MODELING THE SYNCHRONISATION BEHAVIOUR OF MANUAL CAR GEARBOX SYNCHROMESH MECHANISMS

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Abstract: *This paper deals with the modelization of synchronisation process with synchromesh mechanisms of manual gearboxes. The structure of the studied synchromesh mechanism is presented, then a theoretical model of the friction process is mentioned. Developement of this model is proposed by including the effect of the oil viscosity variation due to pressure and temperature, as well as one of churning and bearing losses. Finally, computer simulation and experimental results are presented and compared.*

Keywords: *synchromesh, energy losses, double bump, numerical simulation*

1. INTRODUCTION

Synchromesh mechanisms are important parts of automotive manual gearboxes. They are used to ease gear speed changing by decreasing the needed force and to make speed changing without noise. This is made by synchronising the angular velocity of the gear to be engaged. There are many phenomena influencing this process. One of them is the effect of losses in the gearbox whose simplified model will be presented. Another important phenomenon is the double bump whose origines are described as well. Finally, simulation results and experimental data will be presented, and discussion will give some concluding remarks.

2. THE STUDIED SYNCHROMESH MECHANISM AND THE SYNCHRONISATION MODEL

One of the types manufactured today is the Borg-Warner synchromesh mechanism (Fig. 1). The gear turns free on the gearbox axle. The transmission of the power from the gear to the axle is assumed through the sleeve and the synchro hub, when the gear is engaged.

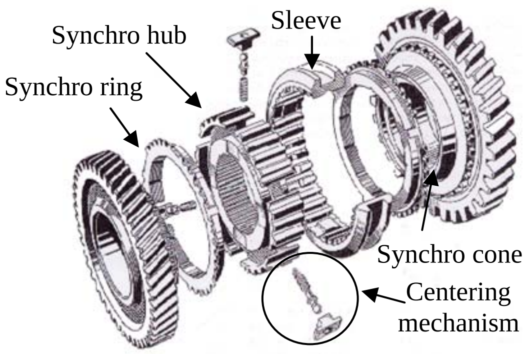

Fig. 1. The studied Borg-Warner synchromesh mechanism

During the gear changing, synchronisation is realized by friction torque. As relative speed and lubrification of the surfaces vary strongly, a model including these effects has to be considered. Such models were described by Progri et al. [5] and Ghaem [3]. They suppose that three successive friction phases can be defined.

Another important phenomenon during gear changing is the “second bump” or “double bump”. In fact, it is a force peak in the axial force versus time diagram, in a well defined zone of the axial displacement. It has important effect on the human shift feeling.

The friction model presented in [5] contains some simplifications which have to be improved. For example, Ghaem [3] considers constant temperature. Of course, temperature varies as energy is dissipated. Spreckels [6] shows the magnitude of temperature variations: on the conical surfaces temperature rise may reach 120°C, while body temperature may increase with 10-20 °C. On the other hand, the climatic conditions may impose a lower body temperature limit of –25°C, while hard working conditions give an upper limit of 120°C. As oil viscosity varies exponentially with temperature, it is important to consider temperature effect.

3. MODEL OF MECHANICAL ENERGY LOSSES

Another important factor to be considered is the effect of internal losses. These energy losses come mainly from axial and radial bearing behaviour, from the input axle joint, and from oil churning losses. In a gearbox, there are as many gear pairs as speeds. One pinion of each gear pair turns always with a different angular velocity that the one of the axle. From here, bearing losses can be calculated. Then, one pinion of the gear pairs is still partially submerged in oil, and churning losses can be calculated from here.

The equation of the input axle joint loss is taken from Roulet [7]:

$$M_{spi}=0{,}06+0{,}06\left(1-e^{-\frac{1000}{n}}\right) \tag{1}$$

where n – angular velocity given in *1/min*. The equation of the radial and axial bearing loss comes from Roulet [7]. The radial bearing loss is given by:

$$M_{pal}=\frac{2\pi\cdot\mu\cdot r^3 l\omega}{R-r} \tag{2}$$

where l – bearing length, R – external radius, r – internal radius, ω – relative angular velocity. The axial bearing loss is given by:

$$M_{butax}=\frac{\pi\cdot\mu\cdot\left(R^4-r^4\right)\omega}{h} \tag{3}$$

where h – thickness of the oil film between mating surfaces, R – external radius, r – internal radius, ω – relative angular velocity. The equations of the oil churning losses are taken from Bonness [1] and Roulet [7]. They are given depending on the wet part k of the radius r:

$$M=\frac{\rho\cdot v^2}{2}\cdot r\cdot A\cdot C_M \tag{4}$$

where A – wet surface, C_M – torque coefficient. Three cases can be discerned: $\frac{k}{r}<0{,}9$, $0{,}9<\frac{k}{r}<1{,}3$ and $1{,}3<\frac{k}{r}<1{,}9$. For the first and the last case, different equations are given to calculate the torque coefficient C_M depending on the Reynold’s number Re. For more details, see in [1] and [7]. In the second case the following linear approximation is considered between the limit values:

$$C_M = C_M(0{,}9) + \left(C_M(1{,}3) - C_M(0{,}9)\right)\frac{\frac{k}{r} - 0{,}9}{0{,}4} \qquad (5)$$

Loss torques always slow down the angular velocity of the gear to be engaged. Consequently, they are useful when engaging a higher gear, because they decrease the axial force. On the other hand, they increase the axial force when engaging lower gear.

4. MEASUREMENTS AND SIMULATION

Measurements has been realised on a test rig of Federal Mogul Sintered Products. On this test rig only one synchromesh can be studied. The structure of the rig is described in [4]. Due to its structure, the test rig presents an important temperature dependent loss torque, whose magnitude is one order bigger then the one of the normal churning and other losses. Measured data can be seen in figure 2.

To simulate the gear changing behaviour, a numerical simulation software has been made under Delphi environment, including all the mentioned models and improvements. It is able to simulate the gear changing either on the test rig or in a gearbox.

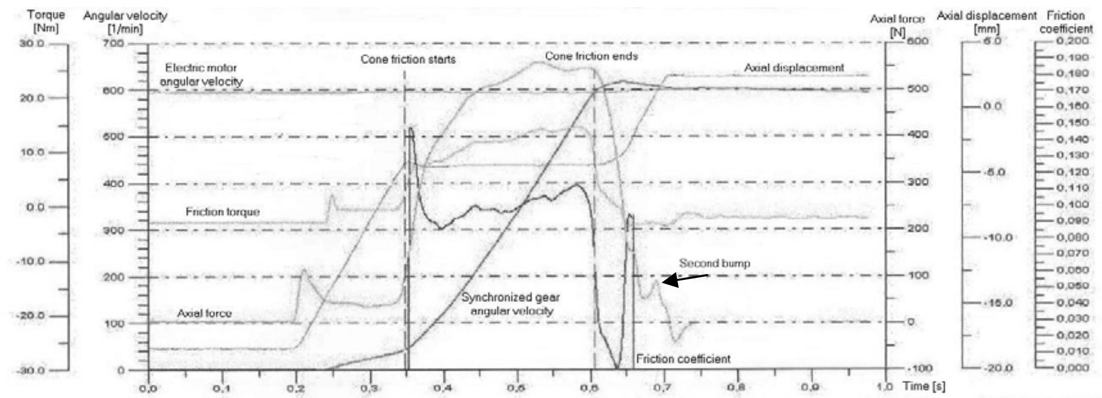

Fig. 2. Measured data

Simulation of energy losses on the test rig has been performed. As only one synchromesh mechanism is studied, and only one gear is turning, the bearing and churning energy losses are low, *0,03 Nm*. The dominant energy loss is that of the rig: *3 Nm*. In the case of a gearbox, this is strongly different. Due to the large number of turning gears, the bearing and churning energy losses are important. In 1st and 2nd gear they have the same magnitude as that of the specific rig loss.

5. STUDY OF THE DOUBLE BUMP

The simulation software allows the study of the double bump as well. The double bump is a peak on the force-time curve, and it appears when the sleeve reaches the splines of the gear, after engaging the synchro ring. As the synchro ring and the gear turn together after the synchronisation, the origin of the bump can be the separation of the ring, which needs an increase of axial force. It should be noted, that the mean value of the bump depends on the type of synchro ring, and the bump is often high.

Figure 3 presents the relative frequency of double bump peak forces for a traditional synchro ring made of brass alloy, in case of gear change 1-2 and 2-1. The dispersion of the non-zero values is small around the mean values of *500-550 N* in both cases. Simulations have been made to understand these phenomena. Simulation parameter was the synchro conicity angle difference $\Delta\alpha$. Static friction coefficient is estimated to be $f_s=0,22$ both on conical surfaces and on spline side chamfer [2].

6. DISCUSSION OF RESULTS

The obtained numerical results allow to understand the dispersion. In the case of a traditional copper alloy synchro ring, dispersion is connected to the conicity angle difference of the clutch, which is the sum of the conicity angle errors of the cone and the ring.

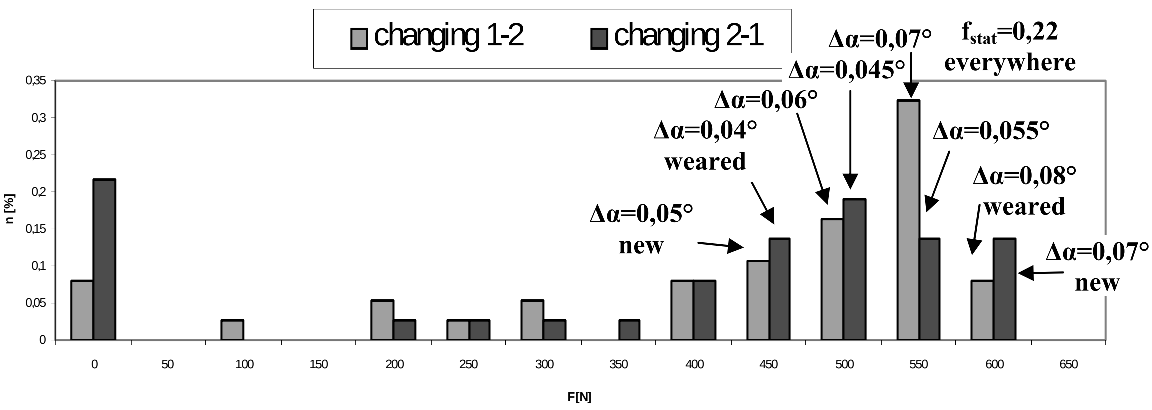

Fig. 3. Comparison of numerical results and measurement data

In figure 3, there are only a few peak values with the original, new angle difference (state new), because the surfaces start to wear in, and the angles change. Then there are many peaks with a weared angle difference, which corresponds to a state of normal wear. Finally, wear becomes excessive by the end of the tests, and this gives a smaller peak with less frequency (state weared). This process can be observed both for 1-2 and 2-1 changing. Simulation data corresponds well to this phenomenon.

This study confirms the importance of the manufacturing precision to minimise the initial angle difference as well as that of the cone wear resistance to limit angle difference variation during use.

7. ACKNOWLEDGEMENTS

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