Krzysztof KUBAS*

MEASUREMENT OF THE STATIC FRICTION COEFFICIENT BETWEEN A poly-V belt 5pk AND A PULLEY UNDER DRY CONDITIONS

POMIAR WSPÓŁCZYNNIKA TARCIA STATYCZNEGO POMIĘDZY PASEM poli-V 5pk A KOŁEM W WARUNKACH TARCIA SUCHEGO

Key words: belt transmission, static friction, dry friction, poly-V belt.

Abstract The paper presents the results of experimental measurements of static friction forces between a poly-V belt 5pk and a pulley on a specialised research stand. An average effective static coefficient is assumed depending on the wrap angle and preload force. Different shapes and positions of curves of the measured values for lower wrap angles are found, with similar curves in the set of measurements with higher angles. The Nelder-Mead optimisation method is proposed to approximate the measured results by a nonlinear function and to achieve good accordance. The dependence of the effective friction coefficient on the rest time between measurements is also presented.

Słowa kluczowe: przekładnia pasowa, tarcie statyczne, tarcie suche, pas wielorowkowy.

Streszczenie W pracy zaprezentowano wyniki badań doświadczalnych sił tarcia statycznego pomiędzy pasem poli-V 5pk a kołem pasowym, wykonane na specjalistycznym stanowisku badawczym. Wyznaczono średnie wartości pozornego współczynnika tarcia statycznego wyrażone w funkcji kąta opasania i siły napięcia wstępnego pasa. Uzyskano różne kształty i położenia krzywych dla mniejszych kątów opasania i zbliżone krzywe dla kątów większych. Celem uzyskania dobrych zgodności, do aproksymacji uzyskanych wyników zaproponowano metodę optymalizacji funkcji nieliniowej Nelder'a-Mead'a. Zaprezentowano również wpływ czasu spoczynku, dokonywanego pomiędzy poszczególnymi pomiarami, na wartości pozornego współczynnika tarcia.

INTRODUCTION

There is a relatively large number of papers that present multidisciplinary research studies of friction between rubber and steel, including the results of clean (e.g., **[L. 1, 2]**) and contaminated or aged rubber (e.g., **[L. 2, 3, 4, 5]**). These papers usually present studies on flat or ball-and-flat surfaces but do not represent real frictional parameters of jammed ribs in the grooves of V-ribbed belts or poly-V belts. The theory of rubber and other materials has been presented, among others, in papers by **[L. 6, 7]**.

Research on belt transmissions started in the 18th century, when Leonard Euler described friction phenomena between a flat belt and a pulley **[L. 8]**. The most interesting research studies since 1981 have been described in Fawcett's widely cited paper **[L. 9]**. Later, particularly interesting seem to be papers by **[L. 10, 11]**, in which, among others, the authors determined the values of static and dynamic effective friction coefficients. Also interesting are papers by **[L. 12, 13]** in which the authors measured the values of a belt slip in transmissions. There are a rare few papers that examined contaminated belt transmissions, e.g., papers by **[L. 14, 15]** present the values of these coefficients of a contaminated belt with ice and water.

The main goal of presented research was to measure boundary friction forces, between the 5pk belt and the water pump pulley causing belt slip in belt transmission. It also decided to check if calculated friction coefficient from classical Euler formula depends on belt initial tension or rest time.

* University of Bielsko-Biała, Faculty of Mechanical Engineering and Computer Science, Department of Mechanics, ul. Willowa 2, 43-309 Bielsko-Biała, Poland, email: kkubas@ath.bielsko.pl.

THE RESEARCH STAND

Figure 1 presents a research stand that allows one to measure both static and dynamic friction forces. It also allows one to measure slip velocity via an incremental encoder connected to the pulley. Finally, the stand can be reconfigured to measure vibrations in the complete belt transmission. The structure and applications of the research stand were described in detail in paper **[L. 16]**.

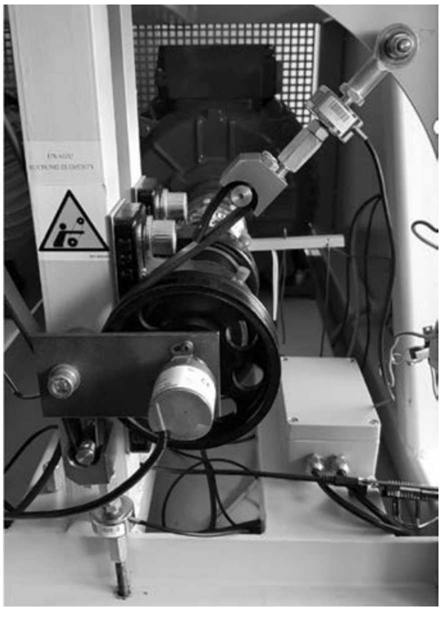

Fig. 1. Research stand for measuring friction in a belt transmission
Rys. 1. Stanowisko badawcze do pomiaru tarcia w przekładni pasowej

The results of the research will be used as input data into a model of belt transmission that is currently being developed and which has already been presented, among others, in **[L. 17]**.

In **Figure 2**, the way of determining the static friction coefficient by measuring the torque required to achieve the belt slip on the pulley is shown. It applied new and original water pump pulley from passenger car (painted coating, diameter 0.12 m).

The belt can be tensioned by axial forces $\mathbf{F}_1$ and $\mathbf{F}_2$, which can be measured by force sensors KMM20 (with a measurement range of 5 kN and a linearity tolerance of 0.2%). When the pulley has no torque applied to it, it can be assumed that the values of forces **F**1 and **F**2 are equal: $F1 = F2 = F0$. They can slightly differ because of the relatively large normal forces in the pulley bearings which increase rolling friction, but they are much smaller than the predicted friction between the belt and the pulley.

Some wrap angle of α can also be assumed. Additionally, the pulley can be loaded by a specified torque (via force **Q**), which rises to a certain arbitrary value and causes the belt slip. The moment of the start of the belt slip can be identified by an infrared reflective sensor connected to the computer via the microcontroller.

It was decided to measure a Poly-V belt 5pk, which is often used in car transmissions and to drive the alternator and air conditioning pump from the engine shaft. This kind of belt is relatively rigid, with small elasticity of the rubber part (in comparison with other kinds of belts, i.e. V-ribbed belts). The longitudinal stiffness make 15 fibres oriented along the belt. The belt is presented in **Figure 3**.

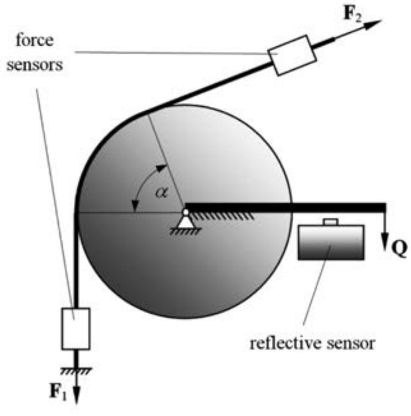

Fig. 2. Configuration of the research stand which allows to measure the static friction coefficient
Rys. 2. Konfiguracja stanowiska badawczego, która pozwala wyznaczyć współczynnik tarcia statycznego

Fig. 3. Analysed poly-V belt 5pk
Rys. 3. Analizowany pas poli-V 5pk

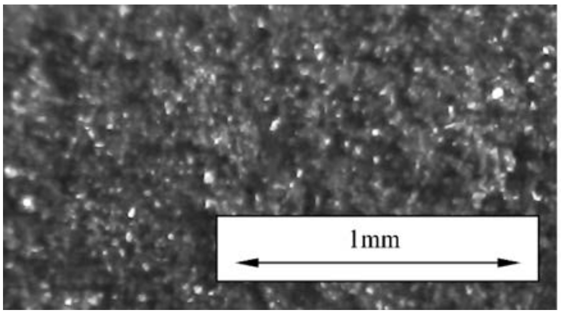

Fig. 4. Porous structure of the contact surface of the analysed belt
Rys. 4. Porowata struktura powierzchni współpracującej w analizowanym pasie

Figure 4 presents the structure of the working surface in the analysed belt. The figure shows a non-uniform composition of the surface. The porous structure and its specific physical and chemical composition influence the relatively high values of the friction forces, which increases the effectiveness of the transmission and reduces the wear of the belt.

MEASURING IF STATIC FRICTION BOUNDARY FORCES DEPEND ON TIME OF STICTION

First, it was decided to measure how the proportion of static friction boundary forces F_1 and F_2 changes in changing periods of time between the measurements made. It was then decided to investigate several cases of friction preload of the belt ($F_1 = F_2 = F_0$ = 200 N, 300 N and 400 N) with a sample wrap angle of α = 90°. The following rest times were assumed: measuring without pause (rest time was less than 1 s), 15 s, 30 s, 45 s, 1 min, 5 min, and 10 min.

Figure 5 shows the measured sample values of proportions F_1/F_2 and the calculated average values for belt preload force F_0 = 300 N and 400 N.

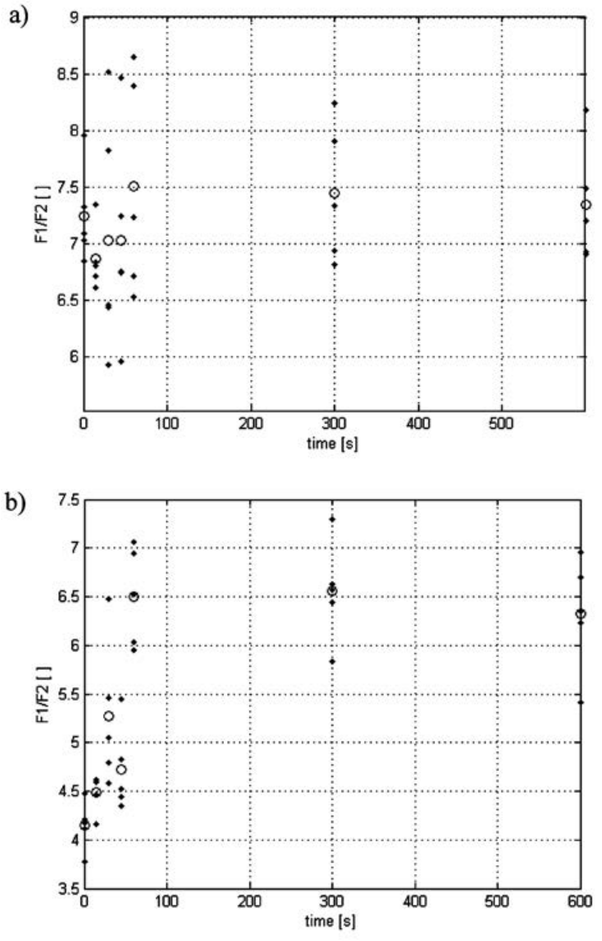

Fig. 5. Values of the proportions of boundary static friction forces as a function of rest time: a) F_0 = 300 N, b) F_0 = 400 N; ◆ measured values, ○ average values

Rys. 5. Wartości proporcji granicznych wartości sił tarcia w funkcji czasu spoczynku: a) F_0 = 300 N, b) F_0 = 400 N, ◆ wartości zmierzone, ○ wartości średnie

As can be observed in the figure above, values of static friction forces change irregularly for rest time periods of ca. 1 min. Rest time periods over 1 min do not significantly influence the value of the friction forces. It should be noted that this regularity was maintained, but the values of the forces were different in all of the cases of the preload forces.

It was decided to measure the friction forces after a rest time of over 1 min.

MEASURING THE EFFECTIVE STATIC FRICTION COEFFICIENT AS A FUNCTION OF THE PRELOAD FORCE

Figure 6 presents the five courses of the tensioning forces F_1 (upper courses) and F_2 (lower courses), measured for the value of the preload force F_0 = 500 N and wrap angle α = 90°. According to earlier observations, rest times between measurements were over 1 min. The time at 0 s corresponds with the moment when the infrared sensor noticed relative movement.

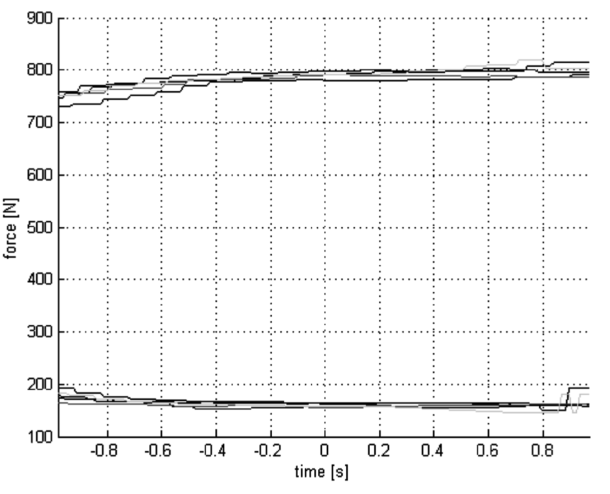

Fig. 6. Sample results of 5 courses of measured F_1 (upper courses) and F_2 (lower courses) forces (F_0 = 500 N, α = 90°)

Rys. 6. Przykładowe wyniki 5 przebiegów sił F_1 (przebiegi górne) i F_2 (przebiegi dolne) (F_0 = 500 N, α = 90°)

Small changing values of these forces can be observed, which are the result of a relatively elastic joint. Moreover, only a slight change in the value of these forces was noted after the start of the relative motion. This is due to slight differences between the coefficients of friction, i.e. static and dynamic. This observation coincides with the observations of other authors involved in research on friction in belt transmissions, including the papers **[L. 10]**.

As can be observed from the measured courses, the values of forces F_1 and F_2 are ca. 795 N and 160 N. Therefore, the proportion of these forces is about F_1/F_2 = 5. This process was repeated for different values of tension preload (200 N, 300 N, 400 N, 500 N, 600 N, 700N, 800N, 900N, and 1000N) and for different values of the wrap angle (30°, 45°, 60°, 75°, 90°, and 105°). **Figure 7** presents the achieved courses of proportions F_1/F_2. A selected case of F_1 and F_2 is shown in **Figure 6**.

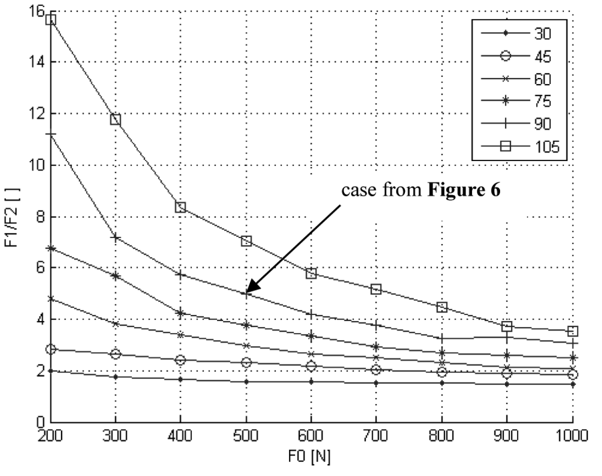

Fig. 7. Dependence of the proportion of F_1/F_2 on force F_0 and wrap angle α

Rys. 7. Zależność proporcji F_1/F_2 od siły F_0 i kąta opasania α

The effective coefficient of static friction can be calculated from the classic Euler-Eytelwein formula:

$$\mu = \frac{\ln F_1 / F_2}{\alpha} \tag{1}$$

The calculated effective coefficient can be easily transformed to static friction coefficient μ_S, using following formula:

$$\mu_s = \mu \sin\frac{\beta}{2} \tag{2}$$

where:

β – groove angle (for pk belts: $\beta = 40°$).

The calculation of the effective coefficient of friction results from the assumption that the multi-groove belt has been interpreted by a model of flat belt. This assumption is convenient during determining the models of belt transmissions with friction considered.

Figure 8 shows the calculated values using the formula above.

An analysis of the results shows that the calculated values of the effective friction coefficient vary along with the preload axial force of belt F_0. As can be observed, the value decreases when this force increases. This is primarily due to changes in roughness on the surface of the rubber when it is pressed to the pulley.

There are also visible significant differences in the characteristics in the section of the smaller wrap angles 30° and 45°. For angle $\alpha = 30°$, a smaller slope of the curve and its significant curvature can be observed. For angle $\alpha = 45°$, the curve only slightly differs in shape

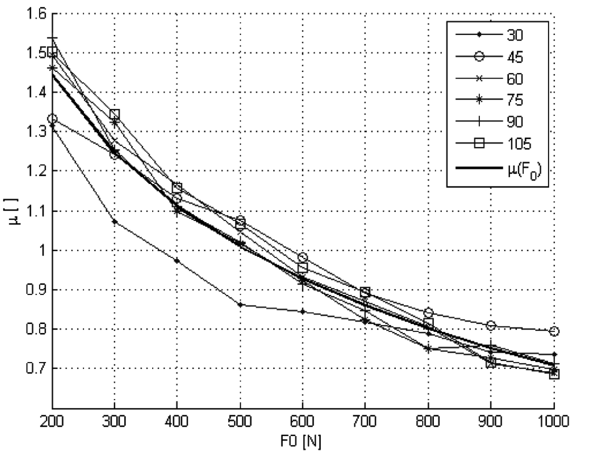

Fig. 8. Dependence of the effective friction coefficient on force F_0 and wrap angle α and the calculated approximation curve

Rys. 8. Zależność pozornego współczynnika tarcia od siły F_0 i kąta opasania α oraz wyznaczona krzywa aproksymacyjna

from those representing larger wrap angles, and it is closer to the rest of them. The characteristics are very similar for larger wrap angles, and this means that the friction coefficient for the specified value of F_0 does not depend much on the wrap angle.

It was decided to approximate the measured values with some power function. During some of the preliminary experiments, it was noticed that it was difficult to approximate the measured courses with the function of only two coefficients. The approximation function of the effective friction coefficient as a function of the F_0 force with three coefficients was assumed as follows:

$$\mu_f(F_0) = aF_0^b + c \tag{3}$$

where a, b, c are unknown coefficients.

The values of the unknown coefficients should be determined using the selected method of nonlinear optimisation by comparing the resulting curves with the results obtained experimentally. The norm in the least-squares sense as an objective function was assumed, which can be described as follows:

$$F_{fit} = \sqrt{\sum_{i=1}^{n} \left(\mu_m - \mu_f\right)^2} \tag{4}$$

μ_m, μ_f are values of effective friction coefficients for the assumed force F_0, achieved from measurements and the optimisation method, respectively.

The random search method was used to find the rough minimum of the objective function, and the Nelder-Mead optimisation method **[L. 18]** was used to find the exact minimum.

Figure 8 presents the calculated shape of the objective function. The calculated values of the coefficients were a = 8.05, b = -0.11, and c = -2.97. The value of F_{fit} that was achieved was 0.43. Because the curves for wrap angles α = 30° and 45° differ so much from the others, it was decided that they would be analysed separately.

Figure 9 shows the achieved curve for the range of wrap angles α = 60°–105° only. The calculated values of the coefficients were a = 10.84, b = -0.07, and c = -5.83. The value of F_{fit} was 0.16, so it gathered significant improvement.

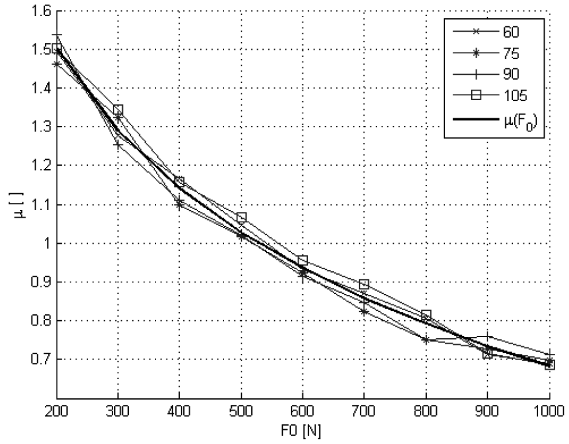

Fig. 9. Course of the calculated approximation curve for wrap angles 60° and above

Rys. 9. Przebieg obliczonej krzywej aproksymacyjnej dla przypadków kątów opasania 60° i większych

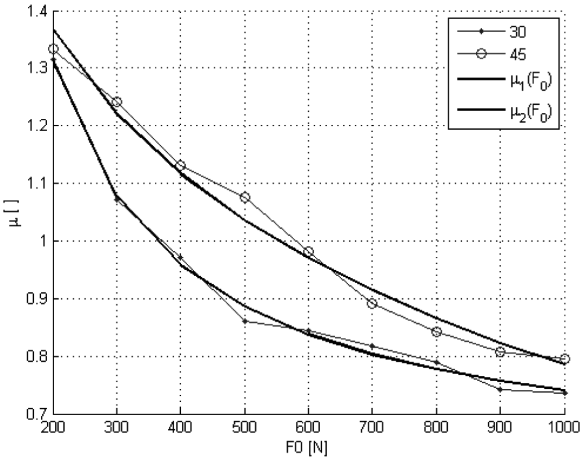

Fig. 10. Courses of calculated approximation curves for wrap angles 30° and 45°

Rys. 10. Przebiegi obliczonych krzywych aproksymacyjnych dla kątów opasania 30° i 45°

By including the values of the coefficients in Equation (3) and using Dependence (1), the proportion F_1/F_2 can be written as follows:

$$\frac{F_1}{F_2}\left(\alpha\,,F_0\right)=e^{\alpha\left(10.84\,F_0^{-0.07}-5.83\right)} \tag{5}$$

It was then decided to use the process as described above to determine the curves of the belt wrap angles of 30° and 45°. **Figure 10** shows the resulting curves of the fitting process.

The values of the coefficients were calculated as follows:

– For angle 30°: a = 125.20, b = -0.97, and c = 0.59 (Function F_{fit} equals 0.04); and,
– For angle 45°: a = 36.99, b = -0.01, and c = -33.64 (Function F_{fit} equals 0.07).

As can be observed, it was difficult to find one general equation for every wrap angle that could describe the effective friction coefficient dependence from F_0.

CONCLUSIONS

As can be observed, the classic Euler-Eytelwein formula cannot be used in this application. It is particularly difficult to estimate how the static friction depends on the rest time between the measurements and lower wrap angles. In the paper, proposed some empirical formulas of static friction not only as a function of the wrap angle but also as a function of the preload force. It should be noted that the proposed simple models do not include such phenomena as belt flexibility or occurring sticking and slipping zones around the circumference of the pulley. The simplicity of the achieved formulas allows dynamic analyses of the belt transmission to be more efficient.

The values of friction forces depend primarily on the type of material from which the belt is made, i.e. its roughness. The quality of the pulley design or the type of coating covering the wheel surface have a big influence on its friction parameters. The tested wheel was covered with paint, which also influenced the determined values of friction forces. It is therefore expected that the frictional parameters will change with the wear of the coating. During the tests, attempts were made to prevent long periods of slip.

It would be advisable to determine how the value of the effective coefficient of friction depends on the number of ribs in the belt. Because it strictly depends on the stiffness of the belt-wheel contact, it probably change with increasing belt ribs. As mentioned in **[L. 10]**, the stiffness value does not depend linearly on the number of grooves. The best choice would be therefore to verify it experimentally.

The measurements of the dynamic friction coefficient and its dependence on relative velocity will be presented in further publications, and the so-called "kinetic and dynamic characteristics of dynamic friction" (measured in a steady and variable changing relative velocity, respectively) seem to be especially interesting. The most important is to investigate what the dynamic friction depends on. After the first measurements were made, it was noticed that, in some situations, it could depend on the relative velocity or even on acceleration.

REFERENCES

1. Yamaguchi T., Umetsu T., Ishizuka Y., Kasuga K., Ito T., Ishizawa S., Hokkirigawa K.: Development of new footwear sole surface pattern for prevention of slip-related falls, Safety Science, 50, 2012, pp. 986–994.
2. Li K.W., Chen C.J., The effect of shoe soling thread groove width on the coefficient of friction with different sole materials, floors, and contaminants, Applied Ergonomics, 35, 2004, pp. 499–507.
3. Cruz Gomez M.A., Gallardo-Hernandez E.A., Vite Torres M., Peña Bantista A.: Rubber steel friction in contaminated contacts, Wear, 302, 2013, pp. 1421–1425.
4. Mofidi M., Kassfeldt E., Prakash B.: Tribological behaviour of an elastomer aged in different oils, Tribology International, 41, 2008, pp. 860–866.
5. Anyszka R., Bieliński D., Mężyński J., Grams J., Rehwinkel C., Moller B.: The influence of surfaces fluorination on tribological properties of rubber (in polish), Tribologia, 4, 2010, pp. 13–20.
6. Persson B.N.J.: Theory of rubber friction and contact mechanics, Journal of Chemical Physics, Vol. 115, No 8, 2001, pp. 3840–3861.
7. Persson B.N.J.: On the theory of rubber friction, Surface Science, 401, 1998, pp. 445–454.
8. Euler M.L.: Remarques sur l'effect du frottement dans l'equilibre. Mém. Acad. Sci., Berlin, 1762, pp. 265–278.
9. Fawcett J.N.: Chain and belt drives – a review. Shock Vibrations Digest, 13(5), 1981, pp. 5–12.
10. Čepon G., Manin L., Boltežar M.: Experimental identification of the contact parameters between a V-ribbed belt and a pulley. Mechanism and Machine Theory, 2010, Vol. 45, pp. 1424–1433.
11. Tambari S., Donaldson P.L., Benjamin I., Lelesi N.: Experimental investigation on the performance of a rope belt friction apparatus. IOSR Journal of Mechanical and Civil Engineering, 2015, Vol. 12, pp. 43–47.
12. Balta B., Sonmez F.O., Cengiz A.: Speed losses in V-ribbed belt drives, Mechanism and Machine Theory, 86, 2015, pp. 1–14.
13. Manin L., Michon G., Remond D., Dufour R.: From transmission error measurement to pulley-belt slip determination in serpentine belt drives: Influence of tensioner and belt characteristics, Mechanism and Machine Theory, 44, 2009, pp. 813–821.
14. Chen G.(S.), Lee J.H., Narravula V., Kitchin T.: Friction and noise of rubber belt in low temperature condition. The influence of interfacial ice film. Cold Regions Science and Technology, 71, 2012, pp. 95–101.
15. Sheng G., Lee J.H., Narravula V., Song D.: Experimental characterization and analysis of wet belt friction and the vibro-acoustic behavior, Tribology International, 44, 2011, pp. 258–265.
16. Kubas K.: A research stand for measuring friction parameters in a belt transmission, The Archives of Automotive Engineering, 2017,75 (1), pp. 69–83.
17. Kubas K.: A two-dimensional discrete model for dynamic analysis of belt transmission with dry friction, The Archive of Mechanical Engineering, Vol. 61, No. 4, 2014, pp. 571–593.
18. Nelder J.A., Mead R.: A simplex method for function minimalization, Computer Journal, 7, 1965, 308–313.