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AN OVERVIEW TO CHOOSE THE PROFILE SHIFT COEFFICIENT FOR INVOLUTE GEARING INCLUDING PLANETARY GEAR DRIVES

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Abstract: At the beginning of 21th century the literature of involute gear drives describe many different methods for the optimum choice of the profile shift factor. These methods had been worked out for one pair of drives, but leaning on the main principles these can be developed for simple planetary gear drives. At first we review the most significant methods for one pair of gears, then we advert to the methods for planetary gear drives.

Keywords: *Gear, profile shift factor, involute, planetary gear drive*

Nowadays the literature of involute gear drives describe many different methods for the determination of the profile shift factor. These methods had been worked out for one pair of drives, but leaning on the main principles these can be generalized for simple planetary gears. At first we review the most significant methods for one pair of gears, then we advert to the methods for planetary gears.

Literatures are known from the beginning of the 20th century that deals with the methods of increasing the load and lifetime of a gear. BÜCHNER [10] determined, that in the aspect of lifetime the fiction of the teeth is significant. He proves that for the amount of the friction the relative sliding is typical, with taking notice on that the one tooth of the pinion contact i times with the gear. This means that friction loss come forward once on the gear it appears i times on the pinion which is important for wearing. It means that for defining of the tooth correction we have to pay respect for the relative sliding on the pinion multiplied by the gear ratio i.

CSERHÁTI [12] defined the relative sliding by drawing it along the tooth profile. He stated that the sliding maximum wear appears on the top of the gear. He suggested to shorten the top of the bigger gear and to lengthen on the smaller gear.

According to VIDÉKI [39] the sliding velocity is significant for the lifetime. He defined by the use of a general center distance between the gears, if aw>a, the sliding velocity reduces. It is practical to enlarge the center distance until the potential of the teething allow.

DIKER [13], SZENICZEI [32] and BOLOTOVSZKIJ [6] described the sizing for balanced relative sliding by that the biggest values of relative sliding on the two side of the line of action have to be equal. This means that calculated values in the contact limit points are corrected, because the relative sliding in the main point is

zero and increasing hyperbolically. The offset values on the line of action divided by the main point are always on the contact points, either equal or not. Their values can be efficiently lowered by increasing the pressure angle. Moreover at a given pressure angle the two biggest sliding can be equalized and minimized.

COLBOURNE [11] worked out a calculating method for internal gears, where the tooth thickness of the external gear is increased to the previously calculated value by a correction factor, while the tooth thickness at the pitch circle of the inner gear is lowered by this factor. The correctional factor is determined to avoid the interferences with the geometry defined by the modified tooth thicknesses.

YU [42] draws attention to that data of the cutting tool have to be known during the design of an internal gear, because after a sharpening the outer diameter of the tool is decreased, the profile shift factor also decreases. If this change is disregarded at the definition of the geometric data of the internal gear, interference can happen during operation or manufacturing.

BOTKA [8], [9] patented in 1954 the Ganz–Botka gearing system. He had shown that if the relative sliding is equalized at the limit contact points, there is triple equalization. It means that beyond the relative sliding, the momentary contact temperature increasing or flash and the two-factor Almen product (products of the Hertz stress and sliding velocity) are also equalized and minimized. He stated that at lower contact angles the temperature increase of the tooth faces are the biggest at the contact limit points. By increasing the contact angle the heat gets its maximum at the contact limit points. In this case he aimed to equalize these heat pitches.

Based on these result Botka found that it is only recommended to use heat equalized gearing when the gear drive has a tendency to seize. In other cases the equalization if relative sliding is the effective method, moreover the root strength is higher. It is especially true for open drives and slow running, heavy loaded drives which are tend to wear faster. At high running speed it is also recommended to use this method instead of the heat equalization, if scoring is not likely to happen.

GAVRILENKO [19] suggested to define the profile shift factors by the equalized relative sliding accelerations, because the relative sliding method has a fundamental problem. Namely at the main contact point the relative sliding is zero, which leads to that there is no wear in turn at the roots should be significant wear. At the same time the practice doesn't confirm this theory. Therefore the places and extents of wear can be determined by the relative sliding acceleration, that is the tangent of the line of act and the sliding curves. In other words the places of wear and pitting should be determined by the changing of the derivative of the relative sliding along the line of action. The best results can be achieved by equalizing the relative sliding accelerations at the contact limit points.

By the studies of NIEMANN [28] stated that the scoring safety factor is optimal when the slip velocities are equal at the contact limit points. It means that the addendum contact numbers are equal. So if these numbers are equal that means the sliding velocities are also equal.

BLOK [4] found during his test about scoring of gears, that happens at high local temperatures. It comes from the difference of the temperature of the gear body and

the local temperature increase at the contact spot. Scoring happens when the sum of the temperature of the gear body and the local temperature increase pass the so called scoring temperature of the lubricant. The seizure temperature can be determined experimentally. The local temperature increase can be determined by the differential equations created by the author, which can be used to define the contact temperatures of the contact points. In this way the optimal dimensions can be defined for the scoring safety knowing the critical temperature of the lubricant.

WINTER [41] created the universally applicable so called integrated temperature criteria method used to define the scoring toughness, based on the works of Blok and Niemann. The main principle of this is that the heat stress is defined by the sum of the temperature of the gear body and the local temperature increase which is considered to be permanent during the line of action. There is no scoring threat if the forming temperature is lower than the experimentally defined temperature which is based on the lubricant, the material of the gear and the loading conditions. With this calculations the optimal dimensions can be defined in the aspect of optimal scoring conditions, but it can be hardly used for the definition of profile shift factors.

TERAUCHI et al. [33], [34] performed experiments on a gear test equipment with closed power flow. At the contact region they achieved the EHD (elasto-hydrodynamic) lubrication and then they seek the optimal parameters for the profile shift factor which gives the best results for the scoring safety. They stated that slightly positive values causes significant improvement. The defined the critical face temperature independently from the lubricant and the material of the gear, where the value of the scoring safety factor is the highest. They determined that the scoring safety can be estimated by the critical face temperature, thus they defined the profile shift factor which causes this critical temperature.

KOLONITS [22], [23] made a computer program for the Ganz–Botka gearing to get the thermal equalization. He uses the newest, mainly Japanese literature which models the heat formation and distribution. He worked out a model similar to the Blok theory, but simpler than the Japanese model. This give adequate results for the industry.

LI–CHIOU–CHANG–YEN [26] made a computational method where during the design of gears using FEM analysis. There they use the pressure angle and the clearance using as parameters get the optimal dimension for the longest lifetime minimizing the Hertz stress.

PEDERSEN–JORGENSEN [29] define the dimensions of a gear be FEM analysis so that the root has as much stiffness as possible. They show that in this way can be achieved the maximum load which means the longest lifespan at given conditions.
KINCZEL [21] examined the straight involute gears in his paper. His work deals with the pairing of the machining and computing methods in details. After the summary of profile shift factor defining methods the author concluded that the refinement of tooth profiles needs the refinement of the correctional methods. By bringing in the universal correctional principle it can be able to get a correct profile

shift factor by using arbitrary friction coefficient and line pressure function. Moreover it tells which kind of correctional factor should be used.

Starting from the general principle (using symmetric linear line pressure and permanent friction coefficient along the line of act) the examination of specific friction energies – in parity with the results of Kolonits – lead to Botka's theory. Furthermore he proved that, there is a pressure angle, at which, using relative sliding equation, the specific friction energie has a minimum. But this minimum is only the 60…30% of the compensated gearing. This pressure angle range is between 24° and 27°.

BAGLIONI–CIANETTE–LANDI [2] worked out a method by defining the slip velocities to improve the efficiency. The proved that by lowering the sliding velocities, the noise, the wear and the performance degradation are also lowering. They made recommendations for choosing the profile shift factor to get the best efficiency.

IMREK–UNUVAR [20] verify the relationship between the tooth profile, the sliding velocity along the line of action, the line pressure and the flank temperature increase. They stated that the driving gear has to made with positive shift factor, while the driven gear with negative shift factor, to get the sliding velocity equalized at the limit contact points. This ensures the smallest value also and minimum wear.
TERPLÁN [36], [37] worked out a calculating and constructing method for the compensated teething of planetary gears. He proved that if we sizing for equalized relative sliding and using ordinates at the limit contact points, the problem of fourfold equalization can be reduced threefold.

With his collaborators (SZOTA–SCHOLTZ) worked out a simpler method based on the VÖRÖS [40] method for the construction of the relative slidings of a planetary gear. The calculation method is iterative which solves the problem of the threefold equalization numerically. It is advantageous to use this method to define the profile shift factors because it is simple and clear. He summarized the values of relative slidings and shift factors in tables [36], [37].

APRÓ [1] found out by examining the planetary gears with on degree of freedom, that by reducing the number of teeth of the planet gear general gearing can be used when the energy-flow is the following: sun gear-arm, fixed ring gear. The main principle of the method was that holding the good contacts it is practical the improvement of external contact which can be achieved by reducing the number of teeth. He summarized in tables the reduction of the number of teeth of the planet gear depending on the sum of teeth on the sun and planet gears and the pressure angle.

BOLOTOVSZKIJ [5] examined the teeth correctional factors planar and spatial limiting contours. These contours are created in the coordinate system of profile shift factors by using the different interference equations as limiting function which create there surfaces. These surfaces are used as limits for the interference free zone. If the sun gear and the planet gear contact is equalized for relative sliding in the defined area, than the profile shift factor for the ring gear can be determined with an equation similar to the collinear assumption of the planetary gear.

In literature, beside the general solution of external contact, the following recommendation can be found for the profile shift factor of the ring gear:
According to BERGSTRÄSSER [3] the definition of the profile shift factor of the ring gear, after the definition of sun and planet gears as DIN 3992, are the following:

$$x_3 \,.\, m = B_{32} \,.\, a_w + x_2 \,.\, m, \tag{1}$$

where the B_{32} parameter is the Kutzbach involute function:

$$\boldsymbol{B_{32}} = \frac{\boldsymbol{inv\alpha_{wt32}} - \boldsymbol{inv\alpha}}{\boldsymbol{tg\alpha}}, \tag{2}$$

in which the α_{wt32} pressure angle of the outer-inner contact and α is the standard presssure angle.

RICHTER [31] defines the profile shift factors for external meshing according to DIN, but uses the following equation for the ring gear:

$$x_3 = x_1 + 2x_2. \tag{3}$$

KUDRJAVCEV [24] gave the same equation as Richter for the profile shift factor of the ring gear, but defines the profile shift factors of the external contact by the equalized relative slidings method.

SUMMARY

We can see that the determination of the profile shift factor is as comprehensive as the widespread of the usage of gears. We can settle also that there is no universal method for the determination of the profile shift factor, which only can ensure the best efficiency and longest lifespan for gears at the same time. Accordingly it is recommended to examine the literature carefully for the specific problem and find the best solution for the given conditions.

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