Strojniški vestnik - Journal of Mechanical Engineering 59(2013)6, 358-366

DOI:10.5545/sv-jme.2012.920
Original Scientific Paper
Received for review: 2012-12-18
Received revised form: 2013-02-28
Accepted for publication: 2013-03-29

A Theoretical and Numerical Study of an Additional Viscosity Term in a Modified Elasto-Plastic Friction Model for Wet Friction Clutch Simulations

Tomaž Petrun[1,*] – Jože Flašker[2] – Marko Kegl[2]
[1] AVL-AST d.o.o., Slovenia
[2] University of Maribor, Faculty of Mechanical Engineering, Slovenia

This paper deals with a theoretical and numerical study of various viscosity terms in the modified elasto-plastic friction model and their influence on the resulting friction force-torque, transmitted through the contact of the friction clutch. Various simple viscous definitions for fluids considering shear rate dependent viscosity were investigated. The Carreau fluid model was chosen as a basis for the research, since it can describe Newtonian, dilatant, and pseudo plastic fluids. In addition to theoretical investigations, numerical simulations under realistic friction clutch operation conditions were carried out. The results were compared to the results of a validation case for a dry friction clutch simulation using the modified elasto-plastic friction model. This research showed significant differences between various viscosity definitions and revealed the drawbacks of such approach, were simple viscous models were used.

In addition to viscosity, calculating the heat generated due to friction and its influence on the contact temperatures are discussed briefly. The basic theory and equations are given along with the directions for future work. The requirements for an accurate temperature calculation in the friction contact are outlined.

Keywords: friction model, friction clutch simulation, simple viscosity models, Carreau fluid

0 INTRODUCTION

In the development process of a complete vehicle powertrain, each part of the whole system must be considered accurately. To be able to do this, numerical software tools with adequate models of single parts, assemblies, and connections between bodies are needed.

The friction clutch is an important part of the vehicle powertrain. Therefore, modelling and simulation of a fully functional friction clutch is vital for accurate powertrain response simulation. In order to extend and upgrade the functionality of the commercial software code for numerical multi-body simulations, AVL EXCITE, such a model with special requirements has been developed and validated experimentally, Petrun et al. [1]. The development and validation was carried out for dry friction clutch applications. In the automotive industry, especially in passenger cars, the most common type of clutch is the dry friction clutch due to its good drivability, comfort, and its simple and relatively cheap construction. However, wet clutches are also used, although they are more common in high power engine applications, where performance is more important than comfort. In these applications the fluid is used to achieve the desired tribological characteristics of the contact pair (usually metal to metal) and also to cool the friction clutch.

This research work focuses on additional viscous terms for the modified elasto-plastic (EP) friction model, proposed by Petrun et al. [1]. The development and experimental validation of the EP Friction Model was focused only on dry friction clutch applications where very good agreement for various commercial friction materials was achieved. However, in order to account properly for wet friction clutch applications, one must also consider the viscous forces in the open and synchronization phase, due to the viscosity of the fluid present.

In the literature, Bukovnik et al. [2], Fajdiga et al. [3], many complex viscous models have been proposed, where the dynamic viscosity η is a function of the shear rate $\dot{\gamma}$, temperature T and pressure p, for example: Vogel, Barus, Cross, Rodermund, Kuss, Roelands, etc. Furthermore, the resulting viscous force in these models also depends on the real geometry and patterns of the contact pair, lubricant flow conditions, pressure distribution in the contact, etc. To get accurate results with these models, complex multi-material simulations, including flexible bodies and an adequate fluid model, are needed to calculate the pressure and gap height distribution in the contact.

The friction model discussed in this work is intended to be implemented in the target software AVL EXITE. This fact imposes some special requirements, outlined as follows. The friction model must connect two bodies in the multi-body system as a joint, Fig. 1. The model must be able to calculate the exact

*Corr. Author's Address: AVL-AST d.o.o., Trg Leona Štuklja 5, 2000 Maribor, Slovenia, tomaz.petrun@yahoo.com

amount of the transmitted torque through the friction contact. Available inputs for the friction model are the relative sliding velocity of the bodies in contact. The model should account for important friction induced dynamics. Currently, the friction model is still 1D. Therefore, the investigated viscous models should also be 1D and only velocity dependent. For this reason, only simple shear rate (velocity and gap height) dependent viscous models will be investigated in this work.

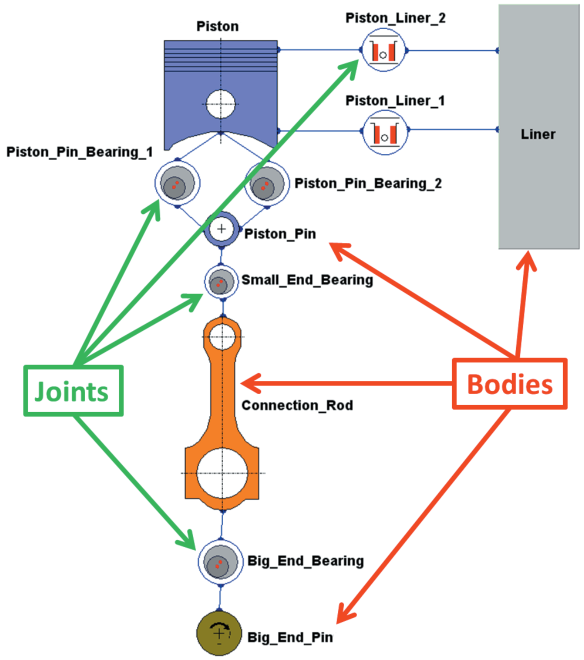

Fig. 1. *Example of a multi-body system in the target software for friction model implementation*

The main focus of this research is a theoretical and numerical investigation of the influence of various viscosity models on the behaviour of the friction clutch and the contribution on the resulting friction force. In particular, the influence on the synchronization process and the so-called drag force in the open phase are of main interest. This work focuses on four different simple viscosity models, which could be used for the application at hand. Furthermore, a basic theory for the calculation of heat generation due to friction will be briefly discussed at the end. Namely, the consequences of the heat generated due to friction are temperature changes at the friction contact. Since temperature has a notable influence on the tribological characteristics of the contact and on friction induced dynamics, this is a very important topic, [1]. The main requirements and demands for an accurate temperature calculation are given in order to indicate the direction of future work.

1 THE MODIFIED EP FRICTION MODEL

The modified elasto-plastic friction model for dynamic friction clutch simulations in a multi-body system, proposed by Petrun et al. [1], is a member of the Single State Variable (SSV) friction model family, Dupont et al. [4] and [5]. These models describe the friction contact as a contact of two brushes, where bristles interact, Fig. 2.

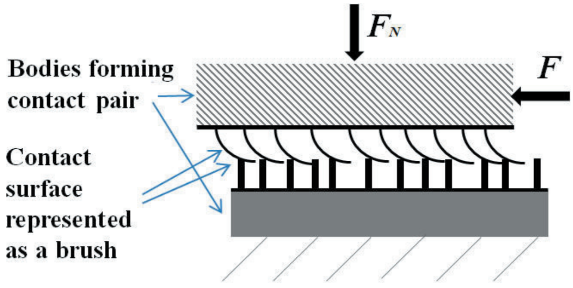

Fig. 2. *Friction contact description by a SSV friction model*

The resulting friction force F_{fEP} of the modified EP friction model is defined as follows:

$$F_{fEP} = \left(\sigma_0 z + \sigma_1 \dot{z}\right) F_N, \qquad (1)$$

where σ_0 represents the stiffens of the brush, σ_1 is the damping coefficient of the brush, z is the bristle deflection, also called the pre-displacement, $\dot{z}$ is the first time derivate of z, F_N is the normal force, and F the external force, see Fig. 2. Note that the state variable z is calculated from the differential equation for $\dot{z}$:

$$\dot{z} = \frac{dz}{dt} = v\left(1 - \alpha(z,v)\frac{sign(v)}{g(v)}z\right), \qquad (2)$$

where, v is the relative sliding velocity, $\alpha(z,v)$ is a function controlling the friction contact state, and $g(v)$ is a function describing the tribological characteristics of the friction contact, e.g. the Stribeck curve. For more details please see [1], [4] and [5].

This modified elasto-plastic friction model is capable of continually providing accurate results of the actual transmitted friction torque through the friction contact in all operational phases of a friction clutch. Furthermore, this model accounts for realistic tribological parameters and friction induced dynamics and was successfully validated against experimental measurements.

The model setup and experimental validation were carried out for dry friction clutch applications, where very good agreement was achieved. For wet friction clutch applications, additional terms for viscosity must be added. An advantage of this model is that additional terms can easily be embedded as additive terms into the expression for the resulting friction force [1], [4] and [5]. For our purpose the extended friction model can be written as follows:

$$F_{fEP+} = \left(\sigma_0 z + \sigma_1 \dot{z}\right) F_N + F_v, \tag{3}$$

where, F_v represents the resulting viscous force defined by an adequate viscosity model.

2 SIMPLE VISCOSITY MODELS

Four different simple viscosity models will be investigated. In these models the dynamic viscosity η is considered either constant or shear rate dependent. For all presented models the same contact area A_C is taken into account and the relative velocity v is averaged at the mean radius of the friction clutch. The mean radius is defined as, [6]:

$$R_m = \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right), \tag{4}$$

where R_o is the outer radius and R_i is the inner radius of the friction clutch contact area.

The simplest viscosity force definition is as follows, [1], [4] and [5]:

$$F_v = \sigma_v v, \tag{5}$$

where σ_v represents a constant viscosity function which can be defined as:

$$\sigma_v = \frac{\eta A_C}{h}. \tag{6}$$

Here η represents the dynamic viscosity of the fluid, A_C is the contact area, and h is the gap height. All these parameters are considered to be constant; the only variable is the relative velocity v. The viscosity function σ_v can be temperature dependent. This viscosity model depends linearly on velocity.

The next 3 viscosity models are special cases of the so-called Carreau fluid definition. Here, the dynamic viscosity $\eta(\dot{\gamma})$ is shear rate dependent and the resulting viscous force is defined as follows, [7] to [10]:

$$F_v = \eta\left(\dot{\gamma}\right) A_C \frac{dv}{dh}. \tag{7}$$

The shear rate is:

$$\dot{\gamma} = \frac{dv}{dh}, \tag{8}$$

where the dynamic viscosity is defined as:

$$\eta\left(\dot{\gamma}\right) = \eta_0 \left(1 + \left(\lambda\dot{\gamma}\right)^2\right)^{\frac{n-1}{2}}. \tag{9}$$

As one can see, this viscosity definition is not linear anymore. Here η_0 is the dynamic viscosity at a given temperature and zero shear rate, the factor λ is a fluid (material) constant, and n is a flow behaviour index, which defines the type of the fluid, Fig. 3, as follows:

$$n \begin{cases} <1 & \text{pseudo plastic fluid} \\ =1 & \text{Newtonian fliud} \\ >1 & \text{dilatant fluid} \end{cases}. \tag{10}$$

The viscosity of pseudo plastic fluids decreases with increasing shear rate (for example; toothpaste). This type of fluid is also known as a shear thinning fluid. Newtonian fluids have a constant dynamic viscosity regardless of the changing shear rate (for example, water, gasses, some oils, etc.). Dilatant fluids react to increasing shear rate with increasing viscosity. They are also called shear thickening fluids. This type of fluid is less common.

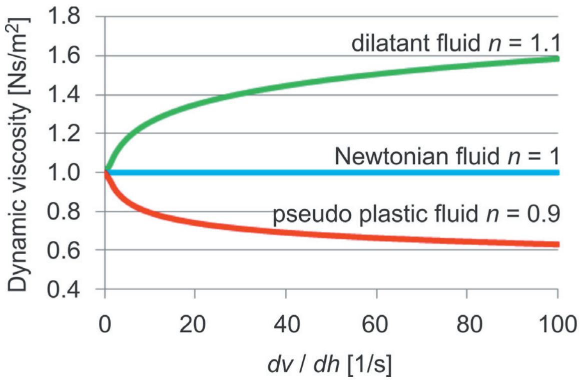

Fig. 3. *The Carreau fluid model – shear rate dependent viscosity for various flow behaviour index values*

In the case of the flow behaviour index $n = 1$, the definition for viscosity becomes:

$$\eta\left(\dot{\gamma}\right) = \eta_0. \tag{11}$$

In this case, the resulting viscous force becomes quite similar to the simple linear definition, i.e. Eq. (5).

In this research work, the following 3 values of the flow behaviour index were chosen: n_1 = 0.9, n_2 = 1 and n_3 = 1.1, shown in Fig. 3. These values were chosen just to represent the differences between various fluid types. One can use any values for the flow behaviour index. The parameter λ is in all cases set to $\lambda = 1$.

3 HEAT GENERATED DUE TO FRICTION

It is commonly known that friction generates heat, which is typically dissipated. The heat generated causes warming up of the friction contact materials. This temperature change changes the tribological characteristics of the friction contact pair. The change in tribological characteristics has a direct influence on the amount of transmitted torque in the slip phase and on the load capacity in the stick phase. It can also have a significant influence on friction induced dynamics, [1]. The influence is even stronger for wet friction clutch applications due to a change in the viscosity of the fluid.

The amount of heat generated depends on the resulting friction force F_f and relative velocity v, [11] to [14], as follows.

$$Q_{slip} = F_f v. \tag{12}$$

This heat is generated during sliding friction. A small amount of heat is also generated when the bodies in a friction contact stick. The proposed modified EP friction model is also dissipative for the stick phase, Dupont et al. [4] and [5]. The amount of generated heat in the stick phase is defined for the modified EP friction model as:

$$Q_{stick} = \sigma_1 \dot{z}. \tag{13}$$

The heat generated in the stick phase is typically negligible, compared to the heat generated in the sliding phase of a friction clutch.

$$Q_{stick} << Q_{slip} = Q. \tag{14}$$

The generated heat Q raised the temperature of the bodies in contact, of nearby bodies, and the surrounding environment. This temperature change depends on the thermal capacity of each body, the contact area, and the heat flux distribution between bodies. For simplification reasons the heat flux is assumed to be limited only to bodies in the friction contact. The total heat flux is defined as:

$$q = \frac{Q}{A_C}. \tag{15}$$

The heat flux into each body depends on the thermal conductivity of the body material and the heat transfer coefficient. Since the friction contact of an automotive friction clutch usually consists of two totally different materials with different thermal characteristics, the heat flux into each body must be defined separately. The heat flux into body 1 can be defined as:

$$q_1 = xq, \tag{16}$$

where, x represents the portion of the total heat flux q flowing into body 1. The heat flux q_2 into body 2 is then defined as:

$$q_2 = (1-x)q, \tag{17}$$

where

$$q = q_1 + q_2 . \tag{18}$$

When the friction contact is lubricated as in the case of a wet friction clutch, submerged in a fluid, the total heat flux must be divided into three parts. In a lubricated, submerged friction contact, a portion of the heat q_3 is conducted into the lubrication - cooling fluid. The total heat flux is then:

$$q = q_1 + q_2 + q_3. \tag{19}$$

To get accurate results for heat flux distribution, temperature change, and local temperature distribution, a full 3D thermal finite element (FE) and finite volume fluid dynamics (CFD) simulation is required, [15]. Since, in this research stage, the friction model used is rather simple and implemented as 1D for now, an accurate thermal calculation is not possible. Besides this, the target numerical code for multi-body system simulations does not currently allow the calculation of temperatures within the involved bodies. Therefore, since temperature is an important influencing factor, it will be addressed in more detail in future research work.

4 VALIDATION MEASUREMENTS

In order to validate the developed modified EP friction model, a special test bed was build. The test bed consists of two shafts. Each shaft is attached at

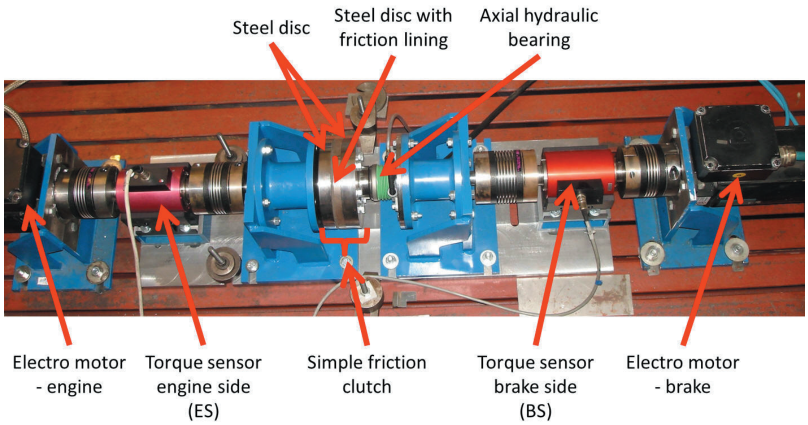

Fig. 4. *Test bed assembly for friction model validation*

one end to a torque sensor and an electro motor and at the other end to a steel disc, as shown in Fig. 4. One electro motor is used as a drive (engine side; ES) and the other one as a brake (brake side; BS). On one disc, an additional and replaceable disk is mounted. In that way, different replaceable discs with various commercial friction materials attached can be mounted. The friction contact occurs by bringing into contact the outer surface of both discs with the help of a hydraulic axial bearing. This friction contact represents a simplified friction clutch, shown in Table 1. No conventional automotive friction clutch was used since the conventional friction clutch is a complex multi-body system where parts are connected using various connection types (rivet connections, springs, dampers, shaft couplings, etc.). The properties of these connections are hard to predict since they depend on manufacturing procedure, tolerances, etc. In the simplified friction clutch, all parts are rigidly connected and the only unknowns are the tribological parameters of the contact.

Table 1. *Simplified clutch parameters*

R_o	0.095 [m]
R_i	0.060 [m]
μ_C	0.350 [/]
μ_S	0.420 [/]
v_S	0.500 [m/s]

This test bed with a simplified friction clutch enables the acquisition of angular velocities on both sides of the test bed, transmitted torques on both shafts, and the pressure (normal force) in the actuation system using a LabView interface. With the help of a LabView program and an electro motor controller, realistic conditions, similar to those in a vehicle powertrain, can be created and measured.

The tribological parameters were calculated for each measurement case from the measured values for the relative sliding velocity, the normal force, and the transmitted torque. For this research work, the values for the tribological parameters listed in Table 1 were used.

5 RESULTS

In this section, the results for three different friction clutch synchronization simulation cases are presented. The first case is an artificial one in order to show the difference between various viscosity models where various flow behaviour index values were used. For simplification reasons, the gap was considered constant in this case. In the second case, a variable gap height was considered and compared to the first case. The third case is a comparison to a real validation case presented in [1]. Compared to the original dry validation case, various viscosity terms were added to the friction model equation. Again, for simplification reasons the same tribological parameters were used as in the validation case. This approach gives a good quantitative comparison of the contributions of viscosity for wet friction clutch applications.

All cases represent a synchronization of two rotating bodies, as was the case during the validation measurements on the test bed. At the beginning, the engine side body rotates with a prescribed constant angular velocity and the break side body rests at zero rpm. After synchronization, both bodies rotate with the prescribed angular velocity of the engine side.

For all simulations of a given case (using various viscosity models), the same initial and boundary conditions were used. The contact area was also the same for all simulations.

5.1 Differences between Various Friction Definitions

In this example, the differences between various viscosity definitions are presented. All models have the same initial viscosity value η_0 at zero shear rate. The value for η_0 is artificial, but is nevertheless within the range of realistic values of oils used for wet friction clutches. For the Carreau fluid, 3 different flow behaviour index n values were used to represent the differences between various values, see Fig. 3.

In Fig. 5, one can see that the synchronization process is quite different for various viscosity definitions. The synchronization starts immediately due to the viscous force and proceeds rapidly after two seconds, when normal force is applied and the friction force prevails. No load is applied to the break side body. The synchronization duration depends on the viscosity definition. For shear thickening fluids – dilatant fluids, the process finishes much faster than

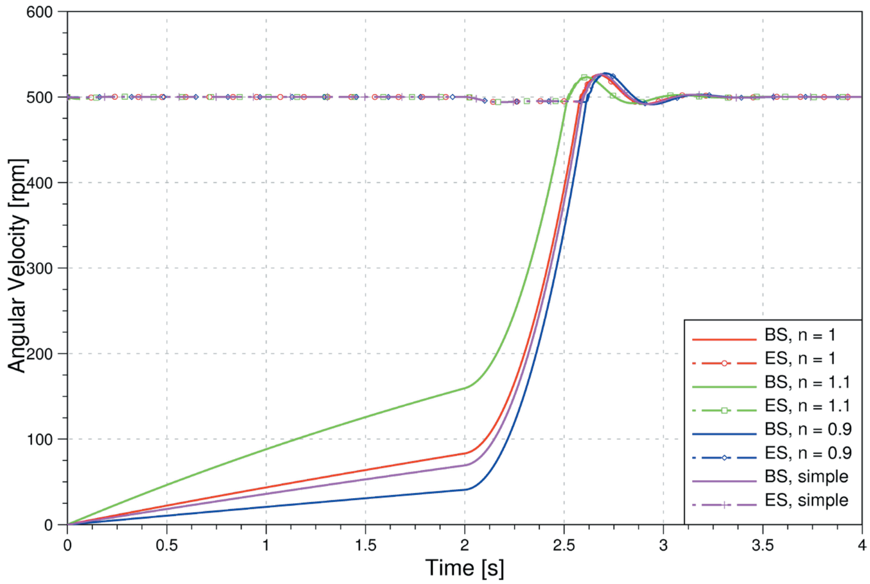

Fig. 5. *Synchronization of two rotating bodies with various viscosity definitions (BS –brake side, ES – engine side)*

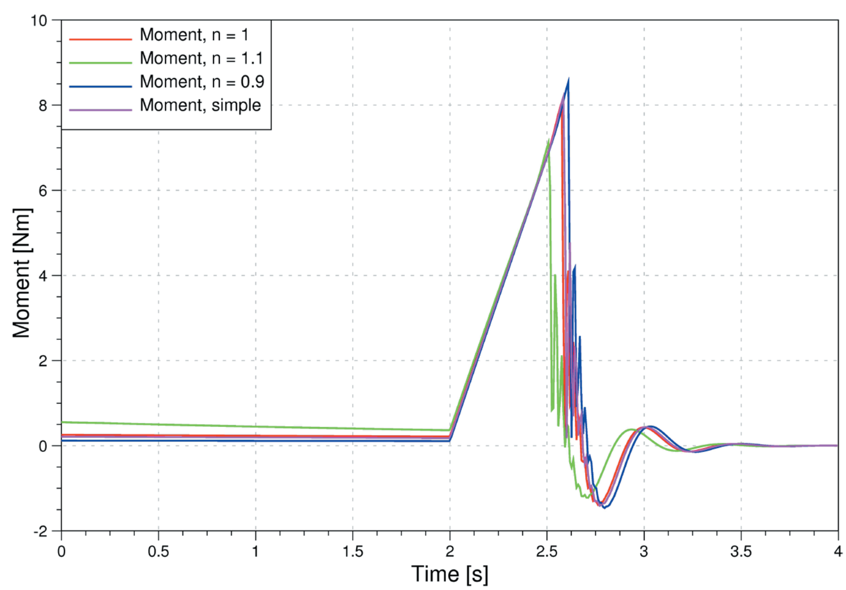

Fig. 6. *Transmitted torque through the friction contact during the synchronization – comparison of various viscosity definitions*

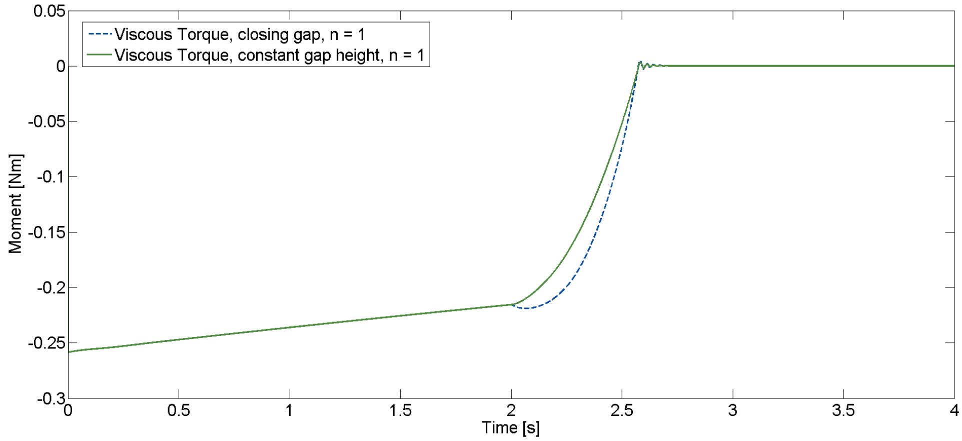

Fig. 7. *Comparison of viscous torque for constant and variable gap height for the Newtonian fluid definition*

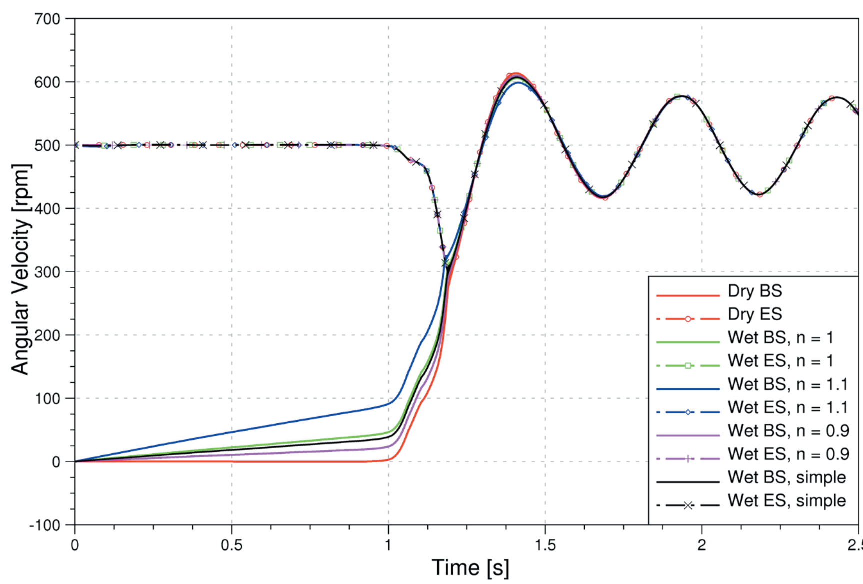

Fig. 8. *Comparison of angular velocities for dry validation case and additional viscosity term for wet applications for various viscosity definitions (BS –brake side, ES – engine side)*

for Newtonian and shear thinning fluids due to the increasing viscosity with increasing shear rate.

One can also see that the results for the simple viscous definition and for Newtonian fluids are quite similar.

In Fig. 6, the torques, transmitted through the friction contact, are plotted. As in Fig. 5, it can be seen that the viscosity definition significantly influences the synchronization process.

As already mentioned, the only input variable in this case is the relative sliding velocity of the bodies in contact, since the artificial gap is considered constant. Next, the results for a variable gap height are presented.

5.2 The Influence of a Variable Gap Height

In this example, the same load was used as in case 1. The gap height is normal force dependent and is, at the beginning, at an initial value of 1 mm. After the normal force is applied, the gap is proportionally reduced to a value of 0.1 mm. These values of gap heights were chosen for simplification reasons. In real cases, the end gap is 0 mm.

In Fig. 7, the resulting viscous torque is plotted for the Newtonian fluid. It can be seen that the difference between the model with a constant gap and the model with a variable gap is minor for the cases calculated here. When compared to the total torque transmitted through the friction contact, the influence of the

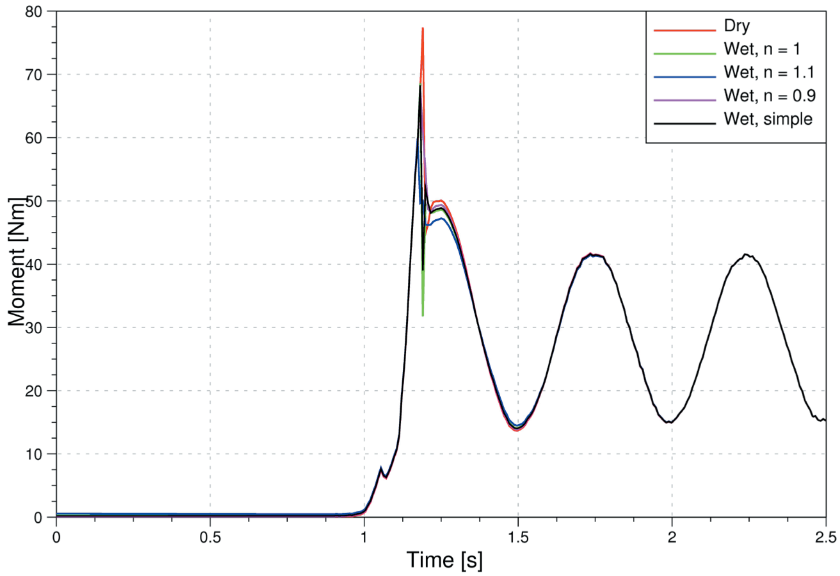

Fig. 9. *Comparison of transmitted torque through the friction contact for the dry validation case and the additional viscosity term for wet applications for various viscosity definitions*

variable gap height can be practically neglected since no visible difference in velocities and synchronization times was observed on the velocity plots. This is also true for all other viscosity models used in this research work.

For the conditions considered so far, one can say that the viscosity of fluids (for the fluid models used here) is mainly velocity dependent, since, when looking at the equation for the shear rate (Eq. 8), the velocity change is much higher than the gap height change.

5.3 Comparison to a Real Validation Case

To investigate the influence of the additional viscosity term in the modified EP friction model, a comparison to a real validation case was carried out. This case was used to validate the modified EP friction model for dry friction clutch applications, where very good agreement of measured and simulated results was achieved, [1]. To demonstrate the influence of various viscosity models, only part of the validation case was used, the open phase, the synchronization, and a small part of the locked phase.

In addition to the simulation results for the dry friction clutch, the results for all four viscous models are plotted. To be able to compare those results, the tribological parameters were assumed to be the same for dry and wet simulations, see Table 1.

In Fig. 8, it can be seen that viscosity influences the response only when the surfaces of the friction clutch are sliding relative to each other. As soon as the velocities of the bodies in contact synchronize (locked phase), the viscous part of the torque becomes zero and the system behaves as if it were dry. The same can be seen in Fig. 9.

6 CONCLUSIONS

In this research work, a theoretical and numerical investigation of simple viscosity models in a friction contact was carried out. The viscosity terms were added to a modified EP friction model equation, which was validated for dry friction clutch simulations and will be implemented into the commercial numerical tool for multi-body analyses AVL EXCITE.

As expected, the results confirm that the viscosity definition of the fluid may have a significant influence on the resulting friction force/torque and consequently on the friction clutch synchronization process. Therefore, such simulations can show only quantitative trends for various viscosity models. For qualitative, realistic results, an experimental validation would be necessary. These simple models probably do not provide highly accurate results since the real 3D geometry of the contact is not taken into account. For the cases presented here, it was also discovered that the shear rate is mainly velocity dependent since high velocity changes are present and the gap change is minimal.

This research also shows that the modified EP friction model enables simple addition of various force terms to the expression for the resulting friction force/torque, transmitted through the friction contact. These additional terms have no influence on the solution process of the main EP friction model equation. They also do not change the structure of the main equation. The only difference is in the resulting friction force/torque, transmitted through the friction contact and, of course, in the contributions of each term (friction force, viscous force, etc.).

In order to introduce at least an approximate temperature dependency, the underlying equations for friction generated heat are relatively simple. However, the hard part is to determine the heat flux into each body and the actual contact temperature, which decisively influences the tribology characteristics of the contact pair. To achieve this, a full 3D FE model and simulation would be needed. When using a wet clutch one would even need a multi-material model, including a CFD simulation. These important topics will need to be investigated in future work.

7 FUNDING

This research work was partly funded by the European Union, European Social Fund and SPIRIT Slovenia, the Slovenian Public Agency for Entrepreneurship, Innovation, Development, Investment and Tourism.

8 REFERENCES

[1] Petrun, T., Flašker, J., Kegl, M. (2012). A friction model for dynamic analyses of multi-body systems with a fully functional friction clutch. *Proceedings of the Institution Mechanical Engineers, Part K: Journal of Multi-Body Dynamics*. DOI:10.1177/1464419312464708.

[2] Bukovnik, S., Offner, G., Čaika, V., Priebsch, H.H., Bartz1, W.J. (2007). Thermo-elasto-hydrodynamic lubrication model for journal bearing including shear rate-dependent viscosity. *Lubrication Science*, vol. 19, no. 4, p. 231-245, DOI:10.1002/ls.45.

[3] Fajdiga, D., Glodež, S., Flašker, J. (1998). Numerical simulation of elastohydrodynamic lubricated line contact problems. *Strojniški vestnik – Journal of Mechanical Engineering*, vol. 44, no. 9-10, p. 285-296.

[4] Dupont, P., Armstrong, B., Hayward, V. (2000). Elasto-Plastic Friction Model: Contact Compliance and Stiction. *Proceedings of the American Control Conference*, Chicago, vol. 2, p.1072-1077.

[5] Dupont, P., Armstrong, B., Hayward, V. Altpeter, F. 2002). Single State Elastoplastic Friction Models. *IEEE Transactions on Automatic Control*, vol. 47, no. 5, p. 787-792, DOI:10.1109/TAC.2002.1000274.

[6] Drexl, H.J., (1997). *Kraftfahrzeugkupplungen: Funktion und Auslegung*. Die Bibliothek der Technik, Band 138, Verlag Moderne Industrie, Landsberg/Lech. (in German)

[7] Gotz, T., Parhusip, H.A. (2005). On an asymptotic expansion for Carreau fluids in porous media. *Journal of Engineering Mathematics*, vol. 51, no. 4, p. 351-365, DOI:10.1007/s10665-004-7468-1.

[8] Barrett, J.W., Liu, W.B. (1993). Finite element error analysis of a quasi-Newtonian flow obeying the Carreau or power law. *Numerische Mathematik*, vol. 64, p. 433-453, DOI:10.1007/BF01388698.

[9] Koh, J.H., Kwon, I., Jung, H.W., Hyun, J.C., (2012). Operability window of slot coating using viscocapillary model for Carrau-type coating fluid. *Korea-Australia Rheology Journal*, vol. 24, no. 2, p. 137-141, DOI:10.1007/s13367-012-0016-z.

[10] Strnadel, J., Simon, M., Machač, I. (2011). Wall effects on terminal faling velociti of spherical particles movingin a Carreau model fluid. *Chemical Papers,* vol. 65, no. 2, p. 177-184, DOI:10.2478/s11696-011-0005-6.

[11] Evtushenko, O.O., Pauk, V.I. (2002). Steady-state frictional heat generation on a periodic sliding contact. *Journal of Mathematical Science*, vol. 109, no. 1, p. 1266-1272, DOI:10.1023/A:1013757030298.

[12] Reibenschuh, M., Oder, G., Čuš, F., Potrč, I. (2009). Modelling and analysis of thermal and stress loads in train disk brakes – braking form 250 km/h to standstill. *Strojniški vestnik – Journal of Mechanical Engineering*, vol. 55, no. 7-8, p. 494-502.

[13] Živanović, Z., Milić, M. (2012). Thermal Load of Multidisc Wet Friction Assemblies at Braking Regime. *Strojniški vestnik – Journal of Mechanical Engineering,* vol. 58, no. 1, p. 29-36, DOI:10.5545/sv-jme.2009.111.

[14] Yevtushenko, A.A., Kuciej, M. (2012). One-dimensional thermal problem of friction during braking: The history of development and actual state. *International Journal of Heat and Mass Transfer,* vol. 55, no. 15-16, p. 4148-4158, DOI:10.1016/j.ijheatmasstransfer.2012.03.056.

[15] Tic, V., Lovrec, D. (2012). Design of Modern Hydraulic Tank Using Fluid Flow Simulation. *International Journal of Simulation Modelling*, vol. 11, no. 2, p. 77-88, DOI:10.2507/IJSIMM11(2)2.202.