Basic Principles of Ship Propulsion

MAN B&W

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Introduction

For the purpose of this paper, the term "ship" is used to denote a vehicle employed to transport goods and persons from one point to another over water. Ship propulsion normally occurs with the help of a propeller, which is the term most widely used in English, although the word "screw" is sometimes seen, inter alia in combinations such as a "twin-screw" propulsion plant.

Today, the primary source of propeller power is the diesel engine, and the power requirement and rate of revolution very much depend on the ship's hull form and the propeller design. Therefore, in order to arrive at a solution that is as optimal as possible, some general knowledge is essential as to the principal ship and diesel engine parameters that influence the propulsion system.

This paper will, in particular, attempt to explain some of the most elementary terms used regarding ship types, ship's dimensions and hull forms and clarify some of the parameters pertaining to hull resistance, propeller conditions and the diesel engine's load diagram.

On the other hand, it is considered beyond the scope of this publication to give an explanation of how propulsion calculations as such are carried out, as the calculation procedure is extremely complex. The reader is referred to the specialised literature on this subject, for example as stated in "References".

Scope of this paper

This paper is divided into three chapters which, in principle may be considered as three separate papers, but which also with advantage may be read in close connection to each other. Therefore, some important information mentioned in one chapter may well appear in another chapter too.

Chapter I, describes the most elementary terms used to define ship sizes and hull forms such as, for example, the ship's displacement, deadweight, design draught, length between perpendiculars, block coefficient, etc. Other ship terms described include the effective towing resistance, consisting of frictional, residual and air resistance, and the influence of these resistances in service.

Chapter II deals with ship propulsion and the flow conditions around the propeller(s). In this connection the wake fraction coefficient and thrust deduction coefficient, etc. are mentioned.

The total power needed for the propeller is found based on the above effective towing resistance and various propeller and hull dependet efficiencies which are also described. A summary of the propulsion theory is shown in Fig. 4, page 7.

The operating conditions of a propeller according to the propeller law are described for free sailing in calm weather, and followed up by the relative heavy/light running conditions which apply when the ship is sailing and subject to different types of extra resistance like fouling, heavy sea against etc.

Chapter III elucidates the importance of chosing the correct specified MCR and optimising point of the main engine, and thereby the engine's load diagram in consideration to the propeller's design point. The construction of the relevant diagram lines is described in detail by means of several examples. Fig. 15, page 21 shows, by means of a load diagram, the important influence of different types of ship resistance on the engine's continuous service rating.

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Chapter I

Ship Definitions and Hull Resistance

Ship types

Depending on the nature of their cargo, and sometimes also the way the cargo is loaded/unloaded, ships can be divided into different categories, classes, and types, some of which are mentioned in Table 1.

The two largest categories of ships are bulk carriers (for bulk goods such as grain, coal, ores, etc.) and tankers, which again can be divided into more precisely defined classes and types. Thus, tankers can be divided into oil tankers, gas tankers and chemical tankers, but there are also combinations, e.g. oil/chemical tankers.

Table 1 provides only a rough outline. In reality there are many other combinations, such as "Multi-purpose bulk container carriers", to mention just one example.

A ship's load lines

Painted halfway along the ship's side is the "Plimsoll Mark", see Fig. 1. The lines and letters of the Plimsoll Mark, which conform to the freeboard rules laid down by the IMO (International Maritime Organisation) and local authorities, indicate the depth to which the vessel may be safely loaded (the depth varies according to the season and the salinity of the water).

There are, e.g. load lines for sailing in freshwater and seawater, respectively, with further divisions for tropical conditions and summer and winter sailing. According to the international freeboard rules, the summer freeboard draught for seawater is equal to the "Scantling draught", which is the term applied to the ship's draught when dimensioning the hull.

Category	Class	Type	
Tanker	Oil tanker Gas tanker Chemical Tanker OBO	Crude (oil) Carrier Very Large Crude Carrier Ultra Large Crude Carrier Product Tanker Liquefied Natural Gas Liquefied Petroleum Gas Oil/Bulk/Ore	(CC) (VLCC) (ULCC) (LNG) (LPG) (OBO)
Bulk Carrier	Bulk Carrier		
Container Ship	Container Ship	Container Carrier Roll On-Roll Off	 (Ro-Ro)
General Cargo Ship	General Cargo Coaster		
Reefer	Reefer		
Passenger Ship	Ferry Cruise Vessel		

Table 1: Examples of ship types

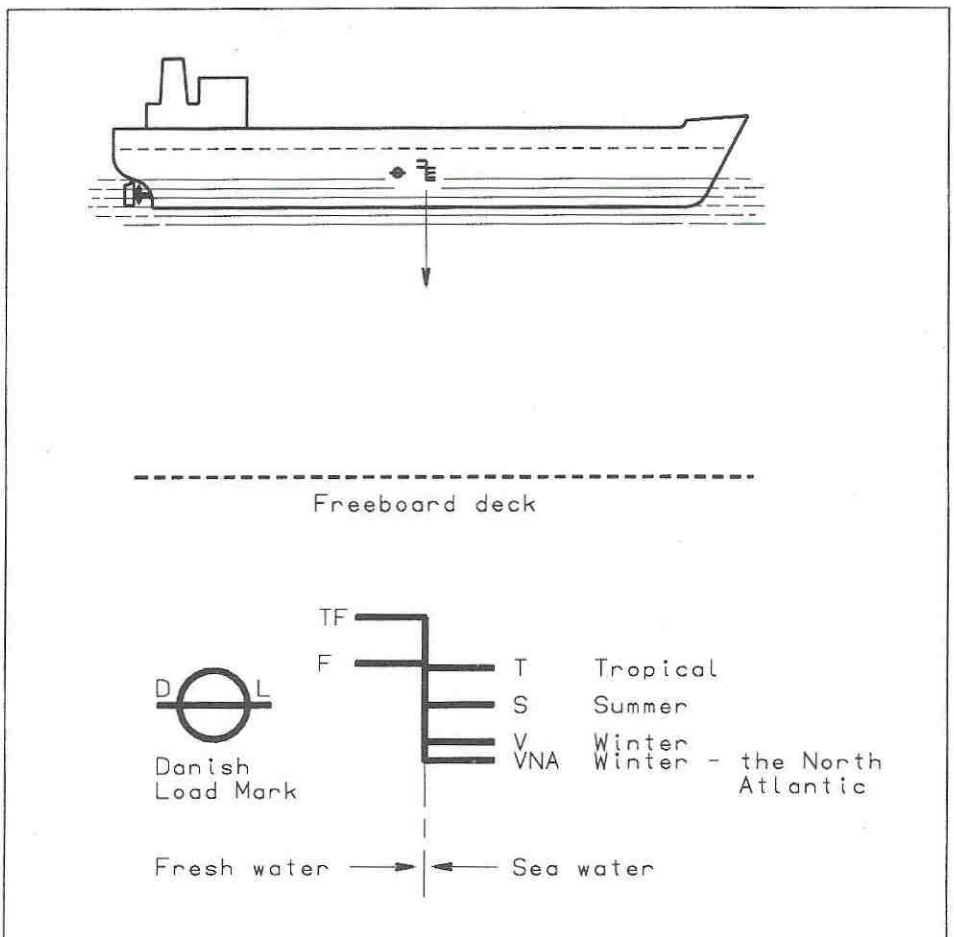

Fig. 1: Load lines – freeboard draught

The winter freeboard draught is less than that valid for summer because of the risk of bad weather whereas, on the other hand, the freeboard draught for tropical seas is somewhat higher than the summer freeboard draught.

Indication of a ship's size

Displacement and deadweight

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, all told, of the relevant loaded ship, normally in seawater with a mass density of 1.025 t/m^3.

Displacement comprises the ship's light weight and its deadweight, where the deadweight is equal to the ship's loaded 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 light weight, all given in tons:

deadweight = displacement – light weight

Incidentally, 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 approximately 907 kg.

The light weight of a ship is not normally used to indicate the size of a ship, whereas the deadweight tonnage (DWT), based on the ship's loading capacity measured in tons at summer freeboard draught, often is.

Sometimes, the deadweight tonnage may also refer to the design draught of the ship but, if so, this will be mentioned. Table 2 indicates the rule-of-thumb relationships between the ship's displacement, deadweight tonnage (summer freeboard) and light weight.

Ship type	DWT/light weight ratio	Displ./DWT ratio
Tanker and Bulk Carrier	6	1.17
Container ship	2.5 – 3.0	1.33 – 1.4

Table 2: Examples of relation between displacement, deadweight tonnage and light weight

A ship's displacement can also be expressed as the volume of displaced water ∇, i.e. in m^3.

Gross Register Tons

Without going into detail, it should be mentioned that there are also such measurements as Gross Register Tons (GRT), and Net Register Tons (NRT) where 1 register ton = 100 English cubic feet = 2.83 m^3.

These measurements express the size of the internal volume of the ship in accordance with the given rules for such measurements, and are extensively used for calculating harbour and canal dues/charges.

Description of hull forms

It is evident that the part of the ship which is of significance for its propulsion is the part of the ship's hull which is under the water line. The dimensions below describing the hull form refer to the design draught, which is less than, or equal to, the summer freeboard draught. The choice of the design draught depends on the degree of load, i.e. whether, in service, the ship will be lightly or heavily loaded. Generally, the most frequently occurring draught between the fully-loaded and the ballast draught is used.

Ship's lengths L_{OA}, L_{WL}, and L_{PP}

The overall length of the ship, L_{OA}, is normally of no consequence when calculating the hull's water resistance. The 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. 2.

The length between perpendiculars is the length between the foremost perpendicular, i.e. usually a vertical line through the stem's intersection with the water line, and the aftmost perpendicular which, normally, coincides with the

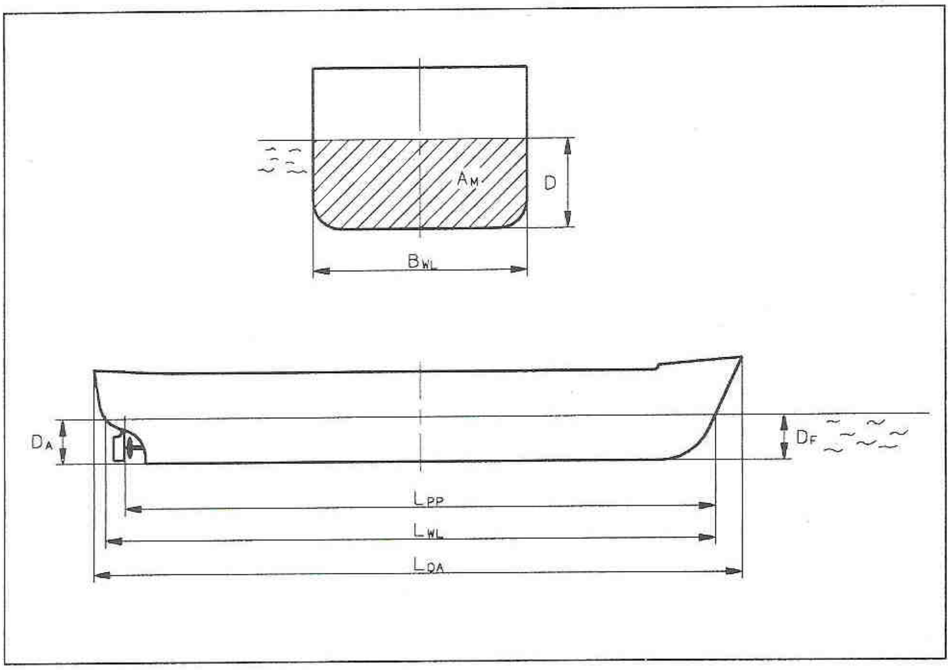

Fig. 2: Hull dimensions

rudder axis. Generally, this length is slightly less than the waterline length, and is often expressed as:

$$L_{PP} = 0.97 \times L_{WL}$$

Draught D
The ship's draught, D, is defined as the vertical distance from the waterline to that point of the hull which is deepest in the water, see Fig. 2. The foremost draught, D_F, and aftmost draught, D_A, are normally the same when the ship is in the loaded condition.

Breadth on waterline B_{WL}
Another important factor is the hull's largest breadth on the waterline, B_{WL}, See Fig. 2.

Block coefficient C_B
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 $L_{WL} \times B_{WL} \times D$, see Fig. 3, i.e.

$$C_B = \frac{\nabla}{L_{WL} \times B_{WL} \times D}$$

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

A small block coefficient means less resistance and, consequently, the possibility of attaining higher speeds.

Table 3 shows some examples of block coefficient sizes, and the pertaining service speeds, on different types of ships. It shows that large block coefficients correspond to low speeds and vice versa.

Ship type	Block coefficient	Approximate speed in knots
Lighter	0.90	5 – 10
Bulk carrier	0.80 – 0.85	12 – 17
Tanker	0.80 – 0.85	12 – 16
General cargo	0.55 – 0.75	13 – 22
Container ship	0.50 – 0.70	14 – 25
Ferry-boat	0.50 – 0.70	15 – 25

Table 3: Examples of block coefficients

Water plane area coefficient C_{WL}
The water plane area coefficient C_{WL}, expresses the ratio between the vessel'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. 3. i.e.:

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

Generally, the waterplane area coefficient is some 0.10 higher than the block coefficient, i.e. $C_{WL} \cong C_B + 0.10$.

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

Midship section coefficient C_M
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 D, See Fig. 3, i.e.

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

For bulkers and tankers, this coefficient is in the order of 0.98-0.99.

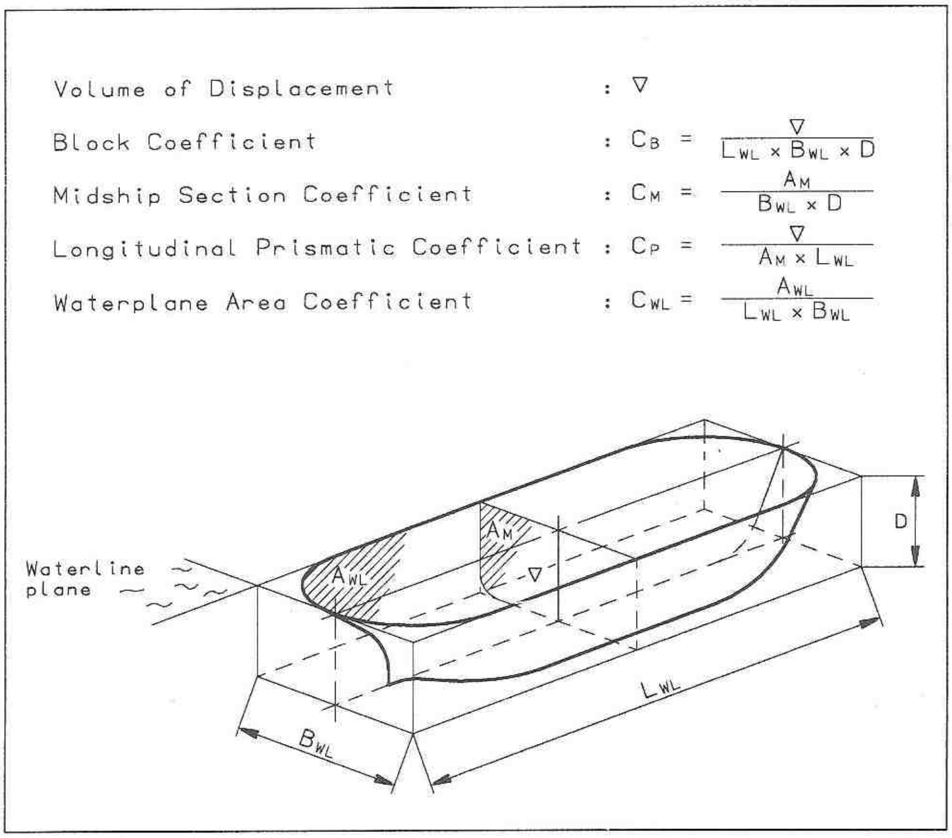

Fig. 3: Hull coefficients of a ship

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

For bulkers and tankers, this coefficient is in the order of 0.98-0.99.

Volume of Displacement : ∇

Block Coefficient : $C_B = \frac{\nabla}{L_{WL} \times B_{WL} \times D}$

Midship Section Coefficient : $C_M = \frac{A_M}{B_{WL} \times D}$

Longitudinal Prismatic Coefficient : $C_P = \frac{\nabla}{A_M \times L_{WL}}$

Waterplane Area Coefficient : $C_{WL} = \frac{A_{WL}}{L_{WL} \times B_{WL}}$

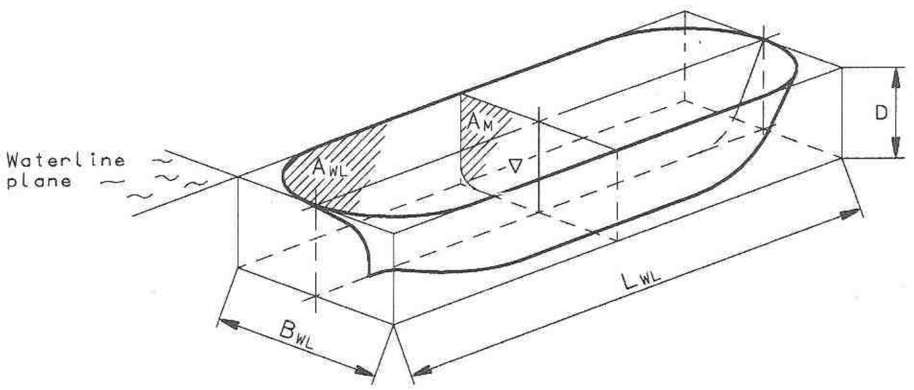


Fig. 3: Hull coefficients of a ship

Longitudinal prismatic coefficient C_P
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. 3, i.e.

$$C_P = \frac{\nabla}{A_M \times L_{WL}}$$
$$= \frac{\nabla}{C_M \times B_{WL} \times D \times L_{WL}} = \frac{C_B}{C_M}$$

As can be seen, C_P is not an independent form coefficient but entirely dependent on the block coefficient C_B and the midship section coefficient C_M.

Longitudinal Centre of Buoyancy LCB
The 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 mid-point between the ship's foremost and aftmost perpendiculars. The distance is normally stated as a percentage of the length between the perpendiculars, and is positive if the centre of buoyancy is located to the fore of the mid-point between the perpendiculars, and negative if located to the aft of the mid-point. 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%.

Fineness ratio C_{LD}
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, i.e.

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

Ship's resistance

To move a ship, it is first necessary to overcome resistance, i.e. the force working against its propulsion. The calculation of this resistance (R) plays a significant role in the selection of the correct propeller and in the subsequent choice of main engine.

General
A ship's 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
2) Residual resistance
3) Air resistance

The influence of frictional and residual resistances depends on how much of the hull is below the waterline, while the influence of air resistance depends on how much of the ship is above the waterline. In view of this, air resistance will have a certain effect on container ships which carry a large number of containers on the deck.

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

$$\tfrac{1}{2} \times \rho \times V^2 \quad \text{(Bernoulli's law).}$$

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 are related to the reference force K, defined as the force which 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's surface is also included in the wetted area. The general data for resistance calculations is thus:

Reference force: $K = \tfrac{1}{2} \times \rho \times V^2 \times A_S$
and source resistances: $R = C \times K$

On the basis of many experimental tank tests, and with the help of pertaining dimensionless hull parameters, some of which have already been discussed, methods have been established for calculating all the necessary resistance coefficients C and, thus, the pertaining source-resistances R. In practice, the calculation of a particular ship's resistance can be verified by testing a model of the relevant ship in a towing tank.

Frictional resistance R_F
The frictional resistance, R_F, of the hull depends on the size of the hull's wetted area A_S, and on the specific frictional resistance coefficient C_F. The friction increases with the growth of, i.a. algae, sea grass and goose barnacles.

When the ship is propelled through the water, the frictional resistance increases at a rate that is virtually equal to the square of the vessel's speed.

Frictional resistance represents a considerable part of the ship's resistance, often some 70-90% of the ship's total resistance for low-speed ships (bulk carriers and tankers), and sometimes less than 40% for high-speed ships (container and passenger ships) [4]. The frictional resistance is found as follows:

$$R_F = C_F \times K$$

Residual resistance R_R
Residual resistance, R_R, comprises wave resistance and eddy resistance. Wave resistance refers to the energy loss caused by waves created by the vessel during its propulsion through the water, while eddy resistance refers to the loss caused by flow separation which creates eddies, particularly at the aft end of the ship.

Wave resistance at low speeds is proportional to the square of the speed, but increases much faster at higher speeds. In principle, this means that a speed barrier is imposed, so that a further increase of the ship's propulsion power will not result in a higher speed as all the power will be converted into wave energy. The residual resistance normally represents 10-25% of the total resistance for low-speed ships, and up to 40-60% for high-speed ships [4].

Incidentally, shallow waters can also have great influence on the residual resistance, as the displaced water under the ship will have greater difficulty in moving aftwards.

The procedure for calculating the specific residual resistance coefficient C_R is described in specialised literature [2], and the residual resistance is found as follows:

$$R_R = C_R \times K$$

Air resistance R_A
In calm weather, air resistance is, in principle, proportional to the square of the ship's speed, and proportional to the cross-sectional area of the ship above the waterline. Air resistance normally represents about 2% of the total resistance.

For container ships in head wind, the air resistance can be as much as 10%. The air resistance can, similar to the foregoing resistances, be expressed as $R_A = C_A \times K$, but is sometimes based on 90% of the dynamic pressure of air with a speed of V, i.e.:

$$R_A = 0.90 \times \tfrac{1}{2} \times \rho_{air} \times V^2 \times A_{air}$$

where ρ_{air} is the density of the air, and A_{air} is the cross-sectional area of the vessel above the water [4].

Towing resistance R_T and effective (towing) power P_E
The ship's total towing resistance R_T is thus found as:

$$R_T = R_F + R_R + R_A$$

The corresponding effective (towing) power, P_E, necessary to move the ship through the water, i.e. to tow the ship at the speed V, is then:

$$P_E = V \times R_T$$

The power delivered to the propeller, P_D, in order to move the ship at speed V is, however, somewhat larger. This is due, in particular, to the flow conditions around the propeller and the propeller efficiency itself, the influences of which are discussed in the next chapter which deals with Propeller Propulsion.

Increase of ship resistance in service, [1], page 244. During the operation of the ship, the paint film on the hull will break down. Erosion will start, and marine plants and barnacles, etc. will grow on the surface of the hull. Bad weather, perhaps in connection with an inappropriate distribution of the cargo, can be a reason for buckled bottom plates. The ship hull has been fouled and will no longer have a "technically smooth" surface, which means that the frictional resistance will be greater. It must also be considered that the propeller surface can become rough and foulded. The total resistance caused by fouling may increase by 25-50% throughout the lifetime of a ship.

Resistance will also increase because of sea, wind and current. The resistance when navigating in head-on sea could perhaps increase by as much as 50-100% of the total ship resistance in calm weather.

The average increase of resistance for ships navigating the main routes is estimated to be as follows:

North Atlantic route navigation westward	25-35%
North Atlantic route navigation eastward	20-25%
Europe-Australia	20-25%
Europe-Eastern Asia	20-25%
The Pacific Routes	20-30%

On the North Atlantic routes the first percentage corresponds to summer navigation and the second percentage to winter navigation.

Unfortunately no data have been published on increased resistance as a function of the type and size of vessel. The larger the ship, the less the increase of resistance due to the sea. On the other hand, the frictional resistance of the large, full-bodied ships will very easily be changed in the course of time because of fouling.

The above information refers to [1], page 244.

In practice the increase of resistance caused by heavy weather depends on the current, the wind, as well as the wave size, where the latter factor may have a great influence. Thus, if the wave size is relatively high the ship speed will also be somewhat reduced even when sailing in fair seas.

In principle, the increase caused by heavy weather could be related to:

a) wind and current against, and to

b) heavy waves,

but in practice it will be difficult to distinguish these factors from each other.

Propeller Propulsion

The traditional agent employed to move a ship is a propeller, sometimes two and, in very rare cases, more than two. The necessary propeller thrust T required to move the ship at speed V is normally greater than the pertaining towing resistance R_T, and the flow related reasons are, amongst other reasons, explained in this chapter. See also Fig. 4, where all relevant velocity, force, power and efficiency parameters are shown.

Flow conditions around the propeller

Wake fraction coefficient w

When the ship is moving, the friction of the hull will create a so-called friction belt or boundary layer of water around the hull. In this friction belt the velocity of the water on the surface of the hull is equal to that of the ship, but is reduced with its distance from the surface of the hull. At a certain distance from the hull and, per definition, equal to the outer "surface" of the friction belt, the water velocity is equal to zero.

The thickness of the friction belt increases with its distance from the fore end of the hull. The friction belt is therefore thickest at the aft end of the hull and this thickness is nearly proportional to the length of the ship [3]. This means that there will be a certain wake velocity caused by the friction along the sides of the hull. Additionally, the ship's displacement of water will also cause wake waves both fore and aft. All this involves that the propeller behind the hull will be working in a wake field.

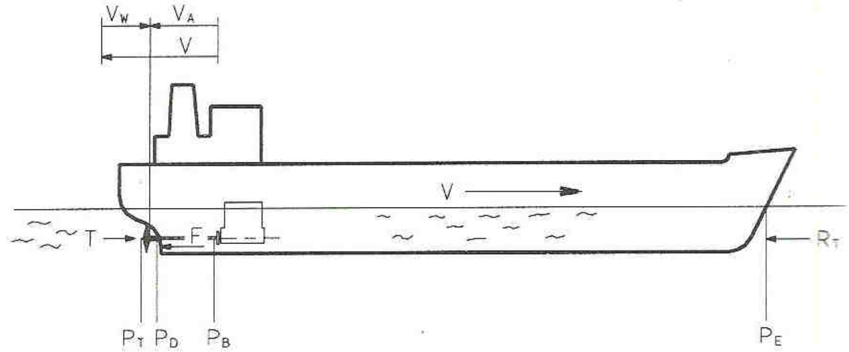
Therefore, and mainly originating from the friction wake, the water at the propeller will have an effective wake velocity V_w which has the same direction as the ship's speed V, see Fig. 4. This means that the velocity of arriving water V_A at the propeller, (equal to the speed of advance of the propeller) given as the average velocity over the propeller's disk area, is V_w lower than the ship's speed V.

Velocities

Ship's speed	V
Arriving water velocity to propeller (Speed of advance of propeller)	V_A
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	$\eta_H = \frac{1 - t}{1 - w}$
Relative rotative efficiency	η_R
Propeller efficiency - open water	η_O
Propeller efficiency - behind hull	$\eta_B = \eta_O \times \eta_R$
Propulsive efficiency	$\eta_D = \eta_H \times \eta_B$
Shaft efficiency	η_S
Total efficiency	η_T

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

Fig. 4: The propulsion of a ship – theory

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. The normally used wake fraction coefficient w given by Taylor is defined as

$$w = \frac{V_W}{V} = \frac{V - V_A}{V}$$

$$\text{(you get } \frac{V_A}{V} = 1 - w\text{)}$$

The value of the wake fraction coefficient depends largely on the shape of the hull, but also on the propeller's location and size, and has great influence on the propeller's efficiency.

The propeller diameter or, even better, the ratio between the propeller diameter d and the ship's length L_{WL} has some influence on the wake fraction coefficient, as d/L_{WL} gives a rough indication of the degree to which the propeller works in the hull's wake field. Thus, the larger the ratio d/L_{WL}, the lower w will be.

The wake fraction coefficient is increased when the hull is fouled. An attempt to avoid fouling is made by the use of modern hull paints (which are toxic), to prevent the hull from becoming "long-haired", i.e. these paints reduce the possibility of the hull becoming fouled by living organisms.

For ships with one propeller, the wake fraction coefficient w is normally in the region of 0.20 to 0.45, corresponding to a flow velocity to the propeller V_A of 0.55 to 0.80 of the ship's speed V, as ships with a large block coefficient have a large wake fraction coefficient. On ships with two propellers and a conventional aftbody form of the hull, the propellers will, normally, be positioned outside the friction belt, for which reason the wake fraction coefficient w will, in this case, be a great deal lower.

Incidentally, a large wake fraction coefficient increases the risk of propeller cavitation, as the distribution of the water velocity around the propeller is generally very inhomogeneous under such conditions.

A more homogeneous wake field for the propeller, also involving a higher speed of advance V_A of the propeller, may sometimes be needed and can be obtained in several ways, e.g. by having the propellers arranged in nozzles, below shields, etc. Obviously, the best method is to ensure, already at the design stage, that the aft end of the hull is shaped in such a way that the optimum wake field is obtained.

Thrust deduction coefficient t

The rotation of the propeller causes the water in front of it to be "sucked" back towards the propeller. This results in an 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. 4. This means that the thrust force T on the propeller has to overcome both the ship's resistance R_T and this "loss of thrust" F.

The thrust deduction fraction F may be expressed in dimensionless form by means of the thrust deduction coefficient t, which is defined as:

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

$$\text{(you get } \frac{R_T}{T} = 1 - t\text{)}$$

The thrust deduction coefficient t can be calculated by using calculation models set up on the basis of research carried out on different models.

In general, the size of the thrust deduction coefficient t increases when the wake fraction coefficient w increases. The shape of the hull may have a significant influence, e.g. a bulbous stem can, under certain circumstances (low ship speeds), reduce t.

The size of the thrust deduction coefficient t for a ship with one propeller is, normally, in the range of 0.12 to 0.30, as a ship with a large block coefficient has a large thrust deduction coefficient. For ships with two propellers, the thrust deduction coefficient t will be much less as the propellers' "sucking" occurs further away from the hull.

Efficiencies

Hull efficiency η_H

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 which the propeller delivers to the water $P_T = T \times V_A$, i.e.

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

For a ship with one propeller, the hull efficiency η_H is usually in the range of 1.1 to 1.4, with the high value for ships with high block coefficients. For ships with two propellers, the hull efficiency η_H is approx. 0.95 to 1.05, again with the high value for a high block coefficient.

Propeller efficiency η_O, working in open water

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, thrust force T, rate of revolution n, diameter d and, moreover, i.a. on the design of the propeller, i.e. the number of blades, disk area ratio, and pitch/diameter ratio – which will be discussed later on in this chapter. The propeller efficiency η_O can vary between approximately 0.35 and 0.75, with the high value being valid for propellers with a high speed of advance V_A [1].

Fig. 5 shows the obtainable propeller efficiency η_O shown as a function of the speed of advance V_A, which is given in dimensionless form as:

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

where J is the advance number of the propeller.

Relative rotative efficiency η_R

The actual velocity of the water flowing to the propeller behind the hull is neither constant nor at right angles to the

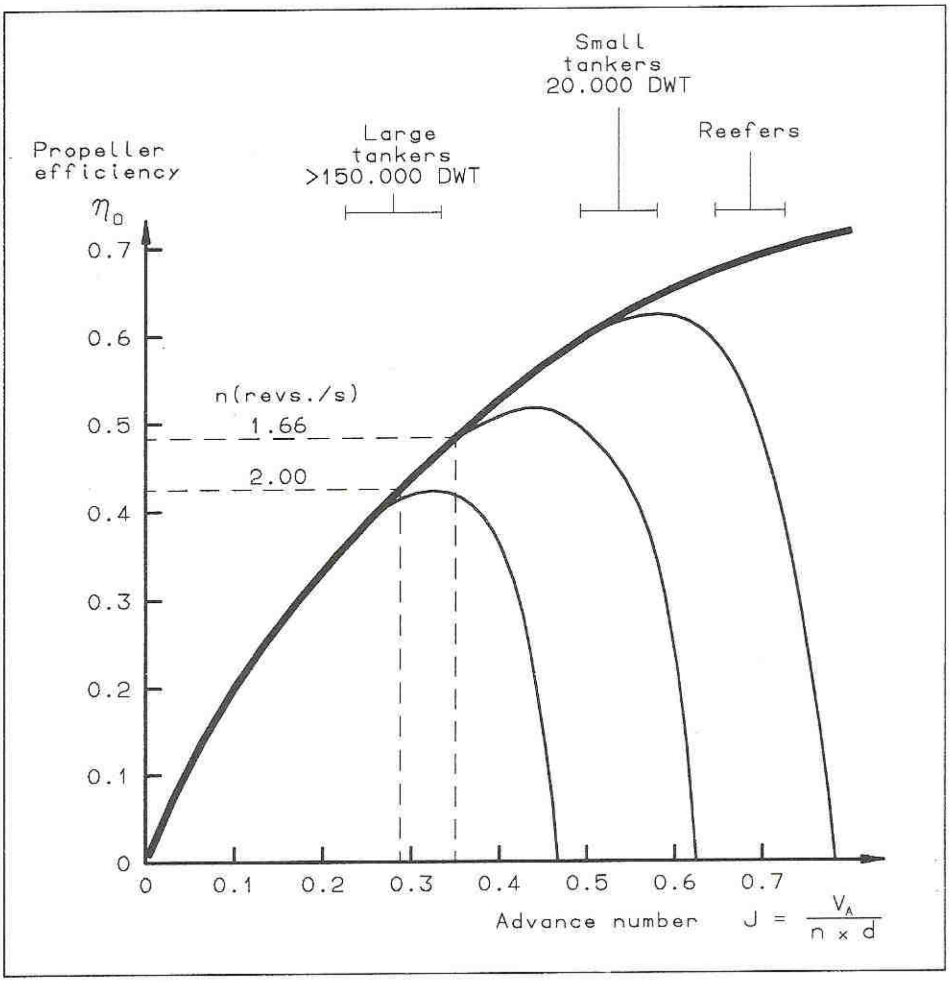


Fig. 5: Obtainable propeller efficiency – open water [1], page 213

propeller's disk area, but has a kind of rotational flow. Therefore, compared to when the propeller is working in open water, the propeller's efficiency is affected by the η_R factor – called the propeller's relative rotative efficiency.

On ships with a single propeller the rotative efficiency is, normally, around 1.0 to 1.07, in other words, the rotation of the water has a beneficial effect. The rotative efficiency on a ship with a conventional hull shape and with two propellers will, normally, be less, approximately 0.98.

In combination with w and t, η_R is probably often being used to adjust the results of model tank tests to the theory.

Propeller efficiency η_B working behind the ship

The ratio between the thrust power P_T, which the propeller delivers to the water, and the power P_D, which is delivered to the propeller, i.e. the propeller efficiency η_B for a propeller working behind the ship is defined as:

$$\eta_B = \frac{P_T}{P_D} = \eta_O \times \eta_R$$

Propulsive efficiency η_D

The propulsive efficiency η_D, which must not be confused with the open water propeller efficiency η_O, is equal to the ratio between the effective (towing) power P_E and the necessary power delivered to the propeller P_D, i.e.

$$\eta_D = \frac{P_E}{P_D} = \frac{P_E}{P_T} \times \frac{P_T}{P_D} =$$

$$\eta_H \times \eta_B = \eta_H \times \eta_O \times \eta_R$$

As can be seen, the propulsive efficiency η_D is equal to the product of the hull efficiency η_H, the open water propeller efficiency η_O, and the relative rotative efficiency η_R, although the latter has less significance.

In this connection, one can be led to believe that a hull form giving a high wake fraction coefficient w, and hence a high hull efficiency η_H, will also provide the best propulsive efficiency η_D.

However, as the propeller efficiency η_O is also greatly dependent on the speed of advance V_A, cf. Fig. 5, that is decreasing with increased w, the propulsive efficiency η_D will not, generally, improve with increasing w, quite often the opposite effect is obtained.

Generally, the best propulsive efficiency is achieved when the propeller works in a homogeneous wake field.

Shaft efficiency η_S

The shaft efficiency η_S depends, i.a. on the alignment and lubrication of the shaft bearings, and on the reduction gear, if installed.

Shaft efficiency is equal to the ratio between the power P_D delivered to the propeller and the brake power P_B delivered by the main engine, i.e.

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

The shaft efficiency is normally around 0.985, but can vary between 0.96 and 0.995.

Total efficiency η_T

The total efficiency η_T, which is equal to the ratio between the effective (towing) power P_E, and the necessary brake power P_B delivered by the main engine, can be expressed thus:

$$\eta_T = \frac{P_E}{P_B} = \frac{P_E}{P_D} \times \frac{P_D}{P_B} =$$

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

Propeller dimensions

Propeller diameter d

With a view to obtaining the highest possible propulsive efficiency η_D, the largest possible propeller diameter d will, normally, be preferred. There are, however, special conditions to be considered. For one thing, the aftbody form of the hull can vary greatly depending on type of ship and ship design, for another, the necessary clearance between the tip of the propeller and the hull will depend on the type of propeller.

For bulkers and tankers, which are often sailing in ballast condition, there are frequent demands that the propeller shall be fully immersed also in this condition giving some limitation to the propeller size. This propeller size limitation is not particularly valid for container ships, as they rarely sail in ballast condition. All the above factors mean that an exact propeller diameter/design draught ratio d/D cannot be given here but, as a rule-of-thumb, the below mentioned diameter/design draught approximations can be presented and a large diameter d will, normally, result in a low rate of revolution n.

Bulk carrier and tanker:
$d/D <$ approximately 0.65
Container ship:
$d/D <$ approximately 0.74

For production reasons, the propeller diameter will generally not exceed 8.5 metres, although the largest propeller manufactured has a diameter of some 12 metres.

Number of propeller blades

Propellers can be manufactured with 2, 3, 4, 5 or 6 blades. The fewer the number of blades, the higher the propeller efficiency will be. However, for reasons of strength, propellers which are to be subjected to heavy loads cannot be manufactured with only two or three blades.

Two-bladed propellers are used on small ships, and 4, 5 and 6-bladed propellers are used on large ships. Ships using the MAN B&W two-stroke engines are normally large-type vessels which use 4-bladed propellers. Ships with a relatively large power requirement and heavily loaded propellers, e.g. container ships, may need 5 or 6-bladed propellers. 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 [3].

Disk area coefficient

The disk area coefficient – referred to in older literature as expanded blade area ratio – defines the developed surface area of the propeller in relation to its disk area. A factor of 0.55 is considered as being good. The disk area coefficient of traditional 4-bladed propellers is of little significance, as a higher value will only lead to extra resistance on the propeller itself and, thus, have little effect on the final result.

For ships with particularly heavy-loaded propellers, often 5 and 6-bladed propellers, the coefficient may have a higher value. On warships it can be as high as 1.2.

Pitch diameter ratio p/d

The pitch diameter ratio p/d, expresses the ratio between the propeller's pitch p and its diameter d, see Fig. 6. The pitch, p, is the distance the propeller "screws" itself forward through the water per revolution, providing that there is no slip – see also the next section and Fig. 6. As the pitch can vary along the blade's radius, the ratio is normally related to the pitch at $0.7 \times r$, where $r = d/2$ is the propeller's radius.

To achieve the best propulsive efficiency for a given propeller diameter, an optimum pitch/diameter ratio is to be found, which again corresponds to a particular design rate of revolution. If, e.g. a lower design rate of revolution is desired, the pitch/diameter ratio has to be increased, and vice versa, at the cost of efficiency. On the other hand, if a lower design rate of revolution is desired, and the ship's draught permits, the choice of a larger propeller diameter may permit lower design rate of revolution and even, at the same time, increase the propulsive efficiency.

Propeller coefficients J, K_T and K_Q

Propeller theory is based on models but, to facilitate the general use of this theory, 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. The three most important of these coefficients are mentioned below.

The advance number of the propeller, J, is, as earlier mentioned, a dimensionless expression of the propeller's speed of advance V_A.

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

The thrust force T, is expressed dimensionless, with the help of the thrust coefficient K_T, as

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

and the propeller torque

$$Q = \frac{P_D}{2\pi \times n}$$

is expressed dimensionless with the help of the torque coefficient K_Q, as

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

The propeller efficiency η_O can be calculated with the help of the above-mentioned coefficients, because, as previously mentioned, the propeller efficiency η_O is defined as:

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

With the help of special and very complicated propeller diagrams, which contain, i.a. J, K_T and K_Q curves, it is possible to find/calculate the propeller's dimensions, efficiency, thrust, power, etc.

Operating conditions of a propeller

Slip ratio S

If the propeller had no slip, i.e. if the water which the propeller "screws" itself through did not yield (i.e. if the water did not accelerate aft), the propeller would move forward at a speed of p × n, where n is the propeller's rate of revolution, see Fig. 6.

However, as the water ***does*** yield (i.e. accelerate aft), the propeller's real speed decreases and becomes equal to the ship's speed V, and its apparent slip can thus be expressed as p × n – V.

The apparent slip ratio S_A, which is dimensionless, is defined as:

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

The apparent slip ratio S_A, which is calculated by the crew, provides useful knowledge as it gives an impression of the loads applied to the propeller under different operating conditions. The apparent slip ratio increases, i.a. when the vessel sails against the wind or waves, in shallow waters, when the hull is fouled, and when the ship accelerates.

The real slip ratio will be greater than the apparent slip ratio because the real speed of advance V_A of the propeller is, as previously mentioned, less than the ship's speed V.

The real slip ratio S_R, which gives a truer picture of the propeller's function, is:

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

At quay trials where the ship's speed is V = 0, both slip ratios are 1.0. Incidentally, slip ratios are often given in percentages.

Propeller law

As discussed in Chapter I, the resistance R for lower ship speeds is proportional to the square of the ship's speed V, i.e. $R = c \times V^2$ where c is a constant. The necessary power requirement P is thus proportional to the speed V to the power of three, thus:

$$P = R \times V = c \times V^3$$

For a ship equipped with a fixed pitch propeller, i.e. a propeller with unchangeable pitch, the speed V will be proportional to the rate of revolution n, thus:

$$P = c \times n^3$$

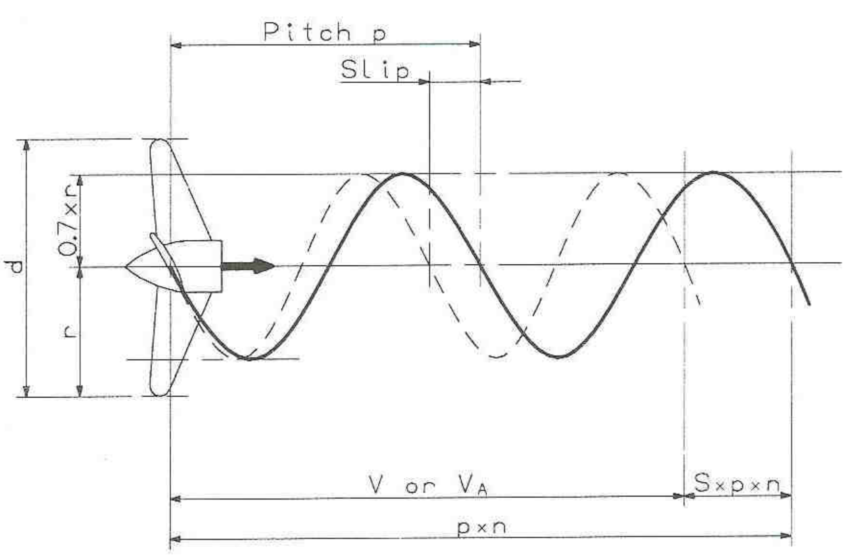
which precisely expresses the propeller law which states that "the necessary power delivered to the propeller is proportional to the rate of revolution to the power of three".

Propeller law, fouled hulls, etc.

The propeller law, of course, can only be applied to identical ship running conditions. When, for example, the ship's hull after some time in service has been fouled and thus become more rough, the wake field will be different from that of the smooth ship (clean hull) valid at trial trip conditions.

A ship with a fouled hull will, consequently, be subject to extra resistance which will give rise to a "heavy propeller condition", i.e. at the same propeller power, the rate of revo-lution will be lower.

The propeller law now applies to another and "heavier" propeller curve than that applying to the clean hull, propeller curve [1], page 243.



The apparent slip ratio : $S_A = \frac{p \times n - V}{p \times n} = 1 - \frac{V}{p \times n}$

The real slip ratio : $S_R = \frac{p \times n - V_A}{p \times n} = 1 - \frac{V_A}{p \times n}$

Fig. 6: The slip ratio S of a propeller and the pitch/diameter ratio p/d

The same relative considerations apply when the ship is sailing in a heavy sea against the current, a strong wind, and heavy waves, where especially the *wave resistance* may give rise to a heavier propeller running than when running in calm weather. On the other hand, if the ship is sailing in ballast condition, i.e. with a lower displacement, the propeller law now applies to a "lighter" propeller curve, i.e. at the same propeller power, the propeller rate of revolution will be higher.

As mentioned previously, the propeller law applying to ships with a fixed pitch propeller is extensively used at part load running. Thus, it is also used in MAN B&W Diesel's engine layout and load diagrams to specify the engine's operational curves for clean and fouled hull, etc. These diagrams are described in detail in the next chapter.

Heavy waves
When sailing in heavy sea against, with heavy wave resistance, the propeller can be 3-4% heavier running than in calm weather, i.e. at the same propeller power, the rate of revolution may be 3-4% lower. On the other hand, in some cases in practice with heavy wind against, heavy running has proved to be even greater.

In order to avoid slamming of the ship, and thereby damage to the stem and racing of the propeller, the ship speed will normally be reduced by the navigating officer on watch.

Ship acceleration
When the ship accelerates, the propeller will be subjected to an even larger load than during free sailing. The power required for the propeller, therefore, will be relatively higher than for free sailing and the engine's operating point will be heavy running.

Shallow waters
When sailing in shallow waters the residual resistance of the ship may be increased and, in the same way as when the ship accelerates, the propeller will be subjected to a larger load than during free sailing, and the propeller will be heavy running.

Influence of the displacement
When the ship is sailing in the loaded condition, the ship's displacement volume may, for example, be 10% higher or lower than for the displacement valid for the average loaded condition. This, of course, has an influence on the ship's resistance, and the needed propeller power, but only a minor influence on the propeller curve.

On the other hand, when the ship is sailing in the ballast condition, the displacement volume compared to the loaded condition can be much lower, and the corresponding propeller curve may apply to, for example, a 3% "lighter" propeller curve, i.e. for the same power to the propeller, the rate of revolution will be 3% higher.

Manoeuvring speed
Below a certain ship speed, called the manoeuvring speed, the manoeuvrability of the rudder is insufficient because of a too low velocity of the arriving water at the rudder. It is rather difficult to give an exact figure for an adequate manoeuvring speed of the ship as the velocity of the water arriving at the rudder depends on the propeller's slip stream.

Often a manoeuvring speed of the magnitude of 3.5-4.5 knots is mentioned. According to the propeller law, a correspondingly low propulsion power will be needed but, of course, this will be higher for running in heavy weather with increased resistance on the ship.

Direction of propeller rotation (Side thrust)
When a ship is sailing, 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 ship's speed decreases. Awareness of this behaviour is very important in critical situations and during harbour manoeuvres.

According to [3], page 15-3, the real reason for the appearance of the side thrust during reversing of the propeller is that the upper part of the propeller's slip stream, which is rotative, strikes the aftbody of the ship.

Thus, also the pilot has to know precisely how the ship reacts in a given situation. It is, therefore, an unwritten law that on a ship fitted with a *fixed pitch propeller* the propeller is always designed for *clockwise* rotation when sailing ahead. A direct coupled main engine, of course, will have the same rotation.

In order to obtain the same side-thrust effect, when reversing to astern, on ships fitted with a *controllable pitch propeller*, CP-propellers are designed for *anticlockwise* rotation when sailing ahead.

Chapter III

Engine Layout and Load Diagrams

Introduction

As is well-known, the effective brake power P_B of a diesel engine is proportional to the mean effective pressure (mep) p_e and engine speed (rate of revolution) n. When using c as a constant, P_B may then be expressed as follows:

$$P_B = c \times p_e \times n$$

or, in other words, for constant mep the power is proportional to the speed:

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

As already mentioned – when running with a fixed pitch propeller – the power may, according to the propeller law, be expressed as:

$$P_B = c \times n^3 \qquad \text{(propeller law)}$$

Thus, for the above examples, the brake power P_B may be expressed as an exponential function of the speed n to the power of i, i.e.:

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

Fig. 7 shows the relationship between the linear functions, y = ax + b, see (A), using linear scales and the exponential functions $P_B = c \times n^i$, see (B), using logarithmic scales.

The exponential functions will be linear when using logarithmic scales, as:

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

which is equivalent to:

$$y = ax + b$$

Thus, propeller curves will be parallel to lines having the inclination i = 3, and lines with constant mep will be parallel to lines with the inclination i = 1.

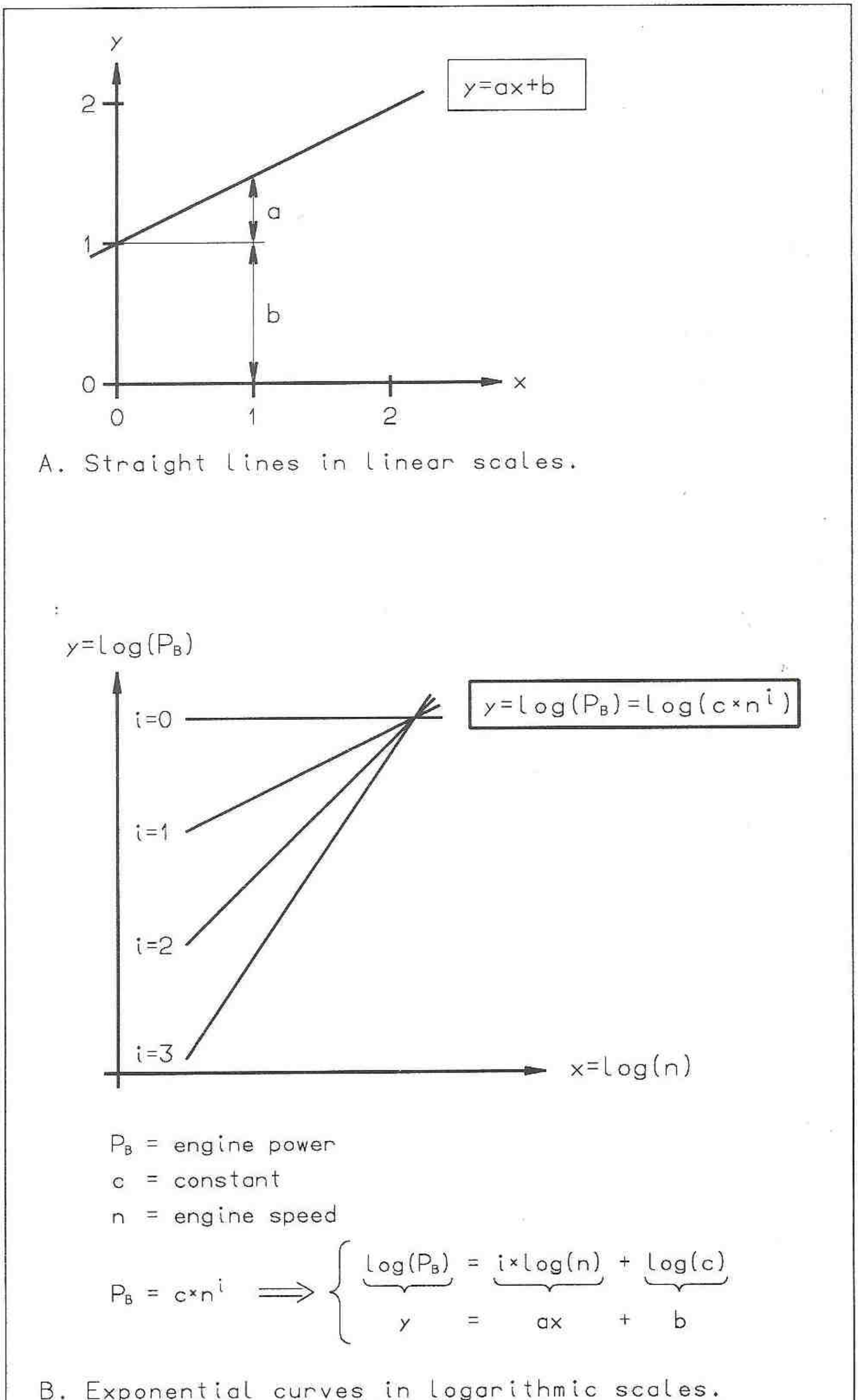


Fig. 7: Relationship between linear functions using linear scales and exponential functions using logarithmic scales

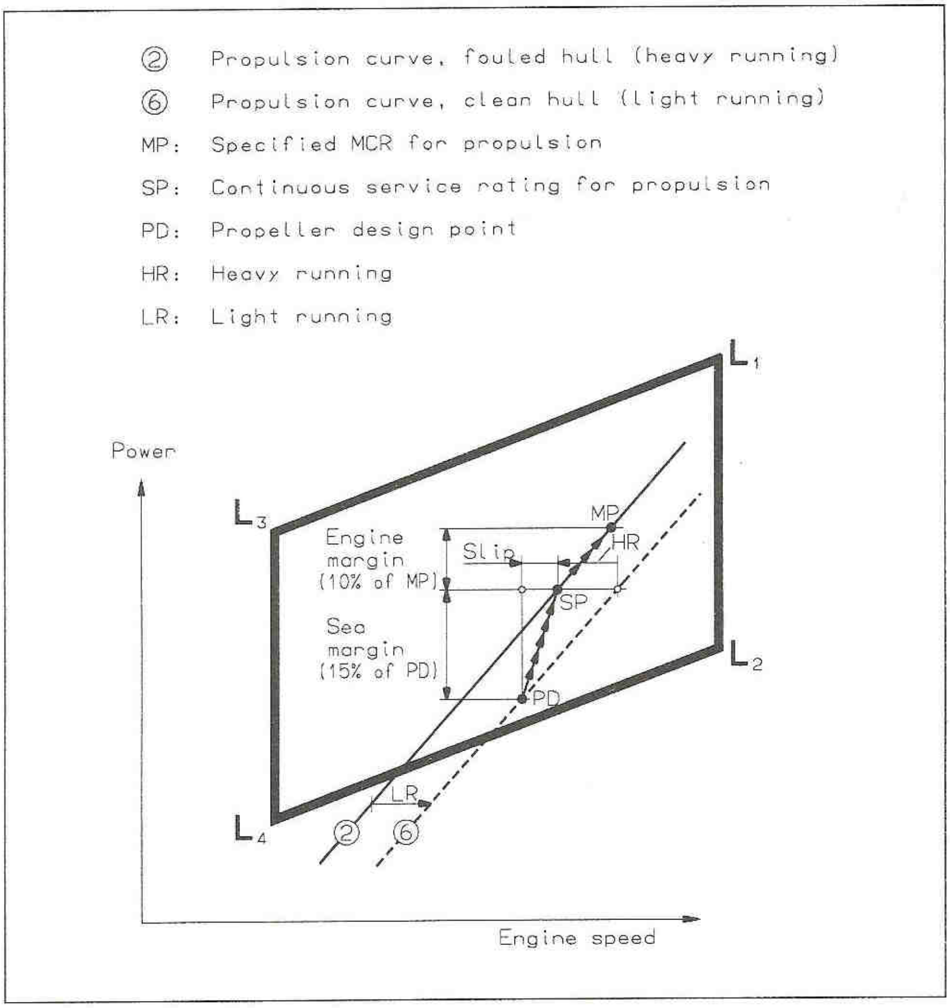


Fig. 8: Ship propulsion running points and engine layout

Therefore, in the layout and load diagrams for diesel engines, as described in the following, logarithmic scales are used, making simple diagrams with straight lines.

Propulsion and engine running points

Propeller design point

Normally, estimations of the necessary propeller power and speed are based on theoretical calculations, and often experimental tank tests, both assuming optimum operating conditions, i.e. a clean hull and good weather. The combination of speed and power obtained may be called the ship's propeller design point (PD), see Fig. 8. On the other hand, some shipyards and/or propeller manufacturers sometimes use a propeller design point that incorporates all or part of the so-called sea margin described below.

Fouled hull and sea margin

When the ship has sailed for some time, the hull and propeller become fouled, causing a change in the propeller's wake field and an increased hull resistance. If, at the same time, the weather is bad, with head winds, the ship's resistance will increase further. Consequently, the ship speed will be reduced unless the engine supplies more power, i.e. the propeller will become further loaded and will be heavy running (HR). In this connection and with only little wave resistance, it is especially the fouling, and thereby the change of the propeller's wake field, that causes the propeller curve to move to the left, see Fig. 8, where propeller curve 6 is valid for loaded ship with clean hull and propeller curve 2, to the left of curve 6, is valid for loaded ship with fouled hull. On the other hand, when sailing in bad weather with *heavy wave resistance* the effect may also be a heavier propeller running and, therefore, the heavy running propeller curve 2 to be anticipated and denoted as "fouled hull" may in principle consider the influence of both fouling and wave resistance.

When the necessary engine power and speed is to be determined, therefore, it is necessary to use a heavy running factor (2.5 to 5%) and to add an extra power margin, the so-called "sea-margin", which traditionally is about 15%. The corresponding speed and power combination, called the "continuous service rating for propulsion" (SP) for fouled hull, see Fig. 8, incorporates a certain "slip" and "heavy running" compared to the clean hull propeller curve 6 through the propeller design point PD.

Light running factor f_{LR}

The propeller curve for a fouled hull (and heavy waves) may be used as the basis for the engine operating curve in service, curve 2, whereas the propeller curve for clean hull (and calm weather) , curve 6, may be valid for running conditions with new ships. Therefore, the propeller curve for clean hull is said to represent a "light running" (LR) propeller and will be related to the fouled hull condition by means of a light running factor f_{LR}, which, for the same power to the propeller, is defined as the percentage increase of the rate of revolution n, compared to the rate of revolution with a fouled hull, i.e.

$$f_{LR} = \frac{n_{clean} - n_{fouled}}{n_{fouled}} \times 100\%$$

Engine margin

Besides the sea margin, a so-called "engine margin" of some 10% is frequently added. The corresponding point is called the "specified MCR for propulsion" (MP), see Fig. 8, and refers to the fact that the power for point SP is 10% lower than for point MP. Point MP is identical to the engine's specified MCR point (M) unless a main engine driven shaft generator is installed. In such a case, the extra power demand of the shaft generator must also be considered.

Note:
Light/heavy running, fouling and sea margin are overlapping terms. Light/heavy running of the propeller refers to hull and propeller deterioration, and bad weather, and sea margin, i.e. extra power to the propeller, refers to the influence of the wind and the sea.

Based on feedback from service, it seems reasonable to design the propeller for 2.5-5% light running. However, the degree of light running must be decided upon, based on experience from the actual trade and hull design.

Engine layout diagram

An engine's layout diagram is limited by two constant mean effective pressure (mep) lines L_1-L_3 and L_2-L_4, and by two constant engine speed lines L_1-L_2 and L_3-L_4, see Fig. 8. The L_1 point refers to the engine's nominal maximum continuous rating. Within the layout area there is full freeedom to select the engines specified MCR point M and relevant optimising point O, see below, which is optimum for the ship and the operating profile.

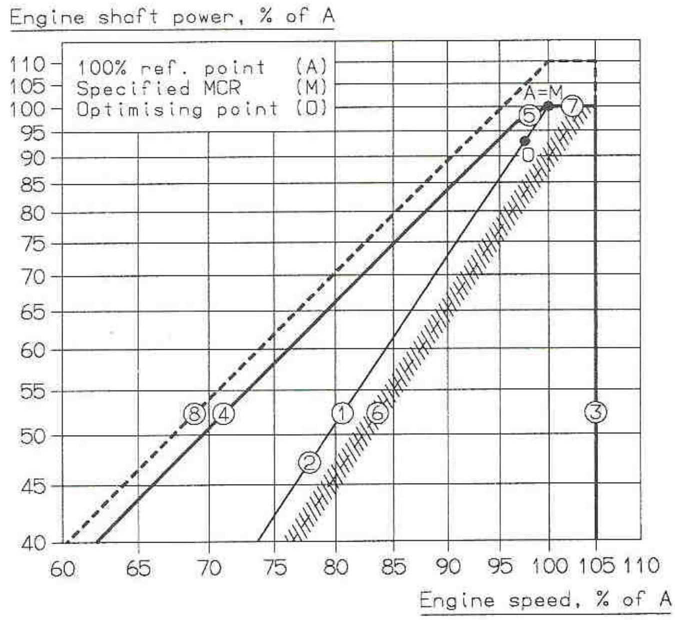
Specified MCR (M)
Based on the propulsion and engine running points, as previously found, the layout diagram of a relevant main engine may be drawn-in. The specified MCR point (M) must be inside the limitation lines of the layout diagram; if it is not, the propeller speed will have to be changed or another main engine type must be chosen. Yet, in special cases, point M may be located to the right of line L_1-L_2, see "Optimising Point" below.

Optimising point (O)
The optimising point O is the rating at which the turbocharger is matched, and at which the engine timing and compression ratio are adjusted.

As is mentioned overleaf under "Load Diagram", the optimising point O is placed on line 1 of the load diagram, and the optimised power can be from 85 to 100% of point M's power, when turbocharger(s) and engine timing are taken into consideration.

The optimising point O is to be placed inside the layout diagram. In fact, the specified MCR point M can be placed outside the layout diagram, but only by exceeding line L_1-L_2, and, of course, only provided that the optimising point O is located inside the layout diagram.

Please note that the S26MC and L35MC type engines cannot be optimised at part-load. Therefore, these two engine types are always optimised in point A, i.e. having point M's power.



Line 1: Propeller curve through optimising point.
Line 2: Propeller curve, fouled hull - heavy running.
Line 3: Speed limit.
Line 4: Torque/speed limit.
Line 5: Mean effective pressure limit.
Line 6: Propeller curve, clean hull - light running.
Line 7: Power limit for continous running.
Line 8: Overload limit.

Point M to be located on line 7 (normally in point A).

Fig. 9: Engine load diagram

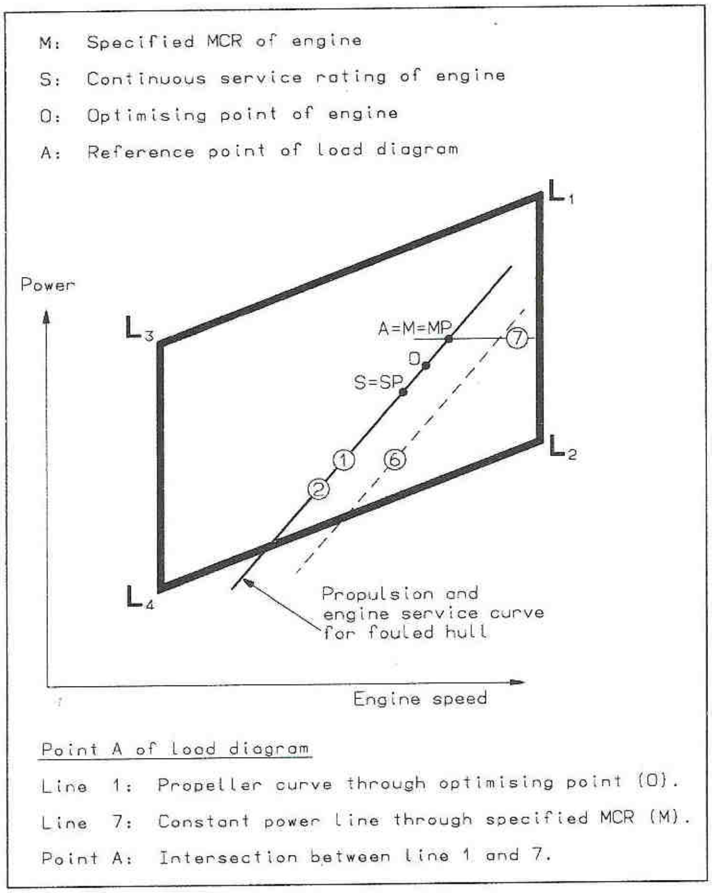


Fig. 10a: Example 1 with FPP – engine layout without SG (normal case)

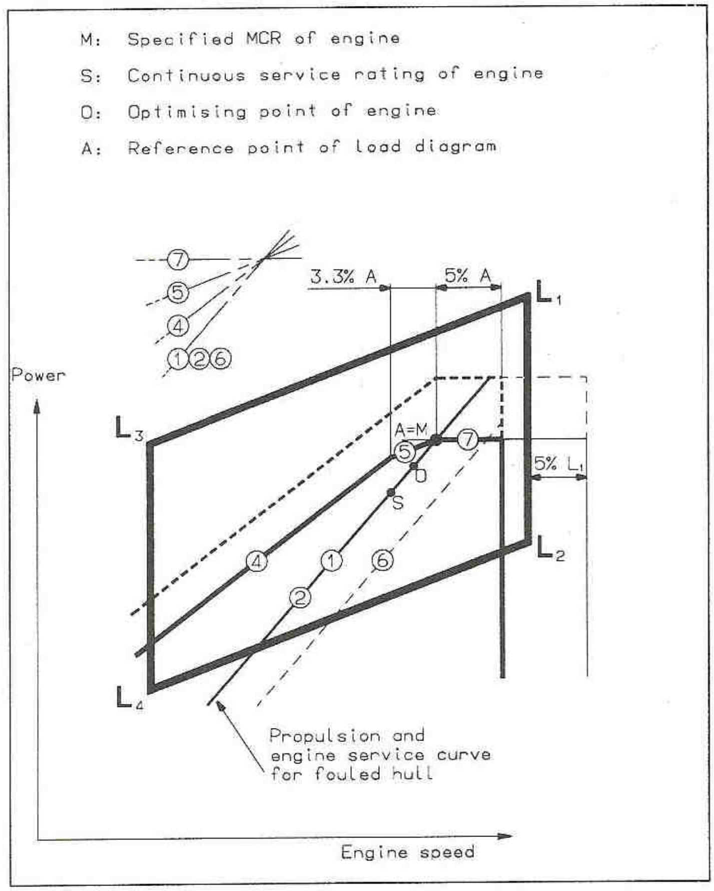


Fig. 10b: Example 1 with FPP – load diagram without SG (normal case)

Load diagram

Definitions

The load diagram (Fig. 9) defines the power and speed limits for continuous as well as overload operation of an installed engine which has an optimising point O and a specified MCR point M that conforms to the ship's specification.

Point A is a 100% speed and power reference point of the load diagram, and is defined as the point on the propeller curve (line 1) through the optimising point O, having the specified MCR power. Normally, point M is equal to point A, but in special cases, for example if a shaft generator is installed, point M may be placed to the right of point A on line 7. The service points of the installed engine incorporate the engine power required for ship propulsion and for the shaft generator, if installed.

Limits for continuous operation

The continuous service range is limited by four lines, see Fig. 9:

Line 3:
Represents the maximum speed which can be accepted for continuous operation, i.e. 105% of A, however maximum 105% of L_1.

The limit may be extended to 105%, and during trial conditions to 107%, of the nominal L_1 speed of the engine, provided the torsional vibration conditions permit.

Running at low load above 100% of the nominal L_1 speed of the engine is, however, to be avoided for extended periods.

Line 4:
Represents the limit at which an ample air supply is available for combustion and imposes a limitation on the maximum combination of torque and speed.

Line 5:
Represents the maximum mean effective pressure level (mep) which can be accepted for continuous operation.

Line 7:
Represents the maximum power for continuous operation.

Limits for overload operation

The overload service range is limited as follows, see Fig. 9:

Line 8:
Represents the overload operation limitations.

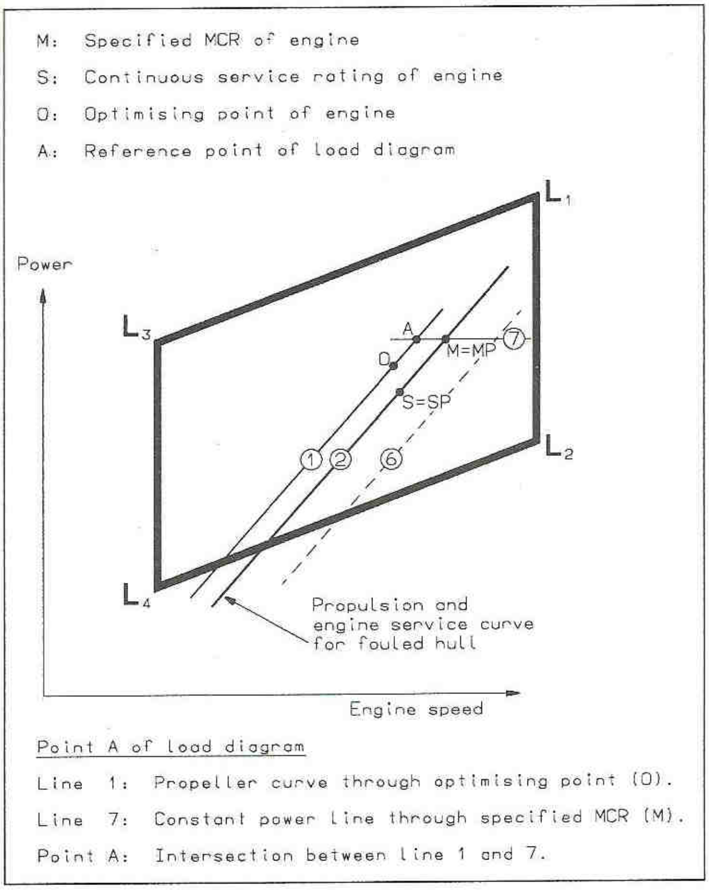


Fig. 11a: Example 2 with FPP – engine layout (special case)

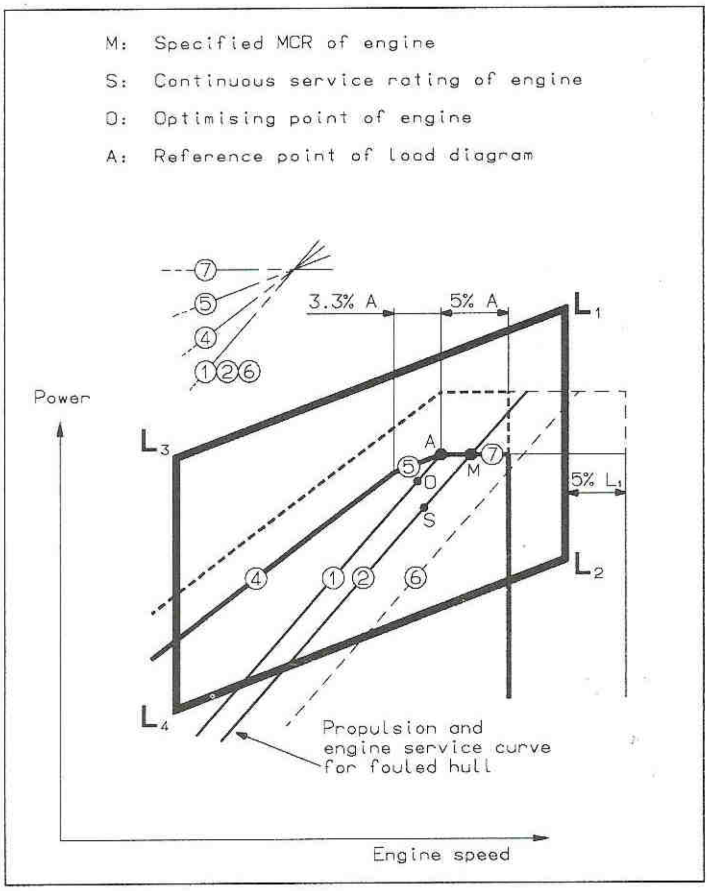


Fig. 11b: Example 2 with FPP – load diagram (special case)

The area between lines 4, 5, 7 and the dashed line 8 in Fig. 9 is available for overload running for limited periods only (1 hour per 12 hours).

Recommendation

Continuous operation without a time limitation is allowed only within the area limited by lines 4, 5, 7 and 3 of the load diagram.

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

After some time in operation, the ship's hull and propeller will become fouled, resulting in heavier running of the propeller, i.e. the propeller curve will move to the left from line 6 towards line 2, and extra power will be required for propulsion in order to maintain the ship speed.

At calm weather conditions, the extent of heavy running of the propeller will indicate the need for cleaning the hull and, possibly, polishing the propeller.

Use of the load diagram – examples

In the following, four different examples based on fixed pitch propeller (FPP) and one example based on controllable pitch propeller (CPP) are given in order to illustrate the flexibility of the layout and load diagrams.

In this respect the choice of the optimising point O has a significant influence.

Examples with fixed pitch propeller

Example 1:
Normal running conditions, without shaft generator

Normally, the optimising point O will be selected on the engine service curve 2 (for fouled hull), as shown in Fig. 10a.

Point A is then found at the intersection between propeller curve 1 (2) and the constant power curve through M, line 7. In this case point A will be equal to point M.

Once point A has been found in the layout diagram, the load diagram can be drawn, as shown in Fig. 10b, and hence the actual load limitation lines of the diesel engine may be found.

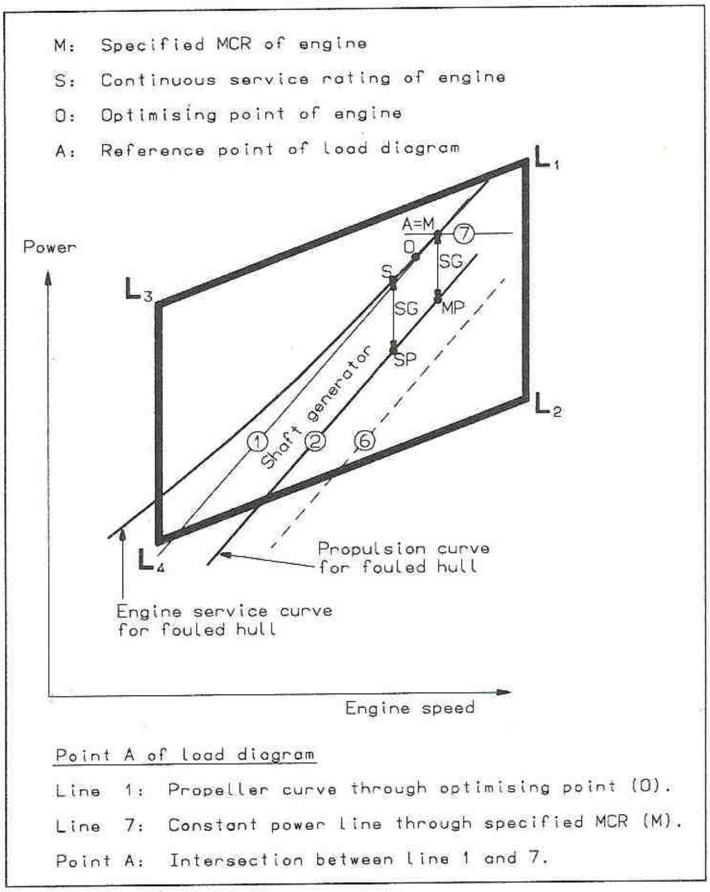


Fig. 12a: Example 3 with FPP – engine layout with SG (normal case)

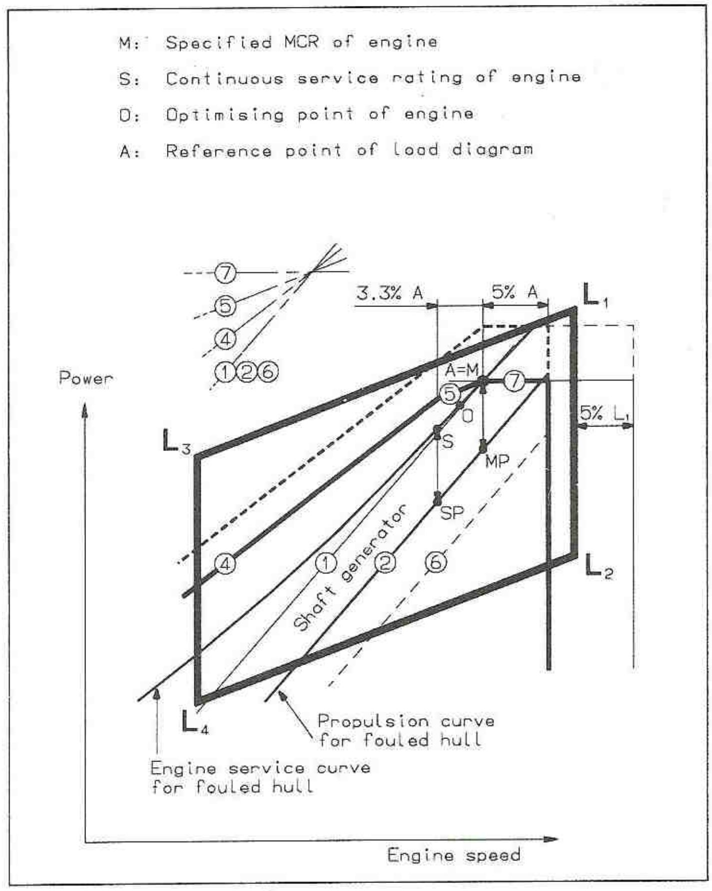


Fig. 12b: Example 3 with FPP – load diagram with SG (normal case)

Example 2:
Special running conditions, without shaft generator

When the ship accelerates, the propeller will be subjected to an even larger load than during free sailing. The same applies when the ship is subjected to an extra resistance as, for example, when sailing against heavy wind and sea with *large wave resistance*.

In both cases, the engine's operating point will be to the left of the normal operating curve, as the propeller will run heavily.

In order to avoid exceeding the left-hand limitation line 4 of the load diagram, it may, in certain cases, be necessary to limit the acceleration and/or the propulsion power.

If the expected trade pattern of the ship is to be in an area with frequently appearing heavy wind and sea and large wave resistance, it can, therefore, be an advantage to design/move the load diagram towards the left. The latter can be done by moving the engine's optimising point O – and thus the propeller curve 1 through the optimising point – towards the left. However, this will be at the expense of a slightly increased specific fuel oil consumption.

An example is shown in Figs. 11a and 11b. As will be seen in Fig. 11b, and compared to the normal case shown in Example 1 (Fig. 10b) the left-hand limitation line 4 is moved to the left, giving a wider margin between lines 2 and 4.

Example 3:
Normal case, with shaft generator

In this example a shaft generator (SG) is installed, and therefore the service power of the engine also has to incorporate the extra shaft power required for the shaft generator's electrical power production.

In Fig. 12a, the engine service curve shown for fouled hull incorporates this extra power.

The optimising point O will normally be chosen on this curve as shown, but can, by an approximation, be located on curve 1 through point M.

Point A is then found in the same way as in example 1, and the load diagram can be drawn as shown in Fig. 12b.

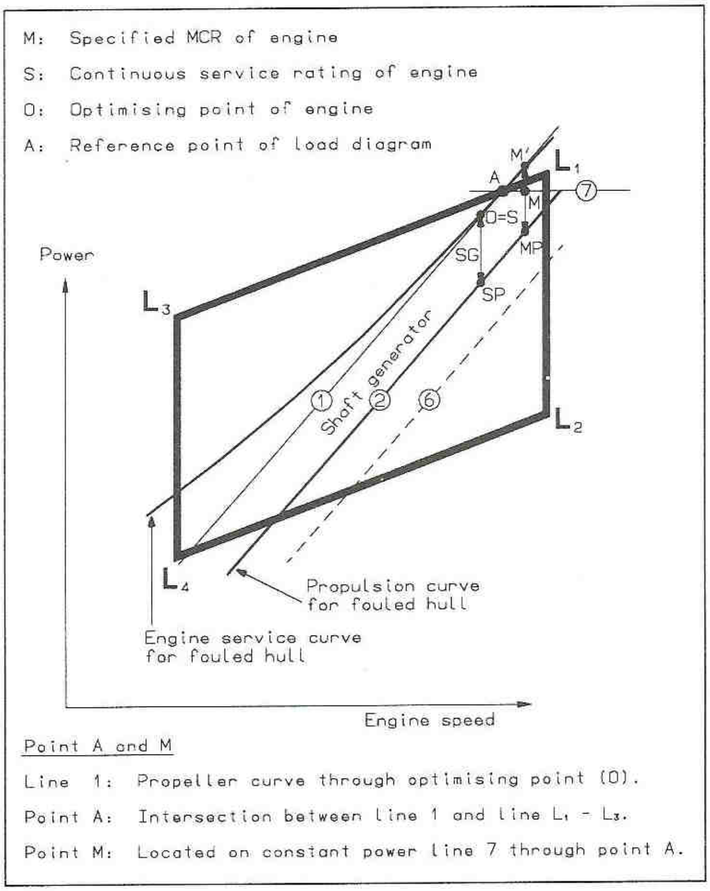


Fig. 13a: Example 4 with FPP – engine layout with SG (special case)

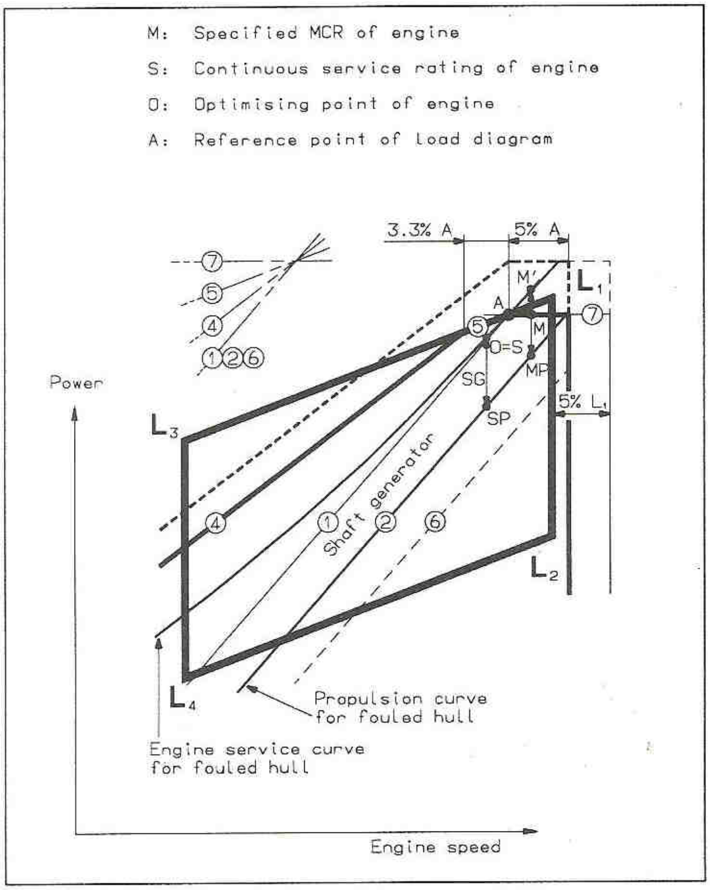


Fig. 13b: Example 4 with FPP – load diagram with SG (special case)

Example 4:
Special case, with shaft generator

Also in this special case, a shaft generator is installed but, unlike in Example 3, now the specified MCR for propulsion MP is placed at the top of the layout diagram, see Fig. 13a. This involves that the intended specified MCR of the engine (point M') will be placed outside the top of the layout diagram.

One solution could be to choose a diesel engine with an extra cylinder, but another and cheaper solution is to reduce the electrical power production of the shaft generator when running in the upper propulsion power range.

If choosing the latter solution the required specified MCR power of the engine can be reduced from point M' to point M as shown in Fig. 13a. Therefore, when running in the upper propulsion power range, a diesel generator has to take over all or part of the electrical power production.

However, such a situation will seldom occur, as ships rather infrequently operate in the upper propulsion power range. In the example, the optimising point O has been chosen equal to point S, and line 1 may be found.

Point A, having the highest possible power, is then found at the intersection of line L_1-L_3 with line 1, see Fig. 13a, and the corresponding load diagram is drawn in Fig. 13b. Point M is found on line 7 at MP's speed.

Example with controllable pitch propeller

Example 5:
With or without shaft generator

When a CP-propeller is installed, the relevant combinator curves of the propeller may be a combination of constant engine speeds and/or propeller curves, and it is not possible to distinguish between running points for clean and fouled hull.

Therefore, when the engine's specified MCR point (M) has been chosen, including the power required for a shaft generator, if installed, *point M may be used as point A* of the load diagram, which may then be drawn.

The optimising point O may be chosen on the propeller curve through point A = M having an optimised power from 85 to 100% of the specified MCR power as before, in the section dealing with Optimising Point O.

An example is given in Fig. 14, which shows two examples of combinator curves that are both contained within the same load diagram.

For specific cases with a shaft generator and where the propeller's combinator curve in the high power range is a propeller curve (similar to the FPP propulsion curve 2 for fouled hull), please also see the fixed pitch propeller examples 3 and 4.

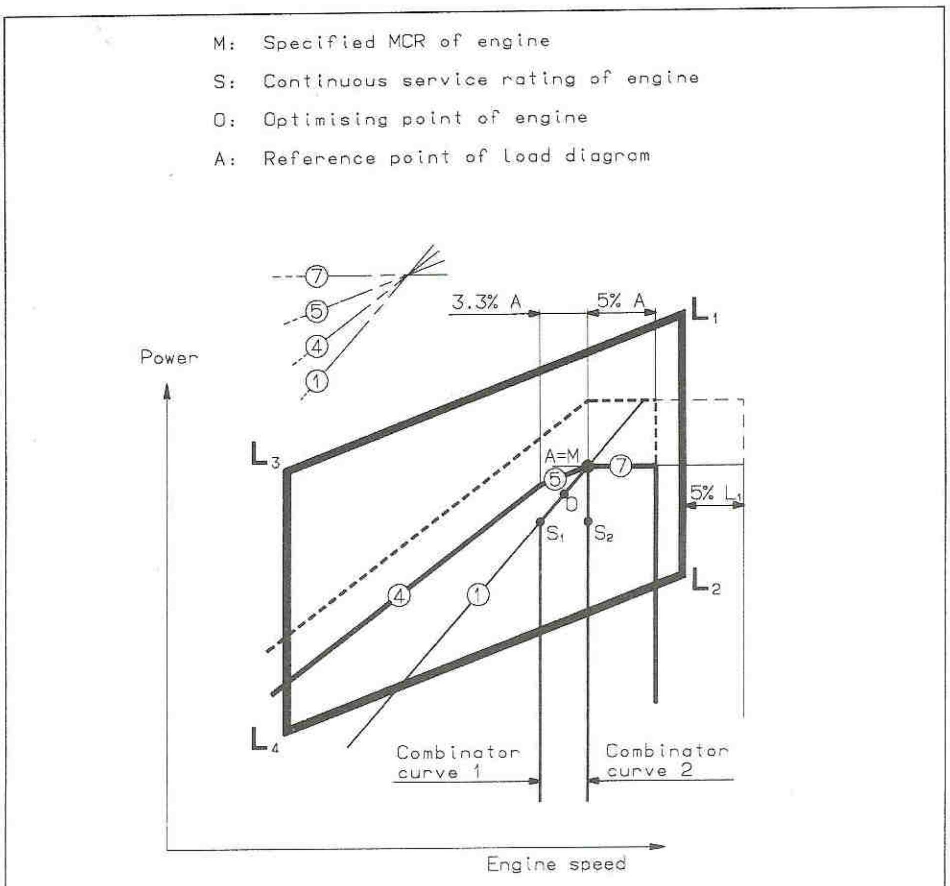


Fig. 14: Example 5 with CPP – load diagram

Influence on engine running of different types of ship resistance

In order to give a brief summary regarding the influence on the propeller and main engine operation of different types of ship resistance, an arbitrary example has been chosen, see the load diagram in Fig. 15.

The influence of the different types of resistance is illustrated by means of corresponding service points for propulsion having the same propulsion power, using as basis the propeller design point PD, plus 15% extra power.

Propeller Design Point PD
The propeller will, as previously described, normally be designed according to a specified ship speed V valid for loaded ship with clean hull and calm weather conditions. The corresponding engine speed and power combination is shown as point PD on propeller curve 6 in the load diagram, Fig. 15.

Increased ships speed, point S0
If the engine power is increased by, for example, 15%, and the loaded ship is still operating with a clean hull and in calm weather (point S0), the ship speed V and engine speed n will increase in accordance with the propeller law (more or less valid for the normal speed range):

$$V_{S0} = V \times \sqrt[3]{1.15} = 1.048 \times V$$

$$n_{S0} = n \times \sqrt[3]{1.15} = 1.048 \times n$$

Point S0 will be placed on the same propeller curve as point PD.

Sea running with clean hull and current against, 15% sea margin, point S2
Conversely, if still operating with loaded ship and clean hull, but now with extra resistance from heavy seas (with relatively small waves) and wind and running against the current, an extra power of, for example, 15% is needed in order to maintain the ship speed V (15% sea margin).

As the ship speed V_{S2} = V, and if the propeller had no slip, it would be expected that the engine (propeller) speed would also be constant. However, as the water does yield, i.e. the propeller has a slip, the engine speed will increase and the running point S2 will be placed on a propeller curve 6.2 very close to S0, on propeller curve 6. Propeller curve 6.2 will possibly represent an approximate 0.5% heavier running propeller than curve 6, [1], page 242.

Depending on the block coefficient of the ship, the heavy running factor of 0.5% may be somewhat higher (high block coefficient) or lower (low block coefficient).

For a resistance corresponding to about 30% extra power (30% sea margin), the corresponding relative heavy running factor will be about 1%, [1], page 242.

If a ship with a fouled hull is used as a basis instead of a clean hull, the same relative heavy running considerations will apply regarding the influence of extra resistance caused by the same weather conditions.

In general, therefore, it must be noted that even a relatively large change in resistance due to weather conditions only, does not alter the ship's propeller curve very much.

Accordingly, the change of ship speed because of a change in resistance can be quite large but has no significant influence on the interaction between the ship, the main engine and the propeller [1], page 242. This applies when the waves are relatively small.

Sea running with fouled hull, and 15% sea margin, point SP

When, after some time in service, the ship's hull has been fouled, and thus become more rough, the wake field will be different from that of a smooth ship (clean hull).

A ship with a fouled hull will, consequently, be subject to an extra resistance which, due to the changed wake field, will give rise to a heavier running propeller than experienced during bad weather conditions alone. When also incorporating some average inlfluence of heavy seas and waves, the propeller curve for loaded ship will move to the left, see propeller curve 2 in the load diagram in Fig. 15. This propeller curve, denoted as fouled hull and loaded ship, is about 4% heavy runnning compared to the clean hull and calm weather propeller curve 6.

In order to maintain an ample air supply for the diesel engine's combustion, which imposes a limitation on the maximum combination of torque and speed, see curve 4 of the load diagram, it is normal practice to match the diesel engine and turbocharger etc. according to a propeller curve 1 of the load diagram, equal to the fouled hull propeller curve 2.

Instead of point S2, therefore, point SP will normally be used for the engine layout by referring this service propulsion rating to 90% of the engine's specified MCR, correspondong to a 10% engine margin.

In other words, in the example the propeller's design curve is about 4% light running compared to the propeller curve used for layout of the main engine.

Running in heavy seas with heavy waves, point S_3

When sailing in heavy sea against, with *heavy waves,* the propeller can be 3-4% heavier running than in calm weather, i.e. at the same propeller power, the rate of revolution may be 3-4% lower. For a propeller power equal to 90% of specified MCR, point S_3 in the load diagram in Fig. 15 shows an example of such a running condition.

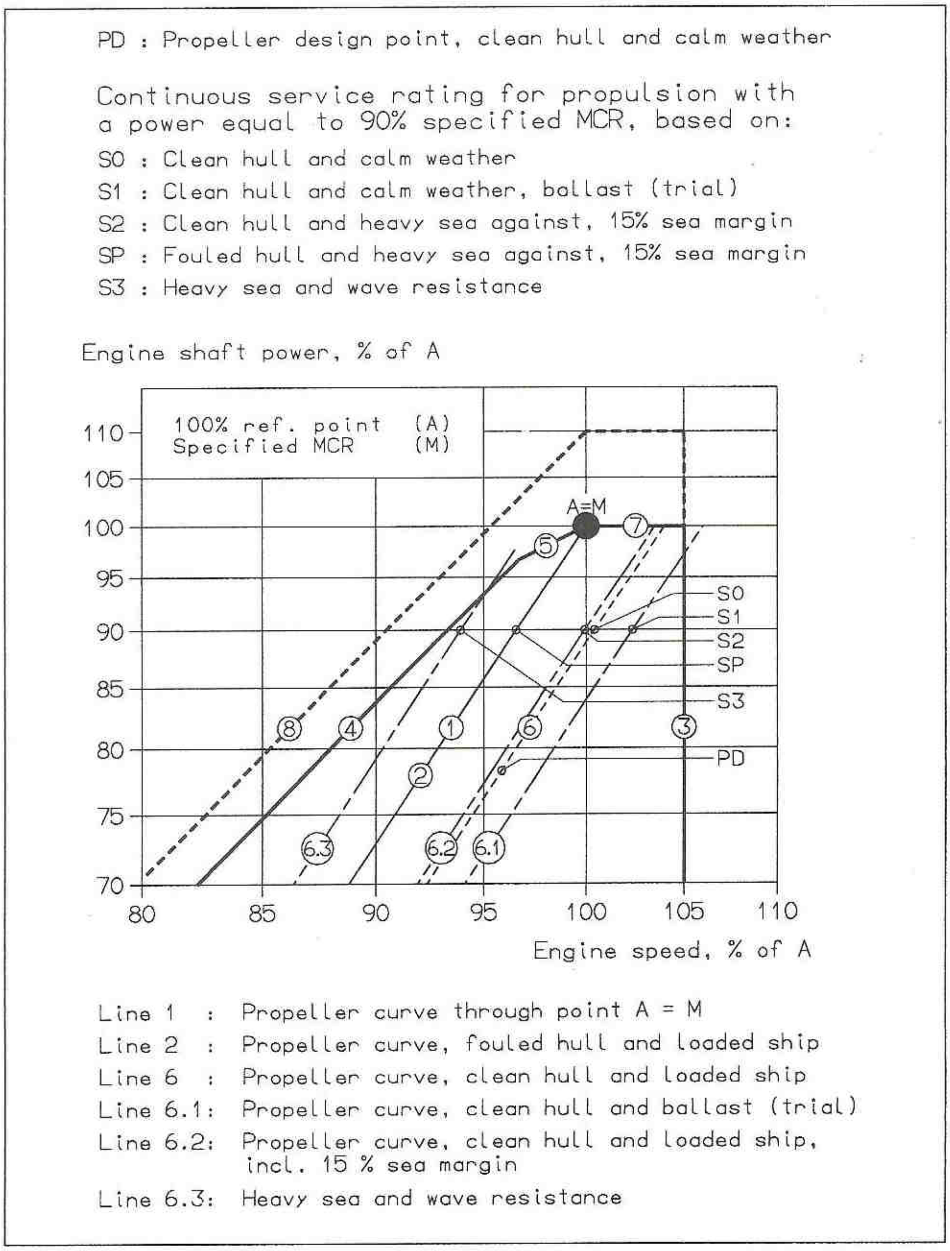


Fig. 15: Influence of different types of ship resistance on the continuous service rating

In some cases in practice with strong wind against, the heavy running has proved to be even greater and even to be found to the left of the limitation line 4 of the load diagram.

In such situations, to avoid slamming of the ship and thus damage to the stem and racing of the propeller, the ship speed will normally be reduced by the navigating officers on watch.

Ship acceleration and operation in shallow waters
When the ship accelerates and the propeller is being subjected to a larger load than during free sailing, the effect on the propeller may be similar to that illustrated by means of point S_3 in the load diagram, Fig. 15. In some cases in practice, the influence of acceleration on the heavy running has proved to be even greater. The same conditions are valid for running in shallow waters.

Sea running at trial conditions, point S1
Normally, the clean hull propeller curve 6 will be referred to as the trial trip propeller curve. However, as the ship is seldom loaded during sea trials and more often is sailing in ballast, the actual propeller curve 6.1 will be more light running than curve 6.

For a power to the propeller equal to 90% specified MCR, point S1 on the load diagram, in Fig. 15, indicates an example of such a running condition. In order to be able to demonstrate operation at 100% power during sea trial conditions, it may in some cases be necessary to exceed the propeller speed restriction, line 3, which during trial conditions may be allowed to be extended to 107%.

Load control system
As mentioned above, there is a risk of heavy propeller running during service in heavy weather conditions, and during acceleration of the ship (and also in shallow waters), which in some cases may cause overloading of the main engine. Therefore, MAN B&W Diesel have designed a load control system for main engines coupled to a fixed pitch propeller.

The load control system is designed to prevent excessive engine load, thus reducing the risk of damaging engine components.

This will ensure that the engine is always operated within the limits of the load diagram.

The system monitors the engine speed, torque, fuel pump index and scavenge air pressure. If the load limit is exceeded, the system automatically takes over control of the speed setpoint and reduces the setpoint until the engine is again running within the load limitations.

In addition to the performance advantages related to the main engine, the system facilitates the work for the engineers. In bad weather and when the ship accelerates, the load control equipment keeps the engine within the specified limitations without the necessity of taking manual measurements and evaluating these.

Furthermore, the captain can at any time utilise the maximum allowable/acceptable power without having to take additional measurements.

Closing Remarks

In practice, the ship's resistance 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.

When the ship's necessary power requirement, including margins, and the propeller's speed (rate of revolution) have been determined, the correct main engine can then be selected e.g. with the help of MAN B&W Diesel's computer based engine selection programme.

In this connection the interaction between ship and main engine is extremely important, and the placing of the engine's load diagram, i.e. the choice of engine layout in relation to the engine's (ship's) operational curve, must be made carefully in order to achieve the optimum propulsion plant. In order to avoid overloading of the main engine for excessive running conditions, MAN B&W Diesel may assist with a specially developed load control system.

If a main engine driven shaft generator – producing electricity for the ship – is installed, the interaction between ship and main engine will be even more complex. However, due to the flexibility of the layout and load diagrams for the MAN B&W engines, a suitable solution will nearly always be readily at hand.

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