

Integration of the Rudder Propeller into the Ship's Structure

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SYNOPSIS

The continuing trend for more powerful engines, coupled with larger revs/min, can cause problems with vibration in tugs, especially when installed in smaller, lightweight hulls. At the same time, the demand for smooth running, combined with low vibration levels, is important for reasons of comfort, as is the awareness of the influence of vibration on a component's lifetime.

At each point where the propulsion systems interact with the ship's, vibration will inevitably be felt within the structure. The ship's construction should be such that it can absorb these forces. On the other hand, the structural design of mainly the shaft line foundation and rudder propeller ambient structure significantly influences the vibration level of the tug's hull. It should therefore be subject to precise engineering not only with regard to strength but also to rigidity.

This paper describes aspects of the mechanical integration of rudder propeller propulsion systems for tugs. It shows modern azimuth propulsion systems (azimuth propellers and shafting systems) and will highlight the advantages and dangers of different installations of shaft lines and rudder propeller fastenings. It will show what can go wrong and will present examples of shafting systems that are less sensitive to misalignment. The paper will also provide an insight into the typical excitation of ducted azimuth thrusters. It will show some results of investigations into the magnitude of these excitations. By means of actual analyses, examples will be given of good and inadequate constructions.

INTRODUCTION

Schottel rudder propellers have been serving tugs for more than 40 years and, with a total number of some 800 tugboats equipped with Schottel rudder propellers, the company has gained wide experience in supplying complete azimuth propulsion systems. These consist of fixed and controllable pitch rudder propellers, shaft lines, hydraulic systems for steering, clutch and brake, plus complete electric control systems, including, for example, electrically-driven azimuth steering, modern bridge panels and the popular Schottel Masterstick multi-unit control system.

Over the years, the tugboat has developed from a simply designed workhorse for use mainly in harbour areas to a highly engineered, complex ship system with varied functions and specialised tasks. Both working hours and power density have increased while hull sizes are often limited by reasons of manoeuvrability or of cost (see *Figure 1*).

The tug industry's need for greater engineering expertise means increased engineering support and continuous development in the essential disciplines of mechanical, hydraulic, electrical and hydrodynamic engineering. Schottel has built up specialised development teams of experienced engineers in these four key areas, supplying the company with the engineering support that is needed to serve its customers' needs. As well as first-class technical support for every individual order, the company has employed a team of engineers to ensure a total technical review of customers' installation. As a result, Schottel ensures its research and development departments bring their state-of-the-art knowledge to each phase of every customer's project.

Typically, this engineering support for a standard tug application includes:

- Bollard-pull calculation and propeller blade design (optional CPP propeller pitch range determination);



Figure 1, from left, **Janus**, L: 24m, B: 8.50m, 2x 360kW, BP: 12t; **Sinalda**, L: 33m, B: 11m, 2x 1,850kW, BP: 65t; **Wesley**, L: 30m, B: 12m, 2x 2,350kW, BP 75t.

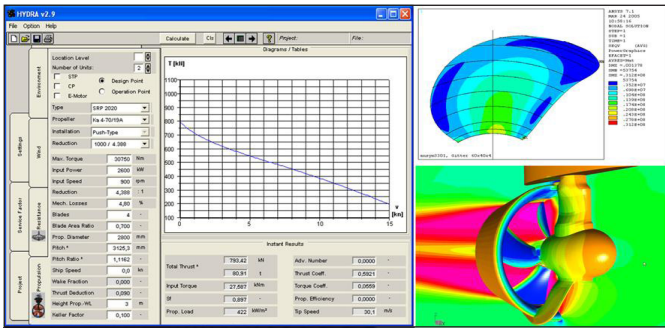


Figure 2: Bollard-pull calculation, blade design and CFD analysis of the thruster.

Design and calculations take into consideration motor capacity, thruster-hull interaction, space and draft. Detailed 3D propeller models are given to the foundry to ensure precise propeller geometry.

- Electrical design of interfaces, switch boxes, operation panels and superior control systems;



Figure 3: Standard panel, modern compact switchboard with bus control, panel layout with Masterstick.

Based upon standard drop-in panel components and a modular switchboard, units can be arranged to fit in with any bespoke design. The CAN-bus system offers a flexible interface, plus secure data and signal transmissions. The generation 3-Masterstick system allows single-handed operation of the ship.

- Hydraulic and pneumatic engineering for steering, clutch, brake;

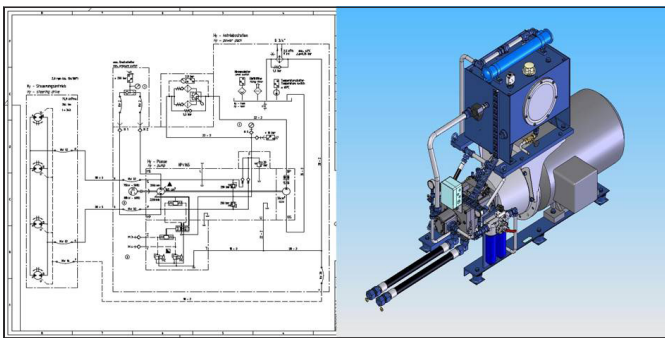


Figure 4: Plan for hydraulic steering, standard hydraulic aggregate.

Hydraulic plans and 3D construction of the hydraulic aggregate give the designer exact interfaces.

- Check resistance of the mechanical connection of the thruster to the hull;

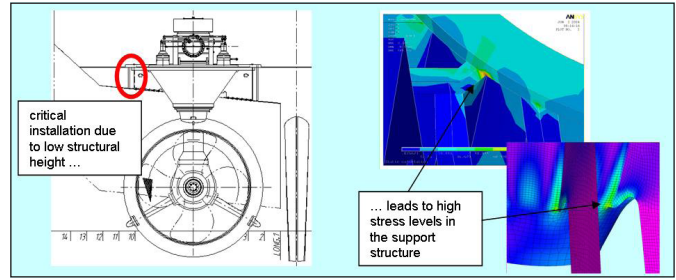


Figure 5: Critical structural height, FEM-calculation of critical installation.

Design of the shaft line includes:

- Strength requirements of the classifying authority;
- Distance and height differences between diesel and SRP;
- Axial movements of the components;
- Rigidity of the foundations and bulkheads;
- Excitations by motion and cardan joints;
- Thermal expansion of the shaft line itself;
- Torsional Vibration Calculation/Lateral Vibration Calculation/Axial Vibration Calculation.

The following graphics show a typical installation space for a medium-sized stern tug application, an example of a mass elastic system of a tug drive line and the result of a modal analysis of an intermediate shaft.

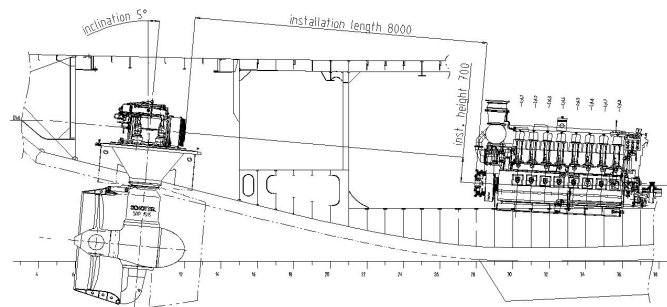


Figure 6: Typical installation space for a shaft line in a stern tug.

The distance between motor and thruster, the axis offset of crankshaft to thruster input shaft and eventually the angle of motor and/or thruster are the initial figures needed to make up a shaft line concept.

A Torsional Vibration Calculation is made for each individual project. Focus is on the dynamic torque load of the thruster manufacturer's remit of supply as a counter-check of the official TVC, which is in the responsibility of the motor supplier. Torsional vibrations are not handled in this paper.

For an image of TVC mass elastic system of a tug drive see Figure 7, at the end of the paper.

Excessive torsional vibration can also affect the shaft line, the motor or even the thruster, resulting in vibration being seen, for example, at the bottom of the shaft line. When examining lateral or axial shaft line vibrations, the torsional vibration system should be checked beforehand to prevent wrong conclusions being drawn.

For an image of LVC vibration mode of a typical fourfold-supported intermediate shaft see Figure 8 at the of the paper.

Different from the TVC, which requires data exchange between motor and thruster maker only, the Lateral Vibration Calculation has an interface to the designer/yard through the draughtsman's estimates of load bearing. There is always a great deal of information for the azimuth issues exchanged between the technical engineers.

To get an overview of Schottel's involvement in the engineering decisions regarding responsibilities and information flow during the project phase, a simple organisation chart may help.

For an image of Organisation chart in chronological order see Figure 9 at the end of the paper.

This paper will address the mechanical engineering, especially the mechanical integration, of rudder propellers (SRP) and shaft lines in tugs only. The most important document given to the customer is a detailed installation drawing showing the mechanical components of the ship's structure, as supplied by the customer.

When the installation drawing is accepted, the initial phase is complete. This period of time should be used by both the drive suppliers and the customer to get all relevant aspects of the installation fixed to ensure trouble-free installation, commissioning and operation.

The following will identify aspects and problems that have to be considered. Thumb values and examples will be given, but it should be noted that, despite the use of standard components and methods, each installation has its own design anomalies. Again, these require a clear and mutual understanding of the questions to be clarified on each side. This paper will help to identify the questions that should be asked.

BESPOKE CALCULATIONS

Extent of calculations by thruster manufacturer

For each order, certain calculations are made to ensure correct engineering of the propulsion system:

- Calculation file of the SRP product is a standard package supplied to the classifying authority;
- Check strength of thruster-to-hull connection (bolting, welding, structure);

- Calculation of strength of shaft line;
- Lateral vibration calculation of shaft line (with assumed or known rigidity values);
- Axial vibration calculation of shaft line (axial rigidity of bulkheads and fixed bearing foundations must be supplied by customer);
- Check of deflection/displacement capacity, both radial and axial – (motor movements of elastically-supported motors must be supplied by customer);
- TVC (torsional vibration calculation) as a counter-check to the official TVC supplied by the motor manufacturer to the classifying authority.

The extent of the calculations is managed by a team of engineers in the production department who are responsible for the technical side of the order and liaise as required between the customer, classifying authority and the experts in the development departments.

To find the best solution for each individual application, a certain amount of information is also needed from the customer's side.

Recommended extent of calculations and information from the customer's side

- Check structural strength or SRP foundation (hull-sided);
- Check of modal behaviour of the SRP units in the hull structure (resonances with first and second blade harmonic to be avoided);
- Shaft foundation rigidity to be guaranteed (design to 500 kN/mm – min. 300 kN/mm);
- Supply axial rigidity of fixed bearing foundations and bulkhead to thruster manufacturer;
- Supply motor movements (Sixth-degree-of-freedom calculation) to Schottel.

The above list is limited to the mechanical aspects of drive integration into the tug hull. Strength aspects are generally controlled and observed by the classifying authority.

With regard to thruster and shaft line dynamics, there is room for improvement and data and knowledge exchange should be improved by all parties involved.

THE SCHOTTEL MODULAR SHAFT SYSTEM

Components used

There are, as can be imagined, a large variety of designs and installations, creating demand over the years for a huge portfolio of components. This not only hinders smooth and logistical ordering but also increases the risk of technical problems. It was therefore decided to develop a standardised modular shaft system with a limited number of components, proven in tug applications.

The “construction kit” for shaft lines comprises the following elements:

Bearings:

For the loose bearing position, split cylinder roller bearings and spherical bearings, both different sections are used. For the fixed bearing position, the same types of bearing are used with an axially fixed outer ring. Additionally, a split double taper roller bearing (developed and tested jointly with Cooper Roller Bearings Ltd) is available in all sizes, allowing for high axial loads at a low minimum-bearing load.

For an image of Types of bearing used in Schottel shaft lines see Figure 10 at the end of the paper.

Shafts

Solid forged steel shafts made of 42CrMo4V, with a maximum length of 6m, are used mostly in the lower speed range, as in stub shafts or for higher speed range with short shaft lengths. Flanges are shrink-fitted with conical oil pressure assembly as a standard. Seamless welded St52-3 with welded solid shaft/flange ends made of 42CrMo4V are mostly used as intermediate shafts, mainly for the higher speed range and for large installation lengths. Distances of some 6-7m are spanned with hollow shafts with only two bearings. Using split bearings allows the shaft flanges to be integrated into the welded end pieces. Composite shafts are normally used in special applications. For the tug application, a steel shaft is clearly more robust and thin-walled hollow steel shafts offer most of the advantages of the composite solutions at lower cost.

For an image of Standard shaft types used in the modular shaft line system see Figure 11 at the end of the paper.

Compensation couplings

To compensate for movements and deflections within the shaft line system a defined variety of components is currently used for the standard installations. Cardan shafts, with the disadvantage of own excitation, are used to compensate for height differences between diesel flywheel and thruster input flange. For the preferred straight shaft applications, steel link couplings with metal joints (lower displacement capacities) or rubber joints (higher displacement capacities) are used at different positions in the shaft line. As these couplings are radially rigid, they can only compensate axial and bending deflection when used singly. To compensate radial deflections or to adjust small height differences between shafts they are used in a double arrangement, with a piece of shaft in between to adjust the radial displacement capacity at a given maximum bending angle of the coupling. Toothed couplings are not often used as they again require regular maintenance and normally only have limited radial displacement capacity. Nevertheless they are often used to link together two pieces of shaft (for example, to a longer intermediate shaft).

Figure 12 gives an overview of the typical maximum displacement values of the different coupling types and the required alignment accuracies.

For an image of Couplings used in the modular shaft line system see Figure 12 at the end of the paper.

The coupling data in Figure 12 is of general value only. Specific values have to take the respective size into account. A particular advantage of the double arrangement with steel link or rubber link couplings is that they do not produce excitations as with the cardan shaft, but have sufficient displacement capacities as required by the installation. It is important to differ between alignment tolerances and displacement capacity in operation. Any alignment needs to leave sufficient margins for operational differences.

The software tool

To make the choice of components quick and easy, and have all necessary data available on one database, Schottel uses its own software tool for the composition of the shaft line. The so-called “shaft line generator” gives the composition of the shaft line with shaft strength, lateral vibration safety, bearing lifetime and adequate cardan shaft size. The basic strategy is to use up to three bearings per shaft, where the centre bearing is the fixed bearing ensuring the minimum bearing load required.

For an image of User interface of the shaft line generator see Figure 13 at the end of the paper.

The user can choose the order of the main data, power and speed and the dimensional restrictions of the installation space. By adding preferences that can be made with help functions from the programme, the user will have a number of useful solutions for the power transmission task. Once the data is generated, further calculations such as TVC and LVC can be done on stored databases and installation drawings can be produced for the customer.

STATE-OF-THE-ART INSTALLATIONS

Principal installation methods of the thruster

Dependent upon the maintenance concept and restrictions in construction of the tug design overall, the principal installation method can be chosen to be either from above or below.

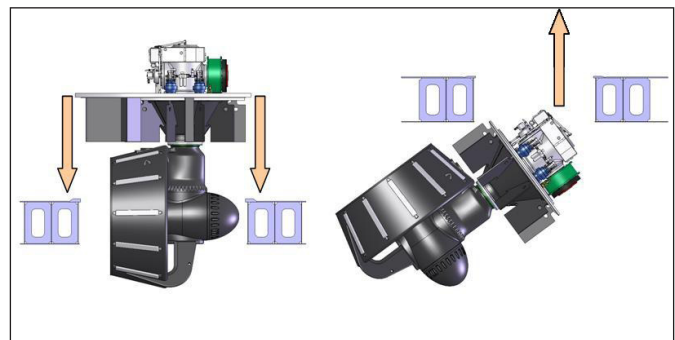


Figure 14: Installation of the azimuth thruster from above and below.

In both cases the connection to the hull has to be designed to be:

- Watertight under pressure;
- Of sufficient strength;
- Of sufficient rigidity.

Installation from above:

Owing to the large size of the structure supplied with the thruster there is a sufficient part of the stiffness already delivered with the thruster itself. Radial web connection to the hull is therefore not normally necessary.

Installation from below:

For this method, the thruster only needs to be connected to the hull flange via its top-plate from a strength point of view. With respect to rigidity and consequently the dynamic behaviour of the thruster in the structure, it is sometimes useful or necessary to also connect the brackets of the thrusters support structure radially with the trunks' cylindrical wall (see *Figure 15a*). Care has to be taken to align the brackets with the hull structures' counter webs (see *Figure 15b*).

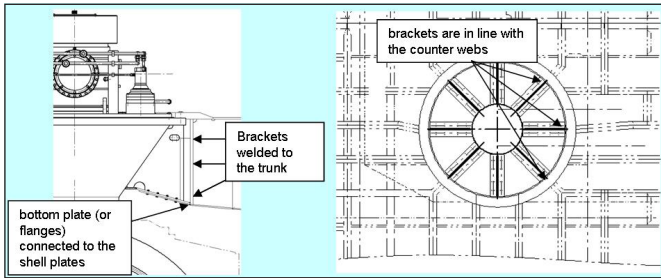


Figure 15a.

Figure 15b.

Compared to the fixation of the main flange, only the bending stiffness increases by approximately by 10-25 per cent when the webs are connected. This results in an offset to higher frequencies of the pendulum mode of the thruster by some 5-10 per cent. Pendulum mode means that the underwater mass of lower gear box, propeller and nozzle as a whole swing in the stiffness of mainly the steering tube of the thruster.

Installation variants

The graphics below show the standard fixation methods of the thruster to the hull structure. From the top left to the bottom right welded connection, bolted steel flange connection with O-ring sealing and bolted rubber flange connection with spacers that define the compression of the rubber flange are shown. A formerly used method, the once-popular epoxy cast foundation is no longer recommended because the other three variants meet the yard requirements with less effort.

All these installation methods can be used either with or without additional bracket connections in the trunk. As the brackets are also used to hold in the bottom cover plate of the hull plating, they are viewed as a standard feature of the support structure of the thruster.

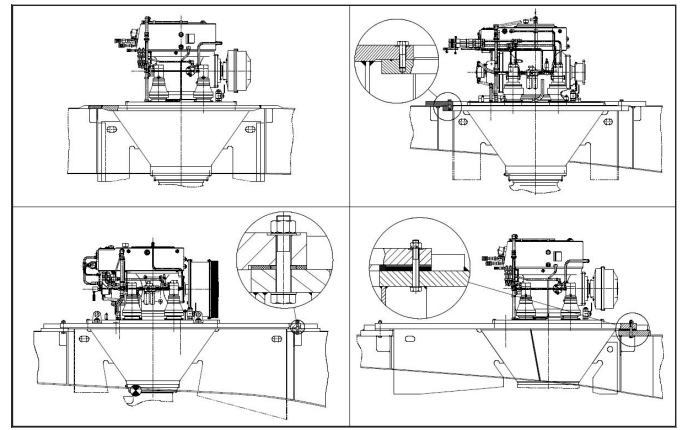


Figure 16: Installation variants for the thruster-hull connection.

Design goal for thruster foundation stiffness

Finally, with a specific thruster design, the mass of the thruster and its mass moments of inertia, especially around x- and y-axis (pendular modes), are known. With the thruster's own vertical (z-direction) and bending stiffnesses (x-, and y-directions) linked to the hull structures' stiffness the Eigenmodes of the thruster are already determined. Furthermore the main excitation frequencies of the propeller (in order of the blade numbers one, two and three) are given by choice of the number of blades (four as a standard, sometimes five as required by the TVC). There are therefore not many options left to tune other than the constructive stiffness of the thruster foundation and, optionally, the connection of the thruster brackets to the hull.

Considering the thruster's own mass-elastic data and its excitation frequencies some typical required values for the thruster foundation stiffness for the power range of 1,000 to 3,500kW can be found in:

$$c_{xx} \text{ \& } c_{yy} > 3 \times 10^{10} \text{ [Nm/rad]} \quad c_z > 3,5 \times 10^9 \text{ [N/m]}$$

Detailed calculations should be made to avoid global effects to the three critical mode shapes with the azimuth propeller orders to avoid needless high vibration levels in the ship or consequential damage in mainly thruster bearings and shaft line components.

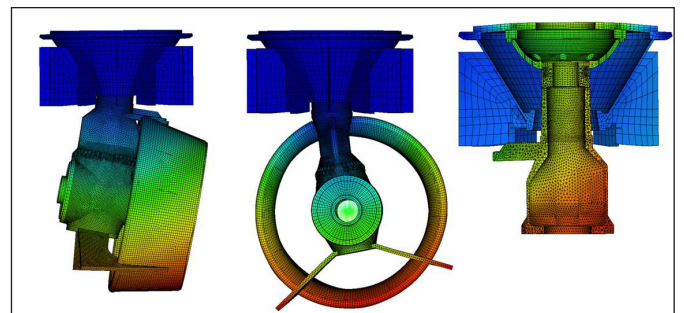


Figure 17: Eigenmodes of the thruster – bending in x- and y-directions, pounding in z-direction.

Traditional and modern shaft line arrangements

For many years the tug drives with azimuthing thrusters were built up with a limited number of diesel motor types and with a tight speed range of approximately 600-1,000 rev/minute.

A typical shaft line was constructed using:

- Massive forged steel shafts only;
- Cardan shafts in Z- or W-arrangements;
- All displacements were taken by the cardan shaft splined connections;
- Spherical roller bearings were used as fixed and loose bearings;
- Standard lithium-soaped multi-purpose grease was used;
- Foundation stiffnesses of 500 kN/mm were commonly reached with massive foundations;
- Bulkhead bearings were meant to be “no go” owing to the radial stiffness deviance compared to the other bearing foundations.

Using a standard foundation stiffness and massive cylindrical shafts, which were considered as simple “bending springs”, the choice of the bearing distance (and hence number of bearings) was a simple job that could be solved by means of the diagram below:

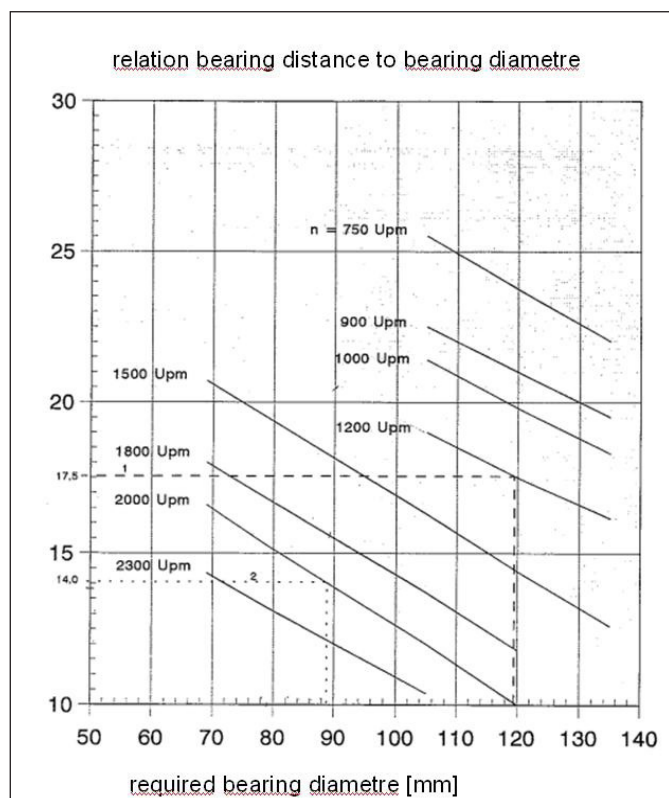


Figure 18: Selection diagram for bearing distance by shaft diameter and speed.

The higher the speed and the longer the shaft line, the more bearings were used to solve the task, which was fine until 1,600 to 1,800 rev/min diesel became widely used. In extreme cases it became necessary to use up

to six bearings in one shaft line and up to 10 bearings in the arrangement of one azimuth drive. This demands a highly competent basic design and requires extensive alignment, which cannot always be realised in practice.

Figure 19 shows a traditional arrangement, often also used with higher rev/min. With a good basic design, proper alignment and maintenance, many of these installations are in operation without problems. But it must be said that the most severe problems happen with arrangements in high-speed applications: complex to analyse and difficult to solve.

For an image of Extreme example of a traditional high-speed arrangement with eight bearings in the shaft line see Figure 19 at the end of the paper.

Today the aim is to solve the power transmission with the lowest possible number of parts (especially bearings) that require maintenance and to ease the installation and alignment. Every bearing position is also a potential excitation contact for structural noise and requires maintenance, alignment and foundation and is subject to wear and therefore has a limited lifetime. Modern installations do not have a cardan shaft any more because it is a vibration exciter on its own when used with typical deflection angles of 4 to 8 degrees. By bringing the thruster into a hydrodynamically acceptable inclined position of some 3 to 6 degrees (which also needs to be applied for the motor) a straight shaft line is reached. Using hollow shafts made of seamless tubes of high strength allows much wider spans of these shaft lines due to their high bending stiffness and low mass without a reduction in the natural frequencies of the shaft line.

This is the way to achieve well-arranged installations which are easy to install, alignment tolerant and with low maintenance.

For an image of Optimum example of a modern high-speed arrangement with one bearing on 8m installation length see Figure 20 at the end of the paper.

In Figure 20, the one and only bearing in the shaft line is located in the bulkhead. So the bearing housing flanged into the bulkhead is serving a double function as a bulkhead sealing. The only real work in the whole shaft line is a hole cut into the bulkhead and occasionally some stiffening flat steels to tune the bulkheads membrane stiffness.

TYPICAL PROBLEMS, OR WHAT CAN GO WRONG?

Attaching the thruster – welded version

The thruster installation via a fully welded connection causes only few problems when used daily. Yards are used to welding. Once the thruster is put into the correct position and deflections of the welding process are controlled within allowable alignment tolerances, the installation offers a

very rigid connection to the ship which is normally resonance free for the azimuth main modes of the thruster. The installation method does not require any machining of the thruster trunk. Nevertheless this installation method is used in only 20-30 per cent of the delivered units.

In spite of the advantages mentioned above, design or installation failures can also happen with this installation:

- Defective alignment (which is difficult to correct afterwards);
- Ship-sided flange too thin, too short or with too much overhang;
- Insufficient connection of the webs, or webs too weak.

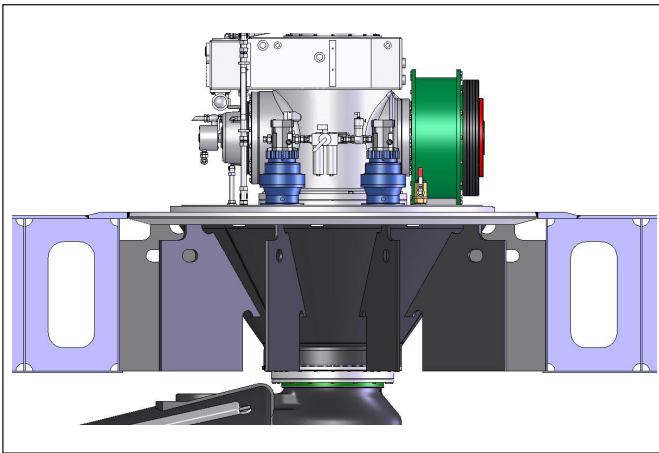


Figure 21: Proper installation of fully welded model with smooth changeover from thruster to well flange and full connection of brackets.

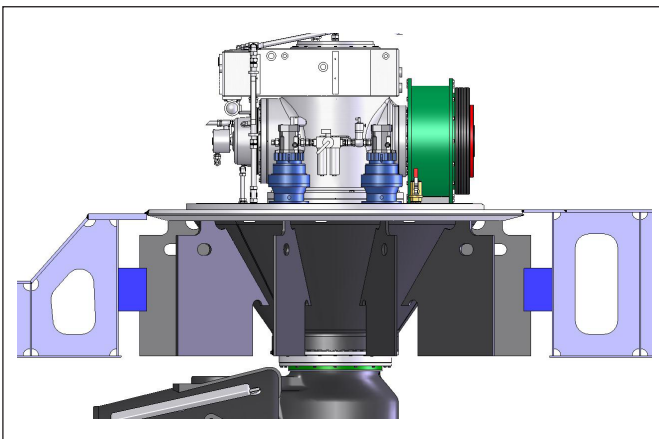


Figure 22: Faulty installation of fully welded model with well flange too thin or with too much overhang/ connection of brackets too weak with shortened lashes.

For alignment of the thruster it is useful to attach provisional installation arms to the thrusters' main flange, which should be set onto adjustable hydraulic plungers or threaded spacers to allow accurate vertical and angular positioning of the heavy unit prior to fixing by welding (See Figure 23).

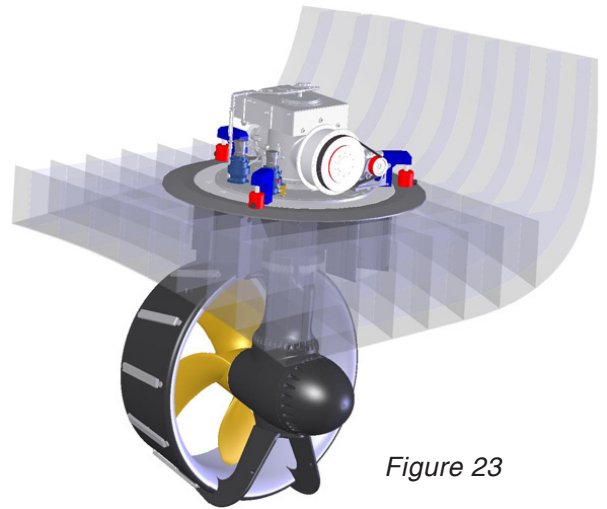


Figure 23

Depending on how the alignment procedure is carried out, a certain vertical, horizontal and angular failure is inevitable and will alter again when the vessel is launched owing to the different hull shape of a dry-docked or a floating tug. Using long hollow shafts with none, or only one, bearing position that can be averaged between motor and thruster makes the whole installation of the shaft line easily achieved. Perfectly straight installation, with almost no unnecessary bending deflection load onto the hollow shafts and their deflection couplings, is reached when the motor is finally positioned after welding the thruster by using, for example, adjustable motor mounts (both rigid or flexible). Care should be taken to take note of the aligned motor position when using flexible mounts. This helps to compensate when setting the rubber elastic motor mounts after a period of time. With height-adjustable flexible mounts these settings should be realigned after time.

Attaching the thruster – bolted version with O-ring sealing

Although this installation method requires proper machining of the ship's flange, it is our customers' most popular version, especially when installed from below (about half the orders). The accuracy requirements of approximately 0.15-0.2mm smoothness and $R_a=3,2\mu\text{m}$ unevenness normally requires machining of the flange *in situ* which is done by specialised companies throughout the world (see Figure 24).

Problems appear when the flange or the trunk is machined in the workshop and afterwards welded into the structure. In most cases the flange is then out of tolerance and the units bolted down to the flange by the high clamping forces of the forceful main bolting. This results in damage to the steering gear and causes hidden problems such as wear etc.

For an image of Typical requirements for flange machining see Figure 24 at the end of the paper.

The typical shape failure of such a machined flange resembles an '8' and can easily reach deflections of a few millimetres, which is obviously the multiple value

of the tolerance. In such cases, as well as in the main bolting of the thruster, it becomes unsafe because the clamping force is used for flange deformation instead of supplying correct and equally distributed friction force over the flange circumference (see *Figure 25*).

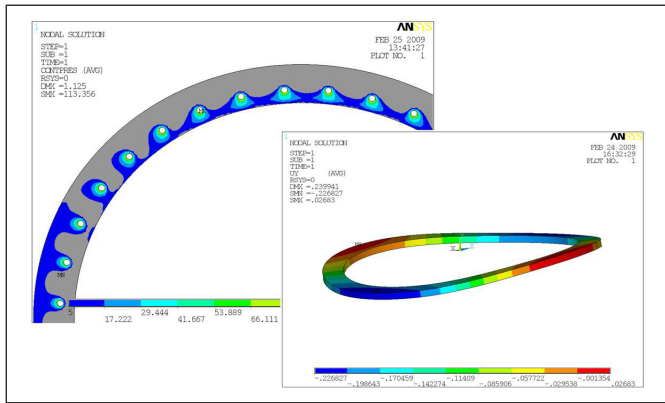


Figure 25: Uneven bolt pre-stress after setting bolts by typical flange deformation due to welding stress.

Attaching the thruster – bolted version with rubber flange installation

The rubber flange method is more tolerant with regard to unevenness and surface quality of the ship flange owing to the spot contact of the bolting, some compensation comes from the flanges themselves (by bending deformation). Sealing is not a problem because of the high compression value (at least 2mm) of the 10mm rubber flange. Unevenness of at least 1.5mm can be supported by this installation method. Even higher unevenness values can be tolerated by using thicker rubber flanges of some 15 or 20mm, where 20 per cent compression is used and up to 3mm unevenness can be equalised.

When making use of these extended tolerances with the thicker rubber flanges, individual machining of the steel spacers between thruster flange and ship flange is required to ensure proper loading of each bolting position and to prevent unacceptable thruster flange deformation.

The gap of each numbered bolting position therefore has to be measured and the spacers (see “washers” in *Figure 26*) have to be supplied with overmatter to allow for quick and easy turning to measure on the lathe.

NB: Stoppers should be used in a radial direction to centre the flange.

For an image of Rubber flange installation showing rubber flange, bolts, spacers and stoppers see Figure 26 at the end of the paper.

The rubber flange installation method has improved in the past few years and accounts for some 20 to 30 per cent of deliveries today. The installation method normally allows the pre-machining of the ship’s flange in the workshop prior to welding-in (welding sequence has to be observed).

Attaching the thruster – global resonance of the thruster in the ship’s structure

Beyond the installation method, the worst thing that can happen is a global resonance of the thruster unit(s) in the ship’s structure. The thruster, together with its sister unit, can perform pendulum and vertical vibrations both synchronously and in opposite phase. This delivers normally at least six relevant Eigenmodes of the thruster pair in the ship’s structure.

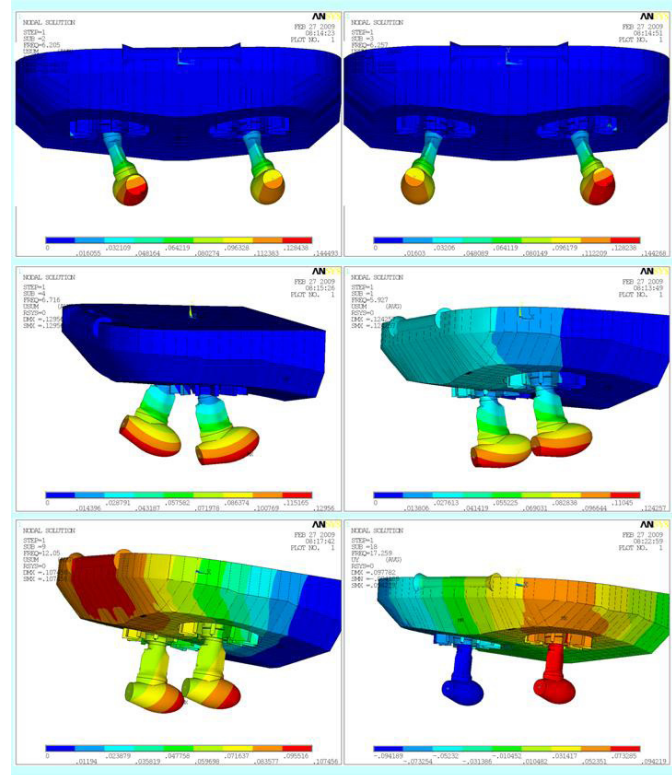


Figure 27: Relevant Eigenmodes of the thruster pair in the ship’s structure.

To determine these Eigenmodes, the structural 3D model of the tug aftship (normally available) and a reduced model of the thruster is used to keep the whole model simple and quick to calculate. Mass supplements for the submerged surfaces have to be made as the system oscillates in water.

When calculating pendulum modes of the thrusters, the hydrodynamic mass supplement for the hull plating can be neglected because of the more localised uneven area of the hull shell. For the thruster submerged body, the water supplement should be added for more exact analysis. In the case of the pounding modes, the water supplement becomes more important for the hull shell and less important for the thruster. As long as comfortable distances to resonances are reached (eg >20 per cent) water supplements do not need to be applied. There are various recommendations available for water supplements for different submerged shapes eg DNV offers values in classification note 30.5.

The reduced thruster model is a simplification of the thruster detailed as a finite-element-model to a pendulum with similar properties.

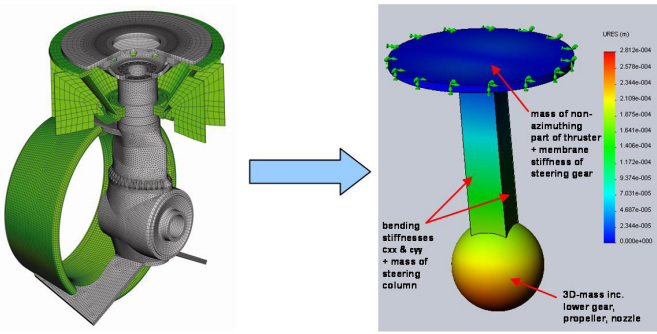


Figure 28: Conduction of the reduced thruster model from the detailed 3D-FEM-model of the thruster.

The task is to prevent resonances of the propeller blade orders with the above shown Eigenmodes. The only parameters to be tuned are the stiffness of the hull fixation and foundation and the number of propeller blades (normally four or five). Concerning the number of propeller blades, the TVC may also require a specific blade number with regard to torsional resonances in the drive train which has to be dealt with.

Shaft line – lubrication problems

Roller bearings lubricated with grease are used as standard for shaft lines in tug installations. It may appear trivial but lubrication problems still occur regularly – insufficient lubrication of the roller bearings owing to thin or insufficient amount of grease is typical. But even overfilling bearing housings can cause operation/maintenance problems after a time. Most problems occur with overheated grease in high rev/min applications (1,600/1,800 rev/min). Few cases are known of insufficient maintenance because essential maintenance of the bearings seems to be obvious to the operator and bearings are normally easy to access and under daily observation.

Bearing housings with a large “dead space”, compared to the grease filling recommendation, may be problematic, especially when using high temperature grease with thicker consistency at operational temperatures (see Figure 29). Fewer problems are experienced from bearing housings with smaller “dead space” for high-speed applications.

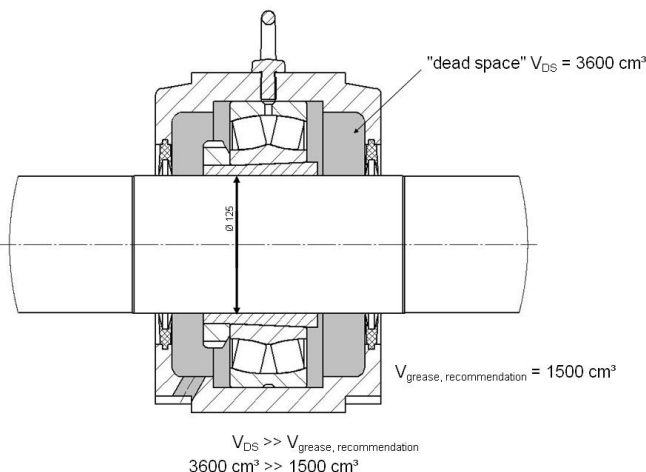


Figure 23

Schottel, meanwhile, uses high temperature grease as standard in all shaft-lines that are supplied to our customers ready-assembled. Standard grease is adequate and can be used for speeds up to 1,000/1,200 rev/min. Maintenance manuals should be consulted and individual service support used in cases of doubtful lubricant or bearing condition or excessive operation temperatures (after running-in).



Figure 30: Overheated grease in under-loaded high-speed application.

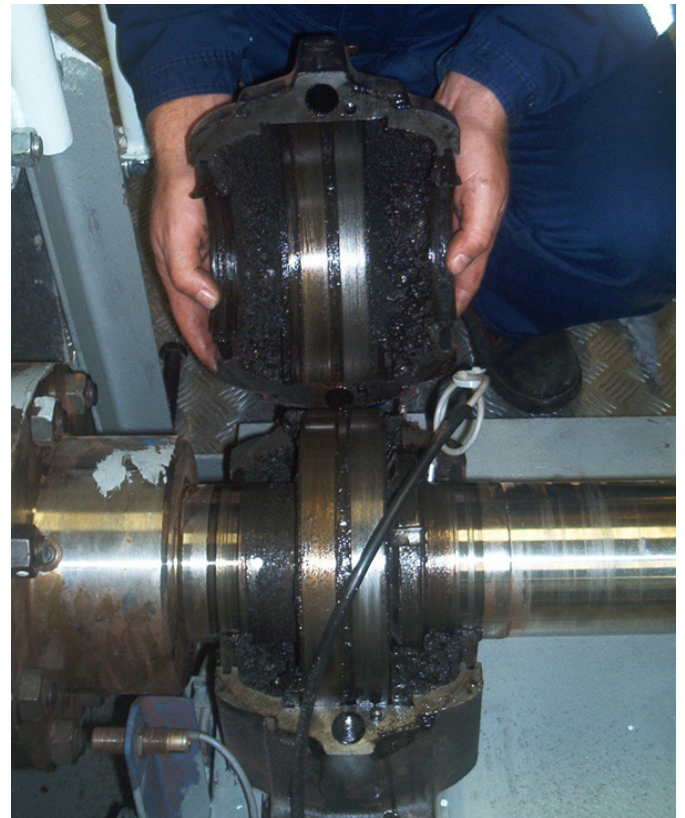


Figure 31: Coloured bearing metal due to insufficient lubricant.

Bearing temperatures measured on the housing surface in way of the outer bearing ring (on the bearing housing) shouldn't exceed some 60 degrees C for standard grease and 80 degrees C for high temperature grease.

grease type	standard grease	high temperature grease
NLGI class	2	1.5
base oil type	mineral oil	synthetic
thickener	lithium	clay
temperature range	-25-130° C	-55-180°C
dropping point	190° C	310° C
ref. viscosity base oil @ 40°C	100 cst	30 cst

Regularly checking these joint bearings for presence and condition of lubricant and condition of raceways is highly recommended to prevent vibration due to play of the joints or, even worse, fatal damage with the joint flying out of the shaft line. A regular check reduces risks and isn't very costly as needle bearings are very cheap spares.

Shaft line – Why do shaft lines vibrate?

Shaft lines vibrate because they are excited by dynamic forces, mainly from dynamic reaction forces of the cardan shafts under torque and deflection and, in rare cases, by an imbalance in the shaft system. These are the main sources for self-excited vibration of the shaft line. Vibration can also occur through the shaft line foundation. This could be called *indirect vibration* and is mainly excited by, for example, diesel motor foundation excitations or thruster excitations occurring via the thruster foundation.

The shaft line is a mass-elastic system and will react with forced vibration, which are amplified when a resonance of the shaft line is hit. As the whole shaft line runs in the open and foundations are normally welded constructions with rigid connections of plummer blocks to foundation, the whole system has a very low damping which leads to high amplification factors when a resonance is hit.

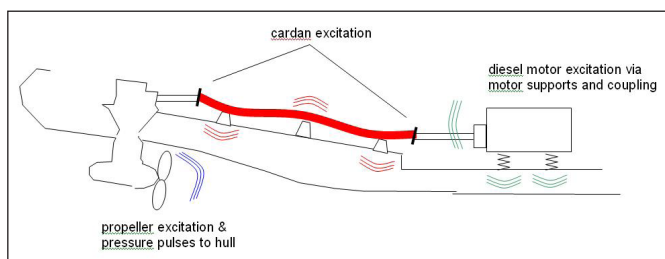


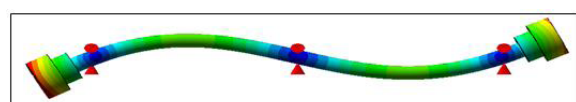
Figure 32: Shaft vibration and its excitation sources.

Problems of imbalance appear in particular with worn equipment or as a consequence of assembly failures. Recommendations for the balance quality of the shaft components can be found in ISO 1940 and VDI 2060. A balancing quality of 6.3 or better should be sought (easy to reach in practice). Most of the problems come from the cardan shaft excitation and vibration from the motor foundation.

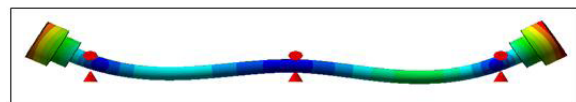
For images of Cardan shaft alignment and dynamic bearing forces under torque and deflection see Figures 33 & 34 at the end of the paper.

Cardan shafts:

- Produce dynamic reaction forces onto the shaft line bearings with higher excitation frequencies at higher shaft speeds (depending on W- or Z- arrangements, torque and deflection angle);
- Additional dynamic components appear due to alignment failures that are unavoidable in real installations;
- An alignment accuracy of a few tenths of a degree is required for cardan shafts;
- Deflection angles are normally chosen below 4 degrees (maximum 8 degrees) to limit dynamic excitation forces;
- Cardan shaft excitation goes with second order of the shaft revolutions (and its higher orders);
- Sufficient distance of the first natural frequency of the shaft line to this cardan shaft excitation frequency is needed;
- The so-called Campbell diagram below shows the aims of the design:



1. Eigenmode: 76.7 Hz



2. Eigenmode: 88.3 Hz

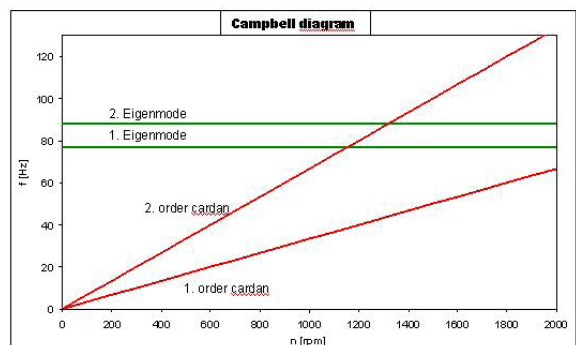


Figure 35: Shaft line resonance shown in a Campbell diagram.

Schottel designs the shaft lines with the first natural frequency at least at 20(15) per cent distance above the cardan shaft excitation frequency at nominal speed (under-critical design). In over-critical design the natural frequency is far below the nominal speed (ie idle speed to 50/60 per cent shaft speed is possible and useful but requires exact knowledge of the parameters).

The natural frequencies of the shaft line masses and stiffnesses of the shafts and bearings are known by the shaft line maker. Bearing foundation stiffnesses must be agreed. Schottel requires a radial design stiffness of 500 kN/mm; 300kN/mm are used in calculations; poor values of down to approximately 100kN/mm are regularly found in real installations and cause severe problems.

The foundation stiffness is a linked serial stiffness of the bearing stiffness (roller contact), the bearing housing stiffness and the welded foundation stiffness. The foundation stiffness in the azimuth range significantly influences the natural frequency of the shaft line.

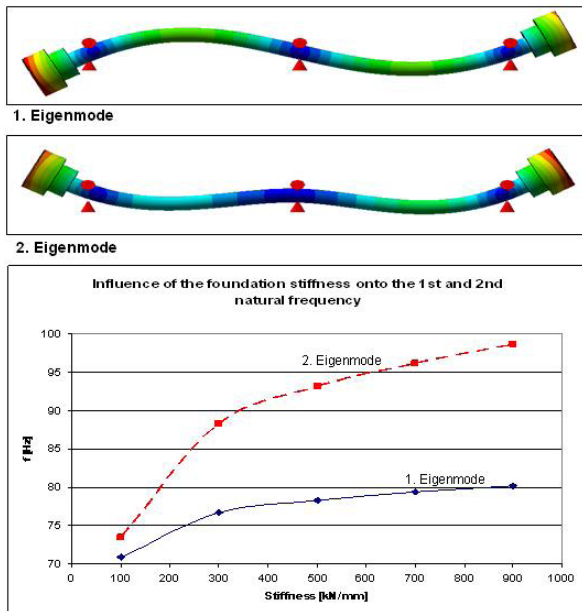
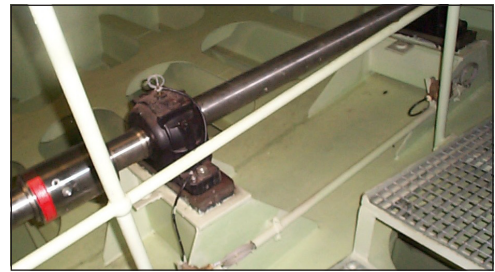


Figure 36: Influence of foundation stiffness on first and second natural frequency of an intermediate shaft.

If there is any doubt, Schottel can evaluate the quality of the shaft foundation if requested. Often the shaft line foundation is not defined at the time of order because steelwork for the shaft line foundation is the remit of the yard and so the designer just leaves some space for the foundation in its steel construction; this sometimes results in “uncontrolled” and poor construction.



Figures 37: Poor construction of shaft line foundations.



Figures 38: Poor construction of shaft line foundations.

Bulkhead bearings are a special case. In the case of traditional shaft line designs with multiple bearings, bulkhead bearings were normally prohibited because of the “jump” in radial stiffness and the resulting undefined load situation for the bulkhead bearing.

With the modern shaft designs, bulkhead bearings are used because they offer high radial stiffness due to the bulkhead itself and because the bearing flanged into the bulkhead substitutes the bulkhead sealing.

Problems occur when the bearing is axially fixed and the axial stiffness of the bulkhead is insufficient or when the bulkhead bearing is used together with more than one additional bearing in the same shaft line.

The worst case scenario is when the shaft line runs into resonance and an axial vibration of parts of the shaft line or the whole shaft line will appear. This normally results in short-term damage to the bulkhead bearing or other axially moveable components of the shaft line, such as the cardan shaft splined connection or deflection couplings.

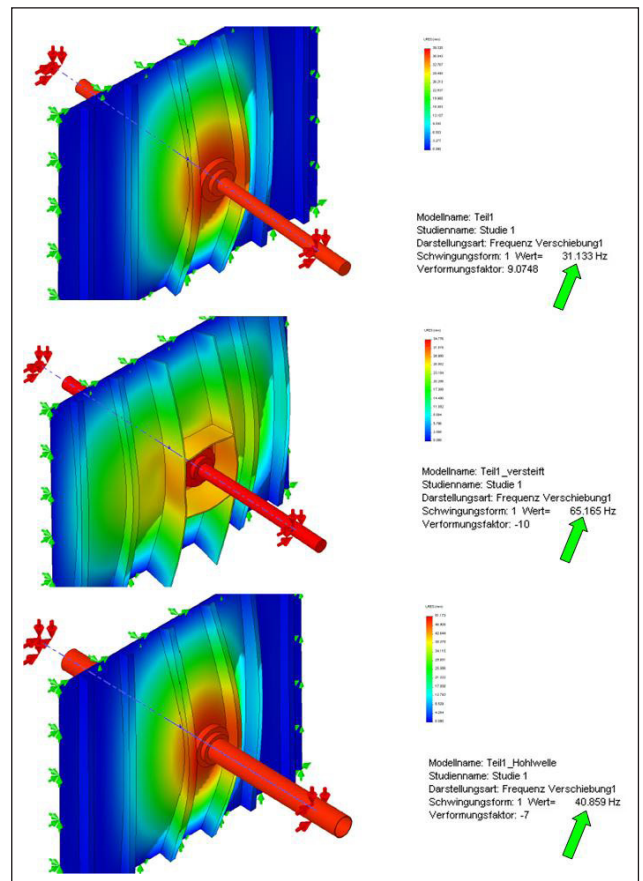


Figure 39: Bulkhead in resonance and possible solutions.

Figure 39 shows what can happen and how it can be solved. For a 1,000 rev/min application the main cardan shaft order of excitation is $f = (1600 \text{ rpm} / 60) \times 2 = 33,33 \text{ Hz}$.

In the first picture the mass-elastic-system of the shaft line (which is connected to the bulkhead via the bulkhead bearing) and the bulkhead (being a membrane spring) has an Eigenfrequency at approximately 31Hz which would result in a resonance with the cardan excitation slightly below the nominal speed. This is unacceptable and will clearly result in, at the very least, a short lifetime of the bearing, or even fatal damage to the shaft line.



Figure 40: Fatal damage due to resonance problems.

Such problems can be solved with a little effort both in the design phase (change of axial stiffnesses or masses of shaft components / shaft separation) and by simple observation when operating (additional stiffeners on the bulkhead).

In the middle picture of Figure 39, two simple flat irons running vertically left and right of the bulkhead bearing are used to shift the Eigenfrequency to twice its original value by stiffening. In the bottom picture, the problem is solved by making the shaft lighter using a hollow shaft of 220 × 8mm instead of the solid 140mm shaft, resulting in a safe increase of the Eigenfrequency by almost 30 per cent.

To prevent this problem the axial bulkhead stiffness should be given to the shaft line supplier.

Shaft line – unacceptable displacement

Displacement of the ship's structure under sea load of the hull, deflection of the hull under thruster forces, thermal expansion and movement of the motor on its elastic mounts have to be taken by the shaft line.

In modern shaft lines with a low number of bearings over the shaft length and deflection couplings between shaft elements, the radial displacement is usually not critical. More problems are found in axial direction both for static and dynamic displacement.

The following example gives some absolute values that can be found in a real application:

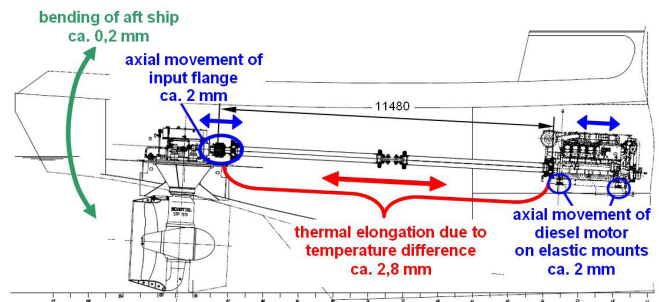


Figure 41: Axial displacements in a standard straight shaft line.

In the above installation, with 11 m distance between motor and thruster, the following displacements occur:

- Axial movement of the thruster input flange under thrust load in both directions (eg +/- 2mm);
- Thermal expansion of shaft line at differential temperature of the shaft of some 10 to 20K (eg + 3mm);
- Axial movement of the diesel motor on its elastic supports (eg +/- 2mm) (value to be taken from 6 degrees-of-freedom calculation of motor manufacturer);
- Bending of aft ship structure under thrust load with effect on thruster-motor distance (can be neglected here).

As can be seen in the above example, the prerequisite axial compensation capacity of the shaft line is about 7mm for a longer shaft line. Values can be somewhat higher for large boats or softer motor mounts.

To manage these deflections, compensation elements are used in the shaft line, most commonly steel link couplings with steel or rubber joints or toothed couplings. These couplings are normally rigid enough not to influence the TVC and require little or no maintenance.

Most flywheel-mounted elastic couplings (based on the Torsional-Vibration-Calculation result and usually responsibility of the diesel motor manufacturer) allow significant axial displacements under a given axial stiffness. This displacement capacity can be used, but the axial reaction force onto the shaft lines' fixed bearings has to be observed (see Figure 42).

With bell-mounted elastic couplings with an integrated output shaft, the problem is often reversed which means that the bell bearings are the weakest part in the axial chain and the occurring problems the motor manufacturer's responsibility. Reaction forces resulting from the stiffnesses of the shaft line compensation couplings will apply axial and radial loads and also bending moments onto the bell bearings.

Additionally these bearings suffer from high vibration loads and increased ambient temperature from the diesel motor. Nevertheless, if properly designed and all these points are kept in mind, this solution is favourable from the point of view of the installation and the low number of components needed.

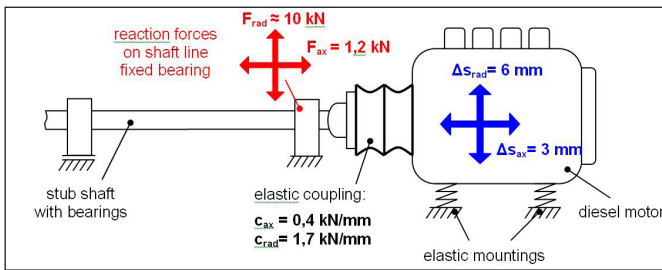


Figure 42: Axial displacement capacity of typical double-row elastic coupling / reaction force **on** shaft line.

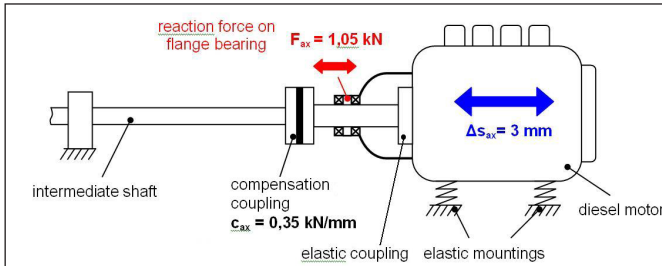


Figure 43: Bell-mounted elastic coupling with integrated stub shaft / reaction forces **from** shaft line.

As explained above, the radial displacements are not critical in today's designs using a minimum number of bearings per shaft length.

The following example gives an idea of what the limits are in radial deflection:

With a required safety margin of 20 per cent, the minimum first Eigenfrequency of a 1,600 rev/min shaft line is $f_{min} = 1,2 \times 2 \times 1600 [\text{min}^{-1}] / 60 [\text{s}] = 0,04 \times 1600 [\text{s}^{-1}] = 64 [\text{Hz}]$ for a shaft line with cardan shaft excitation. In practice, of course, the shaft line is, if possible, designed to a higher Eigenfrequency as an additional safety margin.

Case A: a traditional arrangement with solid shaft line

power $P = 2,200\text{kW}$
 shaft speed $n = 1,600 \text{ rev/min}$
 shaft diameter of the solid 42CrMo4V shaft $d = 140\text{mm}$
 bearing distance for less than critical design $l = 1,720\text{mm}$

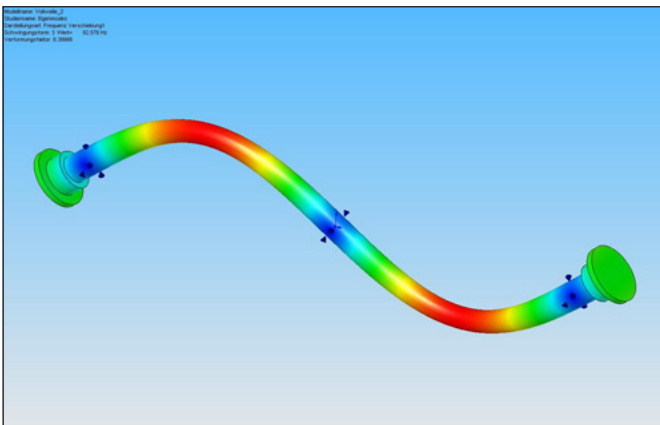


Figure 44: First mode of vibration of the solid shaft at approximately 90Hz.

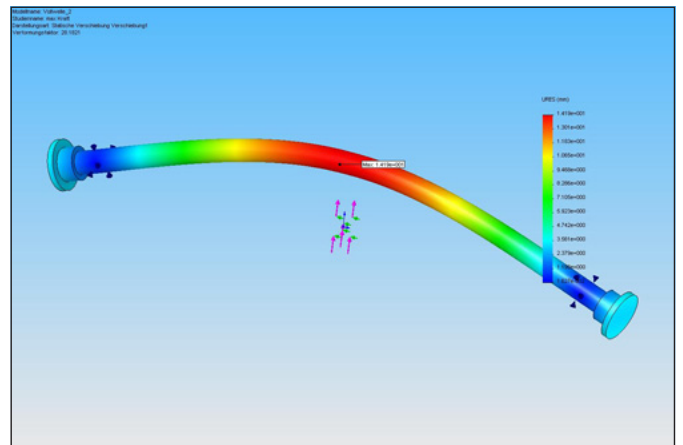


Figure 45: Permissible radial displacement of the centre bearing at bending fatigue limit of the shaft is 14mm:

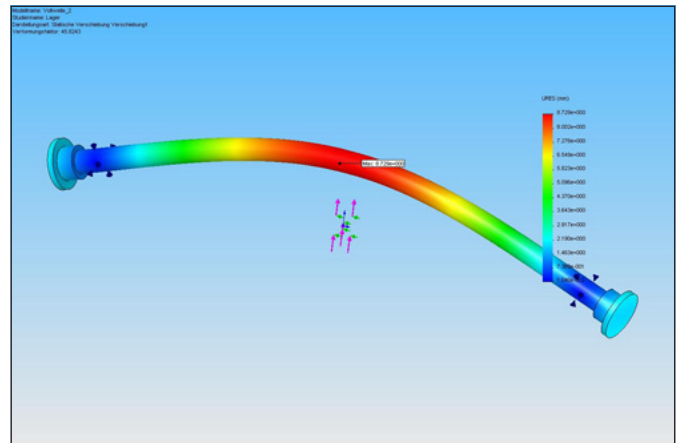


Figure 46: Permissible radial displacement of the centre bearing at bearing lifetime $L_{hna} = 40.000$ hours is 9mm:

With this design, the radial displacement capacity of the shaft line is approximately 14mm at the bending fatigue limit of the shaft and some 9mm at the required bearing lifetime of $L_{hna} = 40.000$ hours. This corresponds to a length of the shaft line of approximately 4,000mm.

Case B: a modern arrangement with hollow shaft line:

power $P = 2,200\text{kW}$
 shaft speed $n = 1,600 \text{ rev/minute}$
 shaft diameter of the solid 42CrMo4V shaft $d = 140\text{mm}$ (at bearing positions)
 bearing distance for under critical design $l = 2,730\text{mm}$

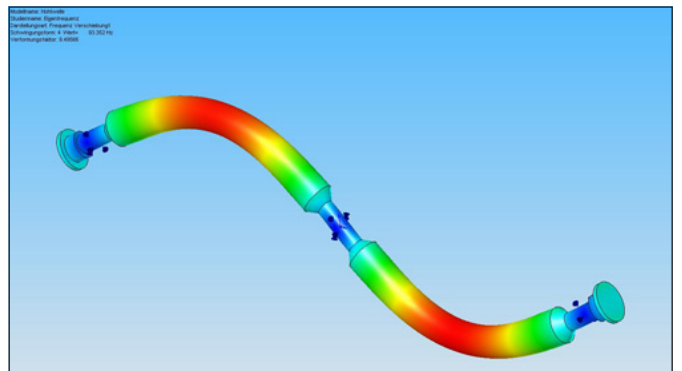


Figure 47: First mode of vibration of the hollow shaft at approximately 90Hz.

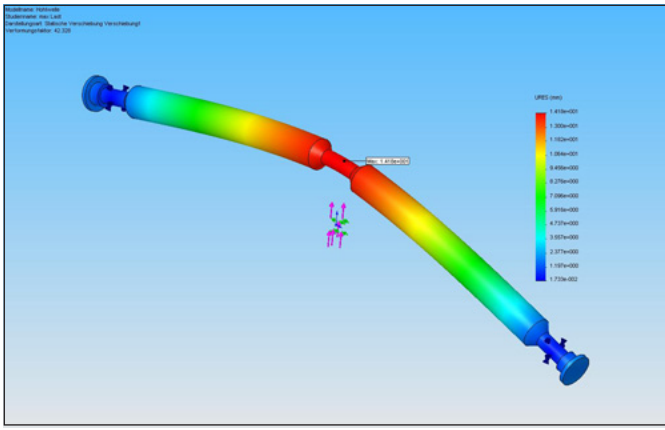


Figure 48: Permissible radial displacement of the centre bearing at bending fatigue limit of the shaft:

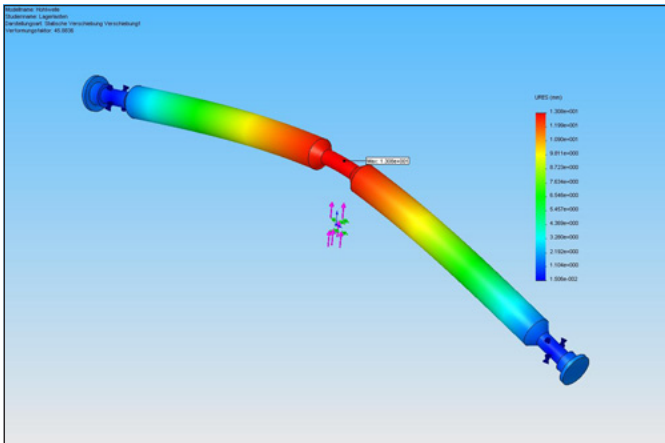


Figure 49: Permissible radial displacement of the centre bearing at bearing lifetime limit for example $L_{hna}=40,000$ hours.

With this design the radial displacement capacity of the shaft line is 14mm at the bending fatigue limit of the shaft and 13mm at the required bearing lifetime of $L_{hna} = 40,000$ hours. This corresponds to a length of the shaft line of 6,000mm.

These values may explain why the radial displacement is not a real problem in shaft line design. In fact, a certain radial displacement is always present and helps to give the bearings additional pre-loading, which reduces the risk of slipping due to under-load. Problems can occur when single bearing positions suffer under load due to radial movements of the foundations. On the other hand, the values above demonstrate that a defined offset of the bearing positions during installation of the shaft line offers sufficient tolerance to load the bearings also by the shaft cross force under targeted bending of the shaft.

This method isn't common practice today, but can be used without worry.

Shaft line – statutory minimum load bearing

Any roller bearing used in shaft lines requires a minimum load to keep the rollers rolling instead of

slipping or shifting (with inevitable damage). If this minimum load is extensively undershot, (see Figure 50) slipping of the rollers will shortly follow.

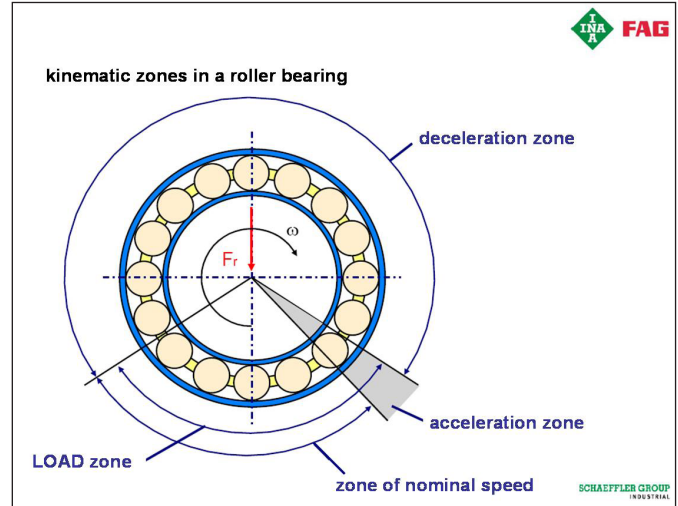


Figure 50: Roller bearing slippage.

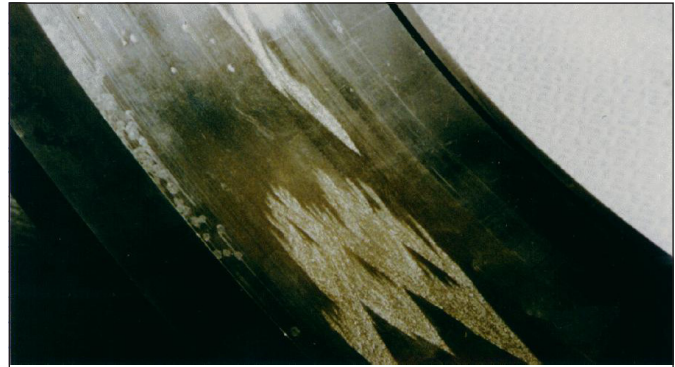


Figure 51: Slipping damage on raceway.

The mandatory values in the bearing manufacturers' catalogues are often not properly reached in real terms, yet most shaft lines run satisfactorily and do not suffer slipping damage. This indicates that deviations can be accepted in practice. Nevertheless, if the required values fall considerably short, it becomes critical and damages can be expected.

For a given shaft size different bearing types with different section sizes and load capacities are used. This results in different minimum loads required for those bearings made for the same shaft size.

The formula below indicates the minimum required load for standard bearing types used in shaft lines.

For spherical roller bearings the minimum radial load needed for grease lubrication is approximately:

$$F_{rad,min} = 0,01 \times C_0 \text{ [kN]}$$

with C_0 being the static load capacity of the bearing taken from the manufacturer's catalogue [kN].

For cylinder roller bearings the minimum radial load needed for grease lubrication is approximately:

$$F_{rad,min} = k_r \times (6 + 4n/n_r) \times (d_m/100)^2 \text{ [kN]}$$

with: k_r being the minimum load factor

n being the actual speed [rev/min]

n_r being the reference speed [rev/min]

d_m being the average of outer and inner bearing diameter [mm].

For a given shaft diameter, bear in mind that the minimum load of the strongest spherical roller bearing can be some 10 times the value of the minimum load of the smallest cylinder roller bearing section.

Forced loading of the bearings can be made by radial offsetting of, for example, the centre bearing of a triple supported intermediate shaft or by radial offsetting of the diesel motor using the radial stiffness of the main elastic coupling to load, eg the stub shaft bearings. In case the calculation results show a bearing position that falls short radially, then this bearing should be used as the fixed bearing of the respective shaft. The rollers all over the bearing circumference will be therefore more or less equally preloaded and slipping risk is significantly reduced. This axial loading can also be done by using the downhill-slope force of an inclined installation.

Shaft line – maximum axial bearing loads

Significant axial movements of the shaft line occur in a tug's shaft line (see **Shaft line – unacceptable displacement** above). These axial movements multiply axial constraining forces by the axial stiffnesses of the installed components. The resulting forces have to be taken by thruster input shaft, motor crank shaft and also by the shaft lines' fixed bearings. Axial forces can be strong and have to be considered when choosing the section size of the fixed bearing.

Another source of, often underestimated, axial forces is the friction force transmitted via the splined connection of the cardan shafts.

The following example of a 3,000kW / 750 rev/min installation demonstrates the problem:

- Cardan shaft size: 390.80;
- Shaft torque: $T = 38,200\text{Nm}$;
- Radius of the cardan shaft splined connection: $r = 92\text{mm}$.

The resulting friction force that can be transmitted axially via the splined connection of the cardan is calculated as:

$$F_{ax} = \mu \times T / r = 62282 \text{ N}$$

with $\mu=0,15$ being the worst case friction coefficient of a poorly lubricated splined connection.

Of course, this load will not appear constantly because vibration and axial movements will make the splined connection slip. But especially when running-in, with steady thermal growth of the shaft line, the high axial force value can act on the fixed bearing over a period of time, critical for the fixed bearing in respect of overheating.

With the azimuth power and speed data, normally a standard cylinder roller bearing could be used as a fixed bearing of the intermediate shaft considering radial loading and required longevity. The axial load capacity of this bearing can be calculated to approximately 20kN, which is much less than required (see value above). The required bearing size is therefore, eg a spherical roller bearing of the section size 23048 with much larger dimensions (see size comparison in *Figure 52*).

Conical sleeves are normally used to clamp the bearing inner ring to the shaft line. It should therefore also be checked to see if the transmittable axial force of this clamped connection is strong enough to take the high axial load in cases such as in that below.

For an image of Necessary over-dimensioning of the bearing because of high axial load see Figure 52 at the end of the paper.

Note that such cases normally only occur when higher axial movements are to be compensated in a shaft line in combination with low speed = high torque applications.

Shaft line – unacceptable bearing vibration

Different reasons for shaft vibration have been addressed beforehand. The vibration level should be measured during the commissioning of the tug. Permissible values are given in the diagram below.

Values should be measured as RMS (root mean square) values over a frequency range of 4-1,000Hz.

Effective value of the vibration velocity (RMS) [mm/s]	$\leq 4,5$	$\leq 11,2$	$> 11,2$
Zone	A	B	C

Figure 53: Recommended evaluation chart for shaft-bearing vibration.

As long as the values shown above are adhered to, short-term failure of a shaft bearing is not likely. Nevertheless, damage can also occur when vibration levels are in the green range (eg lubrication or short load problems due to misalignment). Advice for the execution of the vibration measurements on such shaft lines will be found in ISO 20283-4 which is being developed.

Generally, the fixed bearings should be measured in all three main directions whereas the loose bearings are measured in the two radial directions only. Nevertheless, with very high and soft bearing foundations or with loose bulkhead bearing, excessive axial vibration of the foundations can cause problems with the roller bearings especially with spherical bearings. The outer bearing ring cannot follow the fast vibration movement and will wear out in the housing and the outer ring will axially hammer into the rollers within the axial bearing play. Both effects can cause bearing damage.

Problematical stub shaft vibrations:

The picture below shows the problem of diesel motor vibrations induced by the stub shaft bearings especially to the bearing close to the motor. It is generally recommended that the stub shaft sits on a foundation which is well connected to the motor foundation to prevent excessive radial movements of the motor to the stub shaft. The disadvantage of good foundation connections is the regularly higher vibrations of the stub shaft foundation in excess of the values as given above (see *Figure 53*).

The permissible motor foundation vibrations range from 25mm/s and values in that region and in excess of it are often measured on the stub shaft (see *Figure 54*, an example of vibration measurements on a standard shaft line).

For an image of An example of vibration with excessive stub shaft vibrations due to diesel see *Figure 54 at the end of the paper*.

As the roller bearings accept higher vibration levels at a higher pre-load, a useful trick is to use the radial displacement capacity of the torsional-elastic coupling on the flywheel to radially pre-load the stub shaft bearings within its allowable limits, having a regard to bearing lifetime.

For an image of Loading the stub shaft by defined radial offset of the motor to the stub shaft see *Figure 55 at the end of the paper*.

In this example of a 1,650kW / 1,000 rev/min application, the shaft diameter is 140mm and the radial load of the bearings is approximately 1,800N by the weight of the shaft. The dynamic force component of the cardan shaft running at 4 degrees deflection angle is already 1,500N acting on the horizontal plane.

The radial stiffness of the double-row elastic coupling is 2.4kN/mm. The permissible continuous radial offset of the coupling is 7mm. The limit load of the spherical roller bearing type 23028 (smallest bearing section for that shaft diameter) at $L_{h10} = 20,000$ hours nominal lifetime is 55kN, which gives enough margin for pre-loading. The minimum load (catalogue value) for this bearing to be safe against slipping is 6.8kN.

A radial displacement, therefore, of about 2-3mm can be chosen and will result in about 8kN pre-load of the bearing which will work well in practice. The elastic coupling will still have sufficient radial displacement margin to take tolerances from the alignment and operation deflection of motor to shaft.

TYPICAL EXCITATIONS OF THE NOZZLED THRUSTER

Main static and dynamic force components acting on the azimuthing thruster

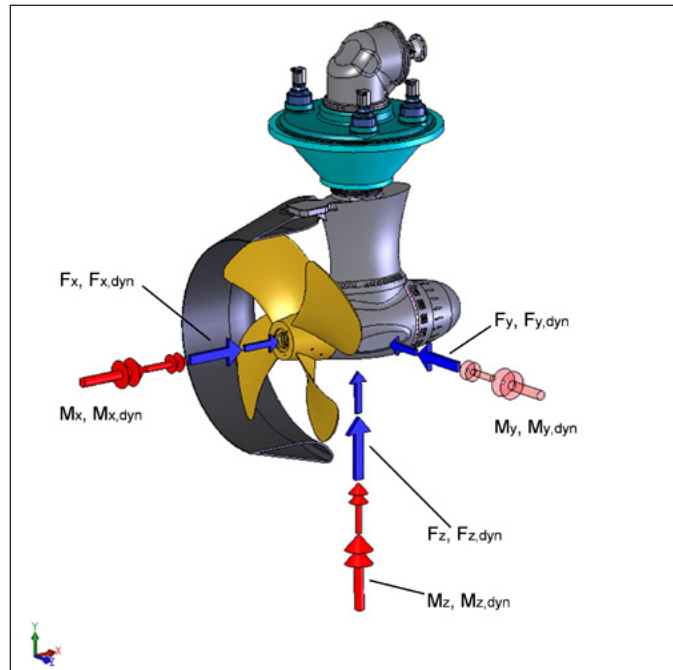


Figure 56: Azimuth thruster with main static and dynamic force components.

- static thrust force + dynamic thrust component F_x & $F_{x,dyn}$
- side force + dynamic component F_y & $F_{y,dyn}$
- weight + vertical dynamic component F_z & $F_{z,dyn}$
- steering torque (highly damped) M_z
- propeller torque M_x & $M_{x,dyn}$
- propeller bending moment due to wake field M_y & $M_{y,dyn}$

For the calculation of the strength of the ship's structure, the maximum values of the static components of F_x , F_y , F_z and M_x , M_z are used in combination. M_y is ignored because of low relative magnitude.

Schottel supplies these values with the document *Forces and Moments* on request by designer or yard. The resulting values of bending moment, steering torque and horizontal force are usually transformed to the coordinates of the centre of the thruster well top flange, so the values can be directly used for calculations of the strength of the ship's structure.

For the modal analysis of the thruster in the structure, either a reduced pendulum model or a complete 3D-thruster model is used. As long as a fair margin to resonances is achieved, the modal analysis is a sufficient dynamic calculation.

For the forced damped vibration calculation of the submerged thruster in the ship structure, the dynamic components of F_x , F_y , F_z and M_x are used. M_z is ignored in addition to M_y because of the very high damping of the steering gears which results in negligible magnitude as well. Forced damped calculation is only used to check the effect of unavoidable resonances.

For an image of Forces and moments for a nozzleed thruster see Figure 57 at the end of the paper.

Exemplary static and dynamic forces of the four-bladed nozzleed thruster

For design purposes some standardised values for the azimuth force and movement components can be used for the four-bladed nozzleed thruster. It has to be considered that these values are only valid for thrusters with nozzles with typical design data for tug application.

The table below gives some standard values that can be used to make calculations on such installations. The values are related to the centre of the lower gear box (section point of propeller and azimuth axis) and can act in any 360 degree rotated direction.

static thrust force	F_x	$P \times 170$ [N/kW]
dynamic thrust component (1 st blade harmonic)	$F_{x,dyn1}$	10% of F_x
dynamic thrust component (2 nd blade harmonic)	$F_{x,dyn2}$	5% of F_x
static side force	F_y	$2 \times M_z / D_{prop}$
dynamic side force (1 st blade harmonic)	$F_{y,dyn1}$	10% of F_y
dynamic side force (2 nd blade harmonic)	$F_{y,dyn2}$	5% of F_y
weight force	F_z	$m_{thruster} \times 9,81$ [m/s ²]
vertical dynamic force (1 st blade harmonic)	$F_{z,dyn1}$	10% of F_y ¹⁾
vertical dynamic force (2 nd blade harmonic)	$F_{z,dyn2}$	5% of F_y ¹⁾
propeller torque	M_x	$16 \times P$ [kW] $\times D_{prop}$ [m]
dynamic propeller torque	$M_{x,dyn}$	$\pm 25\%$ of M_x
max. azimuth torque	M_z	$0,02 \times n_{prop}^2 \times D_{prop}^5$

1) relation to F_y is used consciously

It has to be remembered that most values are considerably dependent on motor torque, steering angle and ship speed and, in practice, can deviate from the above given values. As a coincidental example the diagram below shows some measured values of the relative dynamic side force component $F_{y,dyn}$ at a ship speed of some 9kn. This may illustrate that the am dynamic forces can be occasionally exceeded under dynamic steering operations at higher ship speeds.

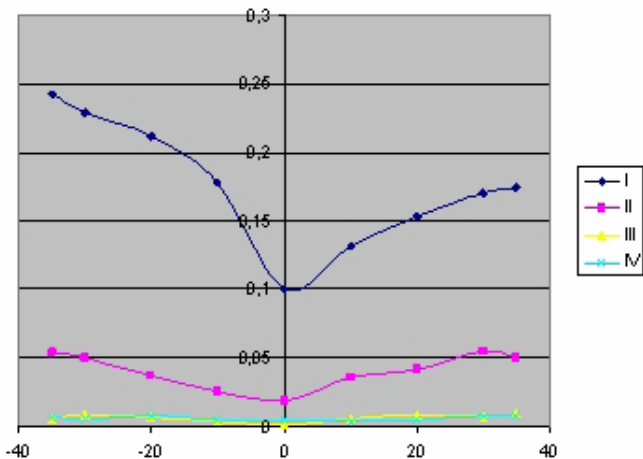


Figure 58: Influences of steering angle and ship speed on exemplary load component $F_{y,dyn}$.

All the values above are thumb values which are good enough for basic designs. In case of problems with resonance a more detailed view of the individual problem needs to be made.

INTEGRATION OF THE THRUSTER IN SHIP CONSTRUCTION

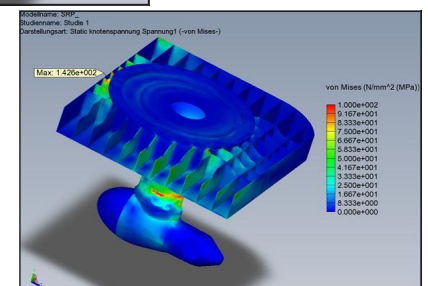
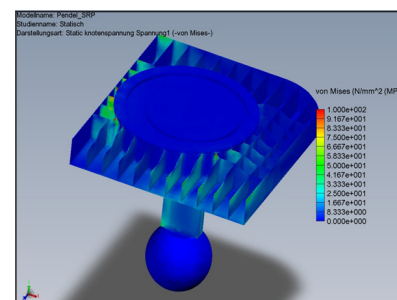
Calculations using detailed and reduced models

The modal behaviour of the thruster in ship structure has already been demonstrated (see Figures 17, 27 and 28). The thruster can be reduced to a simplified pendulum model given to the customer which can than be used for strength calculation of the structure and for modal analysis of the thruster in the hull.

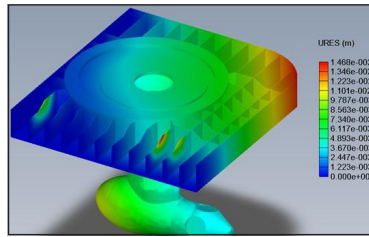
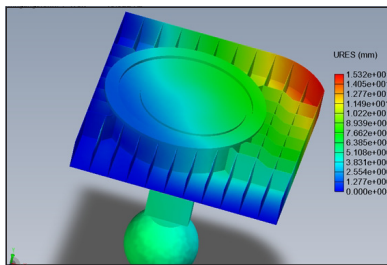
The dummy pendulum is built-up as a 3D mass (including the mass and inertia properties of the underwater parts of the thruster) placed on the end of a beam element with different bending stiffnesses in the two pendulum directions (so it has the same properties as the thruster steering column). This 3D mass, plus beam, is fixed to a circular plate representing the mass and membrane stiffness of the non-azimuthing part of the thruster. The diameter of the circular plate corresponds to the inner diameter of the thruster well in the ship. Thus the dummy pendulum fits into the structural model of the ship.

The following example of a comparative calculation of a scaled model thruster unit and its dummy pendulum model in a ship's structure may tempt yards and designers to make more regular use of this kind of calculation.

The results of the reduced and simplified models normally agree within a few percent with the results of the more extensive high-resolution models (for the rather low frequencies that are examined here).



Figures 59 and 60: Calculation of stress under maximum loads for detailed and dummy models



Figures 61 and 62: Calculation of first Eigenmode under for detailed and dummy model.

In the above example, with a stiff integration of the thruster, the first pendulum frequency (first Eigenmode) of the thruster in the structure is about 40Hz. The first and second propeller blade excitation frequency of the propeller (rotating at 230 rev/min) is 15.3 resp. 30.6Hz, which means the excitation frequency is approximately 25 per cent lower than the first Eigenmode of the thruster. Everything is therefore satisfactory.

If the safety margin is small and the propeller blade excitation is close to an Eigenmode in the speed range of 80-120 per cent of the nominal speed, measures should be taken to prevent excessive vibration and the consequential reduction in component lifetime.

The example below of a forced damped vibration calculation of thruster pair in the structure shows the system response close to the resonance. In the case of a large thruster, the pendulum frequency of 12.5Hz is excited by the first propeller blade frequency of 12.1Hz at nominal speed. The dynamic thrust component is applied with approximately 8 per cent of the nominal thrust. The vertical vibration component that can be measured on the thruster foundation is calculated as $s_{\text{vert}} = 0.2\text{mm}$, which corresponds to a vibration velocity of about 15mm/s. Of course this is not critical from the strength point of view, but the resulting vibration load will hamper thruster component lifetime and comfort on board.

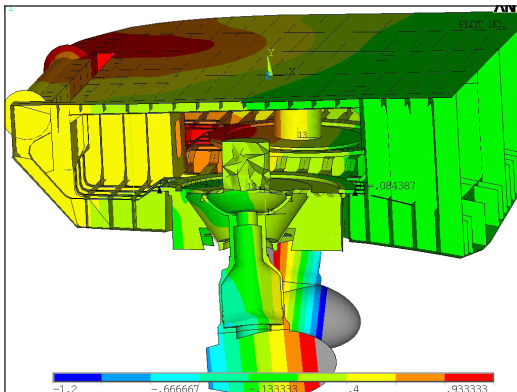


Figure 63: Thruster longitudinal pendulum mode excited by first propeller blade frequency close to the nominal speed.

Evaluation of the results

The above comparison shows the minor deviations of the scaled-down models' results against the detailed calculation. As for the shaft line vibration, a certain "clean" speed range, especially around nominal speed, is the design goal.

Classifying authorities do not have strict rules as to whether or not a structural resonance is accepted or not because this is clearly dependent on what excitation magnitude hits what mode with what damping, and what is finally the vibration amplitude of these constraints.

Schottel generally advises a speed range of 80-120 per cent of nominal speed, free of resonances of the 3(6) main modes of the thruster(s) with the first and second blade harmonic. When calculated with detailed models the clean range can be reduced. In any case the range between 90-110 per cent should be kept clean.

Resonances in the speed range of 50-80 per cent of the nominal speed can often be accepted because the relation of excitation and damping is good enough to deliver non-critical vibration amplitudes only.

From the example shown in Figure 63, the result of the calculation and alternative solutions are given in the Campbell diagram below. It can be seen that there will be a problem with the first blade order hitting the first pendulum Eigenmode of the thruster in the longitudinal direction (see point A). This will require either a more rigid connection of the thruster to the hull ("under-critical" design) with a potential shifting of the natural frequency to point B at approximately 13.7Hz (see above: Principal installation methods of the thruster – influence of connection stiffness) or alternatively the use of a five-bladed propeller with consequently a 25 per cent higher excitation frequency ("over-critical" design) which will leave the resonance point at some 75 per cent of the nominal speed which can normally be accepted (see point C). Point C is clearly outside the required resonance-free speed range of 80-120 per cent nominal speed. Point B is inside this range, but already higher than 110 per cent nominal speed, which could be accepted considering the accuracy of the model.

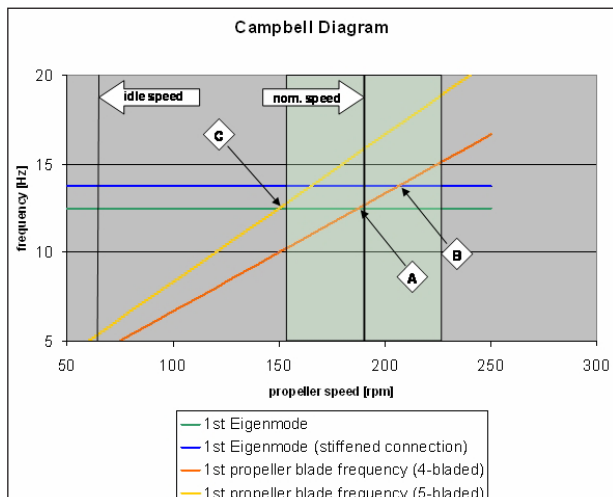


Figure 64: Illustration of the resonance problem and the two alternative solutions in a Campbell diagram.

The examples above may illustrate that the examination of the dynamic behaviour of the thruster in the structure can have different calculation depths starting with a “first shot” done with a simplified pendulum model in a coarsely meshed hull model. If resonance is found within a speed range of 80-120 per cent of the nominal speed, constructive countermeasures should be taken to solve the problem. If resonance is unavoidable, more detailed models to examine the modal behaviour, or even a forced damped calculation, can help to make a decision.

SUMMARY

With its long experience in the development, design and construction of azimuth drives for tug applications Schottel today has methods and experts available to support any task of integration of the drive system into the ship.

The designated project engineer in the sales department supports the project during the important order-entry phase when vital technical questions are clarified, usually within a short period of time, ie a few weeks. Within this period, orders and aspects of design are handled and corresponding calculations generally made by the technical branch of the production department with assistance from the development departments of hydrodynamic, electrical, hydraulic and mechanical engineering.

From a technical point of view, the order-entry phase is completed with the release of the final and official installation drawing showing the drive system integrated into the hull and coupled to the motor. Obviously, relevant information from the customer needs to be included beforehand.

To attach the thruster to the hull, Schottel offers three options, each with its own advantages, to be chosen by the customer. Different premises lead to the decision for either a welded connection, a bolted connection with O-ring sealing or a “rubber flange” installation with water beam cut rubber plates. Any of these alternatives has its own specific advantages and disadvantages from the point of view of the customer, especially considering the yard that has to install the thruster properly.

Furthermore, all three options can be installed from “above” or “below” which influences the degrees of freedom for the construction above the thrusters and the maintenance possibilities. In addition to the main connection between thruster main flange and ship flange, the radial webs of the thruster or even the bottom plate can be connected to the hull structure. This, to a lesser extent, adds strength but also gains stiffness and influences the modal behaviour of the thruster in the hull structure.

For the various installation methods design criteria and installation tolerances are given and tips to prevent problems included.

For the shaft line, the differences are shown between the traditional arrangement, with inclined solid steel shafts and cardan shafts, and the modern straight designs, with hollow shafts and compensation couplings.

The design goal of using the lowest possible number of bearings and other components requiring maintenance is highlighted.

The most common problems of installation and operation of shaft lines, such as lubrication, vibration, insufficient foundation stiffness (including the special case of bulkhead bearings) and over- and under-load – mainly due to radial and axial displacements – are explained and examples are given.

Schottel's efforts to reach a predominantly standardised modular shafting system, which nevertheless comprises solutions for all kinds of installation, is demonstrated. The standard components used, with their specific pros and cons and properties, are addressed. The Schottel modular shafting system allows the order specific design of the shaft line while considering questions of strength, lifetime and vibration safety of the drive train.

An overview of the standard scope of the calculations and information exchanged on both sides (supplier and customer) gives a clear idea of what is expected from whom and what assumptions are made.

With regard to thruster dynamic in the ship structure, the various force and moment components acting on the thruster are explained and thumb values given that can be used for dimensioning and for the basic design. Examples are given for the modal behaviour and for forced vibration of the submerged thruster when excited by realistic dynamic load components. Simplified models with reduced resolution give accurate examples. Schottel gives force and moment, plus reduced thruster models, to be used by the designer for the strength and vibration calculations of the thruster-hull system.

REFERENCES

1. FAG and SKF bearing manufacturers' technical catalogues.
2. FAG support center – information on damages.
3. GEWES technical information on cardan shafts.
4. SVA Schiffbau-Versuchsanstalt Potsdam report No. 3292.
5. Schottel internal technical papers.

Figure 7: TVC mass elastic system of a tug drive.

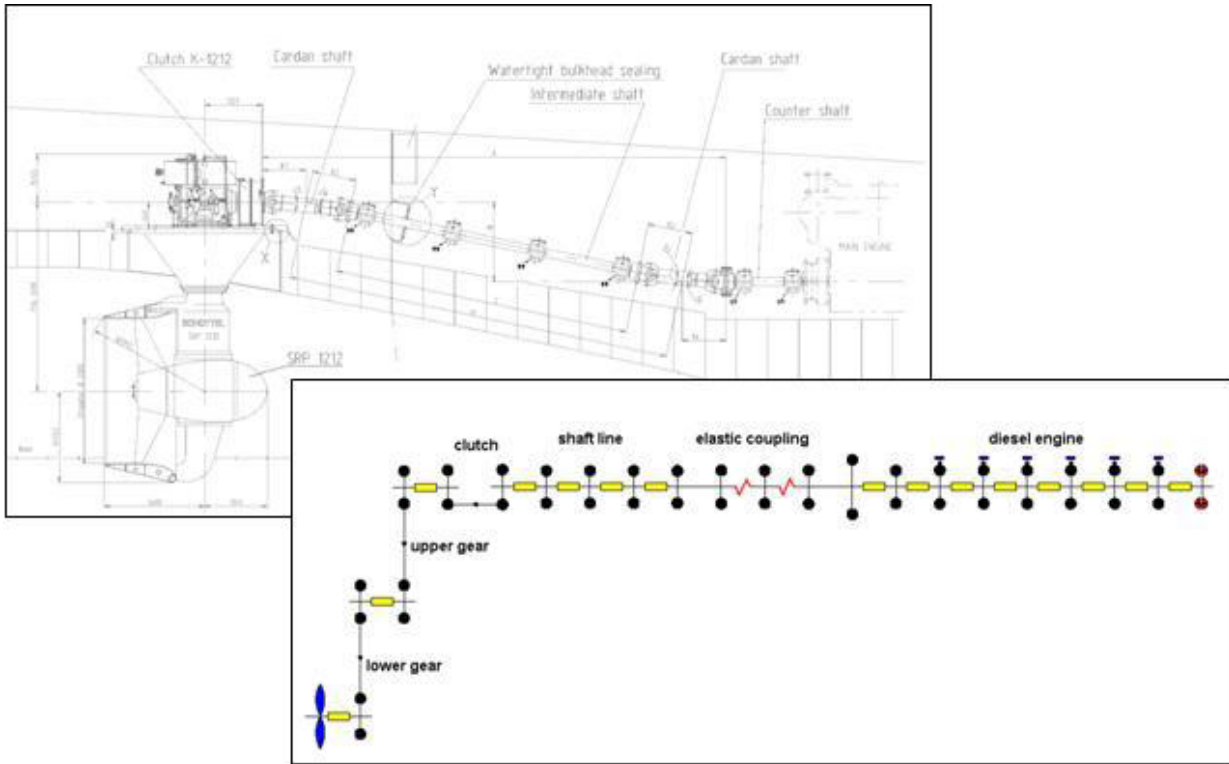


Figure 8: LVC vibration mode of a typical fourfold-supported intermediate shaft.

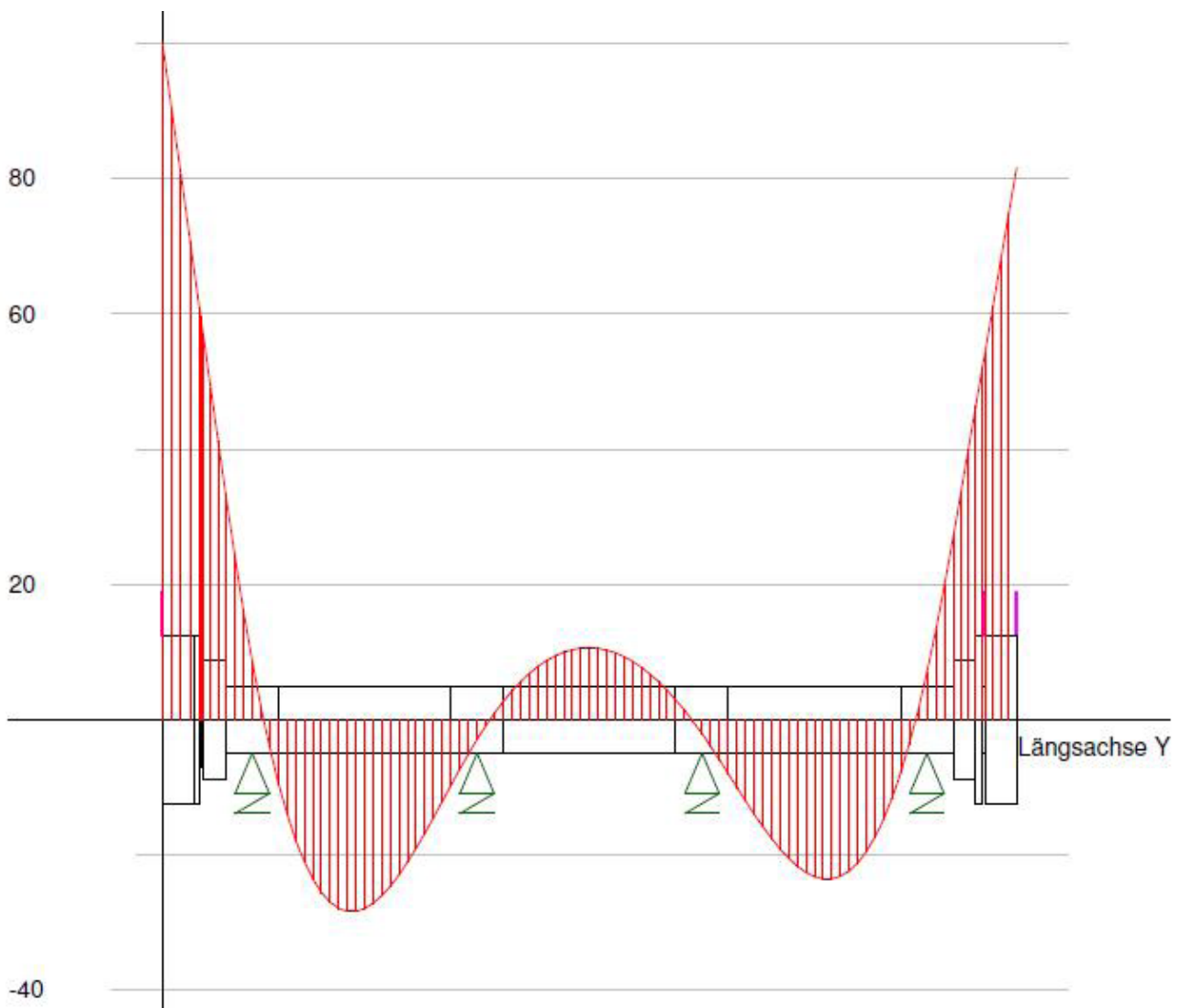


Figure 9: Organisation chart in chronological order.

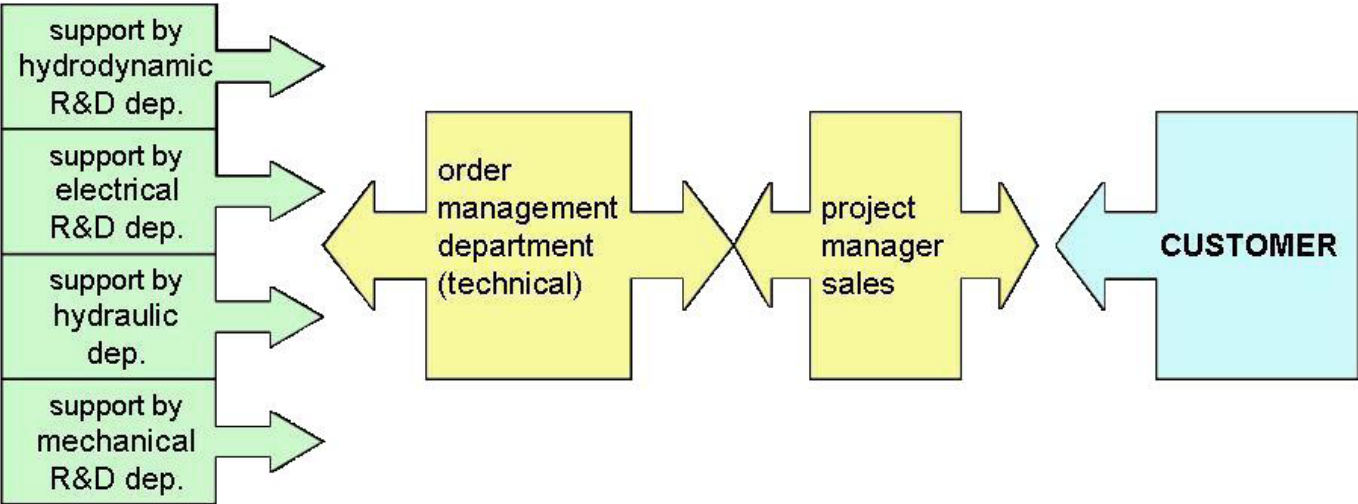


Figure 10: Types of bearing used in Schottel shaft lines.

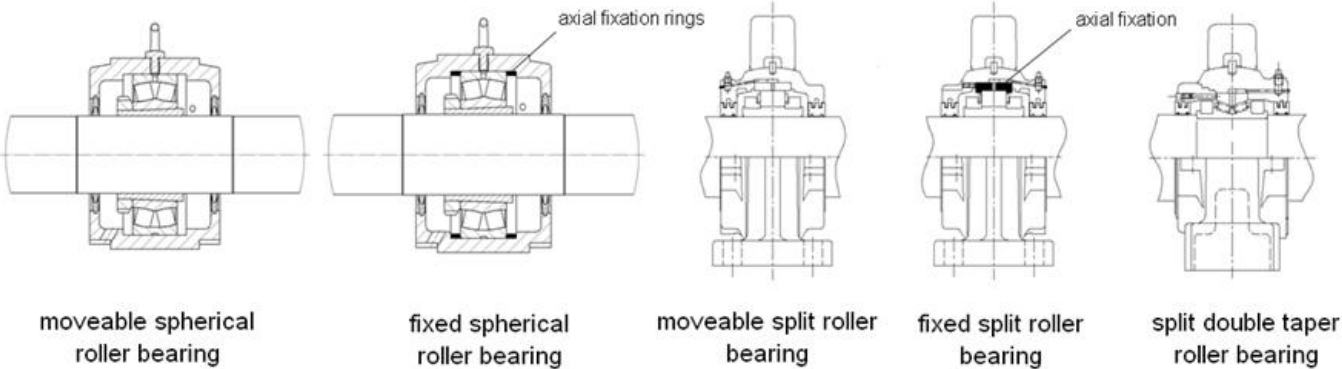


Figure 11: Standard shaft types used in the modular shaft line system.

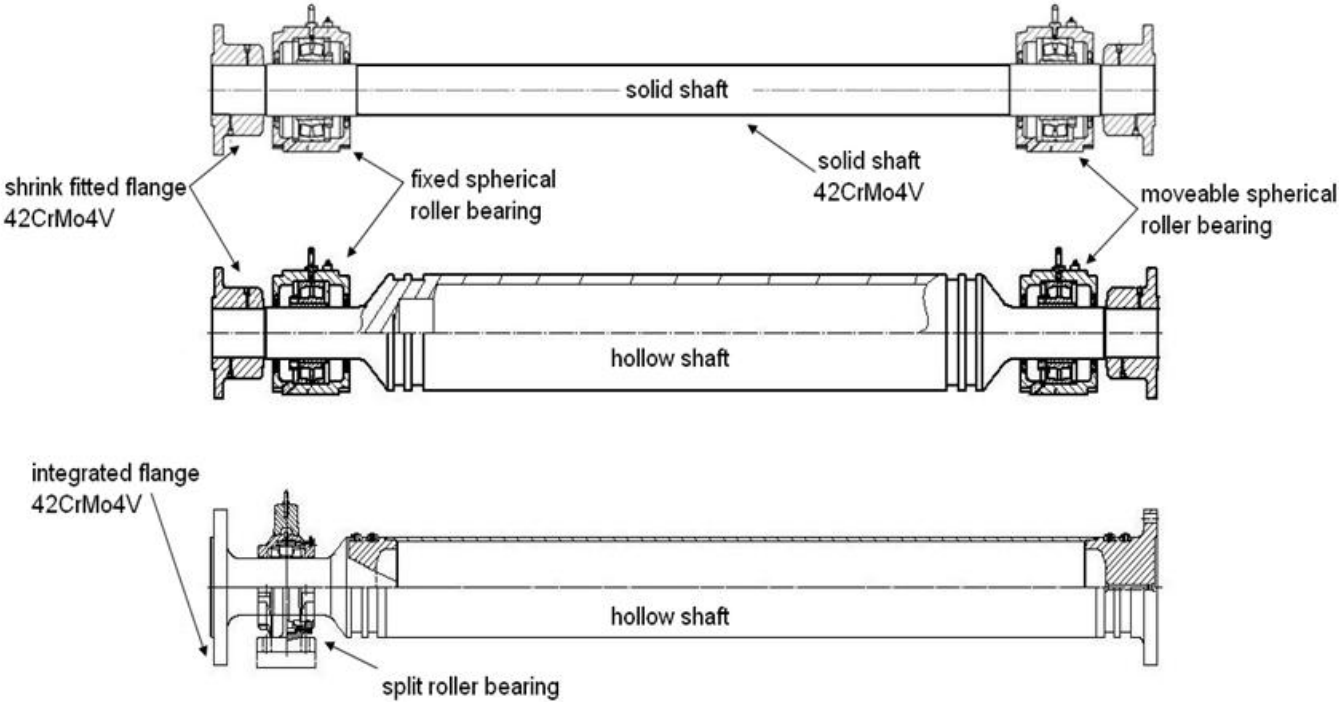


Figure 12: Couplings used in the modular shaft line system.

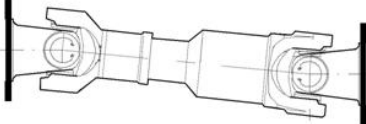
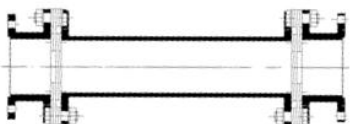
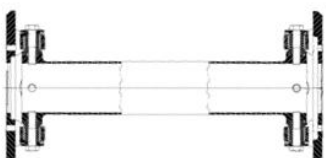
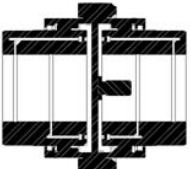
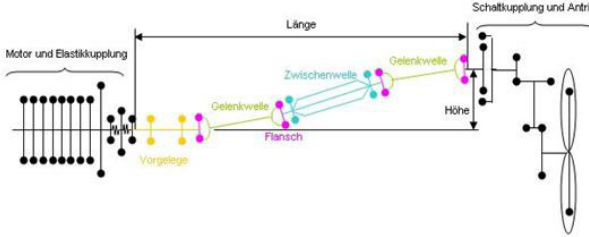
<p>cardan shaft</p> 	<p>coupling with steel joints</p> 
<p>displacement capacity: axial: $\approx 1 \dots 100$ mm radial: $\approx 100 \dots 500$ mm * angular: $\approx 1 \dots 8^\circ$</p> <p>alignment tolerances: axial: $1 \dots 100$ mm radial: depends on angle * angular: $< 0,5^\circ$</p> <p>* radial displacement and radial alignment tolerances depends on length</p>	<p>displacement capacity: axial: $\pm 2,5$ mm radial: $0 \dots < 50$ mm * angular: $< 0,5^\circ$</p> <p>alignment tolerances: axial: $< \pm 2$ mm radial: $0 \dots < 10$ mm * angular: $< 0,2^\circ$</p> <p>* radial displacement and radial alignment tolerances depends on length</p>
<p>coupling with rubber joints</p> 	<p>tooth coupling</p> 
<p>displacement capacity: axial: $\pm 5 \dots 20$ mm radial: $0 \dots < 500$ mm * angular: $< 6^\circ$ (peak)</p> <p>alignment tolerances: axial: $< \pm 2,5$ mm radial: $< 0,10$ mm * angular: $< 1^\circ$</p> <p>* radial displacement and radial alignment tolerances depends on length</p>	<p>displacement capacity: axial: $< \pm 10$ mm radial: $\pm 0,5 \dots 3$ mm angular: $< 0,5^\circ$</p> <p>alignment tolerances: axial: $< \pm 2$ mm radial: $\pm 0,2 \dots 1$ mm angular: $< 0,2^\circ$</p>

Figure 13: User interface of the shaft line generator.

Antriebsgröße	1515	
Leistung	2000	[kW]
Drehzahl	750	<input type="radio"/>
	900	<input type="radio"/>
	1000	<input type="radio"/>
	1200	<input type="radio"/>
	1600	<input checked="" type="radio"/>
	1800	<input type="radio"/>
Länge	8000	[mm]
Höhe	800	[mm]
Horizontaler Versatz	0	[mm]
Vorgelege	kurz	
Riemenscheibe an ZV	nein	
max. Anzahl Lager	3	
GV Hersteller	GEVES	
Klasse	LROS	
Vorschläge machen?	ZV + GV	

Bearbeiter	Heer	Revision	2
Projektnummer	TUG - 4711		
Auftragsnummer	1448454		

Version	26.02.09
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Bei vorhandener ZV	# Lager	ZV-Länge	ZV-Länge	ZV Ergebnis	Auswahl
Hohlwelle	3	5194,6	5194,6	OK	
			3400 < ZV < 6650		

GV-Länge	GV Ergebnis	Auswahl
1.002,1	OK	
1000 < GV < 2500		

Ausführung der Zwischenwelle	Identnummer Vorgelege	Länge Vorgelege	Identnummer Flansch	Ø Flansch	erforderliche Gelenkwellenlänge	Länge Gelenkwelle	Identnummer Gelenkwelle	Ø Zwischenwelle	Anzahl Lager	Zwischenwellenlänge zulässig von	bis	Länge Zwischenwelle
Hohlwelle	1129697	850	1086872	315	90	1002,1		281/125	3	3400	6650	5194,6

Gelenkwellenüberprüfung	
Lebensdauer [h]	10000
Antriebsmaschine	Diesell
Klasse	LROS
Beugewinkel [°]	3,71
Vellenanordnung	w
Längenausgleich	ja

Einbauinformationen	
Pointer [mm]	501,81
Winkel α (ZV) [°]	7,42
Beugewinkel β [°]	3,71
EBY	1157635
Pointer Zeichnung	1158746

Anmerkung für GEVES 30
OK!

Interne Ausgabe Drucken

Figure 19: Extreme example of a traditional high-speed arrangement with eight bearings in the shaft line.

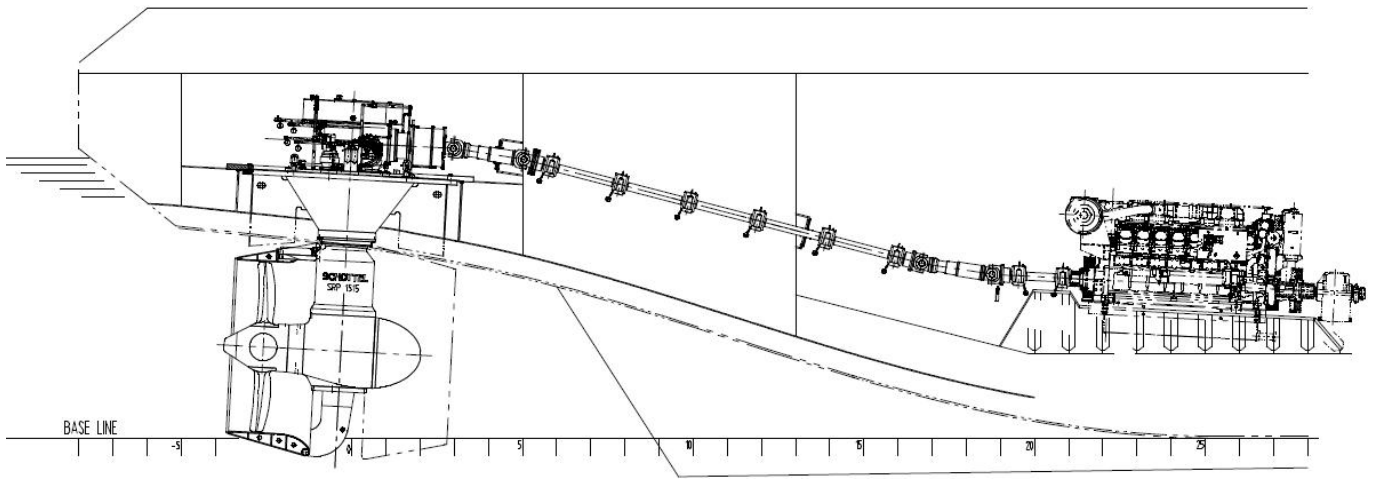


Figure 20: Optimum example of a modern high-speed arrangement with one bearing on 8m installation length.

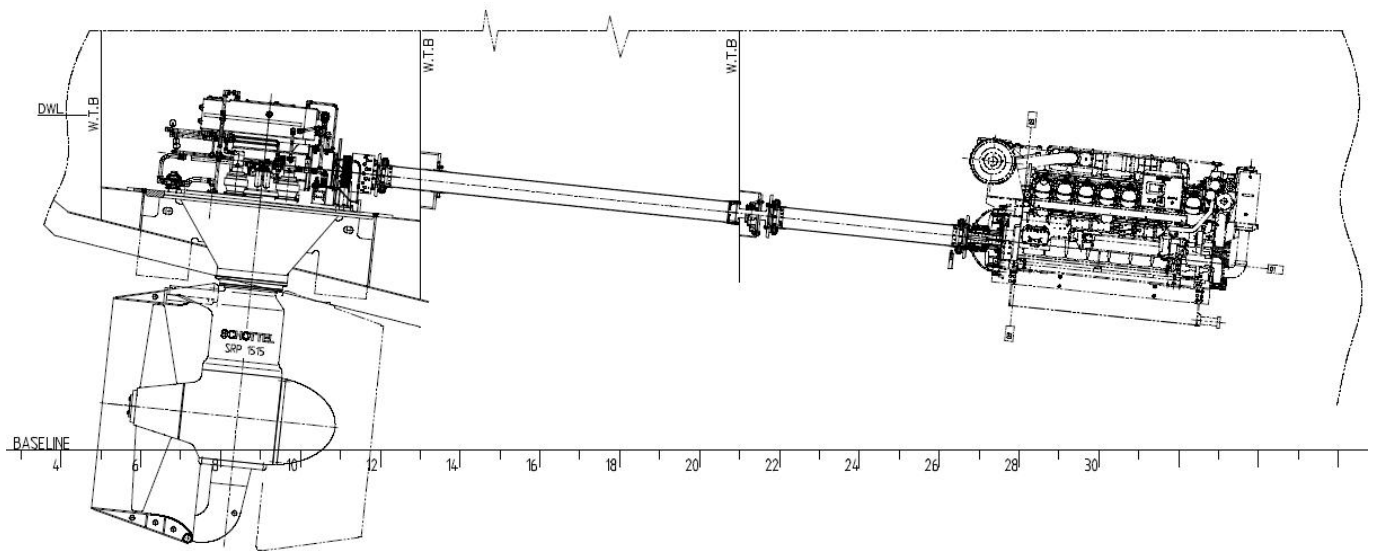


Figure 24: Typical requirements for flange machining.

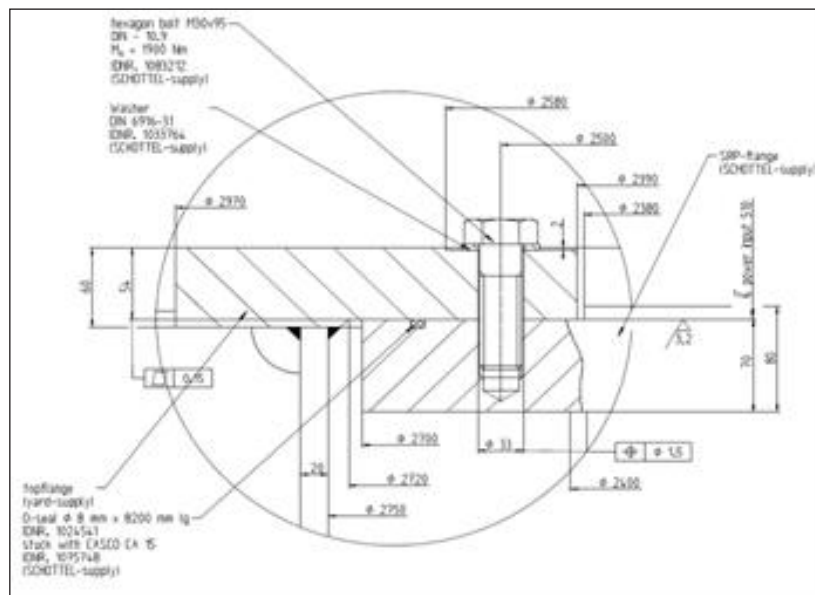


Figure 26: Rubber flange installation showing rubber flange, bolts, spacers and stoppers.

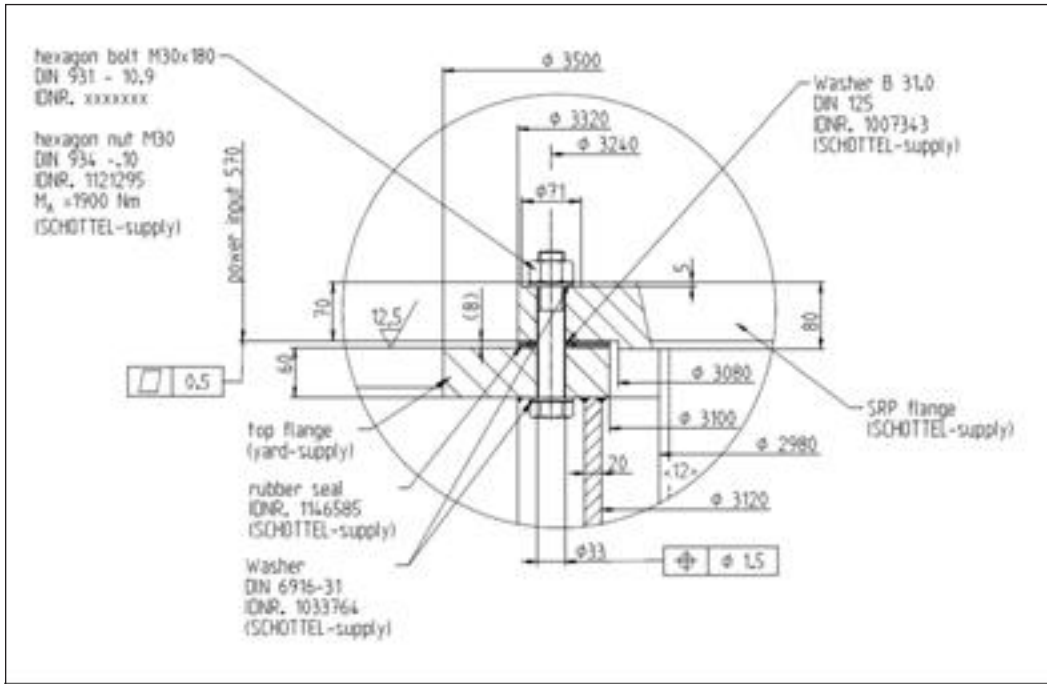
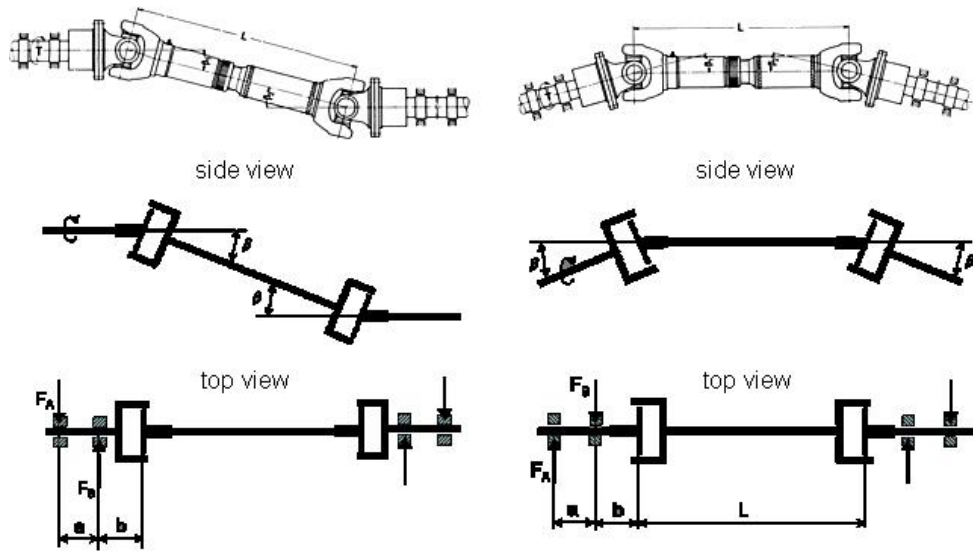


Figure 33: Cardan shaft alignment and dynamic bearing forces under torque and deflection.

Cardanshaft alignment		Alteration index: 0
SCHOTTEL-project:	TUG-050086-05-08	SCHOTTEL-order: 14117253-14117266
A=	8676.8 ± 16 (mm)	
B=	1497.9 ± 8 (mm)	
C=	6022.0 ± 3 (mm)	
m=	7425.8 (mm)	
1		2
Typ:	390.65	Typ: 390.65
Lz=	1280.0 (mm)	Lz= 1280.0 (mm)
Lm=	1398.0 (mm)	Lm= 1398.0 (mm)
K1=K2 ± 1=	701.9 (mm)	K3 = K4 ± 1= 701.9 (mm)
See pointer drawing no.: 1108411		
β = 5.80°		
β1 = β2 = 5.8 ± 0.4°		β3 = β4 = 5.8 ± 0.4°
See In top view max. misalignment engine/clutch: (mm)		
Date: 28.01.2008	Person in charge: T.Klein	Gen. arr. drawing no.: 1160869

Figure 34: Cardan shaft alignment and dynamic bearing forces under torque and deflection.



Universal shaft arrangement with equal deflection angles and equal bearing distances

$\alpha = 0^\circ = 180^\circ$	$F_A = F_B = 0$	$\alpha = 0^\circ = 180^\circ$	$F_A = 2M \cdot \frac{\sin \beta \cdot b}{L \cdot a}$
			$F_B = 2M \cdot \frac{\sin \beta \cdot (a+b)}{L \cdot a}$
$\alpha = 90^\circ = 270^\circ$	$F_A = F_B = M \cdot \frac{\tan \beta}{a}$	$\alpha = 90^\circ = 270^\circ$	$F_A = F_B = M \cdot \frac{\tan \beta}{a}$

Figure 52: Necessary over-dimensioning of the bearing because of high axial.

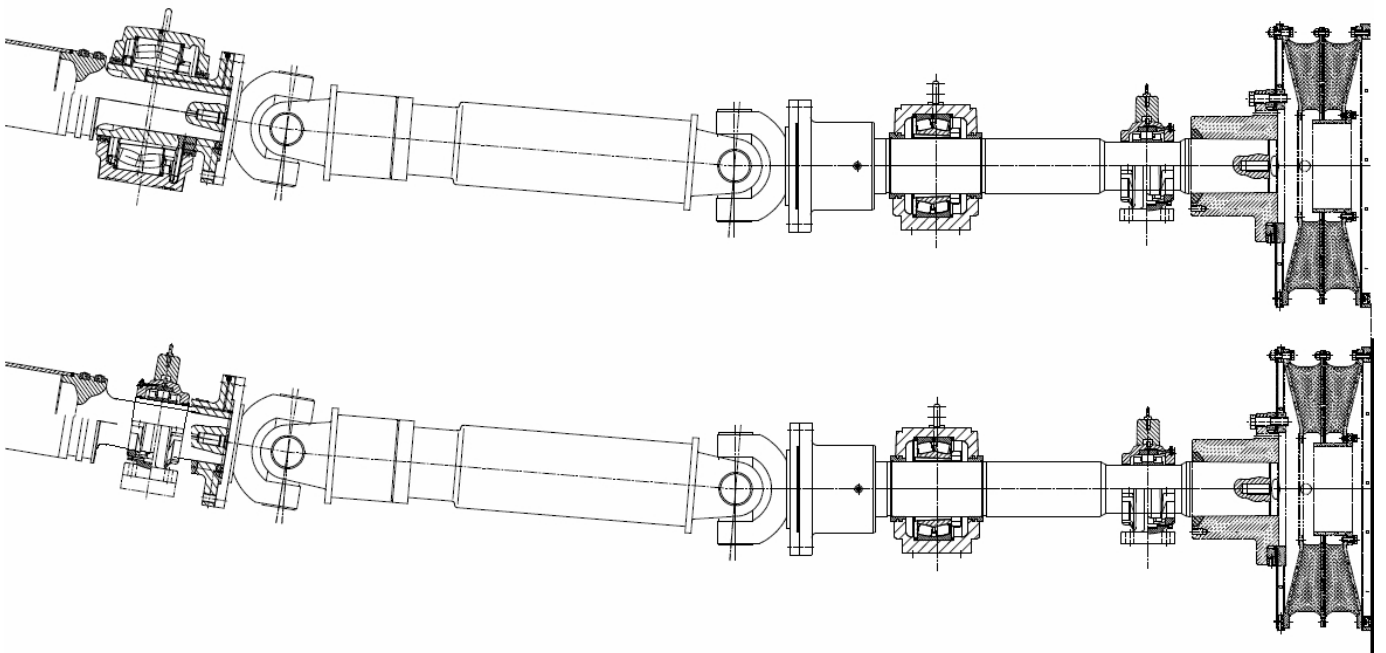


Figure 54: An example of vibration with excessive stub shaft vibrations due to diesel.

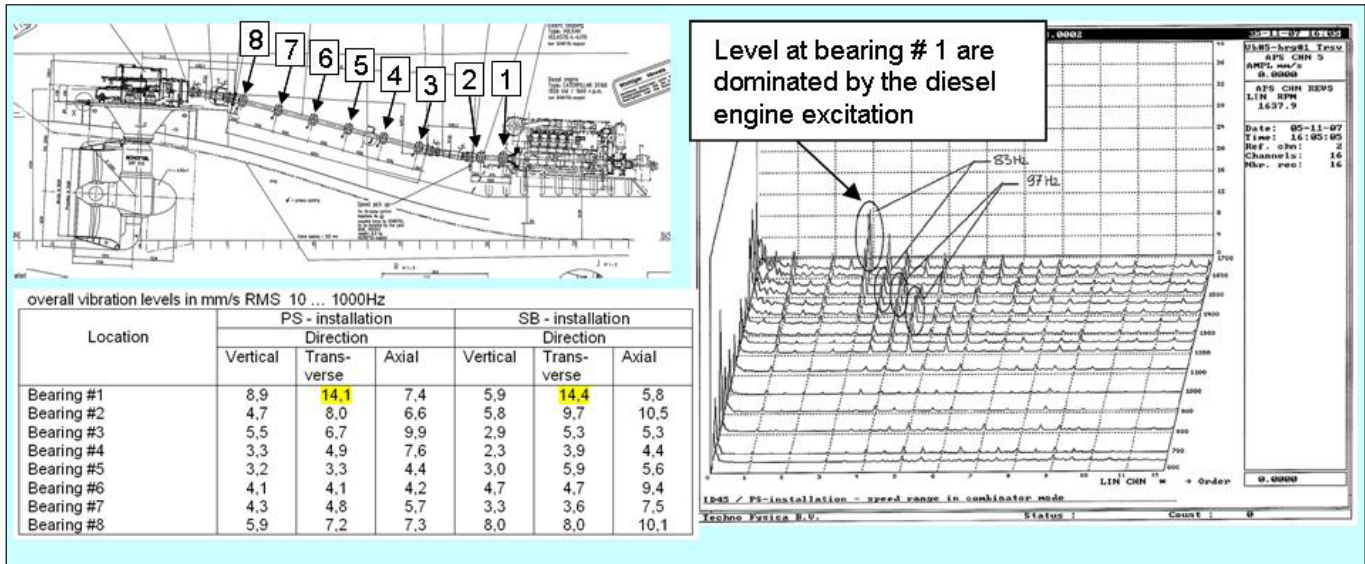


Figure 55: Loading the stub shaft by defined radial offset of the motor to the stub shaft.

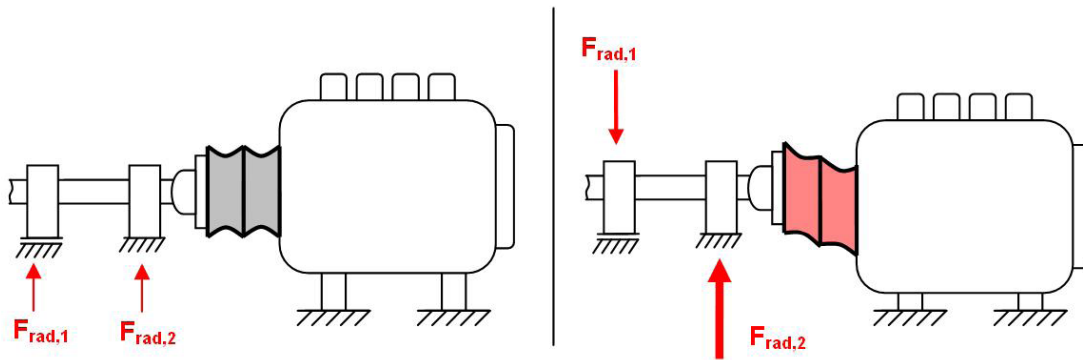

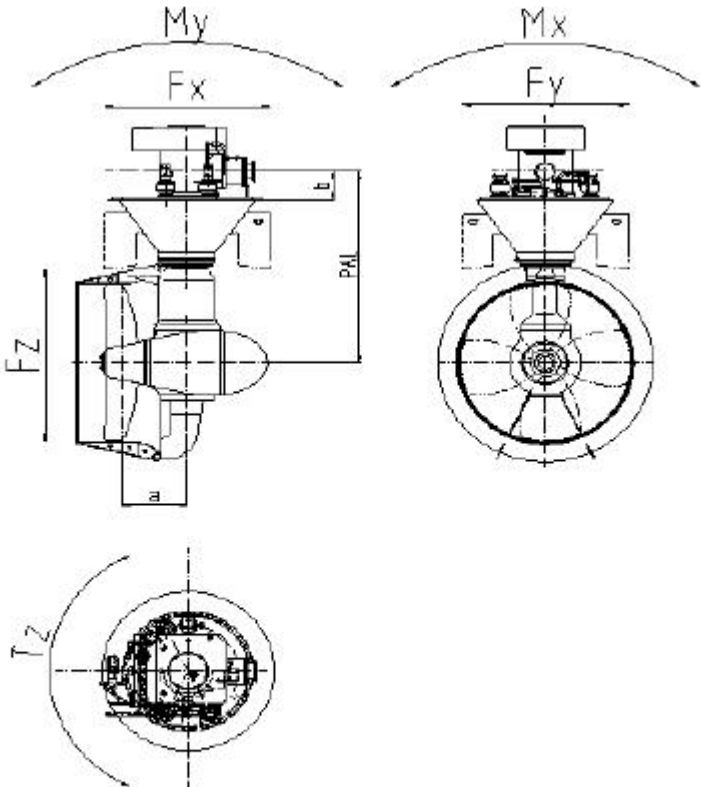


Figure 57: Forces and moments for a nozzled thruster.

Forces and Moments at SCHOTTEL Rudderpropeller			
Type	SRP 1515	Nozzle	
Order Code	ACME-TUG		
Name	M. Heer		
Calculation Number	TKO-01-xxx		
Comments			
Input data		Results	
max. Power	2000 kW	static	dyn. ±
Input speed	1800 rpm	Thrust F_x	340,0 27,2 kN
Reduction ratio	7,235 -	Steering force F_y	0,0 147,1 kN
max. Propeller diameter	2600 mm	Vertical force F_z	255,1 63,8 kN
PAL	3200 mm	Bending moment M_x	76,8 404,7 kNm
Weight	26000 kg	Bending moment M_y	921,4 73,7 kNm
<small>approx. incl. clutch, propeller, oil</small>		Steering Torque T_z	26,2 120,9 kNm
Excentricity propeller/nozzle a =	1000 mm	Total	
Distance input shaft - flange b =	490 mm	Bending moment M_b	1105,5 kNm
Factor dynamic	1,25 -	Torque T_z	147,1 kNm
		Force F	395,6 kN
			
Forces and moments in x- and y-direction rotating around z-axis by steering-manoeuvre of the rudderpropeller !			
Made: M. Heer, 27.2.2009	checked:	TKO-01-xxx	