it is assumed that the control system is capable of controlling the pitch independently of propeller rpm.

As an initial estimate, the power required has been calculated for a FP-propeller and since corrected for an assumed 1.5% less effectiveness due to the larger hub of the CP propeller. The application of a well-designed rudder bulb (see Chapter 2) would in most cases reduce the difference between FPP and CPP efficiency significantly, as a rudder bulb eliminates the low pressure behind the hub.

The draught is limited because the cargo is light, which also limits the maximum size of the propeller.

Considering that there will be no significant difference between ballast and loaded condition, as cargo amounts are assumed to be equal on both crossings, and the fact that the lightweight constitutes the major part of the dis-

Deadweight, scantling	15,000 dwt
Deadweight, design	14,000 dwt
Length oa. L_{OA}	228 m
Length bp. L_{PP}	220 m
Breadth	33 m
Depth to main deck	10.5 m
Draught, scantling, T_s	8.0 m
Draught, design, T_d	7.7 m
Displacement, scantling	34,000 t
Lightweight	19,000 t
Iwt/dwt	1.27
Block coefficient, C_B scantling	0.57
Block coefficient, C_B design	0.57
Design speed	22 kn
Froude number at design speed	0.24
Propeller diameter, d	6.1 m

Table 5.03: Principal parameters of example 3

placement, the propeller size is set relatively large to $d = 6.1$ m with 4 blades. This gives a ratio of $d/T_d = 0.79$, which will require special considerations when designing the aft hull lines.

Twin-screw ships can be designed as a regular hull or as a twin-skeg hull, in this case the first option is selected.

The transmission efficiency is estimated at 99%. The propeller design point (PD) without any margins is located at PD = 10,025 kW at 117 rpm per shaft for a ship speed of 22 knots.

3. CPP operating principles for inclusion of PTO

As the ro-ro ship is intended for a large capacity of reefer units, a PTO is intended on each shaft.

In order to maximise the flexibility of the ro-ro ship and minimise the fuel consumption as much as possible, a solution allowing for variable rpm is selected. This is at the cost of including power electronics, for the conversion of the variable electric frequencies generated by the PTO to the constant frequency of the ship's electric grid.

4. Propulsion margins for CPP

Initially, a sea margin of 25% is set for the ship with an engine margin of 15%. The ro-ro ship is intended for scheduled traffic in the North Sea and therefore a large sea margin is included in order to avoid or catch up delays.

As the CPP control system is assumed to be able to control the pitch independently of rpm, no light running margin is required. If a fixed combinator curve were followed, a rather high LRM, of e.g. 7% if possible, would be relevant as heavy seas can be encountered in the intended operation area, as well as the ship is intended for PTO operation.

Using the margins, the SMCR power and rpm requirements are calculated.

5. Engine layout diagram with SMCR for CPP

The following engines are possible candidates for this project based on the power and rpm requirements calculated:

S50ME-C9-GI with L1 at 117 rpm and S50ME-C8-GI with L1 at 127 rpm.

A high load pitch increase is a typical feature for CP-propellers with a combinator curve. The high load pitch increase is typically applied for the last 10 to 15% power, see Fig. 5.06, left. In order to account for the lower L1 rpm of the S50ME-C9-GI engine design, high load pitch increase is applied to a larger extent for this design, see Fig. 5.06, right.

As the pitch is increased at high loads, it will be slightly off the design pitch and result in a small reduction in efficiency. For reasons of simplicity further considerations of this have been omitted in this example.

$$\text{SMCR}_{\text{power}} = \text{PD} \times \frac{\frac{100 + \text{SM}}{100}}{\frac{100 - \text{EM}}{100}} = 10,025 \times \frac{\frac{100 + 25}{100}}{\frac{100 - 15}{100}} = 14,750 \text{ kW}$$

$$\text{SMCR}_{\text{rpm}} = 117 \times \left(\frac{14,750}{10,025} \right)^{1/3} = 133 \text{ rpm}$$

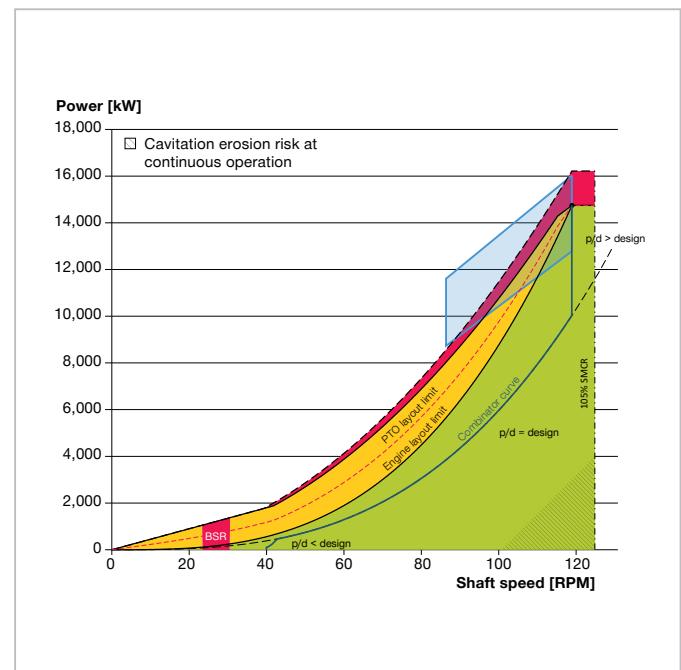
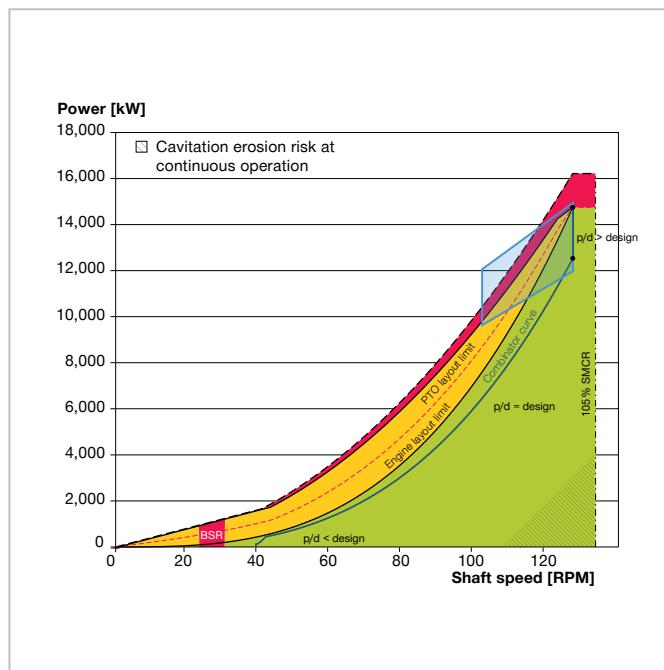


Fig. 5.06: The light propeller curve with conceptual combinator curves plotted on engine load diagram with the layout diagram of the S50ME-C8-GI and S50ME-C9-GI respectively

6. Select engine for CPP

In CEAS there is an FPP option with fuel consumption given along the engine layout curve, and a CPP option with fuel consumption given for constant rpm operation.

In this simple example, the fuel consumption along a conceptual combinator curve can be calculated by interpolating between the SFOC on the engine layout curve and the SFOC on the constant rpm (equal to SMCR-rpm) curve. Both curves can be obtained through CEAS, the result is shown in Fig. 5.07 for Tier III mode.

The S50ME-C9-GI design will be approx. 4.5% more efficient overall and will be preferred. A significant margin is left to the S50ME-C8-GI design, even if a slight reduction in efficiency as a result of the extended high load pitch increase is considered.

7. Engine load diagram for CPP and considerations of PTO power

The difference between the selected combinator curve and the $PTO_{layout\ limit}$ is evaluated, see Fig. 5.06. This indicates up to approx. 3,000 kW capacity per shaft due to the high margins included in this example.

In situations at sea where the margins are exploited and all main engine power is utilised for propulsion, the auxiliary engines on-board must take over the electric load.

8. Compliance with regulation

The ro-ro ship is considered to be constructed after the implementation of the NO_x ECA in the North and Baltic Sea in 2021, and will have to comply with both the NO_x and SO_x ECA regulations in the area. The EEDI reference line with phase 2 (20%) reduction is calculated, Ref. [4.3]. Note that for ro-ro cargo ships the value of the parameter a set for phase 2 and 3 is different from the value set for phase 0 and 1.

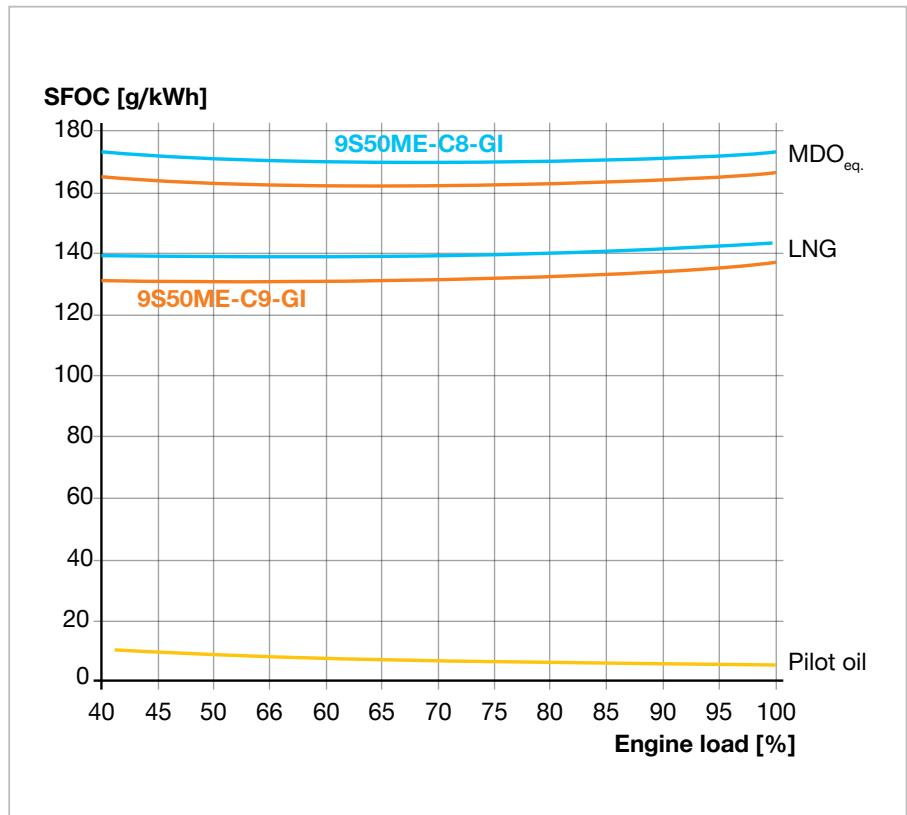


Fig. 5.07: SFOC along the possible combinator curves shown in Fig. 5.06 for Tier III mode. MDO equivalents, LNG, and pilot oil consumption (similar for both engines) are shown separately

$$\text{Reference line value} = a \times b^{-c} = 1686.17 \times 15,000^{-0.498} \times \left(1 - \frac{20}{100}\right) = 11.23$$

The electric consumption at sea is calculated to be 988 kW, Ref. [4.3]. The reference speed at scantling draught and 75% SMCR is calculated to be 22.73 knots without PTO and 22.47 knots with PTO, using Ref. [2.5].

The attained index is calculated based on the EEDI equation in Chapter 4, considering PTO, the 6% tolerance on the SFOC values, and the f_j correction factor for ro-ro cargo ships.

The calculated correction factor of $f_j = 0.3159$ applied to the main engine related term is only displayed here, the calculation is described in Ref. [4.3].

As the specific gas consumption of the engine is stated for pure methane with an $LCV_{methane} = 50 \text{ MJ/kg}$, the specific gas consumption must be corrected to correspond to the LCV value of LNG stated by the IMO, $LCV_{LNG} = 48 \text{ MJ/kg}$, in order to apply the carbon factor of LNG, $C_{F,LNG} = 2.75$, as determined by the IMO.

This correction is represented by the 50/48 term in the calculation of the EEDI attained for LNG, see also Table 4.02 (p. 51).

For reference, the EEDI for MDO is calculated as well. A significant reduction is attained for LNG with MDO as pilot oil.

If the design speed needs to be kept at 22 knots due to a desired operational schedule, LNG is a good option for compliance with EEDI phase 2 and especially for phase 3 (30% reduction), where an index below 9.82 must be attained.

If traditional fuels are employed, energy saving devices and/or a reduction of sea and engine margins will be necessary for compliance with EEDI phase 2.

Similar to example 2, no considerations are currently required with regard to minimum propulsion power for this type of ship.

$$\text{Attained EEDI}_{MDO} = \frac{\left((29,500 - \frac{987.5}{0.75}) \times 0.75 \times f_j + 987.5 \right) \times 3.206 \times 157.8 \times 1.06}{15,000 \times 22.47} = 12.19$$

$$\text{Attained EEDI}_{LNG} = \frac{\left((29,500 - \frac{987.5}{0.75}) \times 0.75 \times f_j + 987.5 \right) \times \left(3.206 \times 6.7 + 2.75 \times 129.0 \times \frac{50}{48} \right) \times 1.06}{15,000 \times 22.47} = 9.42$$

Closing remarks

In practice, the calculated resistance of the ship will frequently be checked against the results obtained by testing a model of the ship in a towing tank. The experimental tank test measurements are also used for optimising the propeller and hull design along with CFD simulations.

The interaction between ship and main engine is important, in order to achieve the lowest possible fuel consumption for the specific ship design. Different paths for optimising the design can be taken by the ship designer. This also explains why two tender designs for the same ship never look the same.

When the ship's necessary power requirement, including margins, and the propeller's speed (rate of revolution, typically rpm) have been determined, the capabilities of different possible main engines can be investigated. MAN Energy Solutions' online engine selection programme CEAS supports this.

The placing of the engine's load diagram, i.e. the choice of engine layout in relation to the engine's (ship's) operational propeller curve, must be made carefully in order to achieve the optimum propulsion plant.

The engine selection spirals for FP and CP propellers, respectively, will ease this optimisation, including considerations of a possible engine driven shaft generator.

Thanks to the flexibility of the layout and load diagrams for the MAN B&W two-stroke engines, a suitable solution will always be at hand.

For questions to specific cases MAN Energy Solutions can be contacted at LEE5@man-es.com.

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List of abbreviations

BSR	barred speed range	MARPOL	The International Convention for the Prevention of Pollution from Ships
BSR _{PM}	barred speed range power margin	MCR	maximum continuous rating
CAPEX	capital expenditure	mep	mean effective pressure
CEAS	computerised engine application system	MPP	minimum propulsion power
CP	controllable pitch	NCR	normal continuous rating
CPP	controllable pitch propeller	NMCR	nominal maximum continuous rating
DF	dual fuel	OPEX	operating expense
DLF	dynamic limiter function	PD	propeller design point
dwt	deadweight tonnage	PM	particulate matter
ECA	emission control area	PTI	power take in (shaft motor)
EEDI	energy efficiency design index	PTO	power take out (shaft generator)
EGR	exhaust gas recirculation	SCR	selective catalytic reduction
EM	engine margin	SFC	specific fuel consumption
FP	fixed pitch	SFOC	specific fuel oil consumption
FPP	fixed pitch propeller	SM	sea margin
GHG	greenhouse gasses	SMCR	specified maximum continuous rating
IMO	International Maritime Organisation	WHR	waste heat recovery
LCB	longitudinal centre of buoyancy		
LCF	longitudinal centre of flotation		
LNG	liquefied natural gas		
LRM	light running margin		
Iwt	lightweight tonnage		

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