2. PROPULSION SYSTEM

2.1. Main requirements of the propulsion system

The basic operating requirement for a main propulsion system is to propel the vessel at the required sustained sea speed for the range (or endurance) required. This propulsive action is obtained from the torque generated at the prime mover (main engine), which is then eventually converted into low-speed drive for the propeller shaft-line by means of a gearbox, and finally reaches the propeller itself. At the propeller shaft-line, the thrust from the propeller is also transferred to the hull of the ship. In complement, the propulsion system should also provide the vessel stopping, backing, and maneuvering capabilities.

In some special-purpose vessels such as military vessels, towboats and fishing vessels, there is more than one single speed requirement. Therefore, the power plant for these vessels must be designed according to their operating profiles. For example, the naval vessel power plant must satisfy the highest speed requirement projected for her, but the maximum power capability is seldom used. Hence, during the majority of the life of a naval vessel the ship is operated at cruising speeds, which correspond to approximately 60% of the maximum speed or about 20% of the propulsion rating. On the other hand, towboats and fishing are clearly established by extremes in the towing modes of operation, but free-running operations must also be given consideration.

Therefore, the selection of the main propulsion system is simultaneously a technical problem of fulfilling the basic operating requirements and an economical problem of life cycle cost compliance with the preliminary allocated budget; otherwise the preliminary studies must be revaluated.

2.2. Aspects preceding the selection of a certain propulsion system

Before a main propulsion plant can be designated, the power required for sustained operation and endurance must be tentatively determined. However, there is an interdependent relationship involved: the space and weight requirements for the propulsion plant vary with power rating and can have a significant effect on the ship configuration, while the dimensional and form characteristics of the hull and its approximate displacement are required to establish an estimate of the propulsive power required. It is apparent that marine engineers must coordinate their activities with those of naval architects from the earliest conceptual design stage to solve this “chicken-egg” type problem.

For future guidance, the following aspects should be primarily identified and determined by the agent responsible for selection of the propulsion system:

a) Determination of ship resistance (power curve);

b) Selection of the propulsor type, since this selection will have a strong influence on the selection process of the main propulsion plant;

c) Choice of the number of propulsors (directly related to maximum draught of the vessel, loading or power density* applied to the propulsor resulting from rating power and rotative velocity conditions, occurrence of cavitation and erosion, manoeuvring capability, maintainability and installation reliability, etc.).

* Note: Limit value of power density on a propeller is a ratio of 800 [kW/m^2], where the numerator corresponds to the shaft power P_S, and the denominator corresponds to the propeller diameter squared.

If a conventional single-screw propeller is assumed (predominant for most applications in merchant ships), a propeller with a larger diameter is theoretically more efficient, providing there is enough space for the propeller including a sufficient clearance between propeller and hull. Due to hydrodynamic effects and/or cavitation the hull bottom structure can be mechanically excited, which can cause heavy vibrations at the stern, specially when the excitation frequency is close to hull natural frequencies of vibration. Still in respect to propeller efficiency, larger efficiencies are obtained at lower speeds. This fact is contradictory with internal combustion engines efficiencies, which tend to have larger efficiencies at higher rotational speeds. Therefore, reduction gearboxes are usually installed in the shaft line between the prime mover and the propeller in order to convert high speed drive from the prime mover into low-speed drive for the propeller. Notice that an increase of rotational speed of the engine corresponds to a reduction in the engine torque, and therefore the robustness of the block can be alleviated so that the total cost of the equipment is also reduced for the same brake power.

2.2.1 Estimating the required power

Closely related to the hull form design is the prediction or estimation of required power to produce a given ship speed.

Initially, a preliminary estimate is made, but as the designer refines the design by sequential steps around the design spiral increasingly refined power estimates should be made. The following lists systematically presents the methods of increasingly improved power estimates:

a) 1st Admiralty Equation – Gross approximation from known ships of similar size and speed by Admiralty constant (N_A):

$$P_S = \frac{\Delta^{2/3} V^3}{N_A} \tag{2.1}$$

, where:

$$N_A = k\left(\sqrt{L} + \frac{90}{V}\right), \text{ and } k = \frac{L}{30} + 14$$

V = Ship's speed, in [kts];

L = Ship's length, in [m];

P_S = Shaft horsepower, in [HP];

Δ = Ship's displacement, in [ton].

b) Estimation of effective power from Taylor-Gertler's Series, with:

$$P_S = P_{EA} \times \frac{1}{\eta_H} \times \frac{1}{\eta_O} \times \frac{1}{\eta_R} \times \frac{1}{\eta_S} \tag{2.2}$$

, where:

$P_{EA} \approx 1.1 P_E = R_T.V$ = effective power with appendages;

$\eta_H = \frac{R_T.V}{T.V_A} = \frac{1-t}{1-w_T}$ = hull efficiency (at design point approx. 0.90-1.10); notice that the velocity of advance of the propeller can be determined using: $V_A = V(1-w_T)$;

t = thrust deduction factor; suction of propeller over the rear of the ship causes added resistance, since the increase in velocity over the stern of the ship will increase frictional resistance. Adding to this, the propeller has a wave system associated to it, which may change the wave resistance of the ship. This augmented resistance may be defined as: $R_T = (1-t)T$, which can be obtained on model tests with self-propelled models by direct measurements of model resistance without propeller and determination of thrust at the self propulsion point.

w_T = (Taylor's) wake fraction; the change of velocity caused by the hull at the propeller position; nominal effective wake can be measured using a pitot tube in order to measure velocity, or directly by means of a Doppler laser device;

η_O = open water propeller efficiency (at design point approx. 0.60-0.75);

η_R = relative rotative efficiency (at design point approx. 0.95-1.02);

$\eta_S = \frac{P_D}{P_S}$ = shaft transmission efficiency (at design point the following approximations can be used: 0.99 no gearbox, 0.98 non reversible gearbox with and without intermediate gear, 0.97 reversible gearbox);

P_D = delivered power or propeller power (power measured at propeller flange, see Figure 2.1);

$PC = \eta_S.QPC$ = propulsive coefficient; QPC is designated the quasi-propulsive coefficient (at design point approx. 0.60);

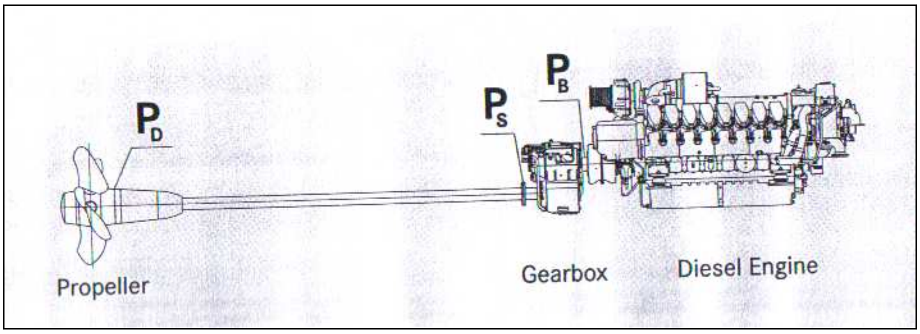


Fig. 2.1 – Scheme of a propulsive unit (side view).

$P_T = T.V_A$ = power thrust of the propeller (this power can be measured at thrust block bearing, see Figure 2.2);

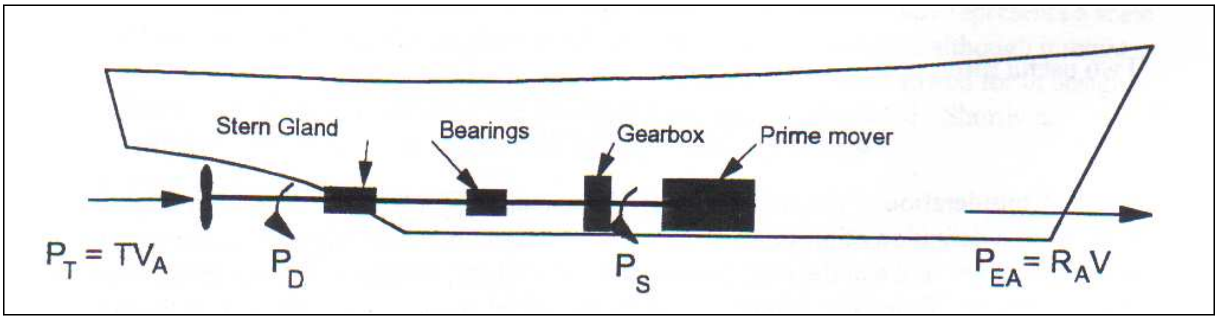


Fig. 2.2 – Scheme of a propelled vessel (side view).

Finally, the shaft power is therefore simply given by:

$$P_S = \frac{P_{EA}}{PC} \tag{2.3}$$

The values of t, w_T, η_O, η_R and η_S may be estimated from similar ships. Values of t and w_T can be obtained from:

- model tests;
- systematic series;
- empirical formulations, as described below:

Taylor: $$\begin{cases} w_T = -0.05 + 0.50.C_B \Leftarrow 1 \text{ single propeller} \\ w_T = -0.20 + 0.55.C_B \Leftarrow 2 \text{ propellers} \end{cases} \tag{2.4}$$

, where: $C_B = \dfrac{\nabla}{LBT}$ = block coefficient.

Bragg: $\frac{t}{w_T} = a - b\frac{C_B}{C_W}$ (2.5)

, where: $C_W = \frac{A_W}{LB} =$ waterplane area coefficient.

Tab. 2.2 – Empirical thrust deduction factor coefficients a and b.

C_B	0.64	0.68	0.72	0.76	0.82
a	2.82	2.72	2.65	2.71	2.80
b	2.54	2.43	2.29	2.28	2.30

Note: These empirical values are valid for one single propeller; for vessels with two shaft lines the ratio $\frac{w_T}{t}$ should be increased 10%.

Values of *QPC* can be also obtained from empirical formulation, as follows:

$$QPC = k - \frac{n\sqrt{L}}{10000} \tag{2.6}$$

, where:

$k = 0.84$;

$n =$ shaft line [rpm];

Taylor-Gertler series power prediction considers the total resistance coefficient, C_T, is given by:

$$C_T = C_R + C_F + C_A \tag{2.7}$$

, where:

$C_R =$ residuary resistance coefficient, which can be extracted from the series charts;

$C_F =$ frictional resistance coefficient, given by ITTC 1957 as:

$$C_F = \frac{0.075}{(\log_{10} R_n - 2)^2}$$

where:

$R_n = \frac{VL}{\nu} =$ Reynolds number (low value corresponds to laminar flow and high values corresponds to turbulent flow;

$C_{F(6m.o.d)} = C_{F(service)}(1 + margin)$, with *margin* = 0.1- 0.2

$C_A = 0.0004$ = model-ship correlation allowance in resistance coefficient, given by Schoenherr.

The value of the total resistance coefficient can now be evaluated, and the clean effective power for each speed can be obtained, using:

$$P_E = \frac{1}{2}\rho.V^3.S.C_T \tag{2.8}$$

, where:

$S = C_S\sqrt{\nabla L} =$ wetted surface area of the hull [m^2];

, where:

$C_s =$ wetted surface coefficient, which can be estimated or graphically obtained, as a function of the midship coefficient C_M and the beam to draught ratio.

c) Estimation of effective horsepower from Taylor-Gertler's Series (TGS) or Series 60, with correction needed to predict the power of the new ship from a known ship (basis ship) of similar size and speed, given by:

$$P_{S\_new} = P_{S\_basis} \times \frac{P_{EA\_new}}{P_{EA\_basis}} \tag{2.9}$$

d) Same as above but with further corrections for:

d.1) bilge keels;

d.2) addition of bulbous bow;

d.3) effect of change in propeller rpm, diameter, blade thickness or advance speed coefficient, etc.;

d.4) change in LCB or any form distortion;

d.5) loss due to bow thruster opening interference;

d.6) change in drag of standard appendages (rudders, struts, bossings, shafting).

e) Same as above except that the P_{EA_new} test results from a small model of the new ship substitute the P_{EA_new} derived from TGS.

f) P_S from self-propelled large model tests with appendages and stock propeller;

g) P_S from self-propelled large model tests with appendages and final designed propeller.

2.2.2 Endurance

There is an empirical formula related to the k factor of the 1st Admiralty equation (2.1) to determine the fuel consumption in 24 hour expressed in tones, given by:

$$k = \frac{\Delta^{2/3} V^3}{q_{24hr}} \qquad (2.10)$$

With the provided information of the shaft power and diesel engine speed from the engine manufacturer, the fuel consumption can be calculated (assuming a 5% margin). Then given a certain operation profile of the vessel and information about the available fuel volume (about 95% of the total capacity of the tanks), the endurance time is easily determined.

2.3. Factors involved on propulsion system selection

Like many other general design projects, the design of a machinery plant largely consists of the integration of a number of units and elements into a functioning system. The process entails selecting components, adjusting each to the constraints imposed by all the others, and arranging them to achieve the required system performance, a satisfactory configuration, and an acceptable life cycle cost.

Fundamental to the design of a main propulsion plant is the coordination of the prime mover with a transmission system and a propulsor. A number of possible machinery combinations may be considered by the marine engineer in making the selection. As indicated in Figures 2.3.a and 2.3.b, even with the range of considerations confined to those most commonly considered, a large number of alternatives is feasible.

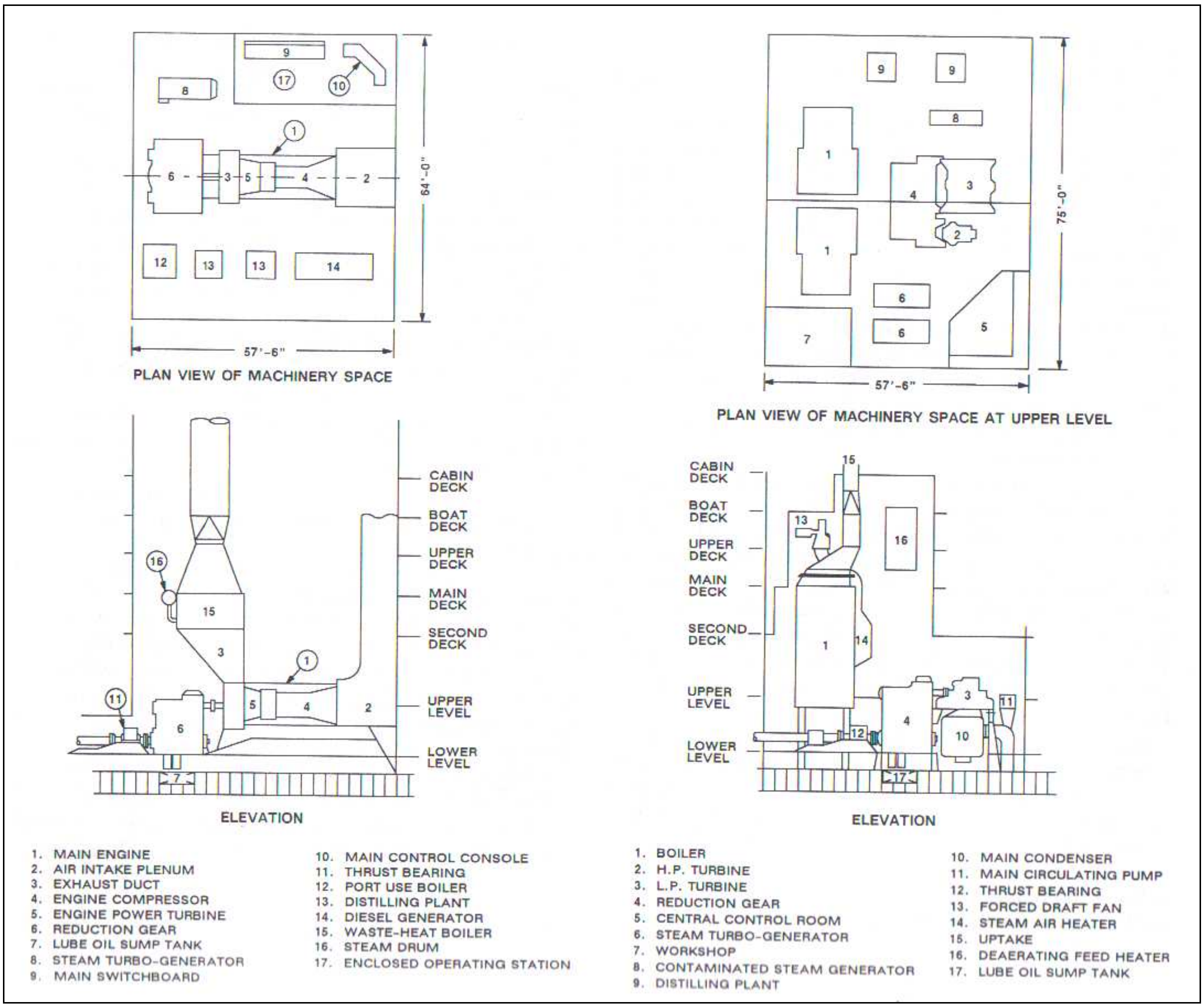


Fig. 2.3.a – Gas turbine power plant (left) and Steam turbine power plant (right).

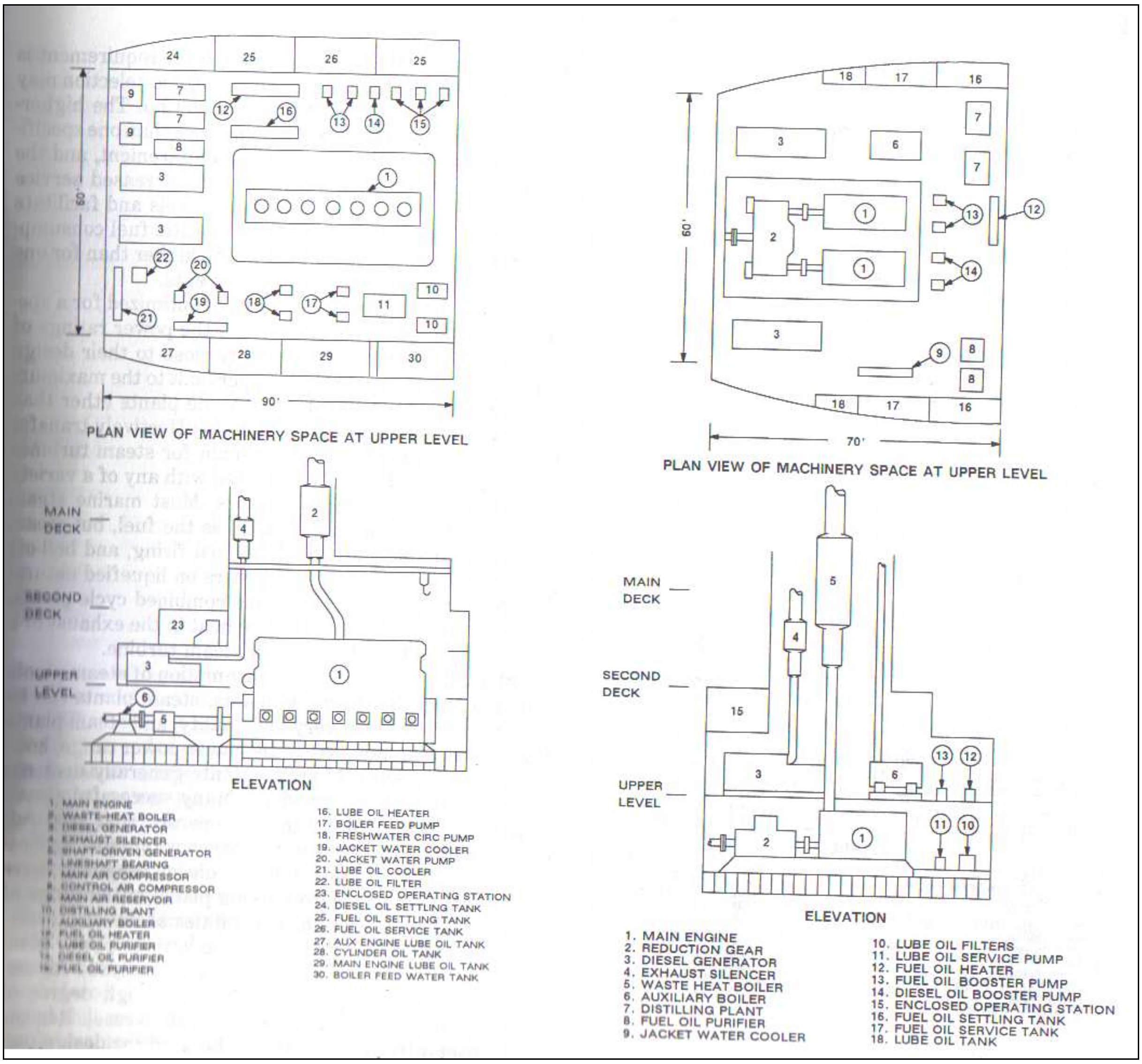


Fig. 2.3.b –Low-speed diesel power plant (left) and medium-speed diesel power plant (right).

Therefore, major considerations used by the designer in the early design stage of the design spiral are:

a) Power range;

b) Specific fuel consumption, [kg/kW.hr];

c) Specific weight, [kg/kW];

Other performance factor often considered simultaneously with the above are:

d) Initial and life-cycle costs;

e) Installation reliability;

f) Maintenance and repair requirements;

g) Type of fuel (lower caloric value);

h) Shaft line reversing capability;

i) Level of manning required for running the plant;

j) Volume requirements in the machinery space and adaptability to ship's configuration.

2.3.1 Main engine selection

Considering all of these factors is in itself a difficult design trade-off problem subjected to optimization procedures. The power range requirements are the most straightforward and for very high power the selection is very simple as can be seen from the table below (where 1.0 [kW] = 0.746 [HP]):

Tab. 2.2 – Typical power ranges of main engines.

Main engine	**Power range [HP]**
Steam turbine	8 000 – 100 000 +
Gas turbine	500 – 45 000
Diesel	15 – 45 000
Gasoline	10 – 500

2.3.2 Propulsor selection

Once the ship speed requirements and resistance have been tentatively established, the next step is to select the type of propulsor, which can be a Fixed Pitch Propeller (FPP) or Controllable Pitch Propeller (CPP), cycloidal propeller, podded propeller, waterjet, etc..

a) Propellers selection

a.1) Propeller geometry

To understand the hydrodynamic action of a propeller it is essential to have a thorough understanding of basic propeller geometry and the corresponding definitions. Figure 2.4 shows what is meant by rake and skew of a propeller. The use of a skew has been shown to be effective in reducing vibratory forces, pressure hull induced vibrations and retarding cavitation development. With rake, the stress in the blade can be controlled and slightly thinner blade sections can be used, which can be advantageous from the hydrodynamic considerations.

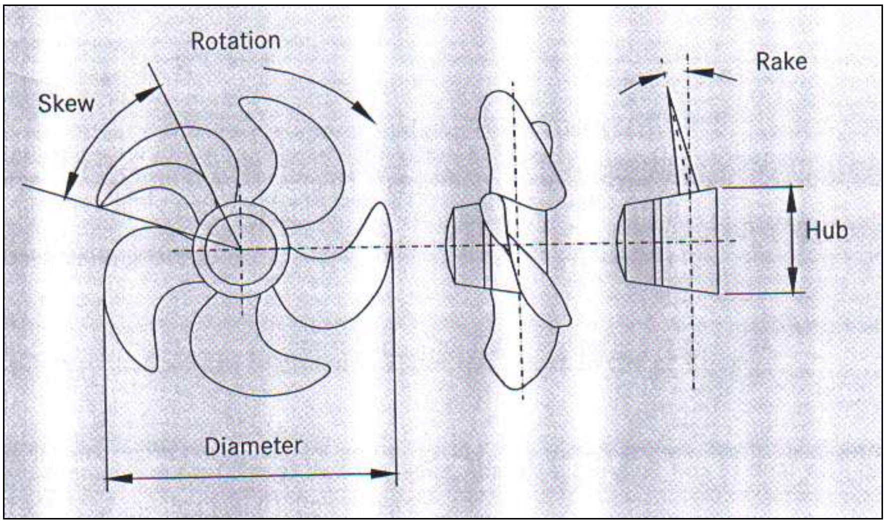


Fig. 2.4 – Scheme of propeller geometry (skew and rate).

Every propeller needs a hub to fix blades and, in case of a CPP, to place the control mechanism for the blades. This results in different hub sizes for a FPP and a CPP and is a characteristic difference between these two types. The hub size of a CPP is 10-15% larger (related to diameter). Hence, the Blade Area Ratio (BAR = A/A_0), defined as the blade area of the outline projection of the blades divided by the propeller disc area, of a CPP are smaller than BAR of a FPP. Moreover, as a CPP is usually fully reversible in the sense that its blades can pass through zero pitch condition, care has to be taken that the blades do not interfere with each other. Therefore, with equal number of blades, a CPP will have a somewhat smaller BAR than a FPP.

The expression P/D is the commonly used pitch ratio (p). Alternatively, the pitch angle θ can be given to relate axial distance to circumferential distance after one propeller rotation (see Figure 2.5). Due to the geometry of the blade the pitch angle varies from the hub to the tip. Hence, the characteristic pitch angle is defined at a propeller ratio x = r/R = 0.7, as follows:

$$\theta = \arctan\left(\frac{P/D}{x\pi}\right) \tag{2.11}$$

Unfortunately, there are several pitch definitions and the distinction between them is of considerable importance to avoid analytical mistakes, namely:

- nose-tail pitch;
- face pitch.

The nose-tail line is today the most commonly used and referenced line (see Figure 2.5). The face line is basically a tangent to the section of the pressure side surface and was used on older model test series (e.g. Wageningen B Series). Although the difference is not large, it can be the reason for using different values for the same propeller.

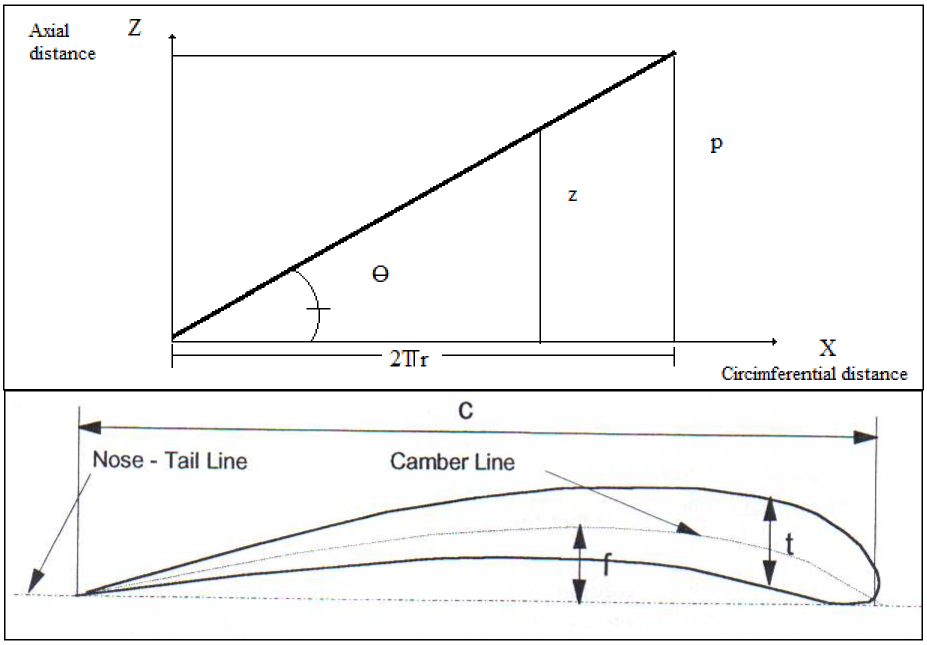


Fig. 2.5 – Relation between axial and circumferential distances after one propeller rotation (above) and scheme of propeller blade geometry: definition nose-tail chord length (below).

a.2) Propeller type selection

The selection of a propeller for a particular application usually is a result of the consideration of different factors. These factors can be determined in pursuit of maximum efficiency with respect to:

- noise limitation;
- ease of maneuverability;
- cost of installation, and so on.

Each vessel has to be considered with regard to its own special application. As indicated by Figure 2.6, some types of propellers are inherently more efficient than others for particular applications.

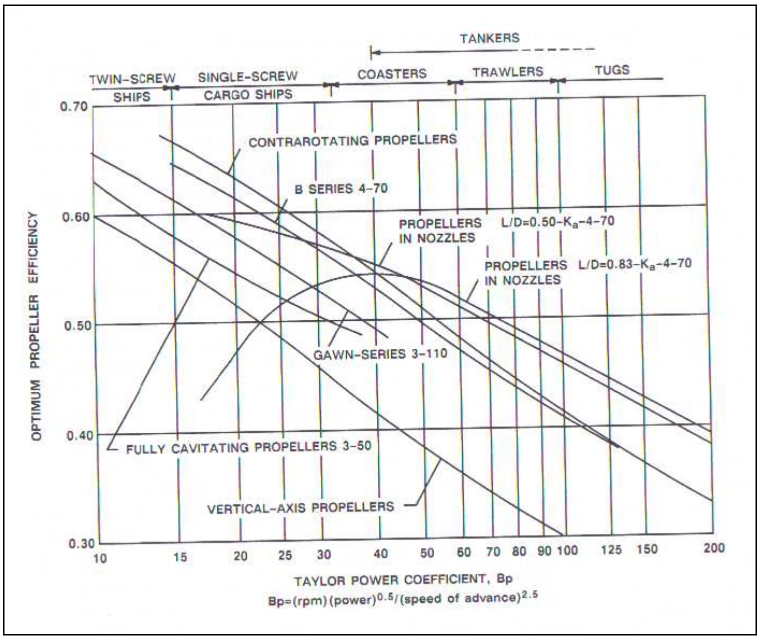


Fig. 2.6 – Comparison of optimum efficiency values for different types of propellers.

Figure 2.6 shows that relatively slow but high-powered vessels such as trawlers and tugs have inherently lower propeller efficiencies, but improved efficiencies are achievable by using propellers in nozzles. For higher speed ships, however, contra-rotating propellers are seen more efficient.

The selection of the propellers may not be a simple process, because in order to establish the type of propellers, it may be necessary to at least tentatively select the type of main propulsion machinery. For example, the gain in efficiency offered by selecting contra-rotating propellers or other complex propellers must be traded-off against the advantage of simplicity provided by conventional main propulsion machinery and shafting arrangement.

Similarly, the selection of the number of propellers may also involve a trade-off. In general, vessels may be single, twin, triple, or quadruple screw. That is to say, the total power required to propel a vessel may be distributed (usually equally) among as many as four propellers. From the point of view of initial and operating costs, fewer propellers are preferred; however, the magnitude of the power requirement, or the constraints on the propeller diameter, may force the selection of multiple-screw arrangement to avoid an excessive propeller loading and the attended cavitation that could otherwise result. In addition, there may be other factors in a given case, such as reduced vulnerability ort improved maneuverability, that may favor the use of multiple propellers.

The choice between fixed pitch (FPP) and controllable pitch propeller (CPP) has been a long contested debate between the proponents of the various systems. Controllable pitch propellers have gained complete dominance in RO/RO vessels, ferry and tug markets with vessels of over 1500 [KW] propulsion power with an operational profile that can be satisfied by a CPP better than by a two stages (output speeds) gearbox. For all the other purposes, the simpler fixed pitch propeller appears to be a satisfactory solution. Comparing reliability between the simply build up FPP and the mechanical complex CPP, the experience shows that CPP has achieved the status of being a reliable component.

The CPP has the advantage of permitting constant speed operation of the propeller. Although, this leads to loss of efficiency, it does readily allow the use of shaft drive generators, if this is a demand in the operational profile of the ship.

During the last years the electric drive podded propeller has been arising on the market. Without the need of a gearbox and controllability of the electric motor, a fixed pitch propeller seems to be the best choice. But it must not be forgotten to compare the economical aspects of an extended motor control with the cost of a CPP.

a.3) Propeller size

To determine the propeller diameter (D) for certain delivered power (P_D) at shaft speed (n) and a design ship's speed (V) is a complex routine. For some propellers, calculation procedures are available, which can be found in literature.

The size of a propeller cannot only be calculated theoretically, but must also be adapted to the ship. The ship must provide the necessary space for the propeller including a sufficient clearance between propeller and hull (see Figure 2.7). Due to hydrodynamic effects and/or cavitation the hull and the rudder can be mechanically excited, which can cause heavy vibrations at the stern or rudder with the possibility of mechanical failures.

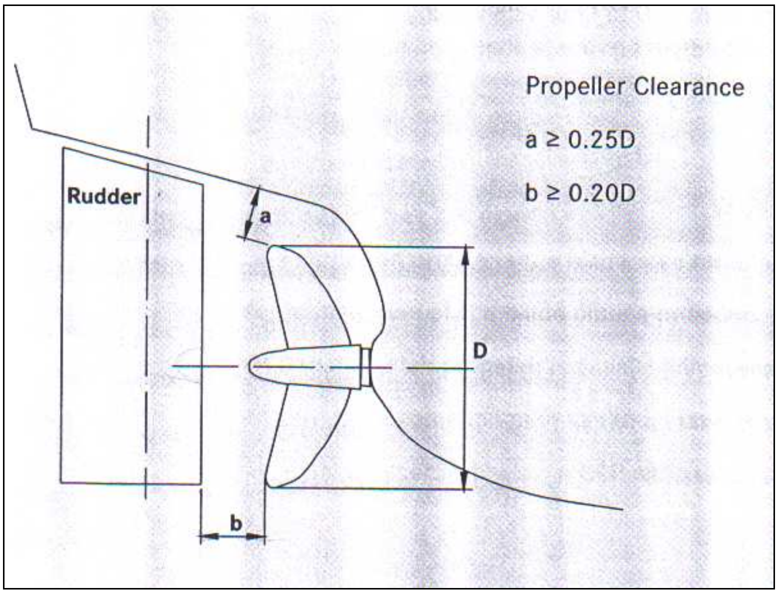


Fig. 2.7 – Scheme of propeller clearances.

The values shown in Figure 2.7 are only a design proposal. For more detailed information, see the recommendations of a Classification Society or responsible yard.

a.4) Cavitation and thrust breakdown

The majority of the vessels of approximately 100 [ton] displacement or more can control, not eliminate, the effects of cavitation. This means to reduce the erosive effect on material and to improve its hydrodynamic performance as well as its behavior as a source of vibration excitation. However, it must be recalled that there are very few propellers that are free from cavitation. Most of them experience cavitation at some position of the propeller disc.

A few words about the effect of thrust breakdown: the power density of a propeller can only be increased to a certain limit, which depends on the propeller parameters and especially on the blade area ratio. Obviously, the cavitation occurs first at the tip section of a suction side of the blade and extends downwards with higher power consumption. It is a matter of definition when these effects are called "thrust breakdown", e.g. if the cavitation exceeds below the 0.5 radius. Subsequently, the propeller efficiency will decrease rapidly.

a.5) Direction of propeller rotation

For vessels with a single propeller the influence on maneuvering is entirely determined by the "paddle wheel effect". When the ship is stationary and the propeller is started, the propeller will move the afterbody of the ship in the direction of rotation. Thus with a fixed pitch propeller, this direction of rotation will change with the direction of rotation, i.e., is ahead or astern thrust. In the case of a controllable pitch propeller the motion will tend to be unidirectional because only the pitch change from ahead to astern position and the direction of rotation remains the same. Therefore, assuming that starboard is the main docking side and there is an advantage to push off from the quay with astern thrust, FPP should have clockwise direction of rotation and CPP should have counter-clockwise (see Figure 2.8).

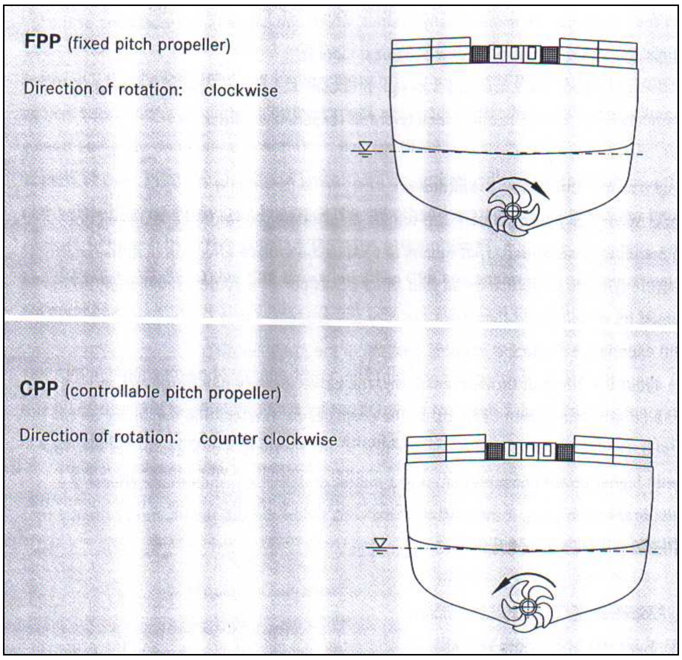


Fig. 2.8 – Single propeller direction of rotation (for prevailing starboard docking side).

For vessels with twin propellers the influence on propeller efficiency is determined by the "wake field contra-rotation effect". It has been found that the rotation present in the wake field, due to the flow around the ship, at the propeller disc can lead to a again in propeller efficiency when the direction of rotation of the propeller is opposite to the direction of the rotation of the wave field. Therefore, in addition to the "paddle wheel effect", for conventional hull forms a supra-diverging rotation will be always preferable for both FPP and CPP (despite for the sake of clarity CPP is shown with supra-converging rotation on Figure 2.9).

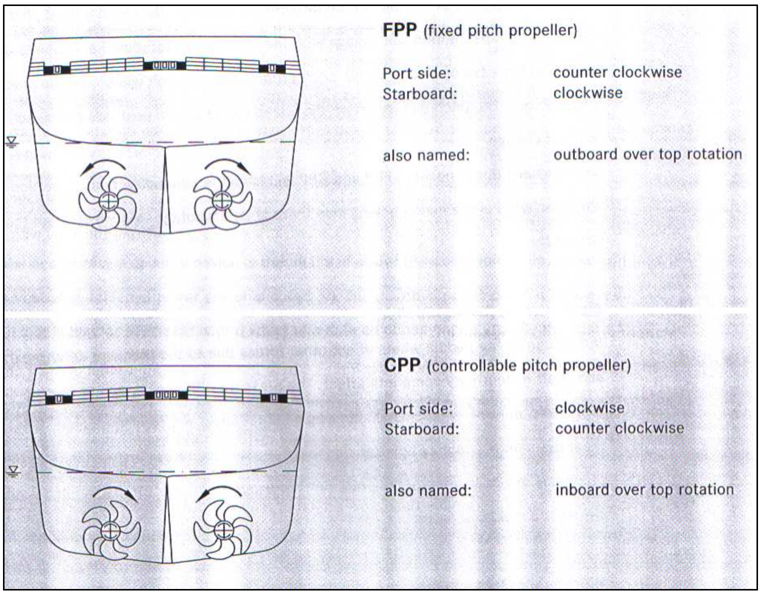


Fig. 2.9 – Twin propellers direction of rotation.

a.6) Selection of propeller blade number

Blade numbers generally range from three to seven. For merchants ships four, five or six are favored, although many tugs and fishing vessels frequently use three bladed designs. In naval applications where the generated noise become important blade number of five and above predominate. In addition, a odd number of blades is always favored in contrast with an even number of blades for reasons associated with propeller induced noise and vibration.

a.7) Propeller for high speed vessels

For high speed vessels where both the advance and rotational speeds are high and the propeller immersion low, a point may be reached where it this not possible to acceptably control the effects of cavitation. To overcome this problem, the blade sections are permitted to fully cavitate, so that the cavity developed at the suction side of the blade extends beyond the trailing edge and collapses into the wake of the blade in the slipstream. Such propellers are called supercavitating propellers and are frequently used in applications on high-speed naval vessels and pleasure crafts.

For small high-speed crafts the concepts of a surface piercing propeller has been successful. This propeller operates partially in and partially out of water. The design immersion measured from the free surface to the centre line, can be reduced to zero or is controllable (e.g. Arneson Surface Drive). In the partially immersed condition the propeller blades are commonly designated to operate such that the pressure side of the blade remains fully wetted and the suction side is dry.

a.8) Ducted propeller or propeller in nozzles

The duct or nozzle can be used to accelerate the flow. This will (in an ideal flow) lead to a gain in efficiency. In real fluids this gain only occurs at high thrust loading. It may be advantageous to the design as a whole to use a small diameter propeller and accept a lower propeller efficiency. This will lead to a high thrust loading and a possible reason for using a ducted propeller. It is also possible to decrease the flow through the duct. By Bernoulli, the pressure will increase and the combination of high pressure and low velocities will delay the onset of cavitation. Finally, the duct can be used to straighten out a non-uniform inflow. This will reduce the vibration which might otherwise be generated by the propeller.

a.9) Contra-rotating propeller

This propeller utilizes rotational kinetic energy in the slipstream of the first propeller and, in so doing, ejects essentially axial flow in the final slipstream. Efficiency is high and, for a given diameter, blade loading is low at the expense of mechanical complexity.

b) Selection of other types of propulsors

Furthermore, attaining the maximum propulsive efficiency does not necessarily constitute the main operational requirement. Special service vessels may require some specific kind of propulsors. Moreover, during the last years other type of propulsors have been made available by the manufacturers to specific applications, as shown in Figures 2.10.a and 2.10.b.

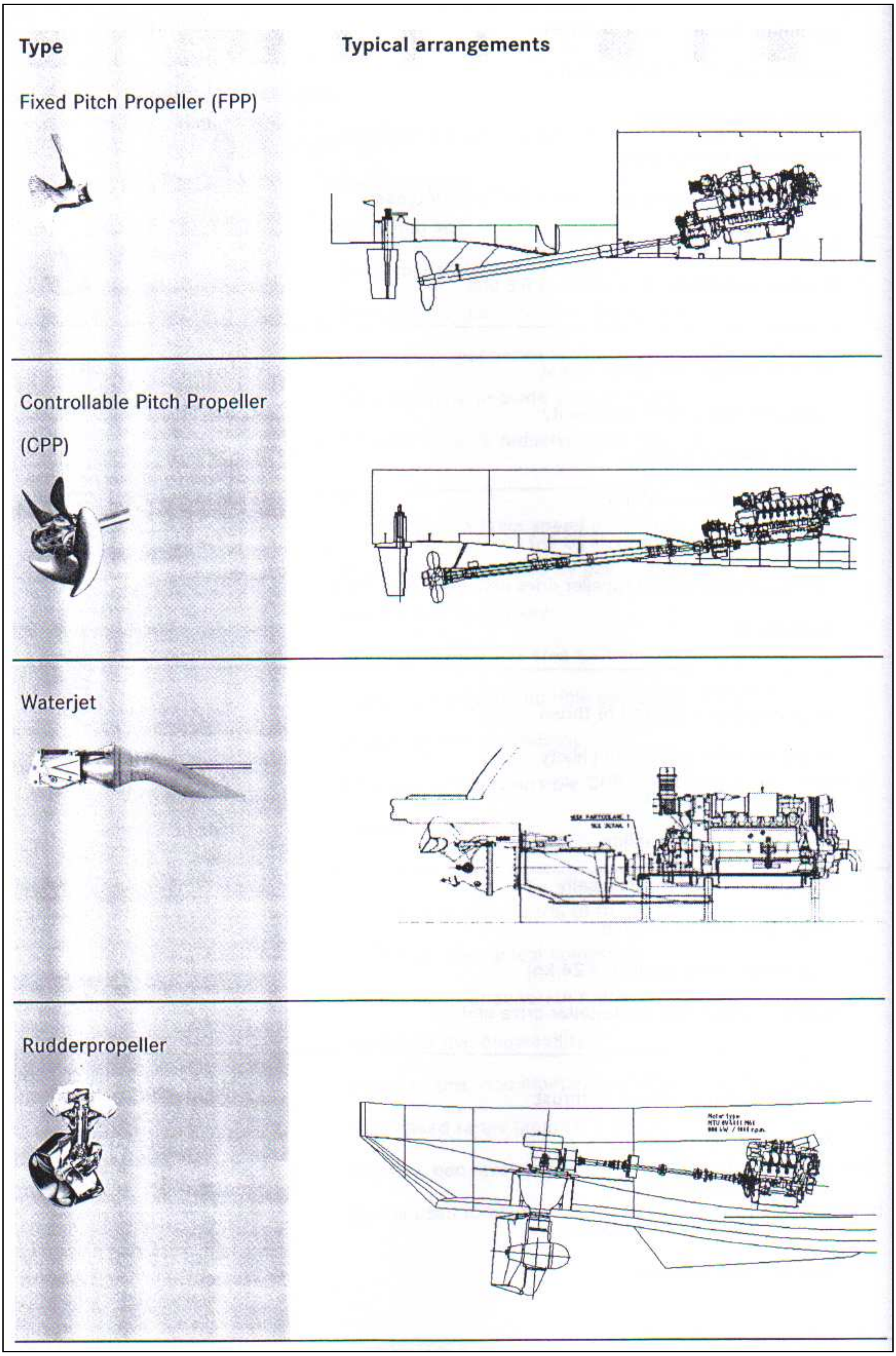


Fig. 2.10.a – Different types and arrangements of modern propulsors.

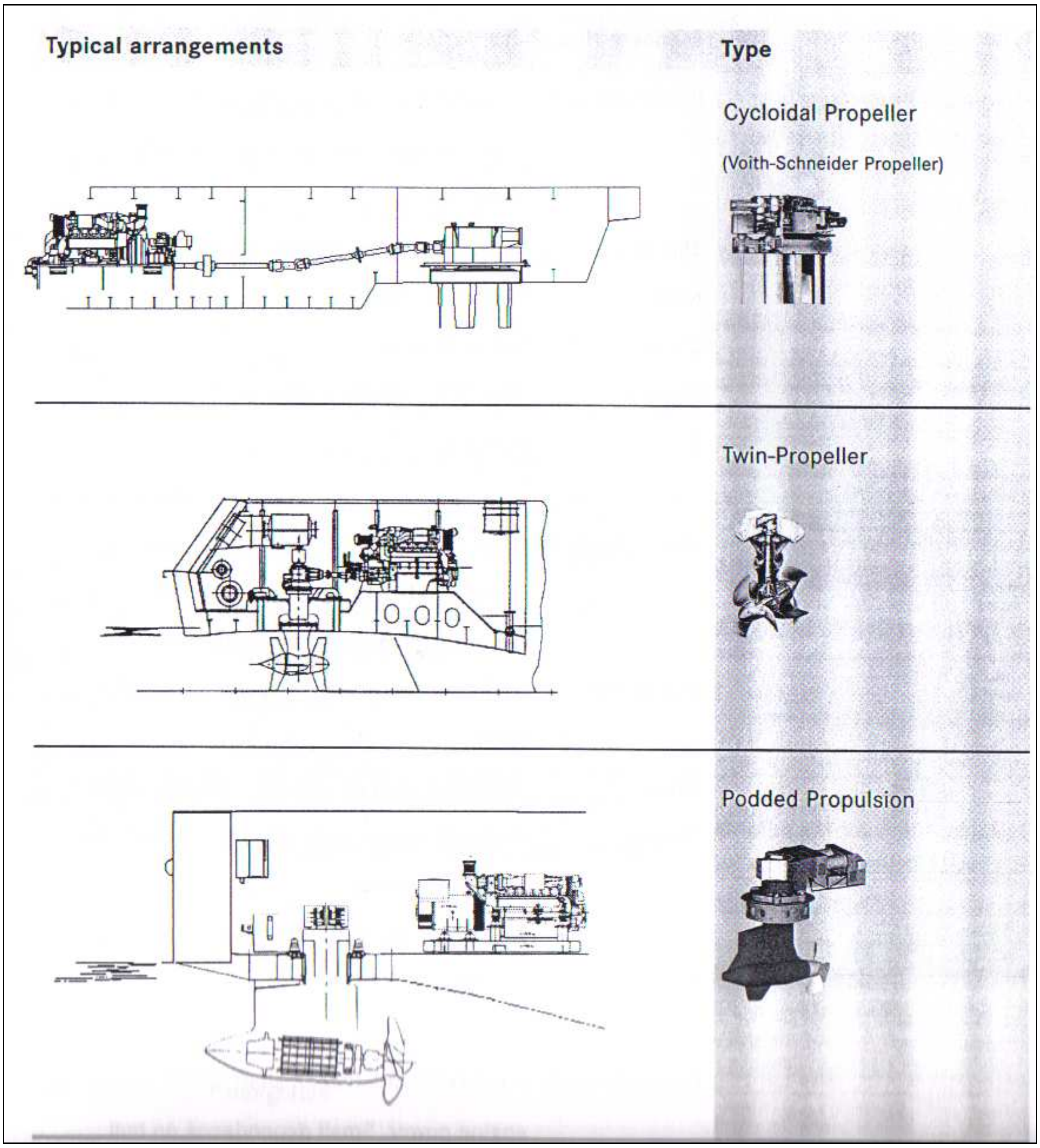


Fig. 2.10.b – Different types and arrangements of modern propulsors.

2.3.3 Main steps on propulsion system selection

Main steps on propulsion system selection are as follows:

a) Owner requirements;

b) Ship preliminary general arrangement and other possible solutions;

c) Ship preliminary structural design;

d) Hull form definition;

e) Calculation of hydrodynamic resistance;

f) Propulsor design (e.g. hydrodynamic characteristics of the propeller);

g) Prime mover selection (manufacturer, model, shaft power and rotation);

h) Gearbox selection – only if necessary (manufacturer, model and type, reduction relation selection);

i) Project optimization and equipments selection following the design spiral procedure.

2.4. Matching between propulsor and diesel engine characteristics

The matching of an engine and its propulsor is a design process that seeks to establish the optimal fuel-to-thrust conversion under rated operating conditions, while ensuring that all possible operating conditions are acceptable to each component. In this section matching between a diesel engine and a conventional Fixed-Pitch Propeller (FPP) will be first discussed, since this is the most common propulsion plant. Secondly, matching between a diesel engine and a Controllable-Pitch Propeller (CPP) and a waterjet propulsor will be briefly discussed, since these plants have been more frequently utilized over the recent years.

2.4.1 Diesel engine performance diagram

The diesel engine performance diagram serves as the basis for a number of calculations, but one of its most important functions is to indicate the rotation speed (n) and the shaft power, or, brake power (P_B) limits that must be observed in different applications.

Figure 2.11 shows the scheme of a typical diesel engine performance diagram with its designations. The diagram can be separated into:

a) operating envelope;

b) operating area;

c) propeller curve;

d) adaption on the application.

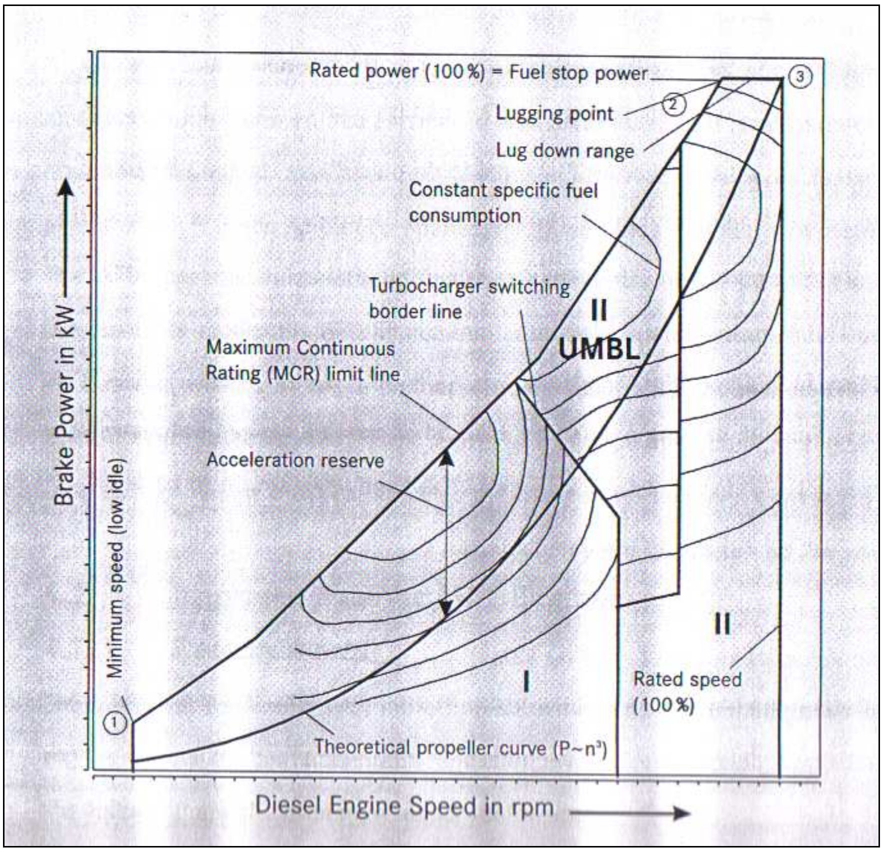
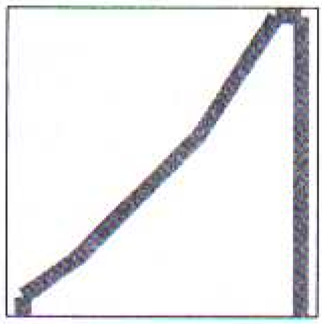
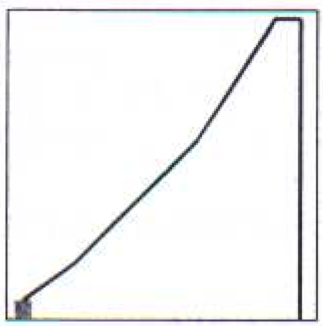
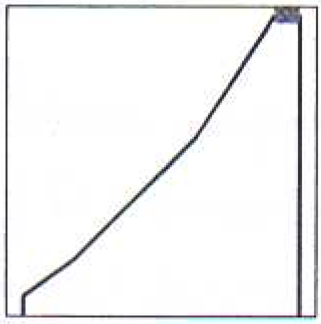
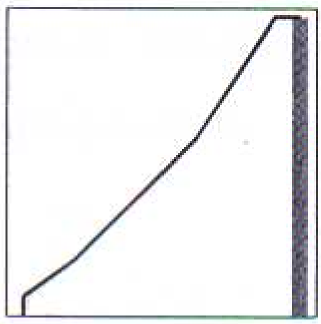


Fig. 2.11 – Structure of a diesel engine performance diagram.

In respect to operating envelope, this is defined by the following limits:

The operating area of the diesel engine is limited by the Maximum Continuous Rating (MCR)	
On the left side the limit will be defined by the minimum speed (n), the lowest self contained speed of the diesel engine. This speed should not be mixed up with the minimum clutch engaging speed, which will be somewhat higher.	

The curve between the minimum speed and lugging point shows the operational limits determined by thermal, mechanical and/or combustion related issues.	
The upper side will be defined by the fuel stop power or rated power (100% brake power). Dependent on the application and the chosen TBO, the maximum available power output of the diesel engine will be called lug-down range. The leftmost point of the lug-down range is the lugging point. Below this speed, the rated power is no longer available. Sometimes this type of rating, where the possible available output power is limited by the diesel engine controller, is called flat rating.	
The right side limit will be defined by the rated speed (100%) of the diesel engine.	

2.4.2 Propeller curve

In Figure 2.11 a simple theoretical propeller curve is shown with its design point at rated power (100% P_B) and rated speed (100% n). The difference between propeller curve and MCR curve is called acceleration reserve. This reserve can be used during dynamic operations. The propeller curve plotted in the diesel engine performance diagram shows only the stationary condition, where all the forces are balanced. In this case, the diesel engine moves only on the propeller curve and all the other points in the operating area are out of reach because there is no power speed relation possible.

In a non-stationary case, this situation changes visibly. Starting at a stationary point the diesel engine tries to accelerate. The speed of the diesel engine increases and so does the propeller. But the relation between propeller speed (n) and ship's speed (V) are not in balance any more. Hence, the propeller tries to move faster for the given ship's speed and it absorbs more power than in a balanced situation. In this situation the power output of the diesel engine at a certain speed (n) can be higher than the stationary propeller curve demands. If the acceleration is too fast the MCR curve will act as limiter. Generally, large acceleration reserves allow for fast maneuvering if gearbox and shaft line are to be designed for this application.

2.4.3 Load curves

Most of the ships have individual hull forms. Nevertheless they can be sorted into three different groups with characteristic propeller curves.

The basis will be formed by a monohull as a typical displacement vessel (see Figure 2.12).

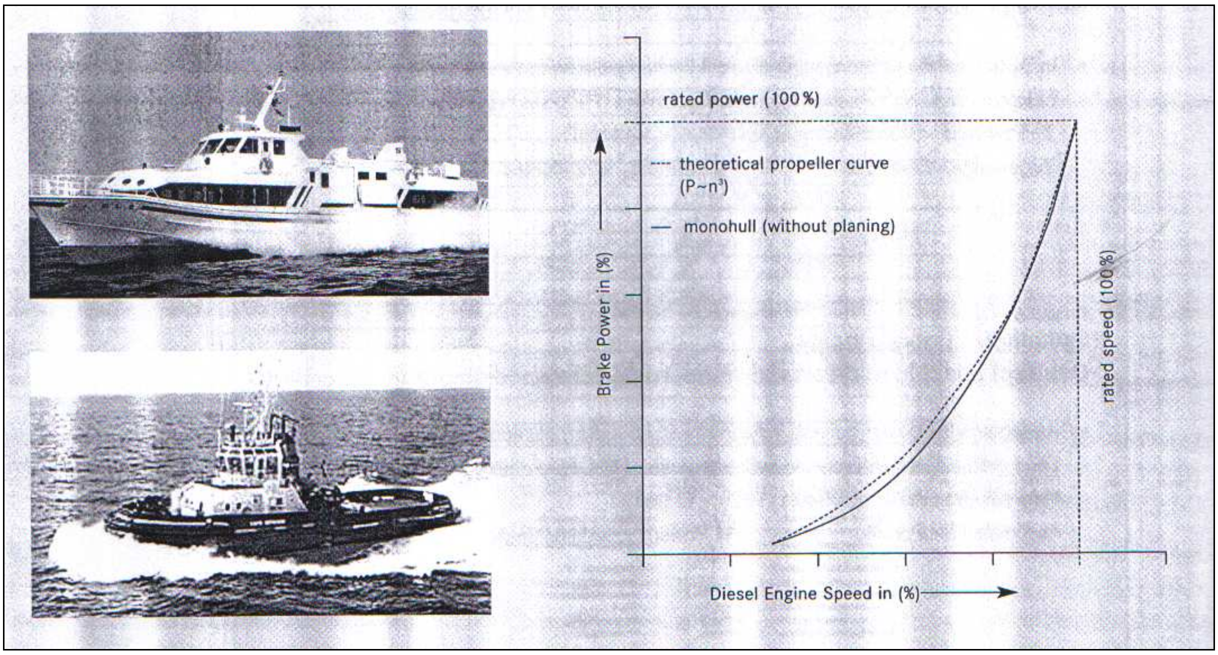


Fig. 2.12 – Typical propeller curve of a monohull (displacement vessel without planing).

As it can be seen from Figure 2.12, there is only a small difference between individual and theoretical propeller curves.

In Figure 2.13 the speed range of the vessel has been increased and the individual propeller curve starts to build a hump. The ship moves from plain displacement into planing regime (hydrodynamic lift forces are generated at the hull bottom and hard chines). The theoretical curve without planing is also plotted in order to show that the resistance of the craft decreases significantly when planing occurs. The difference between individual and theoretical curves is obvious.

Planing depends on many factors such as ship size and speed, and will be mainly influenced by the hullform. A ship designed for planning will enter this regime earlier than other displacement or semi-displacement ships. Therefore, it is not clear how the ship will behave if it is not known for what it is designed.

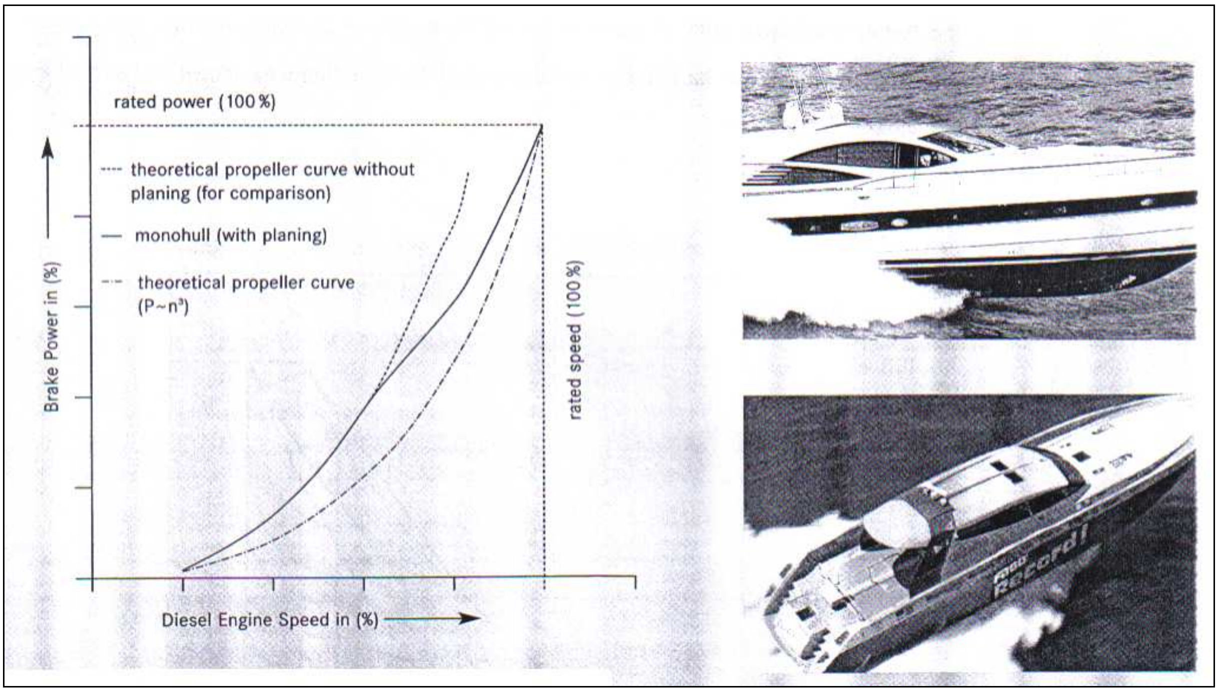


Fig. 2.13 – Typical propeller curve of a monohull (displacement vessel with planing).

There are some hullforms like catamarans or trimarans that will never have a planing phase due to their high draught and small planing area. Their hump is the result of the interaction of waves between the hulls.

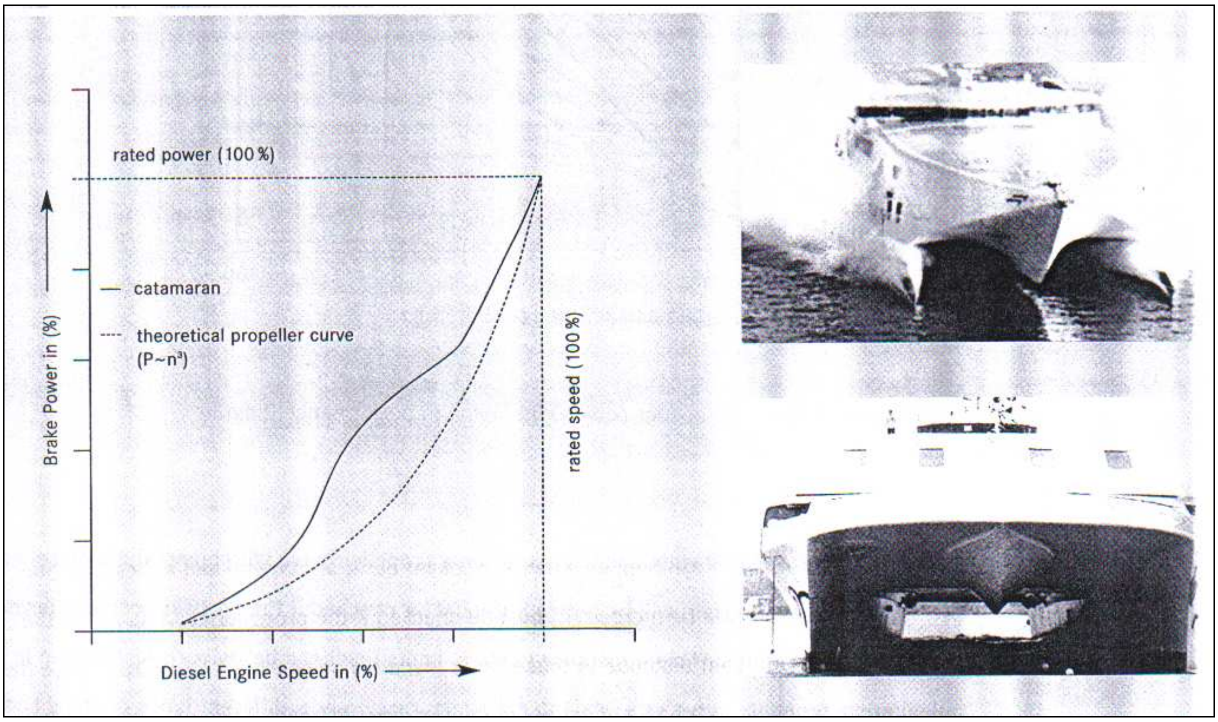


Fig. 2.14 – Typical propeller curve of a catamaran (multihull vessel).

If only the design point of a ship is known, the quality of an approximation with a theoretical propeller curve can be good or bad dependent on the influence shown and should always looked with proper care.

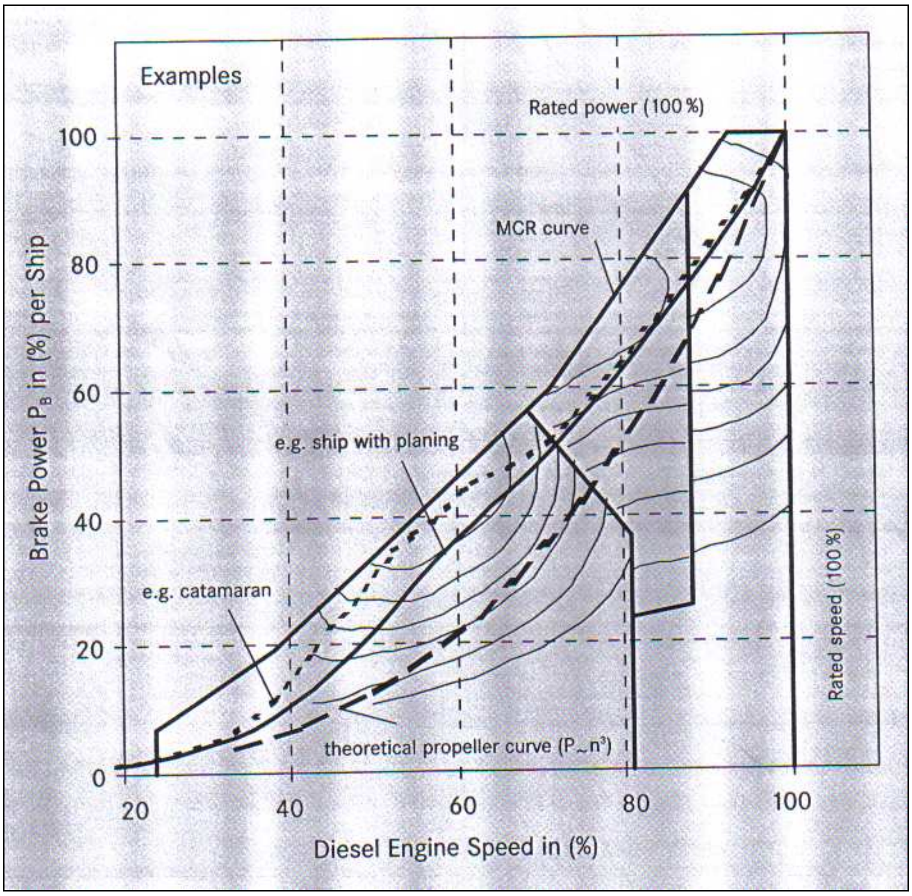


Fig. 2.15 – Propeller curves with hump in performance diagram.

Figure 2.15 shows what can happen when a propeller curve with hump and a diesel engine performance diagram will be merged. It must be checked if the propeller curve is inside the MCR limits with sufficient distance to these limits (due to dynamic behavior). Also shown is the theoretical propeller curve as a basis to get an impression how different types of ships and their operational states can deviate. In such cases, the diesel engine manufacturer should always be consulted. In the worst case scenario when a short overload is not practical a two stage gearbox must be used.

2.4.4 Propeller and performance diagram

a) Basics

The required effective power with appendages or effective power for short (P_E) of the ship does not depend from the ship speed (V) only, but also environmental conditions (wind, seastate), hull roughness (clean, fouling) and actual load condition have to be taken into consideration (see Figure 2.16).

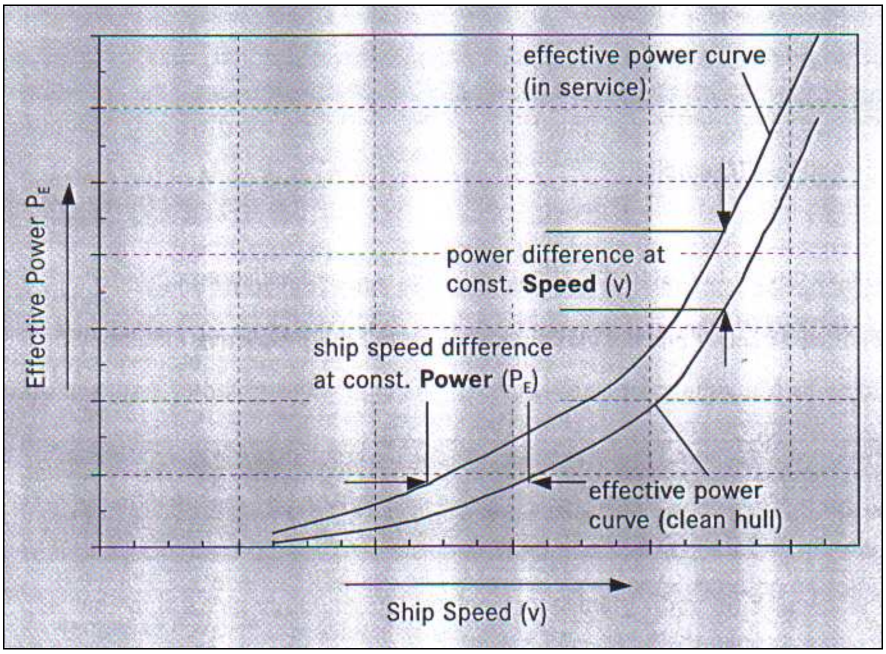


Fig. 2.16 – Influence of change in resistance on effective power curve.

On the basis of a defined effective power curve (see Figures 2.17, part 1) a propeller will be designed. The relation between effective power (P_E) and delivered power (P_D) for different ship speeds (V) can be seen on diagrams using these ordinates (yy-axes). Figures 2.17, part 2 and 3 show the relation between delivered power (P_D) and brake power (P_B) for different ship speeds (V) or propeller speeds (n), respectively. The diagram with the propeller speed (n) as abscissa (xx-axis) has the advantage that the performance diagram of the diesel engine can be plotted in also.

Every change in the effective power curve will be seen in the propeller curve also. The example in Figure 2.16 shows that due to the cubic characteristic of the propeller curve small changes can have great effects.

Although curves in Figures 2.16 and 2.18 are similar in shape they are quantitatively different. The effective and delivered power will be related by the quasi-propulsive efficiency (QPC). This means that the propeller curve is only valid for the designed propeller. Changing the geometry of the propeller (e.g. diameter, blade area ratio, pitch or the number of blades) leads to a new power-speed relation, i.e. a new propeller curve. If the effective power curve changes, e.g. from clean hull to and fair weather to fouled and rough weather the propeller curve will also change. That leads to the conclusion: a change in the propeller curve can be initiated either by the ship (effective power) or by a modification of the propeller geometry.

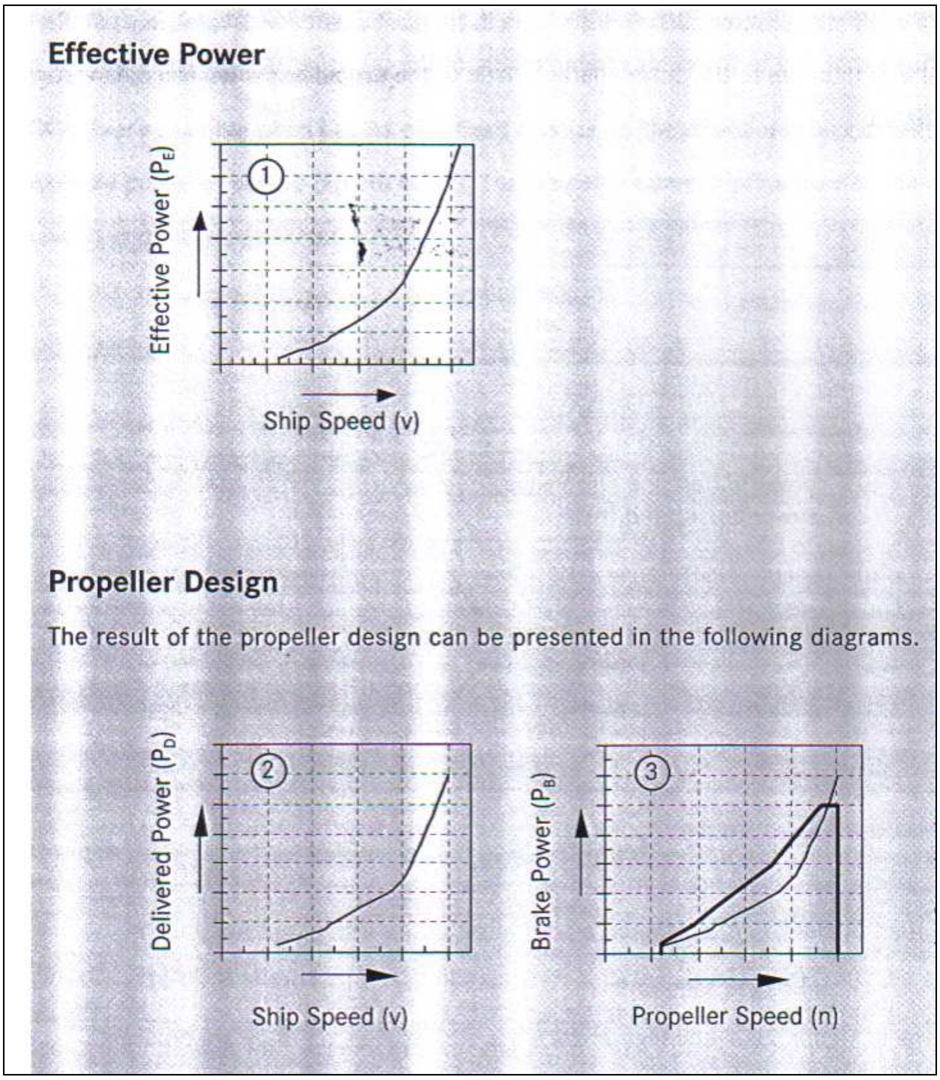


Fig. 2.17 – From effective to brake power curve.

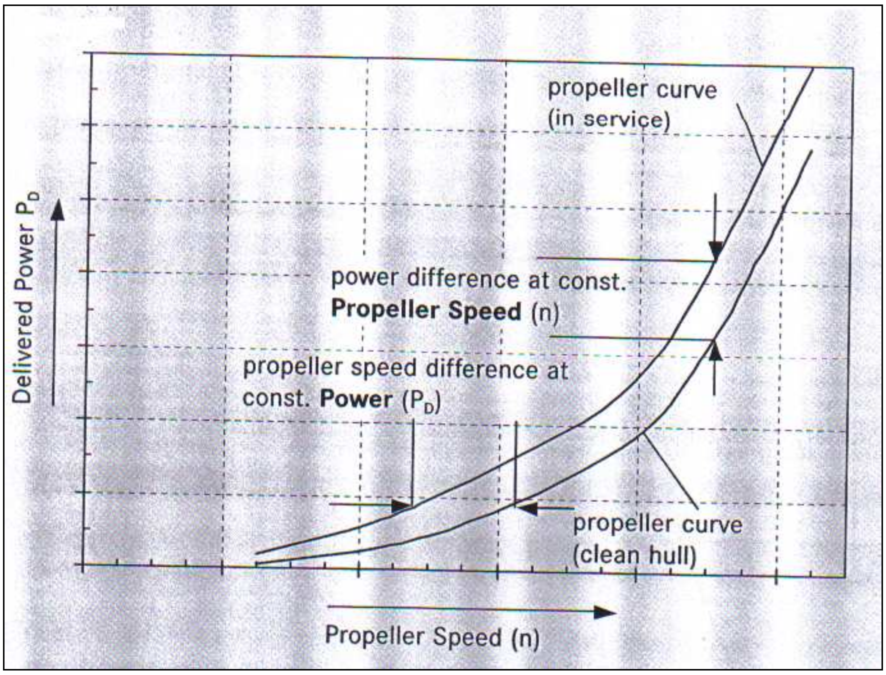


Fig. 2.18 – Effect of change in resistance on delivered power curve.

FPP: The propeller curve has a fixed relation to the effective power curve and will be influenced by the ship (effective power) only.

CPP: Every possible pitch has its own fixed relation to the effective power curve. This leads to multiple propeller curves (see Figure 2.19).

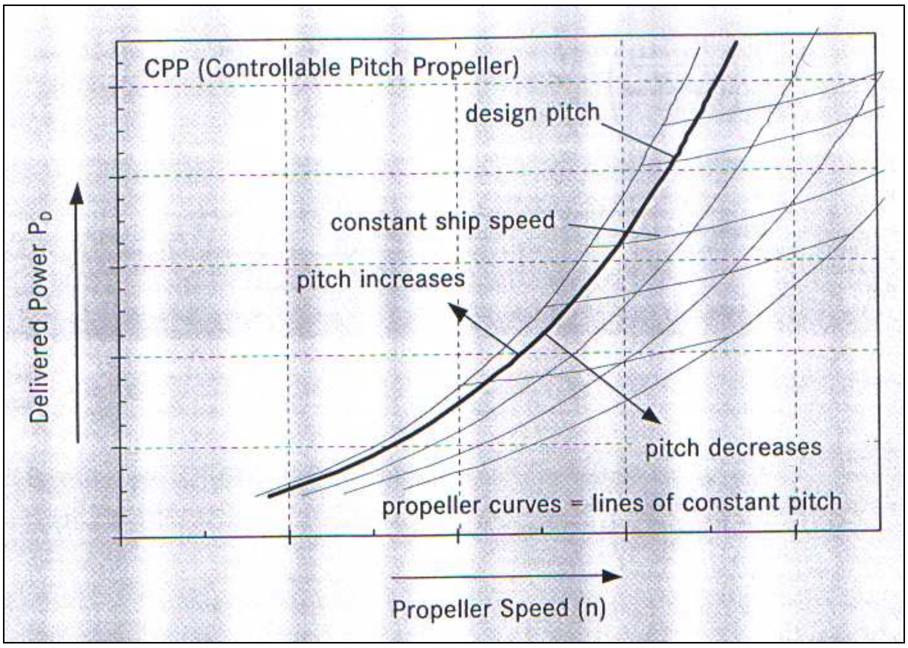


Fig. 2.19 – Effect of different pitches on delivered power curve.

This different behavior will have distinct consequences on the design of the chosen propeller type.

b) Theoretical propeller curve

Diameter (D), delivered power (P_D), and shaft speed (n) of the propeller can be calculated by the propeller manufacturer, when the effective power curve is given and the design speed (V) and the installed brake power (P_B) have been chosen. Power and propeller speed (n) have to match the installed power of the diesel engine.

If only the design point of the propeller or the diesel engine is known, a simple approximation can be done by a theoretical propeller curve:

$$P_D = \left(\frac{P_{D\_rated}}{n_{prop\_rated}^{\ 3}} \right) n_{prop}^{\ 3} \tag{2.12}$$

Notice that volume flow at the propeller disc is given by $Q_V = V_A.A_P = V_A.\pi \dfrac{D^2}{4}$ and $V_A. = \pi.n.D$, which leads to $Q_V \propto n.D^3$. By Bernoulli equation, the pressure rise across the disc is given by: $\Delta p = \dfrac{1}{2}.p.V_A^{\ 2}$, and power delivered is: $P_D = \Delta p.Q_V$, from which: $P_D \propto n^3.D^5$ or $P_D \propto V_A^{\ 3}.D^2$.

Diesel engine and propeller have a fixed relation via the propeller shaft, and, therefore, eq. (2.12) can be used either for delivered power (P_D) or installed brake power (P_B), as follows:

$$P_B = \left(\frac{P_{B\_rated}}{n_{prop\_rated}^{\ 3}} \right) n_{prop}^{\ 3} \tag{2.13}$$

There will be differences to the real curve, depending on the hull form (see section 2.4.3) as the decisive factor, and taking into account that the propeller geometry is fixed. That means the approximation of a CPP is only valid for the design pitch.

In practice, there is another restriction for the lower speed range. Below a certain speed (V), the wind forces can become dominant and the delivered power does not decrease furthermore.

c) Driving mode

Delivered power (P_D) and shaft speed (n) of the propeller have to match the installed brake power (P_B). Only the sea trials show whether estimates are correct or not. At design stage of evaluation, a diesel engine has been selected, and a design point inside the performance diagram of the diesel engine has to be chosen. In addition the hull hydrodynamic characteristics discussed before (see propeller curves on Figures 2.12 - 2.14), manufacturing tolerances have to be taken into account. Therefore:

- Manufacturing tolerance in pitch, surface and profile will influence the power absorption of the propeller;

- Hull resistance can vary due to inevitable differences in load and hull shape of the vessel.

Hydrodynamic and geometrical aspects (see Figure 2.20) can shift the propeller curve A to the left side of the performance diagram as shown in the propeller curve C. Certain models of diesel engines are more sensitive to this shifting than others. As a consequence, the ship may not be able to operate at full speed when the hull is fouled, the weather deteriorates or the draught has increased.

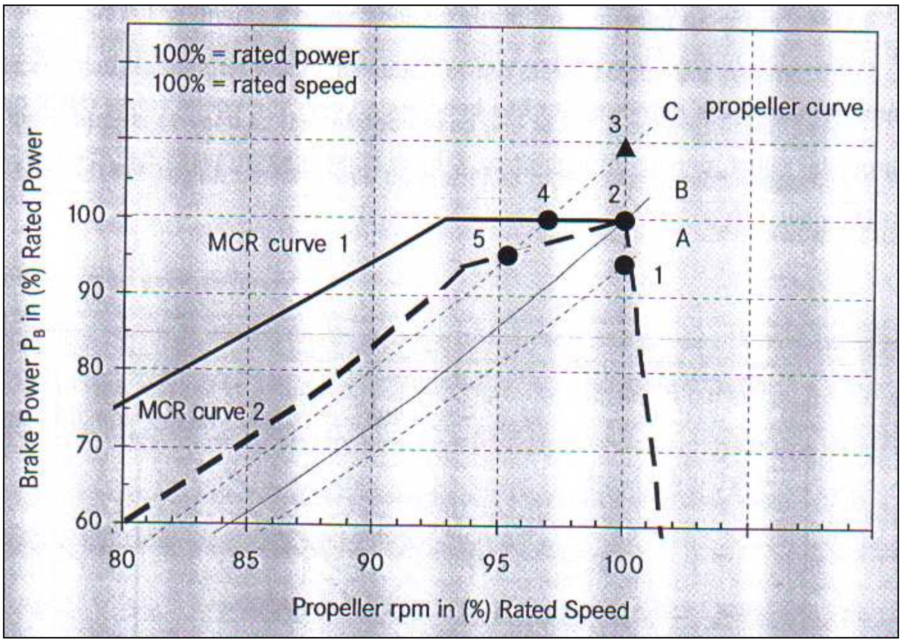


Fig. 2.20 – Change in power delivered due to weather, draught and fouling.

In Figure 2.20 two different diesel engines (MCR curves 1 and 2) with different performance limits are shown. A change in the propeller curve from A over B to C leads to the following behavior:

- The diesel engines can run with full speed (n). No limitation arises (point 1). But the propeller does not absorb the maximum available power;

- The diesel engines can run with full speed (n) and reach their available power. No limitation arises (point 2);

- Due to the load limits (e.g. fuel stop power) both diesel engines are not able to provide the required power for full speed (n) at point 3. In this case the diesel engine controller reduces the speed (n) in order to find a operating point at fuel stop power within the performance limits. For the diesel engine with MCR curve 1 this is point 4, and for the other diesel engine, point 5 is obtained. The differences between the two operating points 4 and 5 are the magnitude of reduction in ship speed (V), which may be considerably high.

d) Bollard pull

For a trawler or a tug the design target is the required bollard pull (or push), which means that for the selection of the diesel engine the required bollard pull – not the effective power and the related ship's speed (V) – is the most important issue. The design point in the diesel engine performance diagram has to be chosen with respect to this condition.

For the bollard pull generally exists three different definitions as follows:

- Maximum bollard pull: the maximum average of the recorded tension in the towing wire over a period of one minute at a suitable trial location. The maximum bollard pull generally corresponds to the maximum diesel engine output;

- Steady bollard pull: the continuously maintained tension in the towing wire, which is achievable over a period of 5 minutes at a suitable trial location;

- Effective bollard pull: the bollard pull that a vessel can achieve in an open seaway but which is not ascertainable in a normal trial location. Generally, it is characterized as a percentage of the steady bollard pull (e.g. 75-80%).

Notice that bollard pull trials should be conducted at allocation providing a sufficient extent of deep and unobstructed water. The recirculation causes thrust reduction/losses since advance speed of the incoming water is not zero.

After a propeller has been chosen (diameter, pitch diameter ratio, blade area ratio, etc.) the bollard pull, also a cubic power curve can be estimated on the basis of the propeller parameters (see Figure 2.21). This curve can also be seen as a result of the limiting effective power curve, which cannot be shifted to the left because the related ship speed is already zero.

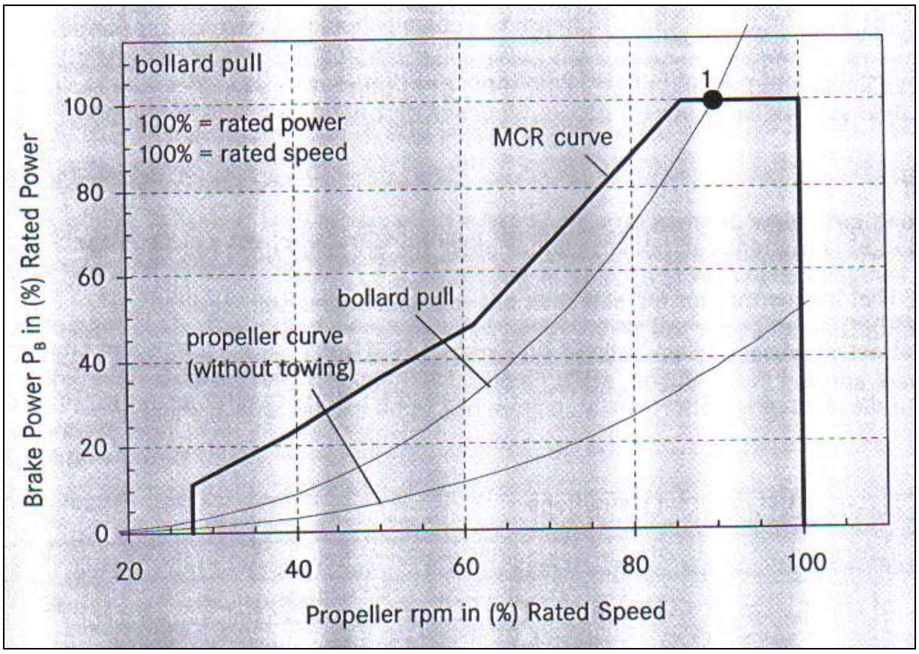


Fig. 2.21 – Bollard pull.

For bollard pull, a design point at rated power (100% P_B) and a diesel engine speed (n) in the middle sector of the lug-down range, e.g. point 1 in Figure 2.21, should be chosen. A reserve/margin in diesel engine speed (n) to the left is necessary to overcome possible design tolerances. The speed reserve/margin to the right allows a higher vessel speed.

Without towing, the vessel runs on its plain effective power curve. The maximum achievable speed (V) depends on the maximum propeller speed (n) using only a part of the available diesel engine power. With a two stage gearbox a higher ship speed (V) would be available. Figure 2.22 shows the relation between diesel engine speed (n) and ship's speed (V).

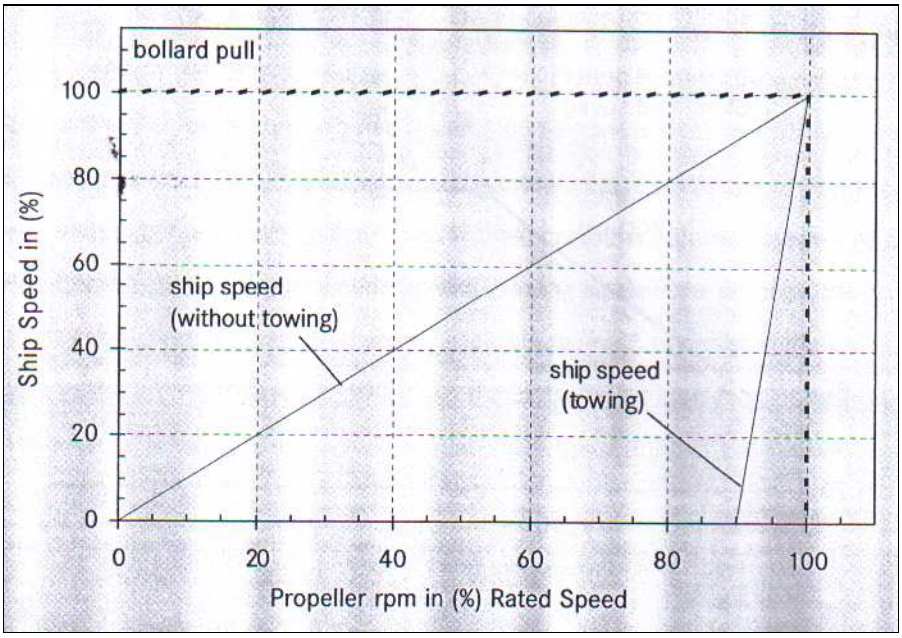


Fig. 2.22 – Bollard pull effect on ship speed.

e) Auxiliary loads

In the case of main propulsion engines, the auxiliary equipment (fuel pumps, lube-oil pumps, cooling water pumps) is driven directly by the engine itself, so that the engine is essentially an independent unit. However, other external auxiliary devices related to ship services (such as electrical generators or hydraulic-system pumps) may also be driven by the propulsion engine. The power required for these units obviously subtracts from that available for propulsion and thus should be a factor in the engine-propeller matching process. If the power is quite small compared to propulsion power the designer might safely neglect it, but if the auxiliary power or Power Take-Off (PTO) is 20% or more of rated propulsion power, we should analyze its effect on the best match as illustrated in Figure 2.23.

Figure 2.23, top sketch, gives a head-flow characteristic for the piping, and head-flow characteristic for the hydraulic pump with rpm as a parameter. Each intersection of a pump curve with the piping curve is an equilibrium point for that particular rpm. Such a pump power-rpm curve (based on an assumed 50% pump efficiency) is plotted in the lower sketch. This curve is then added to the propeller curve to represent the total load as imposed on the engine. Equilibrium occurs where this curve crosses the engine curve; total power, power to the propeller, and power to the pump, can be read from the appropriate intersection. The power taken to drive the pump is, of course, subtracted from the propulsive power. In addition, there is a loss in power because of the slowing down of the engine of about 13% of the original 400 [HP]. The course of action would be the installation of a lower pitched propeller to allow the diesel engine to turn at full rpm when driving the pump, or installation of a two stage gearbox or CPP so that the same rpm could be maintained with and without the pump (PTO). Finally, the loss in rpm may be more significant to the auxiliary load than to the propulsive load. Note that the flow through the piping system is nearly proportional to the pump speed. Thus the decline in engine – and hence pump – rpm from 500 to 460 represents about 8% loss in flow. If the auxiliary load is important, it may dictate an adjustment in propelling matching, or in the speed ratio between engine and pump, to compensate for this loss.

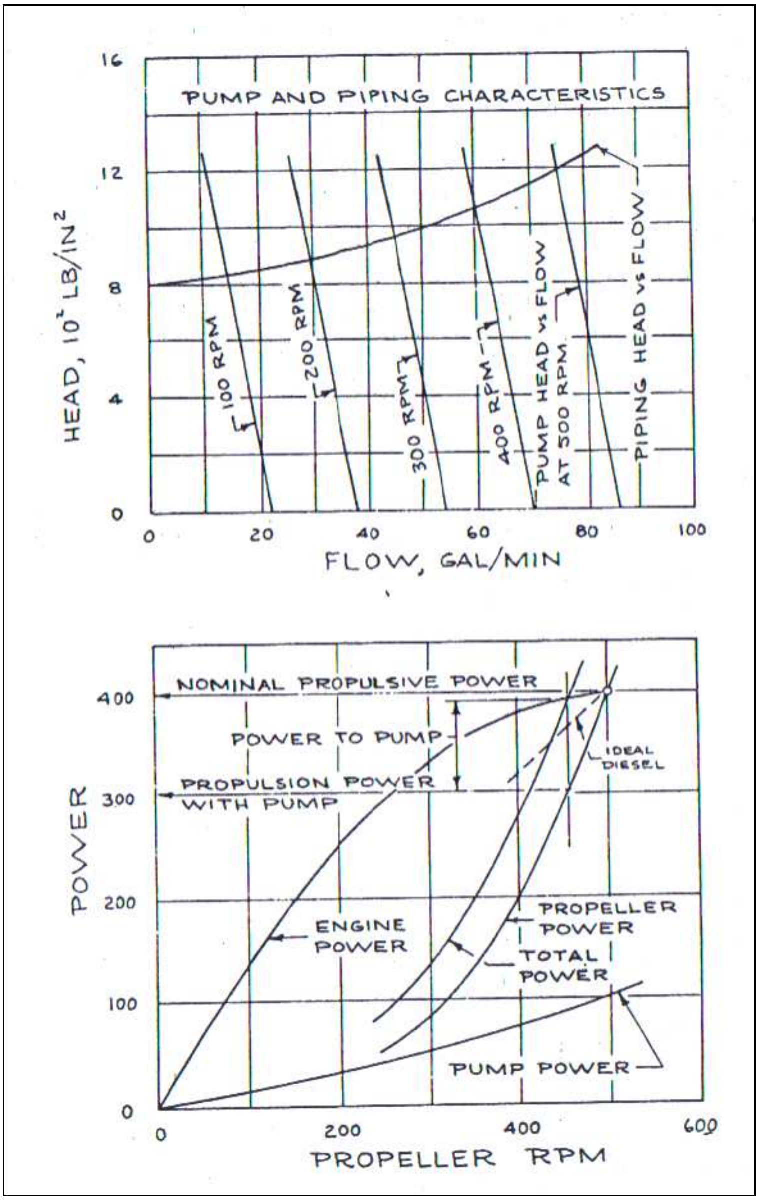


Fig. 2.23 – Hydraulic auxiliary load and diesel engine PTO matching with propeller curve.

f) Fixed pitch propeller and performance diagram

The design of propulsion system with a fixed pitch propeller is absolutely critical to the performance of the ship. As illustrated in Figure 2.24, the brake power (P_B) curve B should pass through rated power at rated speed (point 2) of the diesel engine. But due to geometrical tolerances and deteriorated hydrodynamics, the propeller curve C can be higher than predicted. This situation will be overcome by designing the propeller curve A a few revolutions faster for the new ship.

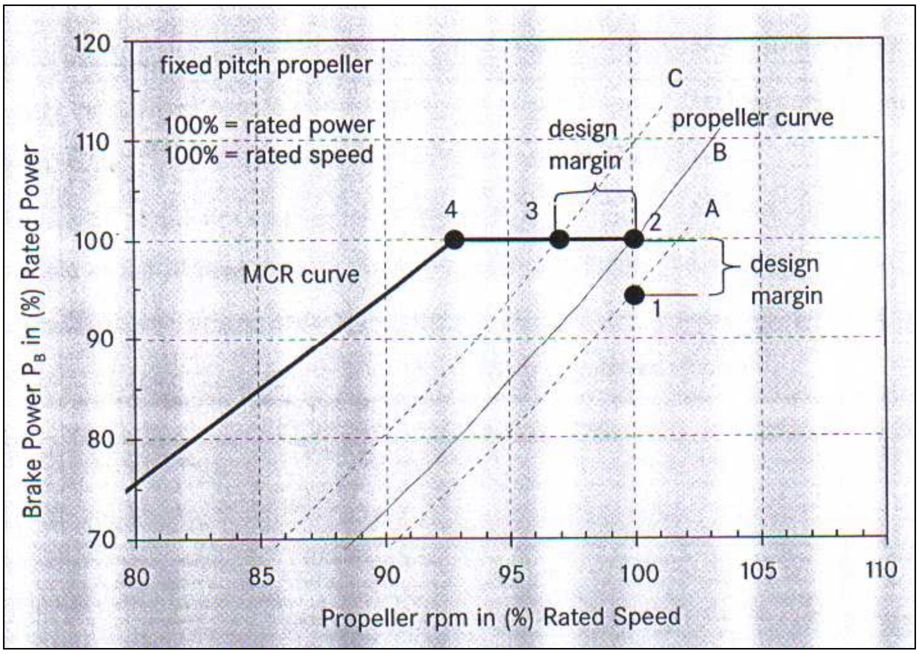


Fig. 2.24 – Choosing a design point for a fixed pitch propeller curve.

Dependent on the type of diesel engine two different approaches are possible:

f.1) Diesel engine with wide lug-down and flat rated power

The characteristic of this diesel engine is the wide lug-down range above a certain speed (n) (fuel stop power). This range can be used as a design margin. In rough weather conditions or at increased hull resistance, the propeller curve will move to the left. This means, at sea trials with design load condition and rated power, the diesel engine should work at the right-most point of the lug down range (point 2, effective power curve for trial condition = propeller curve B), i.e. the design point for the propeller. With growing lifetime the propeller curve will move to the left (e.g. point 3, propeller curve C). The design allows the propeller to run at the rated power (100% P_B) as long as the propeller curve does not pass point 4 (lugging point) but continuous operation at the lugging point should be avoided. The left-most operating point should be 1-2% below the lugging point speed (n). The maximum ship speed (V) will decrease slowly with the left shifting of the propeller curve towards point 3.

f.2) Diesel engine with short lug-down and inclined rated power

In the design point the propeller runs at rated speed (100% n) and a small amount (design margin) below rated power (100% P_B). In this case the diesel engine is effectively working at a de-rated condition (point 1, effective power curve for trial condition = propeller curve A). In rough weather conditions or with growing lifetime the propeller curve will move to the left and the rated power will become available (point 2, propeller curve B). The design allows the propeller to run at 100% rpm (rated speed) as long as the propeller curve does not pass point 2. The ship speed (V) will increase with the shifting of the propeller curve and reaches its maximum at point 2. Using this procedure the designer has to consider that it may not be possible to demonstrate the full speed (V) capability of the ship at the trial conditions, because the speed (n) of the diesel engine is limited to the rated speed (100% n). The difference between rated power and design power is called the "sea margin" (=design margin). If there are no specific demands, a design margin of approximately 6-10% shall be considered. The rated power will be met by propeller curve A at 102-103.5% speed (n) but this is only theoretical value.

A few additional aspects shall not be forgotten by the designer:

- If at the design point the propeller curve does not pass through the region of minimum fuel consumption, the curve cannot be changed afterwards;

If at the design point the propeller curve comes too close to MCR curve, the curve cannot be moved away from this region afterwards with the result of a blocked operation range.

2.4.5 Controllable pitch propeller and performance diagram

The controllable pitch propeller can be seen as an extension to the fixed pitch propeller. Each pitch results in a new propeller curve. A typical example is shown in Figure 2.25 where the controllable pitch propeller characteristic is superimposed on a diesel engine characteristic.

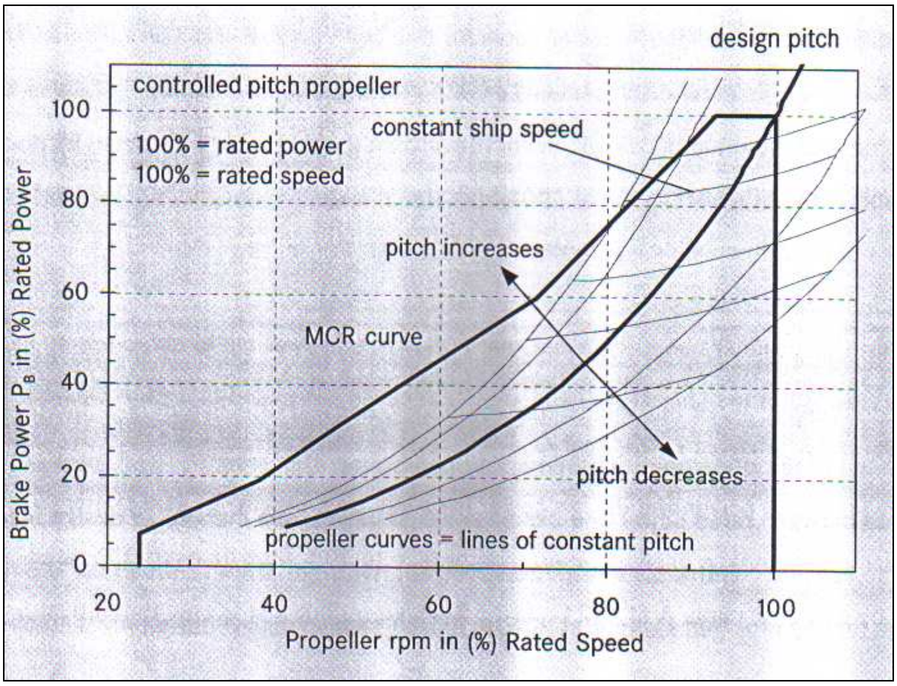


Fig. 2.25 – Controllable pitch propeller curve in a typical diesel engine performance diagram.

Every change in the pitch of the propeller changes the relation between propeller speed (n) and brake power (P_B) for the ship. Due to possible later adjustment of the propeller pitch there are no restrictions for the design point within the diesel engines performance diagram. The point at 100% brake power (P_B) and speed (n) should be chosen (see Figure 2.26).

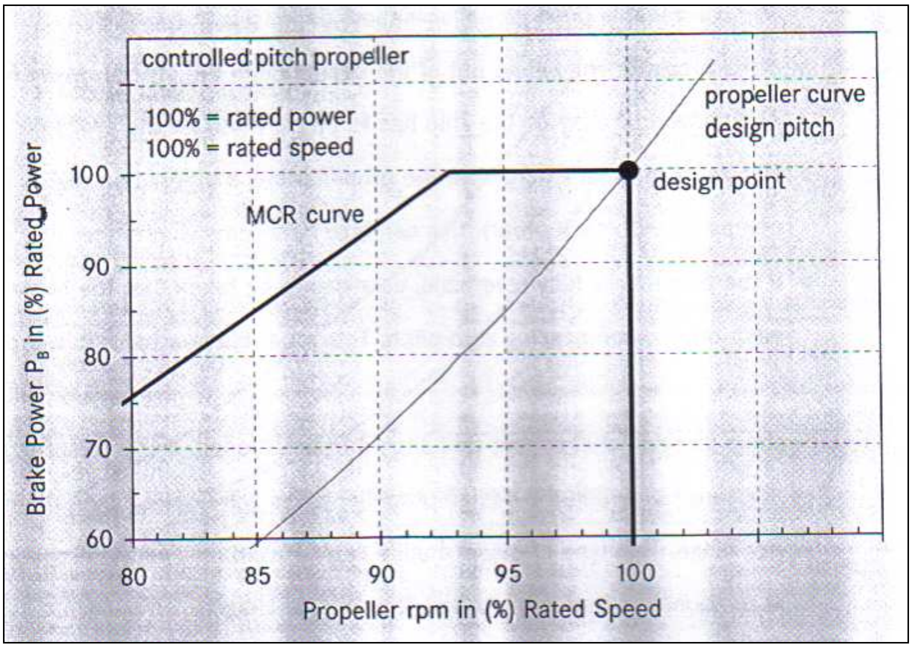


Fig. 2.26 – Choosing a design point for a controllable pitch propeller curve.

The available pitch range is not fixed. It is a part of the owner specification for the propeller. On the manufacture's side it is limited by the size of the hub and the maximum blade forces. Generally, the available pitch range will be related to the design pitch and be given in degrees. The range above the design pitch is very small because there is no general need, except in special applications.

a) Advantages of a CPP:

a.1) If the delivered power curve through the design point (design pitch) does not pass through the minimum fuel consumption region, it is possible to adjust the pitch at partial load conditions;

a.2) If the power curve comes too close to the diesel engine MCR limit, the curve can be moved away from this region;

a.3) If the ship during trial is not able to achieve the design brake power (P_B), the design pitch can be corrected or when the ship resistance increase with service life, the design brake power (P_B) and speed (n) will stay available;

a.4) A CPP can be chosen with a fully reversible position and the ship can move astern without the need of a reversing gearbox. The stopping distance will be significantly lower than with FPP. Generally, the maneuvering characteristics are also better;

a.5) A CPP can be chosen with feathering position (minimum resistance), if a single shaft mode is part of the operational requirements;

a.6) Over a certain power range, the diesel engine can be operated at constant speed. In this mode it can additionally drive a power generator or a fire fighting pump.

b) Disadvantages of a CPP:

b.1) The CPP is more expensive than a FPP;

b.2) If the propeller will be set out of the design pitch the efficiency decreases;

b.3) Additional space inside the ship has to be provided for the propeller control unit;

b.4) Due to its internal mechanism, the propeller has a larger hub than a FPP (approx. 50%, compared with each other), this can lead to a somewhat higher diameter;

b.5) If the propeller is fully reversible, care has to be taken that the blades do not interfere with each other when passing zero pitch. The upper blade area ratio will be limited.

There is an additional aspect that should be mentioned. If the diesel engine has a very slender performance diagram, the design propeller curve will not lie inside the diagram for the lower power range. This type of diesel engines can be used only with a propeller controlled by a pitch-rpm relationship, frequently called "combinator diagram".

2.4.7 Waterjets and performance diagram

The main application for a waterjet is in the higher speed range (above 25 knots). The propulsive efficiency of a waterjet decreases considerably with speed (V) reduction. Below 24 knots a propeller should be preferred.

A waterjet is like a hydrodynamic propulsive device but is arranged inside the ship and behaves more like a pump than as a propeller.

The main differences between a waterjet and a propeller are:

a) The propeller is very sensitive to the velocity and direction of the incoming flow. It senses in its hydrodynamic situation (seastate, wind, draught, etc.), so does the diesel engine;

b) The waterjet works more like a pump as long as there is water in the intake duct and turns the brake power (P_B) into thrust. As illustrated in Figures 2.27.a and 2.27.b, there is only a minor feed back from ship speed (V) into the effective inlet velocity at the waterjet duct.

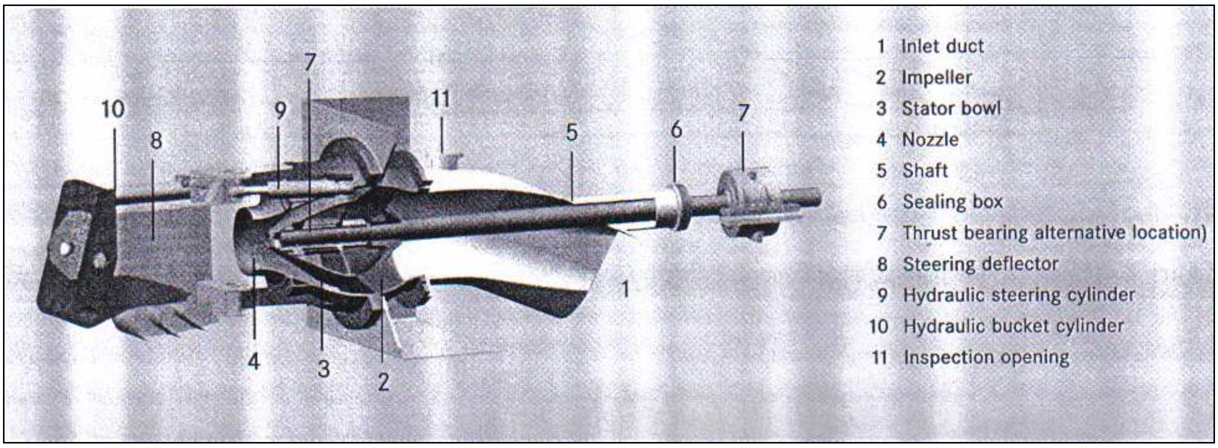


Fig. 2.27.a – Cutaway of a waterjet unit.

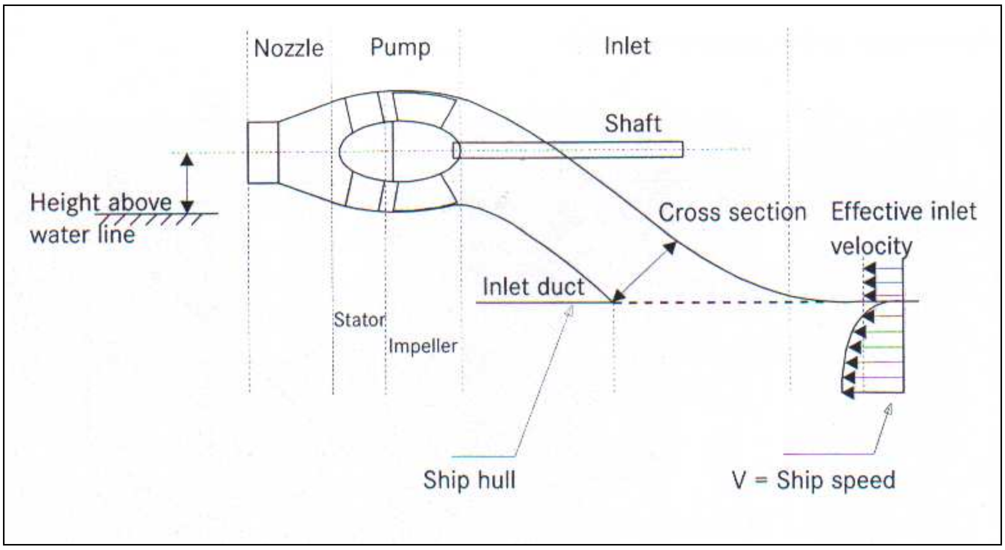


Fig. 2.27.b – Effective waterjet inlet velocity.

For this reason the diesel engine has minor load cycles when is connected to a waterjet.

Due to the insensibility to the ship resistance (effective power curve) there are no restrictions for a design point within the diesel engine performance diagram. But the waterjet, like the propeller, is a mechanical device and manufacturing tolerances have also to be taken into account. This relation can lead to the fact that at 100% shaft speed (n), the waterjet may not absorb the diesel engine brake power (P_B). Therefore, a design point at rated power and approximately 1-2% below 100% diesel engine shaft speed (n) (design margin) should be chosen (corresponding to design point 1 on Figure 2.28). If the effective power curve shifts to the left, the ship speed (V) will decrease but no change will be seen in Figure 2.28 because the waterjet is still running with its demanded speed (n) and brake power (P_B). The behavior of the ship cannot be observed in Figure 2.28. That is the reason why the diesel performance diagram has a limited use for choosing a waterjet design point. It will only give an impression about the relations among the propeller curve, the lines of constant fuel consumption, the design margin, and the margin to the diesel engine MCR limit curve. These relations will remain independent from ship load as before.

With these conditions in mind, design point 2 (Figure 2.28) can be chosen also. The leftmost design shaft speed (n) should be 1.5% above the speed of the lugging point. The advantage is a less fuel consumption but the margin to the MCR curve (acceleration reserve) decreases.

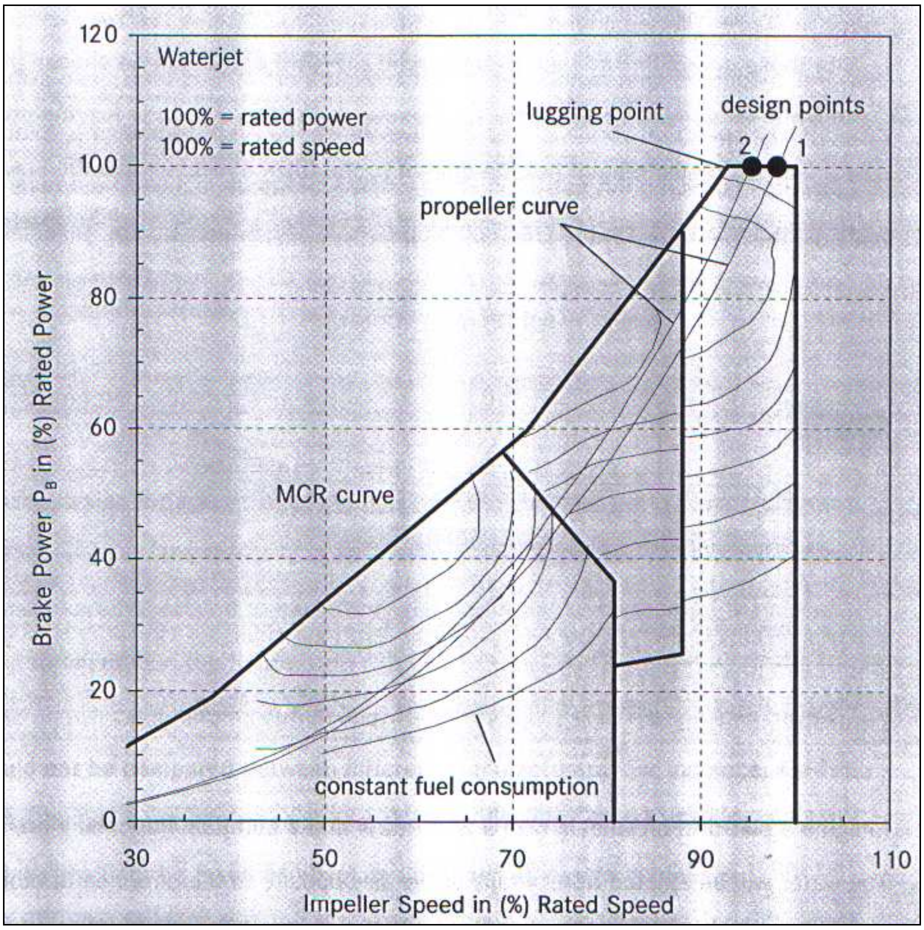


Fig. 2.28 – Waterjet design point.

Because this aspect is critical, an illustrative example shall be given: imagine a platform on wheels with a water tank and a pump installed on its loading area (see Figure 2.29). The water will be ejected horizontally in the air opposite to the direction of motion. The platform will start to move on the ground and no matter how fast the platform will move, the pump will always eject the same amount of water using the same power. This is also true if an obstacle stops the platform. The pump will not be affected by the behavior of the platform. In other words, the generated thrust depends only on the amount of ejected water. Although, this is simplified, it shows the fundamental difference between a propeller and a waterjet. Moving a step ahead: even if there are two pumps on the loading area, they will not interfere with each other, independently whether they are not of equal six or running at different power (i.e. pumping different amounts of water).

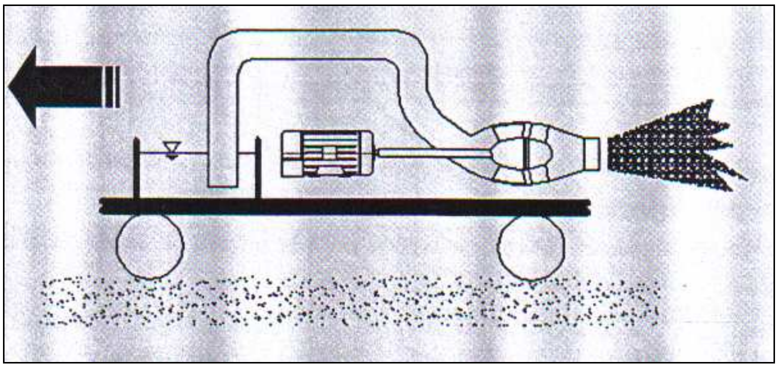


Fig. 2.29 – Sketch of a platform with pump.

For the reasons, a different type of diagram has to be used to do the matching between the waterjet and the diesel engine, as illustrated on Figure 2.30.

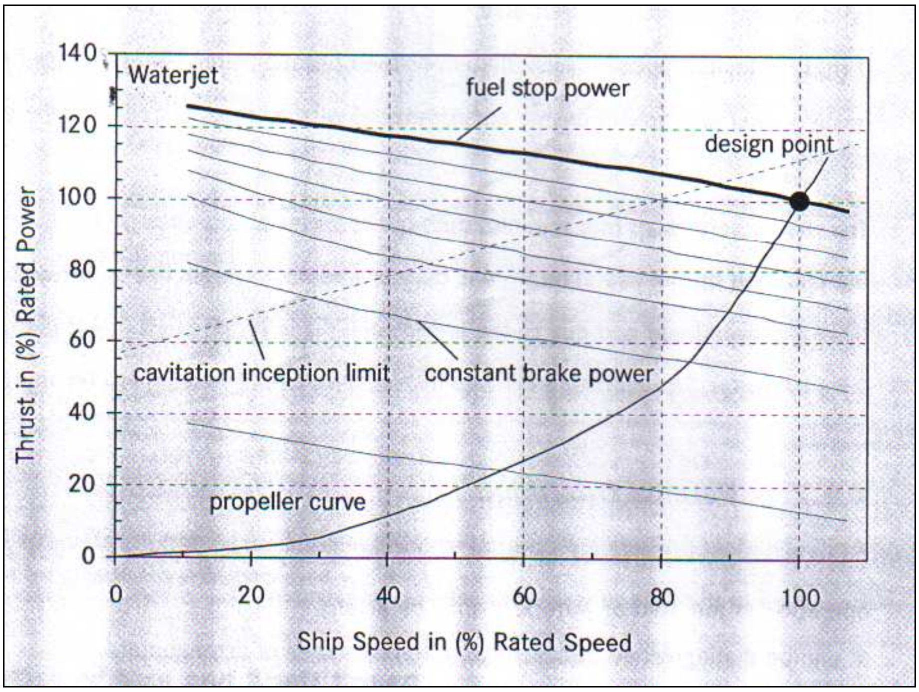


Fig. 2.30 – Waterjet performance diagram.

The figure shows the design propeller curve together with the waterjet performance diagram and instead of effective power the thrust is used. Because the ship's speed (V) and the engine speed (n) of the diesel are not related to each other the performance diagram of the diesel engine cannot be represented in the figure.

The cavitation lines are specific to the chosen waterjet and are defined for different stages of cavitation. Generally, these lines shall not be taken as absolutes limits but rather as design guidelines. If the propeller curve shifts to the left the ship speed (V) will decrease and the distance to the cavitation inception limit will be reduced. The reason for this behavior is that stagnation pressure in the inlet duct goes down and the waterjet starts to suck the water through the duct. The thrust of a waterjet can be obtained from the product of mass flow rate by the speed of the ejected water. That means that a certain thrust can be generated by a larger or smaller waterjet. In the smaller waterjet, the speed of water is higher, i.e. the distance between the design point and the cavitation inception point. If there was a limited space onboard for installation, designer will probably choose a smaller waterjet with lesser distance to the cavitation inception area.

The risk of getting air into the duct inlet of the waterjet depends on specific arrangements in the ship and on the seastate. In this case the control system has to protect the diesel engine from any overspeed and due to the low inertial mass of the shaft line it is more demanding than for a propeller.