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A Study of the Connection Between Bending Stress and Belt Friction Using a Servomotor Controlled by a Computer

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ABSTRACT The aim of the publication is to design an analytical relationship for determining the bending coefficient measurable by belt friction. Following the results of the presented measurements, the theoretical assumption is verified that in the material of the belt stress arises during the friction of the contact surfaces, due to external forces and bending of the belt. Selected experiments of drawing a polypropylene belt at a constant speed over an aluminium cylindrical surface are carried out using a servo drive on a measuring bench with computer-controlled instrumentation. The results of friction and bending coefficient measurements are interpreted in connection with the setting of various input parameters (especially friction surfaces and belt thicknesses), at a constant shaft speed. The newly identified dimensionless bending coefficient of belt friction is a material coefficient with a comparative information value concerning the flexibility of the belt, suitably supplementing the friction Euler-Eytelwein coefficient. The presented publication can be suitably applied especially to the engineering educational process.

INDEX TERMS Engineering educational process, friction, bending coefficient, belt.

I. INTRODUCTION

According to the educational programme for the university field of study Mechanical Engineering (specifically issued by the Ministry of Education, Youth and Sports in the Czech Republic), the GRADUATE PROFILE is determined, in which key and professional competencies are declared. The presented publication corresponds to the key "problem-solving competence", according to which engineering education aims to enable graduates to solve common work and non-work problems independently. The presented publication also corresponds to the professional competencies: "Proposing methods, technical equipment, tools, instruments, production aids and technological conditions"; "Measuring basic technical quantities"; "Using the means of information and communication technologies to support effective work".

The interaction of the surfaces of rigid bodies during their mutual movement (especially in machine systems) has been historically and intensively addressed by tribology as an important interdisciplinary science [1], [2]. In principle, friction is examined as an action characterized by the unity of opposites, i.e. as a specific action conditioned by movement and, at the same time, preventing movement. The friction force is investigated as a dissipative force indirectly measurable, the primary cause of which is the roughness of the contact surfaces [3], [4], hence the effect of cohesion and adhesion. [5] Friction coefficients depend on the combination of contact surfaces, their condition, moisture, the content of impurities, presence of interlayer, rest/movement state [6], [7], the speed of mutual movement of contact surfaces [8], [9], hardness, stress and wear rate [10], various technological processes in the formation of friction surfaces and coating [11]. It is also possible to highlight a significant connection between the coefficient of friction, contact temperature and thermal conductivity of materials [12].

Because these numerous and diverse influences are reflected in changes in material properties to varying degrees, it can be admitted that the friction coefficients have a comparative information value, especially in technical practice [13].

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A necessary condition for accurate measurements of tensile forces is the maintenance of a constant speed of movement, i.e. movement with an artificial subsidy of dissipative losses [14]. The friction mechanism studied at the microscopic level is a complex process in which the cohesive and adhesive effects are applied with different intensity, but always work in synergy. The consequence of the permanent interaction of the particles of the contact surfaces (''stick and slip'' effect) is the fact that the measured value of the tensile force always oscillates around a particular mean value [15]. This mean value is evaluated at the macroscopic level, using a statistically significant number of partial tensile forces, measured stepwise over a relatively long-time interval. The tensile forces evaluated in this way then indicate the actual force action under the given experimental configurations and under the given measurement conditions in the sense that they decide on the overall ratio of forces and also on the energy balance in the investigated system. From the material-quality point of view, they are the basis for indirect measurements of the friction coefficient [16]. Belt friction is, in principle, a specific case of shear friction. It is derived from shear friction at the elementary level, using the ratio of the elements of the tensile force and the normal compressive force. The coefficient of friction is of integral character; in the case of shear friction, it is given by the ratio of the tensile force of the kinetic and compressive force normal; in the case of belt friction, it is given not only by forces acting on the circumference of the cylindrical surface but also geometrically, i.e. the ratio of the moments of the driving and tensile forces increases exponentially with the angle of wrap [17]–[20]. Compared to the surface material of the cylindrical surface, the belt material is significantly elastic; under external forces, well measurable stress arises in this flexible material, depending on the combination of contact surfaces used [21]. Polypropylene, as a thermoplastic is one of the economically important materials used in a wide range of applications, from consumer goods to highly specialized products. The reasons for the wide range of applications are mainly the desired physico-chemical properties, mechanical and chemical resistance, strength and rigidity. Of the hitherto known variants of polypropylenes, isotactic polymers are best used for the production of textile fibres, the fibres of which are particularly soft to the touch, have very good abrasion strength and minimal moisture content. On the contrary, their disadvantages include low ability to recover after extreme deformation and relatively low resistance to the effects of high temperatures. In terms of friction, polypropylenes are sought-after materials with a low coefficient of friction, i.e. they are resistant to wear and the occurrence of stress cracks. Polypropylene surfaces generate relatively little heat during friction; their service life is longer compared to other polymers. The mechanical properties of polypropylenes strongly depend on the rate of deformation; compared to other conventional plastics, they have a good surface hardness and sufficient elasticity at low temperatures [22]–[25]. The necessity to transmit mechanical power between different parts of a machine is a very important aspect of the engineering design in many applications. This transmission mechanism is mostly secured by a belt drive. When the belt rotates around the pulleys, it is exposed to belt friction. To describe this friction between the belt and pulley is a crucial task. As it results from the literature research, there are no friction models generally accepted. There are two the most frequently used approaches in modelling belt-pulley friction. First, one theory is known as the creep model, and the second one is known as a shear model. While the creep model considers only kinetic friction, the shear model developed by Firbank [26] is based on the shear strains along the radial direction of the belt in the adhesion zone (in the arc). He was the first to take the effect of elasticity into account [27], [28]. Leonhard Euler and Johann Albert Eytelwein are known as the most famous scientists in the field of mechanical power transmission when using belts and pulleys. The equation derived by them, known as Euler-Eytelwein Formula, is used to compute how friction contributes to the transmission of power in mechanical assemblies. This formula is used by many researchers to control motion. Eliseev with Vetyukov [29] used this formula in contour motion of the belt, the same authors Vetyukov *et al.* also used this formula for transient dynamics with idealized friction [30]. Bulín and Hajžman [31] made a comparison of the classical belt friction formula with a detailed belt-cylinder interaction model. The absolute nodal coordinate formulation of the finite elements was used to create a contact force model between the belt and cylinder. However, this model has good conformity with the analytical Euler-Eytelwein's formula only when bending stiffness of the belt is neglected. Nevertheless, some authors as Jung *et al.* [32] or Gao *et al.* [33] considered the effect of the belt bending stiffness, and they used the modified classical Euler-Eytelwein's formula. As the belt moves along the angle of wrap of the belt pulley, it is subjected to bending. Bending resistance of the belt reinforced rubber belts was studied by Chołodowski and Dudziński [34]. Because the frictional and stress events in the material interact, they co-determine the material properties of the surface/core and their changes. The output of the presented publication is an analytical proposal of the bending coefficient of belt friction in connection with the friction coefficient. This proposal is based on the assumption that the stress arising during the bending of the belt indicates the degree of external force and the flexibility of the belt. It was predicted and verified that the bending coefficient of belt friction suitably complements the Euler-Eytelwein friction coefficient.

II. THEORETICAL PART

A belt drawn over a cylindrical surface can be viewed as a 3-dimensional object. The material of the belt is compressed/shortened under the action of tensile forces on the inner side of the bend, on the outside of the bend, it is stretched/elongated, while in the intermediate-neutral axis, it is minimally stressed. The neutral axis passes through the centre of gravity of the cross-sectional area, the total stress in

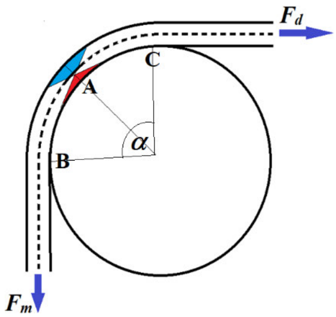

FIGURE 1. Pulling the belt over the cylindrical surface: area of stretched material (in blue); area of compressed material (in red); neutral line (dashed).

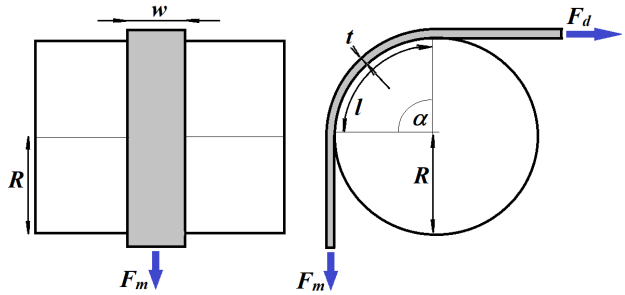

FIGURE 2. Diagram of drawing the belt over a cylindrical surface with an indication of geometric dimensions and applied tensile forces.

the neutral axis is zero because the tensile stress on one side of the neutral axis is positive, while the radial compressive stress on the opposite side of the axis is negative. The tension of the fibres lying on the outside of the neutral layer corresponds to uniaxial tension; the tension of the fibres on the inner side corresponds to uniaxial pressure. The elongation of the fibres of the material in the longitudinal direction on the outside of the bend is accompanied by a decrease in the transverse dimensions; on the contrary, the shortening of the fibres on the inside of the bend is accompanied by an increase in the transverse dimensions, on the contrary, the shortening of the fibres on the inside of the bend is accompanied by an increase in the transverse dimensions. The neutral plane is introduced theoretically with respect to the action of mean, integral tensile and compressive forces, the practical ''compensation'' of the force occurs in the middle region of the friction surface between points A, B and A, C (Figure 1). In reality, at the edge points B, C of the friction surface, the tensile forces F_d, F_m act as a maximum, while in the centre of the friction surface A, the action of the compressive force is minimal. On the other hand, at the edge points B, C of the friction surface, the acting compressive forces are minimal, while at the centre of the friction surface A, the acting compressive-radial force is maximal (Figure 1).

The basic deformations that are considered when pulling the belt over the cylindrical surface are bending, tension and pressure. Deflection deformation and shear stress were not considered to be negligible (e.g. for thin beams and especially relatively thin belts, it is not measurable). With regard not only to the stress state in the contact-stressed area of the belt but especially with regard to measurability, tensile and compressive stresses are considered and evaluated, acquiring extremely smaller values than the values of tensile and compressive strength. If tensile forces act on the belt at a certain angle, the normal tensile stress in the contact-stressed area of the belt can be considered and indirectly measured. At the same time, the belt and the cylindrical surface act on each other with compressive forces (action-reaction). While the consequence of the force acting on the belt on the cylindrical surface is negligible, the consequence of the force acting on the belt on the cylindrical surface is not negligible; normal pressure stress can be considered in the contact-stressed area of the belt. Although actions and reactions are forces of the same magnitude (in this case compressive), the consequences of the action are different (in this case deformation effects). Deformations of both materials (metal and polypropylene) are incomparable in size. The elementary friction coefficient of shear/belt friction is classically defined in tribology as the ratio of the elementary kinetic tangential friction force and the normal elementary compressive force. The mean value as the integral value of the bending coefficient μ (1) in belt friction can be determined in connection with the derivation of the mean-integral value of the Euler-Eytelwein coefficient (Figure 2). Instead of the mean value of friction and bending coefficients, the terms friction and bending coefficients are used to simplify terminology.

$$\mu = \frac{\sigma_T}{\sigma_N} \Leftrightarrow \mu = \frac{\frac{F}{t \cdot w}}{\frac{N}{l \cdot w}} \Rightarrow \mu = \frac{l}{t} \cdot \frac{F}{N} \tag{1}$$

The relation (1) includes:

σ_T normal tensile stress;
σ_N normal compressive stress;
F normal tensile force to the cross-section of the belt $S_T = t \cdot w$;
N compressive force normal to the cross-section of the belt $S_N = l \cdot w$;
l the length of the contact-stressed area of the belt;
w the width of the contact-stressed area of the belt;
t the thickness of the contact-stressed area of the belt;
α angle of wrap;
R the radius of curvature of the cylindrical surface;
μ bending coefficient

In analogy with the classical definition of shear friction along the horizontal plane [14], the ratio of both forces (at the elementary and integral level) can be replaced by the friction coefficient ξ along the cylindrical surface, in accordance with the Euler-Eytelwein derivation based on infinitesimal calculus (2) [18].

In principle, a frictional force is created during friction in the contact surface between two bodies. As for the coefficient of friction, this coefficient is theoretically defined and practically measured by the ratio of kinetic/tensile and normal forces. The older Eulerian derivation is based on the balance of external and internal forces of the system and is performed by infinitesimal calculus, while the newer Eytelwein's derivation is simpler than that, the integration is performed between the edges of the friction surface, and the procedure does not require mathematical negligence. The author's derivation [16] was performed in order to derive the belt friction coefficient in connection with the use of the presented method of semi-automatic measurement. The method of measuring the friction coefficient is based on the assumption that the shaft of the digital servo motor rotates at a constant speed; therefore, the magnitude of the friction force is also constant. Although frictional forces are dissipative in nature, they can still do mechanical work; with the continuous artificial subsidy of energy losses, there are practically no energy losses

$$\begin{aligned}
&\xi = \frac{F}{N} \wedge \frac{F_d}{F_m} = e^{\xi \cdot \alpha} \wedge T = F_d - F_m \\
&\xi = \frac{1}{\alpha} \cdot \ln \frac{F_d}{F_m} \wedge \alpha = \frac{l}{R} \Rightarrow \xi = \frac{R}{l} \cdot \ln \frac{F_d}{F_m} \\
&\wedge \xi = \text{const.} \Leftrightarrow \alpha = \text{const.} \wedge \omega = \text{const.}
\end{aligned} \tag{2}$$

The relation (2) also includes:

F_d tensile force exerted by the actuator;

F_m tensile force exerted by the weight of the load, where $F_d > F_m$ applies;

T friction force;

α angle of wrap;

ω angular speed of rotational movement of the shaft/cylindrical surface;

ξ coefficient of friction.

In the relation (2), the belt friction coefficient ξ is declared to be constant, depending on the tensile forces F_d, F_m, on the angle of wrap α (or on the radius of curvature R and the length of the friction surface l), provided that the angle of wrap α and, at the same time, the shaft speed ω are also constant.

After substituting F/N from the relation (2) to the relation (1), the relation (3) can be declared as a relation of "interconnection" of the bending coefficient μ with the friction coefficient ξ, under conditions of setting the same angle of wrap α.

$$\mu = \frac{l}{t} \cdot \xi \wedge \alpha = \frac{l}{R} \Rightarrow \mu = \frac{\alpha \cdot R}{t} \cdot \xi \tag{3}$$

The relation (3) can then be interpreted as the bending coefficient depending on the bending ratio R/t and on the logarithm of the ratio of the respective tensile forces. The bending coefficient, similar to the friction coefficient, is dimensionless. When the constant value of the angle of wrap and the angular velocity is set, it changes linearly depending on the bending ratio. (3)

$$\begin{aligned}
&\mu = \frac{R}{t} \cdot \ln \frac{F_d}{F_m} \\
&\wedge \mu \text{ is linear function } \Leftrightarrow \alpha = \text{const.} \wedge \omega = \text{const.}
\end{aligned} \tag{4}$$

In classical formulations of belt friction coefficients, the dependence on the friction surface does not explicitly appear, nor is the belt thickness considered. In other words, the connection between the events in the surface and the core of the belt material is not assumed. For a given angle of wrap and a given combination of friction surface materials, the friction coefficients can be measured as the same (the usual accuracy of the friction coefficients is given in hundredths in the tables for designers), regardless of the size of the belt friction surface and belt thickness. The bending factors (for a given angle of wrap and a given combination of friction surface materials) can never be the same because it also explicitly depends on the thickness of the belt.

The bending factors (at a given angle of wrap and a given combination of friction surface materials) may implicitly depend on the size of the friction surface S, in specific cases where the geometric dimensions of the belts are relatively extreme (5).

$$S = l \cdot w \wedge \alpha = \frac{l}{R} \Rightarrow S = \alpha \cdot R \cdot w \tag{5}$$

As far as the length l of the contact-stressed area of the belt is concerned, the neutral length theoretically does not change relatively during bending, i.e. it corresponds approximately to the developed length of the contact-stressed area of the belt. However, when bending a belt with a relatively small radius of curvature, the two lengths are not comparable because the material in the contact-stressed area of the belt is much more compressed than stretched. It is in this case that the calculation needs to be corrected empirically because the neutral length is related not only to the inner bend radius but also significantly to the belt thickness.

For the width w of the contact-stressed area of the belt in real time, it applies that it is related not only to the inner bending radius but also significantly to the belt thickness. In the case of bending wide belts, a simpler planar deformation occurs (the width of the belt is significantly greater than its thickness). Deformations of the belt material in width are relatively small, negligible in the middle region of the width, the change in the length of the fibres in the longitudinal direction is compensated exclusively by a change in the thickness in the radial direction. In the case of bending narrow belts, the cross-section is more complicated to deform, spatial deformation occurs (the width of the belt is significantly smaller than its thickness). Wide belts are more deformation-stable than narrow belts, although they become relatively thin at the bending point during bending, the material is practically not deformed over the entire width of the belt [35].

The thickness of the contact-stressed area of the belt substantially affects the stress in the belt material. The larger the bending angle, the greater the friction between the material and the friction surface, the smaller the bending radius and the less malleable the material, the thinner the material becomes at the bending point [35]–[37].

The dependence on the speed of the movement and especially on the constant speed of the cylindrical surface rotation is implicit in the relations (1) to (4), but it is essential for accurate measurement results! In the case of even a slightly variable speed of the rotation of the friction surface, the belt is usually not stretched exactly in the tangential direction

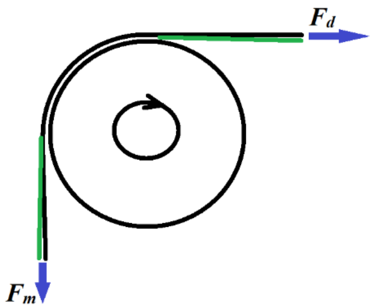

FIGURE 3. Deviation of the real direction of the drawn belt (green) from the ideal direction (black).

of the tensile forces acting both at the entrance to the contact surface and at the exit from the contact surface. The consequence of this phenomenon is the emergence of passive resistance and a corresponding change in the angle of the belt. When pulling the belt in these cases, there is an undesirable ''lifting/sticking'' of the belt, depending on the direction of the rotation (Figure 3). The servomotor shaft can, of course, rotate in both the clockwise and counter - clockwise directions without the direction of rotation, causing different results in the measurement of the applied tensile forces. If a motor which cannot maintain a precisely constant speed is used, the undesired effect shown in Figure 3 could occur. Due to insufficient tension of the belt, inaccuracies would arise in the result of measuring both the wrap angle and the magnitude of the tensile force.

III. EXPERIMENTAL PART

The experimental part was realized using a measuring plane connected to a cylindrical surface and equipped with instruments. The measuring plane served as a horizontal plane for experiments performed with an angle of wrap of 90° (Figure 4) and as an inclined plane for experiments performed with an angle of wrap of 45° (Figure 4) and 135°. The length of the measuring plane was 0.8 m, while the length of the total unwound belt was 0.45 m. The actuator 4 with the shaft installed exerted a total tensile force F_1 for the translational movement of the elements 1, 2, 3 and the rotational movement of the belt. Element 3 loaded with two force gauges and a data logger developed a tensile force F_2 ($F_d = F_2$). The 1 + 2 element as a load composed of a force gauge, a datalogger and an additional weight developed a tensile force F_3 ($F_m = F_3$). A laser sensor 5 with a perpendicular impact of the beam on the belt was installed above the centre of the friction surface in order to continuously measure the thickness of the belt. All clamps for the measuring set components have been made on a 3D printer so that they can be precisely centred in a balanced position.

A. EXPERIMENTAL SET-UP

The traction force F1 (Figure 4) was developed by the Dynamixel MX-64T servo motor (Robotis) externally controlled by the CM-700 unit and monitoring the actual operation data. It is a sophisticated device that offers the user feedback, allows monitoring of the current state of the position, speed, power, or operating temperature (the device worked correctly during the implementation of all experiments, the operating temperature was monitored and was within limits allowed by the manufacturer). Because the servo motor incorporates feedback speed control and has relatively high torque, it can reliably keep the speed as exactly constant. The price for this reliability is only significant dissipative losses, which represents a relatively low efficiency of the device (maximum 50 % at a shaft speed of 30 RPM).

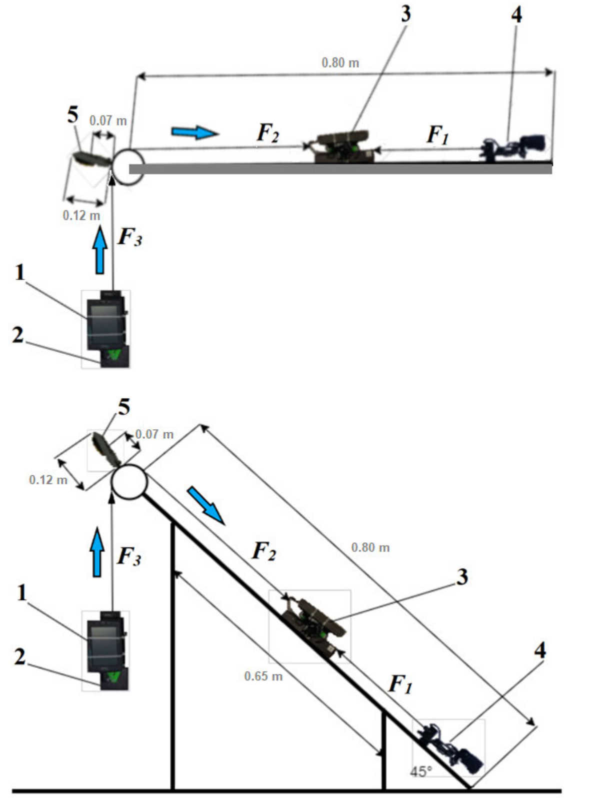

FIGURE 4. A diagram of the assembly intended for measuring the bending and friction coefficient of belt friction for an angle of wrap of 90° (above figure) and 45° (bottom figure): element 1 (a force gauge connected to the datalogger); element 2 (additional weight attached to element 1); element 3 (2 load cells connected to the datalogger); element 4 (actuator with shaft and control unit); element 5 (laser scanner); the blue arrow indicates the direction of the translational movement.

Three digital Dual-Range Force Sensor (Vernier Software & Technology) force gauges connected to data loggers, i.e. external Lab Quest devices, were used to measure the three tensile forces directly. Dataloggers were equipped with LoggerPro software for analysis, display and data processing and read values with a sampling frequency of 100 Hz.

A laser sensor (Allen-Bradley 45BPD-8LTB1-D5) with a measuring range of 30 to 100 mm was used as a digital distance meter to measure the belt thickness. The sensor was calibrated by comparing the results of direct measurements performed on the tested surface of the cylindrical surface and the results of measurements performed on the cylindrical surface with selected calibration pads. Based on the measurement results, the difference between the sensor-cylindrical surface and the sensor-calibration pad distances was determined as the required thickness of the calibration plate.

The original values of the online current measurements were converted to length values (Figure 5) as the examined belt thicknesses, the conversion taking place according

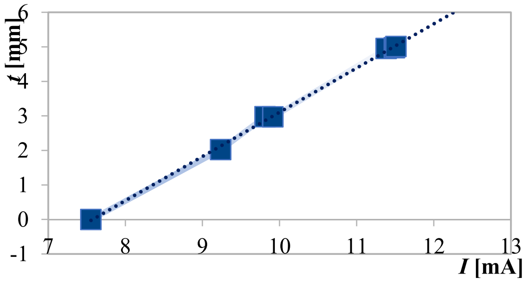

FIGURE 5. Conversion of current values from the laser sensor to the corresponding belt thickness values t.

FIGURE 6. Side view of the measuring assembly with the cylindrical surface set at an angle of wrap 45°.

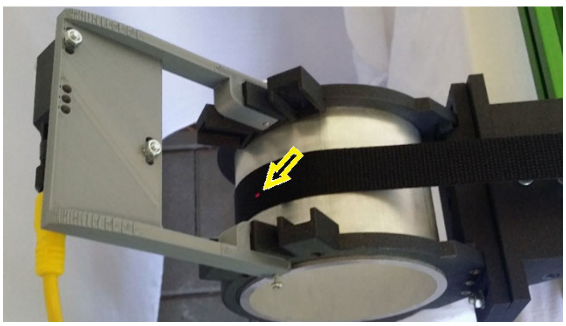

FIGURE 7. An example of a measuring set detail for online measurement of the belt thickness with a laser sensor (the yellow arrow points to the red trace of the beam hitting the belt).

to the relation (6) with the total accuracy of measurement $R^2 = 0.999$. From the graph in Figure 6, the experimental-regression equation was taken as a linear equation, in a relatively narrow interval of the performed measurement (only for calibration).

$$t = 1.281 \cdot I - 9.697 \tag{6}$$

Figure 6 is an overall view of the measuring assembly in an inclined configuration.

Figure 7 is a detailed view of online measurement of the belt thickness by a laser scanner.

B. MATERIALS USED

A single combination of commercial materials was chosen for the presented experiments, namely a polished aluminium cylindrical surface (EN 573-3 AW 6060 T66 EN 755-1,2,8) and polypropylene belts of various thicknesses and widths.

C. MEASUREMENT PROCEDURE

A measuring plane connected to a cylindrical surface and equipped with measuring instruments was assembled according to the diagrams in Figure 4. The weight of element 3 (1.740 kg) was chosen with respect to the weight of element 1 + 2 (0.836 kg) empirically so that there is no accelerated translational movement of the elements (datalogger weight 0.350 kg; the force gauge weight 0.204 kg; additional weight 1.186 kg; storage platform weight 0.078 kg for element 3).

In the context of shear friction between element 3 and the measuring plane, it should be added that the coefficients of shear friction f and belt friction ξ are "connected" by means of tensile force F_2, e.g. for the diagram in Figure 4, the relations (7) apply:

$$\frac{F_2}{F_3} = e^{\xi \cdot \alpha} \wedge f = \frac{F_1 - F_2}{m \cdot g \cdot \cos \alpha} \tag{7}$$

The shear friction of element 3 does not affect the results of the belt friction measurement. The force exerted by the actuator overcomes both acting loads, both frictional forces, and, at the same time, exerts a constant force in order to maintain a uniform movement of both elements 1 + 2, 3; therefore, there is no acceleration of movement (the actuator acts as a "tractor" of elements 1+2, 3). Significant oscillations of the measured tensile forces always occur, but only shortly after the start, when the servo motor starts from zero to the desired constant value of shaft speed. Oscillations occurring during the traffic delay were not included in the results of measuring the mean values of traction forces. Subsequent very slight oscillations of the measured forces around the mean value necessarily occur due to friction.

The experiments were performed at room temperature, no heating of the belt was recorded during the continuous temperature measurements. In all experiments, the speed of rotation of the cylindrical surface, and, therefore, the speed of the installed shaft, was controlled at a constant value of 33.071 RPM; The translational speed of motion of element 3 was determined to be 0.022 m·s^{-1} (as the product of the angular velocity radius of 3.14 and the outer radius of the shaft 0.00705 m).

The belt widths were chosen to be sufficient, i.e. continuous changes in belt width were not recorded during the measurement, even with regard to the fact that the magnitudes of tensile forces were around 31 N at most (changes in the original belt width did not exceed 20 %).

At rest, the basic input data of the cylindrical surface were set and measured: the angle of wrap (by setting the measuring plane using a protractor) and the outer radius of the cylindrical surface (the data given by the manufacturer were verified by a calliper).

TABLE 1. Selected Combinations of Directly Measured Parameters and Results of Indirect Measurements of Friction Coefficient ξ and Bending Coefficient μ.

	w [m]	α [rad]	R [m]	$S\cdot10^{-4}$ [m^2]	$t\cdot10^{-3}$ [m]	F_1 [N]	F_2 [N]	F_3 [N]	ξ [-]	μ [-]
1.	0.025	0.785	0.055	10.79	2.184 ±0.009 0.40 %	27.194	21.545 ± 0.030 0.14 %	17.713 ± 0.003 0.02 %	0.25	0.005
2.	0.025	2.356	0.055	32.40	2.042 ±0.015 0.75 %	27.667	24.917 ± 0.073 0.29 %	17.722 ± 0.002 0.01 %	0.15	0.009
Assessed incomparably, i.e. at a significantly different angle of wrap α: the larger friction surface S has a smaller friction coefficient ξ; comparable belt thicknesses t do not in themselves have comparable bending coefficients μ. It is further assessed comparably, i.e. at the same angles of wrap α:										
3.	0.025	1.571	0.0202	7.93	1.878 ±0.006 0.32 %	24.842	23.139 ± 0.029 0.12 %	17.735 ± 0.003 0.02 %	0.17	0.003
4.	0.025	1.571	0.0095	3.73	1.909 ±0.023 1.20 %	25.986	24.265 ± 0.017 0.07 %	17.712 ± 0.003 0.02 %	0.20	0.002
the smaller friction surface S has a larger friction coefficient ξ; the belt thicknesses are comparable, with the relatively slightly larger thickness t of the belt has a relatively slightly smaller bending coefficient μ.										
5.	0.020	1.571	0.055	17.28	0.601 ±0.008 1.36 %	33.608	31.419 ± 0.067 0.22 %	17.770 ± 0.003 0.01 %	0.36	0.052
6.	0.020	1.571	0.055	17.28	1.865 ±0.012 0.64 %	26.107	24.266 ± 0.039 0.16 %	17.697 ± 0.002 0.01 %	0.20	0.009
the same friction surfaces S do not have the same friction coefficients ξ; a significantly larger thickness t of the belt has a significantly smaller bending coefficient μ.										
7.	0.025	1.571	0.055	21.60	0.835 ±0.004 0.53 %	26.192	24.292 ± 0.046 0.19 %	17.762 ± 0.002 0.01 %	0.20	0.021
8.	0.025	1.571	0.055	21.60	1.907 ± 0.008 0.43 %	26.438	24.562 ± 0.047 0.19 %	17.820 ± 0.001 0.01 %	0.20	0.009
9.	0.025	1.571	0.055	21.60	2.954 ± 0.014 0.49 %	25.301	23.427 ± 0.035 0.15 %	17.757 ± 0.002 0.01 %	0.18	0.005
the same friction surfaces S do not have exactly the same friction coefficients ξ; increasing thicknesses t of the belt show a decreasing bending coefficient μ.										
10.	0.040	1.571	0.055	34.56	0.795 ± 0.013 1.58 %	33.131	30.998 ± 0.080 0.26 %	17.790 ± 0.002 0.01 %	0.35	0.038
11.	0.040	1.571	0.055	34.56	2.262 ±0.024 1.08 %	27.270	25.368 ± 0.048 0.19 %	17.718 ± 0.003 0.01 %	0.23	0.009
the same friction surfaces S do not have the same friction coefficients ξ; a significantly larger thickness t of the belt has a significantly smaller bending coefficient μ.										

In a set of 11 experiments after 10 repetitions, a total of 660,000 partial direct dynamic measurements of tensile forces and belt thicknesses were performed online simultaneously, in 0.01 s increments. Tensile forces were measured by digital force gauges, the force F_1 was measured as the total driving force exerted by the actuator; forces F_2, F_3 were measured as tensile forces. The tensile force F_2 was measured relatively less accurately due to frictional oscillations, the tensile force F_2 was measured more accurately because it represented the load weight of the elements $1+2$. (Table 1).

IV. RESULT AND DISCUSSION

The friction coefficients were evaluated according to the relation (2) and the bending coefficients according to the relation (4). Uncertainties of the results of indirect measurements of the investigated coefficients, including the uncertainty of the relevant directly measured parameter, were plotted using error bars, as a potential 5% of the error value (positive and negative).

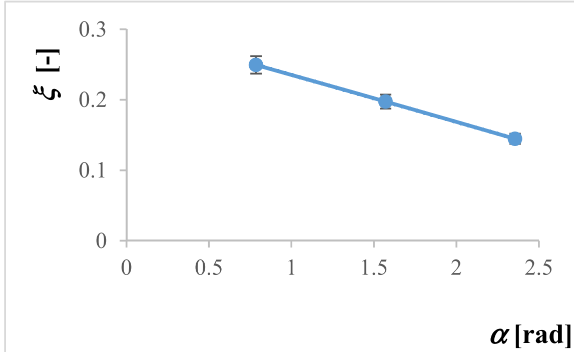

FIGURE 8. Dependence of the friction coefficient ξ on the wrap angle α.

According to the result Table 1, the belt thickness is, therefore, an experimentally important parameter not only for evaluating the bending coefficient but also for evaluating the friction coefficient. As the thickness of the belt increases, not only does the bending coefficient decrease but with a comparable friction surface, the friction coefficient even decreases. Based on the acquired experimental experience, it can be stated that the measurement of both material-geometric coefficients requires not only accurate measurements but also empirical corrections with respect to the thickness of the used belts and the size of the friction surfaces. The classical coefficient of belt friction has been derived and is still measured depending on the length of the friction surface (which is given by the product of the angle of wrap and the radius of curvature), but does not affect the width of the friction surface or the belt thickness. The accurate measurements presented confirmed the expectation that the stresses created when pulling the belt are projected from the core into the friction surface.

A. DEPENDENCE OF THE FRICTION COEFFICIENT ON THE ANGLE OF WRAP

Friction coefficients were evaluated for selected angles of wrap from a relatively narrower interval ⟨45°, 135°⟩. The trend development connecting line was experimentally evaluated as decreasing, according to the linear approximation equation (8) and with the reliability value $R^2 = 1.00$ (Figure 8). The experimental-regression equation was taken as a linear equation in a relatively narrow measurement interval (theoretically and generally when examining a relatively wide range of variables, there is no linear relationship but an inverse relationship).

$$\xi = -0.067 \cdot \alpha + 0.302 \tag{8}$$

Since in the relation (6) there is a negative sign at the angle of wrap α, it is obvious that the factor ξ decreases; also in Figure 10, in accordance with the theory, a decreasing trend of the development of the coefficient ξ can be seen depending on the increasing angle of the belt α.

It can be stated that the measurement results are semantically in accordance with the theory of belt friction according to the relation (2) Euler-Eytelwein. From the experimental point of view, it can be seen that with increasing angle of

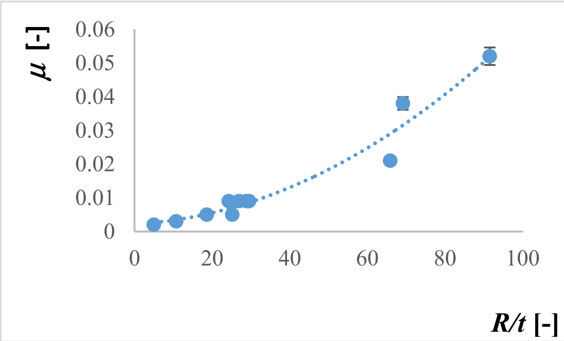

FIGURE 9. Dependence of bending coefficient μ on bending ratio R/t.

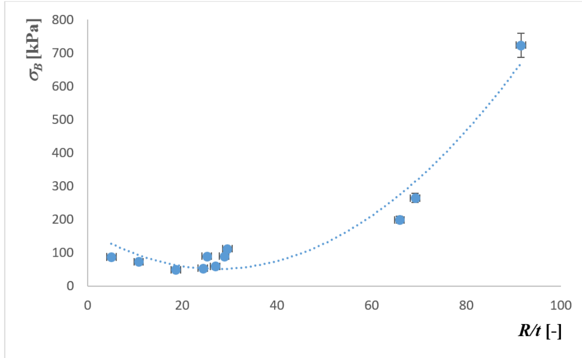

FIGURE 10. Dependence of bending stress σ_B on bending ratio R/t.

TABLE 2. Evaluation of Bending Stress in the Contact-Stressed Area of the Belt.

	$S \cdot 10^{-4}$ [m^2]	$t \cdot 10^{-3}$ [m]	$V_B \cdot 10^{-7}$ [m^2]	F_2 [N]	F_3 [N]	T [N]	R [m]	$(R/t) \cdot 10^3$ [-]	σ_B [kPa]
1.	10.79	2.184	23.565	21.545	17.713	3.832	0.055	25.183	89.436
2.	32.40	2.042	66.161	24.917	17.722	7.195	0.055	26.934	59.813
3.	0.93	1.878	14.892	23.139	17.735	5.404	0.0202	10.756	73.299
4.	3.73	1.909	7.121	24.265	17.712	6.553	0.0095	4.976	87.428
5.	17.28	0.601	10.385	31.419	17.770	13.649	0.055	91.514	722.845
6.	17.28	1.865	32.227	24.266	17.697	6.569	0.055	29.491	112.109
7.	21.60	0.835	18.036	24.292	17.762	6.530	0.055	65.868	199.129
8.	21.60	1.907	41.191	24.562	17.820	6.742	0.055	28.841	90.022
9.	21.60	2.954	63.806	23.427	17.757	5.670	0.055	18.619	48.874
10.	34.56	0.795	27.475	30.998	17.790	13.208	0.055	69.182	264.398
11.	34.56	2.262	78.175	25.368	17.718	7.650	0.055	24.315	53.822

wrap, the friction coefficient decreases, while the accuracy of the result of indirect measurement of the friction coefficient increases, i.e. measurement at small angles of wrap is more demanding than at larger ones.

The static coefficient of belt friction was not investigated (within the exclusion of measurements during the traffic delay). The subject and aim of the publication were not to examine the static friction coefficient, nor the material deformations that occur during static friction. Partial dynamic measurements were the basis for the evaluation of the kinetic friction coefficient.

B. DEPENDENCE OF BENDING COEFFICIENT ON BENDING RATIO

For selected angles of wrap from the relatively narrower interval $\langle 45°, 135° \rangle$, bending coefficients were evaluated, depending on the relative ratio R/t., while the connecting line of the development trend was experimentally evaluated as increasing, according to the quadratic approximation equation (9) with a reliability value of $R^2 = 0.96$ (Figure 9).

$$\mu = 5 \cdot 10^{-6} \cdot (R/t)^2 + 6 \cdot 10^{-5} \cdot R/t + 0.0022 \tag{9}$$

With regard to meaning, it can be stated that the measurement results are in accordance with the prediction of the newly proposed bending coefficient (4). From the experimental point of view, it can be seen that with increasing ratio of the cylindrical surface radius and the belt thickness, the bending coefficient increases, while the accuracy of the result of indirect measurement of bending coefficient decreases, i.e. measuring relatively thin belts is more difficult than measuring wide ones.

C. COMPARISON OF STRESS TRENDS

In order to compare the stress trends, the bending stress arising in the contact-stressed volume of the belt (10) was evaluated using the measured values (Table 1-2).

$$T \cdot R = \sigma_B \cdot V_B \;\Rightarrow\; \sigma_B = \frac{(F_2 - F_3) \cdot R}{S \cdot t} \tag{10}$$

The relation (10) also includes:
V_B volume of the contact-stressed area of the belt;
S size of friction surface;
σ_B bending stress in the contact-stressed volume.

Furthermore, the dependence of the bending stress σ_B on the relative thickness of the belt (R/t) was shown (Figure 10), while the line of the development trend was experimentally evaluated as increasing, according to the approximation quadratic equation (11) with the reliability value $R^2 = 0.95$.

From an experimental point of view, it is clear that with increasing relative belt thickness R/t, the bending stress increases, while the accuracy of the result of indirect bending stress measurement decreases, i.e. the measurement of relatively thin belts is more demanding than the measurement of wide ones.

The bending stress takes into account the stressed volume of the belt, but regardless of the angle of wrap, which is fundamental for the evaluation of the friction coefficient.

The newly designed bending coefficient takes into account the normal tensile and compressive stresses with respect to the frictional force acting at a certain angle of wrap. However, the development trends of compared stress dependences are similar. It can be stated that the exact method of measuring tensile forces during belt friction using a servo motor offers the evaluation of various parameters suitable for tribological research and its applications.

V. SUMMARY AND CONCLUSION

In the presented work, a new analytical relationship was proposed to determine the bending coefficient during belt friction. The model was verified by means of a measuring bench composed of a servo drive and computer-controlled instrumentation. Dynamic measurements of tensile forces and belt thicknesses were performed for a selected combination of polypropylene-aluminium materials at a constant speed of the actuator shaft. Based on the results of direct measurements, indirect measurements of friction surfaces, volumes of contact-stressed areas of belts, friction forces, relative thicknesses of belts, friction coefficients, bending coefficients and bending stresses were performed. The results of accurate measurements were interpreted and compared in the form of numerical values, and development trends expressed approximately by equations and graphs in accordance with the prediction that friction and bending of the belt are closely related through geometric dimensions of the contact-stressed area of the belt and its stress changes.

The novelty of the concerned solution is the analytical determination of the bending coefficient during belt friction, which complements the Euler-Eytelwein friction coefficient. The benefit of the solution is in a simple implementation for other solutions requiring more accurate comparisons of friction and bending properties of belts of different materials used in technical practice.

Regarding the application of the bending coefficient, the ''degree'' of the belt flexibility (assigned to the respective friction coefficient) can be quantified by precise measurements, tabulated for commonly used materials and compared for different materials. Although both ''coupled'' material coefficients are multiple-parametric, the interpretation of their information value can be simplified by evaluating the respective friction coefficient for a given belt material at a given angle of wrap and then evaluating the respective associated bending coefficients for different examined belt thicknesses. It is thus possible to specify the interval of required flexibility depending on the interval of the possible thickness of this belt explicitly for the required material and its friction coefficient (at a given angle of wrap.

In the current tribological research of textile fibres, the dependence of friction forces on the size of the friction surface is investigated empirically. Because the measurement of especially relatively thin belts is relatively demanding with respect to the accuracy, it is possible to recommend the development and use of an original automated device that will measure the tensile forces acting on the fibres in various configurations. The results of these measurements can then be accepted as a basis for research into inter-fibre friction in the textile industry.

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