

# Synchroniser modeling with application specific to the dual clutch transmission

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## Abstract

The synchromesh type synchroniser has been introduced to dual clutch transmission applications for gear selection prior to shifting. Historical research into this type of mechanism has targeted the application to manual transmission systems only. Such work targets the phenomena associated with driver feel such as shift effort and “double bump”. Now automated, the control of the synchroniser is less concerned with such aspects of the actuation, focus not trends towards repeatability of the process as well as speed of engagement.

The actuation of this type of mechanism relies on the balancing of torques derived from engagement chamfers, the cone clutch and losses experienced in the transmission. These torques affect the displacement of sleeve, asynchronisation of the target gear, as well as the unblocking of synchroniser ring and indexing of the gear to the locked position. Thus a simplified model of the synchroniser mechanism and associated gear is developed for the purpose of simulating its operation in a dual clutch transmission. Unlike similar simulations this model targets the actuation of the mechanism using input forces from the controller, rather than using the control of sleeve displacement to determine maximum forces experience by the driver.

To assess how operating characteristics have varied the mechanism has been modelled in the Matlab® environment. This paper presents the techniques used to model the mechanism, including the governing principles of synchroniser actuation and drag torque. Simulation results demonstrate that the primary variation in the mechanism is through the indexing chamfers. Additionally the influence of temperature variation is demonstrated for a fifth gear upshift, and the detrimental effects of cold starts demonstrated.

**Key words:** Synchroniser, Dual clutch transmission, Drag torque

## 1. Introduction and background

Peculiar to the DCT is the integration of the manual transmission synchronisers into an automated process. Though already used in automated manual transmissions the uncontrolled variable, drag torque, substantially changes between these two automated transmission types. This is attributed to sources of drag torque that are related to slipping speeds between components, such as bearings and concentrically orientated shafts. In the model developed herein additional consideration will be placed on to the modelling of the wet clutch drag, which incorporates a significant portion total drag losses (1).

Current research into DCTs is focused on the development of successful control

methodologies for the simultaneous control of both primary clutches. Dynamic modelling has been performed by (2-6), in each instance the control and response of the synchroniser is assumed to have no effect on the process of shifting and is subsequently ignored during modelling. Nevertheless, the synchroniser has been introduced into a new and significantly different operating environment suggesting that modelling and simulation is required for characterisation of changes to the process.

Synchroniser analysis by (7-11) is used as the basis for modelling owing to the development of torque equations for the cone clutch and indexing. When considered in conjunction with torque inequalities that demonstrate the equilibrium between cone, indexing and drag torques, these are the principles that control sleeve transition, which is the fundamental features from which the synchroniser model is developed.

Previous modelling of the synchroniser by (12-14) are used to investigate the process and engagement phenomena experienced in manual transmissions. Results demonstrate that speed matching is stable, whilst indexing is highly dependent on initial alignment of chamfers. In these simulations the sleeve velocity or displacement becomes the controlled variable so that the sleeve force can be determined, in doing so (14) demonstrates that the faster engagements reduces the presence of double bump, a MT phenomenon, whilst increasing the overall load. A point of conjecture with these simulations is the inclusion of drag torque. This torque is not discussed in the papers so it is not possible to identify the effects that it has on results.

Within the context of manual transmissions (13, 15) both demonstrate that the actuation force is not constant, but it rather reaches a maximum during synchronisation, and two local maximums occur during indexing when first the ring is disengaged, and then the chamfers engage on the gear dogs. These are both associated with the initial load engagement of the chamfer contact surfaces.

After reviewing literature relevant to both synchronisers and drag torque, this paper will present models for both the synchroniser mechanism and drag torques that are developed using Matlab/Simulink®. Results will then be presented for four typical gear shifts that may be expected during shifting, and a discussion of these results presented.

## 2. Synchroniser process and modeling

The process of synchronisation has many phases, depending on the author it can be anything from 5 to 11 (10, 13). For the purpose of this paper it will be broken down into the two primary phases and then again into additional sub-phases. The first phase is where the actual speed matching occurs; it is generally called synchronisation or asynchronisation, depending on the literature. Here the cone clutch is engaged by the axial translation of the sleeve and load is applied onto the ring through the thrust piece or sleeve. The frictional engagement causes speed matching of the two halves so that the gear side of the synchroniser matches speed with the shaft side. This is followed by the lockup of the two sides of the synchroniser, called indexing. Where the meshing of sleeve splines with ring chamfers and then hub chamfers occurs, allowing positive locking between gear and shaft, finalising the process, and enabling torque transfer. Each step is discussed below in greater detail; this is based on the work of (10, 12, 13). Figures 1 to 6 provide images of process steps (1) shaft and saddle, (2) sleeve and detent, (3) ring, (4a) target gear, (4b) hub chamfers, and (4c) cone clutch:

(1) Synchronisation is initiated, an axial load is placed on the sleeve which pushes

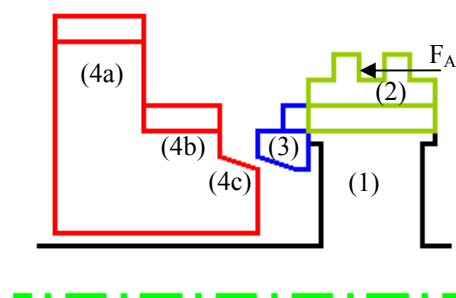


Fig.1 Synchroniser section at process step 1

- against the detent which is forced down and breakthrough is achieved.
- (2) The thrust piece via the ball pin is pushed onto the synchro ring, and translation of the part is initiated. During initial contact with the cone the ring undertakes an angular displacement, aligning the chamfers with those on the sleeve. This will be used to block the sleeve during speed synchronisation.
  - (3) The cone part of the synchroniser ring comes into contact with the cone on the gear and speed synchronisation begins. Circumferential clearances between the ring and sleeve allow a relative displacement and the splines on the sleeve lockup with the chamfers on the ring, preventing continued displacement of the sleeve. The development of cone lock is minimised using a combination of error angles and friction limits during cone design.
  - (4) When speed matching has finished the internal splines on the sleeve can now move over the chamfers as the friction torque is reduced, allowing the indexing torque to rotate and re-align the ring with the sleeve. The thrust piece and ball pin move back to initial positions through the use of a compression spring, and the force is transmitted directly through the sleeve.
  - (5) The sleeve then passes through the ring chamfers, and moves forward until it comes in contact with the hub chamfers.
  - (6) The second indexing phase begins, where the sleeve splines engage the chamfers and force the ring and cone to separate so that relative motion between the two halves of the synchroniser can be occur.
  - (7) With the teeth engaged the indexing torque forces relative motion between the teeth and the splines until they fully mesh and the sleeve moves across to completely cover the gear.
  - (8) With the splines fully meshed with the gear dog, positive locking occurs between the two sets of mesh teeth through undercutting of the chamfer sets.

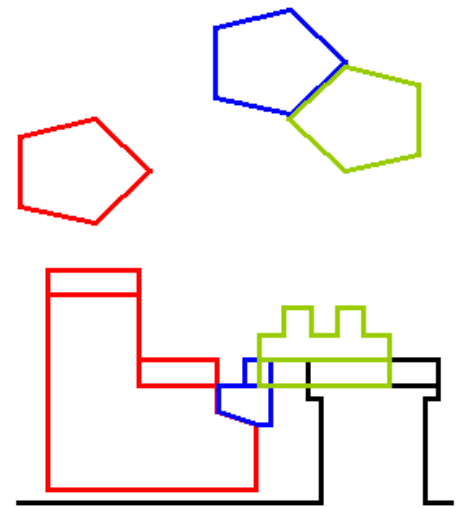


Fig.2 (a) Chamfer tips during blocking (step 2) (b) Synchroniser section at speed synchronisation (step 3)

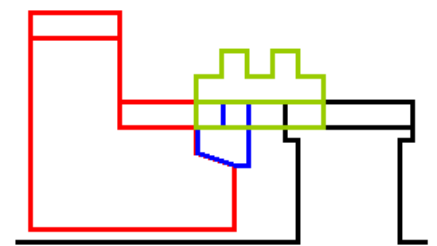


Fig.3 Section at sleeve transition (step 5)

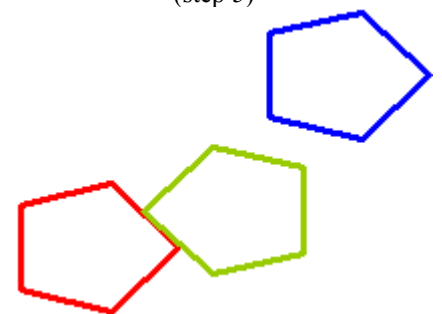


Fig.4 chamfer tips during hub indexing (step 6)

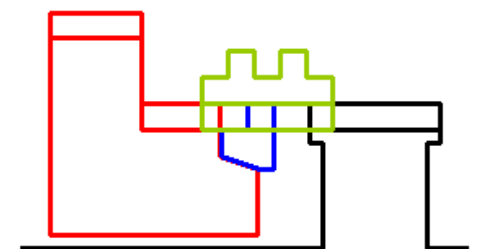


Fig.5 Synchroniser section after lockup (step 8)

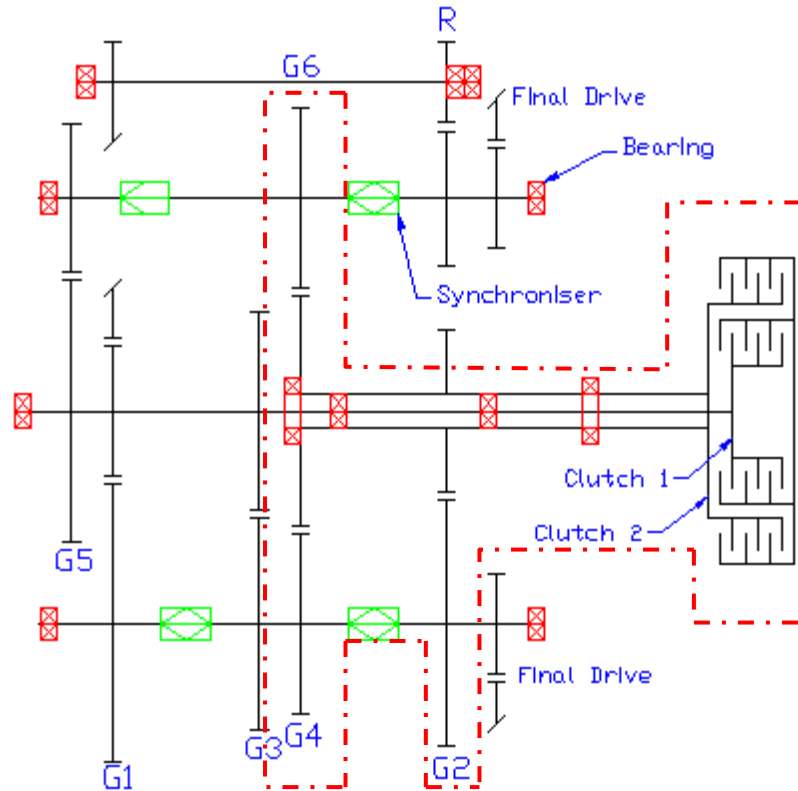


Fig.1 Dual Clutch Transmission Layout

Figure 1 shows a typical transmission layout for a wet clutch DCT which will be used as the basis for modelling. Given that the inertial of the vehicle greatly exceeds that of the reflected inertia of the target gear,  $I_V \gg I_{FW}$ , it is reasonable to simplify the model to only consider the response of the target gear to engagement, thus reducing the model. In this frame of reference the reflected inertia of the target gear is developed from all components between the synchroniser and clutch, hence if synchronising fourth gear, as will be performed in simulations, it is necessary to consider inertia, and indeed drag, from second, fourth, and sixth gears, as well as that of the open clutch pack, and effectively the final drive and vehicle inertia becomes the grounding for the simulation, as per the assumption.

This model requires the development of both linear and rotational rigid body models of the synchroniser to account both the translation of the sleeve and synchroniser and target gear rotation. The sleeve is controlled through the input hydraulic force; whilst torque is applied to the synchroniser in the forms of drag, cone clutch and indexing torques. Only drag torque is constantly applied to rotating components, cone clutch and indexing torque must be considered in a piecewise manner depending on the manner of engagement and load on the mechanism. The cone torque is adopted from (9 & 13) for first free fly, with oil wiping, and dry friction, with the inclusion of a lock up rationale.

$$T_C = \begin{cases} 4\pi\mu R_m^3 b \frac{\dot{\theta}_{syn}}{h} & X_S < 2 \\ \frac{\mu_c F_S R_C}{\sin \alpha} & X_S \geq 2, \dot{\theta} \neq 0 \\ T_D + I_{FW} \ddot{\theta}_{FW} & X_S \geq 2, \dot{\theta} = 0 \\ \frac{\mu_{c,s} F_S R_C}{\sin \alpha} & X_S \geq 2, \dot{\theta} = 0, T_D > T_c \end{cases} \quad (1)$$

This equation is maintained through the first four process steps defined above. However, during the remaining stages the applied load ( $F_S$ ) is considered as a function of net torque on the mechanism and sliding friction of the sleeve over the ring, thence:

$$T_C = \mu_r R_I T_D \times \frac{\mu_{C,S} R_C}{\sin \alpha} \quad (2)$$

The blocking, or indexing, torque is dependent on the direction of sleeve velocity ( $\dot{x}$ ) and side of chamfer engagement ( $\lambda$ ). Indexing torque is represented as:

$$T_I = \begin{cases} FR_I \frac{1 - \mu_I \tan \beta}{\mu_I + \tan \beta} & \lambda + ve, \dot{x} + ve \\ FR_I \frac{1 + \mu_I \tan \beta}{\mu_I - \tan \beta} & \lambda + ve, \dot{x} - ve \\ -FR_I \frac{1 + \mu_I \tan \beta}{\mu_I - \tan \beta} & \lambda - ve, \dot{x} + ve \\ -FR_I \frac{1 - \mu_I \tan \beta}{\mu_I + \tan \beta} & \lambda - ve, \dot{x} - ve \end{cases} \quad (3)$$

The nature of the mechanism requires that successful engagement is a result of the balancing of different torques during different process steps. During the speed matching phase of synchronisation cone and indexing torques are combined with the drag torque in the following inequality to determine if blocking is successful. If this inequality fails the sleeve will push through the blocking ring and engage the gear chamfers prior to the completion of synchronisation causing audible clash of the chamfers, resulting in damage to the chamfer tips. The sign of the drag torque is dependent on the direction of torque, not the type of shift as suggested by Socin and Walters (9) for manual transmissions.

$$T_I < T_C \pm T_D \quad (3)$$

Alternatively, through the unblocking of the ring and the indexing phases of synchronization indexing torque must always exceed that of drag to ensure unblocking is successful. If this fails the sleeve is blocked and prevented from completing the synchronisation process.

$$T_I > T_D \quad (4)$$

Desynchronisation of the mechanism is realised if, after ring unblocking,  $T_D > T_C$ , if there is sufficient speed generated in the mechanism partial clash is realised in the chamfers, but if speed is low indexing will still be successful.

The model developed then incorporates these piecewise equations for cone and indexing torques with a model of drag torque into a rigid body model of the sleeve, ring and free wheeling target gear. In this circumstance the ring has three states, fixed to sleeve, as when under synchronization or post ring unblocking, dynamic, during initial freewheeling, or fixed to target gear, such as during ring unblocking. By maintaining such states effectively the system migrates between 3 and two effective degrees of freedom over the entire engagement process.

### 3. Drag Model

As part of a review into the process affecting DCT different sources of drag torque were identified (16). These include bearing losses, windage of gears, shear in concentric shafts, tooth friction, and clutch shear. Of note are the works of (17) and (18) who both demonstrate the ability to develop reasonably accurate drag models in geartrains. To therefore translate these works onto the synchronisation process requires the consideration of the actual geartrain section that undertakes synchronisation, and is expanded to include a model for clutch drag.

From Figure 1 it can be seen that there are two separate speeds to consider. It is immediately obvious that, during the synchronisation process, components between the synchroniser and the clutch pack will generate drag torque. Gear windage and friction as well as some bearing losses are associated with the absolute speed of shafts, whilst other bearings, clutch slip and concentrically arranged shafts are affected by the slipping speed between the two clutches. Therefore two separate models must be developed to account for the different speeds.

The absolute drag model, using the actual shaft speed, uses the bearing drag equations from (19), widely established as the fundamental basis for determining bearing losses. Tooth friction is developed from (20), whilst gear windage is based on the fluid dynamics model presented by (21). Dimensionless models are avoided here due to questions raised in (22) about validation of this type of model.

The relative drag model is based on the differential speed of the two clutches. Again (19) is used for bearing losses. Additionally, losses developed in the concentrically located shafts are modelled as Couette flow, and can be simulated using the equation presented by (23) for concentrically aligned rotating cylinders. Finally the resistance generated by the wet multi-plate clutch is modelled based on the theory presented by (24), where effects of mass flow, centrifugal force, and capillary action are used to account for the reduction of drag at high speed.

### 4. Simulations and discussion

Simulations have been made for the engagement of fourth gear through upshifts from third and downshifts from fifth gear. The parameters for these simulations are:

Table 1. Simulation parameters

Parameters	Value	Units
Cone angle	7	°
Coefficient of friction	0.1	-
Mean diameter	95	mm
Number of chamfers	45	
Chamfer angle	60	°
Pitch diameter	120	mm
Coefficient of friction	0.1	-

The first set of results is presented for standard upshifts from 3<sup>rd</sup> to 4<sup>th</sup> gear, and downshifts from 5<sup>th</sup> to 4<sup>th</sup>. In both instances the simulation is run from the initial steady state conditions for freewheeling gear speeds, which is highly dependent on drag torque and viscous shear in the cone clutch. Results are presented below for the sleeve displacement and slip speed in the cone.

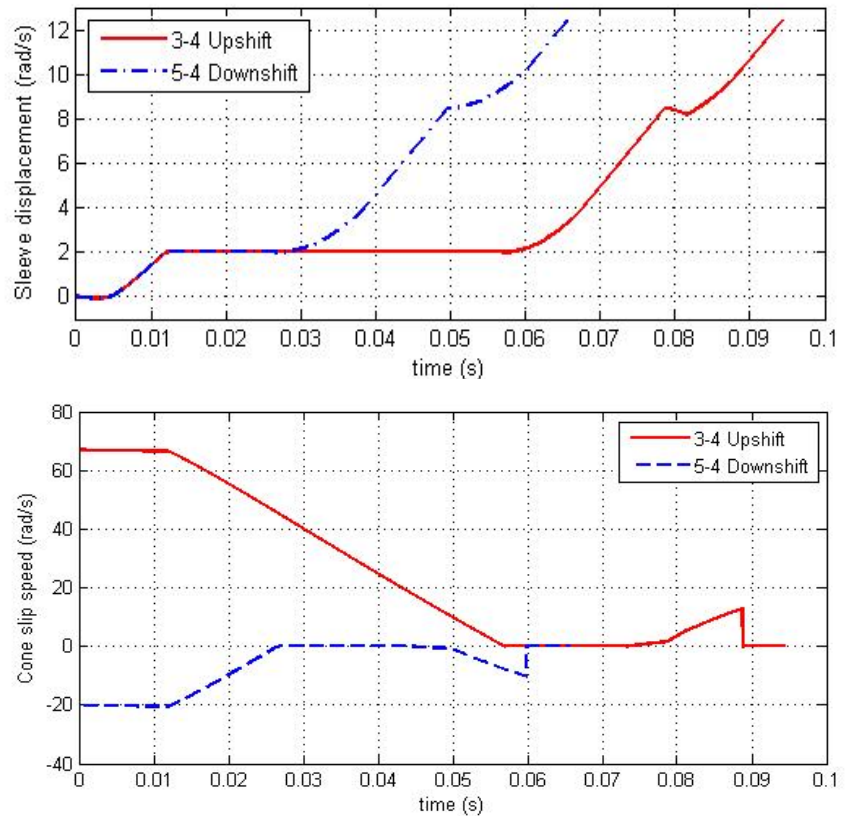


Fig.2 Sleeve engagement and cone clutch slip speed for typical up and down shifts

The results in figure 2 demonstrate the sleeve motion and cone clutch slip speed for the engagement of the synchroniser mechanism. The primary differences between the two shifts are the shorter synchronisation time for the downshift, as a result of lower drag torque and differential speed between target gear and sleeve, and the reversal of the sleeve realised in the upshift, as a result of high drag during indexing forcing the backtracking of the sleeve over the tip of the chamfer. This is countered by higher torques then aiding the engagement process and reducing the overall indexing duration to being similar to that of the downshift. The increased drag results primarily from the high speed differential between third and fourth gears and the subsequent impact on viscous shear in the wet clutch pack.

The key uncontrolled variable in the mechanism is the alignment of sleeve and hub chamfers. Alignment results from a combination of many uncontrolled variables and realistic modeling of this aspect of the process is unreasonable, as such chamfer alignment has been preset in the simulation and can be modified to investigate different aspects of engagement. To demonstrate the influence of chamfer alignment multiple simulations are used to demonstrate this variation. Previous work by Lui, *et al*, (12) demonstrates the possibility of different alignments emerging, including ideal, positive indexing, and negative indexing alignments. This has been expanded to include a head-on-head alignment as an example. The variable " $\delta$ " represents the full range of alignment of any chamfer pair.



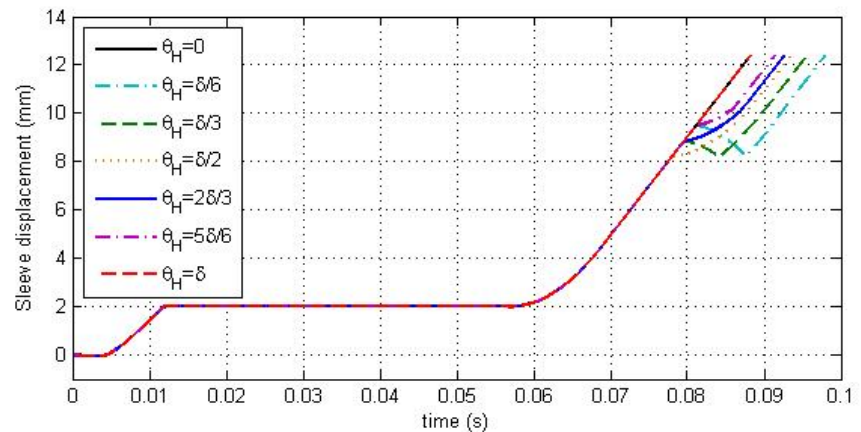


Fig.3 Sleeve engagement with variation of chamfer alignment

From the simulation of different alignments results indicate that the overall process time can fluctuate by approximately 10 percent, with the ideal case being perfect alignment and worst case arising when the chamfers align with adverse indexing torque. The sleeve reversal is a direct result of drag torque exceeding the indexing torque during indexing.

Finally, the influence of temperature variation can be considered for the synchroniser engagement process. The primary consideration when evaluating the influence of temperature variation is the effects on automatic transmission fluid, or ATF. Table 2 below demonstrates the effects of cold start on viscosity and density parameters of a typical ATF. Whilst there is minimum variation of density, less than 5%, the viscosity increases by approximately 800%. Thus it would be reasonable to expect significant influence of the ATF on system drag torques during the synchronisation process.

Table 2. ATF parameters

Parameters	0°C	40°C	60°C	Units
$\mu$	0.239	0.029		N s/m
$\nu$	$2.7 \times 10^{-4}$	$3.4 \times 10^{-6}$		N/m
$\rho$	881.2	853.4		Kg/m <sup>3</sup>

The results presented below are for a four to five upshift in the DCT, three separate operating temperatures for the ATF are applied at 0°C, 40°C, and 60°C. Whilst both 40°C, and 60°C demonstrate quiet similar results, there is evidence that higher drag increases the time required to unblock the mechanism, which is then countered by rapid engagement of the hub chamfers to result in very similar engagement times. However, the 0°C simulation demonstrates significantly different results, where the very high drag resulting from the engagement prevents correct synchronisation of the mechanism. Initial transition of the sleeve is extended as there are significant increases in the time required to squeeze oil film from between the cone surfaces. This is followed by cone torque being insufficient to overcome drag, with the resultant being blackout of the mechanism. This type of failure can also lead to overheating of friction surfaces and permanent damage to the mechanism. It is peculiar to the wet clutch type DCT as the viscous shear in open clutch plates is the most significant contributor to drag torques.



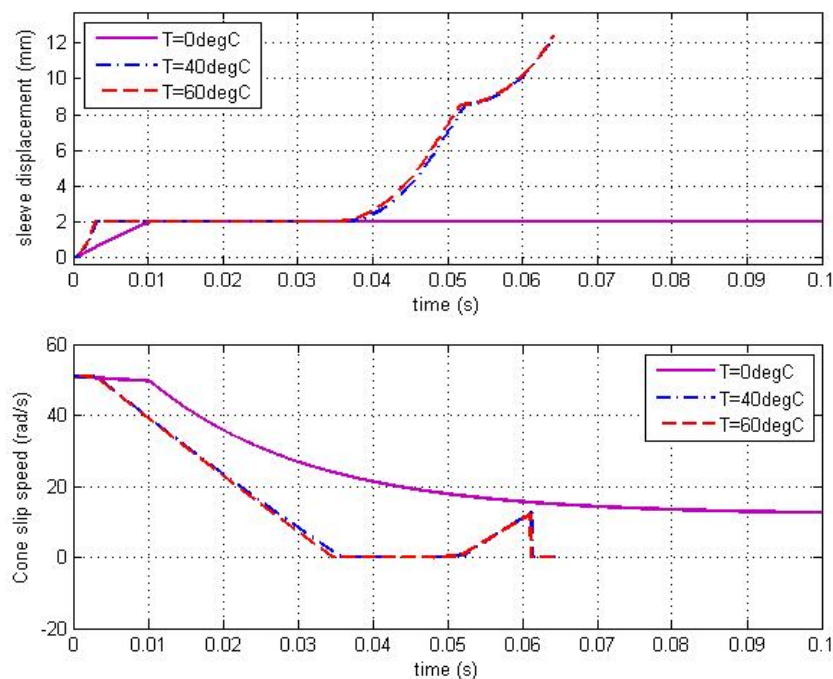


Fig.4 Sleeve engagement and cone slip speed with variation of ATF temperature

## 5. Conclusion

In this paper a model of a typical synchronizer has been developed for applications specific to the dual clutch transmission. Simulations were made to first demonstrate the effectiveness of the model, and second, to explore the influence of uncontrolled variables of chamfer alignment and operating temperature variation. The results of simulations demonstrated the influence of alignment variation which varied engagement by around 10ms, potentially influencing the shift control process. However, the influence of cold starts on the engagement process has been much more significant, where there is a reasonable potential for high clutch drag to cause blockout failure of the mechanism.

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