Approach for Predicting the Friction Temperature between Thermoplastics in Dry-Running Sliding Friction with Periodically Recurring Contacts

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ABSTRACT A variety of analytical and numerical approaches for calculating the friction temperature have been developed in the past. However, none is capable to estimate the friction temperature of thermoplastic friction pairings. Therefore, a semi-analytical model for predicting the friction temperature has been developed. Dry friction and periodically recurring contact is a premise. In the article the derivation is shown und influencing parameters are explained. A validation is made by experimental studies on a conveyor system. The model can be applied to sliding chain conveyor as well as perspectively similar tribological systems.

KEYWORDS sliding friction, dry running, friction temperature, calculation model, tribology, thermoplastics, polymer

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1. Motivation

The processing and packaging industry demand high production rates, lubricant-free operations as well as hundred percent availability on their machines. With increasing requirements, the current conveying technology used for transferring goods is pushed to certain limits. One of them is thermal failure of thermoplastic sliding chains or rails due to a high friction temperature (see Figure 1). A lack in dimensioning fundamentals results in unknown temperatures in sliding contacts. Temperatures above the softening temperature of involved materials leads to melting as well as wear and in the end in malfunction of the system.

If a thermal design is done, usually the PV value is taken, which is fine for bearings. But for other sliding configurations it is insufficient because the PV value represents one geometrical configuration, which is not corresponding to a real system configuration. Therefore, it is reasonable to design friction pairings by a temperature criterion.

A variety of approaches for calculating the friction temperature of a dry-running friction pairing can be found in literature [Blo37, CaJ59, Ash91, Ken01]. With progressing computing possibilities, models become more complex [GeW85, Ash91, Hou00, Lar09, Osm09, Bof12] and require software for solving mathematical problems. The vast majority of models are evaluated

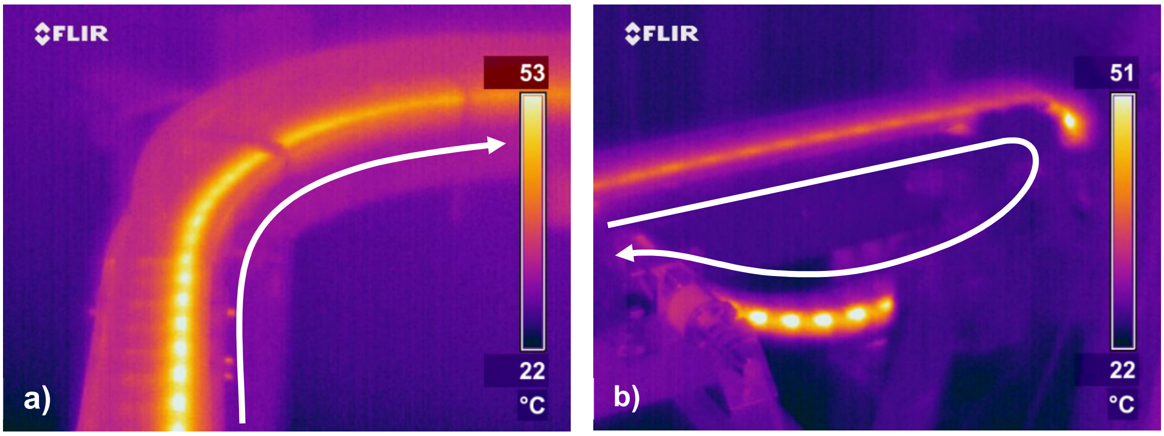

Figure 1: Thermal images of a running sliding chain conveyor: a) horizontal slide curve and b) redirection with chain bag at the drive unit

and compared to calculations in which the material of the two bodies is made of metal. There are a few works that deal with metal-plastic pairings. Models for plastics or their verification on plastic pairings cannot be found.

While conveyor system suppliers are increasingly providing capable software to design their systems, in the engineering practice, feasible and fast calculations are common. Therefore, a simple practicable semi-analytical model is developed.

2. Derivation of the Semi-Analytical Model

In sliding chain conveyors, the chain links ideally slide at a constant speed on a stationary slide rail. Their contacts vary depending on the type of chain and the conveyor section. Looking on a track section, the sliding contacts have a regular pattern, which is defined by the chain links or the chain pitch. Due to the uniform speed and the regular pattern, the system "sliding chain — slide rail" can be converted to a two-body system, which consists of one chain link (body 1) and a slide rail with a length of the chain pitch (body 2). This approach is shown in Figure 2.

Body 1 slides periodically recurring over body 2. The time for a complete sliding over is defined by the periodic time t_p. Looking at an infinitesimal small point on body 2, two states appear during one period: The bodies 1 and 2 are in contact (contact state) and body 2 is exposed to the environment (free state). In both states the surface temperature of the stationary body 2 is supposed to be equal. Therefore, it is assumed that the friction temperature T_f and the frictional

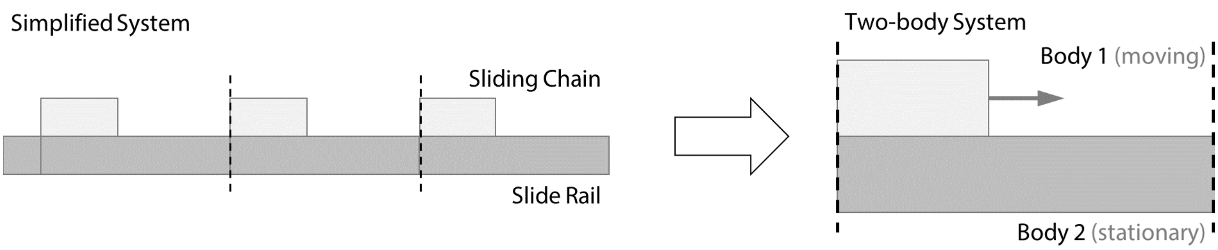

Figure 2: Abstraction of a simplified "sliding chain — slide rail" system to a two-body system

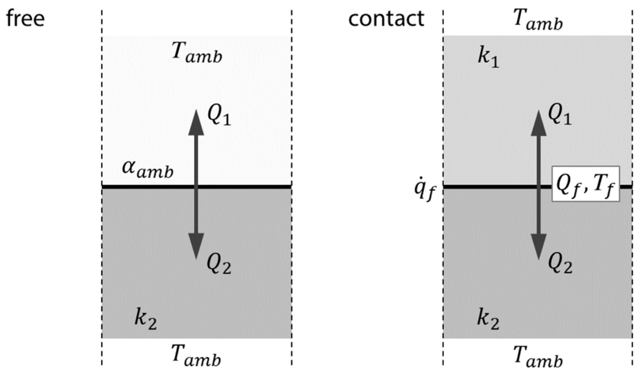

Figure 3: Two-body system with discrete sections and their thermodynamic states (stationary) of an infinitesimal small section

heat Q_f remain the same on both states. For the thermodynamic system following assumptions are made: It is in steady state, the ambient temperature T_{amb} is the same at any point and the highest occurring temperature is the friction temperature in the sliding surface which remains constant in all states.

In both states, a heat Q_2 flows from the sliding surface through body 2 by heat conduction, which has the ambient temperature on the side opposite side. In free state, the sliding surface loses heat Q_1 due to convection and radiation (combined as α_{amb}). In contact state, the heat Q_1 is only defined by heat conduction through body 1. Heat conduction through the bodies is affected by thermal transmittance k. The two thermodynamic states are shown in Figure 3. The periodic time is summed up by the free state duration t_{free} and the contact state duration t_c:

$$t_p = t_{free} + t_c \,. \tag{1}$$

The heat quantities in contact state can be defined as follows:

$$Q_{f,c} = \dot{q}_f \, \mathrm{d}A \, t_c \tag{2}$$

$$Q_{1,c} = k_1 \left(T_f - T_{amb}\right) \mathrm{d}A \, t_c \tag{3}$$

$$Q_{2,c} = k_2 \left(T_f - T_{amb}\right) \mathrm{d}A \, t_c \tag{4}$$

In free state the heat quantities are:

$$Q_{1,free} = \alpha_{amb} \left(T_f - T_{amb}\right) dA \, t_{free} \tag{5}$$

$$Q_{2,free} = k_2 \left(T_f - T_{amb}\right) dA \, t_{free} \tag{6}$$

If heat flows within a period considered in an equilibrium, we obtain a heat balance

$$\dot{Q}_f = \dot{Q}_1 + \dot{Q}_2 \,, \tag{7}$$

with the heat flows

$$\dot{Q}_f = \frac{Q_{f,c}}{t_p} \,, \tag{8}$$

$$\dot{Q}_1 = \frac{Q_{1,free} + Q_{1,c}}{t_p} \tag{9}$$

and

$$\dot{Q}_2 = \frac{Q_{2,free} + Q_{2,c}}{t_p} \,. \tag{10}$$

After applying the heat flows of equations 8 till 10 in equation 7, reducing the heat quantities with equations 2 till 6 and transposing, the following relationship unfolds:

$$\dot{q}_f \frac{1}{T_f - T_{amb}} \frac{t_c}{t_p} = k_1 \frac{t_c}{t_p} + \alpha_{amb} \frac{t_{free}}{t_p} + k_2 \frac{t_c + t_{free}}{t_p} \tag{11}$$

The ratio of contact duration to period duration is introduced as contact ratio

$$\psi = {}^{t_c}/_{t_p} . \tag{12}$$

Inserting equations 1 and 12 in equation 11 and transposing, results in the equation for the friction temperature:

$$T_f = T_{amb} + \frac{\psi}{k_1\,\psi + k_2 + \alpha_{amb}\,(1-\psi)}\;\dot{q}_f . \tag{13}$$

By introducing a contact coefficient

$$C_c = \frac{\psi}{k_1\,\psi\; + k_2 + \alpha_{amb}\,(1-\psi)}. \tag{14}$$

and the heat flux induced by sliding friction

$$\dot{q}_f = \mu\, p\, v\,, \tag{15}$$

a generalized equation for the friction temperature can be obtained:

$$T_f = T_{amb} + C_c\, \mu\, p\, v \tag{16}$$

Equation 16 was firstly presented by Lancaster [Lan73], who defined C_c as a constant characterizing thermal material properties and geometrical configuration, but could describe it only by examined values. His characterization of C_c is still valid. With the new approach it can be calculated. Its unit is that of a specific thermal resistance: $(m^2K)/W$.

Based on equation 13, the friction temperature depends on the frictional heat flux $\dot{q}_f$, the contact ratio ψ, the thermal transmittance of the two involved bodies (k_1 and k_2) and the heat transfer coefficient at the sliding surface (α_{amb}). The frictional heat flux $\dot{q}_f$ which arises in the contact zone by friction depends on the coefficient of friction μ, the contact pressure p and the sliding speed v.

3. Model Parameters

3.1. Contact Ratio

The contact ratio has a significant influence on heat input of the bodies. Small values imply less heat flow into participating bodies and more convective heat release.

At a uniform sliding speed, the contact ratio can be calculated by the length of moving body 1 (contact length l_c) to the length of stationary body 2 (sliding length L):

$$\psi = {}^{l_c}/_{L} \tag{17}$$

For a sliding chain conveyor, the sliding length L is related to the chain pitch.

3.2. *Thermal Transmittance*

The thermal transmittance coefficients k_1 and k_2 are characteristic values which indicate the heat flow from the contact face through the respective body. The body can consist of several parts with different surfaces. The thermal transmittance is characterized by geometry, thermal conductivities of the body parts, thermal resistances between the body parts and heat transfer coefficients of the body part surfaces.

In order to determine the thermal transmittance of a body, the maximum occurring temperature T_{max} and the corresponding heat flux $\dot{q}_c$ at the contact face is required. Thus, the thermal transmittance of the contact face can be obtained:

$$k = \frac{\dot{q}_c}{T_{max} - T_{amb}} \tag{18}$$

Determining the thermal transmittance for complex body geometries, a steady state thermal analysis in a FEM program is recommended. The heat flux at the contact face must be specified and the maximum temperature is determined by FEM. Here, the heat flux does not necessarily have to gain realistic values, since calculations are based on simple heat conduction and the maximum temperature is considered. A higher heat flux leads to imaginary higher temperatures, but the thermal transmittance remains constant.

For a sliding chain conveyor, value k_1 is characterized by the chain and in case of material loading additionally also by the goods. For k_2, the slide rail and guiding/supporting profiles must be considered.

3.3. *Heat Transfer Coefficient*

The heat transfer coefficient α_{amb} can be calculated by Nusselt number. Using the gap length between two adjacent contacts as characteristic length, air for the thermal fluid characteristics and the fact that the temperature influence on fluid characteristics of air is negligible between 20 and 150 °C, the following simplified equation is obtained:

$$\alpha_{amb} = 3.89\sqrt{v/(L - l_c)} \tag{19}$$

v is the sliding speed. Speed and lengths must be applied in SI units (m/s and m), so that the unit W/(m² K) is preserved for the heat transfer coefficient.

4. Validation

A Multiflex sliding chain conveyor similar to FlexLink X85 or Bosch Rexroth VarioFlow 90 system was selected for validation. Because the highest stress occurs in the horizontal curve and therefore the highest temperature is expected [Aue06], a curve section was chosen. For that purpose, a test conveyor has been built up, which is shown in Figure 4. The contacts of the curve section are illustrated in Figure 5.

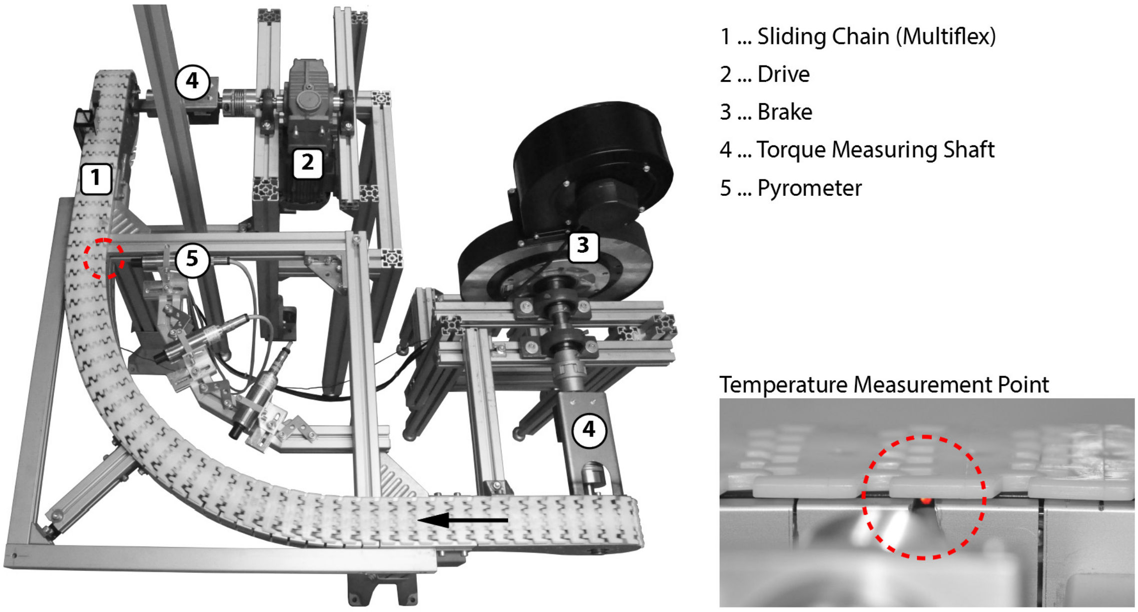

Figure 4: Test conveyor system

Crucial immutable configuration data of the considered conveyor section are:

Material: PE-UHMW (sliding chain), POM (slide rail) and Aluminum (mount)

$L = 25.4$ mm, $l_c = 1$ mm, $h_c = 4$ mm, $T_{amb} = 25$ °C

The contact ratio is calculated by equation 17:

$\psi = 0.031$

In a FEM analysis with a chosen heat flux of 20.000 W/(m² K), the imaginary maximum temperatures are obtained for both bodies (see Figure 6), so that the thermal transmittances are determined by equation 18:

$k_1 = 125\ \mathrm{W/(m^2\ K)}$

$k_2 = 146\ \mathrm{W/(m^2\ K)}$

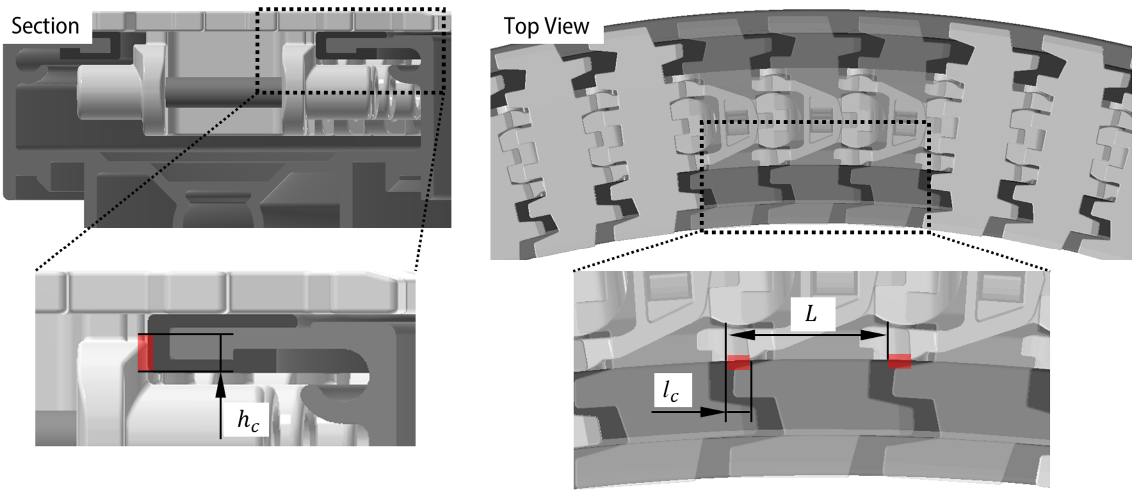

Figure 5: Section and top view of a sliding chain curve (Multiflex system)

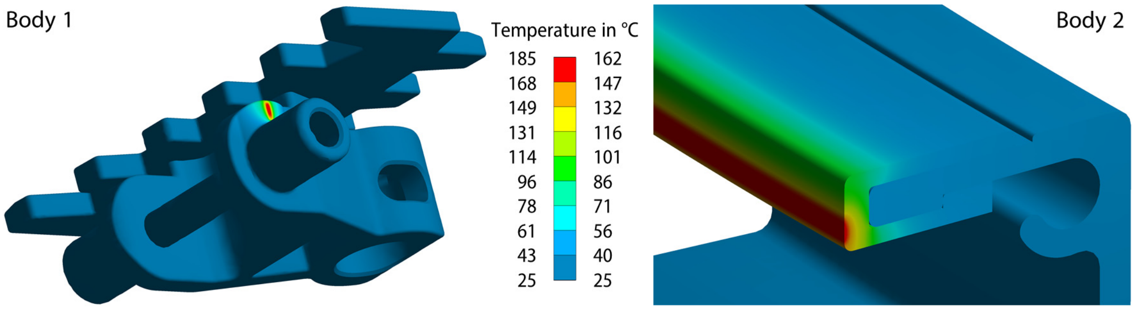

Figure 6: Result of the FEM analysis to obtain the imaginary temperature for a chosen heat flux of 20.000 W/(m² K) in the contact zone

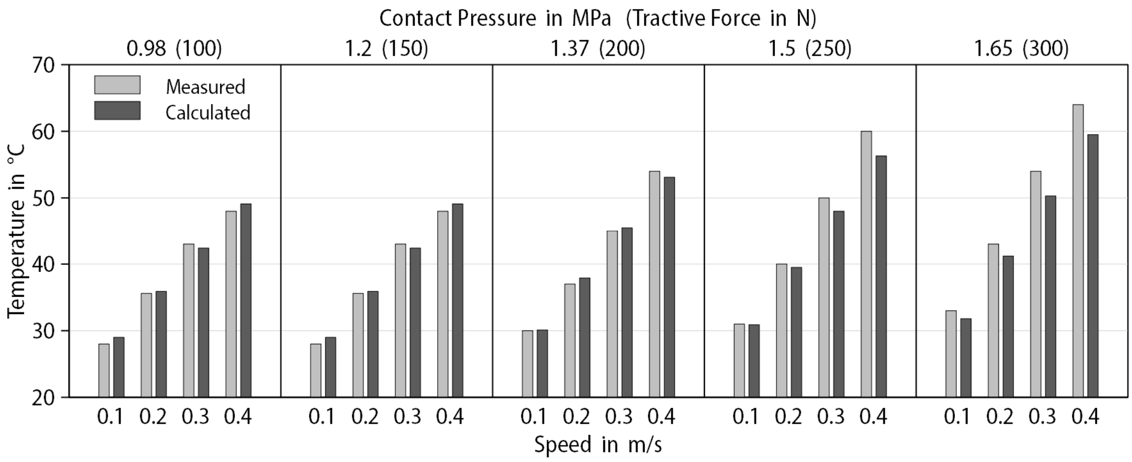

Figure 7: Comparison of measured and calculated friction temperature in a test conveyor as a function of the contact pressure and sliding speed

In the test program, speeds of 0.1 to 0.4 m/s were driven for tractive forces between 100 and 300 N. Higher process parameters could not be performed by the system. From the tractive force, the Hertzian contact pressure was calculated. Here, the curve exit was considered because the highest pressures occur there. The coefficient of friction of the POM — UHMW-PE pairing has been determined in a separate test on an oscillating test bench [Sum11]. To calculate the temperatures, the highest friction coefficient of 0.4 was used which occur in low load combinations (low speed and low pressure).

As shown in Figure 7, the friction temperatures increase with increasing sliding speed as well as pressure. It is also distinctly that the model results correspond well with the measured temperatures. With higher pressures the calculation values drop compared to the measured values. One reason is seen in Hertzian contact pressure which is not designed for thin bodies. The thickness of the soft slide rail is only 2 mm, so the maximum pressure will be higher due to the higher Young's modulus of the underlying aluminum mount.

5. Summary

A calculation model for predicting the maximum occurring friction temperature in dry-running sliding friction was derived. It is intended for periodically recurring contacts, such as those found in continuously working sliding chain conveyors. The semi-analytical model is kept simple in order to use it without high-capacity computing and software technology. Validation were made by tribological tests on a test bench and temperature measurements in a test conveyor [Bar17]. This present article depicts some results of the comparison with the test conveyor. Further validations on a conveyor system with different conveying loads or other geometrical configurations have to be carried out.

With the new approach, sliding chain conveyor systems can be thermally dimensioned depending on conveying speed and surface pressure while specifying heat transmittances, the contact ratio and the coefficient of friction. The calculated temperature can be compared with a maximum permissible temperature. As a temperature limit, for example, the lowest softening temperature of both sliding partners can be used, because here the wear of sliding surfaces starts. This is proposed as a thermal design criterion cue to the fact that the softening temperature of known for every polymer. Conveyor system suppliers have to provide only system-specific parameters for users who evaluate the system it by speed and load. With this method, a gap in thermal dimensioning of conveying systems can be closed.

The new approach can be used hypothetically for all systems with periodically recurring frictional sliding contacts. In addition to the applications on tribological test benches or sliding chain conveyors also mat chain conveyors or the system toothed belt support are conceivable. Also an optimization or new development of sliding chain conveyors is possible with the proposed method.

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