Estimation of the Synchronization Time of a Transmission System through Multi Body Dynamic Analysis

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Abstract—**The essential component in transmission system is synchronizer. Synchronizer developed to obtain gear changing smoothly. Reducing the transmission time will increase the efficiency of the transmission system and minimize the energy loss during the shifting process. In order to achieve the optimized design, the time estimation for synchronizing process is necessary. In this present study, the multi body dynamic model is proposed to predict the synchronization time. For validation of the results two different synchronizer types, single cone and double cone were used in the test rig machine under different loading conditions. The results of multi body dynamic analysis were compared to experimental and analytical results and show that there is a good agreement between simulation and experimental results. Using the multi body dynamic analysis makes more accurate result to predict the synchronization dynamic behavior, especially synchronization time.**

Index Terms—**transmission system, synchronizer modeling, multi body dynamic, time estimation, rigid body motion**

I. Introduction

Energy and fuel consumption are important issues in the automotive industry. In order to satisfy the ecological regulation and to produce an environmental friendly product, car manufacturers are willing to design optimized vehicles [1]. Regarding power and fuel economy, the transmission system is one of the main effective parts of the vehicle [2].

The essential component of the transmission systems is the synchronizer. The synchronizer has to be designed in order to obtain smooth gear changes as well as reduced noise and vibration [3]. However, ease of transmission and comfort are further synchronizer tasks at which recently has been paid attention [4]. Several studies have been carried out in order to increase the shifting quality. Different geometry, materials and lubrication conditions were considered as a solution to improve the shifting performance [5]-[7].

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To better understand the synchronization process, several mathematical and analytical models have been proposed [8], [9]. Moreover, different test rigs were utilized to characterize the significant parameters during the shifting process. In order to evaluate the effect of different forces and drag torques, some mathematical models were developed with different approaches, the main difference between these models being the subdivision of the synchronization process into different phases. Lovas et al. have divided the synchronization process into eight different phases with a detailed analytical formulation of the dynamic equation of each phase. Razaki proposed an analytical formulation in order to identify the design parameters in five different steps: the research was focused on the dynamic behavior of involved components at every step [10], [11].

The implementation of a computational model helps to characterize the effect of most effective parameters such as the friction coefficient and the thermal and lubrication conditions [3], [12]. Although, when 2D finite element models are used, the model simplification so far introduced reduces the model accuracy while the synchronizer is subjected to rotational forces.

A 3D Multi Body Dynamic (MBD) model was used to simulate a heavy-duty synchronizer with different shifting speeds. The author has reported an overall behavior of the synchronizer in different conditions [13].

Shifting time is one of the most important factors that influence the transmission efficiency. The shortest possible shifting time yields to minimize the torque and energy loss [14]. To calculate the shifting time some analytical formulations were proposed [2], [14], while minor attention was paid to computational models to estimate this parameter. However, although, many studies have been done on the synchronization process, the time estimation with numerical method has been neglected [13].

The present paper shows the results obtained by implementing a 3D MBD model for the estimation of the synchronization time for two different types of synchronizers. The experimental data extracted from the

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test rig and the analytical results were used to validate the numerical approach.

II. Synchronizer Mechanism

Generally, synchronizers include sleeve, hub, three strut detents, two synchronizer rings, two friction cones, and two clutch body rings. In order to increase the output torque and to avoid design space limitations, some manufacturers prefer to use more friction cones. Double and triple cone synchronizers are the most common multi-cone synchronizers. Fig. 1(a) and 1(b) illustrate the different components of a single cone and a double cone synchronizer in the exploded view.

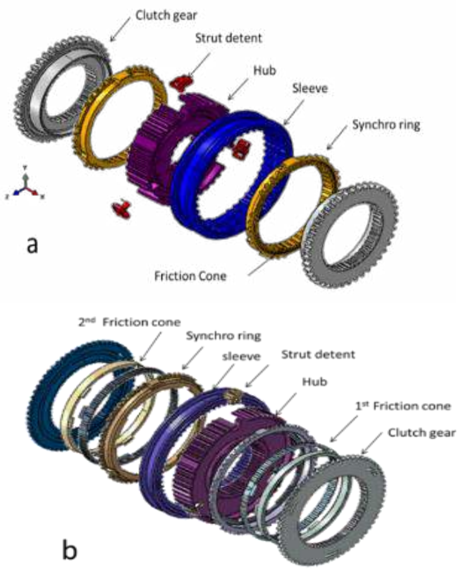

Figure 1. The synchronizer components presented in an exploded view.

At the beginning of the process, the input and output shafts rotate with their specific rotational velocity. At this time, hub and sleeve have the same rotational velocity. When the actuator is activated, the fork moves toward the sleeve axially. In this phase, the springs inside of strut detents are compressed and the strut detents move along hub grooves to the synchronizer ring. Due to the axial motion of the fork, the axial load is increased causing the engagement between the sleeve and the synchronizer ring teeth. The maximum axial force is reached at this step due to the contact between the synchronizer ring cone and the clutch body cone. This phase is called synchronization phase. In order to test the double cone synchronizer, the synchronization phase is occurred between the synchronizer ring, the first friction cone, and the second friction cone. After getting the same rotational velocity between the synchronizer ring and the clutch body cone, the final meshing of the sleeve and the clutch body can be achieved.

III. Methodology

A. Analytical Time Estimation

To estimate the synchronization time, a prediction model was rearranged from the general form of the torque formulation as shown in (1) [2].

$$t_{synch} = \frac{I_R \Delta\omega \sin\alpha}{F \mu_c R_C} \tag{1}$$

where IR is the equivalent rotational inertial, Δω is the difference of angular velocity between input and output shafts, F is the applied axial load, Rc and α are the mean radius and the angle of the cone, respectively.

The synchronization time depends on the friction coefficient, the moment of inertia, the angular velocity, and the applied force on the fork. To evaluate the synchronization time, data from experimental tests were used. In order to reach more accurate result, the applied force was considered as a time dependent parameter. Moreover, in order to calculate the double cone synchronization time, the value of the friction coefficient, the mean radius, and the angle of the cone were considered as the average values between the first and the second friction cones.

B. Numerical Model

A rigid multi body dynamic analysis is appropriate when the overall behavior of the system is concerned [16], [17]. In order to analyze the multi body dynamic behavior of the synchronizer, the ABAQUS 6.14 commercial code was used. All the synchronizer parts were assumed as rigid bodies and the S4R rigid shell element was used to simulate the single and double cone synchronizers. Fig. 2, indicates the applied boundary conditions on the single cone model. Where U and UR demonstrate the axial and the rotational displacement respectively. The hub and clutch body gear are free to rotate around axis 1. The sleeve and synchronizer ring can move axially and rotates around axis 1. Boundary conditions for the double cone are the same as the single cone and only operational conditions are different. The contact between parts was simulated by implementing the surface-to-surface contact option. Different friction coefficients were applied to the different contact areas. The equivalent inertia was assigned to the gear clutch reference point. The sleeve reference point was subjected to ramp axial load in 200 ms. In order to provide the rotational degree of freedom for the model, the connector library of ABAQUS was utilized [15]. The cylindrical connector was used to create axial and rotational motions simultaneously and the hinge connector was used for rotational motion.

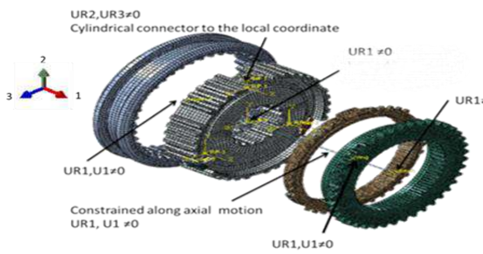

Figure 2. Applied boundary condition and connector element for the synchronizer model.

IV. Experimental Data

Experimental tests were carried out in order to characterize the synchronizer parameters. In this study, two different types of synchronizer, namely, a single cone synchronizer (SC-74) and a double cone synchronizer (DC-170) were installed in the test rig. Fig. 3, shows the schematic of the synchronizer test rig.

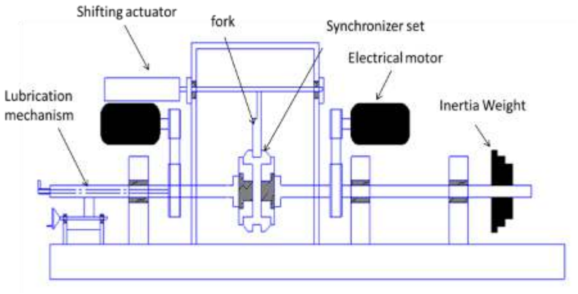

Figure 3. The schematic of the synchronizer test rig.

TABLE I. CHARACTERISTICS OF THE SYNCHRONIZATION PROCESS

characteristic	unit	Code	
		SC-74	**DC-170**
Cone diameter	mm	74	165/170
Angular velocity	rpm	1000/2000	300/900
$\Delta\omega$	rpm	1000	600
μ	--	0.06	0.1
Applied inertia	$kg.m^2$	0.17	0.9
Applied force	N	1400	1500
t_{app}	s	0.2	0.2

Figure 4. (a) single cone and (b) double cone synchronizers used for testing.

In order to test the single cone synchronizer with a 74 mm cone diameter according to the application condition, the rotational velocity of the input shaft was set to 1000 rpm and the rotational velocity of the output shaft was set to 2000 rpm. The mean applied axial force to the fork was 1400 N and the applied inertia was 0.17 kgm2. The 170 mm diameter double cone synchronizer was tested with a 300 rpm rotational velocity at the input shaft and a 900 rpm rotational velocity at the output shaft. The mean applied force on the sleeve, fork was 1500 N, and the applied inertia was 0.9 kgm2. Two different electric motors are attached to the input and output shafts to provide different rotational speeds. An actuator is connected to the shifting fork and is equipped with a load cell to measure the shifting axial load. Based on the testing condition, the appropriate inertia weights were attached to the output shaft for providing the allowable rotational inertia. Through the input shaft, the splash lubrication mechanism can be attached to the synchronizer sample. In order to control wear production and vibrational effect before and after the test the backlash distance is measure. In this study, the nominal distance of the backlash was used in the numerical simulation. Fig. 4, (a) and (b) show the single cone and double cone synchronizers that were installed between the two shafts of the test rig. During the test the required data, e.g. the dynamic friction coefficient, the synchronization time, and the sleeve stroke, are extracted. Table I indicates the dynamic characteristics of the SC-74 and the DC-170 synchronizers.

V. RESULTS AND DISCUSSION

Data from experiments have been introduced into equation 1 in order to calculate the synchronization time of 595 and 310 ms for single cone and double cone, respectively. The estimated error between analytical and experimental results for SC-74 and the DC-170 were 12% and 14% respectively.

In order to validate the results of MBD analysis through experimental data, the angular velocity of the input and output shafts for the SC-74 synchronizer are compared in Fig. 5. Given an axial force applied to the sleeve, after 670 ms the difference between the input and output shaft velocities becomes null. Due to the friction between the cone and the clutch body, the input and output shafts get the same velocity and, after passing the synchronizing time, the whole torque transfers to the output shaft. The absolute error for the numerical solution of SC-74 was 1.8 %.

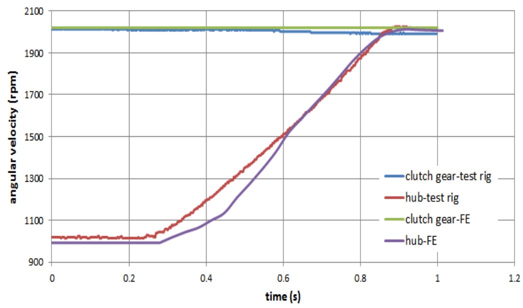

Figure 5. The single cone synchronization time estimation.

In addition, experimental and MBD results of angular velocity for the input and output shafts of the DC-170 synchronizer are shown in Fig.6. The larger inertia linked to the output shaft makes it necessary to apply a larger torque in order to reach the same velocity. With the double cone synchronizer, the input shaft was synchronized with the output velocity after 275 ms with 2% error in compare with the experimental result.

The analysis of the experimental results shows that the increased number of friction cones introduces some velocity fluctuations during the synchronization process. The increased number of components and the effect of different friction materials can be a reason of this phenomenon. Moreover, the transient dynamic behavior between the synchronizer ring, the first friction cone and the second friction cone could introduce some transient effect for the double cone synchronizer. By increasing the number of cones, the output torque and the shifting time can be improved but a possible transient dynamic effect can be introduced.

In this study, the deformation of the elements was neglected and only the overall dynamic analysis of synchronization process was analyzed. In order to verify the numerical solution, the dissipated energy balance has been evaluated: the evaluation of the kinetic, the internal and the total energy shows that the total energy is less than 2% of the internal energy. The rigid MBD results can be used for the investigation of the overall dynamic behavior of the synchronizer. The analysis of the numerical results shows some fluctuations of the angular velocity related to the damping effect of the rigid element.

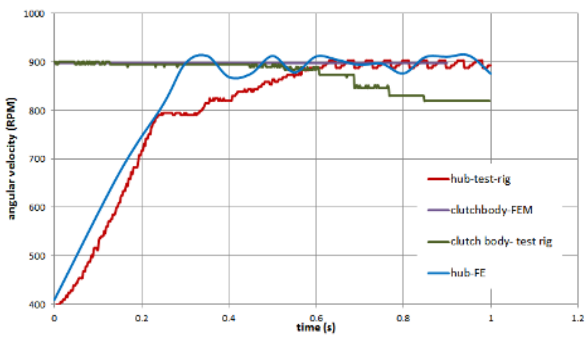

Figure 6. The double cone synchronization time estimation.

The comparison between analytical, numerical and experimental results shows that using MBD analysis the precision of estimated synchronization time can be increased. To simplify the analytical solution some assumptions such as constant applied load and average friction coefficient were assumed that lead to reducing the model accuracy (Table II).

TABLE II. ESTIMATED SYNCHRONIZATION TIME

characteristic		unit	Code	
			SC-74	**DC-170**
t_s	MATH	s	0.595	0.31
	MBD		0.67	0.275
	EXP		0.69	0.27

The MBD model has a better capability to simulate the shifting process as close as to the real test condition. The results show that there is a good agreement between numerical and experimental results. It can be seen that the axial force, and applied time were almost the same but the variation of the radius and of the friction coefficient significantly affect the synchronization time. However, the role of time estimation for designing the more reliable synchronizer can be highlighted more than in the past.

VI. CONCLUSION

A multi body dynamic model of a synchronizer has been developed in order to estimate the synchronization time. Two different synchronizer geometries were used as case studies and experimental tests were conducted on a particular test rig machine. Different angular speeds, inertia, friction coefficient, and axial load were used for the two test cases and the synchronization time was calculated. The dynamic properties were extracted from the test rig and the validity of the model was verified. Moreover, the numerical results were compared with the analytical solution. The results show that there is good agreement between numerical and experimental results under different loading conditions.

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