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Poulios, Konstantinos; Svendsen, G.; Hiller, Jochen; Klit, Peder

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Coefficient of Friction Measurements for Thermoplastics and Fiber Composites under Low Sliding Velocity and High Pressure

K. Poulios[1]*, G. Svendsen[1], J. Hiller[1], P. Klit[1]

[1]*Affiliation 1: Technical University of Denmark, Department of Mechanical Engineering, Nils Koppels Allé , Building 404, 2800 Kgs. Lyngby.*

Abstract

Friction materials for typical brake applications are normally designed considering thermal stability as the major performance criterion. There are however brake applications with very limited sliding velocities, where the generated heat is insignificant. In such cases it is possible that friction materials which are untypical for brake applications, like thermoplastics and fiber composites, can offer superior performance in terms of braking torque, wear resistance and cost than typical brake linings. In this paper coefficient of friction measurements for various thermoplastic and fiber composite materials running against a steel surface are presented. All tests were carried out on a pin-on-disc test-rig at a fixed sliding speed and various pressure levels for both dry and grease lubricated conditions.

Keywords: coefficient of friction, thermoplastics, fiber composites, low speed.

*Corresponding author: Konstantinos Poulios (kopo@mek.dtu.dk).

1. INTRODUCTION

In general, the selection or design of friction materials for brake applications is based on factors like the coefficient of friction, the wear resistance, the thermal stability, the tendency for noise generation and the material cost.

The function of typical brake applications such as vehicles, crane winches or the wind turbine drive train, is to bring the corresponding mechanical systems to standstill by converting their kinetic energy to heat. Such brakes are dimensioned for reaching a certain peak temperature and the friction materials to be used are required to exhibit good thermal stability up to this temperature level. The possible friction material choices include sintered or organic composites or fiber composites specially designed for thermal stability [1].

A different kind of brake applications are the so called positioning or holding brakes. Their function is to maintain the relative position between two components and occasionally permit a relative motion at low sliding speeds. The heat generation in such cases is usually negligible and the coefficient of friction and low-noise operation are the main criteria for the selection of friction materials.

One purpose of this paper is to present the particularities of using a pin-on-disc test-rig for studying the coefficient of friction with respect to positioning brakes applications. Its main objective is however to present coefficient of friction measurements for different thermoplastics and fiber composites running against steel.

2. TESTING EQUIPMENT AND METHODS

Fig. 1 shows a simplified illustration of the used pin-on-disc test-rig. The normal load on the pin (1) is determined by the position of the mass (2) on the loading arm (3). The rotating disc (4) is driven by an electric motor (5). The rotation of the motor housing with respect to the test-rig frame (6) is restricted exclusively through the arm (7), the wire (8) and the load cell (9).

One of the particularities of positioning brakes is their intermittent operation in both sliding directions. In order to take this effect into account the sliding direction had to be alternated periodically during the testing. In order to enable the friction measuring in both rotational directions of the disc the wire (8) and the load cell (9) are preloaded through the weight of an an additional mass (10).

In contrast to typical brake applications, positioning brakes operate at much lower sliding speeds and permit much higher pressure levels. A gearbox of transmission ratio 30 integrated in the test-rig motor was used in order to permit testing speed as low as 2 mm/s and pressures of up to 20 MPa.

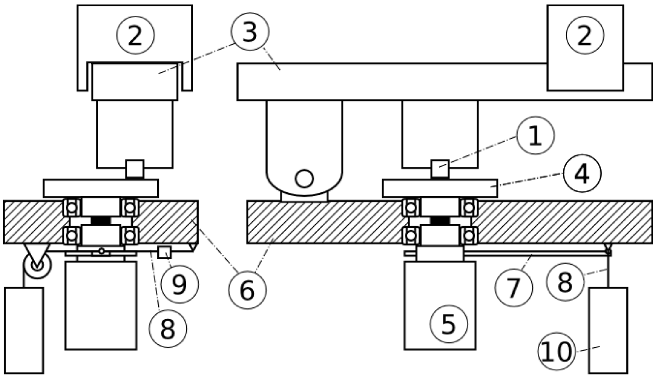

Figure 1: *Simplified illustration of the pin-on-disc test-rig.*

Part of the testing procedure is the calibration of the normal force applied to the pin and the friction force recording load cell (9). The normal force applied to the pin is determined by an initial load due to the weight of the loading arm and a variable load proportional to the linear position of the corresponding mass (2). A calibration line correlating the mass position with the normal force was determined by placing a compressive load cell below the pin and moving the mass to different positions. During the tests the position of the mass is adjusted through a stepper motor. The calibration of the load cell (9) was done in-place by adding and removing weight to the corresponding mass (10).

The dimension of the pins used for all tests was 10 mm in diameter. The pin specimens were manufactured from the different materials to be tested by turning larger material samples in form of plates or bars. The disc material was alloy steel 34CrNiMo6 in unhardened condition and the disc surface was turned with surface roughness corresponding to arithmetic average values (Ra) varying from 0.7 to 1 μm for the different discs used in the tests.

Because of the flat form of the pin surface in contact with the disc, a good alignment between the pin and the disc surfaces is very important. For this purpose, pressure sensitive paper was used before every test in order to verify a uniform contact through the whole pin surface. Fig. 2 shows two photos of a partial and a full pin contact.

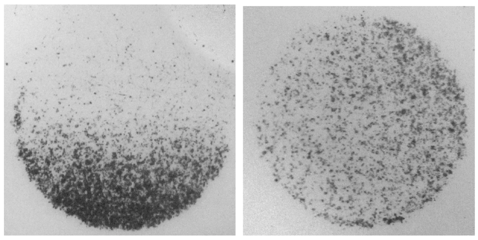

Figure 2: *Pin imprint on pressure sensitive paper for a case of bad (left) and good (right) alignment between the pin and the disc.*

For each of the tested materials an initial running-in test on the disc track with diameter of 80 mm ensured that any remaining geometrical misalignment between the pin and the disc would be eliminated through the wear process. During this initial running-in, the nominal pressure applied to the pin was 6 MPa and the sliding direction was alternated every 0.9 m.

Following, the pin was moved on a disc track with diameter of 102 mm where the actual measurement took place. On this track a second running-in was carried out under the same conditions as the first one until the recorded coefficient of friction approximated a steady state. After that state was reached, the pressure was varied from 3 MPa to 18 MPa in steps of 3 MPa. For each pressure level, a sliding distance of 10 m was covered with a reciprocation stroke length of 0.9 m.

After this phase, the pin was moved to a third track at disc diameter of 124 mm that was covered with an approximately 1 mm thick layer of grease. During a similar running-in period like in the previous cases, the grease layer is removed from the track due to the motion of the pin. However, a small quantity of lubricant will remain in the contact either because of the strong adhesion of a very thin layer of lubricant with the surfaces or because of the lubricant that is stored in the porous structure of the tested materials. It was observed in all tests that were carried out, that the pin runs under boundary lubrication conditions for a very long sliding distance even if no further grease is added to the contact. The base oil viscosity of the grease used in the tests was 46 mm^2/s and its consistency corresponded to NLGI number 2.

3. RESULTS

Table 1 summarizes the tested friction materials that will be presented in this section. All five materials are representative samples of commercially available products of their category.

Table 1: *Description and abbreviated names of the tested materials*

Symbol	Description
PET	Polyethylene terephthalate
PA 6	Polyamide 6
PA 66	Reinforced polyamide 6.6
FCA	Coarse fiber composite
FCB	Fine fiber composite with aramid

The first three materials are semi-crystalline thermoplastics and the last two fiber composites. The PA 66 thermoplastic is reinforced with glass fibers and contains MoS_2 as internal lubricant. The coarse fiber composite FCA consists of a woven textile impregnated in resin and the fine one, FCB, has a more paper like structure.

Fig. 3 shows the evolution of the coefficient of friction during the second running-in phase of the three thermoplastics without lubrication and Fig 4. shows the corresponding curves for the two fiber composites.

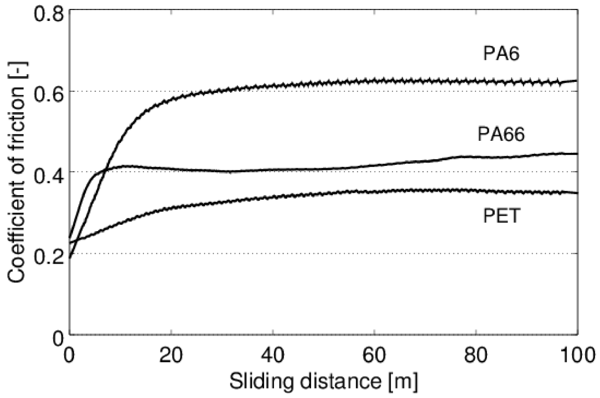

Figure 3: *Running-in of thermoplastics without lubrication.*

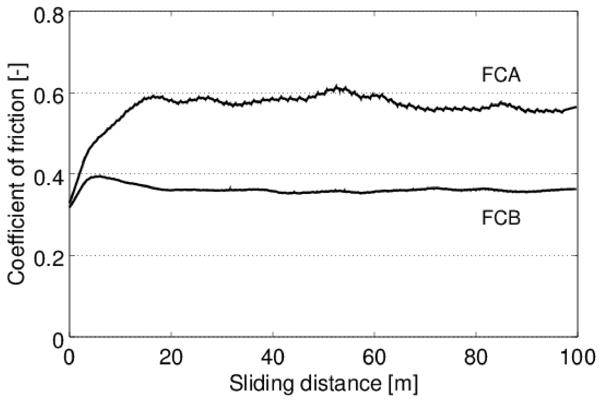

Figure 4: *Running-in of fiber composites without lubrication.*

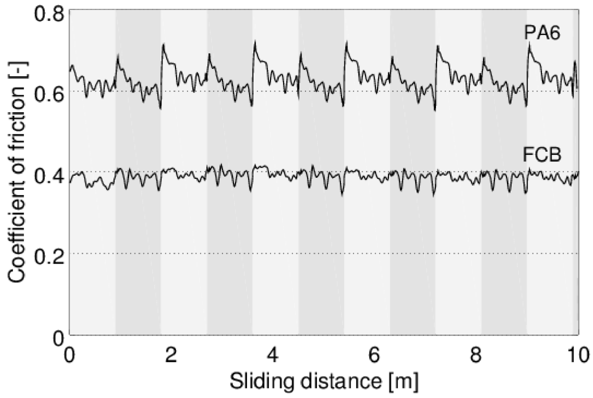

Figure 5: *Steady state coefficient of friction measurement for PA 6 and FCB at the pressure of 6 MPa.*

It must be noted that the filtered curves shown in these diagrams, demonstrate the long-term evolution of the coefficient of friction. However, after each alternation of the sliding direction a short-term transition may occur due to e.g. a reorientation of the surface asperities. This phenomenon was mostly pronounced for PA 6. Fig. 5 shows the unfiltered coefficient of friction measurement during the steady state phase for PA 6 and FCB. The light and dark background of the diagram illustrates the different sliding directions. It is evident that in the case of PA 6 the coefficient of friction increases significantly after each direction change.

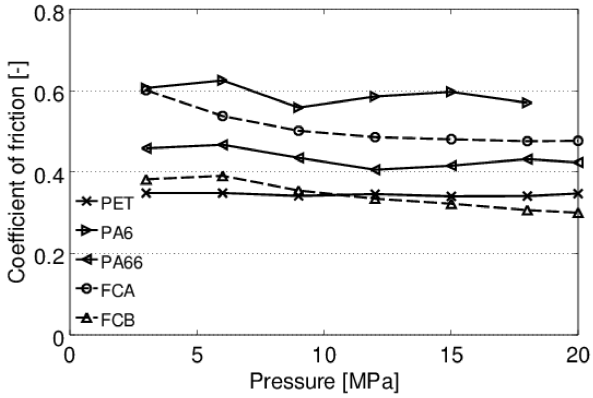

Figure 6: *Impact of the nominal contact pressure on the average coefficient of friction without lubrication.*

Fig. 6 shows the variation of the average coefficient of friction for all five materials over the tested pressure range. It seems that the measured coefficient of friction for all three thermoplastics is quite unaffected by the pressure level. Especially the PET thermoplastic exhibits a completely constant coefficient of friction over the whole pressure range. The measured coefficient of both fiber composites exhibits the same negative tendency with increasing pressure. The corresponding running-in and pressure variation measurements with presence of grease lubricant are illustrated in Figures 7, 8 an 9.

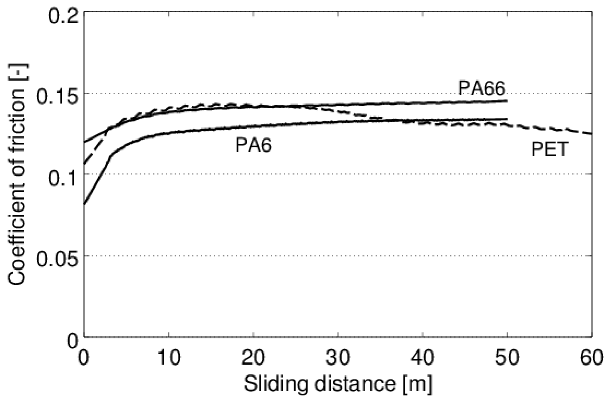

Figure 7: *Running-in of thermoplastics with grease lubrication.*

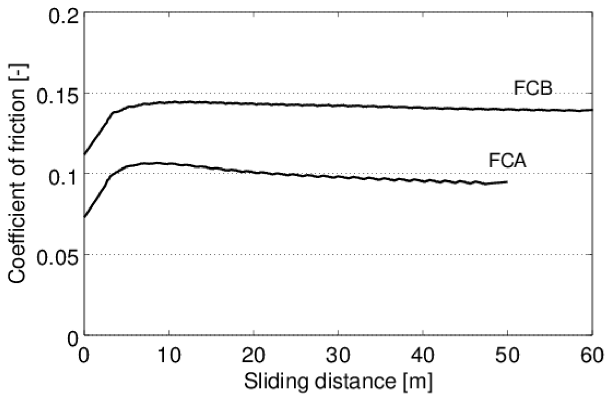

Figure 8: *Running-in of fiber composites with grease lubrication.*

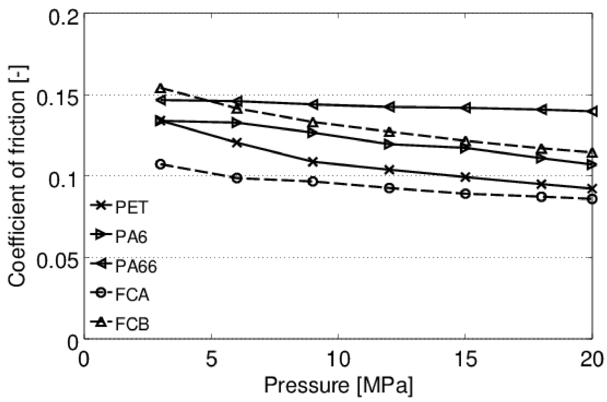

Figure 9: *Impact of the nominal contact pressure on the average coefficient of friction with lubrication.*

4. DISCUSSION

In both dry and lubricated tests the observed coefficient of friction increases during the running in period. In the lubricated case these increase is mainly due to the transition from mixed to boundary lubrication, as the quantity of the grease on the track is reduced to a minimum. However, the increasing coefficient of friction during the dry running-in occurs due to less evident phenomena.

Formation of transfer layers is the first phenomenon to be mentioned [2], [3]. Either material from the pin can adhere to the disc surface or particles from the disc like e.g. iron oxides can adhere to the pin surface. In this way the increased affinity between the two contacting surfaces normally yields to an increase of the interfacial shear strength.

Changes in the topography of the contacting surfaces during the running-in in period is the second factor that contributes to the increasing coefficient of friction. Through the wear process the surfaces tend to become more conformal than initially. This may give rise to a higher real contact area between the two surfaces and an increased coefficient of friction.

All results presented in the previous section are related to the nominal pressure P that is an average pressure over the apparent contact area A_a corresponding to the pin diameter of 10 mm. More relevant for the understanding of the tribological system is however the local pressure p which is distributed over the real contact area A_r. The real contact area is load dependent and much smaller than A_a. The macroscopically observed Amonton's first hypothesis about a pressure independent coefficient of friction can be explained only through consideration of the real contact area [4], [6]. Studying less ideal cases like the pressure dependent coefficients of friction shown in Fig. 6 and Fig. 9 requires consideration of the real contact area A_r as well.

Even if at macroscopical level the coefficient of friction is either independent or weakly dependent on the nominal pressure, microscopically the constitutive law for the interfacial shear may not be proportional to the normal stress. The following constitutive equation can be used for the shear strength of the interface [7]:

$$\tau = \tau_0 + \alpha p \qquad \text{(Eq 1)}$$

In this equation τ_0 is a constant interfacial shear stress related to adhesion and α is a proportionality constant of the pressure dependent term.

The normal force F_N in the contact can be defined as the integral of the local pressure p over the real contact area A_r and the friction force F_T as the integral of the shear stress τ. By integration of Eq. 1 follows:

$$F_T = \tau_0 A_r + \alpha F_N \qquad \text{(Eq 2)}$$

and the coefficient of friction can consequently be expressed as:

$$\mu = \tau_0 A_r / F_N + \alpha \qquad \text{(Eq 3)}$$

Eq. 3 reveals the importance of the real contact area A_r for the observed coefficient of friction μ. During the running-in process A_r increases for constant F_N, causing the coefficient of friction to increase as well. Formation of transfer layers may additionally provoke changes in the constitutive law constants τ_0 and α [3].

The variation of the nominal contact pressure in Fig. 6 and Fig. 9 corresponds to variation of F_N in Eq. 3. The coefficient of friction μ will remain constant only if the real contact area A_r increases proportionally to F_N or if τ_0 is negligible (in comparison to the local pressure p). In most practical cases, τ_0 can not be neglected and therefore it is important to study the relation between A_r and F_N. A constant ratio A_r / F_N means that the average local pressure in the real contact area remains constant with changing normal load. This is the case for both the model of Bowden and Tabor [4] that assumes fully plastic deformation and the fully elastic models of Archard [5] and Greenwood and Williamson [6]. In the more general case, A_r and F_N aren't proportional and their ratio depends on the elastic properties of the contacting bodies and their surface topographies.

The lower friction levels observed under boundary lubrication conditions are attributed to the impact of the lubricant film on the constants τ_0 and α in Eq. 1. However, changes of these two constants are not sufficient for explaining the increased or reduced dependence of the coefficient of friction on the pressure level that is observed with the addition of lubricant. In comparison to Fig. 6, the coefficient of friction dependence on the pressure in Fig. 9 becomes stronger for some materials (e.g. PET, PA 6) and weaker for others (e.g. FCA). A less evident factor affecting the slope of the coefficient of friction curves in these figures is the impact of lubrication on the surface topography. The different wear mechanisms under lubricated conditions yield to different surface topographies in comparison to the dry case. The modified surface topography affects the evolution of the ratio A_r / F_N in Eq. 3. Fig. 10 shows two optical microscope pictures of the fiber composite FCA surface after dry and lubricated runs. From the different wear traces that are visible on the pictures one recognizes how the presence of lubricant can yield to different surface topographies.

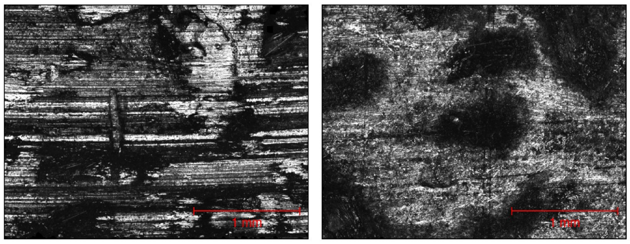

Figure 10: *Microscope pictures for FCA after dry (left) and lubricated testing (right).*

Fig. 11 shows the surface topographies obtained with an infinite focus optical system for the three thermoplastics after the dry testing. At least in the first two cases, it is easy to recognize a characteristic wavelength in the vertical direction which corresponds to the turning marks of the steel counter surface.

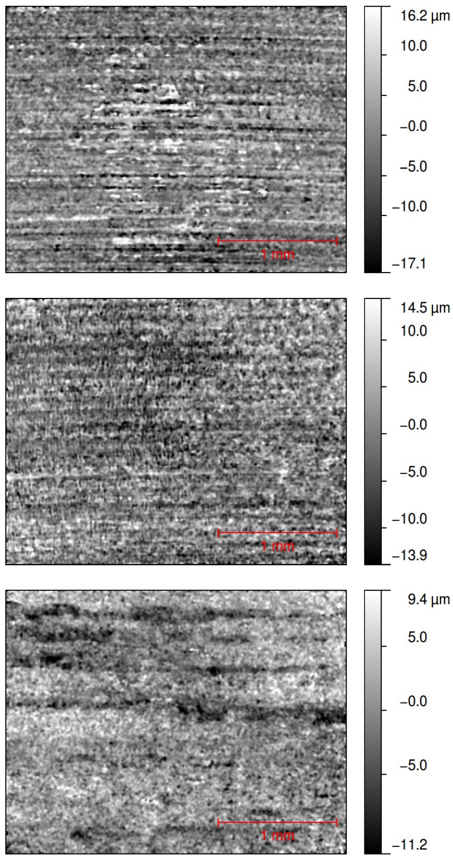

Figure 11: *Surface topographies for thermoplastic after dry runs. PET (top), PA6 (middle), PA66 (bottom).*

5. CONCLUSIONS AND FUTURE WORK

Friction materials from two very dissimilar material categories were tested under conditions that are relevant for holding brakes applications. Some of the practical conclusions that could be drawn are listed below:

- The measured coefficient of friction for all tested thermoplastics appeared to be independent from the nominal pressure under dry conditions.
- The coefficient of friction for fiber composites dropped significantly with increasing nominal pressure under dry conditions.
- With grease lubrication in all cases except one the coefficient of friction dropped with increasing pressure.
- For the design of holding brake applications with friction materials similar to the ones tested here, a coefficient of friction of 0.15 should be considered as an absolute maximum in case that grease reaches the contact surfaces even occasionally.

In this work, some interesting cases concerning the coefficient of friction dependence on pressure were identified. In order to verify the corresponding assumptions and the qualitative explanations presented in this work the following tasks are proposed for future work:

- The relation between the normal load and the real contact area could be studied numerically in order to permit a quantitative comparison between the theoretical friction models and the experimental results.
- An experimental method for estimating the real contact area either in-situ or through analysis of the worn surface topographies would contribute significantly to a more quantitative explanation of the here discussed phenomena.

6. REFERENCES

1. D. Chan, G. W. Stachowiak, Review of Automotive brake Friction Materials, Proceedings of the Institution of Mechanical Engineers, PartD: Journal of Automobile Engineering, 2004, vol. 218, pp. 953-966.

2. S. H. Rhee, K. C. Ludema, Mechanisms of Formation of Polymeric Transfer Films, Wear, 1978, vol. 46, pp. 231-240.

3. Z. Zalisz, P. H. Vroegop, R. Bosma, A Running-in Model for the Reciprocating Sliding of Nylon 6.6 against stainless steel, Wear, 1988, vol. 121, pp. 71-93.

4. F. P. Bowden and D. Tabor, The Friction and Lubrication of Solids, Oxford Univ. Press, 1958.

5. J. F. Archard, Elastic Deformation and the Laws of Friction, Proceedings of the Royal Society of London, series A, 1957, vol. 243, pp. 190-205.

6. J. A. Greenwood and J. B. P. Williamson, Contact of Nominally Flat Surfaces, Proceedings of the Royal Society of London, series A, 1966 vol. 295, pp. 300-319.

7. G. He, M. H. Müser, M. O. Robbins, Adsorbed Layers and the Origin of Static Friction, Science, 1999, vol. 284, pp. 1650-1652.