ADVANCED PROPULSION GEARS FOR LARGE YACHTS

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INTRODUCTION

The market with mega yachts at length of 100 meter and beyond is extremely developing today. Apart from most exclusive designs and interiors, the end customers demand for sophisticated propulsion systems reliably running, easy to operate, providing various operational modes, and generating lowest vibrations and noise. The applications range from conventional diesel engine propulsion to high powered gas turbines, serving for a maximum speed of up to 30 knots and more. With some exceptions, the requirements regarding propulsion technology can be compared to the standards of frigate installations in ways of lowest noise development and operational complexity.

Hence, the main reduction gears transmitting power and reducing speed to controllable pitch propellers or water jets feature similar demands. The general arrangement, reduction ratio, center distance between input and output, and finally auxiliary equipments have to always be adjusted to the shipyard design.

The initiation in this challenging market segment started basically in Italy. The fast vessel “Destriero”, Figure 1, was built by Fincantieri and commissioned in 1991, showing a futuristic design and including a three way water jet propulsion system propelled by 13 MW gas turbines each. However, this boat was not primarily used as a leisure yacht, but focusing in all technical equipments and designed to highest speeds. Consequently, it was awarded for the blue ribbon contest in 1992 crossing the Atlantic ocean at an average speed of 53.8 knots.

Ever since, numerous applications were added, where technical considerations had to be concentrated on low noise radiation as prime, no matter whether combined diesel engine propulsion or sophisticated gas turbine/diesel engine are involved. Naturally, not the under water signature is the ultimate factor to be considered, but structural noise transmission through girders, stiffeners and decks up to the cabins of owner and guests. Today, gear systems on elastic foundations of specific adjustment to noise requirements are state of the art, and even structural refinements considering the machinery lay out in combination with physical interfaces to ship structures are being investigated.

For this, the gear propulsion technology is being steadily developed. Apart from the well introduced high accuracy double helical gearing serving for lowest possible noise generation in the tooth mesh, the gears are equipped with a multitude of components for perfect adaptation to their environment. As examples, elastic supporting elements are available in single or double structures, or propeller thrust bearings are provided as integral version with built in thrust pads or as half integral version in form of a stand alone unit. Even on the level of auxiliaries, noise has to be limited to minimum excitation.

Controls are being continuously refined as well. Digital technology does not only apply to the bridge, but also to main reduction gears. More than 50 % of the gear systems today include even multifunctional computerized local controls interfacing with the prime movers and the ship control system.

1 DESIGN OF LOW NOISE GEARS

Navy ships and mega yachts ask for highly sophisticated propulsion plants based on their special operating profile. The propulsion plants of both types of ships are designed to fulfill the special needs and have to be most reliable.

Luxury mega yachts are well known for the highest quality and comfort standards which presently exist. In
this context comfort means amongst others the lowest possible structure borne and air borne noise levels. Looking at the development of the noise levels as they were reached in the past, design features enabled a significant development.

The bar graph in Figure 2 indicates that significant steps to lower noise levels were possible. This is mainly caused by progressive improvements of the machine tools leading to higher accuracies. At the beginning of the nineties, resilient mountings of the gear box were developed as additional devices for further noise reduction, and established as a practical solution.

But there are also some other design features inside the gear box with regard to the noise behavior, which are most effective in order to reduce airborne and structure borne noise.

The tooth mesh is one of the main sources of the noise excitation transferred from the gear box to the propulsion plant which needs to be observed very carefully. The continuous development of technology to increase refinement of the gear teeth, as the heart of the gear transmission, for the benefit of optimization the lowest noise performance is an important matter. To achieve optimum noise behavior, different criteria of gear design have to be observed, such as selection of bearings (e.g. damping features with regard to the oil film of plain bearings, compared to the noise behavior of roller bearings) and the design of the gear box casing regarding mass and stiffness. In this respect, an extreme importance comes to the correct selection of the macro geometry parameters as well as the tooth correction values. The proper selection has to be done based on the noise requirements of the individual gear box plant, and has to be supported by continuously adapted calculation methods.

After the theoretical part has been finished, it has to be made sure that the appropriate production methods such as heat treatment process, grinding tools etc. are available on the latest stage of technology, to bring the theoretical knowledge into reality.

A gear train is faced with various external influences, such as reaction forces coming from prime movers like diesel engine or gas turbine, reduced and transformed by the corresponding couplings on the input- and output side of the gear box plant.

One of the core decisions is the basic type of gear teeth. In principle, single helical or double helical gears are available.

Involute gears theoretically mesh without periodical angular deviation in rotation and without dynamic excitation. However, due to manufacturing deviations, misalignment and elastic deformations under load, this theoretical optimum is not achieved in reality. Manufacturing and alignment can be addressed by optimum quality with regard to gear grinding, assembly and commissioning. Deformation under load cannot be avoided but addressed properly by smart design and appropriate flank modification. Above all, the macro geometry still is the decisive criteria on noise excitation. That means that only optimized macro geometry allows for optimum noise behavior.

MACRO GEOMETRY

Single helical gears, Figures 3b) and 3c), feature a significantly better performance than spur gears. Normally, narrow angle helices are applied to reduce tooth load impacts due to the overlap ratio, meaning several gear teeth being in contact over the face width and gear teeth getting into mesh continuously starting on one end of the face - a significant benefit compared to spur gears. But, with single helical gears, major disadvantages have to be considered.

Based on their macro geometry single helical gears generate high axial forces which have to be compensated by special bearing arrangements. The axial forces cause tilting moments, which, by bending the shaft and displacement in bearing clearances, have a bad influence on the tooth contact. As a consequence of this, very often an uneven load distribution across the tooth face width is observed, which has a negative influence on endurance life and noise behavior of this gear mesh.

In addition the axial forces generate bending of the gear box casing, which has to be compensated by increased stiffness of the casing. This means added structural weight.

The high angled single helix acc. to Figure 3b), as proposed by ref. (5), would definitely show a much better noise significance compared to low helices due to the high overlap ratio, but generates inadmissible high axial loads which can not be compensated in realistic views with moderately sized thrust bearings and reasonably designed casing structures.

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Finally, the double helical gear, Figure 3a), is regarded as the consequent resolution of the aspects above. The macro geometry shows that a high helix angles can be realized, where the axial forces will be compensated within the gear. This self centering effect has also a positive influence to the contact pattern through out all loading conditions.

Based on the compensation of the axial forces within the gear, the gear box casing can be designed with less reinforcement which saves additional weight and space. The high overlap ratio guarantees a smooth tooth engagement which reduces the noise excitation of the tooth mesh significantly.

MICRO-GEOMETRY

With the calculation and design of the micro-geometry, the shape of the flanks both in profile and lead directions achieve an optimized contact pattern in the gear mesh, respecting the load profile and low vibration impacts. The results of the investigation concerning the micro geometry are transferred to the grinding machine where correction values in the range of micrometers by highly developed machine tools have to be transferred into practice.

To ensure the appropriate quality performance, the referring machine tools and the accompanied high precision measurement equipment are located in a temperature controlled environment, stable within ± 1°C independent from outside temperature variations.

Beside the correct and careful selection of the macro- and micro geometry of the gear teeth based on calculation and the experience and expertise of a gear box manufacturer, there is a certain importance on the arrangement of different components and design principles inside the gear box which finally makes the gear box a reliable high technology component within the propulsion plant. Some of these components and design principles will be introduced in the following.

2 GEAR COMPONENTS

MULTI-DISK CLUTCH

Based on the request of the market for modern marine gear units in the 1970’s, RENK developed and standardized specially designed multi-disk clutches for a wide range of transmitting torque. Already in 1976, extensive tests on a specially built test rig for multi disk clutch plates of a large size (diameter 740 mm) were performed at the RENK facilities.

Based on these test results the basic parameters for the RENK multi-disk clutches have been determined. A principle cross-section of a RENK multi-disk clutch is shown in Figure 4. The hydraulic piston unit which engages the coupling is almost independent from the shaft rotation. The oil pressure in the piston is not influenced by the shaft speed; and therefore no significant engagement effect at high shaft speeds in disengaged condition can be observed. A specially selected sinter bronze material for the disc coating guarantees for a constant characteristic of the dynamic friction coefficient during engagement and over a long operating time. Special spring enforced separation of the disks in disengaged condition guarantees an equal distribution of the axial clearance over the complete disk package. This feature allows high differential speed in the clutch in disengaged condition without wear or excessively high drag torque.
Special oil grooves in the sinter bronze outer clutch plates enable an optimum cooling oil flow even in engaged condition.

Integration in the gear unit components in various arrangements are available, like
• integration inside a pinion, in case the space available allows for this design (Fig 4)
• fitting in line between an input shaft and a pinion,
• assembled on the opposite side of the input shaft in quill shaft design for easy access of the clutch plates.

The clutch is engaged by hydraulic oil pressure, built-up in two controlled steps which results in smooth engagement without unacceptable engine speed drop.

**Figure 5** shows the principle process diagram of a computer controlled engagement of a multi-disk clutch.
These multi-disk clutches have been reliably operating for more than 40 years within a power range from 100 kW up to 18,000 kW. All clutch types and sizes as mentioned above are designed and manufactured according to the rules of the Classification Societies.

**FLUID COUPLINGS**

The fluid coupling is a hydrodynamic coupling acting as a centrifugal pump consisting of two principle elements. The pump wheel transfers the oil volume by rotation to the turbine wheel. Basic principle see Figure 6.

The energy transmission takes place via the liquid filling, and happens with a minimum of transmission losses of approx. 1.6%.

**Figure 6 - Principle cross section of a hydro dynamic coupling (picture VOITH)**

The fluid coupling is preferably used for so called CODA (CODified diesel and gas) or CODAG (CODified diesel and gas turbine).

One of the main advantages of the fluid coupling is the significant reduction of torsional vibration peaks. The two diesel engines which drive into a CODAD plant have certain load differences. The CODAG plants with its total different characteristic of the prime movers (diesel engine and gas turbine) require a proper integration of their different load contribution. Measurements on the diesel engine input shaft of a large CODAG plant show that the load peaks aft of the fluid coupling are significantly reduced, Figure 7.

Fluid couplings can be equipped with means to fill and drain the transmission fluid and hereby engage and disengage the coupling. Specific features of this hydrodynamic clutch are resistance against wear and maintenance free operation.

**Figure 7 - Torsional vibration measurements before and behind fluid coupling**

The above mentioned advantages of a fluid coupling end up in a reduction of the noise and vibration signature of the propulsion plant but have to be paid by an increase of the efficiency which is mainly caused by the slip.

**Figure 8 - Low noise CODAD plant for a luxury mega yacht with quill shaft arrangement, double helical gears and fluid couplings on both input shafts**

**SYNCHRO-SELF-SHIFTING (SSS) CLUTCH**

The initials SSS stand for the “Synchro-Self-Shifting” action of the clutch, where the clutch driving and driven teeth are phased and then automatically shifted axially into engagement when rotating precisely at the same speed. The clutch disengages as soon as the input speed slows down relative to the output speed. Figure 9 shows the basic elements of an SSS clutch.

The SSS clutch has become the most widely used main propulsion clutch in gas turbine propelled naval propulsion plants.
The SSS clutch can be selected under the condition that the prime mover is able to speed up the plant with the clutch engaged which can be done with a gas turbine.

Synchro-Self-Shifting clutches are designed for high power and high speed prime mover applications like gas turbine propulsion systems. Based on a pure mechanical but highly sophisticated design, the SSS clutch is very reliable and able to handle complex change over modes between different prime movers.

3 PROPULSION ARRANGEMENTS

COMBINED GEAR BOX, CODAD

Considering different operating profiles, space requirements and weight restrictions, combining propulsion plants get more and more attractive. But not only pure technical reasons are the basis to prefer a combined engine arrangement, there is the desire for redundancy and therefore for increased flexibility in speed selection and safety.

Figure 10 shows a CODAD plant with a quill shaft arrangement on both input shafts. This special low noise arrangement was built (as shown in Figure 8) for a luxury mega yacht with highest requirements regarding noise and vibration.

The quill shaft arrangement is based on a hollow bored pinion shaft. The input shaft goes through the pinion shaft, and gives the advantage to arrange the spacious fluid coupling on the opposite side of the input flange. This is a significant advantage with regard to restricted space requirements which is a major task especially in yacht engine rooms.

The quill shaft arrangement in combination with a multi disc clutch has beside the space saving aspect the advantage of an easy access to the coupling for inspection.

COGAG PLANT FOR YACHT APPLICATION

The propulsion of fast mega yachts via two gas turbines combined by a COGAG gear box has the highest power density.

The engine rooms of most high speed yachts are on the ship yard side well known for their significant lack of space. In addition to that, the permanent weight problems very often lead the designer to the decision for a propulsion plant with only gas turbines.

Figure 11 shows a gear arrangement of a COGAG gear box with double helical gears and a so called nested arrangement of the second gear stage. Based on this gear arrangement the first gear stage, which is realized between the pinion and the intermediate shaft encircles the second gear stage, being allocated between the intermediate shaft and the output gear.

As it can be seen from the principle sketch in Figure 11, there is a significant space saving effect in the longitudinal direction between input- and output flange of a nested type arrangement.

The quill shaft arrangement of the both input shafts allows to arrange the both SSS clutches on the opposite
side of the input flange which has an additional space saving effect, and guarantees an easy access for inspection of the clutches. Figure 12 shows the input shaft assembly (foreground) and the main casing with the two intermediate shafts and the output flange in the background.

Figure 12 – A tailor-made cast aluminum light weight gear box casing as COGAG arrangement, for a high speed mega yacht in the final stage of assembly.

Beside the above described advantages, the nested arrangement in connection with the double helical gear teeth provides symmetrical load distribution via the whole gear train and thus avoids tilting moments within the gear box.

CODOG - AS A FLEXIBLE ARRANGEMENT FOR DIFFERENT PRIME MOVERS.

In general, diesel engines and gas turbines are used for the propulsion of ships. Both of these prime movers have their specific characteristics with regard to size, specific fuel oil consumption, noise, and weight. All of these aspects are important for the ship designer who needs to find the optimal propulsion concept for a yacht.

The CODOG arrangement enables the designer to combine both prime movers based on their advantages. That means, diesel engines can be used up to a medium speed range where they have e.g. very low specific fuel oil consumption.

For reaching the high sprint speed the diesel engines can be stopped and the propulsion of the yacht will be taken over by the gas turbine. Shifting from diesel engine to gas turbine propulsion is realized by a sophisticated combination of multi disc clutch and SSS clutch supported by a logic control system. Schematic see Figure 13.

A quill shaft arrangement for both input shafts combined with a nested design guarantees for a compact gear box, which fulfills the restricted space requirements in yacht engine rooms.

Figure 13 – Arrangement of a CODOG gear box based on a quill shaft design for both input shafts and a nested arrangement in the secondary gear stage.

A clever combination of different prime movers based on a given speed profile leads to a significant reduction in fuel oil consumption over the whole speed range. The result is a reduction of the fuel oil bunker capacity based on given range within the yacht should operate without refueling over an extended operational time.

4 ELASTIC MOUNTS FOR FURTHER NOISE REDUCTION

As it was already mentioned, luxury yachts have developed in the last decades to highly sophisticated systems which have to cope with the highest technical and quality standards.

Again, one of the major concerns is the reduction of airborne and structureborne noise. There are three main sources where vibration and noise is excited, Figure 14:

- The first main source is the prime mover, in general the diesel engine or the gas turbine. Based on their specifics, the main excitations are in a low frequency range for the diesel engine, due to their output speed and cylinder firing excitation. Vibration is reduced by dampers which are carefully calculated and designed by the diesel engine or gas turbine supplier. The diesel engines in general have a damper inside the engine. In addition to that, single- or double elastic mounts are selected in order to reduce the transmission of vibrations into the steel structure of the ship foundation. The connection from the engine to the gearbox is realized by elastic couplings, or, respectively, a combination of different coupling types.

- The second main source is generated by the different tooth meshes within the gear train of the gear box. The frequencies are in a range of 500 Hz to 1200 Hz, given by the gear teeth design and arrangement of the gears.
The third main source is created by the propeller excitations, and is transmitted via the propeller shaft to the gear box. A smart design of the shape of blades and their pitch, the number of propeller blades and the arrangement of the propulsor within the aft ship design of the hull, lead to a reduction of these excitations.

A significant reduction of the vibrations transmitted from the gear box to the ship foundation can be achieved by elastic mounts. In order to reduce the tilting of the gear box based on the high torque transmitted, and in avoidance of self induced low frequency vibrations, the best results are achieved with hard elastic elements. A single elastic mount is shown in figure 14, the principle arrangement of a double hard elastic element is shown in Figure 15.

![Figure 14 – Main excitation sources from the propulsion plant](image1)

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![Figure 15 – Principle arrangement of a double hard elastic gear box mount](image2)

In general, the hard elastic mount is designed as a multilayered element consisting of rubber elements with different stiffness, which has to be carefully tuned to each other. The rubber elements may be arranged with metal plates in between in order to increase the impedance and to tune the natural frequency.

The stiffness of the individual rubber elements depends not only on the shore hardness of the selected rubber material but also on the height of the rubber element and its ratio between length and width.

This kind of elements are predestined to damp in a higher frequency range starting from 300 to 500 Hz and higher. The damping effect is increasing up to 16,000 Hz. Thus the excitations out of the tooth mesh frequency are significantly reduced and a “pure tone” signature can be avoided.

The design of the hard elastic foundation is tailor made for every gear box application based on the individual noise requirements specified from the client side. As it can be seen from the bar diagram Figure 2, a clear reduction of noise and vibration was possible after the single and hard elastic mounts became a practicable solution for ships with highest noise requirements.

The development in the field of elastic mounting has still potential for further development. The so called “active mounts” are discussed for diesel engines and will be ready for implementation in propulsion plant within the nearest future. The principle of the active mounts is to cancel unwanted sound or vibrations by superimposing a compensation signal exactly in anti phase by electronic means.

It remains to be seen how far this new technology will support a further step to lower noise and vibration with a sensitive vessels like mega yachts.

5 CONCLUSION

Main reduction gears for mega yachts show typical features such as strategically arranged clutches for flexible operation, a variety of gear arrangements specifically designed according to propulsion system requirements, and not lastly perfectly adjusted designs to match with stringent noise specifications. This could be achieved by continuous developments over the years reaching up to high technology products today.

6 REFERENCES