Contents:

Introduction ........................................................................................................... 3
General Description ................................................................................................. 3
  - Propeller equipment ......................................................................................... 4
  - Propeller type VBS .......................................................................................... 4
Mechanical Design ................................................................................................... 7
  - Hub design ........................................................................................................ 7
OD-Box Design ....................................................................................................... 8
  - ODS type .......................................................................................................... 8
  - ODF type .......................................................................................................... 8
  - ODG type .......................................................................................................... 8
Servo Oil System – ODS-ODF-ODG ....................................................................... 10
  - Hydraulic Power Unit (ODS-ODF) ................................................................. 10
  - Hydraulic system, ODG .................................................................................... 11
  - Lubricating oil system, VBS ............................................................................ 11
Propeller Shaft and Coupling Flange .................................................................... 12
  - Coupling flange ............................................................................................... 12
  - Stern tube ......................................................................................................... 12
  - Liners .............................................................................................................. 13
  - Seals ................................................................................................................ 13
  - Hydraulic bolts ............................................................................................... 13
  -Installation ........................................................................................................ 13
Propeller Blade Manufacturing and Materials ....................................................... 14
  - Blade materials .............................................................................................. 14
Propeller Nozzle ..................................................................................................... 15
  - Nozzle length .................................................................................................. 16
  - Propeller induced pressure impulses and nozzle vibrations ....................... 16
Optimizing Propeller Equipment ........................................................................... 17
  - Propeller design ............................................................................................. 17
  - Optimizing the complete propulsion plant .................................................... 17
  - Hydrodynamic design of propeller blades ..................................................... 18
  - Cavitation ....................................................................................................... 18
  - High skew ....................................................................................................... 19
Technical Calculation and Service ......................................................................... 20
  - Arrangement drawings ..................................................................................... 20
  - Installation manual ......................................................................................... 20
  - Alignment instructions ................................................................................... 21
  - Torsional vibrations ......................................................................................... 21
  - Whirling and axial vibration calculations ....................................................... 22
Instruction Manual .................................................................................................. 22
Main Dimensions ................................................................................................... 23
Propeller Layout Data ............................................................................................. 24
Instruction Manual .................................................................................................. 24
CP Propeller Equipment

Introduction

The purpose of this Product Information brochure is to act as a guide in the project planning of MAN Diesel’s Alpha propeller equipment.

The brochure gives a description of the basic design principles of the Alpha Controllable Pitch (CP) propeller equipment. It contains dimensional sketches, thereby making it possible to work out shaft line and engine room arrangement drawings. Furthermore, a guideline to some of the basic layout criteria is given.

Our design department is available with assistance for optimization of propulsion efficiency and propeller interaction with the environment it works in. Prognoses are performed on eg speed and bollard pull, determining power requirements from the propeller, as well as advice on more specific questions like installation aspects and different modes of operation.

All our product range is constantly under review, being developed and improved as needs and conditions dictate.

We therefore reserve the right to make changes to the technical specification and data without prior notice.

In connection with the propeller equipment the Alphatronic Control System is applied. Special literature covering this field can be forwarded on request.

General Description

MAN Diesel have manufactured more than 7,000 controllable pitch propellers of which the first was produced in 1902.

In 1903 a patent was taken out covering the principle of the CP propeller. Thus more than a century of experience is reflected in the design of the present Alpha propeller equipment.

Today the Alpha controllable pitch propeller portfolio handles engine outputs up to 30,000 kW, fig 1.

Controllable pitch propellers can utilize full engine power by adjusting blade pitch irrespective of revolutions or conditions.

They offer not only maximum speed when free sailing, but also maximum power when towing, good manoeuvrability with quick response via the Alphatronic control system and high astern power.

Shaft generators are used simple and cost efficient. These are just a few of many advantages achieved by controllable pitch propellers.

The basic design principles are well-proven, having been operated in all types of vessels including ferries, tankers, container, cruise, offshore vessels, dredgers and navy ships many of which comply with high classification requirements.
Propeller equipment
The standard propeller equipment comprises a four bladed CP propeller complete with shafting, stern tube, outer and inboard seals, oil distributor (OD) box and coupling flange.

The location of the OD-box depends on the propeller and propulsion configuration.

Propeller type VBS
The present version of MAN Diesel’s Alpha propeller equipment is designated VBS. It features an integrated servo motor located in the aft part of the hub and sturdy designed internal components.

A well-distributed range of different hub sizes makes it possible to select an optimum hub for any given combination of power, revolutions and ice class. The different hub sizes are in principle geometrical similar and incorporate large servo piston diameter with low pressure and reaction forces and few components, while still maintaining short overall installation length.

- Oil Distributor box
The VBS propeller equipment can be supplied with three different oil distribution systems for controlling the pitch depending on the type of propulsion system i.e. direct driven two-stroke or geared four-stroke. All three types incorporate the possibility for emergency operation and a valve box that will keep the propeller pitch fixed in case the hydraulic oil supply is interrupted. The latter is required by classification societies and will prevent the propeller blades from changing the pitch setting.
- ODS - Shaft mounted OD-box
For direct driven propellers without reduction gearboxes the oil distribution box must be located in the shaft line.

The ODS type is intended for this type of installations and features beside the oil inlet ring a hydraulic coupling flange, pitch feed-back and the valve box. The unit design ensures short installation length and all radial holes and slots are located on the large diameter coupling flange and are carefully designed to avoid stress raisers.

- ODF - Gearbox mounted OD-box
For geared four-stroke propulsion plants the oil distribution box is usually located on the forward end of the reduction gearbox.

The ODF contains the same elements as the ODS type and comes in different sizes according to the selected type of VBS propeller equipment.

For long shaft lines with one or more intermediate shafts it is recommended to use the ODS type of oil distribution that will ensure a short feed-back system leading to a more precise control of the pitch setting.

- ODG - Gearbox integrated OD-box
For MAN Diesel designed gearboxes (AMG, Alpha Module Gears) the oil distribution and pitch control system is an integral part of the gearbox. Apart from the stand-by pump no external hydraulic power unit is needed thus facilitating a simple and space saving installation.
Fig. 6: Propeller hub type VBS
Mechanical Design

Hub design
The hydraulic servo motor for pitch setting is an integral part of the propeller hub. The design is shown in fig 6. The propeller hub is bolted to the flanged end of the tailshaft, which is hollow bored to accommodate the servo oil and pitch feed-back tube. The servo piston which is bolted to the pitch control head, forms the hydraulic servo motor together with the propeller cap.

The high pressure servo oil system at the aft end of the hub is completely isolated from the pitch regulating mechanism and thus also from the blade flanges, which means that the blade sealings only are subjected to gravitation oil pressure.

By using a large servo piston diameter and balanced blade shapes, the oil pressure and reacting forces are minimized.

Blade sealing rings are placed between blade foot and hub, fig 7. A compressed O-ring presses a PTFE (teflon) slide ring against the blade foot.

This design ensures maximum reliability and sealing without leakages, also under extreme abrasive wear conditions.

Optionally an intermediate flange can be inserted, by which underwater replacement of propeller blades is possible.

For servicing and inspection of the internal parts, the hub remains attached to the shaft flange during disassembly thereby reducing time and need for heavy lifting equipment. Access to all internal parts is even possible without disman- tling the propeller blades thus reducing the time for inspection and mainte- nance during docking.

A hydraulic tube, located inside the shafting, is connected to the piston. With hydraulic oil flowing through the tube, oil is given access into the after section of the propeller hub cylinder, displacing the servo piston forward, into an ahead pitch position. The displaced hydraulic oil from forward of the piston is returned via the annular space between the tube and shaft bore to the oil tank. Reverting the flow directions will move the propeller in astern position.

Fig. 7: Blade sealing rings
OD-Box Design

ODS type
The shaft mounted unit, fig. 8, consists of coupling flange with OD-ring, valve box and pitch feed-back ring. Via the oil distribution ring, high pressure oil is supplied to one side of the servo piston and the other side to the drain. The piston is hereby moved, setting the desired propeller pitch. A feed-back ring is connected to the hydraulic pipe by slots in the coupling flange. The feed-back ring actuates one of two displacement transmitters in the electrical pitch feedback box which measures the actual pitch.

The inner surface of the oil distribution ring is lined with white-metal. The ring itself is split for easy exchange without withdrawal of the shaft or dismounting of the hydraulic coupling flange. The sealing consists of mechanical throw-off rings which ensures that no wear takes place and that sealing rings of V-lip-ring type or similar are unnecessary.

The oil distributor ring is prevented from rotating by a securing device comprising a steel ball located in the ring.

Acceptable installation tolerances are ensured and movement of the propeller shaft remains possible.

In the event of failing oil pressure or fault in the remote control system, special studs can be screwed into the oil distribution ring hereby making manual oil flow control possible. A valve box located at the end of the shaft ensures that the propeller pitch is maintained in case the servo oil supply is interrupted.
**ODF type**
The gearbox mounted unit, fig 10, consists in principle of the very same mechanical parts as the ODS type. However, the pitch feed-back transmitter is of the inductive type that operates contactless and thus without wear.

The drain oil from the oil distribution is led back to the hydraulic power unit tank.

![Fig. 10: ODF type – for gearbox mounting](image)

**ODG type**
The gearbox-integrated unit, fig 11, consists in principle also of the very same parts as the ODF type. The main difference is the use of the gearbox sump as oil reservoir for both the propeller and gearbox.

![Fig. 11: ODG type – integrated in MAN Diesel’s AMG gearboxes](image)
Servo Oil System
ODS-ODF-ODG

A servo oil pump delivers high pressure oil to a high-pressure filter, a valve unit consisting of non-return valves, a safety valve, a pressure adjusting valve and an electrical operated proportional valve. This proportional valve, which is used to control the propeller pitch can also be manually operated.

From the proportional valve the servo oil is led to an oil distributor ring. Servo oil is also used for lubricating and cooling of this ring. This excess servo oil is led back into the servo oil system.

From the oil distributor ring high pressure oil is led through pilot operated double check valves to one or the other side of the servo piston, until the desired propeller pitch has been reached.

The pilot operated double check valves keep the propeller pitch fixed in case the servo oil supply is interrupted.

The propeller is equipped with an electrical pitch feed-back transducer. This feed-back signal is compared to the order signal to maintain the desired pitch.

The pitch setting is normally remotely controlled, but local emergency control is possible.

Hydraulic Power Unit (ODS - ODF)

The hydraulic Power Unit, fig 12, consists of an oil tank with all components top mounted, to facilitate installation at yard.

Two electrically driven pumps draw oil from the oil tank through a suction filter and deliver high pressure oil to the proportional valve through a duplex full flow pressure filter. One of the 2 pumps is in service during normal operation. A
sudden change of manoeuvre will start up the second pump; this second pump also serves as a stand-by pump.

A servo oil pressure adjusting valve ensures minimum servo oil pressure constantly, except during pitch changes, hereby minimizing the electrical power consumption. Maximum system pressure is set on the safety valve.

The return oil is led back to the tank through a cooler and a filter. The servo oil unit is equipped with alarms according to the Classification Society as well as necessary pressure and temperature indication.

Hydraulic system, ODG
The hydraulic components of the ODG type are built on the gearbox and the propeller control valves form together with the gearbox hydraulics an integrated system. The same functions as described by the ODS-ODF type are available with the ODG integrated solution - the major difference being the common oil sump for both the propeller and the gearbox.

In addition to the gearbox driven oil pump, an electric stand-by pump will automatically start-up in the event of missing servo oil pressure.

Lubricating oil system, VBS
The stern tube and hub lubrication is a common system. The stern tube is kept under static oil pressure by a stern tube oil tank placed above sea level, see fig. 13, 14 and 15.

As an option the propeller can be supplied with two separate systems for lubrication of hub and stern tube.

All Alpha propeller equipment with seals of the lip ring type operates with lub oil type SAE 30/40 - usually the same type of lubricating oil as used in the main engine and/or reduction gear.
Propeller Shaft and Coupling Flange

The tailshaft is made of normalized and stress relieved forged steel, table 1.

<table>
<thead>
<tr>
<th>Material</th>
<th>Forged steel type S45P</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yield strength</td>
<td>N/mm²</td>
</tr>
<tr>
<td>Minimum 350</td>
<td></td>
</tr>
<tr>
<td>Tensile strength</td>
<td>N/mm²</td>
</tr>
<tr>
<td>Minimum 600</td>
<td></td>
</tr>
<tr>
<td>Elongation</td>
<td>%</td>
</tr>
<tr>
<td>Minimum 18</td>
<td></td>
</tr>
<tr>
<td>Impact strength</td>
<td>J</td>
</tr>
<tr>
<td>Minimum 18</td>
<td></td>
</tr>
</tbody>
</table>

Table 1

The tailshaft is hollow bored, housing the servo oil pipe.

The distance between the aft and forward stern tube bearings should generally not exceed 20 times the diameter of the propeller shaft. If the aft ship design requires longer distances, special counter-measures may be necessary to avoid whirling vibration problems.

Coupling flange

The tailshaft can be connected, to the flywheel directly or to an intermediate shaft, via a hydraulic coupling flange, fig 16. To fit the flange high pressure oil of more than 2,000 bar is injected between the muff and the coupling flange by means of the injectors in order to expand the muff.

By increasing the pressure in the annular space C, with the hydraulic pump, the muff is gradually pushed up the cone. Longitudinal placing of the coupling flange as well as final push-up of the muff are marked on the shaft and the muff.

Stern tube

Many different installation and stern tube alternatives exist for both oil and water lubrication. The standard stern tube is designed to be fitted from aft

Fig. 16: Shrink fitted coupling flange

Fig. 17: Standard stern tube – VBS
and installed with epoxy resin and bolted to the stern frame boss, fig 17.

The forward end of the stern tube is supported by the welding ring.

The oilbox and the forward shaft seal are bolted onto the welding ring. This design allows thermal expansion/contraction of the stern tube and decreases the necessity for close tolerances of the stern tube installation length.

As an option the stern tube can be installed with a press-fitting and bolted to the stern frame boss. The stern tube is then supplied with 5 mm machining allowance for yard finishing.

Liners
The stern tube is provided with forward and aft white metal liners, fig 18. Sensors for bearing temperature can be mounted, if required. A thermometer for the forward bearing is standard.

Seals
As standard, the stern tube is provided with forward and after stern tube seals of the lip ring type having three lip rings in the after seal and two lip rings in the forward seal, fig 19.

Hydraulic bolts
The propeller equipment can be supplied with hydraulic fitted bolts for easy assembly and disassembly, fig 20. Machining of holes is simple, reaming or honing is avoided.

Installation
Installation of propeller equipment into the ship hull can be done in many different ways as both yards and owners have different requirements of how to install and how to run the propeller equipment. Other designs of stern tube and/or shaft sealings may be preferred. MAN Diesel are available with alternatives to meet specific wishes or design requirements.
Propeller Blade Manufacturing and Materials

The international standard organization has introduced a series of manufacturing standards in compliance with which propellers have to be manufactured (ISO 484). The accuracy class is normally selected by the customer and the table below describes the range of manufacturing categories.

<table>
<thead>
<tr>
<th>Class</th>
<th>Manufacturing accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>S</td>
<td>Very high accuracy</td>
</tr>
<tr>
<td>I</td>
<td>High accuracy</td>
</tr>
<tr>
<td>II</td>
<td>Medium accuracy</td>
</tr>
<tr>
<td>III</td>
<td>Wide tolerances</td>
</tr>
</tbody>
</table>

If no Class is specified, the propeller blades will be manufactured according to Class I but with surface roughness according to Class S.

Blade materials

Propeller blades are made of either NiAl-bronze (NiAl) or stainless steel (CrNi). The mechanical properties of each material at room temperature are:

<table>
<thead>
<tr>
<th>Material</th>
<th>NiAl</th>
<th>CrNi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yield strength (N/mm²)</td>
<td>min 250</td>
<td>min 380</td>
</tr>
<tr>
<td>Tensile strength (N/mm²)</td>
<td>min 630</td>
<td>660-790</td>
</tr>
<tr>
<td>Elongation (%)</td>
<td>min 16</td>
<td>min 19</td>
</tr>
<tr>
<td>Impact strength (Kv at -10 °C)</td>
<td>21</td>
<td>21</td>
</tr>
<tr>
<td>Brinell Hardness (HB)</td>
<td>min 140</td>
<td>240-300</td>
</tr>
</tbody>
</table>

Both materials have high resistance against cavitation erosion. The fatigue characteristics in a corrosive environment are better for NiAl than for CrNi.

Propeller blades are, to a large degree, exposed to cyclically varying stresses. Consequently, the fatigue material strength is of decisive importance.

The dimensioning of a propeller blade according to the Classification Societies will give a 10% higher thickness for the CrNi compared to NiAl in order to obtain the same fatigue strength.

As an example the thickness and weight difference for a propeller blade for a medium-size propulsion system (4,200 kW at 170 r/min) is stated in table 2.

CrNi-steel requires thicker blades than NiAl-bronze, which is unfortunate from the propeller theoretical point of view (thicker = less efficiency). Additionally, the CrNi is more difficult to machine than NiAl. For operation in ice the CrNi material will be able to withstand a higher force before bending due to its higher yield strength and for prolonged operations in shallow water the higher hardness makes it more resistant to abrasive wear from sand.

The final selection of blade and hub material depends on owners requirements and the operating condition of the vessel. In general terms the NiAl material is preferable for ordinary purposes whereas CrNi could be an attractive alternative for non-ducted propellers operating in heavy ice or dredgers and vessels operating in shallow waters.

<table>
<thead>
<tr>
<th>Ice class</th>
<th>C</th>
<th>1A*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>NiAl</td>
<td>CrNi</td>
</tr>
<tr>
<td>Thickness at r/R = 0.35 mm</td>
<td>132</td>
<td>146</td>
</tr>
<tr>
<td>Thickness at r/R = 0.60 mm</td>
<td>71</td>
<td>78</td>
</tr>
<tr>
<td>Thickness at r/R = 1.00 mm</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Blade weight kg</td>
<td>729</td>
<td>877</td>
</tr>
</tbody>
</table>

Table 2: Classification Society: Det Norske Veritas

Fig. 21: Optimum propeller diameter
Propeller Nozzle

Typical offshore vessels, tugs and trawlers are equipped with nozzles around their propellers to increase the bollard pull and the pull at low ship speeds. Maximising the bollard pull has up to now primarily been a matter of having sufficient power installed with little attention paid to the efficiency of the propulsion system in particular the propeller and its nozzle.

Especially the nozzle ‘type 19A’ developed by Wageningen model basin in the Nederlands has for many years been universally used for all sorts of vessels, partly due to its production friendly design. To less extent the ‘type 37’ nozzle is used, normally where high astern thrust is required.

MAN Diesel, however, has seen the potential for improving the existing nozzle designs, using CFD (Computational Fluid Dynamics) and including optimization of the nozzle supports and nozzle position by tilting and azimuthing.

The newly designed nozzle from MAN Diesel - branded AHT (Alpha High Thrust) - can in combination with the optimum choice of support and tilting angels increase the bollard pull by up to 10% compared to a ‘type 19A’ nozzle with conventional head box support.

The improvements can be obtained if the propulsion system is optimised in conjunction with the hull and shaft line.

Figure 23 shows the AHT nozzle profile compared to a 19A profile.
**Nozzle length**

The fixed nozzles are typically supplied in two standard lengths, either 0.4 or 0.5 x propeller diameter, according to the application.

For low loaded propellers a length of 0.4 x propeller diameter is used and for higher loaded propellers and fluctuations in the wake field it is recommendable to use a nozzle length of 0.5 x propeller diameter.

In special cases the propeller nozzle length may be optimized for the specific vessel.

**Propeller induced pressure impulses and nozzle vibrations**

Since the propeller nozzle has an equalizing effect on the wake field around the propeller, the nozzle has a favourable influence on the propeller induced pressure impulses.

Additionally ducted propellers are lower loaded than open propellers contributing to a lower vibration level.

MAN Diesel can carry out vibration analysis of the propeller nozzle with supports to ensure that the natural frequency of the nozzle and excitations from the propeller does not coincide, fig 25.
Optimizing Propeller Equipment

Propeller design
The design of a propeller for a vessel can be categorized in two parts:
- Optimizing the complete propulsion plant
- Hydrodynamic design of propeller blades

Optimizing the complete propulsion plant
The design of the propeller, giving regard to the main variables such as diameter, speed, area ratio etc., is determined by the requirements for maximum efficiency and minimum vibrations and noise levels.

The chosen diameter should be as large as the hull can accommodate, allowing the propeller speed to be selected according to optimum efficiency.

The optimum propeller speed corresponding to the chosen diameter can be found in fig 18 for a given reference condition (ship speed 12 knots and wake fraction 0.25).

For ships often sailing in ballast condition, demands of fully immersed propellers may cause limitations in propeller diameter. This aspect must be considered in each individual case.

To reduce emitted pressure impulses and vibrations from the propeller to the hull, MAN Diesel recommend a minimum tip clearance as shown in fig 26.

The lower values can be used for ships with slender aft body and favourable inflow conditions whereas full after body ships with large variations in wake field require the upper values to be used.

In twin screw ships the blade tip may protrude below the base line.

The operating data for the vessel is essential for optimizing the propeller successfully, therefore it is of great importance that such information is available.

To ensure that all necessary data are known by the propeller designer, the data sheets on page 24 and 25, should be completed.

For propellers operating under varying conditions (service, max or emergency speeds, alternator engaged/disengaged) the operating time spent in each mode should be given.

This will provide the propeller designer with the information necessary to design a propeller capable of delivering the highest overall efficiency.

<table>
<thead>
<tr>
<th>Hub</th>
<th>Dismantling of cap</th>
<th>High skew propeller</th>
<th>Non–skew propeller</th>
<th>Baseline clearance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>X mm</td>
<td>Y</td>
<td>Y</td>
<td>Z mm</td>
</tr>
<tr>
<td>VBS 640</td>
<td>125</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 740</td>
<td>225</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 860</td>
<td>265</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 980</td>
<td>300</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1080</td>
<td>330</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1180</td>
<td>365</td>
<td>20–25% of D</td>
<td>25–30% of D</td>
<td>Minimum 50–100</td>
</tr>
<tr>
<td>VBS 1280</td>
<td>395</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1380</td>
<td>420</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1460</td>
<td>450</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1560</td>
<td>480</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1680</td>
<td>515</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1800</td>
<td>555</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VBS 1940</td>
<td>590</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig.26: Recommended tip clearance
To assist a customer in selecting the optimum propulsion system, MAN Diesel are able of performing speed prognosis, fig 27, fuel oil consumption calculations, fig 28, and towing force calculations fig 29. Various additional alternatives may also be investigated (ie different gearboxes, propeller equipment, nozzles against free running propellers, varying draft and trim of vessel, etc). Additionally MAN Diesel can assist in the hydrodynamic design of aft ship, shaft and brackets arrangement in order to achieve a uniform inflow to the propeller. In connection with the Alpha propeller, a number of efficiency improving devices have been tested and applied comprising Costa bulbs, tip fin propellers, vortex generators, wake equalizing ducts etc. The experience gained in this respect is available for future projects where such devices are considered.

**Hydrodynamic design of propeller blades**

The propeller blades are computer designed, based on advanced hydrodynamic theories, practical experience and frequent model tests at various hydrodynamic institutes.

The blades are designed specially for each hull and according to the operating conditions of the vessel.

High propulsion efficiency, suppressed noise levels and vibration behaviour are the prime design objectives.

Propeller efficiency is mainly determined by diameter and the corresponding optimum speed. To a lesser, but still important degree, the blade area, the pitch and thickness distribution also have an affect on the overall efficiency.

Blade area is selected according to requirements for minimum cavitation, noise and vibration levels.

To reduce the extent of cavitation on the blades even further, the pitch distribution is often reduced at the hub and tip, fig 30. Care must be taken not to make excessive pitch reduction, which will effect the efficiency.

Thickness distribution is chosen according to the requirements of the Classification Societies for unskewed propellers and complemented by a finite element analysis.

**Cavitation**

Cavitation is associated with generation of bubbles caused by a decrease in the local pressure below the prevailing saturation pressure. The low pressure can be located at different positions on the blade as well as in the trailing wake. When water passes the surface of the propeller it will experience areas where the pressure is below the saturation pressure eventually leading to generation of air bubbles. Further down stream...
the bubbles will enter a high pressure region where the bubbles will collapse and cause noise and vibrations to occur, in particular if the collapse of bubbles takes place on the hull surface. Three main types of cavitation exist - their nature and position on the blades can be characterized as:

- **Sheet cavitation on suction side**
The sheet cavitation is generated at the leading edge due to a low pressure peak in this region. If the extent of cavitation is limited and the clearance to the hull is sufficient, no severe noise/vibration will occur. In case the cavitation extends to more than half of the chord length, it might develop into cloud cavitation. Cloud cavitation often leads to cavitation erosion of the blade and should therefore be avoided. Sheet cavitation in the tip region can develop into a tip vortex which will travel downstream. If the tip vortex extends to the rudder, it may cause erosion, fig. 31.

- **Bubble cavitation**
In case the propeller is overloaded - ie the blade area is too small compared to the thrust required - the mid chord area will be covered by cavitation. This type of cavitation is generally followed by cloud cavitation which may lead to erosion. Due to this it must be avoided in the design, fig. 32.

- **Sheet cavitation on pressure side**
This type of cavitation is of the same type as the suction side sheet cavitation but the generated bubbles have a tendency to collapse on the blade surface before leaving the trailing edge. The danger of erosion is eminent and the blade should therefore be designed without any pressure side cavitation, fig. 33.

By using advanced computer programmes the propeller designs supplied by MAN Diesel will be checked for the above cavitation types and designed to minimize the extent of cavitation as well as to avoid harmful cavitation erosion.

**High skew**
To suppress cavitation induced pressure impulses even further, a high skew blade design can be applied, fig 35. By skewing the blade it is possible to reduce the vibration level to less than 30% of an unskewed design. Because skew does not affect the propeller efficiency, it is almost standard design on vessels where low vibration levels are required.

Today, the skew distribution is of the “balanced” type, which means that the blade chords at the inner radii are skewed (moved) forward, while at the outer radii the cords are skewed aft. By designing blades with this kind of skew distribution, it is possible to control the spindle torque and thereby minimize the force on the actuating mechanism inside the propeller hub, fig 36.

The extent of skew is calculated in each case, by rotating the blade in the specific wake field, for determining the optimum skew.

By using advanced computer programmes the propeller designs supplied by MAN Diesel will be checked for the above cavitation types and designed to minimize the extent of cavitation as well as to avoid harmful cavitation erosion.
Technical Calculation and Services

Arrangement drawings
Provided MAN Diesel have adequate information on the ship hull, an arrangement drawing showing a suitable location of the propulsion plant in the ship can be carried out with due consideration to a rational layout of propeller shaft line and bearings.

In order to carry out the above arrangement drawing MAN Diesel need the following drawings:
- Ship lines plan
- Engine room arrangement
- General arrangement

Moreover, to assist the consulting firm or shipyard in accomplishing their own arrangement drawings, drawings of our propeller programme can be forwarded. The disks are compatible with various CAD programmes. Should you require further information, please contact us.

Installation Manual

After the contract documentation has been completed an Installation Manual will be forwarded. As an option the manual will be available in electronic format via our ExtraNet offering you the advantage of easy and fast access to the documentation. When the documentation is released your user name and password for access to your personal folder will be forwarded by separate e-mail.

The Installation Manual will comprise all necessary detailed drawings, specifications and installation instructions for our scope of supply. The manual is in English language.

CAE programmes are used for making alignment calculations, epoxy chock calculations, torsional vibration calculations etc. In the following a brief description is given of some of our CAE programmes and software service.
### Alignment instructions

For easy alignment of the propeller shaft line, alignment calculations are made and a drawing with instructions is given in the Installation Manual, fig 38. The alignment calculations ensure acceptable load distribution of the stern tube bearings and shaft bearings.

### Torsional vibrations

A comprehensive analysis of the torsional vibration characteristics of the complete propulsion plant is essential to avoid damage to the shafting due to fatigue failures.

Based on vast experience with torsional vibration analysis of MAN B&W two-stroke and MAN Diesel four-stroke propulsion plants, the VBS propeller equipment is designed with optimum safety against failure due to fatigue. Stress raisers in the shafting or servo unit are minimized using finite element calculation techniques.

When the propeller is delivered with a MAN Diesel or MAN B&W engine a complete torsional vibration analysis in accordance with the Classification Society rules is performed. This includes all modes of operation including simulation of engine misfiring.

When the total propulsion plant is designed by MAN Diesel, the optimum correlation between the individual items exists. The extensive know-how ensures that the optimum solution is found as regards minimizing stresses in connection with torsional vibration calculations. Fig 39 shows the result of a torsional vibration calculation.

When propellers are supplied to another engine make, a complete set of data necessary for performing the analysis is forwarded to the engine builder in question, fig 40.

---

**Fig. 38: Calculated reactions and deflections in bearings**

<table>
<thead>
<tr>
<th>Bearing</th>
<th>Bearing reaction [kN]</th>
<th>Vertical displacement [mm]</th>
<th>Angular deflection [mRad]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aft sterntube bearing</td>
<td>51.55</td>
<td>0.00</td>
<td>-0.476</td>
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<tr>
<td>Fwd sterntube bearing</td>
<td>22.81</td>
<td>0.00</td>
<td>0.221</td>
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<tr>
<td>Aft main gear bearing</td>
<td>15.67</td>
<td>0.70</td>
<td>0.007</td>
</tr>
<tr>
<td>Fwd main gear bearing</td>
<td>15.16</td>
<td>0.70</td>
<td>-0.003</td>
</tr>
</tbody>
</table>

**Fig. 39: Torsional vibration calculation**

[Graph showing torsional stress amplitude vs. engine speed]

- Rule limit for transient running
- Rule limit for continuous running
- Actual stresses
- Barred speed range
Propeller data

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<thead>
<tr>
<th></th>
<th>kgm²</th>
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<tr>
<td>Inertia in air</td>
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<tr>
<td>Inertia in water (full pitch)</td>
<td>39300</td>
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<td>Inertia in water (zero pitch)</td>
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<tr>
<td>Number of blades</td>
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<tr>
<td>Propeller diameter</td>
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<td>Design pitch</td>
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<tr>
<td>Expanded area ratio</td>
<td>0.48</td>
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<td>Propeller weight (hub + blades)</td>
<td>22230</td>
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Shaft data

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<tr>
<th>Shaft section</th>
<th>Material</th>
<th>Tensile strength N/mm²</th>
<th>Yield strength N/mm²</th>
<th>Torsional stiffness MNm/rad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propeller shaft</td>
<td>Forged steel</td>
<td>min 600</td>
<td>min 350</td>
<td>K1 99.0</td>
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<tr>
<td>Servo unit</td>
<td>Forged steel</td>
<td>min 740</td>
<td>min 375</td>
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<td>Intermediate shaft</td>
<td>Forged steel</td>
<td>min 600</td>
<td>min 350</td>
<td>K3 105.6</td>
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</table>

Whirling and axial vibration calculations

Based on our experience the propeller equipment and shafting are designed considering a large safety margin against propeller induced whirl and axial vibrations. In case of plants with long intermediate shafting or stern posts carried by struts, a whirling analysis is made to ensure that the natural frequencies of the system are sufficiently outside the operating speed regime.

Propeller induced axial vibrations are generally of no concern but analysis of shafting systems can be carried out in accordance with Classification Society requirements.

Instruction Manual

As part of our technical documentation, an Instruction Manual will be forwarded.

The Instruction Manual is tailor-made for each individual propeller plant and includes:

- Descriptions and technical data
- Operation and maintenance guidelines
- Work Cards
- Spare parts plates

As standard the manual is supplied in a printed version, and can as an option be forwarded in electronic document format.
<table>
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<th>HUB VBS-Type</th>
<th>Max shaft Diameter [mm]</th>
<th>ODS/ODG Type</th>
<th>A [mm]</th>
<th>*B [mm]</th>
<th>L [mm]</th>
<th>**M [mm]</th>
<th>*W-min ODS [mm]</th>
<th>W-min ODG [mm]</th>
<th>***F ODF [mm]</th>
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<td>3001</td>
<td>2051</td>
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</tr>
</tbody>
</table>

* Guiding approx dimensions, **M-measure for standard shaft seals, ***F-measure is minimal required space for dismantling
For propeller layout please provide the following information:

1. $S: ________ \text{ mm}$  $W: ________ \text{ mm}$  $I: ________ \text{ mm}$ (as shown above)
2. Stern tube and shafting arrangement layout
3. Stern tube mountings: Exopy mounted or interference fitted
4. Propeller aperture drawing
5. Copies of complete set of reports from model tank test (resistance test, self-propulsion test and wake measurement). In case model test is not available section 10 must be filled in.
6. Drawing of lines plan
7. Classification society: ______________ Notation: __________ Ice class notation: ______________
8. Maximum rated power of shaft generator: ________ kW
9. To obtain the highest propeller efficiency please identify the most common service condition for the vessel:
   Ship speed : ________ kn  Engine service load : ________ %
   Service/sea margin : ________ %  Shaft gen. service load : ________ kW
   Draft : ________ m

Project: _________________________ Type of vessel: _______________________
10. Vessel Main Dimensions  *(Please fill-in if model test is not available)*

<table>
<thead>
<tr>
<th>Nominal</th>
<th>Dimensions</th>
<th>Ballast</th>
<th>Loaded</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars</td>
<td>$L_{PP}$</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Length of load water line</td>
<td>$L_{WL}$</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Breadth</td>
<td>$B$</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Draft at forward perpendicular</td>
<td>$T_F$</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Draft at aft perpendicular</td>
<td>$T_A$</td>
<td>m</td>
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<tr>
<td>Displacement</td>
<td>$\bar{N}$</td>
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</tr>
<tr>
<td>Block coefficient ($L_{PP}$)</td>
<td>$C_B$</td>
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<tr>
<td>Midship coefficient</td>
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<tr>
<td>Waterplane area coefficient</td>
<td>$C_{WL}$</td>
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<td>Wetted surface with appendages</td>
<td>$S$</td>
<td>m$^2$</td>
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<td>Bulb section area at forward perpendicular</td>
<td>$A_B$</td>
<td>m$^2$</td>
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</table>

11. Comments:

_________________________________________________________________
_________________________________________________________________
_________________________________________________________________

Date:_________________________          Signature:___________________________