CALCULATION OF THE EFFICIENCY OF ZTA-TYPE WORM GEAR DRIVES ON THE BASE OF THE ETHD LUBRICATION THEORY

TIBOR BERCSEY, PÉTER HORÁK
Institute of Machine Design, Budapest University of Technology and Economics
Műegyetem rkp. 3., Budapest, Hungary, H-1111

Abstract: *The friction–lubrication conditions are basically influenced on the geometrical–kinematical relations, the contact pressure distribution, the material and the lubricant between the tooth pairs. The tribological conditions of ZTA–type worm gear pairs were investigated – applied the results (geometry, kinematics, load) determined in foregoing publications of the authors – and the efficiency of the drive was calculated.*
Keywords: *worm gear drive, ETHD lubrication theory, efficiency, minimum oil film thickness*

1. INTRODUCTION

To calculate the efficiency, to reduce the friction and wear and to optimise the tribology conditions of highly loaded worm gear pairs information is needed about the geometrical-kinematical relations and the contact pressure between the flanks.

The subject of this investigation is a worm gear pair having circular arch profile in axial section. This type of worm drive (ZTA) was developed at the University of Miskolc and manufactured at the Machine Factory in Diósgyőr.

2. GEOMETRY, KINEMATICS AND CONTACT CONDITIONS

The numerical investigations of the geometry and kinematics of ZTA-type worm gear drives were published by DUDÁS [1], BERCSEY and HORÁK [2]. The kinematics method LITVIN [3] were used for the calculation of the instantaneous contact lines, the changing radii of curvature as well as the hydrodynamically effective velocity and the sliding velocity.

Figure 1 shows the instantaneous contact lines in the section perpendicular to the axes of the investigated worm.

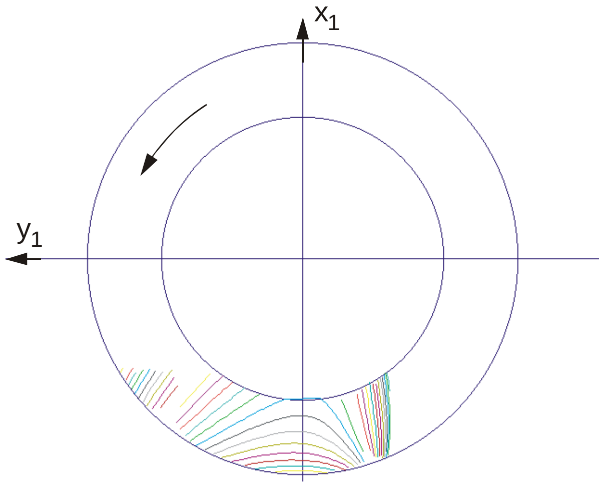

Fig. 1. Instantaneous contact lines in different meshing positions

The calculated reduced normal radii of curvature of the gear flanks and the hydrodynamically effective velocity are shown in Figure 2 and 3.

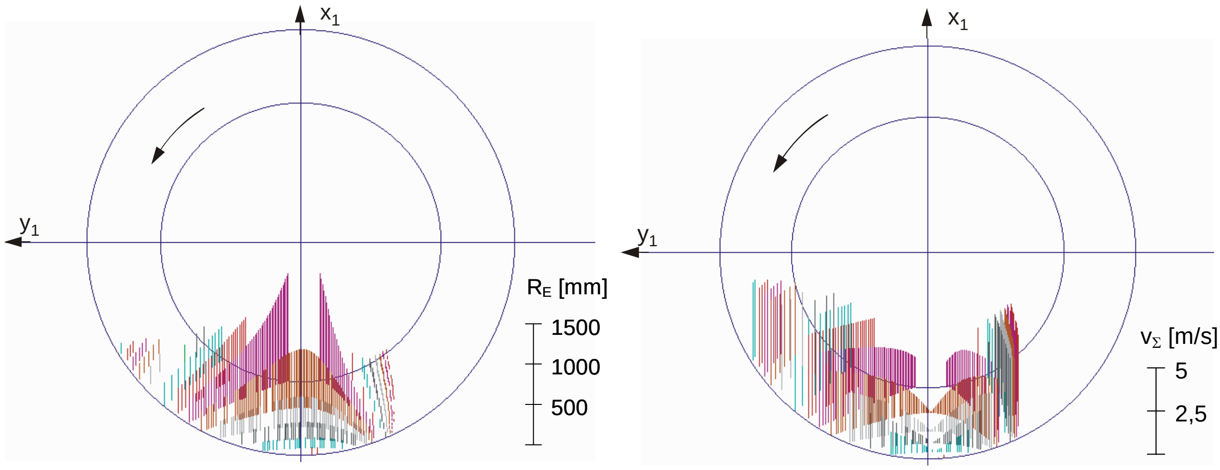

Fig. 2. Calculated reduced normal radii of curvature and hydrodynamically effective velocity along the instantaneous contact lines

A constant Stribeckian-pressure (k_H) was supposed at the calculation of the contact conditions.

$$k_H = \frac{T_1}{2 \cdot R_E \cdot \sum_{j=1}^{n} \left(e_{x(j)} \cdot y_{(j)} - e_{y(j)} \cdot x_{(j)} \right) \cdot \Delta l_{(j)}}, \tag{1}$$

where T_1 is the load on the worm, $\frac{1}{R_E}$ stands for the reduced normal radii of curvature, e_x, e_y are the coordinates of surface normal-, x and y the coordinates of radius vector and Δl is the length of the contact line section.

3. NUMERICAL INVESTIGATION OF THE TRIBOLOGICAL CONDITIONS

The calculation model supposes that the contact surfaces are completely separated by the lubricant film so there is not any metal-metal contact. For the calculation of the minimum oil film thickness, the coefficient of friction as well as the power loss, the parallel solution of the Reynolds, the energy and the Laplace's equations is required. The algorithm developed by the author [4] solves the nonlinear equation system by means of numerical integration considering the effect of the changing temperature in the lubricant film. Figure 4 shows the main steps of the calculation according of the thermal elastohydrodynamic lubrication theory.

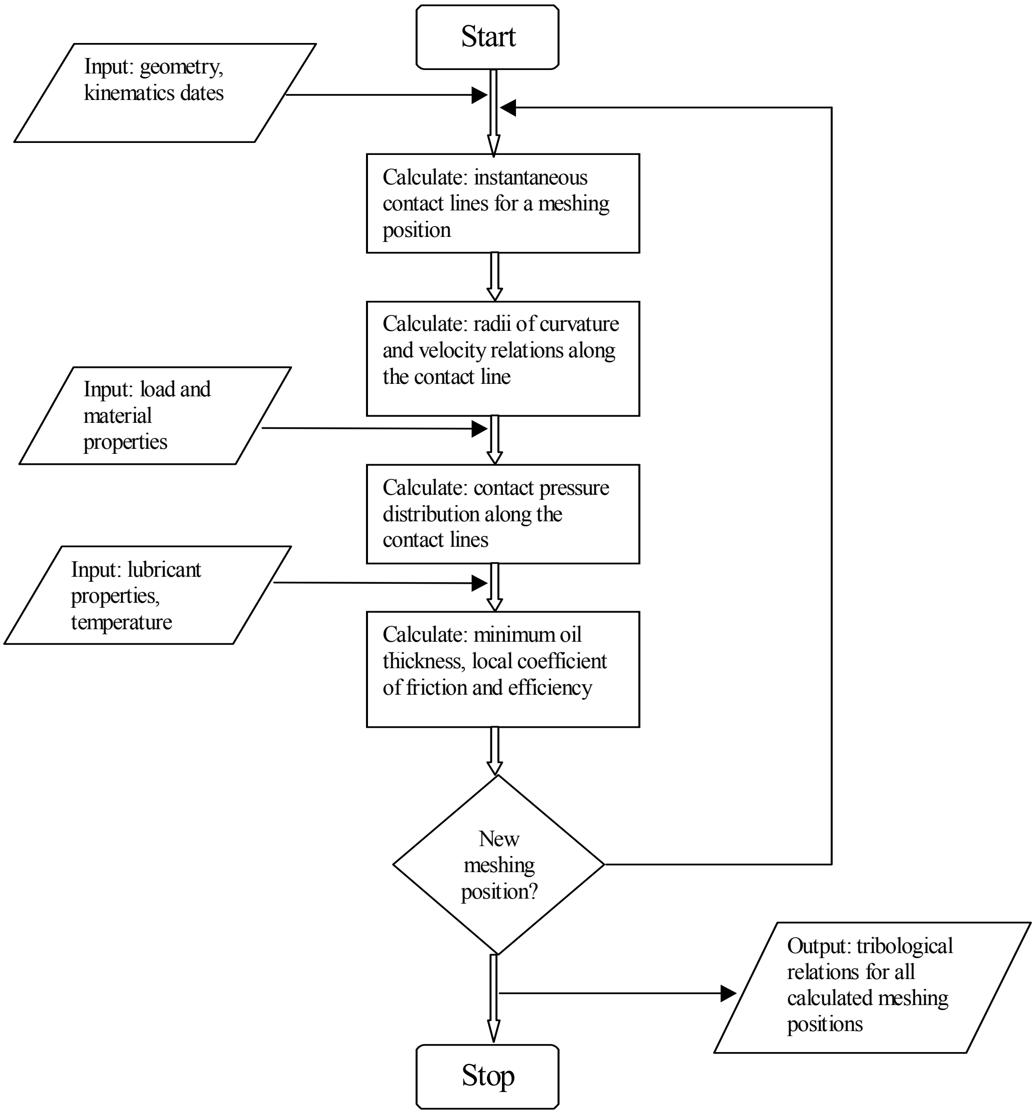

Fig. 4. The main steps of the numerical investigations

The calculated temperature distributions of the worm flanks have maximum values near to the mid plane of the worm (Fig. 5.)

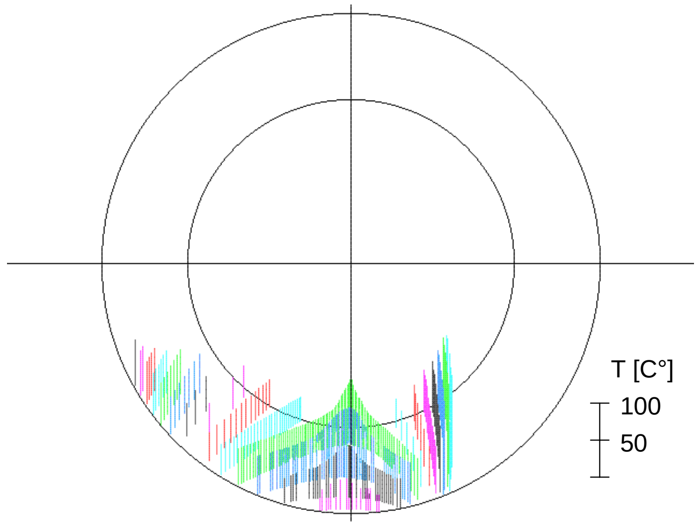

Fig. 5. Temperature distribution of the worm flanks in different meshing positions

The changing coefficient of fluid friction influenced by the temperature of the contacting gear flanks is shown in Figure 6.

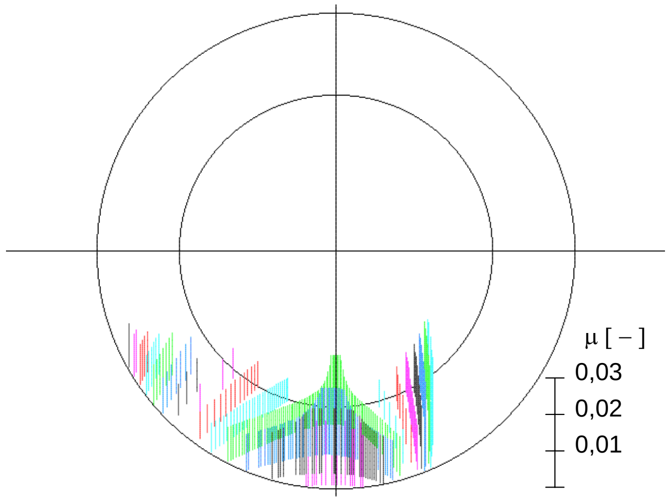

Fig. 6. Calculated coefficient of fluid friction along the instantaneous contact lines in different meshing positions

Mixed lubrication condition can be developed at the mid plane of the worm, because the minimum oil film thickness approaches here to zero (Fig. 7.). The efficiency of the drive can be increased by the elimination of this area [4].

The power loss of the gear flanks along a section of length Δl of the instantaneous contact line can be described by the following equation:

$$dP_v = \mu_h \cdot w_b \cdot \Delta l \cdot v_g \qquad (2)$$

where μ_h is the coefficient of fluid friction, w_b stand for the load along the contact line and v_g is the sliding velocity.

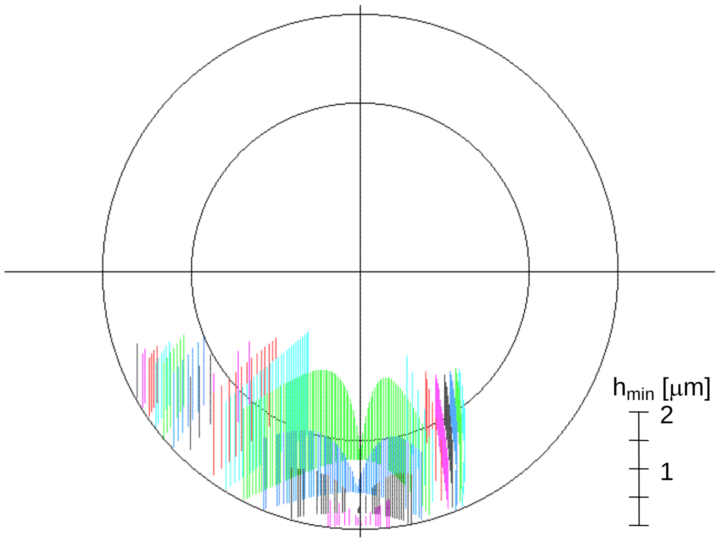

Fig. 7. Distribution of minimum oil film thickness along the contact lines

The power loss belonging to the rotation angle φ_1 along the whole contact line of length l as follows:

$$P_v(\varphi_1) = \int_l dP_v \quad . \tag{3}$$

The average value of the power loss occurring between the gear flanks is

$$P_{ave} = \frac{P_v(\varphi_1)}{n_{\varphi_1}} . \tag{4}$$

In equation (4) n_{φ_1} stands for the number of the investigated different meshing positions.

The efficiency of the meshing

$$\eta_{mesh} = \frac{P_{in} - P_{avel}}{P_{in}} , \tag{5}$$

where P_{in} is the input power.

The calculated results were compared with the efficiency measured by DUDÁS [1] on the same worm gear drive (Fig. 8.).

The 5% difference issues from the power loss at the bearings, from the swirling of oil and from the assumption of fluid lubrication at the numerical solution.

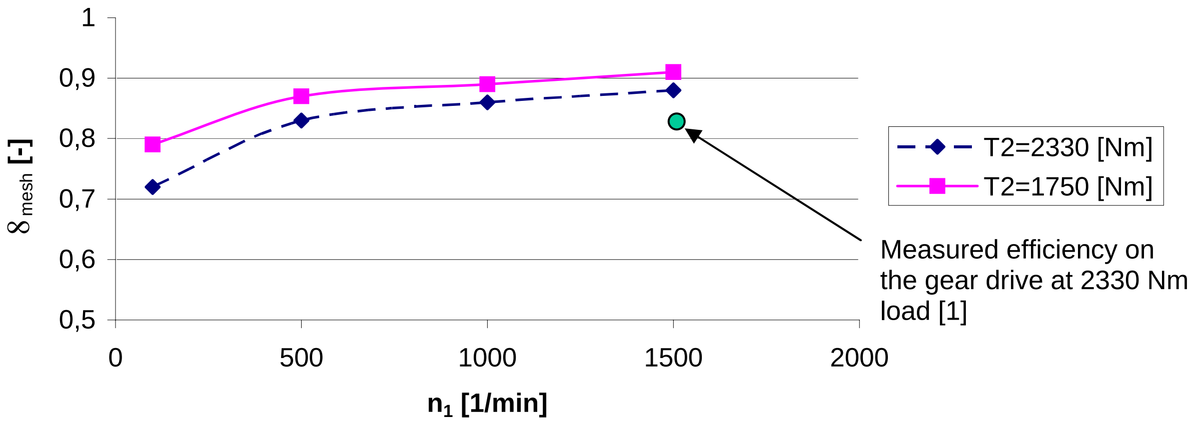

Fig. 8. Calculated and measured efficiency of the gear drives as a function of the speed of the worm at different load

4. CONCLUDING REMARKS

The investigations have proved that the results of the calculation – on the base of the thermo elastohydrodynamic lubrication theory –approximate well the measured values on the worm gear drive.

Further results of the investigations are that the lubrication conditions are unfavourable at the mid plane of the worm, mixed lubrication condition can be developed, and the efficiency can be increased by the elimination of this area.

To consider the micro geometry of the flanks for the calculation of the mixed film lubrication, as well as the manufacturing and assembly errors of the drive, further investigations are required.

ACKNOWLEDGEMENT

The presented research was supported by the OTKA T 034 968 research program.

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