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Theoretical and experimental investigation of wear in internal gears

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ABSTRACT

Internal gears are commonly used in automotive and aeronautic industries as external sun gears of planetary mechanisms. Internal gears have some advantages such as low sliding velocities, low contact stresses, high contact ratios compared with external gears. Therefore, manufacturing is more difficult than external gears. For this reason, it is necessary to determine the working conditions of internal gears carefully. For that purpose, wear in internal gears is investigated theoretically by adapting Archard's wear equation to internal gears and a MATLAB® programme is written to solve this modified equation. The aim is to determine the wear values in different conditions by using this modified equation. In addition, a fatigue and wear test equipment is designed and manufactured which is similar to FZG (Forschungsstelle für Zahnrader und Getreibbau) closed circuit power circulation system in working principle to investigate wear in internal gears experimentally. Internal spur gears which are manufactured from St50 are used in the experiments with different torques and motor speeds. Wear is determined that occurs in tooth profiles of internal gears for different load cycles. It is seen that the results obtained from theoretical and experimental studies are compatible.

1. Introduction

Internal gears differ from external gears in that their teeth are directed to the interior of gear center. An internal gear mechanism meshes on concave and convex surfaces. Thus, internal gears have advantages of low contact stresses, high contact ratios and low sliding velocities when compared with external gears [1]. Internal gears are widely used in gear boxes as in planetary mechanisms, transmission boxes, and differential mechanisms, in cranes, automotive and aeronautic industries [2]. Researchers are usually investigating gear geometry and tooth fillet stresses with the internal gears.

Some programs are developed for determining the geometry of internal gears to simplify the manufacturing of gears [3–5]. With the help of these programs, optimized internal gears can be manufactured that are appropriate for working conditions by changing the geometry of the gear (kinematic limits, minimizing center distance or gear volume) easily. These programs are based on theoretical calculations and are not supported with experimental data's. In studies about the optimization of rim thicknesses on internal gears [6–8], the effects of rim thickness on the tensile and compressive stresses in tooth root are determined. The location and magnitude of maximum tangential stress are aimed to determine in studies for calculating stresses occurring in tooth root of internal gears [9–12]. Finite element method is used for calculating stresses theoretically and experimental measurements are carried out with strain gages and photo-elastic tests.

Wear is the most common failure type in the working surfaces of mechanical systems. It occurs with the breakage of small particles from both surfaces which are in contact and meshing with each other [2]. Wear is commonly seen in gear mechanisms. Mathematical modeling of wear is firstly suggested by Archard [13]. Afterwards, these formulas are used by Flodin and Andersson [14–19] to determine wear theoretically in various external gear mechanisms. The aim of this study is to investigate wear theoretically and experimentally in internal gears along line of action in different working conditions.

2. Wear model in internal gears

In this study, the wear model which is described by Archard [13] and adapted to external gears by Flodin [14] is used by arranging it for internal gears. This model is based on single point observation method [16]. The teeth of mating gear pairs make sliding and rolling motions. As a result of these motions, wear occurs in meshing surfaces. The most used wear model is

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Archard's wear equation;

$$\frac{V}{s} = K\frac{W}{H} \tag{1}$$

where V is the volume of worn material, s is the sliding distance between contacting surfaces, K is the dimensionless wear coefficient, W is the applied load and H is the surface hardness of worn material. It is necessary to take into account the differential sliding and rolling motions from the beginning to the end of meshing tooth contact to use this equation in gears.

At the beginning of tooth contact, tooth root of driven gear (pinion) contacts with the tooth tip of the driving (internal gear) gear Fig. 1(a). During contact, this region is under the effect of sliding and rolling action. Sliding velocity changes continuously to the pitch of the circle. Sliding velocity is zero on the pitch circle because it is equal with opposite direction for the pinion and internal gear Fig. 1(b). In this region, tooth couples make only rolling motion. At the end of contact, sliding is between the tooth tip of pinion and tooth root of internal gear Fig. 1(c).

It is more appropriate to investigate wear on local points rather than during contact because the effect of sliding and rolling are different on interacting surfaces. If wear is described to any point 'p' from meshing surfaces, Archard's wear equation can be written as follows;

$$h_p = \int_0^s kPds \tag{2}$$

where h is the wear depth of point 'p', k is the dimensional wear coefficient and P is the local contact pressure. According to Andersson, if single point observation method [16] is applied to meshing gears, in other words, if it is required to express wear of any point where the tooth profiles of pinion-internal gear couple are meshing with each other during meshing depending on running interval, the wear model can be written as follows;

$$h_{p,(n)} = h_{p,(n-1)} + kP_{p,(n-1)}s_p \tag{3}$$

where $h_{p,\,(n)}$ is the wear depth at point 'p' on the flank after n running interval, $h_{p,\,(n-1)}$ is the wear depth at the same point but one running interval before, $P_{p,\,(n-1)}$ is the pressure on point 'p' and s_p is the sliding distance of point 'p'. Pinion-internal gear couple contact with each other in various positions from the beginning of contact to the end during meshing.

Consider that two points' p_1 and p_2 from the teeth flank of the pinion and the internal gear which are opposite to each other during meshing to determine the sliding distance in the contact points. These points can be in three different positions from the beginning of contact to the end. First position is the case of contact beginning of both points at the same time Fig. 2(a) and in this case there is no distance between the points. In the second position, the point which belongs to internal gear p_2 is still in the meshing while the point which belongs to pinion p_1 is leaving contact, therefore a distance of s_{p1} arises between two selected points Fig. 2(b). In the third position, while p_2 leaving meshing, a distance of s_{p2} arises between it and p_1 Fig. 2(c) [17].

In the contacting gear pair, when the point p_1 moves for a distance of $2a_H$ (semi-herzian contact width) along the contact length, the point p_2 moves for a distance of $2a_H$ (U_2/U_1) where U_1 and U_2 are the peripheral velocities of pinion and internal gear respectively. Similarly, when the point p_2 moves for a distance of $2a_H$, the point p_1 moves for a distance of $2a_H(U_1/U_2)$. Since the

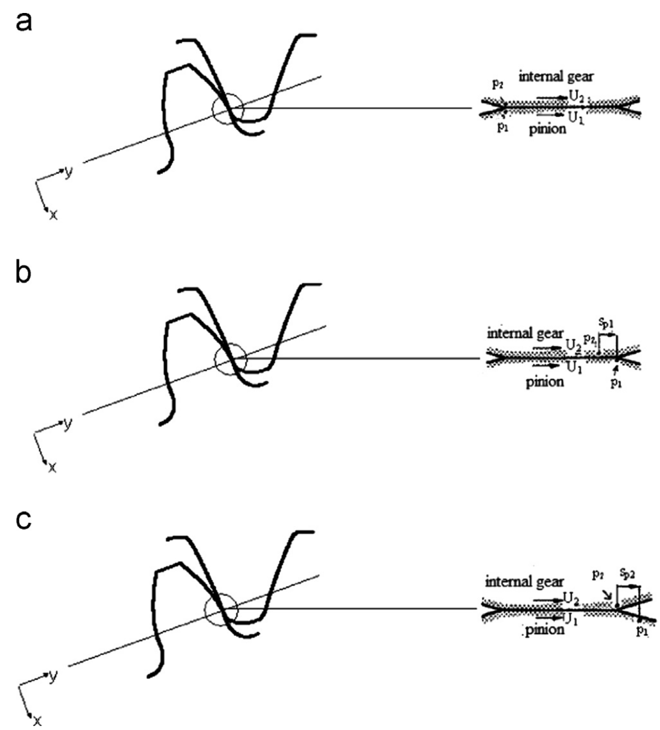

Fig. 2. Distance between points on teeth during contact [17]: (a) beginning of contact, (b) distance between the points is equal to s_{p1} and (c) distance between the points is equal to s_{p2}.

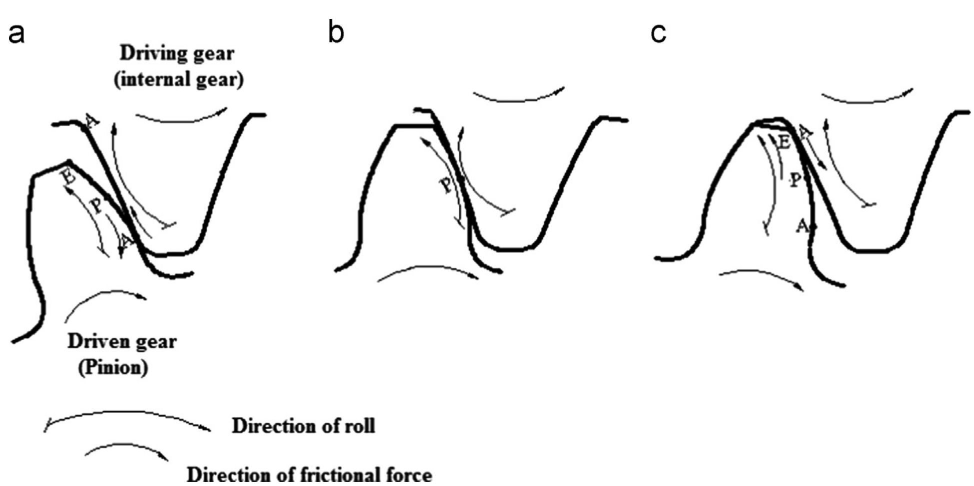

Fig. 1. Mechanics of gear tooth contact [20], (a) beginning of contact, (b) contact on pitch point and (c) end of contact.