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New methodology to reduce the transmission error of the spiral bevel gears

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ABSTRACT

New methods and tools have been developed the last years to improve the understanding of gear meshing. Mechanical industries attach growing attention to the dynamic behavior of mechanical transmissions, including vibration and noise that result. The transmission error of the gear, which measures the intensity of one of the main causes of dynamic phenomena, can be considered as a relevant indicator of gear performance.

This paper presents a new design method of spiral bevel gears, the objective of this method being to reduce their quasi-static transmission error. The proposed approach is based on an optimization process including loaded meshing simulations. The simulation model has been evaluated using a helicopter tail gearbox as bench test. Measurement results are given, showing a good correlation with predictions.

1. Spiral bevel gear optimization for noise reduction

In accordance with the conclusions of Coy and al in their study on behalf of NASA [1], it is established that the noise generated by a helicopter gear box is principally due to the meshing of the gears. The path of the waves is shown schematically in Fig. 1. The vibrations propagate from the gearbox to the cabin through the air and through the airframe. The noise impacts the comfort for pilots and passengers. Different isolations devices can be installed on a Helicopter in order to limit the propagation of the noise. It is now even possible to imagine active noise control systems in the cabin to counteract directly the noise in the passengers environment. However all these devices have significant weight and cost. In addition, Coy [1] highlights two trends related to evolution of the Helicopter design. Thanks to growing performances of gas turbine engines, the power installed on helicopters is becoming bigger and bigger. Unfortunately the noise resulting from gearbox meshing is also increasing with the power transmitted. Also efforts done by designers to always decrease the weight to power ratio of new helicopters lead to a decrease of stiffness for many systems. This could increase the difficulty of controlling the meshing gears under load or of filtering the noise and vibrations through the structural path. Clearly, many benefits are coming if the vibrations and noise can be reduced directly at gearbox level.

According to Coy [1], the transmission error is the main source of dynamic excitation of the gear.

The transmission error is the difference between the actual angular position of the output shaft and the theoretical position it would have if the reduction ratio was constant. The transmission error may be canceled with conjugate surfaces. In practice, modifications of the relative direction of the rotation axes of fully conjugate surfaces may cause contacts on the edges of the flank. This implies a significant increase of the noise and of the contact pressure that finally will lead to the reduction of the efficiency and the life time of the gear. That is one of the reasons why the flanks of a Gleason spiral bevel gear are not conjugate surfaces. They have systematically a transmission error without load, also called kinematic transmission error. It directly affects the noise generated by the meshing. Coy [1] therefore suggests working on how to decrease its amplitude without canceling it completely.

The identification of the origin of noise is based on the results of various vibration and acoustic measurements. Fig. 2 shows the observed vibration spectrum nearby a helicopter gear box. The spiral bevel gear generates the highest peaks in that case.

2. Transmission error analysis of the spiral bevel gear

Numerous papers are dealing with the study of the transmission error, this phenomenon playing a leading role on the performance of gears.

The earlier studies focus on the unloaded transmission error. Vogel [2] determines it without generating the tooth flanks. Its mathematical model combines the equations describing the tooth flank generation with the ones modeling the contact conditions. This is a direct simulation of the meshing. Vimercati [3] uses a method based on a finite element solver. It is a time consuming solution usually reserved for loaded meshing simulations. Astoul [4] proposes a method based on a discretization of one of the two tooth flank surfaces in contact and a specific projection of the

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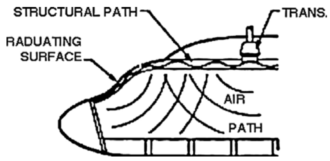

Fig. 1. Transmission gear box noise path [1].

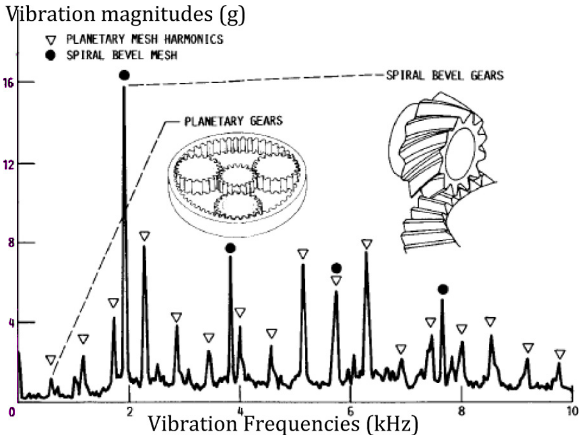

Fig. 2. Linear spectrum of helicopter gearbox vibrations [1].

points on the opposite flank. It gives a good approximation of the unloaded transmission error.

There is a growing interest in the simulation of the loaded transmission error. Gosselin [5] and Litvin [6] present well-known approaches. They model the tooth bending stiffness with the finite element method and estimate the contact deformation using classical Hertz theory. The blank rotation compatibility condition gives the load distribution on the teeth and the loaded transmission error. Kolivand [7] uses semi-analytical Rayleigh–Ritz models which take into account the tooth stiffness in a more efficient manner than finite element models. He defines the Hertzian contact through the Weber formulation. Its computing process is faster than the previous ones.

Several papers present methods to master the unloaded transmission error. Litvin [8–10] seeks to reduce the noise generated by the meshing, and also to increase the life of the gear. This involves reducing the sensitivity of the contact pattern to gear misalignments and controlling the position and shape of the meshing contact ellipse in the middle of the tooth flank. Litvin noted that the behavior of conjugate tooth profiles is extremely sensitive to the relative displacement of the axes. He therefore advises to adjust non-conjugate profiles. He suggests a second method based on the mastery of a kinematic transmission error with a parabolic shape. Litvin wishes both to stabilize the contact pattern position and to reduce the noise. He advocates for a longitudinal shape of the contact pattern. The latter is less sensitive to the relative displacement of the teeth. The contact ratio is increased and the risk of edge effects is reduced. The curve assigned to the transmission error is often of the second order. Wang [11] chooses a form of the fourth order. The latter helps to reduce the noise generated by the meshing of gears. Su [12] defines a predesigned seventh-order polynomial function of transmission error in order to reduce vibrations and noise. A reverse tooth contact analysis method gives the suitable machine-tool settings.

Works on the reduction of the loaded transmission error are quite new. Argyris [13] shows that the finite element modeling of the teeth is accurate enough to determine the loaded transmission error when the structural meshes are constructed automatically from points generated on the tooth flanks without using the support of a CAD software. Simon [14,15] analyses the influences of machine settings and tool geometry on the loaded transmission error. Such a sensitivity study gives clues about the improvement of the gear performance. Artoni [16] optimizes an ease-off modeled by a bivariate polynomial of fourth degree. The coefficients are optimization variables. The objective is the minimal loaded transmission error. Artoni [17] adds more objectives to the optimization problem and shows the influence of the polynomial degree on the solutions. Mermoz [18] presents an automatic optimization process running with loaded simulations. The finite element method is applied to a light model of the gear. The computed maximal contact pressure is minimized. The variables are the six modified roll coefficients of the CNC grinding machine-tool.

The present paper shows that the latter method also provides, in addition to the pressure reduction, a significant contribution in reducing the transmission error. A first part introduces the model worked out to simulate the gear meshing. A bench test is used to measure the loaded transmission error. The correlation between results and measurements is then checked. A second part shows that the new optimization method allows a reduction of the loaded transmission error.

3. Simulation of the transmission error

3.1. Finite element model of the gear

The contact ratio of the gear is assumed to be less than five. It means that no contact can exist simultaneously between more than five pairs of teeth. Only five teeth of the pinion and the gear are modeled with a specific mesh. On each part, rigid bodies link the rim to the pitch apex. The meshing positions are imposed on the pitch apex of the pinion with a constant rotation angle. The torque is applied on the pitch apex of the gear. Fig. 3 shows the model.

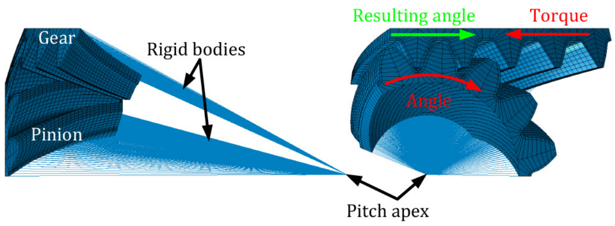

Fig. 3. Simulation model of loaded meshing.

The first unloaded angular position of the gear is computed according to the method described in Ref. [4]. As the CAD software meshers generate interpolation errors, they are not used here. The meshes of the parts are directly built in order to model accurately the tooth flank surfaces. The loaded meshing problem is solved with the SAMCEF MECANO solver using non-linear finite element analysis. The simulation reproduces a quasi-static behavior without dynamic effects. The variation of the resulting rotation angle of the gear is called the quasi-static transmission error. In the literature, it is also defined as the loaded transmission error.

3.2. Bench test and protocol

Simulations are run to compare the results to available loaded transmission error measurements. Icard [19] details the protocol and the bench test. The latter is shown in Fig. 4.

The tests were performed in Airbus Helicopters with a tail gear box. The minimal number of moving parts explains the choice of such a device. The spiral bevel gear and the bearings are the only mechanical components moving. The measurement interference sources are minimized. The pinion position is measured with a magnetic encoder. An optic encoder is installed on the gear. The two encoders are synchronized in order to post-process the gear rotation in relation to the pinion rotation.

The conditions of the test have to comply with the assumptions of the model. The dynamic effects are not simulated. Therefore, the gear runs at low speed to fulfill the quasi-static condition. The displacements of the gear axes are assumed to be zero. The load levels are not high to avoid significant distortions of the gear box components.