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# Predicting friction in synchronizer systems

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ABSTRACT

The coefficient of friction in synchronizers is important from both a performance and a functional point of view. The synchronization process is highly transient, and the parameters affecting the coefficient of friction have strong mutual dependences, making analysis highly complex. A new friction model for a lubricated molybdenum-steel contact has been developed based on integrating results from physical rig tests and FEM simulations. A simplified thermal model has also been developed, with the purpose to quickly assess the coefficient of friction based on transient force and synchronizer dimensions. The results show good correlation with measured data except at very low sliding speed.



## 1. Introduction

To achieve fast gear shifts and robust gear engagements, synchronizers are often used in automotive applications, such as manual transmissions, automated manual transmissions, and dual clutch transmissions for passenger cars as well as commercial vehicles, e.g. trucks. A synchronizer is a mechanical machine element that synchronizes the rotational speeds of the gearbox during gear shifts [1,2]. Synchronizers are usually used at each driving gear. An example of a synchronizer can be seen in Fig. 1. The *driver* component is mounted at the shaft, and the *sleeve*, *wire spring*, and *latch cone* rotate with the driver. The *coupling disc* is mounted on the gear wheel, and the *inner cone* rotates with the *coupling disc*. Due to the gear ratios in the gearbox, there will be different rotational speeds for the latch cone and the inner cone in all cases, except when the gear is engaged or when the entire gearbox is stationary. Thus, when the gear shift is initiated, the rotational speed of the latch cone and the inner cone will differ. To synchronize the rotational speed, the latch cone is pressed into contact with the inner cone, and the frictional torque will synchronize the gear wheel to the shaft. A variant of this synchronizer, a double cone synchronizer, includes an additional cone interface inside of the *inner cone*, to increase the synchronization torque. A prerequisite for successful synchronization is that the gear engagement is blocked during asynchronous speed. The synchronizer, consequently, must have a function that blocks the axial movement of the shift sleeve in the relative position where the latch cone comes into contact with the inner cone. The blocking function depends on the friction torque generated in the conical contact interface, and therefore directly depends on the coefficient of friction between the interacting conical surfaces. Thus, the effect of a reduced coefficient of friction, e.g. because of wear or local thermomechanical effects, not only reduces gear shifting performance, but eventually also result in a gear shift failure [1,2]. The direct effects from an increased coefficient of friction, relative to target, are higher thermomechanical loading of the synchronizer, as well as risk for seizure of the cones, which can cause gear engagement problems [1]. It is therefore of paramount importance to quantitatively understand the frictional behavior during the entire synchronization process.

Two of the most important parameters influencing the frictional properties of a synchronizer are the material pair and the lubricant introduced between the sliding surfaces. One of the mating cones is usually made from hardened steel, while the other cone is often coated with a friction material, such as carbon fiber, a carbon compound, molybdenum, or sintered bronze. In this paper, the frictional properties of a molybdenum coated latch cone, a steel inner cone, and the lubricant Scania oil STO 2:0 Gearbox 75W-90 (kinematic viscosity 115 $mm^2/s$ at 40 °C, 15.2 $mm^2/s$ at 100 °C) are investigated. Synchronization is a highly transient process. In the presented research, the relative sliding speed between the synchronized surfaces was reduced from ~6 to 0 m/s in ~0.15 s. During this time, the axial force applied from the maneuvering system was increased from 0 to 3000 N, giving a nominal surface pressure increase from 0 to 4.5 MPa, and a nominal surface temperature increase of 20 °C. In this paper, the surface temperature is considered to be the nominal temperature of ideally smooth surfaces, and not the local flash temperatures that are found on rough engineering surfaces.

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**Nomenclature**

| | |
|---|---|
| $A_C$ | Cone area [m$^2$] |
| $F$ | Force, axial [N] |
| $\dot{F}$ | Force ramp [N/s] |
| $F_{max}$ | Maximum force [N] |
| $I$ | Inertia to synchronize [kg m$^2$] |
| $M_C$ | Cone torque [Nm] |
| $R_C$ | Cone radius [m] |
| $T$ | Temperature [K] |
| $T_a$ | Temperature at evaluation point $a$ [K] |
| $T_{i,j}$ | Temperature at node $i$, time $j$ [K] |
| $T_{in}$ | Temperature, initial [K] |
| $W_C$ | Cone width [m] |
| $c_p$ | Specific heat [J/(kg K)] |
| $c_{1to8}$ | Coefficients [-] |
| $h$ | Node to node distance [m] |
| $n$ | Number of evaluation points [-] |
| $n_D$ | Number of nodes in thickness direction [-] |
| $n_C$ | Current node [-] |
| $p$ | Contact pressure [Pa] |
| $p_a$ | Contact pressure at evaluation point $a$ [Pa] |
| $\dot{q}$ | Heat flux density [W/mm$^2$] |
| $t$ | Time [s] |
| $t_s$ | Stable time step limit [s] |
| $v$ | Sliding speed [m/s] |
| $\alpha$ | Cone angle [rad] |
| $\gamma$ | Heat partitioning factor [–] |
| $\lambda$ | Thermal conductivity [W/m K] |
| $\mu$ | Coefficient of friction between cones [-] |
| $\rho$ | Density [kg/m$^3$] |
| $\omega$ | Rotational speed [rad/s] |
| $\omega_L$ | Rotational speed limit [rad/s] |

Much research has been published on the coefficient of friction, e.g. for molybdenum synchronizers [3], for systems such as wet clutches in e.g. lock up clutches of automatic transmission and limited slip differentials [4–8] as well as more general models [9–11]. It has been shown that the coefficient of friction for a given material and lubricant combination depends on sliding speed, contact pressure and temperature [12,13], as represented by Eq. (1).

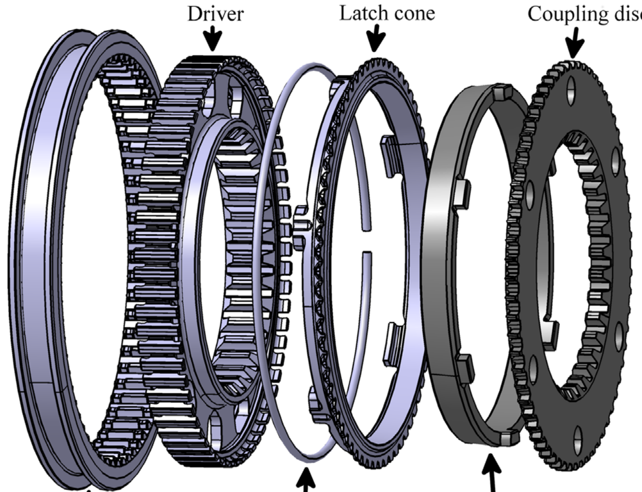


**Fig. 1.** Scania's current (2015) single cone synchronizer.

$$\mu = f(p, v, T) \tag{1}$$

Here $\mu$ is the coefficient of friction in the interface between the pair of cones, $p$ is the contact pressure, $v$ is the sliding speed, and $T$ is the nominal contact temperature. It is assumed that the contact between the conical surfaces can be treated as a nominally flat contact, and thereby the contact pressure is proportional to the applied axial force:

$$p \propto F \tag{2}$$

Under some conditions, typically for manually shifted transmissions, there may be situations where the driver applies shift force that is dependent on engine speed, e.g. gear shifting in steep road inclinations. However, in this study it is assumed that the applied axial force is only a function of time, $t$, i.e.

$$F = f(t) \tag{3}$$

The sliding speed, $v$, in the interface depends on parameters such as the initial relative speed between the cones, the synchronizer design, the inertia to synchronize, the applied axial force, and the coefficient of friction. For a given synchronizer system and gear change, the sliding speed can be described as:

$$v = f(t, \mu, F) \tag{4}$$

The temperature in the synchronizer contact interface depends on the material combination of the contacting surface and the loading parameters. For a given synchronizer design and material pair, the temperature is given by:

$$T = f\left(t, v, \mu, F, T_{in}\right) \tag{5}$$

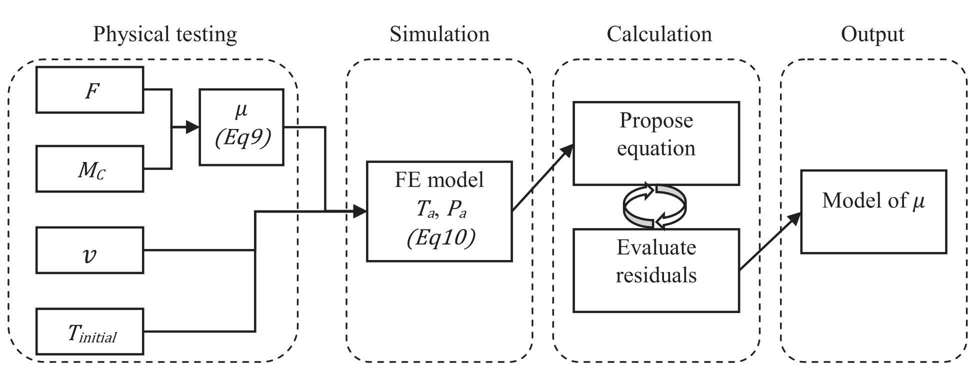


**Fig. 2.** Analysis workflow.