The Role of Cavitation in the Design of Controllable Pitch Propellers

By: T. v. Beek
Technical Manager
Lips B.V.

Drunen, 1995-05-08

Paper prepared for PROPCAV 95

CONTENTS:

1. Introduction
2. The design process
3. Characteristic design features of controllable pitch propellers
4. Trends in vibrations and noise
5. Conclusions
1. Introduction

In recent years the number of controllable pitch propellers compared to the number of fixed pitch propellers has shown a steady growth, which can be derived from official LRS statistics (ref. 1). This regular growth is a reflection of the reputation of the controllable pitch propeller for its increased reliability and reduced costs.

Apart from this nowadays there exist only a few four stroke engines, which are reversible. Thereby for ships with a four stroke engine the controllable pitch propeller appears to be a natural choice.

Several reasons for the selection of controllable pitch propellers are:

- good manoeuvring properties
- constant number of revolutions in view of the use of a shaft generator
- the availability of full engine power for a wide range of ship speeds without overloading the engine (towing conditions but also in heavy weather)
- the possibility to use multiple engines per shaft.

The application of both fixed and controllable pitch propellers can be found in a high ship speed and high power densities (fig. 1.1, ref. 2).

The additional degree of freedom obtained by changing the propeller pitch complicates the design process. Basically the wide range of inflow conditions makes it more complicate to control cavitation.

Most shipdesigners have a history in the application of fixed propellers, the appearance of more controllable pitch propellers requires an extension of this knowledge. This paper is not written to contribute to the theoretical development of the control of cavitation for controllable pitch propellers but concentrates on design procedures, design criteria and boundary conditions.
2. The design process

For propellers in general the highest possible level of propeller efficiency must be achieved while vibration and noise thus cavitation are kept at the lowest possible level.

This leads to conflicting boundary conditions, as less cavitation leads to a large blade area ratio, but with lower efficiency and vice versa.

Therefore any propeller design should be a subtle balance between several extremes resulting in a compromise according to the experience of the propeller designer and the correct tools at his disposition.

In order to overcome the boundary limitations as much as possible Lips has developed blade sections that combine large cavitation-free operation with good structural characteristics and low drag properties. The end result is an optimised design with higher efficiency.

In addition the propeller design system as used and developed by Lips, consists of a series of interactive design and analysis modules as shown in fig. 2.1. Any design can only be initiated after the design criteria have been selected. These design criteria consist of information with respect to shiptype, mission profile and possible limitations regarding propeller diameter, efficiency, ship speed or any other manoeuvring requirement. Each of these criteria relevant to the design of controllable pitch propellers are discussed below.

The shiptype generally is an important factor for the wake distribution and consequential variation in inflow. But it also determines the normal clearance at the propeller. The required maximum pressure pulses are related to the strength of the ship's structure.

The mission profile is vital information for the design of a controllable pitch propeller. The percentage of time at which speed the vessel operates indicates the importance of off-design conditions. For instance in-the case a shaft generator is present it is important in which conditions a constant number of revolutions is used. In the case of tugs the trade-off between relevant towing conditions and the transit speed should be carefully considered.

Even more complication exists in the case multiple engines are connected to a single propeller. In that case the shipowner can operate the ship at lower ships speeds on one or two engines at very different pitch settings.

Manoeuvring requirements for twin screw ferries normally lead to high requirements for the bollard ahead and bollard astern thrust values for defined manoeuvring condition. For instance during a sidesways crabbing manoeuvre one propeller generates forward thrust, which is compensated by the astern thrust of the other propeller.

For ferries operating at the Nordsea the power used for the crabbing manoeuvres is about 60% of the maximum power.
These requirements for the manoeuvring do lead to limitations in the propeller characteristics, which will be explained in the following.

For all the relevant sailing conditions vibration and noise limits are present. In most cases the maximum power condition will limit the maximum induced pressure amplitude on the hull structure as most newbuilding contacts include vibration limits in those conditions. But also other relevant sailing condition should be regarded as to the requirements of vibrations especially in view of comfort. In the case of controllable pitch propellers this can also imply that a lower limit in power exists in order to avoid pressure side cavitation at reduced pitch settings.

Above requirements are considered general design information, which are used in the design of controllable pitch propellers.

The first step in the design process of the CP propeller as given in fig. 2.1 is the design of a propeller (generation of detailed propeller geometry).

The propeller design module runs in an interactive sequence a number of calculations, whereby a low blade area (efficiency!) will be maintained as long as possible. The propeller design consists of three segments related to power absorption, strength and cavitation properties in design condition.

In the power absorption segment the mean pitch of the propeller is determined to accommodate the design condition and takes into account the specified radial pitch distribution.

The strength calculation segment is based on beam theory. The radial thickness distribution is determined taking into account the fatigue load due to varying stress levels during a propeller revolution and the fatigue properties of the propeller material to be used. The resulting thickness distribution will meet the classification requirements.

The cavitation segment determines at each radial station a combination of minimum chordlength and camber. The requirements as stipulated by the designer as to suction side and pressure side cavitation together with a built-in criterium against harmful bubble cavitation are met.

The design of the blade section is done by a variation of camber to chord ratios. The cavitation inception lines of the blade sections, also called cavitation buckets, are shifted drastically dependent on the maximum camber to chord ratio.

The lift of the section and thus the thrust of the propeller is dependent on the power, the number of revolutions, the entrance speed, the wake and the pitch distribution. The optimum blade camber to chord ratio in many cases deviates from those laid down in series.
A design module is always limited to one or two conditions. Therefore at this point the propeller designer has to consider if the designed propeller meets all requirements. Acceptance of the design depends on the calculated results and on the quantity and reliability of the available information.

With the available blade geometry the cavitation properties of the propeller working in its wakefield are determined by the cavitation analysis module for any given combination of shipspeed, number of propeller revolutions, absorbed thrust to be delivered and propeller submergence, this way also off-design condition can be considered. At fixed angular blade positions the loading on each blade is calculated. In the figure the analysis results regarding suction side and pressure side cavitation is given for all blade sections and all angular positions considered. The experience shows that the calculated method compares well with cavitation test results. Therefore a reliable design can be made without testing.

Fig. 2.2. shows a comparison between the calculated cavitation pattern compared to a full scale observation of a controllable pitch propeller for a container ship.

When the cavitation performance of the propeller fulfills the requirements for the respective operational conditions other design criteria have to be fulfilled. These relate to the pressure amplitudes and strength requirements. Pressure pulses are calculated numerically with a combination of lifting line and lifting surface analysis methods (ref. 3) and compared with a maximum allowed level. Fig. 2.3. shows calculation as compared to 17 other institutes as compared to in the 18th ITTC in 1987. In certain cases this should change design properties and lead for instance to an increased skew distribution or increased blade area ratio.

Obviously in the design of controllable pitch propellers the loading of the hub has to be considered. This concerns the actuating forces, the loading of parts of the hub such as blade bolts, blade flange and other local areas of the controllable pitch propeller. Therefore the design of the blade and the hub are strongly related and can not be separated.

Based on the calculated pressure distribution in the wake a finite element calculation is carried out and evaluated using fatigue properties of cast material.

Optimisation of the propeller geometry in the final stage of the design is done by iterative change of the geometry such as: (consider diameter and blade number fixed)

- chordlength distributions
- pitch distribution
- maximum camber distribution
- skew distribution
- rake distribution

In order to investigate in what way the designer can obtain the best possible compromise a number of items are considered fixed or constant.
For instance the propeller diameter should be selected for high efficiency and clearance. During the more detailed optimisations it is impractical to vary this as well. This also holds for the blade number, which normally is selected for avoidance of resonance in the superstructure. Additionally the type of blade section used (a Naca 0.8 modified meanline and some fixed Naca thickness distribution) is assumed constant during the iteration. As a last simplification it is assumed that the blade thickness used is sufficient to accommodate small changes in blade shapes. What is left is the chord distribution, the pitch distribution, the maximum camber distribution and the skew distribution.

In the regular design process a selection of both pitch distributions and chordlength distributions is varied in order to find the proper cavitation behaviour with given wake distribution and operating conditions. The blade camber is varied until a proper margin against pressure side cavitation is obtained. Here also other operating conditions have to be considered. As to the skew distributions, large skew values are normally applied in those cases were the pressure pulses are playing a decisive role. For instance in the case of modern tankers the pressure pulses are so small due to the large tip clearance that a moderate skew is sufficient. Decisive is the efficiency in behind condition whereas the minimum blade area depends on the risk of erosive cavitation in the ballast condition. For large container vessels the level of the pressure pulses is a fixed limit. As a result the skew angle will be larger. It should be noted however that this should be properly balanced against the pitch distribution. Figure 2.4. shows the calculated cavity volumes for given propeller geometry but with increased skew distribution.

It is known that an increased skew leads to a larger loading at the tip. This is reflected in cavitation behaviour and the predicted pressure pulses. Whereas a large skew angle reduces the pressure pulses as a result of the smoother leading edge shape—this effect is partly compensated by the increased tip loading. Therefore, increased skew should be balanced by proper tip unloading in order to achieve the combined effect of low pressure pulses and high efficiency.

In figure 2.5. results are shown for systematic variations in blade contour, pitch distribution and maximum blade camber. Changes are arbitrarily selected but show the effectiveness in reduction of pressure pulses at the cost of efficiency. Based such a variation the designer can select the most promising option, e.g. the maximum efficiency with the smallest increase in pressure pulses. In this case the blade camber is the most effective for the control of cavitation and pressure pulses. However, as usual in this sort of optimization processes better cavitation properties lead to a lower efficiency. Therefore given a maximum pressure amplitude the best efficiency can be obtained.
3. Characteristic design features of controllable pitch propellers

In section 2 the general characteristics of the design process are explained. In this section characteristic differences between controllable and fixed pitch propellers are discussed.

For fixed pitch propellers the blade length and thickness distribution at the hub can be freely optimised. The maximum thickness is determined from the strength requirement whereas the chord length should be long enough to avoid bubble cavitation. Special care has to be given to the fairing of the root fillets.

For controllable pitch propellers the situation is much more complicated. The simple requirement to be able to operate at negative pitch, limits the maximum chord length, especially at the root. Again the maximum thickness follows from strength requirements.

The freedom in choice of design parameters at the hub is fairly limited to a local change in pitch and camber.

Apart from the hub the largest difference between fixed and controllable pitch propellers is obviously the additional degree of freedom which is obtained by changing the pitch. This has consequences for the local inflow angles and the blade section shape. This deformation of the original design geometry is defined as pitch deflection and blade section distortion. Basically the effect is caused by the fact that after significant reduction in the pitch the cylindrical cross sections connect sections, which at the original design pitch were located at different radii. This is further illustrated in fig. 3.1.

Not only the blade section shape changes from character. Also the operating condition changes. For instance at constant number of revolutions the local cavitation of the section remains constant while the lift coefficient is reduced. As a result operation closer to the pressure side cavitation is therefore unavoidable. This is also shown in an open water diagram for a controllable pitch propeller where the lower limit against pressure side cavitation has been shown (fig. 3.2).

Normally pressure side cavitation is avoided in order to avoid cavitation erosion. Pressure side cavitation is found to be erosive which is related to the type of pressure distribution on the blade. Especially the large pressure gradient close to the leading edge causes a type of implosion on the blade which is erosive.

This effect of pressure side cavitation has been known for a very long time. In recent years, especially as a result of the application of resiliently mounted engine noise and vibration problems have arisen as a result of pressure side cavitation. Examples of several of those cases have been given by Lepeix [4]. Lepeix describes two problems, one on a cruise vessel (Sovereign of the Seas 10.2 MW, 4.9 m diameter).
The normal operating number of revolutions is 135 RPM. The minimum number of revolutions was set at 125 RPM! This value is to avoid any resonance of the engines on their elastic mountings. As a result the mean pitch value at low speed was then quite low. Moreover the limited excitation at full speed was partly obtained by a pronounced tip unloading with a strong decrease of the pitch at the top of the blades. By using these propellers with a low mean pitch complicated flow phenomena occur when the main part of the blades produces positive thrust and the outer area of the blades produced negative thrust.

Fig. 3.3 shows three pitch distributions (31 degrees), one for the free running design condition for zero thrust and at 6 degrees pitch. The free running pitch shows a normal distribution and thereby radial load. At zero thrust the outer negative thrust is compensated by the inner positive thrust. It is therefore understandable that at any intermediate condition the pitch at the tip becomes negative. The importance of these intermediate conditions has been illustrated by the noise and vibration problems, which have been encountered in practice. There has not been a proper description of the cavitation phenomena in technical literature.

In the following a first attempt is presented. In the intermediate pitch conditions of fig. 3.3 cavitation observations in a large size cavitation tunnel with axial and tangential simulated wake showed (fig. 3.4):

- Large amount of vortex type pressure side cavitation from 0.7 radius and upwards.
- At the blade tip a secondary tip vortex occurred with lower strength.
- Looking from aft the two vortices are not contracted but the move outwards, such that effectively the tip clearance between the vortex and the hull is reduced.
- The two vortices are counterrotating and therefore show a strong interaction. This interaction causes the individual vortices to become unstable.

The consequences of such behaviour is shown in fig. 3.5. Instead of clear blade rate frequencies, broad band pressure pulses are generated at 3 to 6 times blade frequency.

For the normal design optimisation the avoidance of unstable pressure side cavitation is of vital importance in the condition which regularly occur in the sailing profile of the ship. Although it is possible to theoretically predict the unstable behaviour of interacting tip vortices an engineering approach has been developed which gives a good estimate of the unstable pressure side cavitation. Then the operational limits of the controllable pitch propeller can be outlined as indicated in fig. 3.6. At high number of revolutions a lower limit in pitch exist below which the pressure side cavitation becomes unstable.
Trends in vibrations and noise

From the foregoing it is clear what the limitations are in the use of controllable pitch propellers. At maximum (or high) power the level of pressure amplitudes should be correlated with the vibration limits at multiple blade frequencies.
Not much more than five years ago the levels of blade rate amplitudes were in the order of 3 to 6 kpa. Nowadays, in spite of the increase in power, by careful optimisation of the propeller geometry, values of 1 to 2 kpa have been reached. These values have been measured on full scale ships such as high powered cruise vessels (twin screw) and container vessels (single screw).
It should further be remarked that in spite of the strong reduction in blade rate components the higher order components cannot be affected that much. The level of these components depends on the wake distribution rather than on the propeller geometry.
During recent experiments with several designs for a high powered container vessel it become apparent that blade rate pressure amplitudes vary with about 30 percent. The levels of 2nd and 3rd blade rate pressure amplitudes however varied with only 10 percent in a now consistent way.
A second point should be clear from the observations made in section 3. At low pitch settings the level of excitation is not the problem. Rather it is the broad band character of the excitation which creates local resonance and noise.

From the acoustical point of view the situation is even more complicated.
The shipyards have adjusted the constructions of their new ships to take advantage of the lower levels of pressure amplitudes. In combination with more sophisticated design tools (such as finite element techniques) the lower blade rate pressure amplitudes lead to a lighter and weaker structure.
From acoustical point of view it is then obvious that the properties are worse than before. This is especially troublesome as the control of noise is difficult as:

- It is as yet not common practice to use experiments to predict noise levels from propellers.
- Prediction methods either based on numerical or statistical methods sometimes give quite different results.

In fig. 4.1. two predictions are compared for the noise levels induced by the same propeller. The numerical method is based on unsteady lifting surface theory modelling the cavitation volume as a monopole. The statistical prediction is based upon the method of de Bruijn et al [5] and includes a correction of 10 dB to account for a low noise design. Large deviations are present. In this case the higher level is expected to be more realistic given the simplifications in the physical model. The combination of limitations in prediction methods and the trend of building ships more critical from acoustical point of view makes it necessary to develop a consistent design methodology from acoustical point of view.
The combination of the limitations in propeller noise prediction methods and the building of ships, which are acoustical more critical, makes it necessary to develop a consistent design methodology from acoustical point of view.
5. Conclusions

- The number of controllable pitch propellers compared to fixed pitch propellers is still growing.

- In the design several more complications are present then for fixed pitch propellers such as
  * limited chord length in the root area.
  * many off design conditions to be considered.

- Especially at low pitch settings the occurrence of unstable pressure side cavitation should be avoided in normal operating conditions.

- The cavitation phenomena in such conditions are dynamic of character and creates broadband excitation.

- Blade rate pressure amplitudes at maximum power can be controlled successfully. Absolute low pressure amplitudes are measured at full scale nowadays (1 - 2 kpa).

- The shipyards have taken advantage of the lower pressure pulses by reducing the strength of the ship structure. A negative consequence of this is that also the acoustical properties become worse.

- The development of consistent acoustical design approach is necessary to avoid future problems.
References:


Fig. 1.1. Power density (= power divided by propeller disk area) for fixed and controllable pitch propellers versus ship speed.
Fig. 2.1. Propeller design process.
Fig. 2.2. Model scale (upper) and full scale (lower) observation compared with calculations for a container ship with a controllable pitch propeller.
Fig. 2.3. Calculated pressure pulses for ITTC 87 propeller compared to other predictions.
A - maximum calculated level
B - mean value of 16 calculations
C - Lips calculation
D - minimum calculated level
E - experimental result
Fig. 2.4. Effect of skew angle on calculated cavity volume and cavity volume velocity.
Fig. 2.5. Effect of geometry changes on efficiency and pressure pulses (O - camber, □ - pitch distribution, △ - chordlength distribution)
Fig. 3.1. Effect of change of pitch on sections located on
Fig. 3.2. Open water diagram of controllable pitch propeller with operating lines and pressure side cavitation inception.
Fig. 3.3. Pitch distribution of CP propeller at various conditions.
Fig. 3.4. Cavitation observations at extremely low pitch settings (6 degrees).
Fig. 3.5. Characteristic pressure amplitudes at extremely low pitch settings (6 degrees).
Fig. 3.6. Example of recommended operational limits of a controllable pitch propeller.
Fig. 4.1. Comparison of a numerical prediction of the noise level compared to a statistical.
A - statistical average
B - statistical average with 10 dB correction for low noise design
C - numerical prediction based on calculated cavitation volume
Design aspects of efficient marine propellers

T. van Beek, R. Verbeek, Research & Development Department Lips B.V., Drunen, The Netherlands

1 Introduction

For the design of fixed and controllable pitch propellers efficiency is the main target. Due to boundary conditions the maximum attainable efficiency is limited. For instance if the propeller could operate without a ship the diameter is not limited and very high efficiencies can be reached. The presence of the hull restricts the diameter and implies inhomogeneous inflow velocities. The finite draught of the ship causes that cavitation behaviour and pressure fluctuations, induced by the propeller on the ship's structure, are additional boundary conditions to be dealt with. All these factors by itself have their influence on the efficiency of the propeller. Maximizing the efficiency, while taking into account these boundary conditions, is difficult and the designer has to use sophisticated design tools.

In this paper the basic parameters and their consequences for the efficiency are discussed. The influence of propeller-hull interaction, number of blades, blade area ratio and propeller load are described. The results of cavitation and pressure fluctuation requirements on the blade design are discussed. Effects of skew and its consequences for the blade stresses are shown.

2.1 The efficiency of a single propeller without presence of the ship

The basic action of the propeller is to accelerate the flow through the propeller disk so that an increase in momentum is generated and a thrust force results.

With the aid of simple momentum theory a formula can be derived for the efficiency \( \eta \) under the assumption that there is no friction and that there are no rotational losses:

\[
\eta = \frac{2}{1 + \sqrt{1 + C_T}}
\]

where

\[
C_T = \frac{T}{\frac{1}{2} \rho V_e^2 \pi D^2}
\]

\( T \) = thrust
\( V_e \) = entrance velocity
\( D \) = diameter
\( \rho \) = specific density

\( C_T \) = thrust coefficient

For a given ship and shipspeed the required thrust is known and therefore the efficiency \( \eta \) increases with increasing propeller diameter (Fig. 1). For a more refined theory taking into account the propeller rotational losses the following relation can be found [1]:

\[
C_T = \frac{4(1 - \eta_{ol})}{\eta_{ol}} \left( 1 + (1 - \eta_{ol}) \frac{\lambda^2}{\lambda^2 + \eta_{ol}^2} \right) \left( 2 - \eta_{ol} \right) \frac{\lambda^2}{\eta_{ol}^2} \ln \left( \frac{\lambda^2 + \eta_{ol}^2}{\lambda^2} \right)
\]

Jahrbuch der Schiffbautechnischen Gesellschaft 77 (1983)
The open water efficiency thus calculated includes rotational losses for the propeller with infinite blade number without friction, but is only valid for one radial circulation distribution. The result is plotted in Fig. 2 which clearly shows that the rotational losses increase with the advance ratio $J$.

So far the propeller efficiency is a function of the $CT$ and the $J$-value of the propeller. An estimate of the influence of the viscous drag on the efficiency is given by:

$$\eta = \eta_0 \frac{1 - 2 e_l \lambda_l}{1 + 2/3 e_l \lambda_l}$$

with

$$\lambda = \frac{V_n}{n D} = \frac{1}{\pi}$$

$$\eta_0$$ — ideal open water efficiency

$n$ — rotational speed

The open water efficiency thus calculated includes rotational losses for the propeller with infinite blade number without friction, but is only valid for one radial circulation distribution. The result is plotted in Fig. 2 which clearly shows that the rotational losses increase with the advance ratio $J$.

So far the propeller efficiency is a function of the $CT$ and the $J$-value of the propeller. An estimate of the influence of the viscous drag on the efficiency is given by:

$$\eta = \eta_0 \frac{1 - 2 e_l \lambda_l}{1 + 2/3 e_l \lambda_l}$$

with

$$\lambda = \frac{\lambda}{\eta_0}$$

and

$$e_l = \text{drag lift ratio of the propeller section profile.}$$
Design aspects of efficient marine propellers

Figure 2 shows that $e_i$ has a distinct influence on the efficiency. More sophisticated calculations can be made according lifting line or lifting surface theories. Then also the effect of a finite blade number and the effect of the blade area ratio are taken into account.

The approach has been completely theoretically thus far. For practical applications one often uses experimental results of propeller series, for instance Wageningen B-series, where certain parameters such as propeller pitch and blade area ratio are systematically varied. With the aid of these measured open water characteristics one can choose for given thrust and diameter the optimum number of revolutions and determine the corresponding efficiency. When these resulting efficiencies are plotted, as a function of the blade area ratio per propeller blade ($A_d/A_d/2$), it appears that there is an optimum blade area ratio for a given thrust coefficient (Fig. 3). This figure is within limits valid for all B-series propellers. The physical meaning behind this optimum can be explained by known characteristics of wings with a low aspect ratio. Each wing in a potential flow (no friction) has a lift $L$ and a drag $D$. This drag force is induced by the circulation distribution around the wing:

$$C_D = \frac{C_L^2}{\Lambda}$$

where

- $C_D$ - induced drag coefficient ($D/(b \cdot s \cdot 1/2 \cdot \rho \cdot V^2)$)
- $C_L$ - lift coefficient ($L/(b \cdot s \cdot 1/2 \cdot \rho \cdot V^2)$)
- $b$ - mean chord of the wing
- $s$ - span of the wing
- $\Lambda = \frac{s}{b}$ - aspect ratio
- $k$ - factor depending on circulation distribution and wing platform

Figs. 4 and 5. Propeller open-water efficiency as a function of blade roughness.
A measure for the aspect ratio of a propeller blade is given by:

\[ \Lambda_p = \frac{D^2}{A_0/A_0} \frac{n}{2 \pi} \frac{D^2}{A_0/A_0} \]  

(5)

Thus for a given thrust (lift) and \( A_0/A_0 \), the aspect ratio increases with the number of blades.

In Fig. 3 both viscous friction due to the blade area ratio and induced drag depending on the aspect ratio play a role. With constant aspect ratio an increase of the blade number causes an increase in blade area and therefore the viscous friction. This decreases the efficiency. The lift coefficient \( C_L \) is proportional to \( T/A_0 \), and therefore the induced drag is proportional to \( C_0^1 \)

(6)

For constant aspect ratio and constant thrust an increasing blade number decreases therefore the induced drag. Apparently both effects compensate each other.

From these results it can be concluded that for a given number of blades the blade area ratio can be chosen such that the efficiency is optimum. However, in normal design practice other constraints have to be taken into account. To optimize the cavitation behaviour of the propeller with regard to cavitation erosion and pressure fluctuations, the blade area ratio often must be larger than a certain minimum. Therefore the point of optimum efficiency can not always be reached.

Another effect important for the efficiency of the propeller is the roughness of the propeller blade. The roughness of the blade surface is strongly depending on the quality of the grinding and on the time spent in environmental conditions. Figure 4 shows some typical values for the centre line average roughness (Ra) of propellers with different service conditions:

- New propeller: Class I (ISO 484/1) < 6 \( \mu \)m Ra
- New propeller: Class S (ISO 484/1) < 3 \( \mu \)m Ra
- Propeller 12 to 24 months in service: 20 \( \mu \)m Ra

Figure 4 typical values for propeller roughness.

At increasing roughness of the propeller blade the skin friction and therefore the drag of the blade increases. This increment of the skin friction has been investigated by Schlichting and Prandtl with the aid of sand roughness tests for plates. This has resulted in the following empirical relation for the increase in drag coefficient of a blade section

\[ \Delta C_D = 2 \left[ \left( 1.89 + 1.62 \log \left( \frac{c}{2 \text{kg}} \right) \right)^{-2.5} - 0.455 (\log R_c)^{-2.5} \right] \]  

(7)

with

- \( R_c \) - local Reynolds number \( \left( \frac{c V}{\nu} \right) \)
- \( c \) - chordlength
- \( V \) - inflow velocity
- \( \nu \) - kinematic viscosity
- \( \text{kg} \) - equivalent sand roughness

To estimate the effects of section drag upon the propeller characteristics the equivalent blade section approach of Lerbs [4] may be used. The equivalent sand roughness has to be related to the actual propeller roughness such that realistic changes in friction can be calculated. The usual method for circumventing this is to measure the drag of the rough surface at laboratory scale, compare it with that predicted by equation (7) and thereby determine an equivalent sand grain size. However, some discrepancies exist, the Prandtl-Schlichting formula applies to surfaces having roughness elements which are all geometrically similar (characterized by one parameter). This is not usually the case for surfaces produced in an industrial process, which have textures different from that of uniform sand. The second reason follows from the fact that the first part of equation (7) does not involve the Reynolds number. For industrial surfaces the variation of the roughness is such that this difference should be taken into account. For a more theoretical treatment reference [5] can be useful.

To illustrate the effect of roughness upon efficiency, results of measurements with a B5.75 type propeller with different roughnesses are given in Fig. 5 [3]. The results show that, at these Reynolds...
numbers, increasing \( k \) from 15 to 50 \( \mu m \) reduces the efficiency with about 5 percent. Although this difference is expected to be smaller on full scale, the loss in efficiency will be large enough to pay for a repolishing of the propeller blades.

### 2.2 Propeller hull Interaction

In the preceding section the influence of several parameters on the efficiency have been explained. In reality the vicinity of the ship's hull changes the efficiency due to propeller-hull interaction. The analysis of the propulsion factors by means of the well-known thrust identity method gives values for propeller-hull interaction in behind condition. The total propulsive efficiency is written as:

\[
\eta_D = \eta_\theta \eta_H \eta_R = \frac{R \cdot V_s}{2 \pi Q_p \cdot n}
\]  

with:

\[
\eta_\theta = \frac{TV_s}{2 \pi Q_p \cdot n}
\]

\[
\eta_H = \frac{1 - t}{1 - w}
\]

\[
\eta_R = \frac{Q_o}{Q_p}
\]

All the factors involved have a clear physical meaning. The Taylor wake fraction \( w = 1 - \frac{V_e}{V_s} \) describes the decrease in entrance velocity at the propeller. The thrust deduction coefficient \( t \) accounts for the increase in resistance due to the propeller suction. The relative rotative efficiency \( \eta_R \) accounts for the difference in torque between open water and behind condition. In practical cases \( \eta_H \) ranges from 0.90 to 1.25 and \( \eta_R \approx 0.95 \pm 1.05 \).

Placing the propeller infinitely far from the ship both \( t \) and \( w \) become zero and \( \eta_R \) equals 1, therefore

\[ \eta_D = \eta_\theta \]

When bringing the propeller closer to the ship two things happen:
- the thrust coefficient of the propeller increases due to deceleration of the water, thus the open water efficiency decreases
- the hull efficiency becomes larger than one (in most cases).

The influence of \( \eta_R \), being a value close to unity is for the time being neglected.

To illustrate the influence of the factors \( w \) and \( t \) on the propulsive efficiency formula (1) for the ideal efficiency can be used:

\[ \eta_I = \frac{2}{1 + \sqrt{1 + C_T}} \]

in which \( C_T \) is the thrust coefficient in behind condition:

\[ C_T = \frac{T}{\frac{1}{2} \rho V_e^2 \pi D^2} \]

The last equation can be written as:

\[ C_T = \frac{R}{\frac{1}{2} \rho V_e^2 \pi D^2} (1 - w)^2 (1 - t) = C_T = (1 - w)^2 (1 - t) \]

In Fig. 6 the resulting total efficiency according to formula (13) is given for constant values of \( w \) and \( t \). In Fig. 7 the total efficiency is given as a function of wake fraction for a given value of
$C_T$ ($= 3$) for a range of constant values of $\eta_H$ and corresponding $t$. It is not always favourable to strive at a high hull efficiency as illustrated by the points marked A and B in the figure.

In the wake fraction three components can be distinguished:

- the potential wake due to potential flow,
- the viscous wake due to viscous flow,
- the wave wake due to the orbital motion of the water particles.

Of these three the viscous wake is the largest and thus the most important.

Much effort is put into the prediction of $w$ by describing the viscous wake. Harvald [21] for instance used model tests to show the dependency of the wake fraction upon:

- the breadth to length ratio of the ship,
- the propeller diameter to shiplength ratio ($D/L$),
- the fullness of the ship ($C_9$),
- height above the keel to draught ratio ($E/d$),
- extreme variations of frame shape.

The results are shown in Fig. 8. The main relations can all be explained from the behaviour of the viscous wake. For instance a short ship ($B/L$ high) has a larger viscous wake than a long one. Also when for a given ship the propeller is enlarged the propeller in relation to the boundary layer thickness increases and the wake fraction decreases (see propeller diameter correction in Fig. 8).

The underlying assumption with the thrust identity method is that the propeller thrust is described by the $K_T$-curve of the open water diagram. From the measured thrust the $K_T$ can be calculated while the wake fraction is computed with the aid of Fig. 9. Only if the $K_T$-curve is approximated by a straight line and the number of revolutions is identical behind and far from the ship, it can be shown that a linear relation exists between the thrust deduction coefficient and the wake fraction. This is in accordance with the well-known formula as given by Taylor

$$t = 0.5 + 0.7w.$$  \hfill (14)  

A very important aspect of propeller-hull interaction is the dependency on the propeller diameter in relation to the ship's length. A well-known procedure to reduce the fuel consumption of existing ships is to reduce the shipspeed by several knots and to replace the existing propeller by a larger one. This has been utilized for several large tankers as with these ships the tip clearance of the original propeller is large enough to allow for an increase in propeller diameter.
Under the assumption that the thrust decreases with the ship's velocity squared an increasing diameter leads to a decrease in thrust coefficient:

\[ C_T = \frac{T}{\frac{1}{2} \rho V^2 \pi D^2} \]

The open water efficiency then increases according to equation (1). The variation of \( \eta_D \) with the diameter is then given by:

\[ \frac{\partial \eta_D}{\partial D} = \eta_H \frac{\partial \eta_H}{\partial D} + \eta_0 \frac{\partial \eta_0}{\partial D} \]

\[ (15) \]

With the definitions according to (10) and (13) and assuming that \( \eta_0 \) is only a function of \( C_T \) this can be written as:

\[ \frac{\partial \eta_D}{\partial D} = \eta_H \frac{\partial \eta_H}{\partial C_T} C_T \left( -\frac{2}{D} + \frac{2}{(1-w) D} + \frac{1}{(1-1) D} \right) + \eta_0 \left( \frac{\partial \eta_0}{\partial D} - \frac{\partial \eta_1}{\partial D} \right) \]

\[ (16) \]
Fig. 10. Propeller design conditions; 1 = homogeneous flow; infinite draught; 2 = propeller behind ship; infinite draught; 3 = propeller behind ship; finite draught

From Harvalds diagram (Fig. 8) one can deduce that the value of $\frac{\partial w}{\partial D}$ varies from $-3/L$ to $-6/L$, where $L$ is the ship's length. The value of $\frac{\partial t}{\partial D}$ is considered small.

This equation describes that the increase in the total efficiency due to the increase in open water efficiency is partly cancelled by the reduction in hull efficiency.

2.3 Effect of design criteria on propeller efficiency

In the preceding sections only overall effects are treated. For actual propeller design the local inflow velocities in the wakefield and the cavitation behaviour have to be considered.

The final efficiency of the design will be lower than global considerations indicate. To show that this difference originates from three principal different causes, the following design exercise is made. In Fig. 10 three situations are shown:

- propeller in homogeneous flow, infinite draught,
- propeller behind ship, infinite draught,
- propeller behind ship, finite draught.

If no restrictions are present the propeller operates in a homogeneous flow at infinite draught so that no cavitation or any other related problems exist.

For given propeller diameter the blade area ratio can be chosen such that from Fig. 3 the efficiency is highest. It is also evident that the diameter should be taken as large as possible, as can be
shown by eq. (1). The presence of the ship as a second step in this process geometrically restricts the diameter. Also the inflow velocities in front of the propeller are inhomogeneous. Now the propeller-hull interaction plays a role. From section 2.2 it will be clear that this has effect on the overall efficiency of the propeller. The design is further complicated because fluctuating forces in blade, hub and shaft are to be considered. The effects of cavitation can only then play a role when the finite draught of the ship is taken into account. Then during one revolution a certain amount of cavitation can exist which may lead to:

- cavitation erosion,
- cavitation noise,
- pressure fluctuations on the ship's structure.

The design must be such that no erosion occurs and that cavitation, noise and pressure fluctuation levels are acceptable. Erosion and noise can be controlled by controlling the type and the extent of the cavitation. The pressure fluctuations which are dominated by the cavitation, as will be shown in the next section, can be controlled by changing the pitch and or skew distribution.

All forementioned restrictions on the propeller design decrease the efficiency of the propeller. This is illustrated by a specific design example (Fig. 11) where three different propeller designs are compared. In this case the step from a homogeneous wakefield to a non-uniform wakefield costs 2 percent efficiency. The increase in blade area ratio, necessary to avoid erosion and vibration problems,
costs an additional 3 percent. It can be concluded that the ideal efficiency, as given by equation (1),
cannot be reached due to
- rotational losses,
- frictional losses,
- non homogeneous inflow,
- losses due to cavitation and vibration requirements.

In Fig. 12 results of Lips' designs made during the last two years have been plotted and compared
with equation (1).

Also indicated is the result for optimum B4-55 propellers. Both non-uniform inflow and different
blade area ratio are the reason for the differences between B-series optimum and final design results.
To show the seriousness of this loss in Fig. 13 the differences are given as losses in horsepowers.

2.4 Method to improve propulsion efficiency of C.P. propeller equipped ships

The difference between fixed and controllable pitch propellers is the freedom of pitch variation of
the latter. The degree of freedom can be used to optimize the fuel consumption of the engine. For a
certain sea state and shipspeed the thrust to propel the ship is constant. Several combinations of prop-
peller pitch and propeller speed can be selected to deliver this thrust.

A specially programmed micro processor can be used for automatic selection of the proper combi-
nation of pitch and rpm such that the fuel consumption of the engine is minimal. A further advan-
tage of such a system is that adaption to changes in the motor characteristics and the ship's resistance,
due to fouling, is automatically achieved.

This process was simulated on a computer for a 30 000 tdw container ship. For a certain sea state
and thrust an additional 3% decrease in fuel consumption was achieved by the proper selection of
pitch and rpm. With the control system an optimum adaption of propeller, ship and engine character-
istics can be achieved. Changes in characteristics due to fouling etc. are automatically coped with and
the total propulsive efficiency is continuously optimized. The same system can also be used to
optimize other criteria such as noise.

3 Implications of cavitation and pressure fluctuation requirements on blade design

3.1 Introduction

The primary task for a propeller is to deliver a required thrust with a good efficiency. It has been
shown that, given the wakefield, the number of blades, the operational conditions and the loading
distribution on the propeller, the efficiency mainly depends on the blade area ratio. The choice of the
blade area ratio and the loading distribution however, is limited by the requirement that a good
cavitation performance must be achieved and that the propeller induced pressure fluctuations are
kept within reasonable limits.

Basic requirements to achieve acceptable cavitation performance of the propeller are:
- Face-cavitation should be avoided under all conditions.
- Back cavitation is allowable provided that the extent and type will not lead to cavitation
  erosion.

The way cavitation requirements are incorporated in the overall hydrodynamic design procedure is
described elsewhere (see f.i. [7]) and will not be repeated here.

The requirement that reasonable pressure fluctuations are to be met, influences the blade design
also. Part of the induced pressures are due to the pulsating cavities on the propeller blades. Limita-
tions in allowable pressure fluctuations not only restrict the final blade area ratio in order to keep the
extent of the cavitation limited but also affects other design parameters as skew, tipoff-loading and
cordlength distribution. The effect of pressure fluctuations restrictions on the blade design will be
dealt with in more detail in the next sections. Attention will also be given on the strength aspects of
the propeller.
3.2 Propeller induced pressure fluctuations

The development of highly powered ships has led to highly loaded propeller blades. The blades can be designed in such a way that the propeller operates free of cavitation during most of its passage in the wake. The variation in inflow velocity in the wake peak however, leads to a rapid increase and decrease of cavities on the propeller blades.

This growth and collapse of the cavities gives rise to large fluctuations of the pressure in the fluid surrounding the propeller. The fluctuating pressure field round the propeller leads to excitation forces on the ship's hull which can act as an important source of noise and vibration inboard the ship. For environmental and structural reasons these forces must be kept to a minimum.

Three effects contribute to the propeller induced pressures:

- Pressures induced by the rotating non-cavitating, non-loaded propeller,
- Pressures induced by the loading of the propeller,
- Pressures induced by the rapid growth and collapse of cavities on the blade.

The pressures induced by the thickness and loading effect have a more or less sinusoidal character with a predominant blade frequent component. Phase differences over the aft body are large and the pressures are decreasing rapidly with increasing distance from the propeller. The resulting excitation forces are therefore relatively low.

The characteristics of the pressures induced by the pulsating cavities on the blade are different.

a) Not only blade frequent components are induced but also higher order components can reach significant levels.

b) The decay of the pressures with increasing distance is less rapid, so a large part of the aft body of the ship will be influenced.

c) The phase differences over the aft body are small.

Though the pressure amplitudes induced by the pulsating cavities can have the same magnitude as those induced by the loading and thickness effects, the resulting excitation force due to cavitation can reach several times the value of the excitation force resulting from the thickness and loading effect. This is mainly due to the approximately constant phase angle of the cavity pressure signal induced on the aft body. Therefore in the design of the propeller care has to be given that the cavitation induced pressures are kept within reasonable limits.

Several methods exist to predict the propeller induced pressure loads, reaching from simple ones (see f.i. [18]), based on a statistical analysis of full scale and tank results to the more elaborate ones (see f.i. [19]), requiring large amounts of computer time and only suitable for analysis purposes in the final stage of the propeller design.

A good compromise between the two is the method developed by several authors ([10, 11, 12]) and based on the linearized cavity theory of Geurst [13, 14] and Geurst and Verbrugh [15].

In the calculation procedure the propeller load is represented by rotating pressure dipoles and the non-loaded, non-cavitating propeller by rotating sources and sinks. The pulsating cavities are also represented by rotating sources and sinks, though in this case the source strength will also depend on time.

Only attention will be given to the contribution of the cavitation induced pressures. Detailed information of the complete calculation procedure can be found in literature (see [12]).

The method is based on the calculation of the potential flow around the propeller. Ignoring the vorticity in the inflow field of the propeller and assuming that all perturbation velocities are small in comparison with the ship speed, $V_s$, the equation of Bernoulli in any field point $P$ can be written as

$$P_i = \rho \frac{\partial \phi}{\partial t} + \rho V_s \frac{\partial \phi}{\partial x}$$  \hspace{1cm} (17)

where
- $P_i$ = induced fluctuating pressure
- $\rho$ = density
- $\phi$ = potential
- $t$ = time
- $V_s$ = ship speed
- $x$ = x-coordinate (see Fig. 14 for definition of axis)
In equation (17) constant terms and higher order terms in the perturbation velocities are neglected and the velocity in point P is assumed to be $V_s$. When the sources and sinks representing the cavity rotate steadily with speed $\omega$, equation (17) can be written as

$$P_1 = -\rho \omega \frac{\partial \phi}{\partial \gamma} + \rho V_s \frac{\partial \phi}{\partial x}$$

with $\omega$ - rotational speed
$\gamma$ - propeller position

Consider now a cavity on a blade section at radius $r$ (see Fig. 15). From the thin section theory one can derive an expression for the source density $m$ representing the cavity thickness $\tau$.

$$m = U_s \frac{\partial r}{\partial x_s} + \frac{\partial r}{\partial t}$$

with $U_s$ - local inflow velocity
$x_s$ - local ordinate
$\tau$ - cavity thickness

The cavity thickness distribution along the chord can be calculated when the local inflow velocity, incidence angle, the camber distribution of the blade section and the local cavitation number are given (see [13, 14]). The potential associated with the source density is

$$\phi = -\frac{1}{4\pi} \int \int \frac{m}{d} \, dA$$

with $d$ - distance from point of blade to point P in free space
$m$ - source density of the cavity
and the surface integral is taken over that part of the propeller where cavitation occurs.

The induced pressure is given by (18):

$$P_{cw} = \frac{\rho \omega}{4\pi} \frac{\partial}{\partial \gamma} \int \int \frac{m}{d} \, dA \quad + \quad \frac{\rho V_s}{4\pi} \frac{\partial}{\partial x} \int \int \frac{m}{d} \, dA$$

where $T$ - local cavity thickness
$x_s$ - local ordinate
The expressions for evaluation of the propeller induced pressure require lengthy calculations and are only suitable for processing on a digital computer. Such a program is implemented in the Lips design procedure and gives satisfactory results in comparison with full-scale and tank results (see [16]) and is therefore a useful tool in the design process.

To gain some insight into the induced pressures some simplifications will be made. Inserting equation (19) in (20) gives:

$$\phi_c = -\frac{1}{4\pi} \iint \left( \frac{U_x \frac{\partial \tau}{\partial x} + U_y \frac{\partial \tau}{\partial y}}{d} \right) dx dy$$  \hspace{1cm} (22)

The distance $d$ from a point on the blade to a field point, $P$, is a function of $x$, $y$, $r$, and the coordinates of the field point. Replacing $d$ by $d^*$ ($\gamma$, field point) where $d^*$ represents some average distance from the propeller cavities to the field point, equation (22) can be written as:

$$\phi_c = -\frac{1}{4\pi} \frac{1}{d^*} \iint \left( U_x \frac{\partial \tau}{\partial x} + U_y \frac{\partial \tau}{\partial y} \right) dx dy$$  \hspace{1cm} (23)

This equation can be evaluated to give:

$$\phi_c = -\frac{1}{4\pi} \frac{1}{d^*} \int \left( U_x \frac{\partial \tau}{\partial x} + U_y \frac{\partial \tau}{\partial y} \right) dv$$  \hspace{1cm} (24)

where $V_c$ = volume of the cavities (function of the propeller position)

$\text{d}^*$ = average distance from the cavities to the field point.

Inserting (24) in (18) results in an approximation for the induced pressure by the pulsating cavities:

$$P_c = \frac{\rho \omega^2}{4\pi d^*} \frac{\partial V_c}{\partial \gamma} + \frac{\partial V_c}{\partial x} \left( \frac{1}{4\pi} \frac{\partial \gamma}{\partial x} \frac{d^*}{d^*} - \frac{\rho \omega V_x}{4\pi} \frac{\partial (V_c \frac{\partial \tau}{\partial x})}{\partial x} \right)$$  \hspace{1cm} (25)

It can be shown that the derivatives $\partial/\partial \gamma (1/d^*)$ and $\partial/\partial x (1/d^*)$ are proportional to $(1/d^*)^3$.

In the far field approximation of equation (25) this reduces to:

$$P_c = \frac{\rho \omega^2}{4\pi d^*} \frac{\partial^2 V_c}{\partial \gamma^2}$$  \hspace{1cm} (26)

since the contribution of the second term in (25) becomes small compared with the first term.

Equation (26) leads to the following consideration. In order to keep the pressure pulses originating from the pulsating cavities to a minimum, the extent of the cavitation should be kept small and the growth and collapse of the cavities on the propeller blade should be such that a low value of the second derivative of the volume is achieved. A strong tip-off loading can be applied to minimize the extent of the cavitation. This goes however to the expense of the efficiency of the propeller and in modern propeller design other solutions are used.

To diminish the contribution of the cavitation to the propeller induced pressures it is common practice to modify the leading edge in such a way that cavitation on the blade (or rather the total volume of the cavities) is gradually increasing and decreasing in order to obtain low values of the second derivative of the cavity volume. Since the leading edge contour depends on the chord length and skew distribution of the blade a careful weighted combination of these two is necessary to achieve this goal.

The arithmetic involved in the determination of the propeller induced pressures is rather complex. In normal practice, the proper amount of skew is therefore selected after finalizing intermediate designs. In an iterative way it is then possible to determine a combination of skew, chord length, pitch distribution etc., which without sacrificing too much efficiency, gives acceptable pressure levels. The aid of sophisticated computer programs is inevitable and fully computerized design procedures are a necessary requirement to optimize modern propeller designs.

In the design of highly loaded propeller blades often a large amount of skew is applied to get the proper leading edge contour. The application of skew is often successful and can have a considerable influence on the level of the induced pressures generated by the propeller. The design should however, be checked on strength aspects since in highly skewed blades stress levels differ from the stress levels in conventional blade designs.
3.3 Effect of skew and unequal blade spacing on propeller induced pressures

The leading edge contour of the propeller blade has a significant influence on the pressure fluctuations originating from the pulsating cavities. Alternative leading edge contours are arrived at by giving the propeller a certain amount of skew, keeping the chord length distribution the same. Another method is to modify the chordlength distribution and applying little skew. However, this is not common practice since it will result in excessive blade area ratios. The loss in efficiency will then be significant and the weight of the propeller increases, while in controllable pitch propellers zero pitch passage will not be possible anymore. The application of skew therefore is favourable for several reasons.

To show the effect of skew on propeller induced pressures some designs with different skew distributions have been analysed keeping other geometrical quantities the same. The results of the first and second blade harmonic pressure levels are shown in Fig. 16.

In the shown example the first harmonic reaches a maximum with moderate skew. This indicates that the proper amount of skew must be applied in order to decrease the propeller induced pressure fluctuations. To which extent pressure amplitudes are allowable is not always clear from the beginning. The response of the ship structure to the first and higher order blade frequent components in the pressure signal is of importance.

For equally spaced blades the pressure signal of the propeller contains only multiple blade frequent components. With the concept of unequal blade spacing it is possible to introduce other components in the pressure signal with a lower amplitude. This has been calculated for a 4-bladed propeller as shown in Fig. 17.

The blade spacing can be expressed as the angle $\alpha$ between blade 1 and 2. The resulting pressure signal from the propeller now contains components with multiple twice shaft frequent components.

![Fig. 16. Influence of skew on propeller induced pressure fluctuations](image1)

![Fig. 17. Propeller with unequal blade spacing](image2)

![Fig. 18. Pressure amplitudes as function of blade spacing. 2N: twice shaft frequent etc.](image3)
Design aspects of efficient marine propellers

These are shown in Fig. 18. As can be seen from the figure, for \( \alpha = 45^\circ \) the first blade frequent component is zero and replaced by 2N and 6N components. With the concept of unequal blade spacing it is possible therefore to diminish certain components in the pressure signal.

A more favourable adaption of the pressure signal to the vibration characteristics of the ship can be achieved this way.

4 Strength aspects in relation to the application of skew on propeller blades

Application of skew has an effect on the stress levels and the stress distribution in propeller blades (see e.g. [17, 18]). Compared with a conventional design more torsional loads will act on the several cross sections of the propeller blades and the cross sections will tend to warp. Since the root region of the blade and the hub will resist this warping, additional, in certain cases significant, stresses will arise in the blade. These additional stresses will primarily be located at the lower radii of the blade. When the stresses in the blade are calculated with simple beam theory the predictions of place and magnitude of the stresses in the blades will turn out to be false. Skew results in higher stress levels and the maximum does not occur anymore at the maximum thickness of the blade sections, but moves towards the trailing edge of the blade sections.

To illustrate this effect some designs have been analysed. The operational conditions, blade area ratio, pitch distribution and so on were kept constant and only the skew and rake were altered.

The stresses in the blade are calculated with a finite element program. The applied element mesh is shown in Figs. 19, 20 and 21. The resulting equivalent stresses at the suction and pressure side of the sections at \( t/R = 0.4 \) and \( t/R = 0.6 \) are shown in the Figs. 22, 23 and 24. Also indicated in the figures are the maximum stresses as calculated with simple beam theory. Figures 25, 26 and 27 show contours of equal equivalent stress on the pressure side of the propeller.
Fig. 22. Equivalent stresses propeller A

Fig. 23. Equivalent stresses propeller B

Fig. 24. Equivalent stresses propeller C

Fig. 25. Equivalent stress contour at pressure side of propeller A, level 10 = 4.70 kN/cm²
Fig. 26. Equivalent stress contour at pressure side of propeller B, level 10 = 4.50 kN/cm²

Fig. 27. Equivalent stress contour at pressure side of propeller C, level 10 = 7.30 kN/cm²

Fig. 28. Propeller blade with original and modified trailing edge contour

Fig. 29. Stress distributions

Fig. 30. Stress distributions
It can be seen that for small skew the finite element model and simple beam theory both predict the same stress level. However, at the lower radii ($r/R = 0.4$) with increasing skew, the stress distribution flattens (propeller B) and ultimately the maximum stress occurs at the trailing edge of the profile section (propeller C). At higher radii this effect is less pronounced indicating that the additional stress due to warping is decreasing.

The trailing edge contour of propeller C differs from those of propeller A and B. The trailing edge contour has influence on the stress levels in the propeller blade.
This is illustrated with the following example. For a propeller as shown in Fig. 28, the trailing edge was modified. The pitch distribution was kept the same and camber and thickness were modified in order to keep $f/c$ and $ct^2$ constant.

Here $f$ — camber of the section,
$c$ — chordlength of the section,
and $t$ — thickness of the section.

The induced pressure fluctuations by both propellers were the same. The resulting stresses calculated with simple beam theory were the same for both propellers in equal design conditions. The stress levels calculated with the finite element program show however that the modification has a significant influence on the overall stress levels. The result of the calculation for the radii $r/R = 0.3$, $0.4$, $0.5$ and $0.6$ are shown in Figs. 29, 30, 31 and 32. Contours of equal equivalent stress are shown in Figs. 33 and 34.

As can be seen from these figures, the stress pattern at the trailing edge is strongly influenced.

With the last approach it is possible to decrease the blade area ratio and so influence the efficiency (see (19)) without increasing the induced pressure fluctuations. Care has to be taken, however, to ensure that the resulting design will have enough strength because modifications at the trailing edge can have a significant influence at the stress-levels in the propeller blade.

5 Conclusions

- Important parameters for the propeller efficiency are the thrust coefficient and the blade area ratio per blade.
- Actual propeller efficiency is influenced by the inhomogeneous wakefield and limitations due to cavitation and propeller induced pressure fluctuations.
- Increasing roughness of the propeller surface leads to a significant decrease in propeller efficiency.
- The proper amount of skew diminishes propeller induced pressure fluctuations.
- With unequal blade spacing unfavourable harmonic components in the propeller induced pressure signal can be avoided.
- With skewed propellers careful attention must be given to the strength of the propeller.
- Automatic control of pitch to sailing conditions leads to an increase in overall propulsive efficiency.

References

1 van Mannen, J. D.: Resistance and propulsion Part B. NSMB Publication 132a.
Design aspects of efficient marine propellers

Summary. The design of high efficient fixed and controllable pitch propellers is the main task for propeller designers. The main parameters for the efficiency are analysed. Propeller load and the blade area ratio per blade are shown to have a considerable influence.

In "behind condition" the propeller efficiency is lower due to the non-uniform inflow and higher blade area ratios which results from cavitation and hull pressure limitations.

In the design cavitation erosion must be avoided and propeller induced vibrations should be minimized. The propeller geometry optimization then often leads to the application of a certain amount of skew. Subsequent variations of skew in the design stage reveals that pressure fluctuations are reduced but that attention must be given to the stress levels in the propeller blades.
VT525 Marine Propulsion Systems
working principles, design and application
Zaal E 5th and 6th hour
Starting 1st april 2004

Course by: Teus van Beek

Content
wk 1 Introduction to marine propulsion systems
wk 1 Ship types and propulsors
wk 1,2 Fixed pitch propellers
wk 2 Controllable pitch propellers
wk 3 Steerable thrusters and pods
wk 4 Waterjets
wk 5 Transverse thrusters
wk 5 Hydraulic systems and controls
wk 6 Class requirements, rudder interaction and material aspects
wk 7 Cost driving factors and system
1. Steerable thrusters and pods

Duration: 1.5 hour
Main goal: main working principle and design criteria

Content:
- Why steerable thrusters and pods?
- Ship types and applications
- Hydrodynamic design aspects of thrusters
- Mechanical design and working principle
- Interaction thrusters
  - coanda effect
  - interaction effect

MT525 Marine Propulsion Systems
### Why apply steerable thrusters

Steerable thrusters comply with maritime market demands for:

- Generating full thrust over 360 degrees steering angle:
  - Improved manoeuvrability
  - Accurate Dynamic Positioning
  - Effective braking / stopping power (90 - 180 steering angle)
- Creating compact propulsion arrangements

**Additional benefits:**

- Saving machinery space / increasing cargo volume (short shafting, no gearboxes)
- Saving cost of rudders, stern tubes and shaft brackets

---

### Steerable thrusters types

#### Available thrusters:

- **Modular thrusters** 800 – 7,000 kW
- **Can-mounted thrusters** 800 – 7,000 kW
  (modular thruster bolted in steel mounting can)
- **Small retractable thrusters** 800 – 2,000 kW
- **Large retractable thrusters** 2,000 – 7,000 kW
- **Containerised thrusters** (modular retractable) up to 7,000 kW
- **Underwater de-mountable thrusters** up to 7,000 kW

**Design:** very flexible and easily adaptable to L- and Z-drive

**Application:** in a wide range of Offshore and Seagoing vessels.
Design requirements for thrusters

- Semi submersible
  - high bollard pull requirements
  - safety and reliability aspects
  - dynamic positioning: most of the time low power operation
- Mission profile thruster with CPP:

Steerable thrusters types

Available thrusters:
- Weld-in thrusters
  - 500 - 3.000 kW
- Can-mounted thrusters (compact thruster welded in steel mounting can)
  - 500 - 3.000 kW

Design: standardised and only available as Z-drive

Application: Tugs, Supply Vessel, AH / Tug Supply Vessel, Oil Recovery Vessel, Buoy Laying Vessel.
Design requirements for thrusters

- Tug:
  - High bollard pull requirement
  - Mission profile:

Thruster types

- In power range 800 - 7000 kW
- As L- or Z-drives
- With CPP's or FPP's
- With open propeller or propeller in nozzle
Thruster types

- Cast housing
- Input shaft
- Beveled gear set
- Pinion shaft
- Seals
- Anti-friction bearings

MT 525 Marine Propulsion Systems

Thruster types

- Cast housing
- Steering gear wheel
- Steering pipe upper bearing
- Steering motors

MT 525 Marine Propulsion Systems
Thruster types

- Support pipe
- Steering pipe
- Lower steering pipe bearing
- Steering pipe seal
  (liner with ceramic coating)

- Extra V-type seal

- Bottom well
- Support pipe
- Steering pipe

- Vertical floating drive shaft between upper and propeller gearbox
- Bearing
- Triple lip seal/liner with ceramic coating

MT 525 Marine Propulsion Systems

---

Thruster types

- Cast housing
- Pinion shaft
- Propeller shaft
- Bevel gear set
- Bearings
- Propeller shaft seal
- Rope guard

- Triple Viton lip seal/liner with ceramic coating

- Pinion
- Rigid cast housing
- Crown wheel

MT 525 Marine Propulsion Systems
Applications

- Mounting flange
- Water tight covers
- Air valve
- Holding pins
- Thrust ring

Main input data / thruster selection

To select a fit to purpose thruster following input data are required:

- Type of vessel
- Input power (kW) and input speed (RPM)
- L or Z-drive
- Thruster application:
  - MP = main propulsion
  - DP = dynamic positioning
  - MAN = manoeuvring
- Propeller design / mission profile
  - bollard thrust only
  - free running only
  - free running combined with bollard thrust
- Classification Society, ice class

Note: mission profile represents the average loading during one year of operation.

MT 525 Marine Propulsion Systems
**Main input data / thruster selection**

Size of LGB and UGB depends on input torque by following formula:

\[
\text{Input torque (kNm)} = \text{input power (kW) x } 9550 \times \text{ice factor} \times \text{input speed (RPM)} \times \text{service factor}
\]

Service factor depends on application and mission profile:
- Tug or Drilling rig: DP / bollard pull thrust only \( sf = 1.0 \)
- Product tanker: MP / free running only \( sf = 0.9 \)

Note: each type of gearbox has a range of standard reduction ratios.

**Principles**
- Lower gear reduction ratio allows higher input torque
- Higher power or lower input speed needs bigger gearbox size

---

**Main input data / thruster selection**

Depends on bending moment from propeller thrust by following formula:

\[
\text{Bending moment (kNm)} = \text{Thrust (kN) x arm length (m)}
\]

Note: arm length is the distance from centre of propeller to lower steering pipe bearing

Principle: propeller in nozzle generates more thrust and higher bending moment
Main input data / thruster selection

Depends on steering moment illustrated by following formula:

\[
\text{Steering moment (kNm) = } c \times N^2 \times (\text{RPM})^2 \times \text{Prop. Diam.}^5 \text{ (m}^5)\]

Note: factor \( c \) depends on open propeller or propeller in nozzle

Principles:
- Bigger propeller diameter and higher propeller speed generate higher steering moment
- Higher steering moment requires more steering motors and higher pump capacity

MT 525 Marine Propulsion Systems.

---

Main input data / thruster selection

Diesel motor drive:
Thruster input speed is equal to engine speed

E-motor drive:
Thruster input speed depends of frequency of board net:
- 50 Hz synchronous speeds: 500 – 600 – 750 – 1.000 – 1.500 RPM
- 60 Hz synchronous speeds: 600 – 720 – 900 – 1.200 – 1.800 RPM

Note: L-drive (only one gearbox) has lower input speed than Z-drive

Principles:
- Higher power needs lower input speed (gear wheel ratio / strength)
- Thruster with FPP: needs an expensive variable speed E-drive drive
- Thruster with CPP: needs a simple constant speed E-drive drive

MT 525 Marine Propulsion Systems
**Main input data / thruster selection**

Open propeller:

In principle applied in case of:
- MP / free running designs (V > 13 knots)
- Light DP / free running conditions

Propeller in nozzle:
Is always applied in case of MAN and DP / bollard thrust designs

Nozzle types:
- Standard nozzle is 19A
- Special nozzle

Principles:
- Twin open propellers for MP: always contra rotating
- Twin propeller in nozzle for MP: can rotate same direction

---

**Thruster size / type denomination**

A steerable thruster consists of three main assemblies which are reflected in the thruster size / type denomination:

- **Upper gearbox** (only in case of Z-drive = horizontal input shaft)

- **Stem / steering gear box** (inboard part, enabling steering of the outboard part)
  - or Stem box

- **Lower gear box** (or propeller gearbox)
Vessel types with thrusters

1. Harbour Tug, Escort Tug, Terminal Tug
2. Supply Vessel, AH Tug / Supply Vessel, Diving Support Vessel
3. Cable Layer, Oceanographic Research Vessel
4. Well intervention / Stimulation Vessel, Offshore Constr. Vessel
5. Semi-Sub - Drilling Rig, Heavy Lift, Pipe Layer
6. Drilling ship, Heavy Lift ship, Pipe Laying ship
7. FPSO vessel
8. Product Tanker

Ships with thrusters
Ships with thrusters

MT 526 Marine Propulsion Systems

Ships with thrusters

MT 526 Marine Propulsion Systems
Design requirements

- Reliability and durability
- Modular approach
- Wide power range
- Different types and applications of thrusters
- Special custom-made solutions
- Worldwide service

Applications

- Up to 7000 kW
- As L- or Z-drives
- With FPP's or CPP's
- With open propellers or with nozzles

Example 3000 kW thruster
Applications

- From 2000 to 7000 kW
- As L- or Z-drives
- With FPP's or CPP's in nozzles
- With retraction by spindles

Note
- Maintenance afloat of propeller gearbox is possible by using a so-called habitat

Example 5500 kW thruster

MT 525 Marine Propulsion Systems

Applications

- Up to 2000 kW
- As L- or Z-drives
- With FPP's or CPP's in nozzles
- With retraction by cylinders

Example 1500 kW thruster

MT 525 Marine Propulsion Systems
Applications

- Up to 7000 kW
- As L-drives
- With FPP's and CPP's in nozzles
- As retrievable thrusters

Notes
- Z-drives are very seldom necessary in big offshore vessels
- Thruster maintenance possible without dry-docking by lifting out the container

Example 2200 kW thruster

MT 525 Marine Propulsion Systems

Applications

- Up to 7000 kW
- As L-drives
- With FPP's or CPP's in nozzles

Notes
- Z-drives are very seldom necessary in big offshore vessels
- Maintenance is possible without dry-docking by lifting out the container, or by using a so-called habitat

Example 3000 kW thruster

MT 525 Marine Propulsion Systems
Applications

- Up to 7000 kW
- As L- and Z-drives
- With FPP’s or CPP’s in nozzles

Note
- Maintenance is possible without dry-docking by demounting the outboard part

Example 3000 kW thruster

MT 525 Marine Propulsion Systems

Applications

Receptacle with steering gearbox
Three tubes for hoisting wires
One outboard hoisting wire
Three inboard hoisting wires

Mounting procedure

MT 525 Marine Propulsion Systems
Specials

- Existing equipment removed
  - 6 x Containerised thrusters

- Updated equipment installed
  - 6 x FS3500 retractable thrusters of 4500 kW

MT 525 Marine Propulsion Systems

Specials

Step 1

Step 2

Step 3

Illustration main mounting steps

MT 525 Marine Propulsion Systems
Influence of the hull depends on clearance and hull shape.

The closer the hull and more blockage in front of the propulsor, the larger the influence.
Influence of the hull

- **Stern drive tug:**
- **Hull shape** has an effect on the bollard pull: especially the slope of the aft verticals
- **Thrust**

![Graph showing the influence of hull shape on thrust deduction](image)

**Typical values thrust deduction:**
- Bollard: 0.06 (pull 7% lower than thrust)
- Towing: 0.07 to 0.4
- Free running: 0.12 to 0.30

**A stern**
- Thrust deduction is 2.5 times higher: 0.07 * 2.5 = 0.17
- When thruster is rotated the astern BP is 83/93 ≈ 0.89 times the ahead value
- Because the slipstream hits the hull!!

MT 525 Marine Propulsion Systems
Design of the propeller

Choices to be made:

- Open or in nozzle
- Given power and RPM
- Select propeller diameter based on operating profile
- Number of blades: normally four blades (CPP) and four (torsional vibrations) or five blades for FPP in view of torsional vibrations
- Fixed or controllable pitch propeller

Power and RPM:
- Normally based on requirements for from the operator
- Tug: size of container vessels in harbour, always 2 thrusters
- Semi sub: forces required in heavy seaways, number of thrusters may vary between 4 to 8 (costs, reliability, room, flexibility)
- For us the selection of the gearbox is important
- The larger the torque the larger, heavier and more expensive the gearbox (torque is limiting at lower RPM, power at max. RPM)

Diameter:
- Large diameter in view of better bollard pull
- Pitch not too low in view of performance:
  \[
  \text{thrust/power} = \frac{\text{merit coeff.}}{\text{(kW/m}^2\text{)}}
  \] (the merit coefficient is pitch dependent)
- In nozzle tip speed is restricted due to cavitation erosion (this depends on draught)
Design of the propeller

Merit depends on P/D ratio

\[ \text{thrust/kW} = \frac{\text{merit}}{\left(\text{kW/m}^2\right)^{0.33}} \]

Optimise bollard pull

\[
\frac{T}{P} = \frac{K_t \rho n^2 D^4}{2\pi n K_q \rho n^2 D^5} = c \left(\frac{K_t}{K_q}\right)^{2/3} \left(\frac{P}{\pi / 4D^2}\right)
\]

T thrust, P Power, n rps, D diameter

Kt Kq

MT 525 Marine Propulsion Systems
**Design of the propeller**

- **Cavitation:**
  1. Tip speed depends on the submergence.
  2. For harbour tug, tip speed should be below 33 m/s.
  3. For semi submersibles, this can go up to 36 m/s.
  4. Detailed calculation depends on RPM and submergence.
  5. Blade design for harbour tugs is dependent on operating conditions.
  6. For instance, FPP full ahead RPM at negative ships speed occurs.
  7. For DP low power operations is quite normal (CPP rather flat blades).

**Design of the propeller**

- **Design of CPP:**
  1. Are there any astern thrust requirements? If so, blades may be more flat to get more astern thrust performance.
  2. Will the thruster be rotated or the pitch set to astern?
     1. DP less than 40% max thrust pitch astern otherwise rotate thruster.

- **Design of FPP:**
  1. Choice of design point is crucial when can the full power be absorbed.
  2. In most cases, the bollard pull condition is the design point.
  3. Free running the large propeller will be lighter (given power RPM is higher).

MT 525 Marine Propulsion Systems
**Design of the nozzle**

- Why a nozzle?
- Increased bollard pull order 20% compared to open propeller
- At high speed the nozzle starts to brake (resistance instead of positive thrust)
- For high speed application or lightly loaded propellers a nozzle is not attractive

But is attractive for most thrusters!!!!
- Nozzle types used are 19A or our HR nozzle

- Working principle is based on extra acceleration of the flow due to the nozzle lift into the propeller
- The nozzle itself generates about 50% of the thrust in bollard conditions!!!!

MT 525 Marine Propulsion Systems

---

**Design of the nozzle**

- Nozzle has a fixed standard shape, normally a length of 50% of the propeller diameter.

- Well known nozzle types are:
  - 19A nozzle
  - 37 nozzle

MT 525 Marine Propulsion Systems
Figure 24. Curves for optimum rpm of different propeller types.

Design of the nozzle

Bollard condition  Free sailing

MT 525 Marine Propulsion Systems
Design of the nozzle

Comparison ahead performance

(1 + sub) = (1 + T_nozzle_prop)

Cth related to prop only

HR nozzle
37 nozzle
Optima nozzle
Superior nozzle
19A nozzle
Design of the nozzle

Comparison astern performance

Design of the nozzle

Influence of gearbox housing on performance

- Gearbox size depends on gear ratio
- Creates additional resistance
- A thruster has a lower efficiency than conventional propeller (Order 5% reduction due to hydrodynamic resistance)
Influence of gearbox housing on performance

MT 525 Marine Propulsion Systems
Influence of gearbox housing on performance

Open water test including housing
Compared to propeller without housing: torque higher and thrust lower

MT 525 Marine Propulsion Systems
Diesel driven thruster

FPP design:
- bollard: full power/RPM
- free runn: limited power/full RPM

MT 525 Marine Propulsion Systems

Diesel driven thruster

CPP design:
- bollard: full power/RPM
- free runn: full power/full RPM

MT 525 Marine Propulsion Systems
Diesel electric drive

Electric driven thruster

- Constant torque up to maximum RPM
- Over speed possible, this helps to find a better compromise between bollard condition and free running
Types of bearings

- Two types of bearings:
  - Roller bearings: low friction bearings / anti friction bearings
  - Plain bearings
Forces

External forces:
- thrust
- propeller mass

Internal forces:
- radial
- axial
- tangential

Reaction forces
- radial
- axial

Lifetime

L_{10h}-method:
lifetime fatigue calculation

Average Life
Rating Life (90 % Survival)

% Number of bearings failed
Life 10^6 revs

MT 525 Marine Propulsion Systems
Lifetime

\[ L_{10h} = \frac{1000000}{60n} \left( \frac{C}{P} \right)^p \]

- C: basic dynamic load rating (given construction bearing) N
- P: equivalent load (given loadings) N
- p: life time exponent: 10/3 for roller bearings, 3 for ball bearings
- n: speed RPM
- L: life time in hours

MT 525 Marine Propulsion Systems

\[ L_{10h} = \frac{1000000}{60n} \left( \frac{C}{P} \right)^p \]

- Increase parameter by 10%
- Effect on lifetime:
  - load → -27%
  - load capacity → +37%
  - speed → -9%

MT 525 Marine Propulsion Systems
Lifetime

- Continuous full power
- Specific load profile
- Lifetime increases 3 - 5 times

MT 525 Marine Propulsion Systems

Lifetime

- Life adjustment factors
  \[ L_{na} = a_1 \times a_{23} \times L_{10h} \]

- \( a_1 \): reliability factor
  - normally \( a_1 = 1.0 \) (90%)
  - higher reliability: \( a_1 = 0.62 \) (95%)
    \[ a_1 = 0.21 \] (99%)

- \( a_{23} \): factor for material and lubrication
  - viscosity dependent
  - \( a_{23} \) approx. between 1.2 - 2.0

MT 525 Marine Propulsion Systems
Lifetime

Effects on lifetime:

- Contamination
  - Excessive wear

- Water ingress
  - Oil film deteriorates

- Temperature
  - higher temperature → lower viscosity

Required lifetime:

- tug thrusters: ± 3,000 hours per year
- offshore thrusters: ± 25,000 hours full load
  ± 115,000 hours operating profile
**Bearings**

- Spherical roller bearing
  - two rows of rollers
  - self aligning
  - high load carrying capacity
  - Special type: EXPLORER higher load carrying capacity

---

**Bearings**

- Lower gearbox 3500

---

MT 525 Marine Propulsion Systems
Bearings

Spherical roller thrust bearing
- axial loads (outer ring loose)
- self aligning
- deflection + misalignment
- pre-load required

MT 525 Marine Propulsion Systems

Bearings

Slewing bearing
- radial + axial forces
- tilting moment
- integrated slewing gear
- compact

MT 525 Marine Propulsion Systems
Bearings

- **cylindrical roller bearing**
  - radial loads only
  - axial displacement
  - used for mounting purposes

MT 525 Marine Propulsion Systems
Bearings

- **Taper roller bearing**
  - radial + axial loads
  - radial load generates axial force
  - axial load in one direction

MT 525 Marine Propulsion Systems

---

Bearings

- **CARB™ bearing**
  - high angular misalignment
  - high axial displacement
  - high admissible loads

MT 525 Marine Propulsion Systems
Bearing adjustment

oversized bush
Bearing damage

Main causes of bearing damage:

- outer ring movement in housing: re-bush
- water ingress
- peak loading: split rollers
- standstill damage

MT 525 Marine Propulsion Systems

Bevel Gears

...smooth, curved surfaces...

MT 525 Marine Propulsion Systems
Gears

About: why and where
- Cost ratio gearset / gearbox
- Right-angle power transmission

Theoretical design
- Pinion, Ring gear, Housing
- Spiral direction & Forces: req'ts on bearing arrangement
- Contact: contact sequences, contact line movement

Manufacturing
- Cutting - Heat treatment - Finishing
- Hardening deformation, geometric quality
- Oxidation, purity etc.; material quality

Power rating
- Failure modes
- Optimization by calculation

Design (Housing)

Pinion and Ring Gear shapes
- integral
- shrunk
- bolted

Housing design
- Straddle-mounted
- Pinion & Gear
- Bearings on each side/one side

MT 525 Marine Propulsion Systems
Design (Forces)

- Spiral direction → Causes Forces → These are asymmetrical
  → Different bearing load (& life) → Preferred direction of rotation

Defaults:
- Pinion Left spiral
- Gear Right spiral
- Clockwise input
- Note bearing size

Gear design (Contact-1)

- Contact line movement
  - Contact is a (broad) line
  - Moving inward
  - 1st: Gear tip - Pinion root
  - last: Pinion tip - gear root
Dancing teeth (pinion)

How the pinion moves, seen from the gear

MT 525 Marine Propulsion Systems

Dancing teeth

MT 525 Marine Propulsion Systems
Outside tip circles, there can be no contact
The point of contact travels along the straight line

Contact Travel

Contact zone of various gear ratios, same size crownwheel (LGB)
13-teeth gear ratios are real, 11:39 is for comparison only
Higher gear ratio has smaller pinion and transmits less power
Compare: area of contact zone and length of teeth in contact
13:46 and 11:39 comparison:
same gear ratio and pinion size, less power due to less contact
Contact matters

Gear Design (Contact-2)
Manufacturing (Heat Treatment)

- Carburizing = to enrich the steel with carbon, 0.17 to 1%, to make it hardenable
- But: surface layer only
- hence: case depth
- 'Cook' in carbon shedding gas (methane + natural gas)
- Takes several hours at high temperature (app. 850 °C)
  - An overdosis carbon will create a 'traffic jam' for Carbon travel
  - Carbon must be in solution, not segregated in carbides
  - Avoid burning the steel from the surface inward

MT 525 Marine Propulsion Systems

Power rating (Failure modes)

- Torque rating: what the gears can endure without failure
- One usually refers to a power rating

- Failure modes & Related Safety number
  - Fracture by bending - Sf (fracture)
  - provide massive root section
  - Pitting by sub-surface stress - Sn (Hertz)
  - provide small curvature & case depth
  - Scuffing (welding & tearing) by local high temperature - Sa (scuffing)
    - provide oil dopes & reduce sliding speed
  - Case crushing
    - provide case depth & blend case/core transition
  - wear
    - provide clean oil
  - etc.

MT 525 Marine Propulsion Systems
Power rating (Calculation)

- For each failure mode:
  - a torque limit can be calculated
  - different parameters are determining each limit

- Where parameters have opposed effects, an optimum exists for maximum power
  - example: spiral angle for fracture — pitting optimum

Operational requirements

- Lubrication Oil
  - correct viscosity grade
  - EP-type (i.e. dopes)
  - cooled to stay in condition
  - mineral oil is commonest
  - synthetic oil possible
    - expensive, but lasting much longer
  - bio-degradeable oils
    - just in case

- Acceleration
  - Inertia load can be considerable (E-motor)
  - Rule of thumb: 5-10 sec to run up to full speed
Thruster interaction

FREE JET
(INFINITE FLUID)

MT 625 Marine Propulsion Systems

JET IN PRESENCE OF PLANE SURFACE

MT 625 Marine Propulsion Systems
Thruster interaction

Shape of jet of water behind a working thruster at zero speed (bollard condition)

Line of maximum speed

Line of zero speed

MT 525 Marine Propulsion Systems

Thruster interaction

Maximum jet speeds for thruster in bollard pull condition

$V_{X\text{MAX}}$

$V_{Y\text{MAX}}$

MT 525 Marine Propulsion Systems
Thruster wake in free flow

The Coanda effect

The effect has been discovered in 1930 by a Rumanian aerodynamicist Henri-Marie Coanda (1885 - 1972)

There is friction between water and a surface. This causes for instance the resistance of a ship when moving in the water.
The Coanda effect

Try this, and you will feel that the spoon is strongly pulled towards the water flow.

The water follows the surface of the spoon, an example of the Coanda effect.
Coanda effect

JET IN PRESENCE OF CURVED SURFACE

Interaction with flat plate

Outer boundary of jet hits the plate already around x/D = 1

Flat plate at 0.75D below thruster centre line in bollard condition

MT 525 Marine Propulsion Systems
Interaction with free surface

MT 525 Marine Propulsion Systems

Interaction with barge-shaped hull

Sharp bilge:
- No Coanda effect
- Small jet deflection
**Interaction with barge-shaped hull**

Large bilge radius:
- With increasing distance between thruster and bilge
  - Coanda effect stronger
  - Jet deflection larger

---

**Interaction with barge-shaped hull**

Bilge shape has no relevance for thruster performance if thruster is close to the bilge

---

*MT 525 Marine Propulsion Systems*
Interaction with barge-shaped hull

Close to bilge almost no effect of bilge shape and it is about 0.

For higher thruster angles, further away from bilge and larger bilge radii, it increases up to 15%.

MT 525 Marine Propulsion Systems
Interaction with barge-shaped hull

(from bilge)

Force on opposite hull of twin hull semi sub as a fraction of thrust of thruster on the other hull.

Mutual interaction of thrusters

Effect of mutual distance (bollard condition)

T, Q are values of aft thruster
To and Qo are values of front thruster

Interaction effects in open water persist over a larger distance than under a flat plate.
Mutual interaction of thrusters

Effect of azimuth angle of front thruster (bollard condition)

Interaction dissipates beyond 20 – 30 degrees, unless mutual distance is very small (< 3 – 4D).

Open water

Under a flat plate

T, Q are values of aft thruster
To and Qo are values of front thruster

Effect of (low) speed is very small, especially for larger mutual distances, so for DP speed influence on interaction can practically be neglected.
Thruster interaction

Thruster – hull interaction

- Coanda effect increases with
  - Decreasing velocity in the jet, for instance due to
    - Large distance of thruster to the side of the hull
    - Large azimuth angle (this increases distance the jet has to travel to the side of the hull)
  - Large bilge radius

- Coanda effect can cause up to 15% thrust deduction for a single hull.

- Thrust deduction for twin hull semi-sub can amount up to 75%

MT 525 Marine Propulsion Systems

Thruster interaction

Thruster – thruster interaction

- Can reduce effective thrust of a pair of thrusters by 35%

- Can be noticed more than 20 times the thruster diameter

- Can be avoided by rotating the thrusters over about 30 degrees

MT 525 Marine Propulsion Systems
Thruster interaction

Interaction between two thrusters

MT 525 Marine Propulsion Systems
Thruster interaction

MT 525 Marine Propulsion Systems
Content (outlook to other subjects)

- wk 1 Introduction to marine propulsion systems
- wk 1 Ship types and propulsors
- wk 1,2 Fixed pitch propellers
- wk 2 Controllable pitch propellers
- wk 3 Steerable thrusters and pods
- wk 4 Waterjets
- wk 5 Transverse thrusters
- wk 5 Hydraulic systems and controls
- wk 6 Class requirements, rudder interaction and material aspects
- wk 7 Cost driving factors and system integration

MT525 Marine Propulsion Systems

1. Controllable pitch propellers

Duration: 1.5 hour
Main goal: main working principle and design criteria

Content:
- Why CPP’s?
- Mechanical design and working principle
- Interaction with the engine
  - combinator curves
  - cavitation limits
  - stopping, windmilling and interaction
- Design aspects of CP propellers
  - off design conditions

MT525 Marine Propulsion Systems
Why CPP propellers

CPP propellers are applied to enhance:
- manoeuvring (stopping, steering TS)
- improve overall efficiency (use of shaft generator)
- utilise multiple engines per shaft
- normally in combination with four stroke engines
- four stroke engines have no possibility any more to reverse rotation!!! (not a technical problem)

Main functions in CPP

Two main functions:
- transmit thrust and torque
- change the pitch

This requires a mechanism where the pitch is change via a hydraulic servo system or (old fashioned via a push pull rod system)

The blades have to be suspended in the hub construction: the bearing is the most critical design issue (bearing limits aso)
Figure 6. Blade suspension principles. a = truss type, b = collar type.

MT 525 Marine Propulsion Systems

Figure 7. The truss-type blade suspension leads to the higher forces in the bolt.

MT 525 Marine Propulsion Systems
Figure 8: The hydraulic cylinder can be located inboard (a), or outboard in the hub [b], the latter increasing the overhang moment of the propeller.

Figure 10. The six prime reliability objects in the controllable pitch propeller.
Figure 7: 

Figure 11: LHR type with push-pull rod.
CPP system includes: controls hydraulics shafting and propeller

Pitch mechanism

Figure 11: Pitch mechanism.
Main pitch mechanisms

<table>
<thead>
<tr>
<th>SOME MECHANISMS USED IN CONTROLLABLE PITCH PROPELLERS</th>
</tr>
</thead>
<tbody>
<tr>
<td>PRINCIPLE</td>
</tr>
<tr>
<td>TYPE</td>
</tr>
<tr>
<td>REALIZATION</td>
</tr>
<tr>
<td>NAME</td>
</tr>
<tr>
<td>s=R s=x</td>
</tr>
<tr>
<td>M=F M&gt;0</td>
</tr>
<tr>
<td>FRICTIONLESS CHARACTERISTIC H=H (H=1)</td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems
Overall system

MT 525 Marine Propulsion Systems

Strength aspects

<table>
<thead>
<tr>
<th>Mission profile</th>
<th>Various</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design (nominal)</td>
<td></td>
</tr>
<tr>
<td>off-design</td>
<td></td>
</tr>
<tr>
<td>Severe conditions (Ice)</td>
<td></td>
</tr>
<tr>
<td>Static</td>
<td></td>
</tr>
<tr>
<td>Load factors</td>
<td></td>
</tr>
<tr>
<td>Class reinforcement for blades + equivalent stronger hub</td>
<td>Shock</td>
</tr>
<tr>
<td>Dynamic</td>
<td></td>
</tr>
<tr>
<td>Safety factor</td>
<td></td>
</tr>
<tr>
<td>Lifetime</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Vibrations</td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems
Pyramidal strength

Figure 2. Ice load on propeller blade.
MT 525 Marine Propulsion Systems

Figure 4. Strength comparison diagram.
MT 525 Marine Propulsion Systems
No pyramidal strength

Figure 5. Damage to shifting caused by grounding.

MT 525 Marine Propulsion Systems
Reliability diagram

![Reliability diagram](image)

**Figure 14.** Reliability diagram of a highly parallel-redundant controllable pitch propeller system

MT 525 Marine Propulsion Systems

---

Design improvements

![Design improvements](image)

**Figure 4.** The three stages of post initial design improvement

MT 525 Marine Propulsion Systems
OD box

Figure 8. Schematic oil supply method.

a. Normal situation  b. Incorrectly mounted

Figure 9. Excessive leakage due to deformation under pressure.

MT 525 Marine Propulsion Systems

Figure 10. Arrangement with compensating rings for oil supply to large diameter shafts.

MT 525 Marine Propulsion Systems
Why a small hub is needed

![Diagram showing propeller efficiency with variable RPM (A, B) and constant RPM (C).]

FF versus CPP

![Diagram comparing FF and CPP propeller efficiency.]

Figure 1: Propeller efficiency with variable RPM (A, B) and constant RPM (C).
Spindle torque

\[ C = \frac{M}{S h^2 b^5} \]

Fig. 9—Spindle moment coefficient caused by hydrodynamic forces symmetrical blade outline, no rake
Spindle torque

![Graph showing influence of centrifugal action on spindle moment coefficient](image1)

**Fig. 10—Influence of centrifugal action on spindle moment coefficient**

MT 525 Marine Propulsion Systems

Spindle torque

![Graph showing influence of friction on spindle moment coefficient](image2)

**Fig. 11—Influence of friction on spindle moment coefficient**

MT 525 Marine Propulsion Systems
On a cylindrical section the blade profile changes such that the highest blade spindle torque is at zero pitch.
Spindle torque overview

- Blade spindle torque is normally the highest at zero pitch
- Can limit the size of the hub, affects the basic design of the hub
- Should be tuned to hydraulic system
- Characteristics (dynamic characteristic) depend on blade design
- Skew reduce blade spindle torque
- Note that blade spindle torque per blade rotation is not constant

Some modern hub designs

Check for position of hydraulic cylinder
Check for attachment of the blades
Check for the attachment to the shaft
Check the outside contour (high speed adapted or not)
Check for the number of parts
Check for service friendliness
Check for blade foot sealing
MT 525 Marine Propulsion Systems
Some modern hub designs

Two basic designs
All based on actuating in the hub
Piston either between the blades and hub in one piece
Piston aft requires separate bolted on hub cover (cost, reliability)
Two important strength criteria:
  maximum bearing pressure
  maximum actuating forces
Combinators

Definition

A combinator curve is a pre-set combination of engine/propeller speed and propeller pitch, in order to have an optimum operation of a CP-propeller installation.

The combinator function is included in the Propulsion Control System.

Engine speed and propeller pitch are controlled by one lever on the bridge.

Depending on installation several combinator

---

Combinator

FIG. 4.3. TYPICAL CPP "COMBINATOR"

![Diagram of Combinator](image)

MT 525 Marine Propulsion Systems
Combinators

- Necessary knowledge:
  - propeller open water curve and characteristics
  - propeller performance diagram
  - propeller cavitation limits
  - engine load curves
  - controls and hydraulics
  - effect of environmental conditions
  - design aspects of combinators

Combinator

- Vessel operation profile
  - It is crucial to have the operation profile defined as early as possible in a project
  - This is the only way to make a proper combinator curve design
  - Also, potential problem areas may then be found and solved in an early stage.
Results from open water tests are presented in a diagram giving open water efficiency, and non-dimensional coefficients for thrust $K_T$, and torque $K_Q$ as a function of the advance coefficient $J$.

Open water efficiency can be written as:

$$\eta_0 = \frac{T \cdot V_A}{Q \cdot n \cdot 2 \cdot \pi}$$

Where:

- $T$ is the useful work (Work done by engine)
- $Q$ is the water flow
- $n$ is the engine speed
- $2 \cdot \pi$ is a constant
- $V_A$ is the advance velocity

Definitions:

$$K_T = \frac{T}{\rho \cdot n^2 \cdot D^4}$$

$$K_Q = \frac{Q}{\rho \cdot n^2 \cdot D^5}$$

$$J = \frac{V_A}{n \cdot D}$$
**propeller performance diagram:** Open Water Tests

- Questions
- Show $K_t$ and $K_q$ are dimensionless
- How do open water curves look for other pitch settings
- Which point at bollard pull?
- Which point trailing?

---

**propeller performance diagram**

The best way of determining the propulsive coefficients for a new propeller is to perform both resistance test and a model self-propulsion test with stock propellers.

From these, $t$ and $w$ can be calculated:

- $t$ difference between ship resistance and propeller thrust
- $w$ difference between propeller inflow and ship speed
propeller performance diagram

Self propulsion point of the vessel means:

thrust required to propel the ship is equal to thrust
delivered by the propeller.

Open water is matched to resistance curve

\[
K_T = \frac{T}{\rho \cdot n^2 \cdot D^4}
\]

\[
J = \frac{V_A}{n \cdot D}
\]

\[
K_T / J^2 = \frac{T}{\rho \cdot n^2 \cdot D^4 \cdot (n^2 \cdot D^2 / V_A^2)}
\]

\[
K_T = C \cdot J^2
\]
Due to accelerated flow by the propeller, a low pressure area is formed in the stern, which causes additional resistance (deduction in thrust).

\[
T = R_{th} + t \cdot T
\]

Likewise as with the engine, the propeller has also limits.

These limits are related to noise and vibration.

At high power, too high pressure pulses induce vibration.

At low powers, unsteady pressure side cavitation induces noise and vibration.
Propeller cavitation limits

Unstable pressure side cavitation

Broad band excitation: noise and vibration hindrance

MT 525 Marine Propulsion Systems
Combination with the engines

Constant RPM vs. Combinator mode
- PTO
- manoeuvring
- engine response

Inception of cavitation
- suction side
- pressure side

Propeller cavitation limits
Measured noise level and hull vibrations above propellers

MT 525 Marine Propulsion Systems
Propeller performance diagram

For each pitch ratio a line results:
- power, RPM, shipspeed
- for different values of the pitch a complete power absorption diagram is found with pitch, shipspeed, power, RPM

MT 525 Marine Propulsion Systems
**Propeller performance diagram**

**Questions**
- What is the point of best efficiency given ship speed?
- How do you get to that point?
- Is the curve representative for acceleration?
- Is the curve representative for bollard condition?
- How can we use this curve in view of combinator diagrams?

**Engine load curve**

**Load limits due to:**
- Maximum torque related to maximum fuel
- At reduced RPM amount of air is playing a role (turbo charger)
- The limits are stationary conditions of the engine
- For combinators we have to consider margin!!
Engine load curve

Example W20 engine

Engine load curve

Example W38

Nominal propeller curve

Fig. 5. Operating area for CF-propeller

MT 525 Marine Propulsion Systems
MT 525 Marine Propulsion Systems

**Engine load curve**

**Engine power [kW]**

- 2400 kW
- 3200 kW
- 4000 kW
- 4800 kW
- 5600 kW
- 6400 kW
- 7200 kW
- 8000 kW

**Propeller speed [RPM]**

- 40 RPM
- 60 RPM
- 80 RPM
- 100 RPM
- 120 RPM
- 140 RPM
- 160 RPM
- 180 RPM

- P1 = 0.4
- P1 = 0.6
- P1 = 0.8
- P1 = 1.0

MT 525 Marine Propulsion Systems
Controls and hydraulics

Overall system consists of:

- bridge telegraph handle position
- controls cabinet required pitch and RPM
- hydraulic power pack oil flow
- oil distribution box oil in/pitch feedback
- controllable pitch propeller response on pitch given oil

Schematics of CP propeller

MT 525 Marine Propulsion Systems
Controls and hydraulics

When in combinator control the following other functions may be active:

- **load control** this protects the engine from being overloaded
- **windmilling** this protects the engine from being underloaded at reduction of pitch
- **running up/down** this takes care of the slow increase or decrease of power when changing the handle
- **load sharing** makes sure that load of 2 engines per shaft is shared

MT 525 Marine Propulsion Systems
**Combinator Curves**

**Design principles 1**
- Design of an "optimum" combinator curve is a compromise between engine- and propeller related issues:
  - Load / torque limitations for the engine
  - Lowest specific fuel consumption for the engine
  - Highest propeller efficiency
  - Lowest pressure impulses and noise from propeller cavitation
  - Highest thrust response during manoeuvring

**Combinator Curves**

**Design principles 2**
- The combinator curve should be positioned with some margin under the load limit curve for the engine
- It is recommended to have a separate combinator curve for manoeuvring
- High propeller speed in combination with low propeller pitch should be avoided, as there is a risk for pressure side cavitation. Also, the propeller efficiency will be low in this condition.
Combinator Curves

Effect of environmental condition 10% resistance increase

Fig. 2. Effect of resistance increase on optimum propeller curve
Combinator Curves

Effect of shallow water

Fig. 4. Influence of water depth on speed-power relation.

Combinator Curves

Typical ferry crossing

MT 525 Marine Propulsion Systems
Modes of operation

- Based on the mission profile:
  - freesailing
  - maneuvering
  - towing
  - environmental conditions like seastate
  - shallow water
  - draft of the vessel

MT 525 Marine Propulsion Systems
Modes of operation

Based on engine configuration:
- number of engines per shaft
- number per propellers per ship

- for each possibility one can consider constant RPM or variable RPM
- for manoeuvring consider the bollard ahead and astern conditions and base the selection of the pitch on those conditions

Modes of operation

Example: twin propellers twin engine
- transit I: 2 engines per shaft variable RPM
- transit II: 2 engines per shaft constant RPM
- Maneuvering I: 2 engines per shaft constant RPM
- Maneuvering II: 1 engine per shaft variable RPM

limitations: engine power modes possible in the controls system
- in theory many modes (maybe 14 are possible) in practice not all of them are being used
- restriction is wise as every mode need to be tested during trials

MT 525 Marine Propulsion Systems
Example

1. Review the power absorption diagram
2. Indicate for 26 kn the point of best efficiency
3. Indicate at 20 knots the point of best efficiency
4. Draw a proposal for a combinator curve with one engine
5. Draw a proposal for a combinator with 2 engines

Example

1. Review the power absorption diagram including the combinator line
2. Why is the pitch at 26 knots lower then answered in previous slide?
3. Check the pitch and power difference between 26 knots and hp 10
4. Draw the combinator curve HP versus pitch and RPM

MT 525 Marine Propulsion Systems
Example

- Review the power absorption diagram including the combinator line including the cavitation limits
- Confirm the pitch at 26 knots
- The ship wants to cruise at 20 knots. What is your recommendation: one or two engines per shaft?
Example: twin screw ferry

- Power absorption
- twin screw ferry
- part load 1 engine per shaft
- FREE SAILING AHEAD at VAR RPM
  Wärtsilä Engine type 46C
Example: twin screw ferry

- Power absorption
- twin screw ferry
- 2 engines per shaft
- FREE SAILING AHEAD at VAR RPM
- Wartsila engine type 46 C

MT 525 Marine Propulsion Systems
Example: twin screw ferry

- Constant RPM vs. Combinator mode
- PTO
- Manoeuvring
- Engine response

Inception of cavitation

Power absorption diagram

Fast ferry

Inception suction side cavitation

Engine power per shaftline [kW]

Inception free cav.
Example: dredger

- Wide variation in operational profile
- Large variation due to pump loading (significant part of the system)

MT 525 Marine Propulsion Systems
Example: dredger

- Power absorption
- Twin screw dredger
- Modes of operation at const rpm

Cavitation inception and calculated pitch vs ship speed for a Twin Screw Dredger.
Example: dredger

MT 525 Marine Propulsion Systems

Example: dredger

MT 525 Marine Propulsion Systems
Example: dredger

Example: Fast ferry

Fast ferry

2 CPP for normal cruising
one waterjet to be used as booster

MT 525 Marine Propulsion Systems
MT 525 Marine Propulsion Systems
Combinator Curve in
Power absorption diagram
Final Propeller Design
Based on model tests HSFA for T = 6.20 m

constant ship speed: 
constant pitch angle:
back side cavitation
face side cavitation inception:
face cavitation inception with increased risk:
combinator curve

Total Engine Power [kW]
Engine Power per shaftline [kW]
propeller speed [rpm]
Combinator Curve in
Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

constant ship speed:
constant pitch prop.
back side cavitation
face side cavitation inception:
face cavitation inception with increased risk:
combinator curve

Total Engine Power [kW]
Engine Power per shaftline [kW]
propeller speed [rpm]
Combinator Curve in
Power absorption diagram
Final Propeller Design

Based on modeltests HSVA for $T = 6.20 \ m$

- Constant ship speed: 
- Constant pitch
- Back side cavitation
- Face side cavitation
- Inception:
- Face cavitation inception with increased risk:
- Combinator curve

**Graph Details:**
- Horizontal axis: Propeller speed [rpm]
- Vertical axis: Total Engine Power [kW]
- Engine Power per shaftline [kW]

- Marked points and lines indicate specific speeds and powers for different HP levels.

**Notes:**
- PS 15 kW (40 RPM)
- 30 kW (20 RPM)
Combinator Curve in Power absorption diagram
Final Propeller Design
Based on model tests HSLA for T = 6.20 m

constant ship speed: 
constant pitch angle:
back side cavitation:
face side cavitation inception:
face cavitation inception with increased risk:
combinator curve

Total Engine Power [kW]
Engine Power per shaftline [kW]

propeller speed [rpm]
Combinator Curve in
Power absorption diagram
Final Propeller Design
Based on model tests HSVA for T = 6.20 m

Constant ship speed:
Constant pitch angle:
Back side cavitation:
Face side cavitation inception:
Face cavitation inception with increased risk:
Combinator curve:

Propeller speed [rpm]
Combinator Curve in Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

constant ship speed: ---
constant pitch angle

back side cavitation
face side cavitation inception: -----

face cavitation inception with increased risk: -----
combinator curve

Total Engine Power [kW]
Engine Power per shaftline [kW]

45000
40000
35000
30000
25000
20000
15000
10000
5000

28 kn
26 kn
24 kn
23 kn
22 kn
21 kn
20 kn
18 kn
16 kn

110 rpm
115 rpm
120 rpm
125 rpm
130 rpm
135 rpm
140 rpm
145 rpm
150 rpm

0
5000
10000
15000
20000
25000
50000
Combinator Curve in
Power absorption diagram
Final Propeller Design

Based on modeltests HSV A for T = 6.20 m

propeller speed [rpm]

Total Engine Power [kW]

Engine Power per shaftline [kW]

constant ship speed:

constant pitch

back side
cavitation

face side cavitation
inception:

face cavitation inception
with increased risk:

combinator curve

119 rpm
134 rpm
140 rpm

HP 10
27
31

HP 9
24
26

HP 8
22
26

HP 7
21
25

HP 6
18
21

HP 5
16
18

HP 4
15
10

HP 3
10
0

HP 2
0
0

HP 1
0
0

PS 15 kW 140 RPM

PS 10 kW 120 RPM
Combinator Curve in Power absorption diagram
Final Propeller Design

Based on model tests HSVA for r = 6.20 m

- Constant ship speed:
- Constant pitch angle:
- Backside cavitation:
- Face side cavitation inception:
- Face cavitation inception with increased risk:
- Combinator curve:

Propeller speed [rpm]

Total Engine Power [kW]

Engine Power per shaftline [kW]

PS 15 MW 140 rpm
CM 9 MW 120 rpm
Combinator Curve in Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

constant ship speed: 
constant pitch 
back side cavitation 
face side cavitation inception: 
face cavitation inception with increased risk: 
combinator curve

propeller speed [rpm]
0 100 105 110 115 120 125 130 135 140 145 150

Total Engine Power [kW]
0 5000 10000 15000 20000 25000

Engine Power per shaftline [kW]
0 5000 10000 15000

119 rpm: 28 kn
134 rpm: 25 kn
140 rpm: 22 kn
28 kn:
24 kn:
23 kn:
21 kn:
20 kn:
18 kn:
16 kn:
15 kn:
14 kn:
13 kn:
12 kn:
11 kn:
10 kn:
9 kn:
8 kn:
7 kn:
6 kn:
5 kn:
4 kn:
3 kn:
2 kn:
1 kn:
0 kn:

PS 15 MW 140 RPM
30 MW 120 RPM
Combinator Curve in Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

![Diagram showing propeller performance with various speed and power levels.](image)

- Constant ship speed
- Constant pitch
- Back side cavitation
- Face side cavitation inception
- Face cavitation inception with increased risk
- Combinator curve

**Engine Power per shaftline [kW]**

- 50000
- 45000
- 40000
- 35000
- 30000
- 25000
- 20000
- 15000
- 10000
- 5000

**Total Engine Power [kW]**

- 50000
- 45000
- 40000
- 35000
- 30000
- 25000
- 20000
- 15000
- 10000
- 5000

**Propeller speed [rpm]**

- 110 rpm
- 130 rpm
- 150 rpm

**Engine Power per shaftline [kW]**

- 28 kn
- 26 kn
- 25 kn
- 24 kn
- 23 kn
- 22 kn
- 21 kn
- 18 kn
- 16 kn

**Propeller speed [rpm]**

- 110 rpm
- 130 rpm
- 150 rpm

**Notes:**
- PS 15 MW 140 RPM
- COM 120 RPM
Combinator Curve in Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

constant ship speed: 
constant pitch angle: 
back side cavitation: 
face side cavitation inception: 
face cavitation inception with increased risk: 
combinator curve

propeller speed [rpm]

Total Engine Power [kW]

Engine Power per shaftline [kW]
Combinator Curve in Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for $T = 6.20$ m

constant ship speed:

constant pitch angle:

back side cavitation

face side cavitation inception:

face cavitation inception with increased risk:

combinator curve

constant speed:

constant pitch angle:

back side cavitation

face side cavitation inception:

face cavitation inception with increased risk:

combinator curve

Total Engine Power [kW]

Engine Power per shaftline [kW]

propeller speed [rpm]
Combinator Curve in
Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

119 rpm  134 rpm  140 rpm

Total Engine Power [kW]
Engine Power per shafttime [kW]
propeller speed [rpm]

constant ship speed:
constant pitch

back side cavitation
face side cavitation inception:
face cavitation inception with increased risk:
combinator curve

constant shipspeed:
constant pitch

back side cavitation
face side cavitation inception:
face cavitation inception with increased risk:
combinator curve

100 105 110 115 120 125 130 135 140 145 150

0 5000 10000 15000 20000 25000 30000 35000 40000 45000 50000

15 kn 18 kn 21 kn 23 kn 25 kn 31 kn 34 kn 35 kn

PS 15 MW 140 RPM
PS 10 MW 120 RPM
Combinator Curve in
Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

constant ship speed: 
constant pitch angle: 
back side cavitation: 
face side cavitation inception: 
face cavitation inception with increased risk: 
combinator curve: 

Total Engine Power [kW]
Engine Power per shaftline [kW]

propeller speed [rpm]
Combinator Curve in Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

constant ship speed:
constant pitch angle:
back side cavitation:
face side cavitation inception:
face cavitation inception with increased risk:
combinator curve:

Total Engine Power [kW]
Engine Power per shaftline [kW]
propeller speed [rpm]

constant ship speed:
constant pitch angle:
back side cavitation:
face side cavitation inception:
face cavitation inception with increased risk:
combinator curve:

119 rpm 134 rpm 140 rpm
HP 0 HP 1 HP 2 HP 3 HP 4 HP 5 HP 6 HP 7 HP 8 HP 9 HP 10
0 5000 10000 15000 20000 25000 30000 35000 40000 45000 50000
0 5000 10000 15000 20000 25000 30000 35000 40000 45000 50000

100 105 110 115 120 125 130 135 140 145 150

15 MW 140 rpm
10 MW 120 rpm

constitutive speed:
constant pitch;
thrust ala.
back side;
final face side cavitation
TDH000000326
Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 6.20 m

constant ship speed: 
constant pitch angle:

Propeller speed [rpm]

Engine Power per shaftline [kW]

Total Engine Power [kW]
TDH000000326
Power absorption diagram
Final Propeller Design
Based on model tests HSVA for T = 6.20 m
constant ship speed:

constant pitch
Power absorption diagram
Final Propeller Design
Based on model tests HSVA for T = 6.20 m

Constant ship speed: ---
Constant pitch angle: ---

propeller speed [rpm]
Power absorption diagram
Final Propeller Design
Based on model tests HSVA for $T = 6.20$ m

Propeller speed [rpm]

Engine Power per shaftline [kW]

Total Engine Power [kW]

constant ship speed: 
constant pitch 

$119 \text{ rpm}$ $134 \text{ rpm}$ $140 \text{ rpm}$

$28 \text{ kn}$

$110$ $115$ $120$ $125$ $130$ $135$ $140$ $145$ $150$

$0$ $5000$ $10000$ $15000$ $20000$ $25000$

$0$ $5000$ $10000$ $15000$ $20000$ $25000$
TDH000000326
Power absorption diagram
Final Propeller Design
Based on modeltests HSVA for T = 8.20 m

constant ship speed: _________________________
constant pitch angle

119 rpm 134 rpm 140 rpm

propeller speed [rpm]

Total Engine Power [kW]

Engine Power per shaftline [kW]
Content (outlook to other subjects)

Content
wk 1 Introduction to marine propulsion systems
wk 1 Ship types and propulsors
wk 1,2 Fixed pitch propellers
wk 2 Controllable pitch propellers
wk 3 Steerable thrusters and pods
wk 4 Waterjets
wk 5 Transverse thrusters
wk 5 Hydraulic systems and controls
wk 6 Class requirements, rudder interaction and material aspects
wk 7 Cost driving factors and system integration

MT525 Marine Propulsion Systems

3. Fixed pitch propellers

Duration: 1.0 hour
Main goal: main working principle and design criteria
Content:
Design requirements
Main parameters on propeller efficiency
Propeller design aspects
Interaction with the engine
  design point
  stopping: robinson curves
Mechanical design
  keyless hub
  shaft forces off design

MT525 Marine Propulsion Systems
Market Overview

Between 1997 and 2002 over 1600 vessels above 4000 grt were built with an FPP.
Application

- FPP are used for:
  - containers 35%
  - TANKERS 41%
  - BULKCARRIERS 20%
  - DRY CARGO VESSELS 4%

The design process

- Design criteria depend on shiptype
- Tankers and bulkcarriers
- Container vessels
tankers and bulkcarriers

Requirement by owner:
- no broken blades
- power/RPM correct
- high efficiency
- no vibration
- erosion
- dock costs
- engine repairs
- fuel costs
- complaints
- propeller damage

MT 525 Marine Propulsion Systems

tankers and bulkcarriers

Requirement by yard:
- price
- delivery time
- efficiency
- no vibration
- other aspects
- contract speed
- contract
- owner complaints

MT 525 Marine Propulsion Systems
Design aspects for tankers and bulkcarriers

- Efficiency
  - depends on choice of diameter
  - match power and RPM
- Cavitation behaviour
  - maximum power in ballast condition
  - no erosion
- Pressure pulses
  - large clearance thus not critical

MT 525 Marine Propulsion Systems

Design aspects for fast container ships

- Efficiency
  - high speed thus large fuel consumption
- Cavitation behaviour
  - high power on limited propeller diameter
- Pressure pulses
  - low pressure pulses to avoid vibration
  (largest risk !!)

MT 525 Marine Propulsion Systems
fast container ships

- Requirements by yard
  - price
  - delivery time
  - high efficiency
  - no vibration
  - other aspects

MT 525 Marine Propulsion Systems

- Requirements by owner
  - no broken blades
  - power/RPM correct
  - high efficiency
  - no vibration
  - erosion/damage

- dock costs
- engine repairs
- fuel costs
- complaints
- propeller repair

MT 525 Marine Propulsion Systems
The design process

- Requirements for
  - efficiency $\rightarrow$ shipspeed or fuel costs
  - cavitation $\rightarrow$ noise
  - pressure pulses $\rightarrow$ vibration

Choice of propeller diameter

- Select maximum possible in view of tip clearance
- Match propeller diameter and RPM of main engine
Actuator disk

Propeller disk

Figure A1, Definition of axial and tangential velocities

Deviations from ideal efficiency

- **Axial losses** function of $C_t$ or axial induced velocity squared
  \[ \eta = \frac{2}{1 + \sqrt{1 + C_t}} \]
  \[ \frac{V_a}{V} = \sqrt{1 + C_t} - 1 \]
  \[ C_t = \frac{7 \frac{D^2}{2} \rho v^2}{4 \rho v^2} \]

- **Rotational losses** function of $C_q$ or tangential velocity squared
- **Frictional losses** function of BAR and tipspeed squared
- **Finite blade number**
- **Non optimum design**

MT 525 Marine Propulsion Systems
In case the blade friction is zero, the power supplied to the propeller is the difference between the kinetic energy aft and ahead of the propeller, or:

\[ P = \frac{1}{2} \rho \pi r (V + \frac{Va}{2}) \times \left[ (V + Va)^2 + Vt^2 \right] dr \]

which can be written as:

\[ P = \frac{1}{2} \rho \pi r (V + \frac{Va}{2}) \times \left[ 2 VaV + Va^2 + Vt^2 \right] \]
Equation A.3 can then be written as:

\[
\eta_i = \frac{\int_{\rho}^{R} 2 \pi r \left( V + \frac{V_a}{2} \right) V_a V dr}{\int_{O}^{\rho} 2 \pi r \left( V + \frac{V_a}{2} \right) (V_a V + \frac{V_a^2}{2} + \frac{V_t^2}{2}) dr}
\]

It can be concluded that the tangential losses are related to:

\[
E_t = \int_{O}^{\rho} 2 \pi r \left( V + \frac{V_a}{2} \right) \frac{V_t^2}{2} dr
\]

and the axial losses related to:

\[
E_a = \int_{O}^{\rho} 2 \pi r \left( V + \frac{V_a}{2} \right) \frac{V_a^2}{2} dr
\]

The difference in momentum then becomes:

\[
T = \int_{O}^{\rho} 2 \pi r \left( V + \frac{V_a}{2} \right) V_a dr
\]

The efficiency of a propeller can be defined as the ratio of the effective power delivered by the propeller divided by the power supplied to the propeller. Neglecting blade friction this is:

\[
\eta_i = \frac{T.V}{P}
\]

where \( T.V \) = effective power delivered by the propeller,

\( P \) = power supplied to the propeller.
Given diameter and thrust
Find opt RPM

\[ 0.8 \, \text{Kt/J}^2 \]

\( u \) - C

\( 0.6 - u \)

\( 0.5 - r_1 \text{ICT} \)

Friction - QL

0.8

0.5

0.2

0.1

0.0

0.2

0.4

0.6

0.8

1.0

MT 525 Marine Propulsion Systems

Fig. 2. Rotational and frictional losses as function of the advance ratio

Ideal vs. true efficiency (impulse theory)

Axial losses \( \eta_1 \)

Friction losses

Rotational losses

MT 525 Marine Propulsion Systems
Fig. 3. Efficiency derived from B-series propeller

MT 625 Marine Propulsion Systems
Propeller efficiency

<table>
<thead>
<tr>
<th>Ct</th>
<th>axial losses %</th>
<th>rotational losses %</th>
<th>drag %</th>
<th>efficiency %</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>16</td>
<td>7</td>
<td>16</td>
<td>61</td>
</tr>
<tr>
<td>1.5</td>
<td>23</td>
<td>6</td>
<td>14</td>
<td>57</td>
</tr>
<tr>
<td>3.5</td>
<td>32</td>
<td>5</td>
<td>13</td>
<td>50</td>
</tr>
<tr>
<td>6</td>
<td>41</td>
<td>7</td>
<td>11</td>
<td>41</td>
</tr>
</tbody>
</table>

Ct: thrust loading coefficient

Increasing propeller diameter leads to lower hull efficiency.
Propulsive efficiency

Suction force

Propeller thrust \( T \)
Thrust deduction factor \( t \)
Suction force \( S \)

\[ T = R_{TS} + t \cdot T \]

More loading at the tip increases thrust deduction

MT 525 Marine Propulsion Systems

Propulsive efficiency

Propeller efficiency \( \eta_P = \frac{P_T}{P_D} = \frac{T \cdot V_A}{P_D} \)

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Cruiseline</td>
</tr>
<tr>
<td></td>
<td>Product tanker</td>
</tr>
<tr>
<td></td>
<td>Boliard pull</td>
</tr>
</tbody>
</table>

- Propeller thrust \( T \) [kN]
- Ship speed \( V_S \) [m/s]
- Wake fraction \( W_T \)
- Advance speed \( V_A = V_S(1-W_T) \) [m/s]
- Thrust power \( P_T = T \cdot V_A \)
- Brake power \( P_B \) [kW]
- Mechanical efficiency \( \eta_M \)
- Delivered power \( P_D = P_B \cdot \eta_M \) [kW]

MT 525 Marine Propulsion Systems
Propulsive efficiency

Total efficiency
\[ \eta_D = \frac{P_E}{PD} = \frac{T \cdot V_A}{PD} \cdot \frac{(1-t)/(1-WT)} = \eta_P \cdot \eta_H \]

Cruiseline: 0.7
Product tanker: 0.7

High thrust, \( T \) => High \( \eta_P \), Low suction force, i.e., low \( t \) => High \( \eta_H \)
High advance speed, \( V_A \) => High \( \eta_P \), Low advance speed, i.e., high \( WT \) => High \( \eta_H \)

(High advance speed, \( V_A \) => Low \( T \))

- Effective power: \( P_E = R_{TS} \cdot V_B \) [kW]
- Ship resistance: \( R_{TS} \) [kN]
- Ship speed: \( V_B \) [m/s]
- Delivered power: \( P_D = P_E \cdot \eta_M \) [kW]
- Thrust deduction: \( t \)
- Wake fraction: \( WT \)
- Hull efficiency: \( \eta_H = \frac{(1-t)/(1-WT)} \)

Examples

<table>
<thead>
<tr>
<th>Cm. no.</th>
<th>Cases</th>
<th>Na.</th>
<th>0.01</th>
<th>0.00</th>
<th>0.001</th>
<th>0.1</th>
<th>0.01</th>
<th>0.01</th>
<th>10</th>
<th>100</th>
<th>1000</th>
<th>10000</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>70</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>71.5</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>72</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>70</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>75</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>80</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>7</td>
<td>7</td>
<td>85</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
<td>90</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>9</td>
<td>9</td>
<td>95</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
<tr>
<td>10</td>
<td>10</td>
<td>100</td>
<td>0.01</td>
<td>0.00</td>
<td>0.001</td>
<td>0.1</td>
<td>0.01</td>
<td>0.01</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems
Effect of diameter on efficiency

Take account of effect on hull efficiency

Gain in open water is reduced due to smaller wake fraction and less hull efficiency

Basically 0.5 to 0.66 of the open water improvement is found in the total efficiency (based on experience with model tests)
Effect of diameter on wake and thrust deduction

Propeller diameter

MT 525 Marine Propulsion Systems

Effect of diameter on open water and total efficiency

Propeller diameter

MT 525 Marine Propulsion Systems
Effect of diameter on pressure pulse

MT 525 Marine Propulsion Systems

Clearances

SG040403

Propeller clearances acc. to DnV
Single screw

A = min. 0.12xD
generally used 0.16...0.2xD

C = min. 0.2xD

MT 525 Marine Propulsion Systems
Clearances

Propeller clearances acc. to DnV
Twin screw

\[ C = \text{min} \ 0.26D \]

Cavitation - Practical Considerations

- Wake
  - The influence from the wake on the pressure impulses is principally depending on:
    - Maximum wake peak
    - The wake gradients (how rapid the wake changes as a function of blade positions)
  - There are generally large differences in wake between different hull forms. Only minor modifications of the lines may have a big impact on the wake field
Example of change in wake field from modification of hull lines

Original lines

Max wake 0.7

Modified hull lines

Cavitation - Practical Considerations

* Dotted lines: Modified hull form

MT 526 Marine Propulsion Systems
Axial wake as a function of blade position

Cavitation - Practical Considerations

- Influence from skew, maximum wake field and tip clearance on pressure impulses
Cavitation - Practical Considerations

Load distribution
By changing pitch at blade tip, the load can be distributed away from the hull and pressure impulses are reduced.

<table>
<thead>
<tr>
<th>Cruise ship - load dis.</th>
<th>Elliptical load dis.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
</tr>
<tr>
<td>0.9</td>
<td></td>
</tr>
<tr>
<td>0.8</td>
<td></td>
</tr>
<tr>
<td>0.7</td>
<td></td>
</tr>
<tr>
<td>0.6</td>
<td></td>
</tr>
<tr>
<td>0.5</td>
<td></td>
</tr>
<tr>
<td>0.4</td>
<td></td>
</tr>
<tr>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>0.2</td>
<td></td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems

Case study

<table>
<thead>
<tr>
<th>Length waterline (m)</th>
<th>Draft (min) (m)</th>
<th>Displacement (t)</th>
</tr>
</thead>
<tbody>
<tr>
<td>195.0</td>
<td>6.0</td>
<td>10,073</td>
</tr>
</tbody>
</table>

Blade coefficient: 0.828

Table 1.1

<table>
<thead>
<tr>
<th>Propeller no.</th>
<th>Angle (deg)</th>
<th>Suction distribution</th>
<th>Discharge distribution</th>
</tr>
</thead>
<tbody>
<tr>
<td>P-1032</td>
<td>3.45</td>
<td>Naca 0012</td>
<td>Naca 16 elliptical 36°</td>
</tr>
<tr>
<td>P-1031</td>
<td>3.50</td>
<td>Naca 0012</td>
<td>Naca 16 elliptical 36°</td>
</tr>
<tr>
<td>P-1033</td>
<td>3.50</td>
<td>Naca 0012</td>
<td>Naca 16 elliptical 36°</td>
</tr>
<tr>
<td>P-1034</td>
<td>3.50</td>
<td>Naca 0012</td>
<td>Naca 16 elliptical 36°</td>
</tr>
</tbody>
</table>

Comprehensive propulsion efficiencies related to actual propeller – P-1002 – at same speed – thrust.

<table>
<thead>
<tr>
<th>Propeller no.</th>
<th>Compressible</th>
<th>Open water</th>
<th>Open water</th>
<th>Open water</th>
<th>Open water</th>
<th>Open water</th>
<th>Open water</th>
<th>Open water</th>
<th>Open water</th>
</tr>
</thead>
<tbody>
<tr>
<td>P-1031</td>
<td>6.5</td>
<td>2.6</td>
<td>3.2</td>
<td>2.8</td>
<td>4.2</td>
<td>2.2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>P-1032</td>
<td>6.4</td>
<td>2.7</td>
<td>3.3</td>
<td>2.8</td>
<td>4.4</td>
<td>2.4</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>P-1033</td>
<td>6.2</td>
<td>2.5</td>
<td>3.1</td>
<td>3.2</td>
<td>4.6</td>
<td>3.2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>P-1034</td>
<td>6.0</td>
<td>2.4</td>
<td>2.9</td>
<td>2.4</td>
<td>4.8</td>
<td>3.2</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems
Case study

Figure 2.5. Pitch distribution.

MT 525 Marine Propulsion Systems

Cavitation behaviour

Figure 4.4. Extent of cavitation φ = 0°.

Figure 4.5. Extent of cavitation φ = 220°.

25 Marine Propulsion
Case study

MT 525 Marine Propulsion Systems
Figure 27. Relation between axial wake variation and half angle of flow line at $0.8 \cdot R$.

MT 525 Marine Propulsion Systems

$$T + 0.61 \left[ N \cdot D^3 \cdot V_s \left( \frac{a_{0.8R} + 29}{83} \right) \right]$$

$$D.I. = \frac{T + 0.61 \left[ N \cdot D^3 \cdot V_s \left( \frac{a_{0.8R} + 29}{83} \right) \right]}{(h + 10) \cdot D^2}$$

in which

- $D.I.$ = difficulty index
- $T$ = thrust in kg
- $N$ = number of revolutions per minute
- $D$ = diameter in m
- $V_s$ = ship's speed in kn
- $a_{0.8R}$ = half angle of flow line at $0.8 \cdot R$ in 12 o'clock position
- $h$ = water column above propeller tip in m.

MT 525 Marine Propulsion Systems
Cavitation and pressure fluctuations

The pressure field induced by a cavitating sheet is:

\[ P = \frac{\rho \omega^2}{4\pi d^*} \frac{\delta^2 V_c}{\delta^2 \gamma} \]

In other words:
second derivative of the cavitation volume
dependent on tip clearance

Statistical pressure pulse prediction

Holdens method
- Very practical way of reviewing pressure pulse levels on main parameter in early stage
- Based on statistical analysis of measurements carried out in the early 70's
- Statistics should be verified on recent cases but basic formulation is straightforward

- Limitation are from old data set
- Basic setup ok, but to be correlated with new data sets

Cavitating pressure pulse = constant \* effect of wake peak/ sqrt cavitation number/ tip clearance

MT 525 Marine Propulsion Systems
Cavitation - Practical Considerations

Minimizing noise / vibrations problems

1. Improve wake-field
   - Improve hull lines
   - Wake improving devices
2. Increase propeller-hull clearance
3. Avoid resonances
   - Analyse local and global hull stiffness
   - Analyse excitation sources
4. Insulation
   - Engine installations
   - Compartments
5. Optimize propeller
   - Propeller speed
   - Skew & Rake
   - Load distribution
   - Blade area & Chord length distribution
   - Profile type & Thickness
   - Model tests

MT 525 Marine Propulsion Systems
Engine and propeller torque

- Propeller law

- (fixed pitch operation)

"The power absorbed by the propeller is proportional to the shaft speed cubed - within the ship’s speed range in which the power is proportional with the vessel speed cubed."

\[ P_2 = \left( \frac{n_2}{n_1} \right)^3 \cdot P_1 \]

MT 525 Marine Propulsion Systems

---

Engine and propeller torque

- Propeller law

- In case the vessel resistance curve is steeper - or less steep than \( R = \text{const} \, V^2 \), the propeller law does not apply

- In spite of this, the propeller law is widely used to estimate the relation between engine power and propeller rpm at constant pitch

MT 525 Marine Propulsion Systems
Engine and propeller torque

- Fixed pitch propellers

- The torque and power absorbed by a FP-propeller will only depend on the rpm and advance speed (relative water speed into the propeller disc)

- For a given rpm the power absorbed will therefore depend on:
  - The loading condition of the vessel
  - Added resistance from weather and fouling of hull
  - Free running or towing/bollard pull conditions

Design point

- Definitions:
  - sea margin: % power increase given ship speed
  - power margin: % difference between required power and MCR
  - light running margin: how light is the propeller for given conditions (trial or service) and in most cases MCR
Lay out fields

MT 525 Marine Propulsion Systems
Lay out fields PTO

Engine and propeller torque

"Traditional" propeller law and engine limit curve
Engine and propeller torque

Propeller law for bollard pull - Free running design

Engine and propeller torque

Propeller law at free running - Bollard pull design
Engine and propeller torque

FP-propellers for twin screw vessels are generally not recommended.

1. With only one propeller running, the engine will be overloaded.
2. And if we try to design the propellers so that running one engine is possible, it will not be possible to utilize the full power with both engines running.
Stopping of the propeller

- Control of RPM via fuel rack
- Reduce RPM by reducing fuel rack
- This will get problematic in conditions were the propeller is not absorbing power
- Too rapid reduction of RPM will generate negative torque which the engine can not absorb
- For that purpose Robinson curves are made interpolated from four kwadrant measurements (function beta)
- Robinson curve present the instantaneous torque or thrust given shipspeed and RPM (given beta)
- Horizontal always plot RPM so that easy comparison with engine limits can be made
- Apart engine load limits there may also be other limits such as thrust bearing limits!

MT 525 Marine Propulsion Systems
Theoretical Thrust Curves

**Kelyless connection**

- Shrink fit between shaft and hub
- no key
- hydraulically operated, pressure increase results allow hub to be moved up
- distance controls torque and safety factor
- limits are: material of the hub
- temperature effect of fit due to fact that hub bronze elongation due to temperature is larger than that of steel shaft
- only for smaller propellers other connections are applied

MT 525 Marine Propulsion Systems
Keyless connection

- More secure due to accurate calculations
- No weakening of hub and shaft due to keyway
- Controlled hydraulic mounting and dismounting
- No heat applied during mounting process
- Fit to be checked

Fig. 14 Section through a typical propeller boss showing the arrangement for keyless oil injection fitting to the shaft

MT 525 Marine Propulsion Systems
\[ p_v = \frac{M_m}{A_v} \text{ [N/mm²]} \]

\[ F = \mu F_w \]

\[ T = \sqrt{T_T^2 + T_{TV}^2} \]

\[ M_T = M_m - (1+TV) \]

\[ F_{max} = S \cdot F \]

\[ p_v = \frac{F_{max}}{\mu} \cdot D_{eqm} \]

\[ S \cdot F' = 1000 \cdot \mu \cdot p_v \cdot D_{eqm} \cdot \ell \]

\[ p_v = \frac{S \cdot F' + 1000}{\mu \cdot D_{eqm} \cdot \ell} \]

MT 625 Marine Propulsion Systems
Vlaktedruk wordt verkregen door opperflaank:

\[ \sigma_{eq} = \sqrt{\sigma_1^2 + \sigma_0^2 + \sigma_T^2} \]

Relatie:

\[ \frac{\sigma_0}{\sigma_1} = k \]

Spanningen:

Maximum spanningen (\( \sigma_{eq} \)) treden op bij:

- Naaf: \( \rightarrow \) aan binnenzijde
- As: hol: \( \rightarrow \) aan binnenzijde
- Massief: \( \rightarrow \) aan buitenzijde

Volgens keur:

\[ \sigma_{eq} = f \cdot R_p \text{ as} \]

\[ \sigma_{eq} = f \cdot R_p \text{ naaf} \]

opm.: KEYLESS

Verschil in temperatuuruitzettingscoëfficiënt [m/m/°C] tussen as- en naafmateriaal

- \( \Delta \text{AS} \) versus \( \Delta \text{NAAF} \)

Naaf zet sterker uit bij hogere temperaturen:

- 35°C \( \rightarrow \) \( p \) \( \rightarrow \) minimum over te brengen koppel
- 0°C \( \rightarrow \) \( p \) \( \rightarrow \) grootste materiaalspanningen

arbitraire grenzen, door keur bepaald
Keyless Fitting

5.15.2(a) Design criteria. The factor of safety against slip of the propeller hub on the tail shaft taper at 35°C (95°F) is to be at least 2.8 under the action of maximum continuous ahead rated torque plus torque due to torsional vibrations. See 6-1-1/33.7 for propellers requiring ice strengthening. For oil injection method of fit, the coefficient of friction is to be taken no greater than 0.13 for bronze/bronze propeller hubs on steel shafts. The maximum equivalent uniaxial stress (von Mises-Hencky criterion) in the hub at 0°C (32°F) is not to exceed 70% of the minimum specified yield stress or 0.2% proof stress of the propeller material.

Stress calculations and fitting instructions are to be submitted (see 4-3-3/1.5.4) and are to include at least the following:

- Theoretical contact surface area;
- The maximum permissible pull-up length at 0°C (32°F) as limited by the maximum permissible uniaxial stress specified above;
- The minimum pull-up length and contact pressure at 35°C (95°F) to attain a safety factor against slip of 2.8;
- The proposed pull-up length and contact pressure at fitting temperature; and
- The rated propeller ahead thrust.

MT 525 Marine Propulsion Systems

Theoretical Contact Surface Between Hub and Shaft

MT 525 Marine Propulsion Systems
5.15.2(c) Equations. The taper on the tail shaft cone is not to exceed 1/15. Although the equations given below are for ahead operation, they may be considered to provide an adequate safety margin for astern operation also.

The minimum mating surface pressure at 35°C (95°F), \( P_{35} \), is to be:

\[
P_{35} = \frac{ST}{AB} \left[ 90 + \frac{1}{2} \left( \frac{F_r}{T} \right) \right] \text{N/mm}^2 (\text{kgf/mm}^2, \text{psi})
\]

The rated propeller thrust, \( T \), submitted by the designer is to be used in these calculations. In the event that this is not submitted, one of the equations in 4-3-3/Table 3 may be used, subject to whichever yields the larger value of \( P_{35} \).

The shear force at interface, \( F_r \), is given by

\[
F_r = \frac{2TR}{D_2}
\]

Constant \( B \) is given by:

\[
B = \mu^2 + 20^2
\]

The corresponding [i.e., at 35°C (95°F)] minimum pull-up length, \( \delta_{53} \), is:

\[
\delta_{53} = \frac{P_{35}D_2}{20} \left[ \frac{1}{E_0} \left( \frac{1}{K} - 1 \right) \frac{1}{E_0} (0 - 1) \right] \text{mm (in.)}
\]

\[
K = \frac{D_2}{D_1}
\]

The minimum pull-up length, \( \delta_6 \), at temperature \( t \), where \( t < 35°C (95°F) \), is:

\[
\delta_6 = \delta_{53} + \frac{D_2}{20} (\alpha_0 - \alpha_5) (35 - t) \text{mm (SI and MKS units)}
\]

\[
\delta_6 = \delta_{53} + \frac{D_2}{20} (\alpha_0 - \alpha_5) (35 - t) \text{in (US units)}
\]

The corresponding minimum surface pressure, \( P_6 \), is:

\[
P_6 = \frac{\delta_6}{\delta_{53}} \frac{P_{35}}{D_2} \text{N/mm}^2 (\text{kgf/mm}^2, \text{psi})
\]

The maximum permissible mating surface pressure, \( P_{\text{max}} \), at 0°C (32°F) is:

\[
P_{\text{max}} = \frac{0.778t_0 (K^2 - 1)}{\sqrt{3K^2 - 1}} \frac{P_{35}}{\delta_{53}} \text{N/mm}^2 (\text{kgf/mm}^2, \text{psi})
\]

and the corresponding maximum permissible pull-up length, \( \delta_{\text{max}} \), is:

\[
\delta_{\text{max}} = \frac{P_{\text{max}}}{P_{35}} \delta_{53} \text{mm (in.)}
\]
MT525 Marine Propulsion Systems
Content (outlook to other subjects)

Content
wk 1 Introduction to marine propulsion systems
wk 1 Ship types and propulsors
wk 1,2 Fixed pitch propellers
wk 2 Controllable pitch propellers
wk 3 Steerable thrusters and pods
wk 4 Waterjets
wk 5 Transverse thrusters
wk 5 Hydraulic systems and controls
wk 6 Class requirements, rudder interaction and material aspects
wk 7 Cost driving factors and system integration

2. Ship type and propulsors

Duration: 0.5 hour
Main goal: explain main design criteria of ships in relation to their propulsion systems
Content:
Main transportation routes
Ship design aspects
Ship concepts
Main ship types and their requirements for the propulsion system
MAIN REQUIREMENTS FOR PROPULSION SYSTEMS

- THE APPROACH STARTS WITH THE REQUIREMENTS PER SHIP TYPE
- THEN WITH THE MAIN REQUIREMENTS FOR THE MAIN DRIVE
- THEN THE MAIN REQUIREMENTS FOR THE PROPULSION SYSTEM
- THEN WE HAVE A PRODUCT MARKET COMBINATION
Sea transportation of iron ore

Sea transportation of grain
Ferry operator

MTS25 Marine Propulsion Systems

Ship Design Spiral

Design process is iterative; e.g. main dimensions depend on the weight of the ship, and the weight depends on the main dimensions...

MT 525 Marine Propulsion Systems
Ship design process

1. Trade / Ship type
2. Limitations due to field of operation
   - Canals (max. draught/breadth)
   - Rivers/harbours (max. draught/length)
3. Requirements from classifications societies
   and governmental authorities
   - Strength (light ship weight)
   - Safety (load lines)
4. Deadweight (DWT) and ship speed ($V_s$)

MT 525 Marine Propulsion Systems

Ship design process

5. Principal dimensions based on the previous
   - Displacement, $V$ or $\n$
   - Length, $L$
   - Breadth, $B$
   - Draught, $T$
   - Depth, $D$

6. Hull Lines
   - Ship resistance
   - Propulsive coefficients
   - Wake field

MT 525 Marine Propulsion Systems
Ship design process

7. Other requirements
   - Electricity (shaft generator)
   - Noise/Vibrations
   - Operating profile

8. Propeller
   - Diameter
   - Propeller speed
   - Propeller nozzle
   - Skew back
   - Number of blades
   - Blade area ratio
   - Load distribution

MT 525 Marine Propulsion Systems

Ship designing process

9. Powering
   - Main engine and auxiliary engine
   - MCR (Sea Margin)
   - Fuel type

MT 525 Marine Propulsion Systems
Outset of design

- Total costs per DWT and nautical mile
  - Capital cost
  - Operating costs
    - Fuel
    - Crew
    - Lube oil
    - Insurance
    - Harbour
    - Administration
    - Maintenance

MT 525 Marine Propulsion Systems

Ship Concepts

Reason for the existence of different concepts

Different cargoes...

have different transportation requirements...

which calls for different ship concepts

MT 525 Marine Propulsion Systems
Ship Concepts

Lift Triangle

Aerostatic:
Works with an air cushion...

Aerodynamic:
Works with wings...

Hydrostatic Lift
Hydrodynamic Lift

MT 525 Marine Propulsion Systems
Ship Propulsion

What is ship propulsion?

Ship propulsion is...
- Installing devices that can make a ship move...
  or in other words...
- Installing devices that can counter-act the forces that stop a ship from moving...
  or in other words...
- Installing devices that can overcome a ship's resistance...

MT 525 Marine Propulsion Systems
Power Requirements

Power Requirement Characteristic

- displacement ship (third power curve)

Ship speed

MT 525 Marine Propulsion Systems

Limitation for ship speed

- No technical limitation
- Thus an economic limitation
- The economic limitation is given by the value of the cargo...

Ship speed

1 knot = 1 nautical mile / hour
1 nautical mile = 1.852 kilometer
1 knot = 1.852 kilometer / hour
15 knots = 28 kilometer / hour
25 knots = 46 kilometer / hour

MT 525 Marine Propulsion Systems
Power Requirements

Justification for advanced ship concepts

Required propulsion power

- displacement ship (third power curve)
- non-displacement ship

Ship speed

MT 525 Marine Propulsion Systems

Propulsion Concepts

- FPP: Direct drive...
- CPP: Indirect drive...
- FPP: Diesel-electric drive...

MT 525 Marine Propulsion Systems
Ship Types

World fleet development

- Tankers
- Container vessels
- General cargo vessels
- Others

Merchant fleet [million dwt]

Source: I. L.

MT 526 Marine Propulsion Systems

Ship Sizes

Various units to express ship size

- Deadweight tonnage [dwt]

MT 526 Marine Propulsion Systems
Ship Sizes

Deadweight & Payload

- Archimedes...
- Weight of the displaced water...
- Weight of the steel structure...
- Weight of the machinery...
- Weight of the consumables...
- Displacement
- Steel weight
- Machinery weight
- Deadweight
- Fuel & stores weight
- Payload

MT 525 Marine Propulsion Systems

Ship Sizes

Various units to express ship size

- Deadweight tonnage [dwt]
- Container capacity [TEU]
- Gross tonnage [grt]
- Passenger capacity [Pax]
- Hold or tank volume [m³ or cuft]

TEU
Length = 20 feet
Width = 8 feet
Height = 8, 8.5 or 9.5 feet

MT 525 Marine Propulsion Systems
Ship Types

Major ship types
- Tankers
- Bulk carriers
- Container vessels
- General cargo vessels
- Others (RO-RO vessels, Reefer vessels, Ferries, Cruiseships, etc.)

Fleet segmentation 1999
- Tankers: 42%
- Bulk carriers: 8%
- Container vessels: 10%
- General cargo vessels: 4%
- Others: 36%

MT 525 Marine Propulsion Systems

Ship Types

Tankers

Afrenax tanker - 100,000 dwt

MT 525 Marine Propulsion Systems
Ship Types & Sizes

Tankers
- < 5'000 dwt
- 5'000 to 10'000 dwt
- 10'000 to 80'000 dwt
- Aframax (80'000 to 120'000 dwt)
- Suezmax (120'000 to 200'000 dwt)
- Very Large Crude Carrier (VLCC) (> 200'000 dwt)

MT 525 Marine Propulsion Systems

Ship Types

Tankers
- Very Large Crude Carrier (VLCC) (> 200'000 dwt)

Berge Stadt
MT 525 Marine Propulsion Systems
**Ship Types**

**Tankers**

- *Magdala*
  - MT 525 Marine Propulsion Systems

- *Bunga Cenderawasih*
  - MT 525 Marine Propulsion Systems
Ship Types

Panamax bulk carrier - 74'000 dwt

MT 525 Marine Propulsion Systems

Ship Types & Sizes

Bulk carriers
- Handysize (10'000 to 35'000 dwt) - low speed engine FPP
- Handymax (35'000 to 50'000 dwt) - low speed engine FPP
- Panamax (50'000 to 80'000 dwt) - low speed engine FPP
- Capesize (> 80'000 dwt) - low speed engine FPP

MT 525 Marine Propulsion Systems
Ship Types

Bulk carriers

Lyra
MT 525 Marine Propulsion Systems

Ship Types

Bulk carriers

Pina Prima
MT 525 Marine Propulsion Systems
Ship Types

Bulk carriers

Handymax bulk carrier - 38'000 dwt

Incatrans
MT 525 Marine Propulsion Systems

Ship Types

Container vessels

Cross section - Cargo hold

hatch covers

cell guides

Post-Panamax container vessel - 6296 TEU
MT 525 Marine Propulsion Systems
Ship Types & Sizes

<table>
<thead>
<tr>
<th>Container vessels</th>
<th>Low speed engines</th>
</tr>
</thead>
<tbody>
<tr>
<td>Feeders</td>
<td>low speed engines</td>
</tr>
<tr>
<td>Sub-Panamax (B &lt; 32.2 meters)</td>
<td>low speed engines</td>
</tr>
<tr>
<td>Panamax (B = 32.2 meters)</td>
<td>low speed engine</td>
</tr>
<tr>
<td>Post-Panamax (B &gt; 32.2 meters)</td>
<td>low speed engine</td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems

Ship Types

<table>
<thead>
<tr>
<th>Container vessels</th>
<th>8500 TEU</th>
</tr>
</thead>
<tbody>
<tr>
<td>Post-Panamax container vessel</td>
<td></td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems

P&O Nedlloyd Southampton - Sulzer 12RTA96C
Ship Types

**Container vessels**

*Ever Dainty*

MT 525 Marine Propulsion Systems

Ship Types

**General cargo vessels**

*General cargo vessel - 9'000 dwt*

MT 525 Marine Propulsion Systems
Ship Types & Sizes

<table>
<thead>
<tr>
<th>General cargo vessels</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 5'000 dwt</td>
<td>medium speed engines and gearbox</td>
</tr>
<tr>
<td>5'000 to 10'000 dwt</td>
<td>medium speed engines and gearbox</td>
</tr>
<tr>
<td>10'000 to 35'000 dwt</td>
<td>low speed engines and FPP</td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems

Ship Types

General cargo vessel - 12'000 dwt.

General cargo vessels

Aalsmeergracht

MT 525 Marine Propulsion Systems
Ship Types

Others - RO-RO vessels

MT 525 Marine Propulsion Systems

Ship Types

RO-RO vessel - 25000 dwt

Grande America
MT 525 Marine Propulsion Systems
Ship Types

Others - Reefer vessels

Reefer vessel - 10'000 dwt

Brazilian Reefer - Sulzer 6RTA58

MT 525 Marine Propulsion Systems
Ship Types

Cross section cargo hold

Others - Cruiseships

Cruiseship - 137'000 grt

MT 525 Marine Propulsion Systems

Ship Types

Cruiseship - 137'000 grt

Others - Cruiseships

Voyager of the Seas - ex Wärtsilä 12V48C

MT 525 Marine Propulsion Systems
Ship Types

Carnival Fantasy - 4x Sulzer 12ZA405 + 2x Sulzer 8ZA405
MT 525 Marine Propulsion Systems

Shipping

Different parties involved in shipping:
- Shipper: Person who has cargo that he wants to transport...
- Cargo broker: Helps persons who have cargo that they want to transport...
- Ship broker: Helps persons who have ship with which they want to transport cargo...
- Ship owner: Person who owns a ship...
- Ship manager: Person who takes care that one could transport cargo with a ship...
- Ship operator: Person who actually transports cargo with a ship...
- Charterer: Person who has hired a ship to transport cargo with it...

Different charter forms:
- Bareboat charter: Ship owner covers capital costs...
- Time or trip charter: Ship owner covers operating costs too...
- Voyage charter: Ship owner covers voyage costs too...

MT 525 Marine Propulsion Systems
Shipping

Basic kinds of shipping

There are two basic kinds of shipping...

- Bulk shipping
  - The cargo decides where the ship goes...

- Liner shipping
  - The ship decides where the cargo goes...

- Tramp shipping
  - A mix form...
    - Oil, ore, grain, coal, etc.

- Project cargo
  - Containers, pallets, bales, other unitized cargo, etc.

MT 525 Marine Propulsion Systems

Charter Party
Responsibility

<table>
<thead>
<tr>
<th>Capital Costs</th>
<th>Bareboat</th>
<th>Time &amp; Trip</th>
<th>Voyage</th>
<th>Voyage Liner</th>
</tr>
</thead>
<tbody>
<tr>
<td>Investment Cost</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Interest</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating Costs</td>
<td>Manning Costs</td>
<td>Shipowner's Costs</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maintenance &amp; Repairs</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Survey &amp; Drydock</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stores, supplies &amp; lube oil</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>H&amp;M &amp; P&amp;I Insurance Management</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Voyage Costs</td>
<td>Bunkers</td>
<td>Charterer's Costs</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hold Cleaning</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Port Charges</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Canal Dues</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fairway Fees</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pilotage</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tug Costs</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Voyage Insurance</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Loading Costs</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Unloading Costs</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems
Stimulating questions

- Can you imagine that for some ship types it does not matter so much what the fuel consumption is?
- Indicate some ship types for which maneuvering is very important.
- Is astern operation important?
Marine Propulsion Systems
Content (outlook to other subjects)

- **Content**
  - wk 1 Introduction to marine propulsion systems
  - wk 1 Ship types and propulsors
  - wk 1,2 Fixed pitch propellers
  - wk 2 Controllable pitch propellers
  - wk 3 Steerable thrusters and pods
  - wk 4 Waterjets
  - wk 5 Transverse thrusters
  - wk 5 Hydraulic systems and controls
  - wk 6 Class requirements, rudder interaction and material aspects
  - wk 7 Cost driving factors and system integration

MT 525 Marine Propulsion Systems

Why this course

- Marine propulsion systems are an important part of the ships cost
- This may amount to 5% of the total ship
- The main operation of the vessels and the ship are determined by the character of the main propulsion plant
- The reliability of the vessel is mainly determined by the reliability of the propulsion plant (engine, propeller, .. Rudder)
- From economical point of view: due to the complexity production in Europe still is possible. The main players in marine propulsion systems still have their basis in Europe.
- Do you have other reasons?

MT 625 Marine Propulsion Systems
Main purpose

- To introduce the main working principle to fourth years student who later will be active in shipyards, ship owners, consultants, surveyors or marine equipment supplier.
- Focus on first principle
- Start with main hydrodynamic function
- Include basic mechanical system and main strength parameters
- Cover system integration issues
- Cover reliability aspects (what can go wrong)

Main targets

After the course students are expected to:

- Have basic knowledge on the type and working principle of most marine propulsion systems
- Understand the main working principles
- Understand how to integrate propulsion systems in a ship
- Know the main critical strength parameters
- Know which questions to ask to make it work
- Know how to review technical specifications or how to review them
1. Introduction to Marine Propulsion Systems

Duration: 0.5 hour
Main goal: main purpose and explain historical development and achievements
Content:
- Some definitions
- When did it all start
- Maximum power available per propulsor system
- Basic considerations in the design of propulsor system
- Basic principles to generate thrust

Historical development

- The basic propeller invented in 1835
- Before that time sails were used to push a ship
- Increase of density (air to water) decreased the size of the propulsor
- Via rotation of a helix thrust was generated in the water
- Via rotation thrust is generated
Sorts of propulsors

How can thrust be generated:

- accelerate the water by pushing
- accelerate water by electric force field
- boiling water
- accelerate by burning fuel
- generate thrust by moving up and down
- accelerate water via suction pump
- accelerate water via rotating wings
- paddle wheel
- emhd
- plof plof
- ramjet
- fishtail propulsion
- waterjet
- propeller

Paddlewheel

MT 525 Marine Propulsion Systems
Plof Plof

MT 525 Marine Propulsion Systems

Ram jet

MT 525 Marine Propulsion Systems
**Helix**

NB! If you walk in a circular stairway, you move in a helix geometry...

**Propellers utilise favourable characteristics of blade section**

- From model tests, lift and drag coefficients are obtained:
  - Measured lift = $C_L \frac{1}{2} \rho V^2 S$
  - $S =$ Wing area
  - Measured drag = $C_D \frac{1}{2} \rho V^2 S$
  - $V =$ Flow speed

---

MT 525 Marine Propulsion Systems
Pressure upon a wing section

With camber and/or angle of attack of the wing section, pressure differential is achieved.

Pressure distribution on wing section

Performance comparison

Note: Efficiency of waterjets has gone up considerably since 1978 (typical values WJ 0.7, SPP 0.65 around 40 knots) (Allison, 1978)

Figure 1.7. Approximate maximum installed efficiency envelopes for different propellers. Taken from [Allison 1978].
Performance comparison

Available power on propulsors

- Maximum power resembles the learning curve in the development
- Besides the absolute power the power density and the tip speed are important parameters for the difficulty of the design
- Parameters: power, RPM, diameter, ship speed, draught of the ship

- Maximum power is limited by:
  - what is needed from ship perspective (maximum ship size)
  - what is possible for instance the maximum weight or production limits
    - maximum weight fixed pitch propeller 125 tons (but this is not a real physical limit, more important is for instance the draught limitation of the ship it is intended for)
There are various types of propellers...

- Fixed-pitch propellers
- Controllable-pitch propellers
- Ducted propellers
- Podded propellers

MT 525 Marine Propulsion Systems

Historical development of maximum power applied per ship propulsor

MT 525 Marine Propulsion Systems
Relevance of power density

Power density:

Power per disk area:

\[
2 \pi Q_n \approx \frac{T}{\frac{\pi}{4} D^2} = \frac{\pi}{4} D^2
\]

MT 525 Marine Propulsion Systems
Consequences

- Large variation especially ship speed (majority <40 knots)
- Large variation in power
- Little or now series effect
  - Tailor made design approach
  - Design problem: how to make first time right design
  - Compare with airplane: even a Fokker 50 has been built 200 times

MT 525 Marine Propulsion Systems
Controls

Steerable thruster

Transverse thruster

Waterjet

FPP

MT 525 Marine Propulsion Systems

Steerable thruster

Propulsion Concepts

FPP

CPP

low-speed

Direct drive...

Gearbox

medium-speed

Indirect drive...

FPP

medium-speed

Diesel-electric drive...

MT 525 Marine Propulsion Systems
Questions

- Why no FPP with gearbox driven engines?
- Why no FPP with more than one engine?

Ship Types

Source: RSL

- Tankers
- Container vessels
- Bulk carriers
- General cargo vessels
- Others

World fleet development
Stimulating questions

- When we refer to marine propulsion systems what are the system boundaries?
- Why is it so difficult to verify the working of mps before the actual seatrial?
- What do you consider the most difficult aspects of designing marine propulsion systems?
5a Pods

Content:
1 Introduction to marine propulsion systems
2 Ship types and propulsors
3 Fixed pitch propellers
4 Controllable pitch propellers
5 Steerable thrusters
5a Pods
   wk 5 Waterjets
   wk 6 Transverse thrusters
   wk 7 Hydraulic systems and controls
   wk 7 Class requirements, rudder interaction and material aspects
   wk 7 Cost driving factors and system integration

5a. Podded propulsors

Duration: 1 hour
Main goal: main working principle and design criteria
Content:
   Why pods?
   Historic development and applications
   Mechanical design and working principle
   Hydrodynamic design aspects and comparison conventional propulsors
   Electric drive and E motor
   Steering forces, maneuvering and crash stop

MT625 Marine Propulsion Systems
Why pods

- Make use of electric drives
- Large torque capability
- Take advantage of pods type propulsors such as propulsive efficiency and inflow
- Electric motor is necessary given gear limits at 7000 kW

History

Origin
- The wish to combine the thruster maneuvering capabilities with the high torque of electric drives for powers higher than 7500 kW.

- 1990 Support Vessel "Seili" 1500 kW (Azpod prototype)
- 1993 Icebreaking Tanker "Uikku" 11.4 MW (Azpod prototype)
- 1998 Cruise vessel "Elation" 2*14 MW (Azpod)
- 1999 Ultra Deep Water Semisubmersible Drilling Rig "Sedco Express" 4*7 MW (Mermaid)
- 2000 Chemical Tanker "Prospero" 5.1 MW (SSP)
- 2000 Cruise vessel "Millennium" 2*19.5 MW (Mermaid)
- 2001 Cruise vessel "Norwegian Dawn" 2*20 MW (Azpod)
- 2001 Ferry "Nils Holgersson" 2*11 MW (SSP)
- 2002 Cruise vessel "Seven Seas Voyager" 2*7 MW (DOLPHIN)
- 2004 Cruise vessel "Queen Mary 2" 4*21 MW (Mermaid)
Products

- ABB Azipod
- STN-WP
- Dolphin
- Kamewa
- Mermaid

MT 525 Marine Propulsion Systems

Systems overview

MT 525 Marine Propulsion Systems
**Principle**

Azipod, Mermaid, SSP and DOLPHIN are **Podded Propulsors**:

- **Principle**

  An electric motor with controllable and reversible rpm is connected directly to the propeller and is mounted in an underwater housing that can rotate continuously over 360°.

- A podded propulsor is similar to a steerable or azimuthing thruster, but usually for larger powers (7 MW and up)
Pod system

Steering Unit
Slewing Bearing
Steering Hub
Seal Support Tanks
Aft Bearing Position
Electric Motor
Forward Bearing Position

Cooling air in- and output
Lubrication Oil Pumps
Bilge Pumps
Pod Housing

Note: some systems do not need external cooling (power lt 7 MW)

MT 525 Marine Propulsion Systems

Steering

- Slewing bearing

MT 525 Marine Propulsion Systems
**Forward Bearing Position**

- Radial CARB Bearing

- Bearing and inboard seal can be replaced without removing the rotor or propeller shaft

**Aft Bearing Position**

- Aft bearing block
  - Thrust bearings
  - Shaft earthing
  - Rotor position pickup
  - Radial bearing
Steering Mechanism

- SAI hydraulic motor
- Brevini reduction gear
- Counter balance block

MT 525 Marine Propulsion Systems

Assembly and Installation

MT 525 Marine Propulsion Systems
Seal system in operation

MT 525 Marine Propulsion Systems

Modified venting system

Fig. 3 - Old gear rack venting system

Fig. 4 - New gear rack venting system
Seals

Outboard Seal System

Bearing problems

MT 525 Marine Propulsion Systems
Experience in practice

- Problem with water ingress
- Problem with lubricating and bearing loads
- Problem with electrical components in the pod (overheating)

Propulsive efficiency

Efficiency

\[
\text{Propulsive Power (} P_0 \text{)} = \frac{\text{Ship Resistance (} R \text{)} \times \text{Ship Speed (} V_s \text{)}}{\text{Propulsor Efficiency (} \eta_p \text{)} \times \text{Hull Efficiency (} \eta_h \times \eta_R \text{)}}
\]

- Podded Propulsion
  - Resistance (} R \text{)} is lower (no appendages)
  - Hull Efficiency (} \eta_h \text{)} is better, depending on hull form
  - The Propulsor Efficiency (} \eta_p \text{)} includes the pod drag and is slightly lower
  - Pod diameter should be as small as possible (close to 0.5 ft), this requires special design electric motor

MT 525 Marine Propulsion Systems
**Propulsive efficiency**

Efficiency
- Conventional Twin Screw
  \[ P_D = \frac{(R + R_{appendages}) V_s}{\eta_o \eta_h \eta_r} = \frac{110 P_E}{0.69 \cdot 0.88 \cdot 10} = 181 P_E \]

- Podded Propulsion
  \[ P'_D = \frac{R V_s}{\eta_{cp} \eta_h \eta_r} = \frac{10 P_E}{0.66 \cdot (0.88 \text{ to } 0.93) \cdot 10} = \text{163 to 171} P_E \]

**Propeller inflow**

Conventional Wake Field

MT 525 Marine Propulsion Systems
Propeller inflow

Podded propulsor

Almost uniform
Only disturbed by the ships boundary layer

Effect of pod orientation on pressure pulses

Fig. 8 - Change of Pressure Fluctuations with Pod Inclination
MT 525 Marine Propulsion Systems
Design of pod and electric motor

- Normal electric motor diameter = length
- Pod motor length about 2 time diameter
- Small reduction in electrical efficiency

- Pod diameter is close to 0.5 this reduces the open water efficiency 5 to 10%

Electromotor:
- \( \frac{D^3 NL}{P} \) = constant

MT 525 Marine Propulsion Systems

Design of pod and electric motor

Fig. 2 - What is an optimum POD configuration?

Fig. 3 - Total power at design speed as function of \( \frac{Dm}{Dp} \) - Raytext
**Design of pod and electric motor**

**Fig. 6** - Total power at design speed as function of weight to power ratio for Cargo

### Design of pod and electric motor

**Table 1** - Pod technology targets proposed

<table>
<thead>
<tr>
<th>Ship type</th>
<th>ROPAX</th>
<th>CARGO</th>
<th>TRIMARAN</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Motor parameters</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Weight/Power ratio (kg/kW)</td>
<td>3.0-3.5</td>
<td>4.0-4.5</td>
<td>3.5</td>
</tr>
<tr>
<td>DPLN/kW (m³/kW.min)</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
</tr>
<tr>
<td>D₉/D₈</td>
<td>0.50-0.55</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td><strong>Hydrodynamic Parameters</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>CD coefficient</td>
<td>0.010</td>
<td>0.010</td>
<td>0.010</td>
</tr>
<tr>
<td>Propeller efficiency</td>
<td>0.75</td>
<td>0.75</td>
<td>0.75</td>
</tr>
</tbody>
</table>
Design of pod and electric motor

\[ D = C_d \frac{1}{2} \rho \left( V_e + \frac{1}{2} V_a \right)^2 S. \]

Strut reduce rotation:
- total torque increase
- Drag of pod and strut:
- total thrust reduces

Estimation of pod effect

Hoerners formula's

\[ C_f = \frac{0.075}{(\log R_e - 2)^2} \]
- Friction coefficient

\[ C_{d_t} = C_f \left[ 3 \left( \frac{l}{D} \right) + 4.5 \left( \frac{D}{l} \right)^{0.5} + 21 \left( \frac{D}{l} \right)^3 \right] \]
- Gondola, l length, D diameter

\[ C_{d_p} = 2C_f \left[ 1 + 2 \left( \frac{t}{c} \right) + 16 \left( \frac{t}{c} \right)^4 \right] \]
- Strut, t thickness, c chord

\[ C_{d_i} = \left[ 17 \left( \frac{t}{c} \right)^2 - 0.05 \right] \]
- Interference drag
Pod efficiency case study

Fig. 1 - Body Plan of a bulk carrier model (unit: cm)

MT 525 Marine Propulsion Systems

Fig. 3 - Propeller open-water characteristics

Fig. 6 - Characteristics of podded propellors
Tractor or Pusher

- **Tractor**
  - Rotation behind the propeller interacts with the strut
  - Effect is largest at bollard condition

- **Pusher**
  - Drag of strut is speed dependent
  - Minor torque effect

- At low speed use pusher
- At high speed use tractor (better wake and inflow, but higher steering torque)

Pod efficiency case study

*Fig. 7 - Comparison of propeller only. Fig. 8 - Estimated characteristics of podded propellers*
### Electrical systems

- **Synchronous motor:**
  - alternating current in the stator winding
  - rotor with permanent magnet or
  - rotor with rotating exciter (small generator with rectifier generates power for rotor)

- **Asynchronous motor**
  - squirrel cage motor

- **Cyclo convertor**
  - unlimited power level
  - efficient
  - many components

- **Synchro convertor**
  - unlimited power
  - efficient
  - only with synchronous motor
  - power factor part load is poor

- **PWM**
  - very efficient
  - limited to 7-8 MW

---

**Table 2 - Comparison of η, EHP, n, and DHP**

<table>
<thead>
<tr>
<th>Type</th>
<th>Pusher</th>
<th>Tractor</th>
<th>Pusher</th>
<th>Tractor</th>
<th>Pusher</th>
<th>Tractor</th>
<th>Pusher</th>
<th>Tractor</th>
</tr>
</thead>
<tbody>
<tr>
<td>GAL Method</td>
<td>1</td>
<td>ITTC 1978</td>
<td>1</td>
<td>ITTC 1978</td>
<td>1</td>
<td>ITTC 1978</td>
<td>1</td>
<td>ITTC 1978</td>
</tr>
<tr>
<td>EHP (kW)</td>
<td>2857</td>
<td>2864</td>
<td>2858</td>
<td>2868</td>
<td>3195</td>
<td>3194</td>
<td>3200</td>
<td>3200</td>
</tr>
<tr>
<td>η</td>
<td>0.483</td>
<td>0.491</td>
<td>0.498</td>
<td>0.496</td>
<td>0.526</td>
<td>0.546</td>
<td>0.575</td>
<td>0.554</td>
</tr>
<tr>
<td>DHP (kW)</td>
<td>5977</td>
<td>5874</td>
<td>6198</td>
<td>6128</td>
<td>5873</td>
<td>5829</td>
<td>6093</td>
<td>3771</td>
</tr>
</tbody>
</table>

---

**Fig. 21 - Comparison of calculated propulsion coefficients (Method I)**

**Fig. 22 - Comparison of calculated propulsion coefficients (Method II)**

---

MT 525 Marine Propulsion Systems
Cyclo-converter

- number of thyristors: 36

The cyclo-converter exists of min. 3 thyristor bridges, each supplied from propulsion switchboard.

These bridges are required to allow positive and negative current into the winding.

Basic arrangement for drive systems features 36 switching devices (thyrstors) for full four quadrant operation.

More commonly a propulsion motor has 2 sets of 3-phase windings, each with its own network of 36 devices.

Cyclo-converter drive system provides high torque at low shaft speeds ——> Typically suitable for icebreakers.

MT 526 Marine/Propulsion/Systems
Synchro-converter

Designed with fewer thyristors than cyclo-converters

MT 525 Marine Propulsion Systems
Synchro-converter

- The synchro-converter is associated with AC synchronous motors.
- The converter arrangement is fully reversible and allows motor drive operation in the four quadrants of the torque versus speed diagram.
- The input bridge and line reactor in the DC link provide current source for the output inverter bridge.
- By controlling the switching of the thyristor pairs in the inverter, controlled current at variable frequency can be applied sequentially to the motor windings.
- Does not have the same overload characteristic as the cyclo-converter, but is perfectly suitable for typical propeller law characteristics of marine propulsion.

Single line diagram synchro-converter

MT 525 Marine Propulsion Systems
**Cyclo- versus Synchro-converter**

- Comparison for propulsion drives 2x6.5 MW
- (12 pulse characteristic)

<table>
<thead>
<tr>
<th></th>
<th>Cyclo-conv.</th>
<th>Synchro-conv.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of thyristors</td>
<td>72</td>
<td>24</td>
</tr>
<tr>
<td>Number of prop. transformers</td>
<td>12</td>
<td>4</td>
</tr>
<tr>
<td>Number of prop. circuit breakers</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Length of converter panel (mm)</td>
<td>27600</td>
<td>15600</td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems

---

**Synchro-Converter Board**

[Image of Synchro-Converter Board]

MT 525 Marine Propulsion Systems
Pods: modes of operation

ⅰ SAFETIES:
1. Automatic steering angle limitation as function of measured ship speed and steering torque, both in follow-up as in back-up

ⅱ EFFICIENCY AND COURSE STABILITY:
1. While in mode with master azimuthing lever, common steering wheel and auto-pilot asynchronous steering to avoid too high loss of ship-speed during turning.

MT 525 Marine Propulsion Systems
Pods: modes of operation

**MANOEUVRING:**
1. Slow sailing in confined waters on Joystick or manual on azimuthing levers

**BERTHING:**
1. Crabbing maneuver towards or from berth at near zero longitudinal speed by joystick or by azimuthing levers
Manoeuvring

- Crabbing with podded propulsors
  - Increase in transverse forces of 10%
  - More efficient since both propellers are operating 'ahead'
  - Less noise and vibration

MT 525 Marine Propulsion Systems

Crabbing

Figure 12. Set-up and sign definition for crabbing tests

Figure 13. Crabbing whiffle plug for a ship with pods

MT 525 Marine Propulsion Systems
Pods improve turning circles

Figure 6. Turning circle statistics of ships with podded propulsion

Figure 7. Turning circle data of ships equipped with podded or conventional propulsion units

Forces at zero speed

Figure 10. Longitudinal (left) and lateral (right) forces on the ship

Figure 11. Longitudinal (left) and lateral (right) force coefficients

MT 525 Marine Propulsion Systems
### IMO Requirements

#### Table 2 - IMO Turn Standards For Ship Manoeuvrability

<table>
<thead>
<tr>
<th>Ability</th>
<th>Test</th>
<th>Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Turning Ability</td>
<td>Turning Test, With Max. Bailer Angle</td>
<td>Track Result = 2.5s, by the time that 10° deviation is reached, from the original heading in exactions of 10° order angle</td>
</tr>
<tr>
<td>Varying Checking and Course Keeping Ability</td>
<td>10°/10° Manoeuvring Test</td>
<td>(1) First overshoot angle = 10°, if L/V is less than 10 sec. or more than 30°</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(2) Second overshoot angle = 25°, if L/V is less than 10 sec. or more than 40°</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(2) Second overshoot angle = 35°, if L/V is less than 30 sec. or more than 50°</td>
</tr>
</tbody>
</table>

**MT 525 Marine Propulsion Systems**

![Diagram of ship manœuvre](image1)

![Diagram of turn](image2)
Turning ability

<table>
<thead>
<tr>
<th>Approach</th>
<th>Speed</th>
<th>Type of propulsion</th>
<th>Radii/PGD (deg)</th>
<th>Advance</th>
<th>Thrust (kN)</th>
<th>Torsional Diameter (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conventional</td>
<td>35° STR</td>
<td>3.75</td>
<td>2.96</td>
<td>4.5</td>
<td>2.63</td>
<td>2.46</td>
</tr>
<tr>
<td>Propulsion</td>
<td>35° STR</td>
<td>2.31</td>
<td>2.47</td>
<td>4.5</td>
<td>1.97</td>
<td>1.85</td>
</tr>
<tr>
<td>Conventional</td>
<td>35° PORT</td>
<td>2.76</td>
<td>2.93</td>
<td>4.5</td>
<td>2.64</td>
<td>2.32</td>
</tr>
<tr>
<td>Propulsion</td>
<td>35° PORT</td>
<td>2.32</td>
<td>2.42</td>
<td>4.5</td>
<td>1.98</td>
<td>1.89</td>
</tr>
</tbody>
</table>

Forces at high speed

- Steering forces and moments are very large on pulling design
- Reason is oblique inflow of the propeller
- Distance of propeller to turning axis results in steering moment
- Ship need to be reinforced to counteract steering moments and forces
Steering forces

MT 525 Marine Propulsion Systems
### Effect on pod on efficiency

#### Table 2 - Propeller hub geometry

<table>
<thead>
<tr>
<th>Propeller arrangement</th>
<th>Hub length ratio $h/D$</th>
<th>Hub diameter ratio $d_h/D$</th>
<th>Hub diameter ratio fore $d_{hF}/D$</th>
<th>Hub diameter ratio aft $d_{hA}/D$</th>
<th>Hub diameter ratio $d_h/D$</th>
<th>Hub diameter ratio fore $d_{hF}/D$</th>
<th>Hub diameter ratio aft $d_{hA}/D$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pull propeller arrangement</td>
<td>0.3193</td>
<td>0.2568</td>
<td>0.2648</td>
<td>0.4102</td>
<td>0.3530</td>
<td>0.4106</td>
<td>0.2899</td>
</tr>
</tbody>
</table>

#### Table 3 - Pod housing main data

<table>
<thead>
<tr>
<th>Propeller arrangement</th>
<th>Length ratio of the gondola $L/D$</th>
<th>Diameter ratio of the gondola $d/D$</th>
<th>Clearance ratio between propeller plane and steering axis $c/D$</th>
<th>Strut length ratio $b/D$</th>
<th>Strut height ratio $h/D$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pull propeller arrangement</td>
<td>1.7557</td>
<td>2.2159</td>
<td>0.8125</td>
<td>0.8325</td>
<td>0.4545</td>
</tr>
<tr>
<td>Push propeller arrangement</td>
<td>1.7977</td>
<td>2.5000</td>
<td>0.832</td>
<td>1.0247</td>
<td>0.4545</td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems

### Effect on pod on efficiency

#### Table 4 - Open water characteristics for the design torque coefficient, steering angle $\theta$

<table>
<thead>
<tr>
<th>$\theta$</th>
<th>$c_N$</th>
<th>$c_{PM}$</th>
<th>$c_{D}$</th>
<th>$c_{VL}$</th>
<th>$c_{AM}$</th>
<th>$c_{AM}$</th>
<th>$c_{MP}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1777</td>
<td>0.0489</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1773</td>
<td>0.0489</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1616</td>
<td>0.0328</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1675</td>
<td>0.0365</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1153</td>
<td>0.0198</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.0772</td>
<td>0.0691</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.0086</td>
<td>0.0659</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.0151</td>
<td>0.0371</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.0064</td>
<td>0.0458</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.0016</td>
<td>0.0346</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.0009</td>
<td>0.0637</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

MT 525 Marine Propulsion Systems
Pod efficiency case study

Fig. 6 - Open water characteristics of podded drives with push propellers CP 1374 and CP 1375

MT 525 Marine Propulsion Systems
**Effect on pod on efficiency**

**Fig. 4 - Open water characteristics of the free running propellers CP 1374 and CP 1375**

MT 525 Marine Propulsion Systems

**Effect on pod on efficiency**

**Fig. 5 - Open water characteristics of podded drives with pull propellers CP 1374 and CP 1375**

MT 525 Marine Propulsion Systems
Fig. 7 - Force and moment coefficients of the podded drives with pull propeller, \( J = 0.847 \)

Fig. 8 - Force and moment coefficients of the podded drives with push propeller, \( J = 0.850 \)
**Steering forces**

Fig. 1 - Thrust, torque, and normal forces resulting from a propeller in oblique inflow

\[ F_p = \rho A_p (V_s + \frac{1}{2}V_o) V_s \sin \delta \]

Momentum equation due to steering entrance speed reduces

\[ V_s = V_o \cos \delta \]

Side force is generated due momentum difference perpendicular to shaft

MT 525 Marine Propulsion Systems

---

**Effect of steering angle on side force**

![Effect of steering angle on side force](image)

MT 525 Marine Propulsion Systems
Steering forces

- Side forces and moments load slewing bearing
- Largest forces occur at highest speed and largest steering angle
- Note the propeller gets heavier while steering

MT 525 Marine Propulsion Systems