

Gear Noise – Challenge and Success Based on Optimized Gear Geometries

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American Gear Manufacturers Association



TECHNICAL PAPER

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Abstract

For gear transmissions different levels of requirements with regard to noise excitation have to be matched. Industrial applications for conveyor belts or cement mills are without doubt much less sensitive with respect to noise emission than military applications, e.g. for navy ship propulsion. However, also for industrial applications the air borne and structure borne noise behavior more and more becomes an important feature.

RENK has been developing optimum gear units for all applications with a transmitted power level ranging up to 145 MW. This paper describes requirements and solutions with regard to noise behavior focusing on examples taken from navy applications and wind turbine gear transmissions. The individual approaches have to be a suit-able compromise to meet the challenge of noise requirement and cost optimization without any restriction on gear load carrying capacity. Therefore, there is no general but individual solution for optimum design.

The paper comprises basic considerations with regard to gear noise, noise requirements and measurements at shop and field tests in comparison to gear geometry and calculation results.

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1 INTRODUCTION

Power generation or Navy vessel propulsions are two examples for the application of advanced gear systems. Typical applications are wind turbine multiple stage step up gears, or CODOG and CODAG marine gears for up to 30 MW gas turbines and diesel engines up to 7 MW to controllable pitch propellers. In any case, the supplier of main reduction gears has to provide the optimized technology for any kind of installation at utmost interface flexibility. This can only be achieved in close cooperation with power train suppliers, or Navies and shipyards.

Furthermore, in continuous development of technology enhancement, single components are subjected to increased refinement such as gear teeth as the heart of gear transmissions for the benefit of optimization of the load carrying capacity and lowest noise performance. To achieve optimum noise behavior, different criteria of gear design have to be observed, such as selection of bearings (damping features) and housing (mass, noise dissipation features) and gear geometry. In this respect, an extreme importance comes to the correct selection of macro geometry parameters as well as tooth correction values, where the proper evaluation of both is supported by continuously adapted calculation methods. However, theory needs to be transferred to real operable gears with the required accuracy applied, and therefore production means such as heat treatment processing and grinding tools are simultaneously to be maintained on the latest stage of technology.

In completion of a modern main transmission gears, the environmental demands on control aspects require an ongoing adaptation. Today, integral programmable logic control (PLC) systems provide an utmost flexibility, safety and comfort for the operation of reduction gears. The gear is not just a mechanical transmission – it is developed to a system with functional sub stations interfacing with the plant's operational environment.

2 GEAR NOISE BASICS

2.1 Tooth Geometry Basics for Low Noise

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

In view of all these impacts, and also respecting in most marine field cases low noise generation requirements, the tooth design is to be focused on in particular, as pinions and gears represent “the heart” of a gearbox. First priority, the decision on the basic type of gear teeth is of importance, where principally spur gears, single helical or double helical gears are available.

Figure 1 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 practical sense realized only with double helical gears, achieving $\epsilon_\beta > 3$. Apart from significant noise reduction at increased ϵ_β , excitation appears to be minimum with integer value of overlap ratio. The fundamental results as depicted in Figure 1 are 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. 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 criterion on noise excitation. That



means that only optimized macro geometry allows for optimum noise behavior. The basic background will be described in the following paragraphs.

Gears will change their geometric position under load due to the following influences:

- Deflection due to Hertzian pressure,
- bending of teeth,
- elastic deflection of gear bulks and shafts.

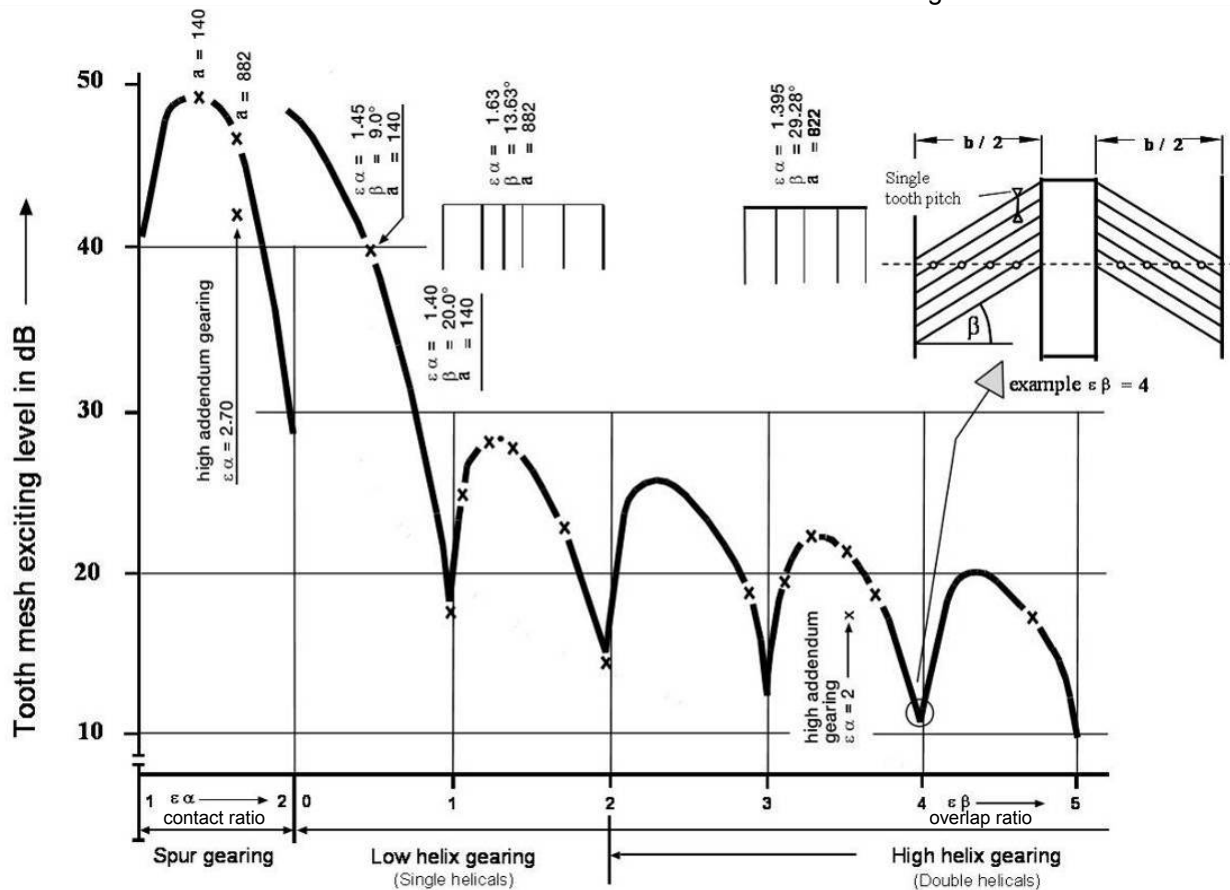


Figure 1 Noise in dB generated in tooth mesh, dependent on basic layout and overlap ratio, ϵ_β , acc. to ref. (2) and (3). Note: Minima achieved with even ϵ_β values.

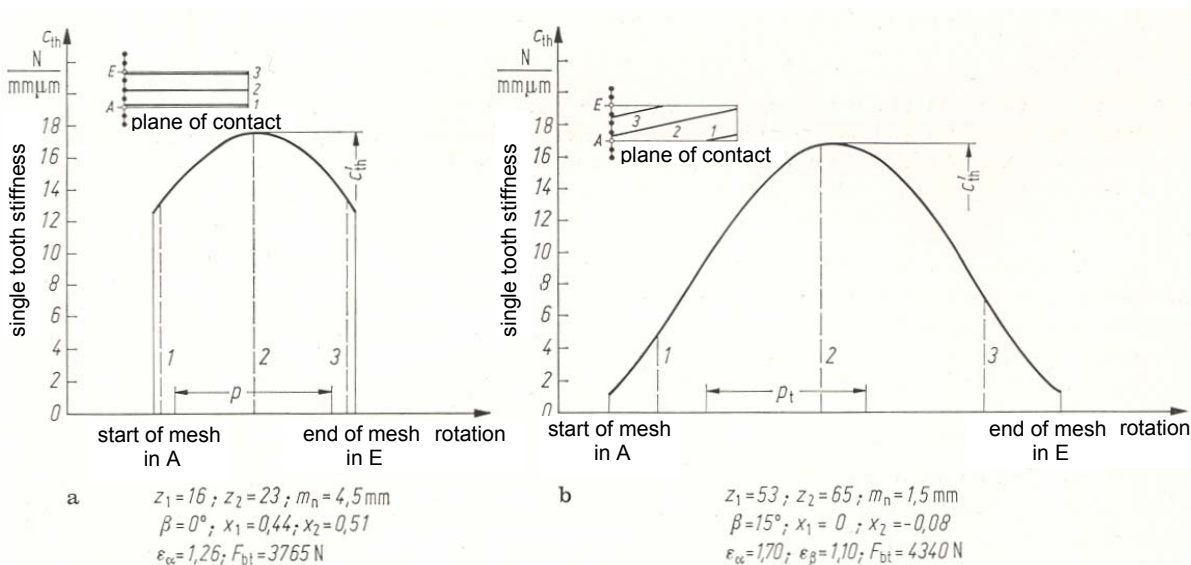


Figure 2 Single tooth stiffness, a) spur gear mesh; b) helical gear mesh; acc. to (6)

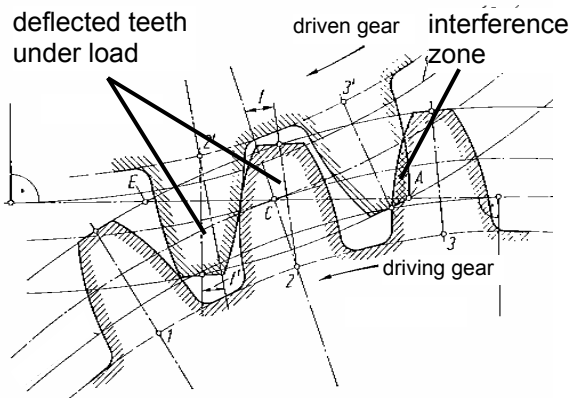


Figure 3 Tooth deflection under load and subsequent interference zone in following gear mesh, acc. to (6)

Along the path of contact, the total stiffness of all teeth in mesh will vary according to the referring bending moment arms and active face width available, **Figure 2**. For spur gears, the active flank width is constant, for helical gears it is low at beginning and end of path of contact. Therefore, the stiffness varies much more for helical gears than for spur gears.

Even when considering deviation free tooth flanks - which is almost achievable with modern manufacturing methods- due to this elastic deformation un-

der load, the flank position will change and therefore there will be interference between the gear teeth, see **Figure 3**, causing periodical noise excitation. This interference can be compensated for by appropriate flank modification, which only can be optimized for one load level and one stiffness value for the mesh stiffness. Gears requiring low noise emission at different load levels, e.g. navy gears, can not be only optimized for one single load stage. Therefore, an utmost constant total mesh stiffness, c_v , considering all teeth being in mesh at a time, should be obtained for the complete plane of contact.

Figure 4 shows the course of mesh stiffness over rotational angle for a spur and a helical gear example. 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. It is obvious, that with 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; 0 for spur gears, up to 8 for double helical gears). The decisive total contact ratio, ϵ_v , is the sum of ϵ_α and ϵ_β . Integer figures for ϵ_β further improve the mesh quality (Figure 1).

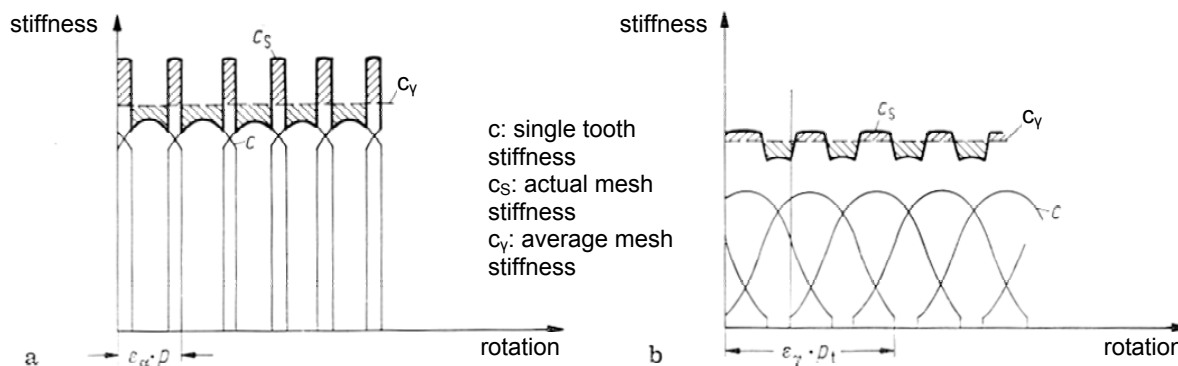


Figure 4 Mesh stiffness, a) spur gear mesh; b) helical gear mesh; acc. to (6)

Conclusion: Maximum total contact ratio, ϵ_v , as well as integer value for overlap ratio, ϵ_β , are the optimum basis for noise optimized gears.

2.2 Gear Lay Out with Marine Applications

Coming to more distinct views on these principles, without any doubt, **spur gears**, with zero helix angle, are less suitable for high performance marine gears, neither with commercial vessels nor for smooth run of gears on Navy ships. As described before, such a design provides the tooth load transferred from one tooth pair in contact to the next by a

sudden load alteration over the tooth mesh causing periodical force impacts and, subsequently, increased generated noise. Any attempt to decrease these impacts by profile modification may only slightly improve the noise behavior on one specific load level, with the basic disadvantage at remaining lack of contact ratio and still high sensitivity for misalignment and concurring tooth edge overload.

Single helical gears, Figures 5a) and 5b) comprise at a first glance some benefits against spur gears. Normally, low helices apply in practice, where tooth load impacts are diminished by helix