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MODELING AND SIMULATION OF FRICTION DISC CLUTCH OPERATION IN AUTOMATIC TRANSMISSION

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ABSTRACT: *Friction disc clutches and brakes have very important role in quality of gear shifting. Thus, it is necessary that they have electric-hydraulic control and stable characteristics, which depend on a type of a friction material. Also, it is necessary that they have sufficient heat capacitance in order to be able to resist high operation temperatures during gear shifting. By application of friction disc clutches and brakes in automatic transmissions, gear shifting under load is achieved without power flow interruption. Gear shifting time is considerably reduced, driver's load is decreased and the task of gear shifting is simplified.*

One of the heaviest and the most important tasks of the system for gearbox automatic control are to provide continuos engagement of friction clutches and brakes. If this task is not successfully resolved, there are some interruptions in vehicle's operation and it makes its exploitation more difficult and causes dynamic loads of vehicle's transmission and engine.

This paper presents a mathematical model of engagement of disc clutches and brakes during gear shifting from higher to lower level, as well as the results of simulation performed by MATLAB-Simulink computer program.

KEY WORDS: *friction disc clutches, mathematical model, simulation*

1. INTRODUCTION

In the last ten years, a large number of authors have presented plenty of models for simulation of gearshift in automatic transmissions of the vehicles. Friction disc clutches and brakes are controlled during gearshift. Before corresponding model is developed, it is necessary to set differential equations of clutches behavior during their lock and unlock (stick-slip effect).

2. MATHEMATICAL MODEL OF FRICTION ASSEMBLY ENGAGEMENT

The friction assembly moment of friction (figure 1) is determined according to expression [2], [3], [5]:

$$M_{spoj} = \int_{R_1}^{R_2} 2 \cdot \pi \cdot \mu \cdot n \cdot p \cdot R^2 \cdot dR = R_c \cdot F \cdot \mu \cdot n, \qquad (1)$$

where is:

F [N] - total activating force of the friction assembly,

p [Pa] -pressure between contact surfaces of the friction assembly,

μ [-] - friction coefficient,
R [m] - current radius of the operating fluid position,
n [-] - the number of contact surfaces.

Total activating force of the friction assembly, realized as in figure 1, is defined by the following expression:

$$F = F_{st} + F_{cf} - F_{op}\,, \tag{2}$$

where is:

F_{st} [N] - the force acting on a piston due to static oil pressure;
F_{cf} [N] - the force acting on a piston due to centrifugal oil pressure;
F_{op} [N] - the force acting on a piston of the system of operating rod springs.

The force acting on a piston due to static oil pressure is:

$$F_{st} = p_{st} \cdot A_{sp}\,. \tag{3}$$

It is recommended that the static oil pressure, p_{st}, in the operating cylinder, varies from 5 to 15 [bar] [2], [3]. If this pressure was greater then 15 [bar], difficulties in sealing the hydraulic system would occur.

The total contact friction surface is calculated by:

$$A_{sp} = (\,R_2^{\,2} - R_1^{\,2}\,) \cdot \pi \cdot n\,, \tag{4}$$

where:

R_2 and R_1 [m] - are external and internal radii of disc contact surfaces.

The force acting on a piston of the system of operating rod springs is obtained as a product of spring stiffness, deflection (*f*) and number of springs (z_{op}):

$$\begin{aligned} F_{op} &= c \cdot f \cdot z_{op}\,, \\ f &= f_u + h, \end{aligned} \tag{5}$$

where:

f_u [m] - is the initial deflection,
h [m] - is the operating piston motion.

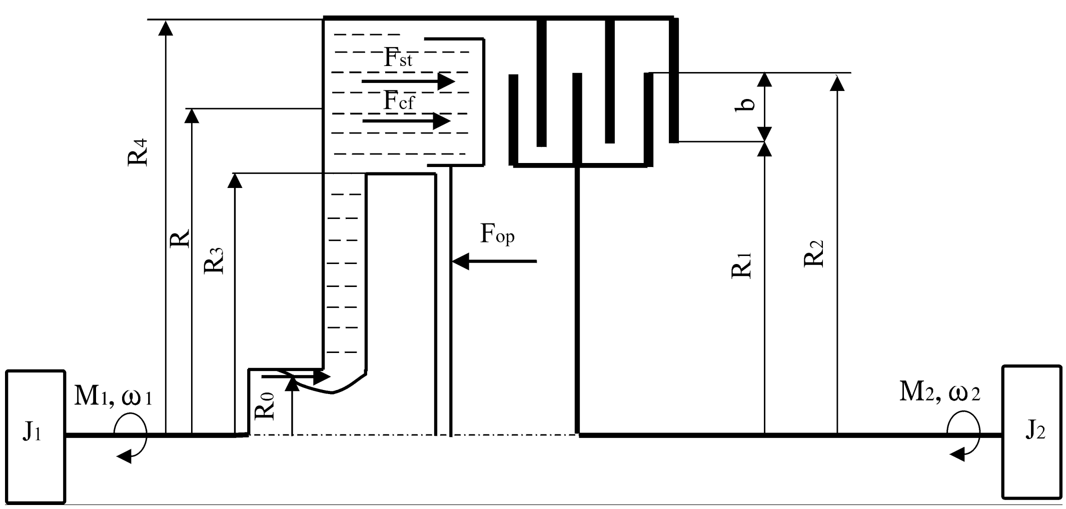

Figure 1. *Mathematical model of friction clutch*

Variation of pressure during action of centrifugal forces of the operating fluid, which appears during rotation of friction clutch's operating cylinder, is:

$$dp_c = \rho \cdot \omega^2 \cdot R \cdot dR \quad , \tag{6}$$

where is:

ρ [kg/m^3] - density of the operating fliud;

p_c [Pa] - the pressure in hydraulic cylinder due to centrifugal force of the operating fluid;

ω[s^{-1}] - angular velocity of the operating cylinder;

R [m] - current radius of the operating fluid position.

By integration of the equation (6), the value of pressure on current radius is gained:

$$p_c = \int_{R_0}^{R} \rho \cdot \omega^2 \cdot R \cdot dR = \frac{1}{2} \cdot \rho \cdot \omega^2 \cdot (R^2 - R_0^2)\,, \tag{7}$$

that is, the value of centrifugal force of the operating fluid with internal radius being R_3 and external radius being R_4:

$$F_{cf} = \int_{R_3}^{R_4} p_c \cdot 2 \cdot R \cdot \pi \cdot dR = \frac{\pi \cdot \rho \cdot \omega^2}{4} \cdot \left[R_4^4 - R_3^4 - 2 \cdot R_0^2 \cdot \left(R_4^2 - R_3^2 \right) \right]. \tag{8}$$

Since the largest radius, R_0 , is much lesser then R_4, the second term in equation (8) can be neglected, and relation (8) becomes:

$$F_{cf} = \frac{\pi \cdot \rho \cdot \omega^2}{4} \cdot \left[R_4^4 - R_3^4 \right]. \tag{9}$$

Mean friction radius (R_c) is determined based on division circle radius of the external toothing (R_4) and on division circle radius of the internal toothing (R_3) of friction elements and with help of the expression:

$$R_c = \frac{2 \cdot (R_2^3 - R_1^3)}{3 \cdot (R_2^2 - R_1^2)}. \tag{10}$$

The value of the friction coefficient, μ [-], depends on the shape of the lubrication channels, specific pressure and relative slide velocity between the contact surfaces. Its functional dependence may be graphically or analytically presented based on experimentally acquired data, equations (11) or (12) [1],[2]:

$$\begin{aligned} &\mu = 0.1316 + 0.0001748 \cdot \left| \Delta\omega_{spoj} \right| \\ &\Delta\omega_{spoj} = \omega_1 - \omega_2 \end{aligned}\,, \tag{11}$$

$$\mu(t) = 0.17 - 0.16 \cdot \overline{v}(t) + 0.16 \cdot \overline{v}(t)^2\,, \tag{12}$$

where is:

$\overline{v}(t)$ [m/s] - mean slide velocity.

2.1 ANALYSIS OF THE ENGAGEMENT PROCESS OF FRICTION ASSEMBLY

Differential equations describing the engagement process of the friction clutch (figure 1) are presented by equation [4]:

$$J_1 \cdot \dot{\omega}_1 = M_1 - b_1 \cdot \omega_1 - M_{spoj}$$
$$J_2 \cdot \dot{\omega}_2 = M_{spoj} - b_2 \cdot \omega_2 \qquad (13)$$

where is:

J_1 [kgm^2]- inertial moment of rotating masses reduced to input shaft,
J_2 [kgm^2]- inertial moment of rotating masses reduced to output shaft,
M_1 [Nm]- engine torque,
M_2 [Nm]- total moment of resistance to motion,
b_1 [Nm/rad/s] - damping coefficient of the input shaft,
b_2 [Nm/rad/s] - damping coefficient of the output shaft.

During the sliding phase, maximal friction moment transmitted by the friction clutch is:

$$M_{spoj\,max} = R_c \cdot F \cdot \mu \cdot n\,, \qquad (14)$$

that is:

$$M_{spoj} = sgn(\,\omega_1 - \omega_2\,)M_{spoj\,max}\,.$$

At the moment the clutch is engaged, balance of angular rotation speeds of driving and driven discs occurs, that is $\omega_1 = \omega_2 = \omega$, so, equation (13) is transformed into:

$$(\,J_1 + J_2\,)\,\dot{\omega} \cdot = M_1 - (\,b_1 + b_2\,) \cdot \omega\,. \qquad (15)$$

By solving the equations (13) and (15), expression for calculation of friction assembly moment of friction is obtained for the moment when the clutch is engaged:

$$M_{spoj} = \frac{J_2 \cdot M_1 - (\,J_2 \cdot b_2 - J_1 \cdot b_1\,) \cdot \omega}{J_1 + J_2}\,. \qquad (16)$$

2.2 ANALYSIS OF THE RESULTS OF SIMULATION OF FRICTION ASSEMBLY ENGAGEMENT

The case when the clutch S_3 is engaged is observed during simulation of the engagement process of a friction clutch as a subsystem of domestic design automatic transmission (figure 2 - a). Values of geometrical parameters are chosen from constructive documentation [4] (table 1), and values of the friction coefficient and inertial moments are taken from references.

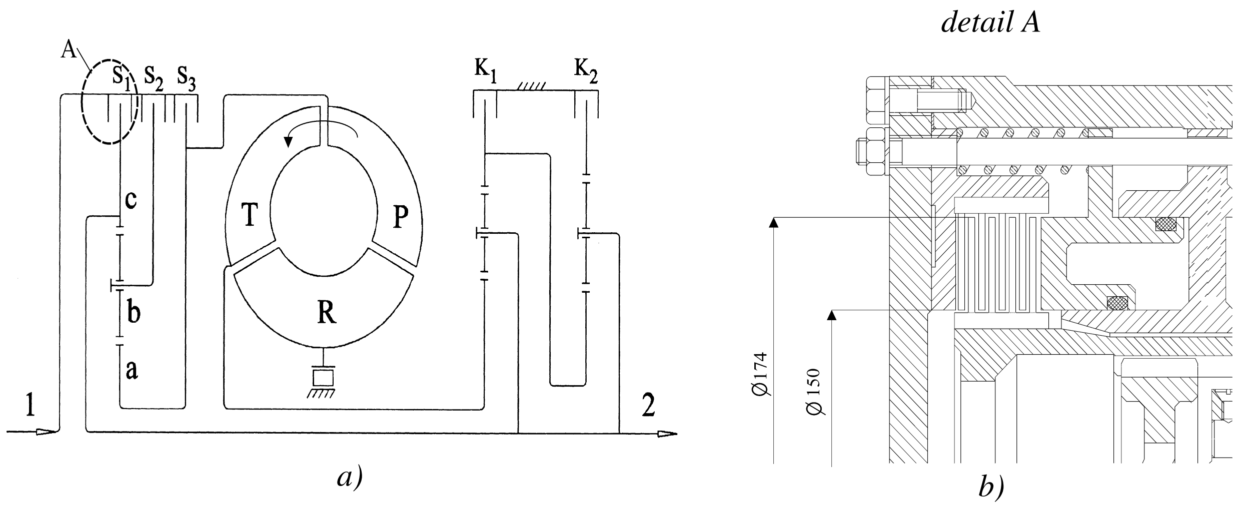

Figure 2 *Kinematic scheme of the automatic transmission of a city bus - a) and appearance of one friction clutch - b)*

Table 1 *Data used in calculation of disc clutches*

Parameters used in calculations	
R_1 [m]	0.15
R_2 [m]	0.174
μ_s [-]	0.173
μ_d [-]	0.1331
n [-]	4
J_1 [kgm^2]	0.012
J_2 [kgm^2]	0.012
b_1 [Nm/rad/s]	0.2
b_2 [Nm/rad/s]	0.2

Numbers of revolutions of driving and driven discs of the friction clutch S_3 during its engagement are shown in figure 3, while a diagram of friction moment, transmitted by the clutch, are shown in figure 4. Results are obtained by simulation of friction clutch operation using MatLab Simulink, with electromagnetic valve being hydraulically controlled. The cases of engagement of remaining clutches and brakes are not discussed here, because all other clutches of the automatic gear, shown in figure 2, have the same dimensions.

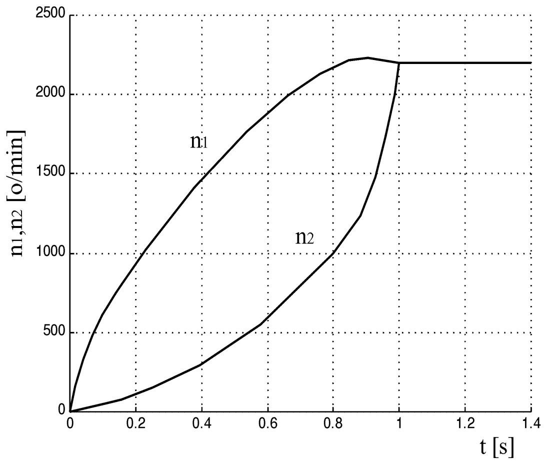

Figure 3. *Dependence between number of revolutions of driving and driven clutch discs obtained by simulation program*

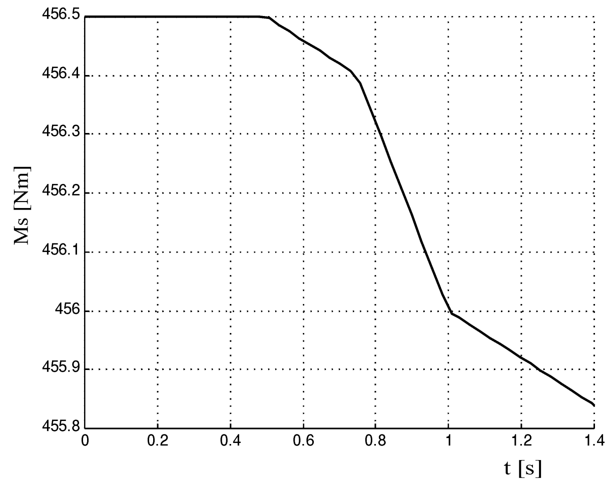

Figure 4. *Dependence of friction moment of the friction clutch*

3. CONCLUSIONS

Application of friction clutches and brakes in automatic transmissions as devices for gear change enables minimal duration of transient process, reduction of dynamic loads of elements and of reduction of driver effort. At the same time, they considerably simplify the automation task and enable gear change under load.

Mathematical model of the friction clutch engagement in automatic gear is presented in this paper, as well as the results obtained by simulation in Matlab Simulink of operation under hydraulic control.

4. REFERENCES

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