Effect of bearings preload on the tribological performance of elastohydrodynamic conjunctions in automotive manual transmissions

Effect of Bearings Preload on the Tribological Performance of Elastohydrodynamic Conjunctions in Automotive Manual Transmissions

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1. Introduction

Fuel efficiency and reduced emissions are key objectives for the automotive industry. There has been significant focus on engine technologies in order to mitigate thermal and engine frictional power losses, but limited attention has been paid to the efficiency enhancement of the transmission losses, which include churning losses, windage losses, frictional losses in engaged and non-engaged gears, as well as bearing losses.

One of the key transmission components in terms of efficiency and durability are the bearings. Amongst the different types employed in transmissions, the tapered roller bearings are vastly used due to higher and controllable stiffness and durability. One of the main parameters in controlling bearing efficiency, durability and functionality is the preload or clearance. The preload is usually prescribed and applied at the design stage, but can change significantly under different working conditions, particularly under varying temperatures.

In this paper, the effect of preload on the tribological behaviour of the elastohydrodynamic conjunctions of tapered roller bearings in manual transmissions has been investigated. An objective approach has been proposed to assess the effect of preload on bearing efficiency.

The bearing preload can be defined as the static contact force between the rolling elements and the races. The thermally induced preload appears when the transmission parts expand unevenly due to temperature change during operation [1].

The thermally generated preload is considered to develop in four stages:

1. Friction heat generation
2. Heat transfer
3. Thermal expansion
4. Contact force inducement (induced preload).

The last stage of the thermally induced preload is the result of the expansion of the bearing, casing and the shaft [2].

There are factors that can affect the generation of thermally induced preload. Some of them are:

1. Assembly tolerances
2. Load [3]
3. Frictional heat
4. Rotational speed [4]
5. Heat transfer [5]

2. Methodology

Friction contribution on tapered roller bearings comprises boundary and viscous contribution, which combined give the total friction that generated on a tapered roller bearing during operation.

The boundary friction [6] is given by the following equation:

$$F_{boundary} = \tau_L * A_a \tag{1}$$

Where, τ_L is the limiting shear stress and A_a is the asperity contact area. The limiting shear stress is given by equation (2):

$$\tau_L = \tau_0 + \varepsilon * P_{mean} \tag{2}$$

Where τ_0 is the Eyring stress, ε is the slope of the lubricant limiting shear stress-pressure curve and P_{mean} is the mean pressure applied on each roller at each instant of time during operation, as described by equation (3):

$$P_{mean} = \frac{W_a}{A_a} \tag{3}$$

If a Gaussian distribution of asperity peaks is assumed, then there is a fraction of load that can be carried by these asperities, as given by equation (4):

$$W_a = \frac{16\sqrt{2}}{15} * \pi * (\xi * \beta * \sigma)^2 * \sqrt{\frac{\sigma}{\beta}} * E' * A * F_{\frac{5}{2}}(\lambda) \tag{4}$$

In equation (4), ξ is the asperity density per unit area, β is the average asperity tip radius, σ is the composite RMS surface parameter, A is the apparent contact area, E' is the reduced elastic modulus of contact and $F_{5/2}(\lambda)$ is a statistical function for a Gaussian distribution of asperities.

The asperity contact area A_α is given by:

$$A_a = \pi^2 * (\xi * \beta * \sigma)^2 * F_2(\lambda) \tag{5}$$

Where $F_2(\lambda)$ is another statistical function for Gaussian distribution of asperities.

The viscous friction [6] is described by:

$$F_{viscous} = \mu * q \quad (6)$$

Where μ is the coefficient of friction and q is the applied load at each roller at each instant of operation which depends on the input applied load on the bearing. The coefficient of friction can be described by equation (7):

$$\mu = 0{,}87 * \alpha * \tau_0 + 1{,}74 * \frac{\tau_0}{P_{mean}} * \ln\left(\frac{1{,}2}{\tau_0 * h_{c0}} * \left(\frac{2 * K * \eta_0}{1 + 9{,}6 * \xi}\right)^{1/2}\right) \quad (7)$$

Where α is the pressure-viscosity coefficient, h_{c0} is the dimensionless central film thickness, K is the conductivity of the lubricant and η_0 is the dynamic viscosity of the lubricant at atmospheric pressure.

The film thickness of the lubricant during operation, is given by:

$$h_{c0} = 4.31 * U_e^{0.68} * G_e^{0.49} * W_e^{-0.073} * \left\{1 - exp\left[-1.23 * \left(\frac{R_y}{R_x}\right)^{2/3}\right]\right\} \quad (8)$$

In equation (8), R_x, R_y are the radii of curvature along the direction of sliding and side leakage, respectively. The non-dimensional governing groups, are described by the equations that follow.

$$U_e = \frac{\pi * \eta_0 * U}{4 * E_r * R_x}$$

$$W_e = \frac{\pi * W}{2 * E_r * R_x{}^2}$$

$$G_e = \frac{2}{\pi} * (E_r * \alpha)$$

$$h_{c0} = \frac{h_0}{R_x}$$

In the case of tapered roller bearings this load is the equivalent of axial and radial loads applied.

For the calculation of the total applied load on the rollers of the bearings on the normal plane an iterative method is used. Considering that on tapered roller bearings both axial and radial loads are applied, there are also axial and radial deflections that need to be calculated. [7] The axial and radial deflections are given by equations (9) and (10), respectively:

$$\delta_a = \left(\frac{F_a}{z * K_n * J_a * \sin\alpha}\right)^{0.9} + P_r \quad (9)$$

$$\delta_r = \left(\frac{F_r}{z * K_n * J_r}\right)^{0.9} + P_r \quad (10)$$

Where F_a and F_r are the applied loads on the axial and radial directions respectively, z is the number of rolling elements of the bearing, K_n is the load deflection factor, α is the contact angle and P_r is the applied preload. J_a and J_r are the axial and radial integrals, respectively. They depend on the load distribution factor e, which is presented in equation (11), and their values are given by tables [7].

$$e = \frac{1}{2} * \left(1 + \frac{\delta_a * \tan\alpha}{\delta_r}\right) \quad (11)$$

When the values of both deflections are calculated, then the total applied load on the normal plane of the rolling elements can be calculated using equation (12):

$$q = \frac{F_r}{J_r * z * \cos\alpha} = \frac{F_a}{J_a * z * \sin\alpha} \quad (12)$$

Since both boundary and viscous friction have been calculated, the total friction load on the bearing is given by:

$$F_{friction} = F_{boundary} + F_{viscous} \quad (13)$$

2.1. Results

In this paper a typical tapered roller bearing of a manual transmission gearbox is examined. Figure (1) shows the bearing structure with its main components.

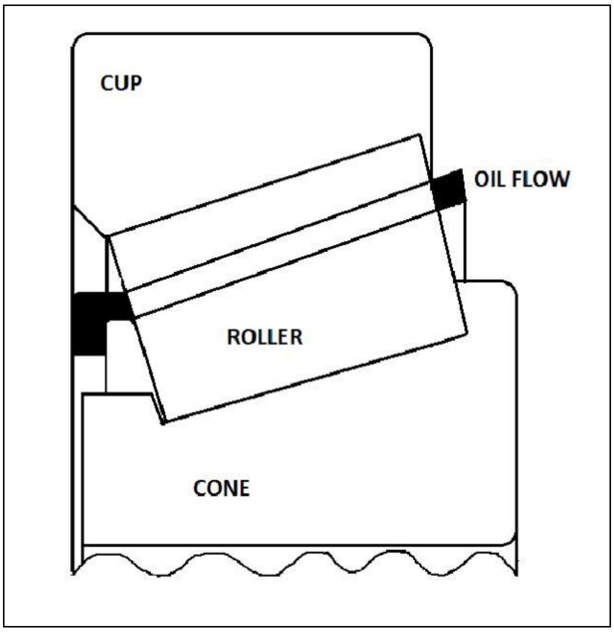

Figure 1 Tapered roller bearing

A model has been developed using Matlab to calculate the effect on both boundary and viscous friction on each rolling element due to the applied preload. The input data of the system examined are summarized in the following table.

Table 1 Input data of examined system

Number of Rollers	15
Roughness Parameter	0.056
Reduced Elastic Modulus of Contact	$3.451*10^{11}$
Axial Applied Load [N]	6514.9
Radial Applied Load [N]	3455.3
Initial Preload [mm]	0.030
Lubricant dynamic viscosity at atmospheric pressure[Pa*s]	0.00404
Pressure viscosity coefficient[1/Pa]	$1.05*10^{-8}$
Lubricant conductivity[W/m*K]	0.137
Rotational Speed [rpm]	3000

For the system examined, three different values were considered for the applied preload, so that its effect on the friction can be assessed. For the first case, preload was added at 30 μm, introduced as clearance on the roller of the bearing, whereas for the second case an increase of 20% of the initial value was applied and for the third case the increase was 40%.

In the following figure, the difference of the actual equivalent of the applied load at one of the rolling elements of the bearing can be seen. It is shown that, based on the position of the roller on the shaft, there is a load contribution leading to positions of zero load. For the loaded conditions, the difference of the load due to the application of preload can be seen.

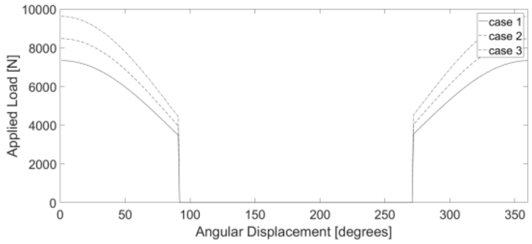

Figure 2 Applied Load on one bearing roller for a complete rotation

In figure (3), the thickness of the film that is established during operation can be seen, for the three different values of the applied preload. The maximum film thickness is applied when the roller takes no load. For that reason, the difference on the film thickness for the three different values of preload can be seen for the loaded positions.

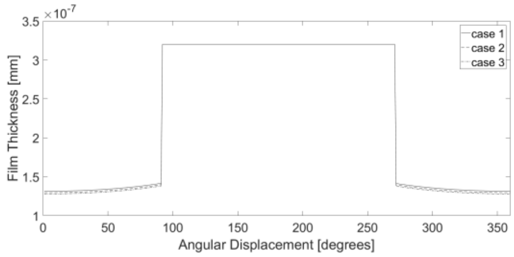

Figure 3 Film Thickness on each bearing roller for a complete rotation

The effect of preload on viscous friction can be seen in figure (4). For the loaded period, as mentioned before, there is an obvious increase on the friction that is created as preload increases.

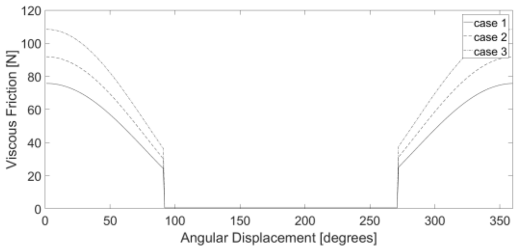

Figure 4 Viscous Friction on each roller at each instant of operation for the three tests

The values of boundary friction on the bearing for all three preloads are shown in figure (5). In that case, there is an increase on maximum values of boundary friction, as was shown for viscous friction as well, but the effect is slightly smaller.

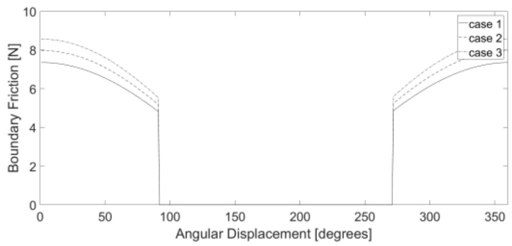

Figure 5 Boundary Friction on each bearing roller for a complete rotation

Finally, in figure (6) the total friction torque can be seen. The increase on the torque for the three different case is noticeable. However, it should be mentioned that for each case the torque is almost constant, with minor fluctuations, since the total load that applies on the bearing as a whole is constant as well.

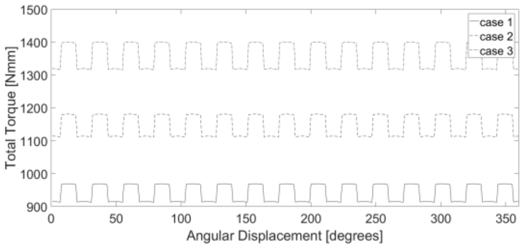

Figure 6 Total torque on bearing for complete rotation

3. Conclusion

The effect of preload on the tribological behaviour of the elastohydrodynamic conjunctions of tapered roller bearings in manual transmissions has been investigated. More specifically, a model has been built featuring the effect of preload on the generated boundary and viscous friction on typical tapered roller bearings used in manual transmissions. The effect is noticeable even from the change that it creates on the applied load for constant external load. For 20% increase of preload, the actual load on the roller of the bearing increases by around 1000 N and an additional increase of 1000 N applies when preload is increased by 40%.

The effect can be seen as well when viscous friction is examined. For the second case, there is a 25% increase on the maximum value of viscous friction. For the third case, an additional increase of 22% for the maximum viscous friction value occurs. The same effect can be observed in boundary friction as well. The second case shows a 30% increase of the maximum value while for the third case an additional increase of 12% of the maximum value is observed. Lastly, following the difference on the applied load for the three cases examined, the total friction torque shows also noticeable differences. The increase between the first and the second case is around 21%. The difference between the second and the third case is around 17 %.

The results of the simulations presented show the need for controlling the applied preload on transmission bearings in order to decrease the frictional losses generated. The latter could increase the efficiency of manual transmissions.

4. Acknowledgment

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