Multi-body simulation of dog clutch engagement

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Multi-body simulace řazení zubové spojky do záběru

Abstract: *This paper presents a multi-body simulation model in Adams created for optimization of the dog clutch engagement process, especially minimization of the gearshift time regarding to the shape of the dogs. The simulation model was validated using measurements of a prototype dog clutch engagement on the inertia test bench. This dog clutch uses positive tapers of the dog sides. Dog clutch gearshift process and expected influence of the shape of the dogs on the gearshift time are also described.*

Key words: *dog clutch, multi-body simulation, gearshift, engagement, MSC Adams*

INTRODUCTION

The task of multi-body simulation of dog clutch engagement emerged during further development of the dog clutch with blocking mechanism. This newly developed gearshift mechanism was presented in the previous year. It consists of a dog clutch with positive side angles ensuring minimal circular backlash in the engaged state and a blocking mechanism holding the clutch in engaged position under the load of axial force arising on the sides of the dogs during torque transmission.

Major benefit of the dog clutch compared synchronized gearshift mechanisms should be shortening the gearshift time. Besides other factors this shortening can be reached by optimization of the dogs’ shape and size. This is possible thanks to the fact that geometry of the dogs is independent from the blocking mechanism and both can be tuned separately. As an experimental optimization this process would include manufacturing many dog clutch variants, assembling and testing. This would be time and money consuming. Therefore, our goal is to create a multi-body simulation model which would represent the real engagement process.

DOG CLUTCH WITH BLOCKING MECHANISM

Figure 1 shows the original dog clutch with blocking mechanism design. Even though this design was improved since then, this one was manufactured and used for the experimental testing described later. Also, the following description of the function principle remains valid.

The clutch consists of clutch body (1) which is fixed to the gearbox shaft (radially using internal splines and axially sitting between the pair of idler gear wheels). Gearshift is provided by sliding dog (2) which is connected to the clutch body using internal toothing and can be moved axially. Its dogs mesh with the dogs at idler gear wheels in engaged state. Movement of the sliding dog is controlled by gearshift sleeve (3) through gearshift pins (6). There is an axial backlash between these pins and the sliding dog which is important for the mechanism functionality. Gearshift stones (5) are fixed to the gearshift sleeve. Gearshift stones encircle the sliding dog through the slots and control blocking rings (4). Blocking rings are placed in the clutch body grooves and cannot rotate freely around the clutch axis of revolution due to detent pins (7). However, the internal diameter of the blocking rings is bigger than the external diameter of the grooves. As a result, the C-shaped blocking rings can shrink slightly when under pressure of the gearshift stones. Blocking rings are unshrunk in the neutral and engaged positions of the clutch and secure the sliding dog against unwanted movement. During engagement, they are shrunk by the gearshift sleeve and allow free movement of the sliding dog. The clutch does not need any changes in common gearbox design and also the gear linkage and gear selector system can stay unchanged. Further information can be found in [1] or [2].

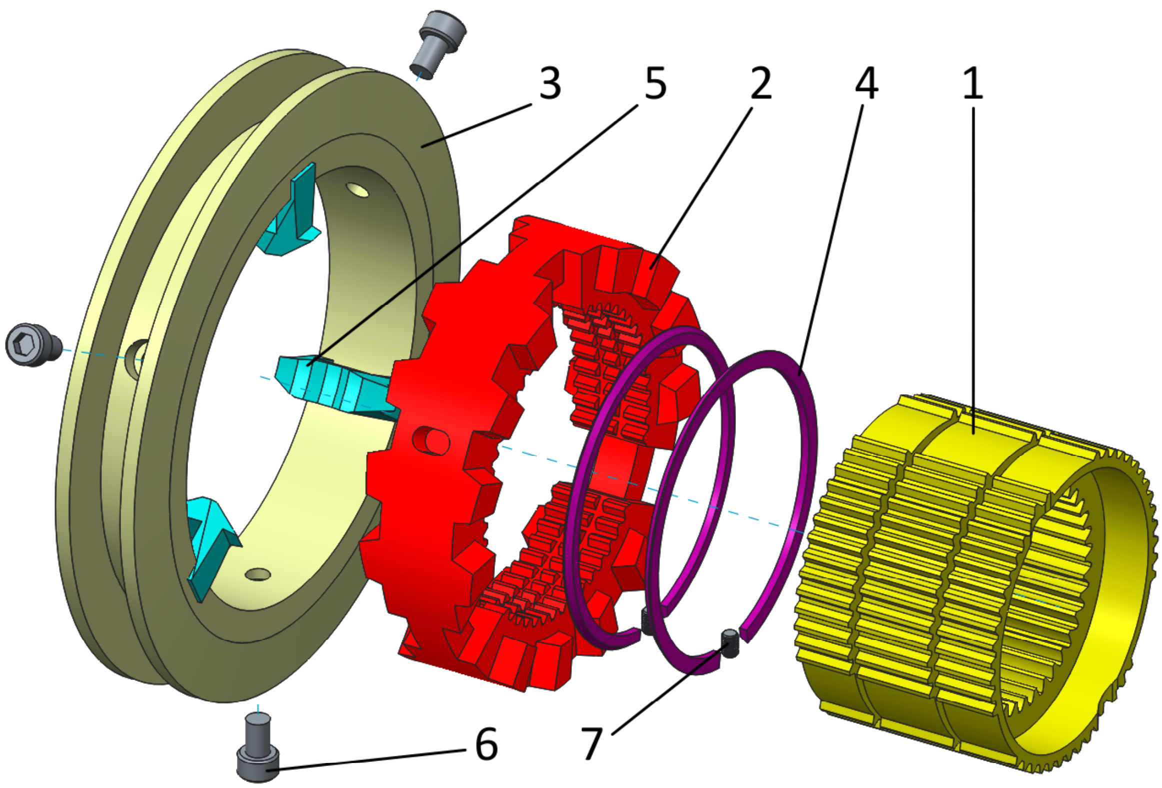

Figure 1 *Dog clutch with blocking mechanism (exploded view)*
1 – clutch body; 2 – sliding dog; 3 – gearshift sleeve; 4 – blocking rings;
5 – gearshift stones; 6 – gearshift pins; 7 – detent pins

DOG CLUTCH ENGAGEMENT

The process of basic rectangular face dog clutch engagement can be divided into multiple phases. It begins with the actuation of the sliding dog in time t_0. Initial mismatch speed of sliding dog and gear wheel has a value of $\Delta\omega_0$. The dog clutch transmits no torque, and the engagement phase is called free fly, lasting until the sliding dog reaches the gear wheel at t_1 time. Because of the possible minor changes in vehicle speed and more importantly, due to the friction losses and the eventual countershaft brake torque acting on the gear wheel, the mismatch speed changes to $\Delta\omega_1$ by that time. It is usually the dogs' faces which first come in contact, resulting in an impact.

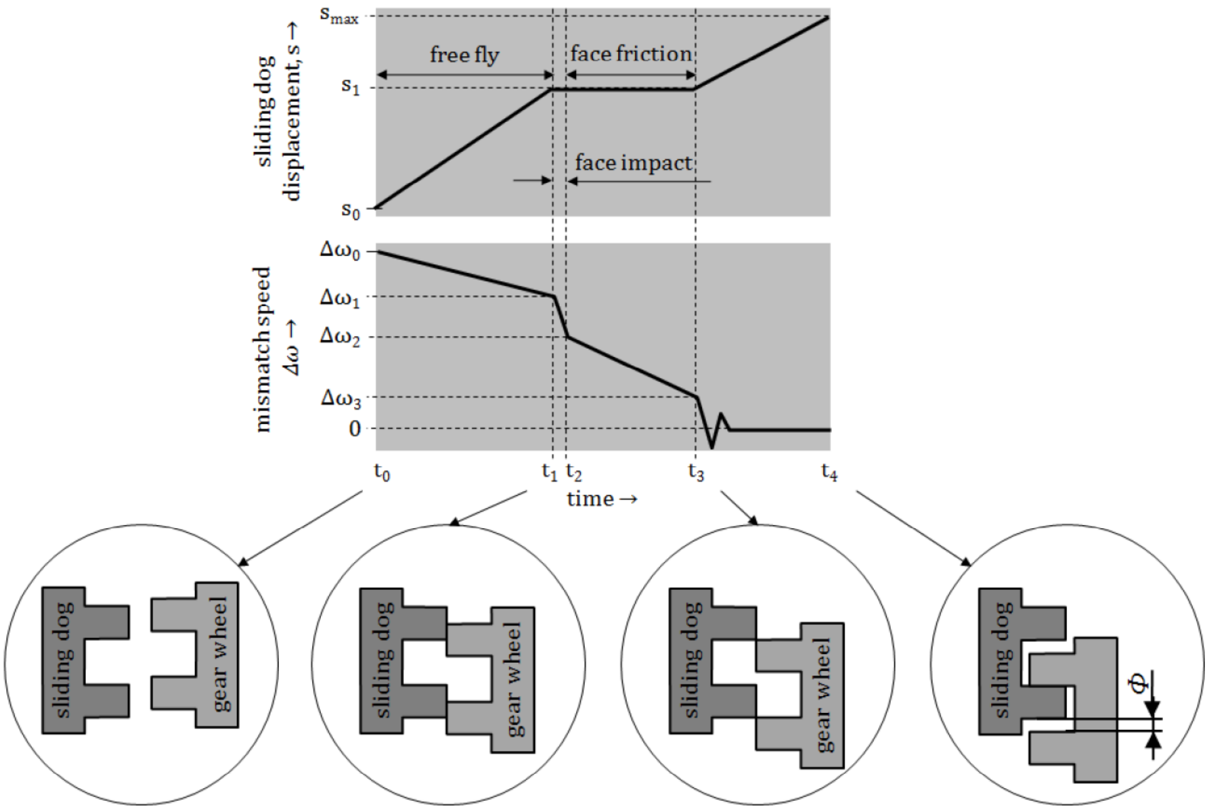

Figure 2 *Engagement process of a face dog clutch with rectangular dogs* ***[3]***

The high impact force peak stops the sliding dog by consuming the motion energy of the whole actuation mechanism attached to it. The normal force between the sliding dog and the gear wheel and the relative turning of them imply a face friction torque acting against the relative turning. As the peak in the normal force results in a peak in the face friction torque as well, the mismatch speed reduces to $\Delta\omega_2$ in the negligible time range of the face impact.

After the impact, the face areas slip on each other, until the dogs turn against the slots by the time t_3. That phase is called the face friction phase, and the face friction torque still acting against the relative turning further reduces the mismatch speed to $\Delta\omega_3$. As the dogs of the sliding dog are now free to enter the slots in the gear wheel, the dog clutch engages, and the mismatch speed quickly reduces from $\Delta\omega_3$ to zero provoking torsional vibrations damped in the driveline. Then the process becomes steady again and the gearshift is finished at time t_4, clutch is engaged; [3].

SUCCESSFUL AND UNSUCCESFUL ENGAGEMENT

Peak value of the torsional vibrations is proportional to the torsional stiffness of the gearbox and other driveline components and their moments of inertia. Another crucial factor is the mismatch speed $\Delta\omega_3$. In order to minimize the torsional vibrations and maximize the driveline service life we try to keep the value $\Delta\omega_3$ as low as possible. This also reduces the noise during gearshift. However, for low values of mismatch speed there is a risk of the engagement being unsuccessful. As the mismatch speed is reasonably reduced in the disengaged state of the dog clutch and is further decreased by the face friction torque, it may be completely vanished during the face friction phase before the engagement. The result is a permanent dog-on-dog situation, when the face contact is not resolved. The gearshift cannot be completed (Figure 3 right).

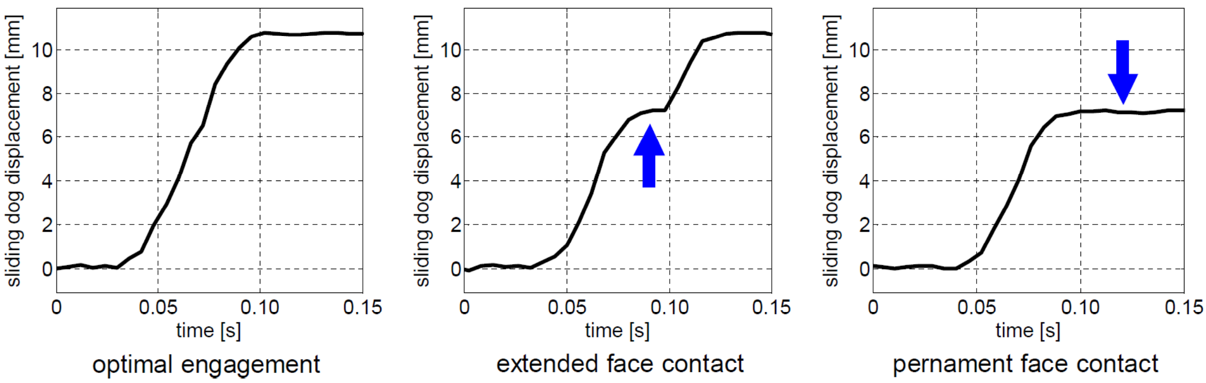

Figure 3 *Variations of sliding dog displacement during engagement* ***[4]***

The face friction phase can be skipped in special case when the dogs of the sliding dog move into the slots of the gear wheel immediately (Figure 3 left). Probability of this phenomenon depends on the number of dogs and the width ratio of dogs and slots. This ratio can be described using the angular backlash ϕ measured in the distance s_1 of the sliding dog axial displacement (Figure 3). Further details of this phenomenon can be found in the article [4].

Problems can also be caused by another phenomenon called bounce-back. This means that at the time t_1 the impact is so hard that the sliding dog is bounced back and its dogs miss the corresponding slots in gear wheel because of high remaining mismatch speed. This makes the gearshift longer and increases the probability of unsuccessful engagement described in previous paragraph (Figure 3 middle). Choice of appropriate initial mismatch speed and gearshift force is crucial; [3].

SHORTENING ENGAGEMENT TIME

Successful engagement regardless to current mismatch speed can be secured by the usage of tapered dogs' faces. If the taper angle (marked α in Figure 4) is bigger than the self-locking angle, the engagement is successful even with zero mismatch speed. Disadvantage of this modification is reducing the contact height and therefore the contact surface of the dogs in mesh. To preserve the strength of the dogs it might be necessary to make them higher (marked h in Figure 4), consequently the clutch becomes axially longer. Furthermore, the clutch becomes sensitive to the initial mismatch speed sign. In other words, the probability of bounce-back is determined by which one of the two parts rotates faster. In one way it is lower than for rectangular dogs, in the other it is bigger. This places higher demands on the external synchronization of the initial mismatch speed.

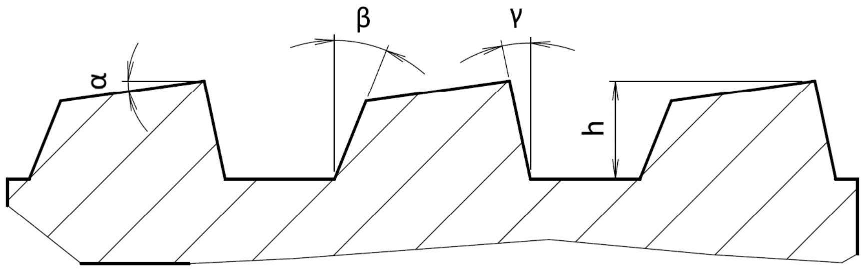

Figure 4 *Basic geometric parameters influencing the dogs' shape*

Another effect of the tapered face is that for bigger α values the face friction phase is impossible to happen for big values of initial mismatch speed. The breaking point is situation, when the vector *v* (see Figure 5) of the engaging dog becomes parallel with the face of the opposite dog. Omitting the face friction phase should shorten the gearshift. Therefore, we want to have the v_{radial} (depends on the initial mismatch speed) as big as possible and the v_{axial} as low as possible. However, the v_{axial} together with the length of the gearshift (which increases with bigger α as mentioned before) effects the gearshift time as well. Apparently, this situation leads to a compromise while looking for shortest possible time.

In case of the tested prototype of the dog clutch with face taper α = 8°, the minimal mismatch speed eliminating face friction phase is above 130 RPM for gearshift time 0,1 s. However, this assumes constant movement of the sliding dog. For real gearshift the necessary mismatch speed would therefore be higher.

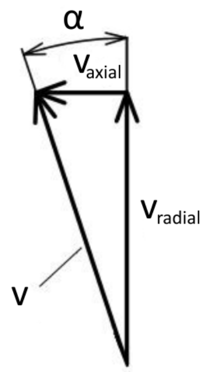

Figure 5 *Speed vectors during engagement*

Next parameters are the side angles β and γ. These might be generally different to each other. For the dog clutch without circular backlash, their value must be big enough so that the dogs are not self-locking. For lubricated steel the minimum value is therefore roughly 5°. Bigger values are preferable for lower bending stress and contact pressure under load but lead to bigger axial forces. They are also expected to make the gearshift longer (especially the taper γ) since big values are comparable to the gearshift with the mismatch speed sign provoking bounce-back (small values of taper α).

TESTING AT THE INERTIA TEST STAND

To verify the simulation model, experimental data of the gearshifts were necessary for comparison. To obtain as clear measured data as possible, the blocking rings were removed from the clutch. There was no torque transmitted, therefore the blocking mechanism was not necessary. Furthermore, the clearance between the sliding dog and the gearshift sleeve was filled and the mechanism therefore behaved the same as a standard dog clutch without blocking mechanism.

The inertia test stand (Figure 6) was developed for testing parallel shaft MT and AMT gearboxes under different operating conditions. It can also test gearshift mechanisms (1) if they fit into the tested gearbox (2) – Škoda MQ200 is used for gearshift mechanism tests. Since gearshifts in this gearbox occur without power flow (with the friction clutch disconnected), there is only a revolution sensor connected to the input shaft. Power comes from an asynchronous motor (5) which keeps the revolutions of inertia masses (3) constant during the test through belt drive (4). Energy of the rotating flywheel (3) represents the energy of a moving vehicle. The flywheel is connected to one of the differential output flanges using a tensometric shaft (measuring torque and revolutions), torque limiting clutch and standard homokinetic joint shaft. The differential is locked.

The gearshifts are carried out by a pneumatic shift robot connected to the gear selector mechanism of the gearbox using standard gear linkage. It can shift up to eight gears plus reverse. The test stand is controlled by a program compiled in National Instruments LabVIEW software. It also collects and saves all the measured data for each gearshift. The shifting force is measured on the shift robot. The sensor is placed on the connection between pneumatic cylinder and cable for gearbox actuation. The same is valid for the lift sensor. Further details of the test stand can be found in [5] and [6].

MQ200 is a manual parallel shaft gearbox for front wheel drive cars with transverse engine layout produced by Škoda for the VW group. It exists in 5-speed and 6-speed variants. All forward gears are synchronized and shifted by Borg-Warner synchromesh unit. Differential is included in the gearbox construction; [7].

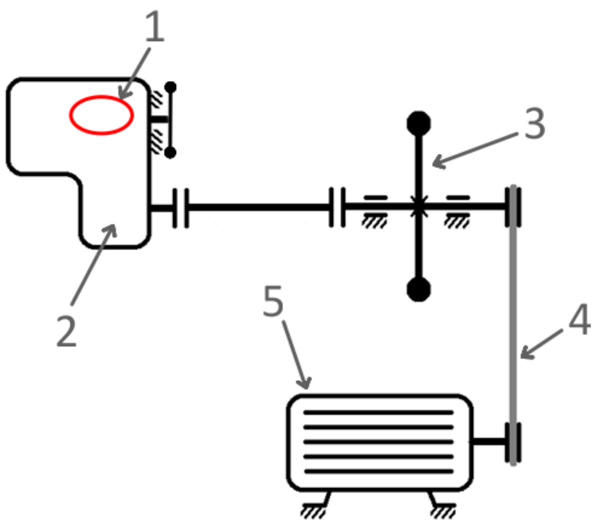

Figure 6 *Inertia test stand scheme* ***[8]***
1 – tested gearshift mechanism; 2 – tested gearbox; 3 – inertia mass; 4 – belt drive; 5 – asynchronous motor

EXPERIMENTAL RESULTS

Typical result of gearshift measurement can be seen in Figure 7. Note that the force and displacement values are measured at the shift robot, therefore their values have to be modified by gearshift mechanism ratio and in case of the gearshift force also by efficiency to obtain values at the gearshift sleeve (dog clutch itself). This gearshift ratio was calculated from the displacement values and efficiency of the gearshift mechanism was estimated regarding to [9]. At the displacement curve the face friction phase is clearly visible. The initial oscillations of the force curve are related to the initial sudden pressure rise in the pneumatic cylinder. Around 200 ms timestamp oscillations in input speed, force and torque are visible. These are caused by the dogs' engagement.

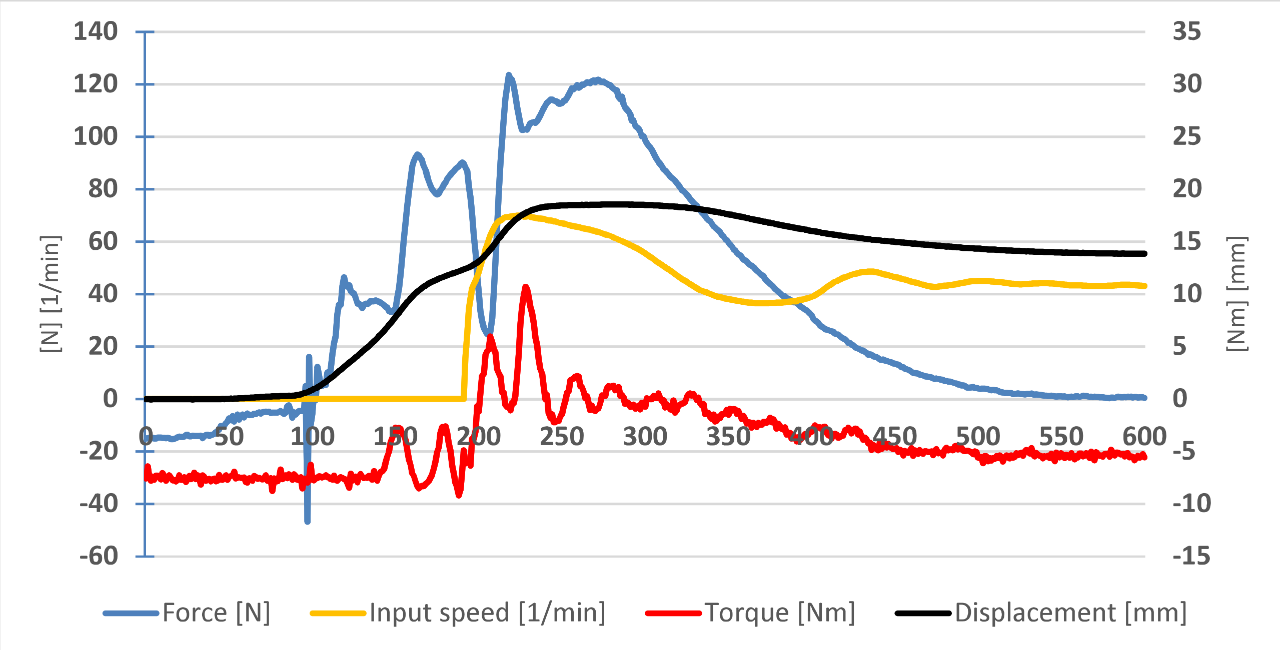

Figure 7 *Typical results of gearshift measurement (mismatch speed 50 min^{-1}; gearshift force 315,5 N)*
Note: Gearshift force is measured at the shift robot.

Gearshift time for various mismatch speeds and gearshift forces was measured regarding to the displacement curve. Since the initial position of the dogs and slots is generally random, the final value is an average of 10 measurements. Measured values and gearshift scenarios were then used as a reference for simulation results to compare and optimize the simulation model.

Table 1 *Average gearshift time for various mismatch speeds and gearshift forces*

Average gearshift time [s]		Gearshift force [N]			
		235,2	315,5	469,0	641,5
Mismatch speed [1/min]	5	0,481	X	X	X
	10	0,332	0,381	0,269	0,176
	25	0,208	0,206	0,180	0,139
	50	0,198	0,191	0,159	0,137
	100	0,206	0,176	0,156	0,136
	149	0,197	0,180	0,155	0,139
	249	0,196	0,171	0,150	0,136

SIMULATION MODEL

In order to sufficiently explore the important phenomena of dog clutch engagement the Adams software, which is a powerful computational tool used for multibody dynamics simulations, was employed. Utilization of multibody approaches is faster and accurate enough for the case of dog clutch simulations contrary to the usage of the finite element method, which could be very time consuming. Multibody approaches allow to consider large motion characterized by large displacements and finite rotations [10]. Rigid and flexible bodies can be constrained by joints and many force (torque) elements can be used during the model creation. Mutual contacts between bodies are also considered as special force elements based on the actual geometry of bodies and contact and friction parameters. The mathematical form of multibody models in Adams is systems of differential-algebraic equations (DAEs) where each free body has originally six degrees of freedom forming six ordinary differential equations and constraints are represented by algebraic equations. Such set of DAEs can be numerically solved using various methods based on the character of particular simulation models.

The investigated computational model was created by two rigid bodies representing two parts of the dog clutch (input and output side). The first body (blue in Figure 8) is representing a geared wheel and is coupled to fixed ground by a revolute joint. The second body is representing a sliding dog (gray in Figure 8) and is coupled to the ground by a sliding joint, which allows rotation around the wheel axis and sliding in the same direction. Inertia of both bodies was estimated to represent important inertia properties of experimental stand elements. The shifting force is further applied on the sliding dog. Both bodies can have different initial conditions on the angular velocities and positions. Alternatively, an impose motion could be applied to a selected body to represented driving motion.

A very important part of the model is the definition of the contact interaction. Real CAD geometry of both parts was used in order to define contact kinematics. Evaluation of normal contact forces is based on local geometry penetration and on a material stiffness coefficient. For realistic contacts there are also defined regularized damping forces in the normal contact direction based on a material damping coefficient and local penetration rate. The friction forces are represented by regularized Coulomb friction model with Stribeck effect.

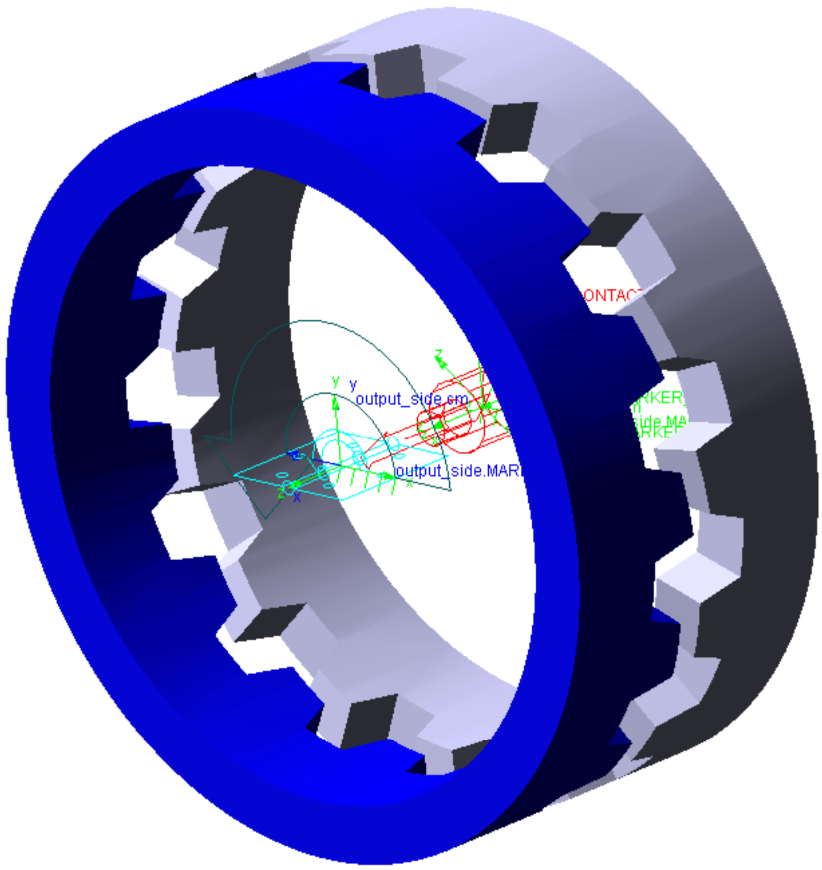

Figure 8 *Visualization of the computational model in Adams software*

SIMULATION RESULTS

Chosen results of the numerical simulations will be shown and commented in this section. The main advantage of the simulation model is that it allows cheaply observe many configurations of the dog clutch with various parameters. Operational quantities (i.e. angular velocities, shifting force value and its speed) can be variated as well as design quantities (such as angles of teeth, their height, inertia of wheels).

Main input parameters for performed simulations were the difference between angular velocities of the geared wheel and the sleeve dog, magnitude of the shifting force and its duration (the time period of its growth from zero to maximal value) and relative angular positions of both wheels. In reality the relative angular position is random variable and for statistically correct analysis huge set of numerical simulations should be performed. For sake of this paper, only selected scenarios were studied.

The first simulation (case 1) is determined by relative angular velocity difference of 50 rpm and shifting force of maximal magnitude 235,2 N (see Figure 9). The calculated sleeve dog displacement with respect to time is shown in Figure 10. It can be seen that this is the case of an optimal engagement and no face-to-face contact occurs. On the other hand, the displacement in

Figure 11 was calculated for same conditions but with different relative angle (10 degrees) and there is apparent face-to-face contact.

Case 3 (Figure 12) is the same as case 1 but with higher shifting force magnitude (469 N). Due to the higher force and therefore to the higher dog sleeve acceleration there is short contact occurrence of particular sides of teeth during the engagement, which could be observed in the response. Results for case 4 (the same as case 3 with 10 degrees of relative angle) are affected by face-to-face contact (see Figure 13).

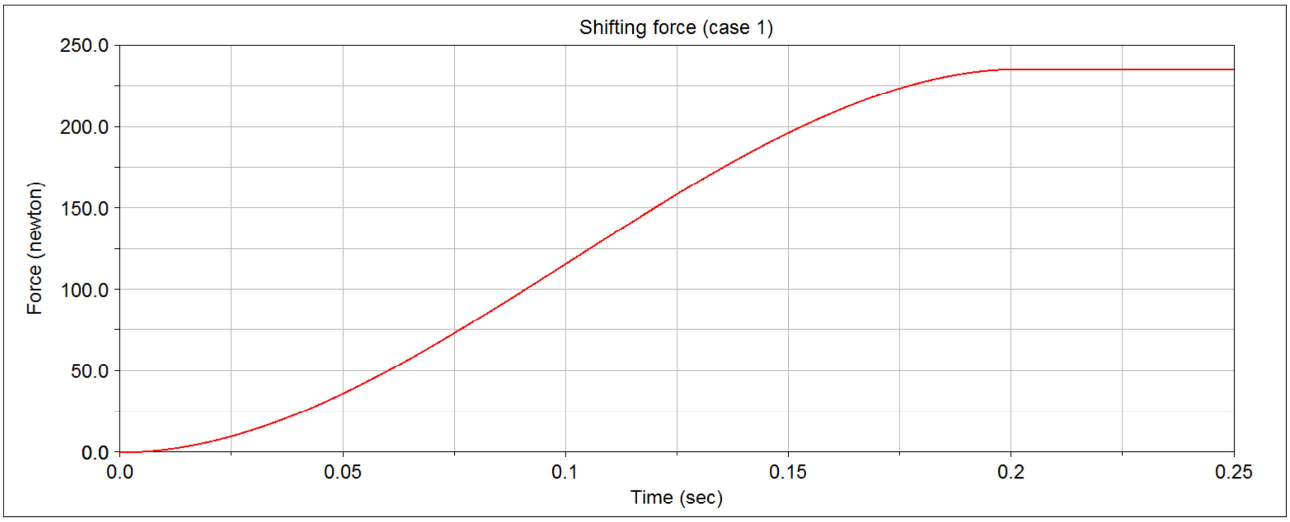

Figure 9 *Shifting force with respect to time*

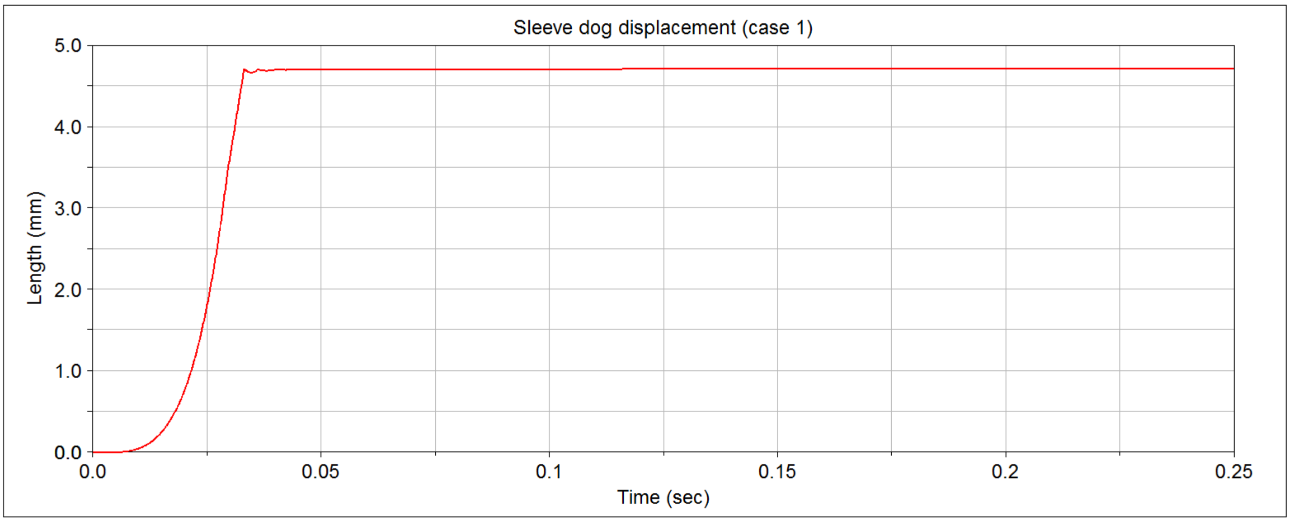

Figure 10 *Sleeve dog displacement for case 1 (50 rpm difference, 0 degrees shift, force magnitude 235,2 N)*

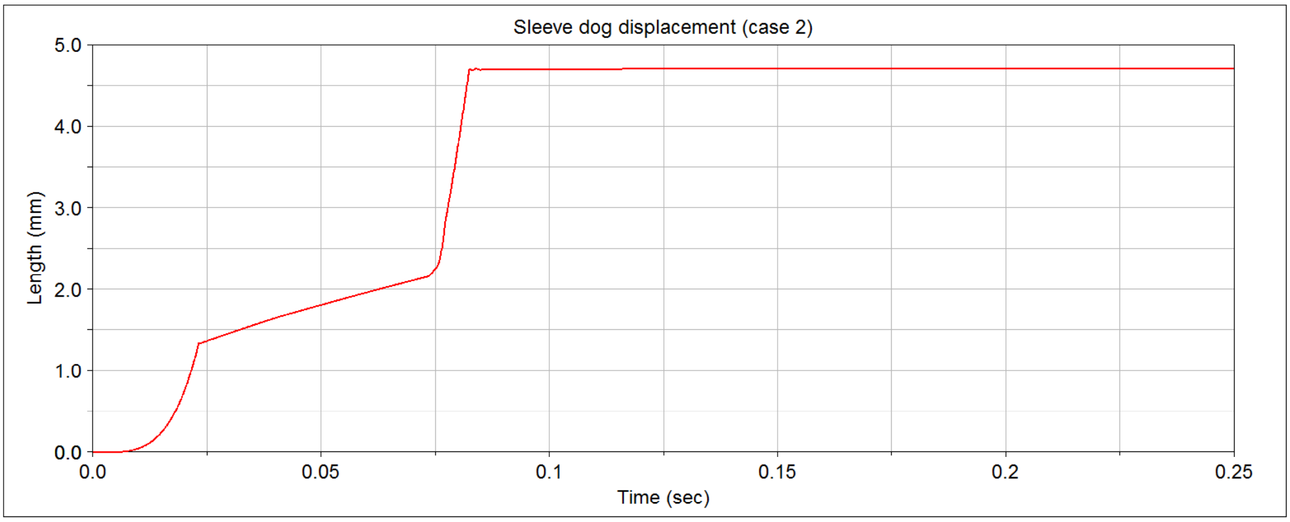

Figure 11 *Sleeve dog displacement for case 2 (50 rpm difference, 10 degrees shift, force magnitude 235,2 N)*

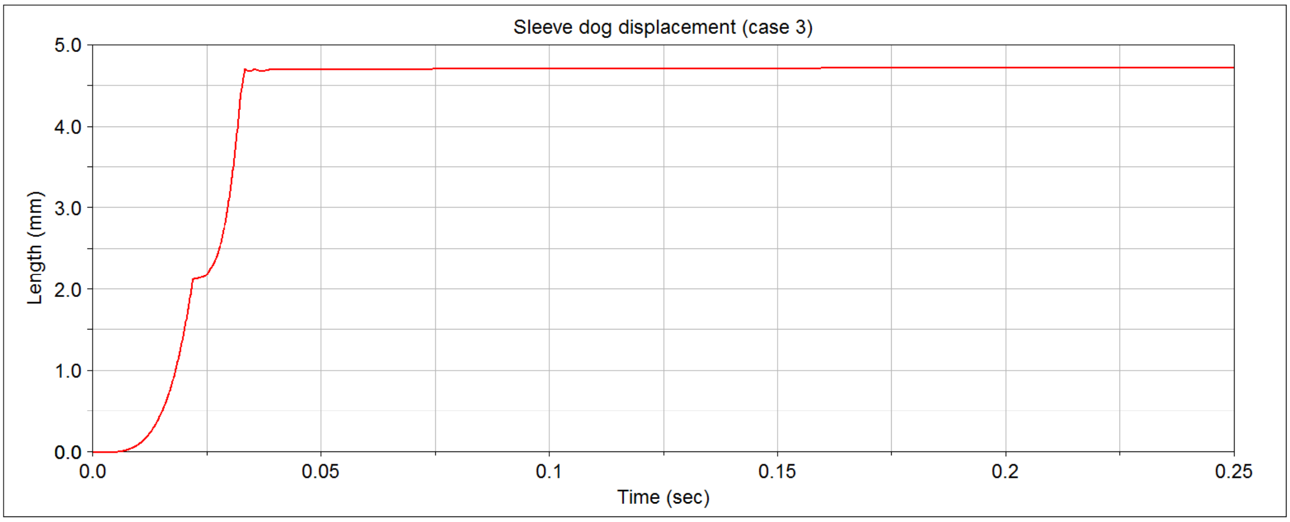

Figure 12 *Sleeve dog displacement for case 3 (50 rpm difference, 0 degrees shift, force magnitude 469 N)*

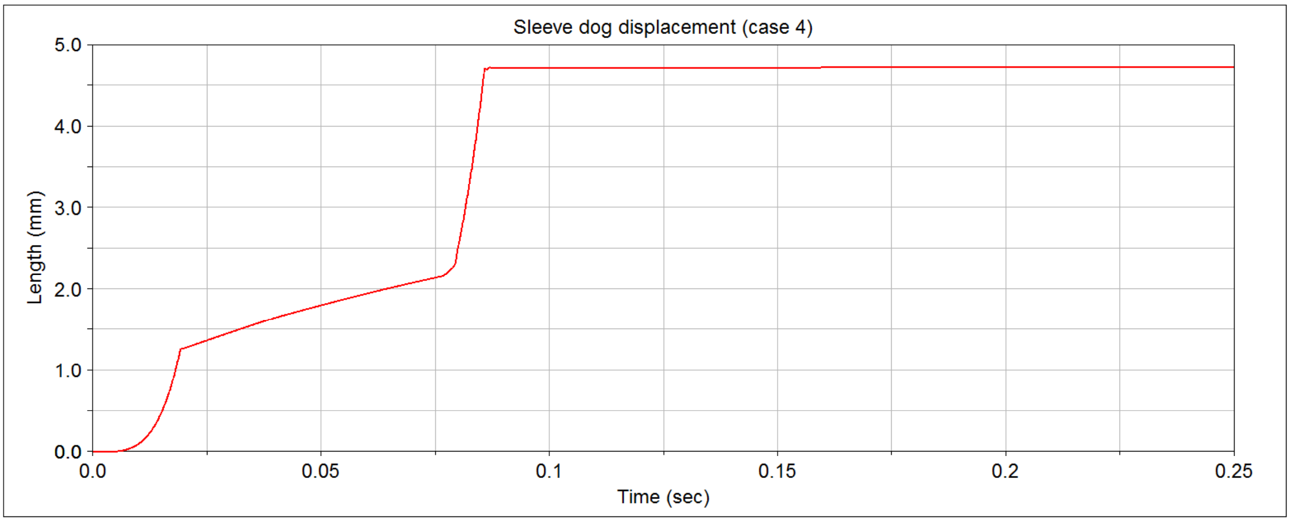

Figure 13 *Sleeve dog displacement for case 4 (50 rpm difference, 10 degrees shift, force magnitude 469 N)*

The last example of the simulation results (case 5) is the engagement for higher angular velocity difference of 149 rpm and shifting force with magnitude of 235,2 N. It can be seen from Figure 14 that there are several contact occurrences in the dynamic response. However, the time of the engagement is relatively short. All presented results are mainly affected by relative angles between two dog rings.

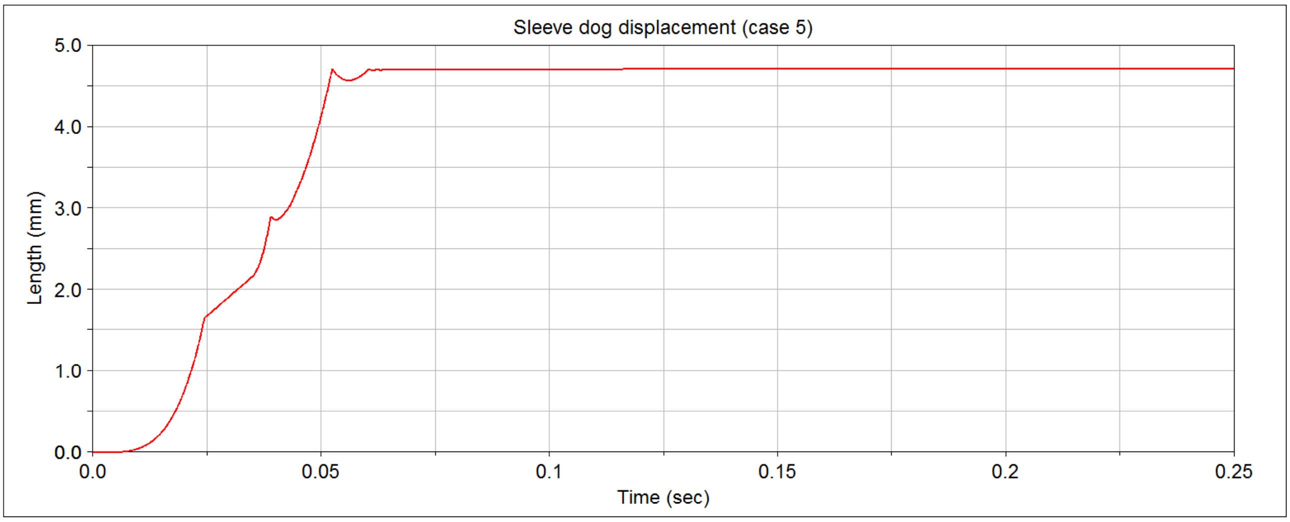

Figure 14 *Sleeve dog displacement for case 5 (149 rpm difference, 0 degrees shift, force magnitude 235.2 N)*

FURTHER PROGRESS

The simulation result show reasonable values and characteristics of the gearshift. These results are promising in terms of optimizing the shape of the dogs especially in order to minimize the gearshift time under particular circumstances (mismatch speed, gearshift force).

For better simulation optimization it turns out that clearer measured data is necessary. Especially measuring gearshift force and displacement at the gearshift robot outside the gearbox is very inaccurate. Between the dog clutch and measuring point there is plenty of parts including gearshift Bowden cables. These reduce the stiffness of the whole gearshift mechanism and reduce the accuracy of measuring. This lower stiffness can be seen in Figure 7. After engagement the displacement curve is heading towards lower values when the gearshift force vanishes even though the dog clutch itself remains in its position. This is caused by untensioning of the gearshift mechanism. Currently a brand-new test bench for synchromesh units and dog clutches [11] is under construction in the laboratories of Czech Technical University in Prague and new measurements are planned to improve the correlation between simulation model and experimental results.

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