THE USE OF SIMULATION MODELS FOR THE DESIGN OF FRICTION SHIFTING ELEMENTS USED IN PLANETARY TRANSMISSIONS

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ZHRNUTIE

Otázka projektovania a navrhovania transmisíí nadobúda pri súčasnom využívaní softvérových prostriedkov nové rozmery. Výpočtový systém Matlab ponúka široké možnosti pri vytváraní a skúmaní nových riešení v prostredí matematických modelov. Jeho súčasťou je aj modul SimDriveline, ktorý nadobúda význam hlavne čo sa týka zjednodušenia zostavenia modelu transmisie, vytvárania vzťahov medzi objektmi modelu, názornosti, prehľadnosti, prezentácie výsledkov a pod. V tomto prípade sa však jedná o simulačné modely fyzikálne, ktoré umožňujú modelovať správanie sa systému priamo popisom jeho fyzikálnej štruktúry. Takéto simulačné modely nachádzajú využitie pri návrhoch rôznych častí pohonov, napríklad i radiacích prvkov planétových prevodoviek – spojok a brzd. Pomocou simulácii možno sledovať priebehy základných veličín, teda aj správanie sa jednotlivých členov mechanizmu, čo umožňuje značnou mierou prispieť k optimálnemu riešeniu transmisie.

KLÚČOVÉ SLOVÁ: MATLAB, SIMDRIVELINE, SIMULÁCIE PREVODOVIEK, PLANÉTOVÉ PREVODY, TRANSMISIA, POHON

ABSTRACT

Software tools bring a new dimension to the designing and proposing of transmissions. The Matlab computing system provides a great opportunity to develop and explore new solutions in the area of mathematical models. It also includes the SimDriveline module, which is important especially in terms of simplification of transmission model creation, the creating of model object relationships, clearness, clarity, results presentation etc. In this case, it's a physical simulation model that allows the modeling of the behavior of the system directly by a description of the physical structure. These simulation models are useful for designing various driveline parts, as well as planetary transmission shifting elements – clutches and brakes. Using the simulation enables monitoring of the variation of basic variables, as well as the behavior of the individual mechanism members. This provides a significant contribution to the optimal solution of a transmission.

KEYWORDS: MATLAB, SIMDRIVELINE, SIMULATION OF TRANSMISSION, PLANETARY GEARS, TRANSMISSION, DRIVELINE

1. INTRODUCTION

Transmission calculation is in most cases focused on determining the dynamic load and carrying capacity, or the strain of friction shifting elements which shift individual gears and it's shifting under load. Evaluation of carrying capacity of friction elements is particularly important in transmissions with high power, where the friction shifting elements usually crucially affect the whole transmission reliability. Most complex in terms of calculation, given their diversity and complexity, are transmissions based on a planetary mechanism [1].

Planetary transmissions have gears permanently meshed, and gear changes are carried out by friction elements. The gear shift process happens under load and it can easily be mechanized and automated [1,2].

2. THE GEAR SHIFTING SEQUENCE

2.1 MATHEMATICAL DESCRIPTION OF THE FRICTION ELEMENT ACTIVATING PROCESS

The engagement (activating) process of a friction element may according to [2] be explained using the simple mechanical system in Figure 1 after adoption of the following simplifications:

- at time $t = 0$: $M_1 = \text{constant}$, $\omega_1 = \text{constant}$ and $\omega_2 = 0$
- during engagement (run-up): $M_T = M_S = \text{constant}$, $f = \text{constant}$, $M_2 = \text{constant}$
- during engagement the progress of angular velocities is linear

FIGURE 1: Mechanical system with friction member – friction clutch; M_1 and M_2 – torque of driving and driven part, $M_S = M_T$ – friction torque of clutch, I_1 and I_2 – inertia of driving and driven part, ω_1 and ω_2 – angular velocity of driving and driven shaft.

OBRÁZOK 1: Mechanická sústava s trecím členom – trecou spojkou; M_1 a M_2 – moment hnacej a hnanej časti, $M_S = M_T$ – trecí moment spojky, I_1 a I_2 – hmotný moment zotrvačnosti hnacej a hnanej strany, ω_1 a ω_2 – uhlová rýchlosť hnacieho a hnaneého hriadeľa.

According to [2,6] during friction member engagement (in this case a friction clutch) inertia I_1 will decelerate and inertia I_2 will accelerate. Both inertias achieve the same angular velocity ω_{12} at point B (Figure 2), and it occurs when the friction torque of clutch M_S is higher than M_2 . Friction torque will accrete after engagement of the clutch in a short time to the value $M_S = M_2$, and it's assumed, that friction torque of the clutch will be constant during the next engagement process and that speed change will be linear.

FIGURE 2: Values during the engagement process of friction element – friction clutch.

OBRÁZOK 2: Pribeh veličín počas zopínania trecieho člena – trecej spojky.

During clutch slipping the system has two degrees of freedom and kinetic equations have the following form:

$$M_1 + I_1 \cdot \dot{\omega}_1 - M_T = 0 \quad (1)$$

$$M_T - M_2 - I_2 \cdot \dot{\omega}_2 = 0 \quad (2)$$

Angular accelerations of shafts by linear speed progresses will be constant, therefore:

$$\frac{M_T - M_1}{I_1} = \dot{\omega}_1 = \varepsilon_1 = \text{constant} \quad (3)$$

$$\frac{M_T - M_2}{I_2} = \dot{\omega}_2 = \varepsilon_2 = \text{constant} \quad (4)$$

while both shaft speeds of at point B will be the same the following applies:

$$\omega_{12} = \omega_1 - \varepsilon_1 \cdot t_B = \omega_2 + \varepsilon_2 \cdot t_B \quad (5)$$

where t_B is the time from the beginning of clutch engagement until the end of this process at point B. Time t_B may be determined as follows:

$$t_B = \frac{\omega_1 - \omega_2}{\varepsilon_1 - \varepsilon_2} \quad (6)$$

From equations (3) and (4) it follows that:

$$\varepsilon_1 + \varepsilon_2 = \frac{(M_T - M_1) \cdot I_2 + (M_T - M_2) \cdot I_1}{I_1 \cdot I_2} \quad (7)$$

The effect of contact force F results in friction surfaces pressure and friction torque, which is according to [2] defined as follows:

$$M_T = k \cdot M_S = i \cdot F \cdot R_s \cdot f \quad (8)$$

where R_s is the mean friction surface radius, f is the friction coefficient and i is the number of friction surfaces (Figure 3). For normal contact force the following applies:

$$F = p \cdot S_T \quad (9)$$

where $p \leq p_D$ is the pressure at the contact friction surfaces, S_T is the contact area. The mean friction radius depends on the pressure distribution in the frictional contact surfaces:

- if $p = \text{constant}$ (hard friction linings: cermet, hardened steel etc.):

$$R_s = \frac{1}{3} \cdot \frac{D_1^3 - D_2^3}{D_1^2 - D_2^2} \quad (10)$$

- if $p \cdot v = \text{constant}$ (soft friction linings: Ferodo etc.):

$$R_s = \frac{D_1 + D_2}{4} \quad (11)$$

FIGURE 3: Parameters of the friction member.

OBRÁZOK 3: Parametre trecieho člena.

2.2 PROGRESS OF VALUES DURING GEAR SHIFTING SEQUENCE

A mechanical drive system model with asynchronous engine and two-stage planetary transmission can be used to describe the gear shifting sequence of planetary gears (Figure 4).

FIGURE 4: Schematic drive machine with two-stage planetary transmission.

OBRÁZOK 4: Schéma pohonu stroja s dvojstupňovou planétovou prevodovkou.

Figure 5 shows a typical example of the most important parameters as they vary with time during gear shifting sequence at maximum engine power [3]. It's evident that dynamic

transients arise from the gradual engagement or disengagement of friction elements during gear shifting, from the power needed to overcome the resistance and from the power that is at that time supplied from the engine.

FIGURE 5: Example of variance of most important parameters during gear shifting sequence of a planetary transmission [3]; M – output torque of transmission, ω_e – engine speed, p – pressure in the hydraulic circuit of shifting members.

OBRÁZOK 5: Príklad priebehu sledovaných veličín pri preradzovaní planétovej prevodovky [3]; M – výstupný moment prevodovky, ω_e – otáčky motora, p – tlak v hydraulickom okruhu radiacích členov.

The main objective in designing the shifting sequence is approximation of the ideal pattern of driving torque and thus reducing undesirable jerk on the output shaft. The plots of pressures should intersect at the moment when one element engages and a second disengages, so that the step change of drive torque is minimal. The hydraulic circuit layout crucially affects the progress of pressures.

The recommended increase (decrease) nature of pressure in engaging (disengaging) the friction component is shown in Figure 6 [1]. This characteristic plot was obtained from the analysis of pressure changes in automatic transmissions. Characteristic points in the engaging component are marked Z_0 to Z_8 and in disengaging component are marked V_0 to V_2 . The coordinates of each point Z or V are defined by gear shifting time t_{Z0} to t_{Z8} , respectively t_{V0} to t_{V2} and by pressure p_{Z0} to p_{Z8} , respectively p_{V0} to p_{V2} .

FIGURE 6: Variation of pressure in engaging and disengaging friction component [1]. Legend:

t – time from the beginning of engagement sequence
 t_{zk} – end of engagement sequence
 t_{z0}, t_{v0} – issuing of electrical signal to turn on of the hydraulic distributor of engaging, disengaging component
 $t_{z1} - t_{z0}$ – reaction time of the hydraulic distributor
 $t_{z2} - t_{z1}$ – filling of the supply pipe
 $t_{z3} - t_{z2}$ – clearance adjustment between plates
 $t_{z4} - t_{z3}$ – exponential increase of the compressive force
 $t_{z7} - t_{z4}$ – shifting pressure modulation (for initial proposals it can be linear modulation, which is marked dashed between Z_4 and Z_7)
 $t_{z8} - t_{z7}$ – increase of working pressure to the maximum value
 $t_{v1} - t_{v0}$ – reaction time of the hydraulic distributor
 $t_{v2} - t_{v1}$ – controlled pressure drop in disengaging component
 Z_4 – starting point of the pressure modulation
 Z_m – point of friction component fully engaged
 Z_7 – endpoint of the pressure modulation

OBRAZOK 6: Priebeh tlaku v zaraďovanom a vyraďovanom trecom prvku [1].

From the variation of control pressures according to [1] there result three characteristic phases of engaging component activity:

The preparatory phase lasts from issuing of electrical signal to turn on of the hydraulic distributor of engaging component to initial control pressure increase. Thus, the preparatory phase is $t_I = t_{z3} - t_{z0}$.

The slipping phase is characterized by the transfer of torque, but there is a slip between the engaging component plates. At point Z_m occurs full engagement of the friction member, i.e. the slip ends between the plates of this element. Time of this phase is $t_{II} = t_{zm} - t_{z3}$.

The final phase is characterized by the completion of the increase shifting pressure to its maximum value. At this stage is required transferred torque intended by the value of external shaft torques, which are dependent on the load torque. The final phase lasts until the point Z_k and duration of the final phase is $t_{III} = t_{zk} - t_{zm}$.

3. USE OF SIMULATION MODELS

3.1 SOFTWARE SIMDRIVELINE

Design of gear shifting sequences by experimental procedures can sometimes be rather inefficient, difficult and demanding. In contrast, simulation models provide options that can't be obtained by experimental methods [3].

SimDriveline is an extension of Matlab and extends its capabilities to include tools for modeling and simulation of car and machinery powertrains [5,7]. Since blocks used to create a model directly describing the properties of specific transmission parts, it's not necessary to mathematically formulate the properties of the objects or relationships between themselves etc. In this environment all necessary input data and parameters can be entered directly into the block properties. Similarly, the relationship between blocks isn't defined as a method of signal transmission, but as a mechanical bond.

3.2 PHYSICAL SIMULATION MODEL OF DESCRIBED DRIVELINE

The basis of the simulation model (Figure 7) is a subsystem of planetary transmission (Figure 8). It is defined by its individual parts and their parameters, their involvement and relationships both mutual and between other parts of the model. The engine overcomes passive resistance torque and torque of dynamic effects of the system. For the simplification of the model, the effects of inertia are reduced to the load. Transmission brakes (gear shifting members) are controlled by signals from the control unit.

FIGURE 7: The complete driveline simulation model with described planetary transmission. The model consists of engine, transmission, load and control unit subsystems.

OBRAZOK 7: Celkový simulačný model pohonu s uvažovanou planétovou prevodovkou. Model pozostáva zo subsystému motora, prevodovky, záťaže a riadenia.

FIGURE 8: The simulation model of the transmission subsystem.
OBRÁZOK 8: Simulačný model subsystému prevodovky.

Control signals in this model can take values of from 0 to 1 and in this case are equivalent to real transmission control pressures. A value of 0 corresponds to 0 MPa, a value of 1 corresponds to maximum pressure in the hydraulic circuit [5].

3.3 SIMULATIONS OF GEAR SHIFTING SEQUENCE

Plots of monitored values and friction elements control signals obtained from simulations using the software SimDriveline are shown in Figures 9 and 10. In this case real variations during engagement and disengagement are considered.

The graphs plot 1/10 of torque values (M_A and M_X) and angular velocities (ω_A and ω_X) on the input and output shaft. Gear ratio i_{AX} is determined by the ratio of angular velocities. Control signals correspond to controlling friction elements pressures and for illustration are displayed as 15-fold the actual values.

The first simulation (Figure 9) is for determining torque values, angular shaft velocities and resulting gear ratios during constant gear shifting sequence from first to second gear and from second to first gear. Simulations were performed for varied transmission loads; the inertias of all members of the driveline system were constant. The system was powered by an asynchronous engine with starting torque 173 Nm and peak torque of 200 Nm.

Plots of the monitored variables obtained during simulations at constant load and gradually decreasing shifting times are shown in Figure 10. The simulation provides torque values, angular shaft

FIGURE 9: Simulation at varying loads from -100 Nm to -200 Nm and constant gear shifting sequence from first to second and from second to first gear.

OBRÁZOK 9: Simulácia pri meniaci sa záťaži od -100 Nm do -200 Nm a nemennej sekvencii radenia pri preradzovaní z prvého na druhý a z druhého na prvý rýchlostný stupeň.

velocities and resulting gear ratios during different gear shifting sequences. Simulations were performed at constant loads of transmission; the inertias of all members of the driveline system were constant. The system was powered by the same engine as in the previous simulation.

3.4 REAL GEAR SHIFTING SEQUENCE

There remains an important fact that any simulation or mathematical model only describes a simplified system and its behavior over time, and doesn't consider all variables and values that affect the described action [4]. Since plots of variables are also an output of simulations, their real values may vary. It's therefore appropriate to compare values obtained from simulations with values obtained experimentally, at least for calibration and tuning of the simulation model.

Actual control pressure measurements taken during gear shifting of a terrain vehicle from second to third gear, and also from third to second gear are shown in Figure 11 [1]. The illustrations describe the variation of engine speed n_E and control pressures p_2 and p_3 in the friction element with time.

4. CONCLUSION

Simulations can be used to obtain valuable information for design and dimensioning, and together with experimentation it is a basic method for obtaining input parameters in the design process. From performed simulations it's possible to design the basic parameters and dimensions of the friction members – in this case the brakes and basic parameters of hydraulic control circuit. During the compilation of a simulation model in SimDriveline, the benefits of this environment were recognized with respect to the simplicity of model creation.

From the first simulation it's clear that during a gear shifting process with a planetary gear with increasing load under the same conditions and parameters, the slip also increases. This arises due to the mutual synchronization of shaft speeds, and if the same friction member carries a higher load, a higher friction force in the friction plates must be applied. From the above simulation can be determined the load torque at which the acting friction force is already inadequate, and then friction member parameters, hydraulic control circuit etc. can be dimensioned. From the second simulation is clear that the length of gear shifting sequence affects the slip. Slip is greater when disengaging time

FIGURE 10: Simulation at constant load -175 Nm and various gear shifting sequences from first to second and from second to first gear. In this simulation there occurs a gradual shortening of friction element engaging time.

OBRAZOK 10: Simulácia pri meniaci sa sekvencii radenia a konštantnej záťaži -175 Nm pri preradzovaní z prvého na druhý a z druhého na prvý rýchlostný stupeň. Pri tejto simulácii dochádza k postupnému skracovaniu času zopínania tretieho prvku.

of the first and engaging time of the second friction member is longer. From this it is clear that the interval must be as short as possible. This requirement is in real terms limited resulting from the inertia of friction mechanism components, dynamic properties of hydraulic fluid, capabilities of individual members and hydraulic circuit etc. From this simulation can be obtained valuable information for the designing of hydraulic circuit dynamics and various parts of friction members.

The advantage of simulation models is that they allow for different conditions in a simple way to monitor the behavior of a transmission in any state. Simple change of input parameters without necessary changes in the overall model creates opportunities to find the optimal drive solution, but also for tuning the mathematical model.

FIGURE 11: Variation of control pressures and engine speed with time of gear shifting in a terrain vehicle transmission.

OBRÁZOK 11: Priebehy riadiacích tlakov a otáčok motora v závislosti od času radenia, pri radení v prevodovke terénneho vozidla.

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