On a Possible Way of Size and Weight Reduction of a Car Transmission

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This study is focused on some features of the geometry and kinematics of gear hobbing operations. The principal goal is to determine the minimal hob idle distance that is required for complete generation of the gear teeth. This task is important in many aspects, especially for reducing the axial size of a hobbed shoulder-gear. Reduction of the axial size of a hobbed shoulder-gear leads to reduction of size and weight of the shoulder-gear itself and of the gear train housing. Developed by E. Buckingham, methods of analytical mechanics of gears are applied to determine an exact minimal allowed length of the gear hob idle distance. The results reported in the paper are applicable for manufacturing of spur and helical involute gears. Applying the results allows one to reduce the axial size and weight of a gear train and its housing. The consideration below is focused on hobbing of involute gears. Slightly modified results obtained are applicable for hobbing of splines, sprockets, ratchets and other form tooth profiles. The results obtained are important for the application of multi-start hobs of small diameter.

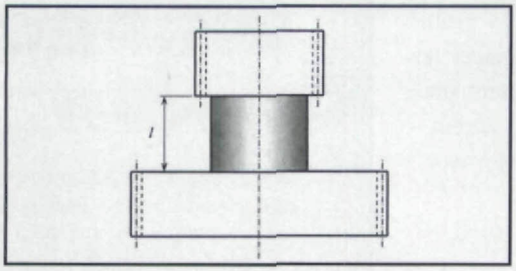

Figure 1—An example of a cluster gear.

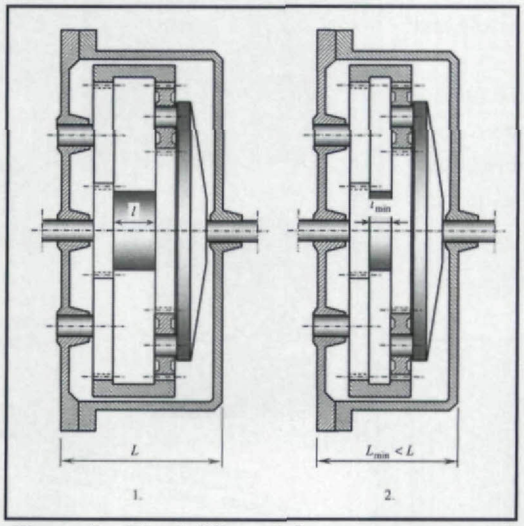

Figure 2—Fixed differential gear arrangement of Type D.

Introduction

Almost any external tooth form that is uniformly spaced around a center can be hobbed. Hobbing is recognized as an economical means of producing spur and helical gears with involute tooth profiles. Although the hobbing process is often associated with manufacturing of gears and splines, many other forms can be cut. Due to its versatility, hobbing is also recognized as an economic means of producing ratchets, sprockets and other special forms.

But a problem appears in mass production of cluster gears (Fig. 1), for example, in manufacturing transmissions in the automotive industry. For economic reasons, both gears in a cluster gear need to be hobbed and not machined with a gear shaper cutter. Productivity of gear hobbing operations significantly exceeds that of gear shaping operations. Moreover, hobbing of a cluster gear can be performed on the same hobbing machine and in the same set-up without work resetting. Therefore, hobbing of cluster gears does not waste time: a) for unloading the gear hobbing machine, b) for transporting gear blanks to a gear shaping machine, c) for installing gear blanks again on the gear shaper, and d) for solving a specific problem of gear facing. This last issue is important for manufacturing a multi-flow gear train, for example, in manufacturing planetary reducers, where torque to be transmitted is splitting on two or more flows.

However, the principal disadvantage of gear hobbing operations is that usually only cluster gears with long enough necks can be hobbed. Increasing length l of a neck increases a cluster gear's axial size, and thus increases the weight of the gear cluster itself. It also increases the weight of gear shafts and gear housings. In the case of $l=l_{min}$, size L could be significantly reduced to L_{min} (Fig. 2). This size reduction corresponds to a reduction of size and weight in a car transmission. The example above (Fig. 2.1) is taken from References 3, 5, and 11. The example discussed above clearly indicates the necessity of developing methods of hobbing cluster gears with the shortest neck, i.e., cluster gears with $l=l_{min}$.

In the paper, notation of parameters recommended by ANSI/AGMA (Ref. 1) is used.

Literary Survey

A few directions of hob feed are recognized. Downward feed direction (in conventional gear hobbing) and upward feed direction (in climb gear hobbing) are mostly used.

The reduction of total hob travel distance leads to a reduction of total gear hobbing time and is widely recognized. The parameters of a gear to be machined are given and a gear manufacturer cannot change them. Gear manufacturers cannot reduce gear face width. A total hob travel distance can be reduced if one gets to know the shortest allowed length of approach distance and the shortest allowed hob idle distance. In this paper, the shortest allowed hob idle distance is considered.

A hob rotates about its axis (Fig. 3), with a constant angular velocity ω_h. Simultaneously, the hob moves along the work axis in the feed direction (or in the opposite direction in a conventional gear hobbing operation).

Cutter total travel distance AD (Fig. 3) is equal to the sum of the hob approach distance AB, face width BC, and hob idle distance (or hob overrun) CD.

The hob approach is the distance that the hob has to travel parallel to the gear axis, from the point of initial contact between the hob and the point where center distance reaches the first gear face (at the point B). The hob approach distance can be computed by knowing the appropriate formula, which is available in Reference 4.

The gear face width is indicated on the part print. Its length cannot be changed by gear manufacturers.

The hob idle distance, a distance required to complete generating of the entire set of gear teeth, is the linear hob carriage travel beyond the second gear face C.

There is no complete understanding of what the hob idle distance exactly is and how to compute it minimally, allowing length for hobbing a given cluster gear. For example, in accordance with recommendations given by hob manufacturer FETTE GmbH: "No idle distance, except for a safety allowance, is required for straight teeth" (Ref. 4).

Impact of the Hob Idle Distance on Axial Size of a Hobbed Cluster Gear

The length of the hob idle distance is not equal to the length of the hob approach distance (Fig. 4).

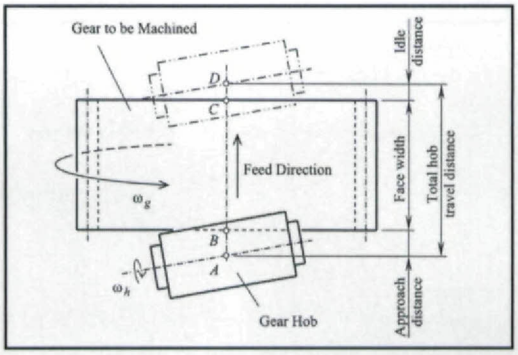

Figure 3—Kinematics of climb hobbing of an involute gear.

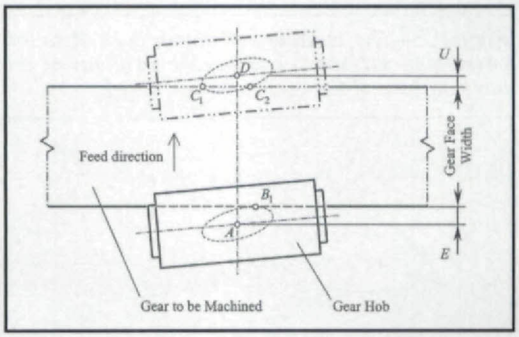

Figure 4—Comparison of length of the hob approach distance E and of the hob idle distance U.

Length of the hob approach distance is equal to the height of the highest point, B_1, on the penetration curve above the horizontal plane through the center distance—through the intersection of the cutter and gear axes (Fig. 4). When the hob is away from the upper face of the gear, the penetration curve intersects the upper face at points C_1 and C_2. This makes it evident that the length of the hob idle distance U is always shorter than the length of the hob approach distance E. In order to reduce axial size, hobbing cluster gears usually starts from the open end, where there is enough room for hob approach distance, and finishes at the opposite end, where there is a lack of space for the hob.

The hob idle distance U affects the axial size of a hobbed cluster gear and a hobbed shoulder-gear as well. When a longer idle distance U is required, a longer cluster gear neck l_{min} is necessary, and the axial size of the cluster gear becomes larger and vice versa (Fig. 4). As is evident in Figure 4, the minimal length of a cluster gear neck l_{min} depends on the hob idle distance U, i.e. $l_{min} = l_{min}(U)$.

To cut gear hobbing time and minimize the axial length of a cluster gear, the hob idle distance must be as short as possible. At the same time, the length of the hob idle distance must be sufficient for completing generation of the entire gear tooth surface. As mentioned above, in gear hobbing

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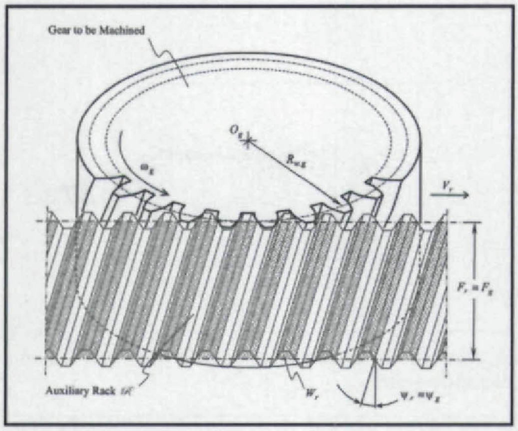

Figure 5—The auxiliary phantom rack R as an enveloping surface to consecutive positions of the tooth surface of the gear to be machined.

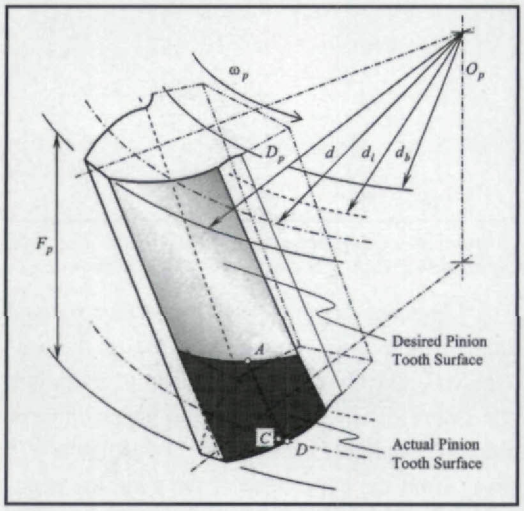

Figure 6—Deviation of an actual gear tooth surface from the desired gear tooth surface caused by lack of the hob idle distance.

operations, the gear blank is rotating around its axis O_g with a constant angular velocity ω_g. The auxiliary phantom rack R is traveling with a constant speed V_r. The velocities ω_g and V_r are synchronized in the manner that the auxiliary rack R pitch surface W_r is rolling without sliding over the gear pitch cylinder of radius $R_{w.g}$. For machining the entire length of the gear, it is required that the gear hob generates the auxiliary rack R of width F_r that is equal to or exceeds width F_g of the gear to be machined (Fig. 5).

A lack of length of the hob idle distance ΔU causes deviation of the actual tooth surface from its desirable shape (Fig. 6). The length of ΔU depends only on the relative location of the gear hob in the axial direction of the gear to be machined. It does not depend on hob diameter, while maximal deviation δ^* of the actual gear tooth surface from the nominal gear tooth surface does.

The initial solution to the problem is enhanced and developed here. A numerical example is also provided.

Impact of a Gear Hob-Setting Angle on the Hob Idle Distance

It is convenient to evaluate the impact of a gear hob-setting angle on the hob idle distance by considering a simple degenerated case, like when the hob-setting angle ζ_h is equal to zero ($\zeta_h = 0°$). In machining a spur gear, this occurs when $\zeta_h = 0°$, the crossed-axes angle Σ, i.e. the angle that makes axes of the gear O_g and of the hob O_h, is equal to a right angle ($\Sigma = 90°$).

For climb hobbing (Fig. 7.1), the hob idle distance U is measured from the upper face of the gear to be machined. Generating the gear tooth profile with the hob begins the instant that straight generatrix $E_1^{(r)}$ of the recessing flank of the hob tooth profile in its horizontal position reaches the lower face of the work (Fig. 7.2). When the hob travel distance exceeds the length $l_1 = d_{b.h}$ (here $d_{b.h}$ is the hob base diameter), the opposite flank of the hob tooth profile enters the gear profile. In this case, both flanks of the gear tooth are generated simultaneously. Generating the gear tooth profile accomplishes this in an instant, when the straight generatrix $E_1^{(a)}$ of the opposite flank (of the approaching flank) of the hob tooth in its horizontal position reaches the upper face of the work (Fig. 7.2). In the case under consideration, the length of the hob idle distance U is equal to half a base diameter of the hob, i.e $U = 0.5d_{b.h}$. This is the shortest allowed hob idle distance. To accomplish the generation of the gear tooth profile, it is required that the cutter travel distance is equal to or exceeds $L \geq F_g + d_{b.h}$. It is important to point out here that the shortest allowed hob idle distance could not be less than $0.5d_{b.h}$.

The above approach (Fig. 7.2) can be utilized when the gear hob-setting angle is not equal to zero and this angle is positive. In this case, the straight generatrices $E_2^{(a)}$ and $E_2^{(r)}$ (Fig. 7.3) are similar to the straight generatrices $E_1^{(a)}$ and $E_1^{(r)}$ in Figure 7.2, additionally shifted on the distance δ relative to each other along the gear axis. The value of the parameter δ depends on the length of the gear tooth profile-generating zone and on the gear hob-setting angle. This question is explained in more detail below. Distance δ increases the length of the hob idle distance U and the total hob travel distance, which is necessary for accomplishing the generation of the gear tooth profile.

It is evident that, in general, Figure 7.3 provides us only with qualitative results. It cannot be applied for computing the length of the hob idle distance in quantities.

When reducing the axial length of a cluster gear or hobbing an involute gear with any given value of pitch helix angle ψ_g it is preferable to design something that enables a crossed-axes angle $\Sigma = 90°$ with a gear hob. To continue the analysis, it is required to clearly understand the gear hob-setting angle ζ_h.

While generating the machining surface of a gear hob, the auxiliary rack *R* performs a screw motion with the hob axis as the axis of the screw motion. First of all, the axis of rotation of a cylindrical hob is parallel to the auxiliary rack pitch surface W_r. Secondly, orientation of the hob axis of rotation may vary relative to the pitch plane W_r (Fig. 8).

The idea of the gear hob-setting angle ζ_h can be traced back to a publication by E. Buckingham (Ref. 2).

The auxiliary rack tooth makes an angle ψ_r with the gear axis O_g. Very often, this angle is called the auxiliary rack pitch helix angle (Ref. 1). The gear hob axis of rotation intersects the center distance *C*. In Figure 8, the center distance is depicted as a point C of intersection of the center distance and of the auxiliary rack pitch surface W_r.

The straight line $^{\circ}O_h$ (parallel to the plane surface W_r) through the point C is orthogonal to the auxiliary rack tooth. For a spur gear, it is parallel to the gear face. The gear hob axis of rotation makes a certain angle with the straight line $^{\circ}O_h$. The gear hob axis of rotation $^{+}O_h$ makes a positive angle $+\zeta_h > 0°$ with the straight line $^{\circ}O_h$. The gear hob axis $^{-}O_h$ makes a negative angle $-\zeta_h < 0°$ with the same straight line $^{\circ}O_h$.

We suggest naming the angle ζ_h the gear hob-setting angle (Ref. 7). It is necessary to stress here that the gear hob-setting angle ζ_h is a parameter of a hob design. Its value does not have to equal the value of the gear hob lead angle $\lambda_h = 90° - \psi_h$. However, due to the restriction explained below, the difference between the angles ζ_h and λ_h cannot be significant. The equality $\zeta_h = \lambda_h$ may or may not take place. For example (Fig. 9), the same spur gear with a given parameter of its design can be machined with a few gear hobs of various hob-setting angles ζ_h, but with the same hand of hob lead angles.

Usually, the gear hob-setting angle is positive ($\zeta_h > 0°$) and is approximately equal to the hob lead angle λ_h. In this case, crossed-axis angle $\Sigma = 90° - \zeta_h$ is less than 90° (Fig. 9.1).

The method of spur gear hobbing with right

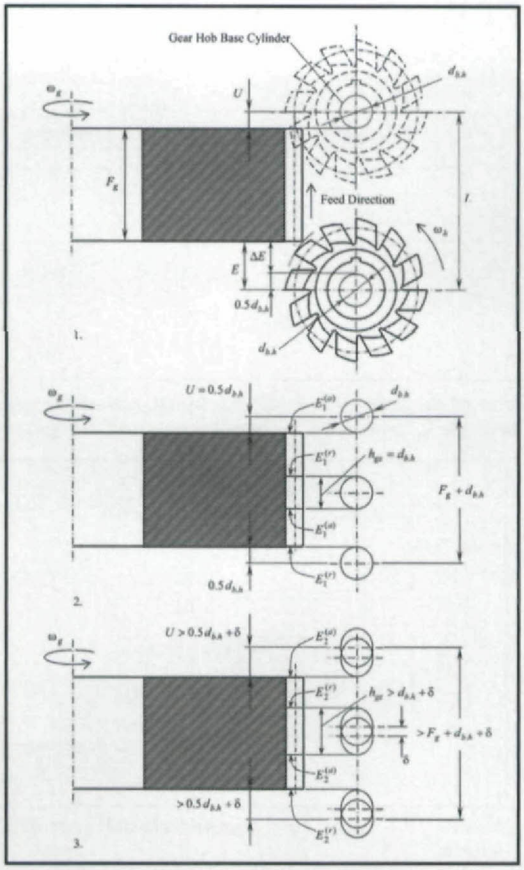

Figure 7—The impact of gear hob-setting angle on the length of the hob idle distance in climb hobbing of involute gears.

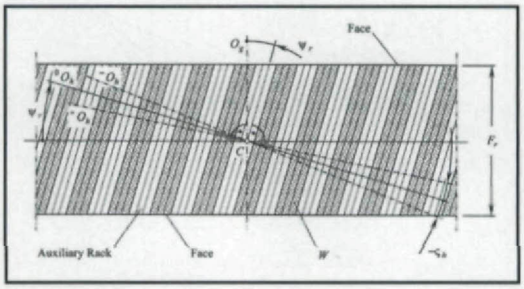

Figure 8—Front view on the auxiliary phantom rack* R *of an involute hob (see Fig. 5).

crossed-axis angle $\Sigma = 90°$ is known. In this case, the gear hob-setting angle is equal to zero (Fig. 9.2). The same method can be applied to hobbing a helical gear. In the last case, the value of the crossed-axis angle can be computed by the formula $\Sigma = 90° - \psi_g$.

Finally, an involute gear can be machined with a hob with negative hob-setting angle: $\zeta_h < 0°$ (Fig. 9.3). In this case, crossed-axis angle $\Sigma = 90° + \zeta_h$ exceeds 90°. It is important to stress that such an opportunity for gear hobbing operations exists mostly theoretically. The top cutting edge of a gear hob tooth with negative hob-setting angle ζ_h becomes too short or has negative length. For this reason, the gear hob with $\zeta_h < 0°$ does not have a wide application in contemporary manufacturing

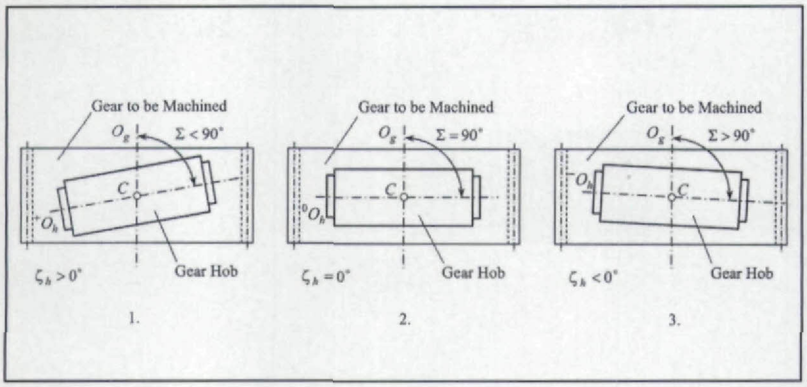

Figure 9—Relative orientation of the spur gear to be machined and of the gear hob with the gear hob-setting angle ζ_h 1-positive, 2-equal to zero, and 3-negative.

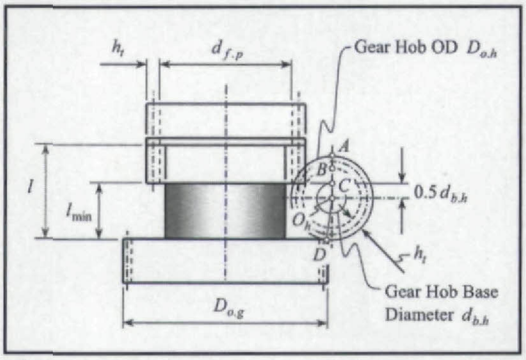

Figure 10—Computing minimal cluster gear axial length.

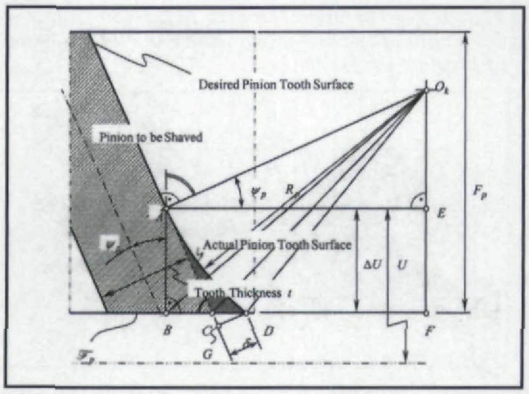

Figure 11—Unfolded section of the gear in Figure 6 by the pitch cylinder of diameter d that is co-axial to the gear. Letters "a", "c" and "d" in Figure 6 correspond to the same points in Figure 11.

of gears.

For machining a spur gear, it is required to maintain the gear hob-setting angle $\zeta_h = 0°$ (Fig. 9). In this case, the gear hob helix angle becomes very small, ψ_h approaching 0°. However, the angle ψ_h remains positive ($\psi_h > 0°$).

For machining a helical gear with a certain helix angle ψ_g, it is required to maintain the gear hob-setting angle $\zeta_h = -\psi_g$. In both cases, this yields the cross-axis angle $\Sigma = 90°$. The difference $(\psi_h - \psi_g)$ is equal to 0°. The hand of the gear hob start is opposite to the hand of the gear start.

Figure 9 clearly illustrates that the hob lead angle λ_h, and the gear hob-setting angle ζ_h have to be considered as two completely different parameters of the design of the gear hob. In some cases, the parameters of a gear hob design, λ_h, and ζ_h can be equal (or almost equal) to each other. In general, this requirement is not mandatory. In the example above (Fig. 9) the hand of the gear hob lead angle λ_h remains the same for all three cases, and for all of these cases it is positive ($\lambda_h > 0°$). However, the value of the lead angle λ_h for each of the three cases does not remain the same. The hob-setting angle ζ_h also can vary. It can be positive (Fig. 9.1), equal to zero (Fig. 9.2), or even negative (Fig. 9.3) as well. Under such a variation, the value of the lead angle λ_h also varies, but its hand remains the same. The sign of the lead angle λ_h does not change.

Determining Parameters of a Gear Hob for Machining the Shortest Cluster Gear

Gear hob outside diameter is the most critical issue in machining a cluster gear with the hob when the crossed-axes angle $\Sigma = 90°$. Usually, a gear hob outside diameter $D_{o.h}$ can be expressed as: $D_{o.h} = 2(AB + BC + CO_h)$ (Fig. 10). This formula yields an expression for $D_{o.h}$ which is explained further in Equation 1 in the box on page 49.

In Equation 1, whole tooth depth is designated as h_t (Ref. 1).

Aiming for a reduction of size and weight of a car transmission, the portion BC of the hob outside diameter has to be eliminated. It is required that $BC \equiv 0$. Under such an assumption, Equation 1 reduces to a formula shown as Equation 2 in the box.

Hob base diameter $d_{b.h}$ can be computed from the formula (Refs. 7, 9 and 10) shown in Equation 3 in the box; note that:

m is the gear hob modulus;
Z_h is the number of the gear hob starts;
ϕ_n is the gear hob normal pressure angle.

Further reduction of the hob outside diameter $D_{o.h}$ is restricted with the necessity of satisfying the first condition of part surface generation (Ref. 8). In cases, $D_{o.h} < 2h_t + d_{b.h}$, first necessary condition of proper part surface generation is not satisfied, and thus, the gear tooth cannot be machined in accordance to the gear drawing.

Figure 10 reveals that for the cases of right crossed-axis angle $\Sigma = 90°$, length l of the cluster gear neck can be calculated from the formula in Equation 4.

In Equation 4:

$D_{o.g}$ is the outside diameter of the cluster gear's larger gear, i.e. of the shoulder (see Fig. 10);
d_f is the form diameter of the cluster gear's

smaller gear, i.e. of the pinion (see Fig. 10).

In all cases when $d_{f.p} + D_{o.h} \le D_{o.g}$, length l of the cluster gear neck is equal to $l = 0.5(d_{b.h} + D_{o.h})$.

The results yield the resultant formula for l, shown in Equation 5.

To ensure right crossed-axes angle $\Sigma = 90°$, the gear hob-setting angle is assigned equal to the gear helix angle, but with the opposite sign, i.e. $\zeta_h = -\psi_g$. For computing the minimal length l_{min} of the gear cluster, it is required to resolve the set of two equations with two unknowns:

$$\frac{\partial l}{\partial Z_h} = 0 \text{ and } \frac{\partial l}{\partial \phi_n} = 0$$

Equation 5 yields both of these equations. The number of gear hob starts Z_h, calculated from the above set of two equations, usually is a fraction. It is required to approximate it to the nearest integer number, and afterwards to recalculate the final value of the corresponding hob normal pressure angle ϕ_n. In cases where the calculated value of Z_h is close to mid-interval of two consequent integer numbers, it is required to recalculate the value of the corresponding hob pressure angle for both cases. Afterwards, it is necessary to select that value which ensures a longer gear hob top cutting edge.

Another way for computing l_{min} could be developed. The number of starts Z_h of a gear hob could be assigned several consequent integer values: $Z_h = 1$, $Z_h = 2$, and so on. Furthermore, it is required to resolve the equation

$$\frac{\partial l}{\partial \phi_n} = 0$$

for each integer value of Z_h. Finally, it is required to select the smallest possible Z_h. It is also strongly preferred that the difference between the gear hob normal pressure angle ϕ_n and the hob normal pressure angle would be the smallest possible. This results in the longest top cutting edge of the hob.

Impact of Tolerance on Length of the Hob Idle Distance

Figure 11 shows the unfolded section of the gear (Fig. 6) by the pitch cylinder of diameter d, which is co-axial to the gear. Points A, C, and D in Figure 6 correspond to the same points A, C, and D as in Figure 11.

In order to complete the generation of a perfect gear tooth in its longitudinal direction, it is required that the gear hob overrun would be equal to or would exceed the hob idle distance U.

EQUATIONS

$$D_{o.h} = 2h_t + 2BC + d_{b.h} \quad (1)$$

$$D_{o.h} = 2h_t + d_{b.h} \quad (2)$$

$$d_{b.h} = \frac{m \cdot Z_h \cdot \cos\phi_n}{\sqrt{1-\cos^2\phi_n \cdot \cos^2\zeta_h}} \quad (3)$$

$$l = 0.5\sqrt{2 \cdot D_{o.h} \cdot (D_{o.g} - d_{f.p}) - (D_{o.g} - d_{f.p})^2} \quad (4)$$

$$l = \sqrt{(D_{o.g} - d_{f.p}) \cdot \left(h_t + \frac{m \cdot Z_h \cdot \cos\phi_n}{2 \cdot \sqrt{1-\cos^2\phi_n \cdot \cos^2\zeta_h}} - D_{o.g} + d_{f.p} \right)} \quad (5)$$

$$\Delta U(\delta) = \frac{-[\delta] \cdot \cos\phi_n \cdot \tan\psi_p + \sqrt{[\delta]^2 \cdot (\cos\phi_n)^2 \cdot (\tan\psi_p)^2 - \left[[\delta]^2 \cdot \left(\frac{\cos\phi_n}{\cos\psi_n}\right)^2 - 2 \cdot R_p \cdot [\delta] \right]}}{\cos\psi_p} \quad (6)$$

$$\Delta U(\delta) \cong \frac{\sqrt{2 \cdot R_p \cdot [\delta]}}{\cos\psi_p} \quad (7)$$

$$R_{1.h} = \frac{\sqrt{d^2 - d_{b.h}^2}}{2\sin\psi_h \sin\phi_n} \cot\lambda_{b.h} = \frac{\sqrt{d^2 - d_{b.h}^2}}{2\sin\psi_h \sin\phi_n} \tan\psi_{b.h} \quad (8)$$

$$\tan\psi_{b.h} = \frac{\sqrt{\sin^2\phi_n + \tan^2\zeta_h}}{\cos\phi_n} \quad (9)$$

$$d_{b.h} = \frac{3 \cdot 5 \cdot \cos 14.5°}{\sqrt{1-\cos^2 14.5° \cos^2 12.4°}} = 44.623\ mm \quad (10)$$

$$\tan\psi_{b.h} = \frac{\sqrt{\sin^2 14.5° + \tan^2 12.4°}}{\cos 14.5°} = 0.344 \qquad \psi_{b.h} = \tan^{-1}(0.344) = 18.992° \quad (11)$$

$$\psi_h = \tan^{-1}\left(\frac{d}{d_{b.h}} \cdot \tan\psi_{b.h}\right) = \tan^{-1}\left(\frac{70}{44.623} \cdot 0.344\right) = 28.365° \quad (12)$$

$$R_p \equiv R_{1.h} = \frac{\sqrt{70^2 - 44.623^2}}{2\sin 28.365° \sin 14.5°} 0.344 = 78.025\ mm \quad (13)$$

The hob overrun must not be less than the hob idle distance U. In cases where the actual hob idle distance is shorter than or equal to U, the actual longitudinal profile AD of the gear tooth deviates from its desirable longitudinal profile AC (Fig. 11). Maximal deviation occurs at the point D, at which deviation is equal to δ. It is required that the maximal deviation δ does not exceed tolerance $[\delta]$ on the gear tooth accuracy. The length ΔU depends on the relative location of the work and on the involute hob in axial direction of the work.

Incorporating a tolerance on gear tooth longitudinal profile yields a significant reduction of the length of the required gear hob idle distance. Analysis of Figure 11 yields a formula for computation of the allowed shortening of the hob idle distance that is shown in Equation 6.

For engineering computation, a simplified approximate formula is valid and presented in Equation 7.

In that equation, $R_p \equiv R_{1.h}$ is the first principle radius of curvature of the machining surface of the involute hob.

It is already proven (Refs. 9, 10) that the value of the first principle radius of the machining surface of the involute hob can be computed. The computation method is explained in Equation 8.

Base helix angle $\psi_{b.h}$ can be calculated from the formula laid out in Equation 9.

For example, for the left-hand involute gear hob of modulus $m = 3$ mm with pitch diameter $d = 70$ mm, normal profile angle $a_n = 14.30°$, number of starts $Z_h = 5$, and hob-setting angle $\zeta_h = 12.4°$, one can compute the hob base diameter by using the formula in Equation 10.

The hob base helix angle is equal to two formulas listed in Equation 11.

The hob helix angle (Ref. 6) can be found in Equation 12.

The above results yield a final calculation shown in Equation 13.

That result yields $U = 28.471$ mm. In the case, with the tolerance equal to $\delta = 0.1$ mm, one can compute that $\Delta U = 4.561$ mm, and therefore the actual hob idle distance required for perfect involute gear tooth profile generating is equal to $U^* = 28.471–3.949 = 23.910$ mm. Hob idle distance $U^* = 23.910$ mm is 19.1% shorter than the hob idle distance $U = 28.471$ mm. This reduces the length of the hob idle distance that leads to corresponding reduction of the size and weight of a cluster gear and of the gear train housing as well.

Conclusions

Methods of analytical mechanics of gears are applied to determine the exact minimal axial length of a cluster gear for conventional and climb hobbing of spur and helical gears. The results reported in the paper allow users to cut hobbing time and to reduce the size and weight of a gear train and its housing. The approach is especially important for applying the multi-start hob of small diameter. A similar approach can be utilized for hobbing of a non-involute profile, for instance, while machining splines, sprockets, ratchets, etc. The results presented might be incorporated as a part into software for CNC hobbing machines. ✪

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