The Design and Testing of a Low Noise Marine Gear

J. J. Bos

Introduction

This article offers an overview of the practical design of a naval gear for combined diesel or gas turbine propulsion (CODOG type). The vibration performance of the gear is tested in a back-to-back test. The gear presented is a low noise design for the Royal Dutch Navy's LCF Frigate. The design aspects for low noise operation were incorporated into the overall gear system design. Therefore, special attention was paid to all the parameters that could influence the noise and vibration performance of the gearbox. These design aspects, such as tooth corrections, tooth loading, gear layout, balance, lubrication and resilient mounting, will be discussed.

The back-to-back configuration was built with two gears, intermediate shafts and a torque actuator for load simulation. The tests were done for gas turbine and diesel engine propulsion modes at approximately 3,000 kW power input. This corresponded to a propeller shaft speed of 93 RPM. The torque actuator for this test configuration was rated for a maximum torque of 45 kNm and a maximum speed of 3500 RPM. The required torque during testing amounted to 33 kNm at 875 RPM.

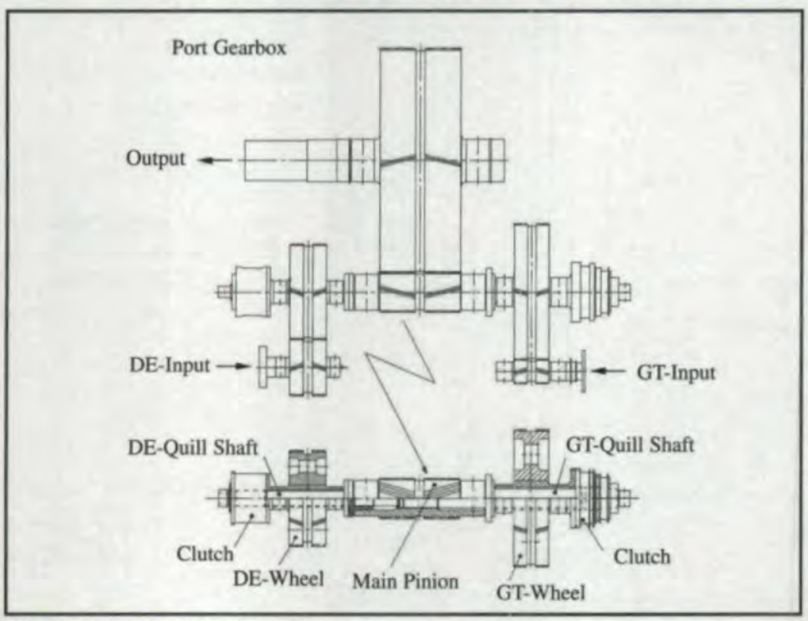

Fig. 1—Layout of the gear elements.

Design Requirements

The propulsion system for the LCF consists of two independent, opposite-handed propulsion lines. One is for the starboard and the other is for the portside propeller shaft line. Each is equipped with a Controlled Pitch Propeller (CPP) as well as boost gas turbine and cruising diesel engine propulsion modes.

The design of each gear set had to meet specific requirements, the most important of which are listed below:

- Gear Ratings

<u>Diesel Engine Propulsion</u>	
Power	5,000 kW
Input Shaft Speed	1,000 RPM
Output Speed	103 RPM
<u>Gas turbine Propulsion</u>	
Power	19,500 kW
Input Shaft Speed	5,450 RPM
Output Speed	164 RPM

- Propeller Shaft Speed Range:
  min. 50–max. 164 RPM
- Input Power Range:
  min. 1,200 kW–max. 19,500 kW
- Oil supply for the gear by gear-driven pump in the propeller speed range of 64–164 RPM.
- Shock resistant for shocks up to 13 g.
- The structure-borne noise requirements for above and below the resilient mounting are defined for a shaft speed of 93 RPM.
- The fulfillment of the structure-borne and air-borne noise requirement shall be demonstrated in a back-to-back test.
- Noise requirements for airborne and structure-borne noise according to Navy specification.

In respect to the above mentioned design requirements, to have optimal corrections for loaded conditions and to optimize tooth loading, the number of rotating elements under load was minimized for both gas turbine and diesel engine propulsion systems. The gears are mounted on a resilient mounting in order to optimize the damping of higher frequency range vibrations to reduce underwater noise levels.

The thrust block for this design is a separate item that is rigidly mounted in the propeller shaft line. The whole gear train in the gearbox has a 15 mm freedom of movement to accommodate the relative movement of the gearbox to the shaftline. This movement is caused by displacements of the gear case due to the resilient mounting and shock loads.

The overall size of the gearbox is approximately 5 x 5 x 4 meters. There are 35 flexible mounts, which require a rigid casing for optimal performance. The stiffness of the casing was checked by means of a finite element analysis. In combination with the flexible mounting, the stiffness of the gear casing is an important feature.

Layout

The first objective was to design a gear layout (see Figure 1) that could meet the requirements as specified within the available space of the gear compartment and with the correct geometric positions for the input shafts of the diesel and gas turbine engines as well as the output shaft. Important features for the design are a balanced tooth load, a minimal number of rotating elements, and the elimination of element rotation when in an unloaded condition.

The layout that was chosen for this purpose was a two-stage reduction for both diesel and gas turbine propulsion modes. The second stage is a common stage for both propulsion modes.

The thrust block is a separate item in the shaft line. The whole gear train is axially positioned by this thrust bearing. The first reduction of diesel and gas turbine input requires, therefore, a connection to the second reduction pinion in the axial direction. The clutches are, therefore, equipped with axial bearings. The rotating parts, therefore, float in the gear casing in an axial direction.

System Design

Gears and Clutches. The construction of the first reduction pinions and wheels is based on solid forgings. The construction of the second reduction is based on assemblies. The main wheel is a fabricated structure shrunk onto the main shaft. A center driven second reduction pinion was required to be able to establish a good tooth contact pattern in both gas turbine and diesel engine propulsion. The construction of the second reduction pinion is a center shaft with a specially designed intermediate sleeve on which the pinion body is shrunk. Due to this special design, the assembly of pinion, quill shafts and sleeve consists of 5 items.

For the several operational modes, two clutches have been built in. The second reduction pinion is connected to these clutches by means of quill shafts (Figure 1). The gas turbine reduction wheel is equipped with a self-shifting, synchronizing clutch connected to the quill shaft of the second reduction pinion. The diesel engine reduction wheel is equipped with a hydraulically operated multi-plate type friction clutch connected to the quill shaft of the second reduction pinion. Table 1 gives an overview of all rotating parts.

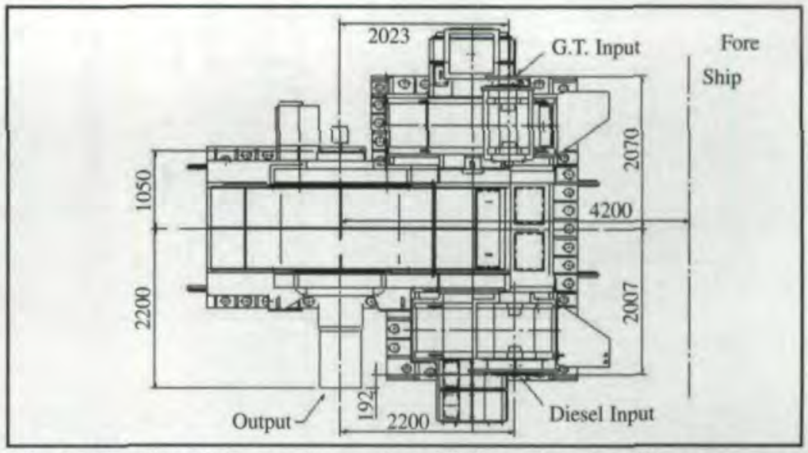

Fig. 2—Top view of the portside gearbox. All dimensions in millimeters.

Table 1—Overview of all rotating parts.

	GT Pinion	GT Wheel	DE Pinion	DE Wheel	Main Pinion	Main Wheel
Material	17CrNiMo6	17CrNiMo6	17CrNiMo6	30CrNiMo6	17CrNiMo6	32CrNiMo12
Heat Treatment	Carburized	Carburized	Carburized	Through Hardened	Carburized	Nitrided
Module	6.5	6.5	6.5	6.5	8	8
Number of Teeth	40	231	94	158	53	305
Quill Shafts		30CrNiMo8		30CrNiMo8	30CrNiMo8	
Main Shaft						C50E+QT

The underwater noise spectrum of the ship can be identified by the first and second order tooth frequencies if they dominate the noise spectrum. The distinctive role of the tooth frequencies in the vibration spectrum of the gear should be avoided.

Therefore, the gear design required a high total contact coefficient, which is realized with a double helical gear design with a pametrada tooth, a 16° pressure angle and a module as small as possible with acceptable tooth load. All tooth calculations are based on Lloyd's Rules of Shipping, DIN and ISO regulations.

The oil pumps mounted on the gearbox also required an optimal design with regard to noise generation.

Tooth corrections are made by correction of the helical angle and tip relief in combination with tooth end relief. The tooth corrections for this gear were based on our own experience and programs for tooth corrections, and secondarily on calculations done by the Design Unit of the University of Newcastle, UK. The program for tooth correction calculation is the DU-GATE program, designed for tooth correction calculations in order to minimize the transmission error. The accuracy level for the

Ir. Johan Bos

is the technical manager of Schelde Gears. He received his degree in mechanical engineering at the Technical University of Delft in the Department for Tribology in 1980. After working in the fields of research and general mechanical engineering, he joined Schelde Gears in 1993. His responsibility is the design and development of low noise reduction gears. Schelde Gears specializes in reduction gears for naval and merchant marine vessels, where stringent noise specifications are applied.

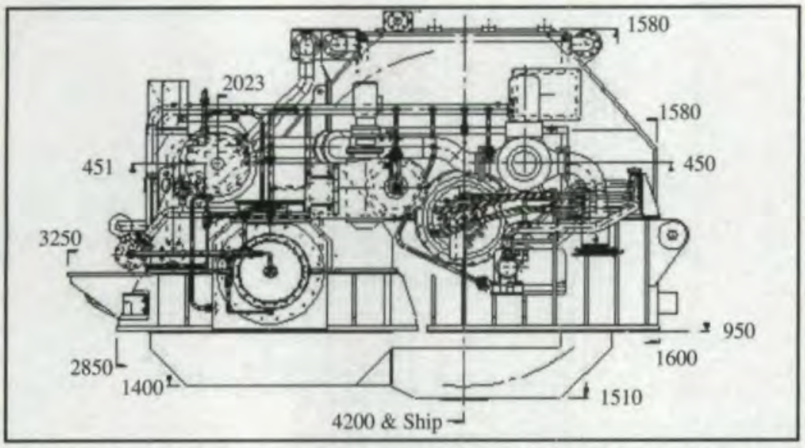

Fig. 3—Front view of the portside gearbox. All dimensions in millimeters.

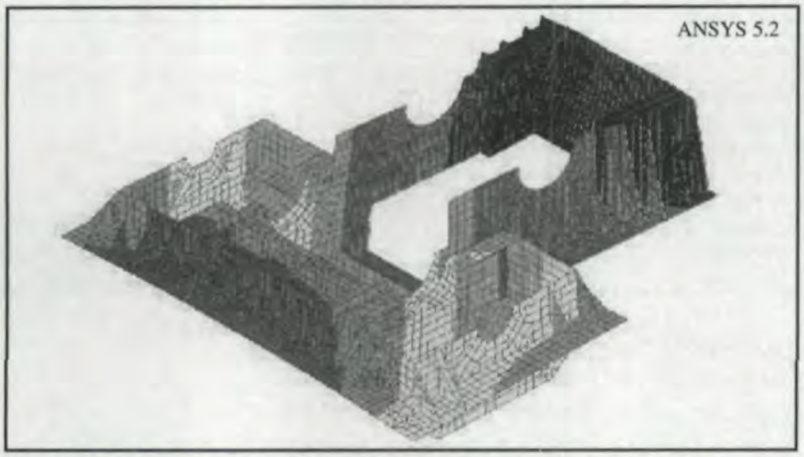

Fig. 4—Displacement calculation of lower part casing for GT propulsion.

gear elements is in accordance with ISO 1328 Class 3 requirements. All these design aspects contribute to low airborne and structure-borne noise levels for the gearbox.

Optimal tooth corrections are based on full load conditions. The calculations for transmission error were made for loads in the operational modes with noise requirements. The tooth corrections are based on bending, including shear, torsion and the bearing position. Other parameters such as wheel deformation and housing deformation were in this case negligible. The design of the second reduction pinion required a central driven construction in order to have optimal tooth loading for both gas turbine and diesel propulsion modes.

Gear Casing Design. The design of the overall gear casing calls for separate casings for the diesel first reduction, the gas turbine first reduction, and the second reduction. The casings are fabricated constructions with solid walls. Noise requirements led to the decision to use a solid wall. For stiffness purposes, a fabricated structure with double wall construction would have served this purpose best. However, because the distance between the shaft center lines and the foundation is relatively low (<800 mm), a double wall casing would be difficult to design and produce. The casings are assembled to one bolted construction before machining is performed. The casing has overall dimensions of 4 x 5 x 4 meters (Figures 2 and 3) and has a separate sump integrated into the hull of the ship.

The stiffness of the gear casing is an important item for two reasons. First, to perform correctly, the resilient mounting requires a stiff gear casing. Secondly, due to asymmetric loading of the casing, misalignment is possible in combination with the resilient mounting and insufficient stiffness of the casing. The stiffness of the casing is checked using a finite element analysis (ANSYS) calculation.

In Figure 4, the calculation result for one operational load is presented. The bearing loads for full torque are applied to the structure for both gas turbine and diesel engine modes with an applied torque of 33 kNm on the gas turbine input shaft and 48 kNm on the diesel input shaft. The output shaft has a torque of 1106 kNm for the gas turbine engine mode and 464 kNm for the diesel engine mode.

These kinds of analyses are performed on all new gearbox concepts, and the acceptance limits for the deformation results are generally based on the bearing requirements. The acceptable misalignment between two radial bearings is limited to an angle of approximately 10^{-5} radians. For axial bearings the acceptance depends on the type of bearing tilting pad or tapered land bearings being used, but the limits are approximately 10^{-3} to 10^{-5} radians.

Bearing Concept. The bearings are located close to the gear elements, giving the most effective stiffness to the pinions and the wheels. The bearing manufacturer uses an accurate program for calculating the bearing dimensions, clearances, required flows and losses. An important aspect for the journal bearings is to define all possible modes of operation. These consist of all relevant combinations of load, load angle and speed. For this gearing, all bearings are journal bearings. The axial bearing for all reductions is integrated into the axial thrust bearing located in the propeller shaft line, approximately 10 m aft of the gearing. As the whole gear train has only one axial bearing, the first reductions of diesel and gas turbine input need to be axially locked to the second reduction for times when the reductions are disengaged or are running engaged. These possible modes of operation, therefore, require that all bearings have an oil supply in all operational modes.

Oil Supply System. A main gear driven oil pump is used for the oil supply to the bearings and the tooth lubrication under operational conditions. The gear driven pump supplies the gear with oil over an output shaft speed range of 50 to 164 RPM. The oil consumption of the gear is almost constant over this speed range, while the oil supply from a spindle type oil pump increases with the speed. To avoid large overcapacity and large overflows at higher speeds, a special pump with constant output pressure and variable flow is used. The result is that for each mode of propulsion in this design, the oil supply is just the required amount. The control of this flow is based on a constant pressure in the main supply line. The input pressure for the various users will, therefore, be constant.

In the propeller speed range of 0 to 50 RPM, and in emergency cases, an electric pump is used. The takeover from electric to gear driven pump is done by a trigger signal at a shaft speed of 50 RPM. The pump will take over within a fraction of a second. The required oil flow is presented in Figure 5.

The oil for the friction clutch engagement is supplied by a separate electric driven oil pump. A gear driven pump directly coupled to the diesel engine input pinion supplies the oil required for keeping the friction clutch engaged.

A separate skid is mounted between the gears to accommodate the lubrication oil filter, cooler and the electrically driven oil pumps and cooling water pumps for both gears.

The Resilient Mounting

As the gear is mounted on a resilient mounting and the gear elements are axially positioned by the thrust block, the whole casing will have movement relative to the gear elements. Under normal conditions this movement could be approximately 0.2 to 0.5 mm in all directions. This, of course, depends on sea conditions. For extreme shock conditions, the movement of the gear casing is limited in the vertical direction to ± 2 mm by shock limiters. However, due to the movement of the thrust block position relative to the position of the gear casing foundation, the total required relative movement could be ± 15 mm in the axial direction. A sketch of the resilient mounting is shown in Figure 6.

The purpose of the resilient mounting is to reduce underwater noise level, thus increasing the difficulty in detecting and recognizing the ship. The noise requirements below the mounts and above the mounts are calculated based on the impedances from the ship's structure and the water.

The resilient mounting was specified to have a natural frequency of 20–25 Hz. The reason for this frequency is that an optimal damping of frequencies is required for frequencies over 60 Hz. A choice in this respect has always to be a compromise. Lower frequency vibrations are normally caused by imbalance and misalignment forces. An effective damping of these frequencies should then require a very soft, resilient mounting with

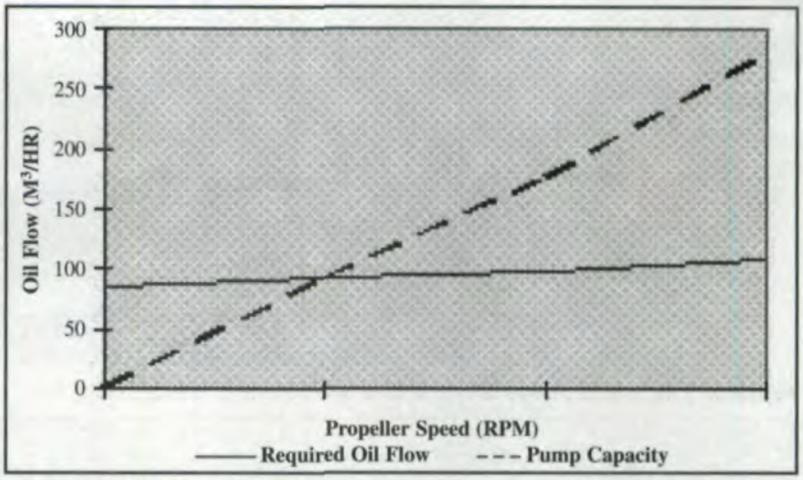

Fig. 5—Oil flow requirements.

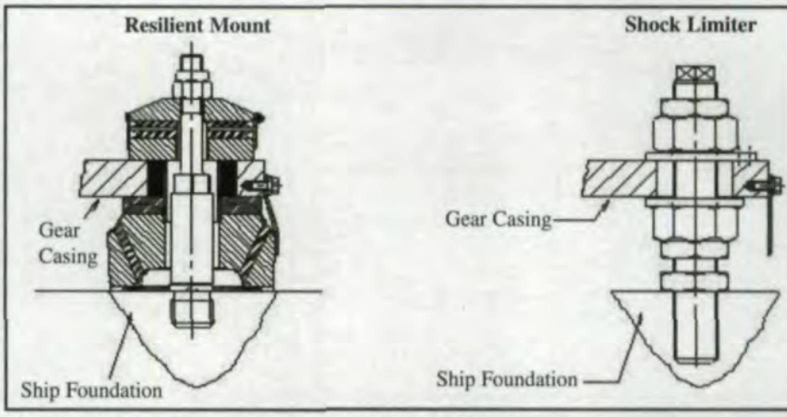

Fig. 6—Resilient mounting and vertical shock limiter.

Table 2—Shaft and tooth frequencies for a low noise operation mode.

POSITION	SHAFT FREQUENCY	TOOTH FREQUENCY
Main Wheel	1.5 Hz	400 Hz
Main Pinion	7.5 Hz	400 Hz
DE Pinion	11.5 Hz	850 Hz
GT Pinion	50 Hz	2000 Hz

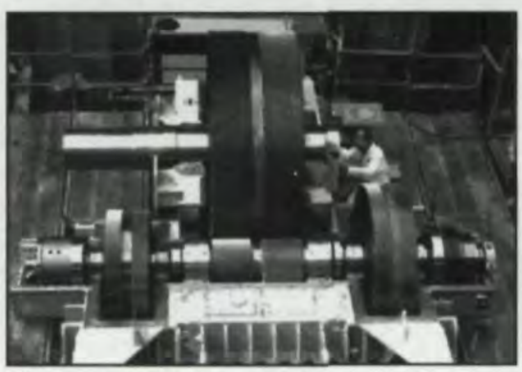

Fig. 7—Assembly floor, portside gearbox.

low natural frequencies. The frequency range for the resilient mounting is especially chosen to reduce the levels for the tooth frequencies in the underwater noise. A list of shaft and tooth frequencies is given in Table 2.

The gearbox, mounted on 35 such resilient mounting devices, is isolated in the vertical and horizontal direction from the ship structure. The rubber compound is tuned with the requirement for the performance of the resilient mounting, e.g. the damping

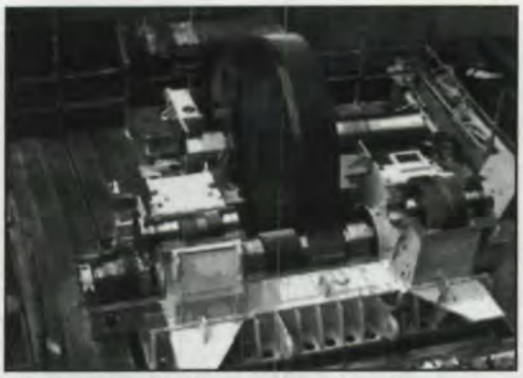

Fig. 8—Assembly floor, starboard gearbox.

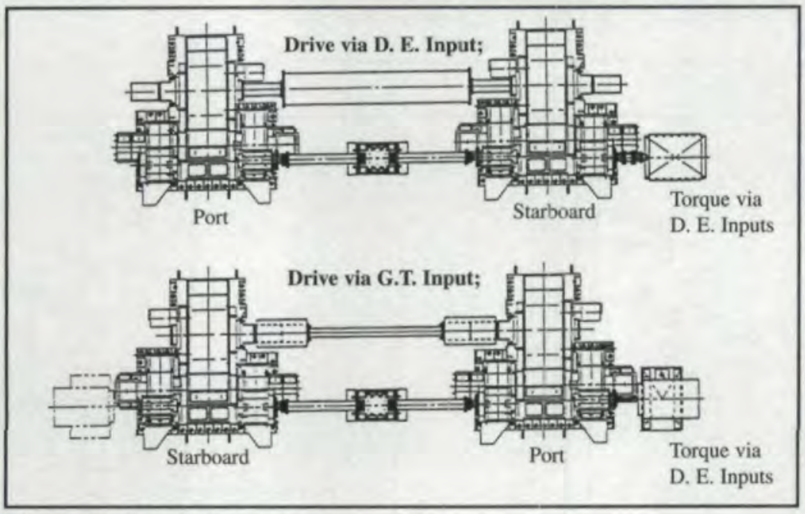

Fig. 9—Back-to-back arrangement.

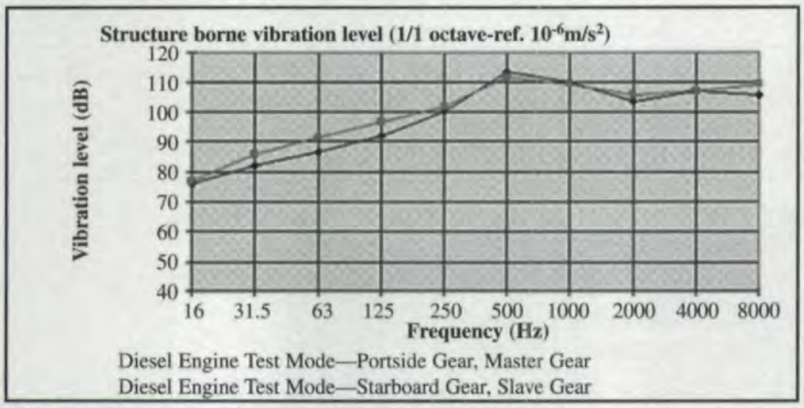

Fig. 10—Measurement results from back-to-back test.

Table 2—Test configurations and conditions.

	Portside Gear	Standard Gear
G.T. Test Mode		
Test Portside Gear	Master	Slave + E-drive
Test Starboard Gear	Slave + E-drive	Master
Position Torque Actuator	G.T. Input	G.T. Input
Load Characteristic	Master-Positive Torque	Master-Positive Torque
	Slave-Negative Torque	Slave-Negative Torque
D.E. Test Mode		
Test Portside Gear	Master	Slave + E-drive
Test Starboard Gear	Slave + E-drive	Master
Position Torque Actuator	D.E. Input	D.E. Input
Load Characteristic	Master-Positive Torque	Master-Positive Torque
	Slave-Negative Torque	Slave-Negative Torque
Test Conditions		
Load		3,000 kW
Diesel Propulsion		875 RPM
Gas turbine Propulsion		2,950 RPM

of the natural frequency as well as the dynamic damping of higher frequencies. The available space for movement is limited by separately mounted shock limiters. These are separate devices that block a further movement of the gearbox.

These requirements are valid for a ship speed of 18 knots. For speeds above that, propeller noise will be dominant.

Manufacturing

The production of the gear casing is an important part of building the total gear system. The casing itself is fairly large and complex with a total weight of 28 tons. The welding process is monitored to maintain constant quality in the welding and dimensions. After welding, the different parts are partly machined. The casing is assembled and the bearing seats are premachined. The final machining of the gear casing, an essential operation by which the center distance of the several shaft lines are machined within narrow limits, is done in a temperature controlled production shop. The precision of this part, achieving optimal alignment between the shaft lines, is an important aspect of low noise design. All rotating elements are ground to a Class 3 quality, ISO 1328. The tooth contact pattern of all interacting gears are checked before they are released for final assembly. All stages in the production of the gears are followed and recorded with regard to the important parameters of each specific stage. Balancing is done separately for each component and partly in the assembled position.

The assembly of the casing with the rotating elements and oil system is the last control to see if all the required tolerances are really matched. During the assembly, dimensions are always carefully checked.

Test Conditions

All gears are submitted to spin and partial load tests. The spin test demonstrates the functional performance of the gear and verifies the stability of the bearing temperatures, the electric system and the functioning clutches. The partial load test demonstrates the performance of the gear with regard to noise requirements. The loaded test is done in a back-to-back test arrangement as shown in Figure 9. The structure-borne and airborne noise of this gearbox requires this back-to-back test configuration for both diesel and gas turbine drive modes at the power ratings for a ship speed of 18 knots. The tests were performed for an equivalent power of 3,000 kW.

The acceptance of the gears required absolute certainty about the performance of structural and airborne noise levels because a possible deviation from the expected data can be corrected better in the factory than it can in the built-in situation aboard the ship. The different test conditions during the back-to-back test are listed in Table 3.

Building the test rig was something new for the engineers at Schelde Gears. Because of this, the influence of each part of the test rig needed to be evaluated, as direct experience with this type of test rig was not directly available. Each component in the test rig, as well as the gear itself, could influence the test results in either a positive or negative way. Therefore, the first objective was to recognize those parameters of influence and eliminate them as much as possible.

In the back-to-back configuration, the main shafts are coupled. Then, depending on the test mode, either the diesel input shaft or the gas turbine input shaft is also coupled to drive the slave gearbox (see Figure 8).

The gears were mounted on a resilient mounting like those designed for the ship's foundation during all the tests. The alignment of the gearboxes in the back-to-back test needed special attention because the bearing loads had to be about equal to the loads expected under operational conditions. This required different alignment procedures for the diesel and gas turbine engines. The main concern for this part is the flexibility of the main shaft line. In this shaft line, torsional stiffness needed to be combined with a certain degree of bending flexibility in order to maintain the proper bearing load division on the main bearings.

The input shafts were connected to the torque actuator with flexible couplings on the intermediate shafts. The weight of the intermediate shafts was limited in order to realize a bearing load distribution that is equal to that in reality. The balancing and alignment of all those parts is of significant influence on the test results. Some of those results are presented in Figure 10.

During the back-to-back test, the gear driven pumps supplied the lubricating oil to the gear components. The skid with all the oil equipment is placed close to the test bed. The test conditions for the gear were to be close to normal operational conditions.

The tooth load is generated with a torque actuator. This torque actuator is designed for a torque of 45 kN at a maximum speed of 3,500 RPM. This actuator is designed and built by the Design Unit from the University of Newcastle. The concept of this design has been presented in Ref. 2. The tooth load is adapted in accordance with the output shaft load curve of the gear during operation.

The torque actuator is a vane-type coupling, which enables the torque to be changed during running. The actuator is mounted between the intermediate shafts. Although the shafts had flexible couplings, the influence of the alignment and stiffness of the actuator foundation was considerable. From the actuator, a constant peak of one times the shaft speed influenced the measurements. Improvements of the foundation stiffness and the shaft balancing improved the results. Therefore, the flexible couplings were balanced in their mounted position. The shafts were well balanced, but tests showed that the flexible part in the shaft had a negative influence on the measurements. Balancing the hubs at the primary and secondary sides of the flexible elements of the coupling showed improvement. The shaft orbit was changed from a diameter of approximately 50 microns to less than 10 microns.

The oil pump characteristics also had a great influence on the results. This influence was clearly shown in the frequency area of 60 to 400 Hz and was greatly alleviated by improving the pump design. The pulsation in the oil flow and the stiffness of the pump foundation was shown to have a considerable influence on the vibration levels of the gear. Both of these aspects were improved during the testing phase. The tests for the diesel propulsion mode were influenced by the internal alignment in the multiplate friction-type clutch. Due to the low engaging energy required for this test, special engaging procedures were used for better plate alignment. In Figure 10, the achieved level of vibration is given.

Results. The results of the tests show that the requirements of the specification were met. The realization of the test rig required a careful setup, alignment and local balancing. Above the mountings, the required values are met. At higher frequencies, the line is even below the specification. Each component mounted on the gearbox has its own contribution to the vibration spectrum.

Conclusions

The design of low noise gears requires careful attention for all components, not only for design but also during the manufacturing process. This is in respect to the gear elements and to all rotating equipment that is mounted on the gearbox, e.g. gear-driven pumps.

The back-to-back testing of a gear can only be successful and representative when all operational conditions can be reproduced. This is valid, especially for the balancing of all shafts and couplings, especially for the high-speed shafts. The engagement sequences should be as close as possible to the conditions on board the ship. In case these conditions are not met, the result will give an approximation, but will be contaminated with disturbances from the test rig. ○

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