

Computerized Design and Generation of Low-Noise Gears With Localized Bearing Contact

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

ABSTRACT:

The paper represents the results of accomplished research projects directed at reduction of noise caused by misalignment of the following gear drives: double-circular arc helical gears, modified involute helical gears, face-milled spiral bevel gears and face-milled formate cut hypoid gears. It is proven by the authors that due to misalignment, periodic almost linear discontinuous functions of transmission errors occur. The period of such functions is the cycle of meshing when one pair of teeth is changed for another one. Due to the discontinuity of such functions of transmission errors high vibration and noise are inevitable. The authors propose to predesign a parabolic function of transmission errors that is able to: (i) absorb linear discontinuous functions of transmission errors, and (ii) turn out the resulting function of transmission errors into a continuous one. The proposed idea was successfully tested by the Gleason Works and NASA Lewis Research Center for a set of spiral bevel gears manufactured by the Bell Helicopter Co. It was found out that the noise was reduced at 12–18 decibels in comparison with the existing design. The idea of the predesign of a parabolic function is applied for the reduction of noise of helical and hypoid gears. The effectiveness of the proposed approach has been investigated by developed TCA (Tooth Contact Analysis) programs. The bearing contact for the mentioned above gears is localized. Conditions that allow to avoid edge contact for the gear drives have been determined. Manufacturing of helical gears with new topology by hobs and grinding worms has been investigated.

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COMPUTERIZED DESIGN AND GENERATION OF LOW-NOISE
GEARS WITH LOCALIZED BEARING CONTACT

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1. Introduction :

Vibration and noise of gears are caused due to errors of gear alignment, and deflection of teeth and shafts under the load. The main attention in this paper is paid to the reduction of transmission errors caused by misalignment, and the stabilization of the bearing contact.

Gear misalignment and errors of manufacturing cause transmission errors that act as an oscillator of vibrations. Fig. 1(a) shows the transmission function $\phi_2(\phi_1)$ for a misaligned gear drive. This function deviates from the theoretical transmission function $\phi_2^{(o)}(\phi_1)$ that is a linear one. The function of transmission errors $\Delta\phi_2(\phi_1)$ is represented as (fig. 1(b)).

$$\Delta\phi_2(\phi_1) = \phi_2(\phi_1) - \phi_2^{(o)}(\phi_1) = \phi_2(\phi_1) - \frac{N_1}{N_2}\phi_1 \quad (1.1)$$

where N_i ($i = 1, 2$) is the pinion 1 (gear 2) tooth number. Function $\Delta\phi_2(\phi_1)$ of transmission errors is a piecewise almost linear function with the frequency of the cycle of meshing determined by $\phi_1 = \frac{2\pi}{N_1}$.

The shape of existing function of transmission errors shown in fig. 1(b) is very unfavorable because the transfer of meshing from one pair of teeth to the neighboring one is accompanied with a stroke of the teeth, and vibration and noise are inevitable. Since errors of misalignment cannot be avoided, we have to expect that the transformation of motions by a gear mechanism will be accompanied by transmission errors. However, although transmission errors cannot be avoided, there is a possibility to change the shape of the function of transmission errors for a more favorable one. This goal can be achieved by absorption of a linear function of transmission errors caused by misalignment by a predesigned parabolic function of transmission errors as shown in fig. 2. Fig. 2(a) illustrates the interaction of two functions: (i) the linear function $\Delta\phi_2^{(1)} = b\phi_1$ caused by misalignment, and (ii) the predesigned parabolic function $\Delta\phi_2^{(2)} = -a\phi_1^2$. Function $\Delta\phi_2^{(2)}(\phi_1)$ is provided for the absorption of linear function $\Delta\phi_2^{(1)}(\phi_1)$. The resulting function $\Delta\phi_2(\phi_1)$ of transmission errors is represented as follows

$$\Delta\phi_2(\phi_1) = \Delta\phi_2^{(1)}(\phi_1) + \Delta\phi_2^{(2)}(\phi_1) = b\phi_1 - a\phi_1^2 \quad (1.2)$$

It is easy to verify that equation (1.2) represents in new coordinate system $(\Delta\psi_2, \psi_1)$ a parabolic function with the same parabola parameter a . The parabolic function (fig. 2(b)) is designated as

$$\Delta\psi_2 = -a\psi_1^2 \quad (1.3)$$

Axes of coordinate system $(\Delta\psi_2, \psi_1)$ and $(\Delta\phi_2, \phi_1)$ are parallel but the origins are different. The coordinate transformation from $(\Delta\phi_2, \phi_1)$ to $(\Delta\psi_2, \psi_1)$ is represented with the following equations

$$\Delta\psi_2 = \Delta\phi_2 - \frac{b^2}{4a}, \quad \psi_1 = \phi_1 - \frac{b}{2a} \quad (1.4)$$

The difference between functions $\Delta\phi_2(\phi_1)$ and $\Delta\psi_2(\psi_1)$ in addition to the change of location of the origin of coordinate system is the difference of location of couples of points (A,B) and the respective points (A^*, B^*) (fig. 2(a)). The symmetrical location of (A,B) is turned into asymmetrical location of (A^*, B^*) . However, the interaction of several functions $\Delta\psi_2(\psi_1)$ determined for several tooth surfaces being in mesh may provide a symmetrical parabolic function of transmission errors as shown in fig. 2(b). This can be achieved if the parabolic function $\Delta\phi_2(\phi_1)$ will be predesigned in the area (fig. 2(a))

$$\phi_1(B) - \phi_1(A) \geq \frac{2\pi}{N_1} + 2c \quad (1.5)$$

where $c = \frac{b}{2a}$. Requirement (1.5), if observed, enables to provide a continuous function $\Delta\psi_2(\psi_1)$ for the range of $\frac{2\pi}{N_1}$.

The idea of a predesigned parabolic function for absorption of a linear function of transmission errors caused by gear misalignment have been applied by the authors of the paper for various types of gears (see sections 2, 3, 4, 5, and 6). However, the application of a predesigned parabolic function of transmission errors needs the modification of gear tooth surfaces.

The other ideas discussed in this paper, the localization of the bearing contact and its stabilization, are based on the following principals: (i) The instantaneous line contact of gear tooth surfaces is substituted by the instantaneous point contact; (ii) Due to elastic deformation, the real contact of gear tooth surfaces is spread over an elliptical area, and the bearing contact is formed as the set of instantaneous contact ellipses; (iii) The path of contact is chosen in the longitudinal direction, and the center of symmetry of the contact ellipse moves as well in the longitudinal direction; this makes the bearing contact more stable and, probably, favors the conditions of lubrication.

The discussed above ideas are applied for involute helical gears, double circular-arc helical gears, spiral bevel gears, and hypoid gears (see below).

2. Involute Helical Gears with Modified Topology :

Preliminary Considerations. Ideal involute helical gears are in line contact at every instant. The line of contact is the line of tangency to the helix on the base cylinder (fig. 3). The ideal gears perform rotation with constant gear ratio represented as

$$m_{12} = \frac{\omega^{(1)}}{\omega^{(2)}} = \frac{r_2}{r_1} = \frac{r_{b2}}{r_{b1}} = \frac{N_2}{N_1} \quad (2.1)$$

where $\omega^{(i)}$ ($i = 1, 2$) is the pinion (gear) angular velocity; r_i is the radius of the operating pitch circle for nonstandard gears; $r_i = r_{pi}$, where r_{pi} is the pitch circle, for standard gears; r_{bi} is the radius of the base circle; N_i is the tooth number. The gear axodes are cylinders of radii r_i , and the line of tangency of these cylinders is the instantaneous axis of rotation. The cross-section of the gear tooth surface is an involute profile (fig. 4(a)). The intersection of the tooth surface by a cylinder of radius ρ is a helix (fig. 4(b)). The helix is turned out into a straight line when the cylinder is developed onto a plane (fig. 4(c)). The lead H represents the axial displacement for one revolution when a point moves along the helix. The lead H of a helical gear is the same for a cylinder of any radius ρ , and can be expressed as follows (fig. 4(c))

$$H = 2\pi\rho_i \tan \lambda_{pi} \quad (2.2)$$

The screw parameter p represents the axial displacement along the cylinder for the angle of one radian, and is represented as

$$p = \frac{H}{2\pi} = \rho_i \tan \lambda_{pi} \quad (2.3)$$

Considering two mating helical gears and using equations (2.1) to (2.3), we obtain that the lead angles of the gears are equal only for helices on the gear cylinders r_1 and r_2 , and r_{b1} and r_{b2} . However, the mating gears for the case of external meshing have opposite directions of the helices. Equations (2.1) to (2.3) yield that

$$\frac{H_2}{H_1} = \frac{N_2}{N_1} = m_{12} \quad (2.4)$$

Existing Methods of Crowning. The ideal helical gears are very sensitive to the angular errors of alignment such as crossing or intersecting of gear axes (instead of to be parallel), errors of lead angles (caused as error of leads), and the profile angle of the tool. Such errors may cause an edge contact, and therefore the manufacturers used various methods of crowning of gear tooth surfaces to obtain a favorable localized bearing contact. However, it was not observed that the existing methods of crowning provide the undesirable shape of the function of transmission errors shown in fig. 1(b), and vibration and noise of the gears are inevitable. We will illustrate this statement with two particular cases of crowning.

Case 1: Crowning by modification of involute profiles

Let us assume that the leads H_1 and H_2 are observed as the same

as for ideal involute helical gears, and they are related with equation (2.4). The crowning is based on the modification of the involute profiles as shown in fig. 5. The point of tangency M of modified profiles is considered in the cross-section of the gear tooth surfaces. Point M coincides with a current point of tangency of the ideal involute profiles. The normal at point M to the theoretical and modified profiles is the same and passes through the instantaneous center of rotation I (fig. 5). However, there is a big difference between the conditions of meshing of ideal involute helical gears and the gears provided with modified tooth surfaces, that can be described as follows:

(i) The ideal involute helical gears are in *line* tangency at every instant. The modified involute helical gears are in *point* tangency at every instant.

(ii) The instantaneous point of tangency of modified gear tooth surfaces is point M in the respective cross-section. In the process of meshing point M moves along a straight line that passes through M and is parallel to gear axes.

(iii) The bearing contact is localized, and the path of contact is a helix on the cylinder of radius O_1M for the pinion, and radius O_2M for the gear.

(iv) The described method for crowning allows to avoid the edge contact of the gears. However, the function of transmission errors for the crowned gears has the unfavorable shape (fig. 1(b)), and vibration and noise are inevitable. This is the reason why this method of crowning cannot be recommended for application.

Case 2: Crowning by modification of leads

We will consider the following sub-cases:

(i) The profile modification, as described in case 1, is provided and in addition the leads are modified. The modified leads H_1^* and H_2^* satisfy equation (2.4). In this sub-case the main conditions of meshing are the same as in case 1, but there was not a need in modification of the leads.

(ii) The profile modification, as described in case 1, is provided but the modified leads H_1^* and H_2^* do not satisfy equation (2.4). In this sub-case the bearing contact is localized, the path of contact on the gear tooth surface is a helix, the line of action is a straight line that is parallel to the gear axis, but the gear ratio, although it is constant, differs from the theoretical one. The gear ratio is determined as

$$m_{12}^* = \frac{H_2^*}{H_1^*} \neq \frac{N_2}{N_1} \quad (2.5)$$

Since the real gear ratio m_{12}^* differs from the theoretical one determined as $m_{12} = \frac{N_2}{N_1}$, the function of transmission errors has the shape shown in fig. 1(b), and the transformation of motion is accompanied by a stroke at each cycle of meshing.

(iii) The theoretical involute profiles are not modified, but the leads are changed, and $\frac{H_2^*}{H_1^*} \neq \frac{N_2}{N_1}$. The contact of gear tooth surfaces is localized, and the path of contact on the gear tooth surface has the shape shown in fig. 6. The disadvantage of such method of crowning is that the contact of gear tooth surface (due to small modification of leads) is close to the line contact, the gears are very sensitive to angular errors of misalignment, there is a possibility of an edge contact, and the function transmission errors has the unfavorable shape shown in fig. 1(b).

New Approach for Modification of Gear Tooth Surfaces. In the authors' opinion the existing methods of crowning do not satisfy the requirements of the design and manufacture of low-noise helical gears.