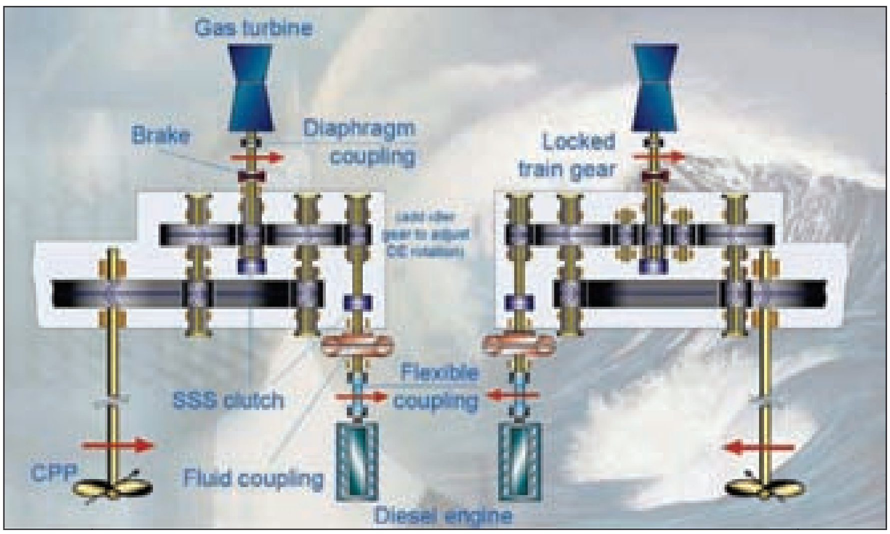
High Speed Gears for Extreme Applications in Industrial and Marine Fields

By Dr. Ing. Toni Weiss, Dr. Ing. Franz Hoppe



Management Summary

Today, advanced gear systems in both the marine and industrial fields are exposed to increasing demands. No matter whether stationary high-speed gears up to 140 MW, or COGAG and CODAG power transmission marine gears are addressed, a supplier of heavy-duty reduction gears has to provide—at the utmost interface—flexibility and the optimized technique for any kind of installation in close co-operation with power plant suppliers, or Navies and shipyards.

Optimization of load carrying capacity and lowest noise performance require ultimate refinement of toothed gears and bearings in continuous development of technology enhancement. In this respect, it is extremely important to select the appropriate macro geometry parameters and tooth correction values, supported by continuously adapted calculation methods. Experimental investigations as carried out by highly respected institutes lead to results transferred directly into the gear design.

Moreover, as theory needs to be transferred to real operable gears with the required accuracy applicable, heat treatment processing and grinding tools are to be continuously adjusted to the latest stage of technology.

Above all, a gear is not just a mechanical transmission, but is developed to a system fulfilling multiple demands, such as clutch integration, selectable output speeds, and controls of highest electronic standards. This paper shows the basics for high-speed gear design and a selection of numerous applications in detailed design and operational needs.

Introduction

Tooth corrections for high-power gears have to be evaluated and manufactured to compensate for all influences which disturb even-load transmission and even-load distribution during operation.

Load distribution in circumferential direction. Today's gear manufacturing technology for high-precision, high-power gears typically provides deviations in a range where an impact on load distribution in circumferential direction can merely be neglected, as remaining pitch deviations are much less than tooth deflections under load.

This, of course, also is an inevitable precondition for adequate tooth and tip relief and in turn even optimization of how tooth and tip relief are affected. The major difficulty of such an optimization is to choose the load for which the amount of correction is selected. The calculation of the necessary amount is relatively easy, as gear mesh stiffness values and calculation procedures are widely available.

When extreme loads are applied to gears designed for normal operation, there is a certain risk of hard contact, e.g., in the dedendum area near the tooth root where tooth contact with the mating gear tip starts. Similarly the influence of tooth and tip relief on gear scuffing has to be observed. A "soft start" of the gear coming in mesh of course also influences gear noise, which is discussed later.

Load Distribution Across Face Width

Elastic deflections. Based on the assumption that manufacturing deviations on even relatively large gears can be minimized to very high quality, especially for any actual gear pair in mesh, elastic deviations will be the predominant influence factors on load distribution, which have to be taken into consideration. Again different load conditions will result in different deflections.

From a load carrying point of view, in most cases the lowest maximum stress is obtained if the deflection at maximum load is considered for tooth geometry corrections. It may be advisable, however, to study these influences for different loads.

As the number of influence factors—such as bending and torsion of pinion and wheel, bearing and housing deflection—will increase, the more accurate a calculation must be. It becomes relatively difficult to evaluate these influences in their common effect, and it is widely accepted that only proven computer programs like RIKOR in its latest edition are a suitable tool for the designer to find the best values for tooth correction.

In particular, helical gears are very sensitive when the tooth ends come into mesh. The effect of typical end relief, which has been state-of-the-art for a long time, can be analyzed in more detail by such a computer program and it is even possible to manufacture adequate three-dimensional corrections.

In any gear calculation procedure, according to valid standards, the load distribution factors are described in more or less detail. In reality, however, the influences are not simply meant to be superposed, but rather observed at any location of the gear mesh field and at various loads. The result can be shown and analyzed in a three-dimensional graph.

A very simple example for a helical gear and a certain load is shown in Figure 1. Whereas this distribution is easy to understand and most likely reflects the conventional expectations, Figure 2 shows a helical gear with a very uneven load distribution and its effect. It can clearly be seen that due to the high local load, the correct tip relief cannot avoid high local stresses, even higher than represented by the load distribution factor.

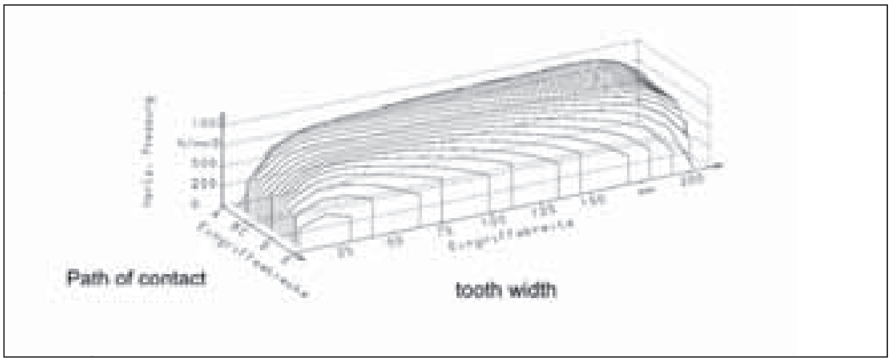


Figure 1—Hertzian pressure distribution for a helical gear with optimized tip and root relief as well as end relief.

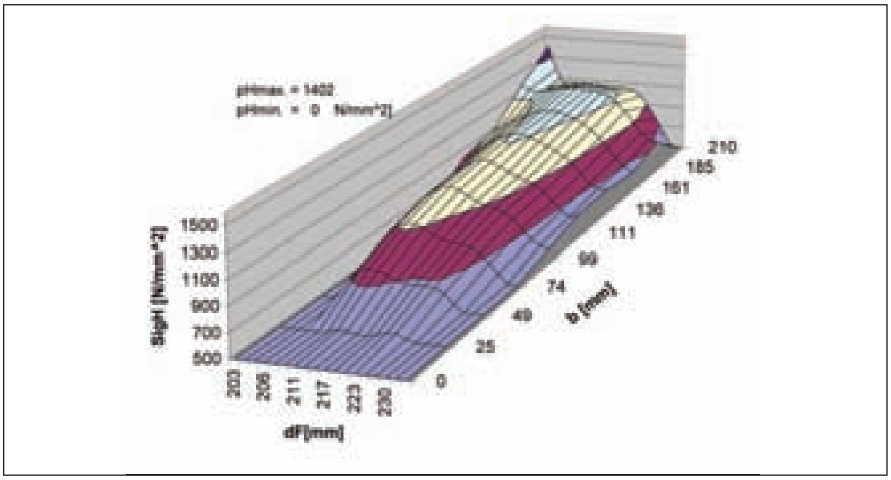


Figure 2—Hertzian pressure distribution for a helical gear with optimized tip and root relief as well as end relief, but assuming a very uneven load distribution ($K_{H\beta} > 2$) for study purposes.

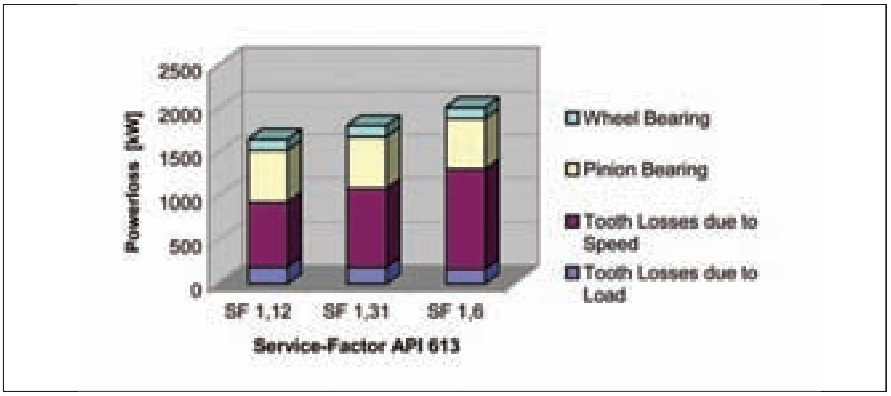


Figure 3—Power loss for a high-power gear calculated for different theoretical safety requirements, i.e., different API Service Factors.

Non Load-Related Deviations

Deviations affecting load distribution due to manufacturing, assembly and alignment are typically measured and to a large extent eliminated for high-precision, high-power gears. They are not discussed further here.

High-power gears, however, are normally running at high circumferential speeds. To minimize inertia or overall gear weight, it is sometimes necessary to reduce the mass of the gear body. Thus the design will be similar to a gear rim and a web. In this case, centrifugal forces can deform the gear rim in addition to the variation in stiffness.

Another effect is gear bulk temperature, which obviously is of great importance for large gear sizes and high circumferential speeds, generating high power loss, because there is always a certain temperature gradient. Therefore due to the negative effect of high temperature on scuffing risk, it is important to reduce gear size to a reasonable minimum. This is not reducing the actual gear safety when all influence factors are sufficiently considered. This must be taken into consideration when choosing appropriate safety factors for comparatively simple calculation procedures. Figure 3 gives an example of gear size and power loss, based on different API service factors.

The higher the temperature and the size, the greater the effect on the gear diameter and length of pinion and wheel will have to be considered. Such effects of temperature on the load distribution can be minimized if the typical temperature distribution is known by a modification of the lead correction for a certain steady state condition or by minimizing the temperature distribution itself by optimized cooling. It goes without saying that this influence is reduced when the gear is optimized when running in vacuum-like conditions as provided with etaX technology, as shown in Figure 4.

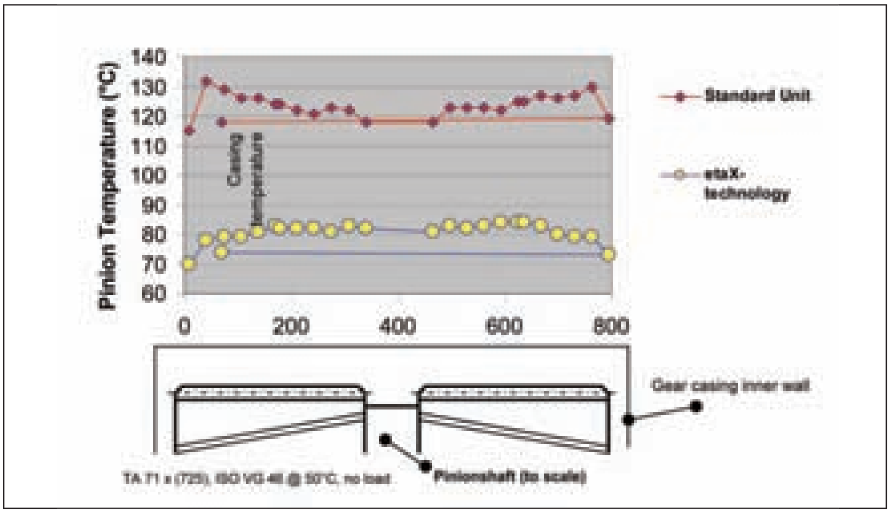


Figure 4—Oil temperature distribution of a 100 MW class gear measured after the mesh for an already optimized standard unit and with etaX technology.

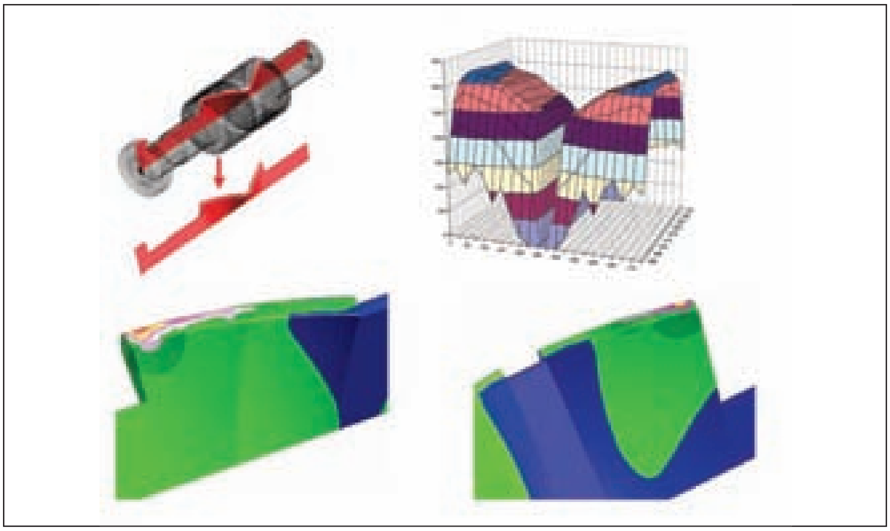


Figure 5—Temperature distribution of pinion and wheel, calculated with an FEM model and its effect on contact stress distribution (Ref. 7).

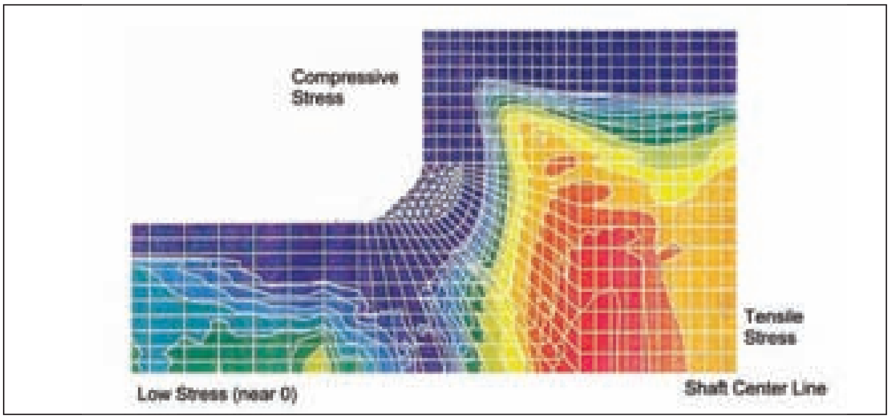


Figure 6—Calculated residual stress distribution for a (half) double helical model gear.

The influence of such an effect can only be calculated using an extensive FEM calculation which takes into consideration all heating and cooling sources and heat dissipation. Several boundary conditions have to be estimated, but when different measured temperature distributions can be realized with good correlation, it can be assumed that the calculation provides realistic results. Figure 5 shows an FEM model of a large double-helical pinion and the temperature distribution with the highest value at the outer end of the helix as well as the corresponding temperature distribution for the mating bull gear. The theoretical influence on load distribution of this temperature distribution then is calculated as an example neglecting all other influences like elastic deformation, etc. It can be seen that there may be significant influence.

As high-power gears for 100 MW and more must have hardened gears to limit the size, the gears and their integrated shaft still have enormous dimensions, especially for case hardening, with the quenching process of a very large solid part. The material for such parts has to be carefully selected and the process of manufacturing the gear blank needs to fulfill restrictive quality requirements. This is especially true because of unavoidable residual stress in the gear body after quenching. The result of such an evaluation is shown in Figure 6, which can be taken as a principle distribution, whereas details are still questionable as to whether boundary conditions and some material behavior have an important effect, but are only known to a limited extent. Stress-relieving heat treatment after hardening is limited in order to not negatively influence the tooth hardness. Various calculations and measurements have proven such residual stress distribution. Stress relief in operation after a long period of time—due to a high number of stress cycles (e.g., start ups)—has been experienced to a certain extent.

This stress relief obviously depends on its original value and on the gear dimensions. It will therefore be different for pinion and gear. In any case, it results in some permanent deformation of the gear body. It therefore is essential to evaluate the expected influence of this deformation on the tooth load distribution and to find an optimum compromise of its effect on gear life.

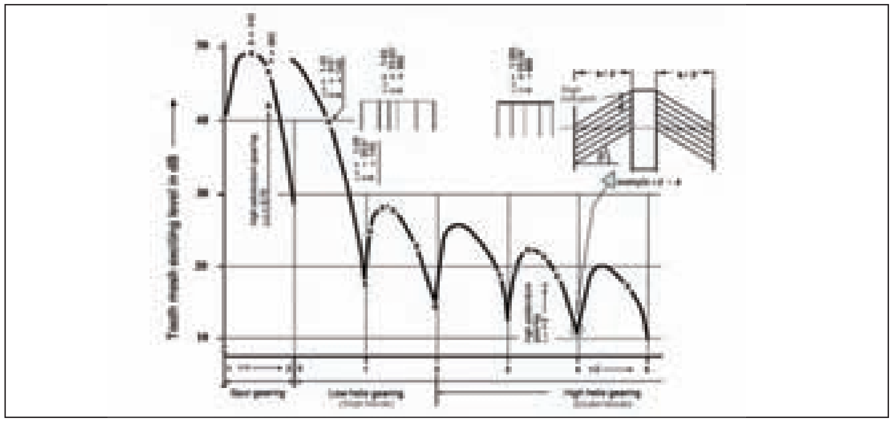


Figure 7—Noise in dB generated in tooth mesh, dependent on basic layout and transverse contact ratio εβ. Note minima achieved with integer ε_β values.

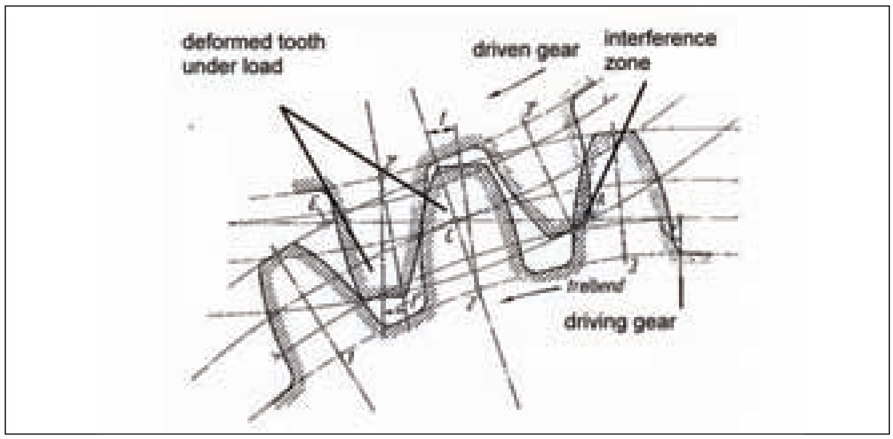


Figure 8—Tooth deflection under load and subsequent interface zone in following gear mesh (Ref. 6).

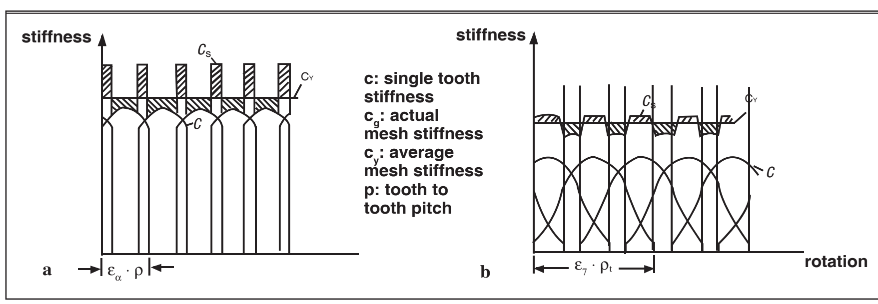


Figure 9—Mesh stiffness: a) Spur gear mesh; b) Helical gear mesh (Ref. 6).

Optimized Geometry of Low-Noise Gears

Source of Noise. A main power transmission gear is subjected to various external influences, such as reaction loads from adjoined external couplings, foundation distortion, or dynamic mass forces caused by heavy sea states, and, not lastly, heat expansion due to the power loss generated by gear teeth and bearings.

In light of all these impacts and with respect to low noise signature, the tooth design has to be specifically observed, as pinions and gears represent "the heart" of a gearbox. The decision on the basic type of gear teeth is important, where principally spur gears, single-helical or double-helical gears are available.

Figure 7 shows the principle coherence between tooth mesh noise excitation and overlap ratio, ε_β. With spur gears, ε_β equals zero, with low single helices, ε_β values up to 2 are achievable. High helices are in a practical sense realized only with double-helical gears, achieving $\varepsilon_\beta > 3$. Apart from significant noise reduction at increased εβ, excitation appears minimal with integer value of overlap ratio. The fundamental results as depicted in Figure 7 are still considered as state-of-the-art and have been confirmed throughout the past 25 years with numerous research programs, supported by experience with countless applications in service.

Involute gears theoretically mesh without periodical angular deviation in rotation and without dynamic excitation when the gear teeth get into mesh. However, due to manufacturing tolerances, misalignment and elastic deformations under load, this theoretical optimum is not achieved in reality without use of appropriate design criteria.

Hence, these influences are to be compensated by adequate tooth flank corrections generated on high-precision grinding tools. Above all, the macro geometry still is the decisive criterion on noise excitation, as shown here.

The geometric ideal position is changed under load due to the Hertzian deflection on gear flanks, bending of the teeth, and elastic deflection of gear bulks and shafts. Even when considering deviation-free tooth flanks—as almost achievable with modern manufacturing methods—interference between the gear teeth in mesh occurs due to elastic deformation, causing periodical noise excitation as shown in Figure 8. This interference can be compensated by appropriate flank modification, achievable normally for one load level and one average stiffness value. Thus, gears requiring low noise emission in a wide power range, e.g. naval gears, need a minimized amplitude of the total mesh stiffness, c_γ.

Figure 9 shows the course of mesh stiffness over rotational angle for spur

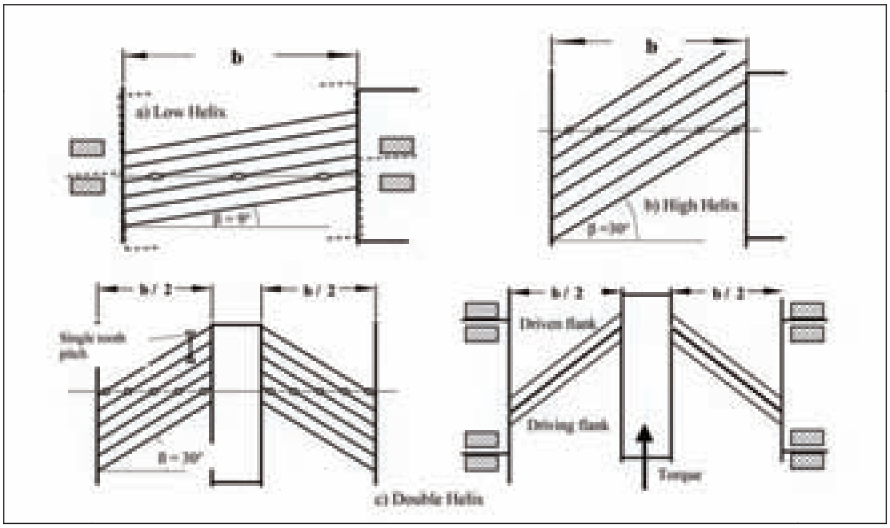


Figure 10a—Low helix, maximum overlap ratio ε_β =3; Dotted: position in bearings under load (overscaled). Figure 10b—High helix (Ref. 5). Figure10c—Double helix at overlap ratio ε_β = 5, here sole radial displacement in bearings at no shaft tilting, thus equal tooth contact over entire face width at all load conditions.

Table 1—Gear teeth base (macro) geometry of investigated gears, gear layouts, Figure 10, center distance 400 mm, ratio 3:1.

Tooth Design	*a) Low Helix*	*c) Double Helix*	*b) High Helix*
Module m_n mm	6	4.5	6
Profile angle α	15°	15°	17.5°
Helix angle β	14.3°	31.7°	28.7°
Transverse contact ratio ε_α	2.0	2.0	2.0
Overlap ratio ε_β	3.0	5.0	5.0

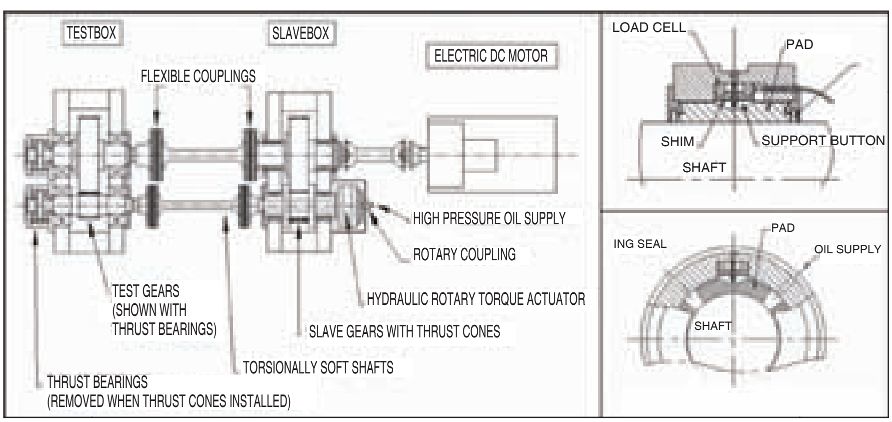


Figure 11—Test rig arrangement (left) and bearing load cell installation (right) of marine noise test rig (Ref. 5).

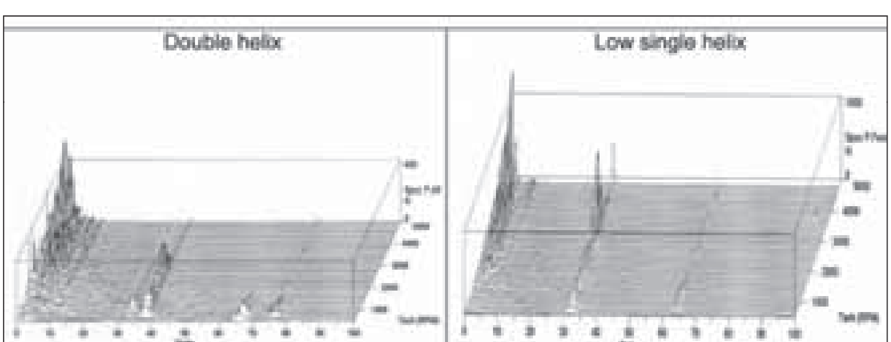


Figure 12—Typical waterfall plots showing significant frequencies in order and peaks with low helical and double helical gears. Gear mesh frequency at 38th order with double helical gears at 32nd order with single helical gears.

and helical gears. The mesh stiffness is the sum of the individual single tooth stiffness values for the teeth in contact shown on the bottom side of the diagram. Obviously, with an increasing number of teeth being in mesh at the same time, the amplitude of the periodical change in mesh stiffness decreases. The number of teeth in contact is determined by the transverse contact ratio, ε_α, (number of teeth in mesh along the path of contact; 1–2 for most gear applications) and the overlap ratio, ε_β (number of teeth in mesh along the face width; zero for spur gears, up to eight for double helical gears). The decisive total contact ratio, ε_γ, is the sum of ε_α and ε_β. Integer figures for ε_β further improve the mesh quality (see Figure 7).

Selection of Basic Geometry for Noise Attenuation. Coming to more distinct views on these principles, spur gears are not suitable for high-speed gears, according to their zero helix angle. As can be seen with Fig. 9a, the tooth load transferred from one tooth pair in contact to the next is followed with a sudden load jump, causing periodical force impacts and, subsequently, increased excited noise. Any attempt to decrease these impacts by profile correction grinding would just end up in some marginal excitation-reduced levels, with the remaining disadvantage of lack of contact ratio and high sensitivity for misalignment and concurring tooth edge overload.

Single helical gears, as shown in Figures 10a and 10b, comprise some benefits against spur gears. Normally, low helices apply in practice, where tooth load impacts are diminished by helix-angle-influenced transverse axial travel of the tooth mesh—a significant benefit against spur gears. But, with single helical gears, major disadvantageous aspects are to be considered:

- Limited helix angle $\beta < 10$ ° is not to exceed axial loads, and still need of separate thrust bearings;
- Low transverse contact ratios, thus limited minimization of noise performance;
- Axial forces cause tilting shafts

in bearings, thus unpredictable non-contact areas and uneven load distribution across tooth face width;
- Axial forces cause bending of casing structures, to be compensated with added up structural weight.

In spite of these aspects, single helical gears could apply if measures such as reinforced casing structures, carefully selected tooth flank corrections, and an increased demand to foundation rigidity are realized. The high helix according to Figure 10b, would definitely show a much better noise significance compared to low helices, due to the high overlap ratio, but generates inadmissible high-axial loads not compensable in realistic views with moderately sized thrust bearings and reasonably designed casing structures (Ref. 5).

Finally, double helical gears, (Figure 10c), include the consequent resolution of the aspects above. They combine insensitivity regarding external forces to casing structures, followed by a high degree of load pattern consistency throughout the full power range—due to their symmetric design—with low noise performance for maximum achievable helix angles at optimized macro geometry of the gear teeth.

In summary, the following aspects support double helical gears:

- Maximum total contact ratio, ε_γ, for the benefit of smooth tooth engagement and lowest noise performance, as shown in Figure 7;
- Radial symmetric tooth forces, no axial impact to bearings, due to self centering effect, as shown in Figure 10c;
- Even contact pattern throughout all loading conditions;
- No axial loads to casing structures generated, thus most light weight designs achievable, by avoidance of excessive casing structural reinforcements other than ultimately required;
- Tooth corrections by grinding respecting just bending and torsional deflection of pinions, at no tilting load or casing deflection unpredictable impacts.

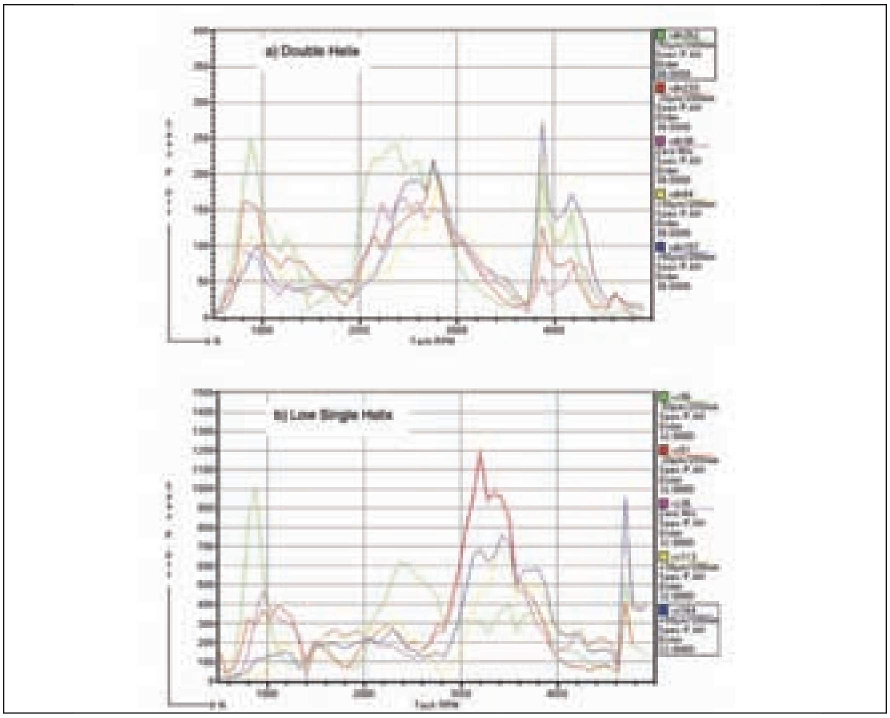


Figure 13—Sensitivity of double helicals (a) and single helicals (b) to variation of misalignment between pinion and gear, single curves as orthogonal slice through waterfall at gear mesh frequency.

Research on Gear Noise. For fundamental investigation purposes, a research program on dynamic bearing forces with different macro tooth geometries, manufactured to ISO quality level 2, was performed at the Design Unit Institute, Newcastle University, UK, supported with funding of the British Ministry of Defense (MoD). In effect, the tests were related to a comparison between the three basic gear designs as per Figure 10. The parameter combinations a) and c) as given with Table 1 were tested on a 8 MW back-to- back test rig and compared to previous results obtained with version b) under the same conditions (Ref. 5).

The test setup and measurement principles are given with Figure 11. In a back-to-back test rig, the slave gear and the testing gear are connected to each other by flexible couplings, the electric motor provides just enough power to overcome the losses of both gear boxes. The operating torque is introduced to gear circle by the hydraulic torque actuator variable up to 15,000 Nm. The test gearbox is supported by servo-pneumatic soft mounts for foundation isolation.

The dynamic excitation generated in the tooth mesh is measured as dynamic forces directly in the journal bearings of pinion and wheel of the testing gear. The bearing assemblies consist of four tilting pads with piezo-electric load cells mounted against the bearing shell, capable of discriminating dynamic forces down to 0.1 N per 100 kN. A special dynamometer at each bearing combines the load cell signals at high resolution and effective frequency response to waterfall results as shown with Figure 12.

Various parameters were subjected to investigation, such as torques up to 15,000 Nm, pinion speeds up to 5,000 rpm, and misalignments between pinion and gear ranging from –100–100 µm per 200 mm face width in distinct steps. For good comparison between the various gear designs, especially related to marine gearing applications, all tests were performed following the propeller law T ~ n 3.

Results of Noise Investigation. In comparison of two typical waterfall plots shown with Figure 12, some interesting basic results are being

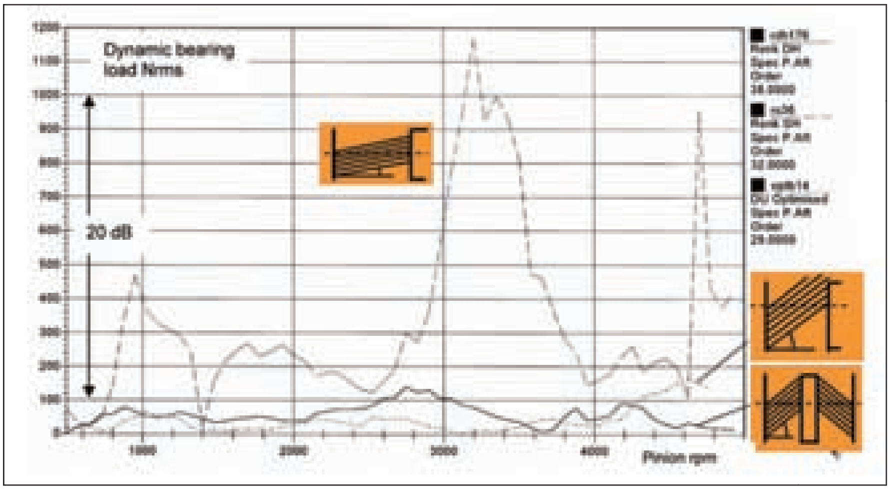


Figure 14—Representative comparison of dynamic bearing loads v. pinion speed (and related propeller law torque) with three gear designs, as detailed in Table 1.

revealed. Double helicals have significantly lower rotational and mesh frequency excitation than single helicals, combined with a broader band behavior. These fundamental results were observed throughout all parameter variations, as obtained with more than 200 single measurements taken.

Figure 13a and b give an impression on the sensitivity of double helicals and single helicals depending on various adjusted misalignment values between pinion and wheel. Basically, with all measurements, a single curve represents an orthogonal slice at gear mesh frequency orders. Evidently, double helicals show lower excitations by a factor of four compared to single helicals, and a fairly low insensitivity to misalignment. This effect is interpreted as the consequence of the self-centering effect in the gear mesh.

As a conclusion, high helices and double helices show a similar performance regarding their dynamic behavior, whereas low helices—even at equally high accuracies—present higher dynamic forces due to their limited macro geometry and structural impacts as discussed above. As an overall representative result, Figure 14 shows single helicals generating up to 20 dB higher structural noise compared to double helicals.

Conclusion

The stat-of-the-art high-speed gear includes a sophisticated gear design respecting any kind of external impacts to operational conditions. Heat expansion and low noise signature requirements are to be compensated by the appropriate selection of macro tooth geometry combined with refined lead and profile tooth corrections. Double helicals show the best performance throughout numerous applications as can be seen with various installations in industrial fields or aboard vessels of any kind. Moreover, if specific solutions are required, such as complex gear trains with multi-functional clutch arrangements, the gear layout is to be flexibly adjusted to environmental conditions, without leave of design principles.

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