

Numerical Simulations and Experimental Investigations of Contact Phenomena in a Mechanical Friction Clutch

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Abstract

Numerical simulations and experimental investigations of contact phenomena occurring on the contact surface of frictional shields of a mechanical friction clutch have been studied in the work described in this paper. The mentioned problems have already been studied earlier, but only simplified mathematical models have been used and applied so far. In this work elasticity as well as wear properties of the material of frictional shields rubbing themselves are taken into account. A general non-linear differential model of wear and a wear model in the integral form are considered. Moreover, numerical and experimental results describing thermal phenomena (heat generation and its propagation) occurring in the considered system are presented. Many interesting numerical and experimental results are obtained, illustrated and discussed. Finally, numerical simulations agree with the experimental data.

Keywords: clutch, friction, wear process, heat production, thermal phenomena, numerical simulations, experimental investigations.

1 Introduction

A clutch belongs to the elements of a mechanical system used for coupling shafts and transmitting torque between them. Historically, the oldest simple clutches were used for direct connection of coaxial shafts. Future demands and tendencies in technology of producing clutches make many requirements referring their structure, functioning, strength or life. For this reason, the basic issues in designing power transmission systems are, among others, increase of productivity and improvement of functioning quality of driven machines, as well as enhancing reliability degree and obtaining better economic indices of these systems. In order to meet these assumptions in friction clutches, appropriate knowledge of mathematical modeling and description of this

system, as well as contact phenomena and tribological processes (friction, wear, heat generation and its propagation) occurring therein, is important. This approach allows better predicting the behaviour of real systems of this type.

Issues related to interaction of dynamics and contact phenomena as well as accompanying tribological processes in different types of mechanical systems (bars, transmissions, gears, guides, bearings, clutches, brakes, and others) have been the object of interests and investigations of many scientists for many years. Different mathematical models have been used in investigations related to dynamics and tribological processes occurring in the systems with a friction clutch or brake to describe them. Usually, during analysis of them the contact phenomena and accompanying tribological processes were not taken into account. In most cases simplified mathematical models were used therein, as a rule separately for particular issues and without mutual interrelations between them.

In this work, a considered friction clutch is treated as the frictional connection of elastic bodies taking into account elasticity in axial direction of the material of friction linings. A general non-linear differential model of wear and an integral model of wear taking into consideration hereditary and memory processes were described. Non-uniform contact pressure distributions on the contact surface of clutch linings have been determined. Changes of the friction torque transmitted by the clutch resulting from the change of contact pressure distribution have been calculated. The proposed numerical model allowing determination of non-uniform temperature distribution on the contact surface of friction linings for any instant of time has been used. Finally we show, that presented numerical results coincide with the experimental data.

2 State-of-the art

Wear is a dynamical process, which is related to a change of surfaces of bodies moving relative to one another, as a result of mechanical interaction between them. This process depends on many factors and parameters, like geometry of contacting surfaces, normal force applied, rubbing speed, material hardness, etc. [5]. Studies on wear process and its modeling have been carried out for many years. One of the first scientists investigating wear processes was Archard, who proposed a linear model of wear for metals [4]. Since then many mathematical models describing wear processes in friction joints of different type, in different external conditions and for different materials, have come into being yet. In literature related to tribology there may be found about 300 different models of wear, from simple empirical equations to complex mathematical relationships [5]. Numerical calculations of wear processes in different type of friction joints may be found in the works [5, 11, 18, 19, 21], and many others.

In the general case of modeling wear processes in different type of friction connections of mechanical systems general models of wear are used. According to Archard [4], the model of wear written in the differential form takes the following form

$$\frac{dw(t)}{dt} = K^{(w)} |V_r(t)| P(t), \quad (1)$$

where t is time, $w(t)$ is wear, $K^{(w)}$ is material wear coefficient, $V_r(t)$ is relative rubbing speed of rubbing surfaces, while $P(t)$ is contact pressure between them. It is the linear model of wear, considering contact pressure and rubbing speed of rubbing surfaces. In this work for modeling processes of wear of clutch friction linings the general non-linear differential model of wear has been used, governed by the Equation [19]

$$\frac{dw(t)}{dt} = K^{(w)}(T'(t))|V_r(t)|^\beta P^\alpha(t), \quad (2)$$

where wear coefficient $K^{(w)}(T'(t))$ is a function of temperature $T'(t)$ on the contact surface, and α and β coefficients are quantities dependent on the model of wear, grade of machining and lubrication of rubbing surfaces. Thus, it is the non-linear model of wear, wherein the speed of wear is the non-linear (exponential) function of contact pressure and rubbing speed of rubbing surfaces. The presented non-linear model of wear was used earlier in the works [10, 19], and others.

At variable conditions of external load so called delay effects may be observed [19, 21]. For some frictional materials, in spite of stable conditions of wear processes, the wear coefficient changes with time, e.g. as a result of ageing or wearing-in of friction linings. Then, there is a necessity to use other models of wear than presented above. An adequate mathematical description of such wear process is the integral model of wear in the form of [11, 19]

$$w(t) = \int_0^t K^{(w)}(T'(t'))|V_r(t')|K'(t, t')P(t')dt', \quad (3)$$

where $K'(t, t') = K'_1(t')K'_2(t - t')$, wherein $K'_1(t') = 1 + c \cdot \exp(-\gamma't')$ and $K'_2(t - t') = 1 - \exp(-\gamma''(t - t'))$ are so called hereditary and memory kernels. The exponential functions in the presented model of wear are responsible for decreasing the speed of wear process, even in stationary conditions. The model of wear (3) for $K'_1(t') = 1$ and $K'_2(t - t') = 1 - \exp(-\gamma''(t - t'))$ was used in the work [21], where for a model of contact of a thermoelastic layer with a thermally insulated plate the solution was obtained with taking into account wear and heat generation. An integral model of wear was also applied, among others, in references [18, 20].

Thermal processes are objects of interest of many researches working in mechanical engineering. Different friction laws have been applied in various machine members like gears, sprockets, bearings, clutches, brakes, and others. In general, the mentioned processes are very difficult for the proper and reliable analyses, and in addition they usually may depend on many parameters. For instance, the work [17] describes how to determine the surface temperature of the clutch taking into account the whole studied object as well as the temperature of friction surfaces. In addition, to study the clutch as a whole mechanical system, also approximate computations of the temperature of its working surfaces are carried out. In this case, as a model the plate with unlimited diameter and being heated from both sides by a uniform heat flux is considered. This model has also been presented in [17].

In order to obtain in a more accurate manner a description of the processes of heat production and propagation associated with friction in the clutch or brake, general relationships and interactions of these processes should be used. One of the basic equations describing the mentioned phenomena is the heat conduction equation. For example, in the work [1] these equations were used to determine the temperature of the shaft of cylindrical bearing, while in the work [2] the same authors have used them to determine the temperature of the spherical bearing. In reference [16] heat conduction equation is used to determine the temperature at the interface between two thermo-elastic semi-spaces pressed together by a normal loading, and application of this equation to describe the thermal phenomena occurring in the aircraft carbon-carbon composite multi-disc clutch (similar to an aircraft brake) can be found in [22]. In general, the heat flux density $q(t)$ generated on the clutch working surfaces can be determined as a part of the work of friction force generated per unit time, and penetrated through the unit area. The heat flux density has the form [3]

$$q(t) = (1 - \chi)\mu|V_r(t)|P(t), \quad (4)$$

where χ is a part of the work of friction force not converted into heat (this part of the work is associated with wear), μ is the coefficient of friction, $V_r(t)$ is the relative sliding velocity, and $P(t)$ is the contact pressure distribution at the interface of bodies. Formula (4) of the heat flux density is used in references [14, 15] for a three-dimensional thermo-elastic contact problem with frictional heat generation. This formula has also been used in the work [8], where the finite element method analysis of thermal phenomena in the friction clutch with ceramic disc is carried out. The presented relationship of the heat flux density is also applied in references [16, 18] for various friction approximations. Heat conduction in the material is governed by the Fourier's law of the following form

$$q(t) = -k^{(p)} \text{grad} T', \quad (5)$$

where $k^{(p)}$ is the thermal conductivity coefficient (thermal conductivity) of the material, and $\text{grad} T'$ is the gradient of temperature T' perpendicular to the isothermal surface. The heat flux density exchanged at the border between the body and its surroundings environment is

$$q(t) = \lambda(T' - T'_{ot}), \quad (6)$$

where λ is the coefficient of heat transfer between the body and its surroundings environment, T' is the temperature of the body at the border of its surroundings, and T'_{ot} is the ambient temperature. Formula (6) describes the mechanism of heat exchange between body and environment known as the second type boundary condition. An extensive literature review on scientific papers devoted to heat various types can be traced in review paper [12]. In the review work [9] papers published in recent years devoted to the analysis and calculations of thermal phenomena occurred in different kinds of friction brakes are also presented, illustrated and discussed.

3 Model of the system

Figure 1 presents a model of two-disc mechanical friction clutch and a cross-section of friction linings of this clutch with a computational grid (plotted on the cross-section of the linings divided into m equal segments along the radius, in nodes of which temperature values are being calculated).

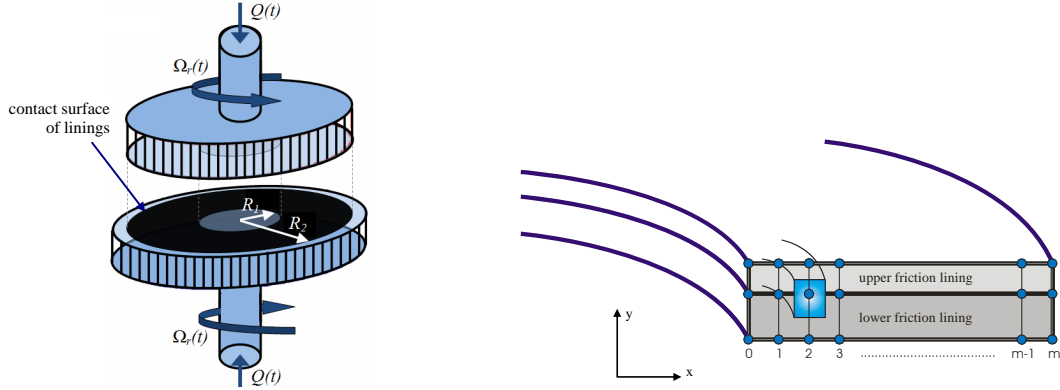


Figure 1: Model of two-disc mechanical friction clutch and a cross-section of linings of the considered clutch with a plotted computational grid.

The friction linings are attached to both discs of the clutch. The friction contact between these linings occurs in the ring area $R \in [R_1, R_2]$. The mentioned discs are being pressed by axial force $Q(t)$ and their relative angular velocity is equal to $\Omega_r(t)$, while contact pressure in any contact point and time is equal to $P(R, t)$. Material wear coefficients for the left and right friction linings, dependent on temperature $T'(R, t)$ in a given contact point and time, are $K_1^{(w)}(T'(R, t))$ and $K_2^{(w)}(T'(R, t))$, respectively. Stiffness coefficients, in turn, in axial direction of material of these linings are k_1 and k_2 , respectively. Thicknesses of the upper and lower linings are H_1 and H_2 , while thermal conductivities of respective linings are $k_1^{(p)}$ and $k_2^{(p)}$, respectively. Heat transfer coefficients between the upper (lower) friction lining and the upper (lower) clutch disc made of aluminium are λ_1 and λ_2 , respectively. Heat transfer coefficients between the upper/lower lining and environment in turn are λ_3 and λ_4 , respectively. Specific heats of materials of which the linings are made are c_{w1} and c_{w2} , respectively, while densities of material of which these linings are made are ρ_1 and ρ_2 . The detailed mathematical description of processes of wear and heat generation and propagation in the mechanical friction clutch is presented in our previous works [6, 7, 13]. Integrals occurring in differential, integral and integro-differential equations obtained in dimensionless form were written applying the trapezium method. Appropriate equations were solved using the fourth order Runge-Kutta method and the Gauss-Jordan elimination method. In this work the results of numerical simulations of obtained relationships are presented and compared to our own experimental investigations.

4 Numerical and experimental results

Presented computer simulations are obtained using own numerical algorithms written in C++. Moreover, our numerical simulations are verified experimentally. Main parts of the experimental stand are presented in Figure 2.

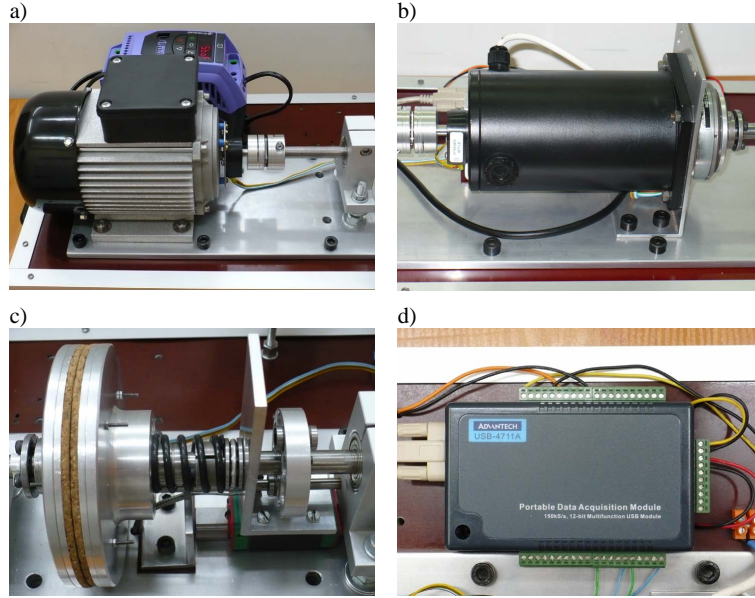


Figure 2: Main parts of the experimental stand.

The stand comprises a typical mechanical system with a friction clutch operating in a mechatronic system. It consists of a driving part (Fig. 2a), driven part (Fig. 2b) and friction clutch (Fig. 2c). The driving part includes an asynchronous motor controlled with a single-phase AC inverter. For determining angular position of the active part of the clutch (motor) serves an optical incremental encoder. The driven part includes a DC motor working as a generator which causes appropriate anti-torque depending on a connected load. Furthermore, in the driven part of the system there is a friction brake and an optical incremental encoder. The transmitted torque is measured with a dynamic torque sensor. A member coupling both devices is the mechanical friction clutch. A member coupling the mechanical system with computer software is the control and measuring module USB-4711A (Fig. 2d).

The results of conducted experimental investigations for stationary conditions ($Q(t)$, $\Omega_r(t) = \text{const.}$) have been compared to the numerical computations. The friction linings used for investigations were made of pressed cork, which is a natural material used for friction linings. In Figure 3 a comparison of the results of numerical simulations with the experimental results is presented, that is: the changes of friction torque transmitted by the clutch (Fig. 3a) and the contact pressure distributions on the contact surface of the linings (Fig. 3b).

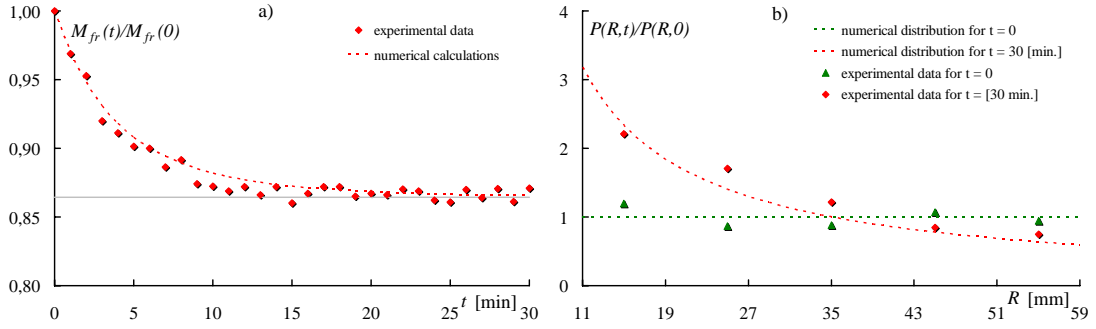


Figure 3: Comparison of numerical with experimental results: a) the changes of transmitted friction torque; b) the contact pressure distributions.

On the ground of obtained results may be assumed approximately that, according to the numerical solution, the actual values of friction torque transmitted by the clutch decrease as the process of wear of the clutch friction linings advances. Even though the experimental results of contact pressure distribution do not correspond with the numerically obtained results with relatively good accuracy, yet substantial changes of the contact pressure distribution according to the simulations may be noticed.

As examples, the numerical results of the model describing thermal phenomena in the friction clutch are presented below. At the beginning the case of pressures uniformly distributed ($P(R, t) = \text{const.}$) on all contact surface of the clutch friction linings is considered. Figure 4a presents the distributions of dimensionless temperature $T(r, \infty)$ in a steady state, i.e. when these distributions have a steady form on all contact surface of the linings, independently of time. Similar numerical analysis of the mathematical model describing thermal phenomena in the friction clutch has been also carried out for the case of pressures non-uniformly distributed ($P(R, t) = A/R$, $R = \text{const.}$) on all contact surface of the clutch friction linings and presented in Figure 4b. Moreover, Figure 5 presents, as examples, the real (obtained using infrared camera) surface distributions and the profiles of temperature along the radius of the lining obtained for the uniform (on the left) and the steady non-uniform (on the right) pressure distribution. In both cases the temperature distributions achieve a steady state due to establishing of thermal equilibrium between the heat generated in the clutch and the heat transferred to the environment of the linings, i.e. to the clutch discs and to a medium surrounding them (e.g. the air). In the first case (uniform contact pressure distribution) inside the contact surface of the linings temperature changes linearly with the dimensionless radius r , whereas in the second one (non-uniform contact pressure distribution) the temperature is constant independently of the radius r . In both cases at boundaries of contact of linings temperature is significantly lower. This decrease of temperature at the boundaries of contact of the linings results from heat exchange between the linings and the surrounding air. Deviations from the linear or constant relationship result from the fact that actually there takes place a heat exchange between the lining and its environment at the contact boundaries and hence, the values of temperature in these places of the lining are adequately lower.

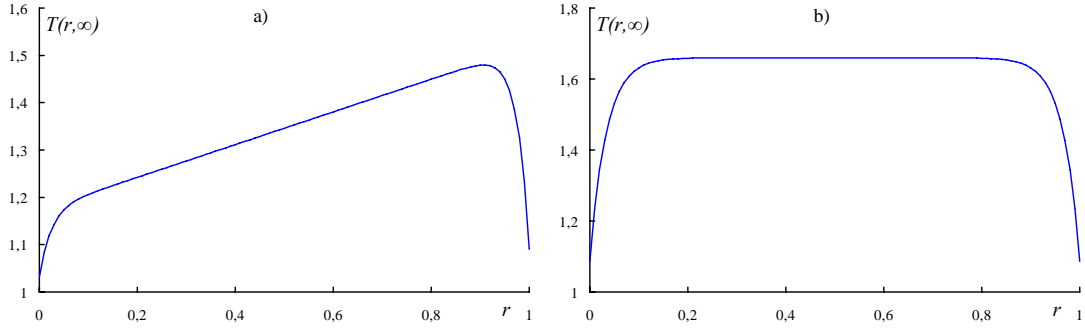


Figure 4: Steady temperature distributions on the contact surface of the linings: a) for pressures uniformly distributed; b) for pressures non-uniformly distributed.

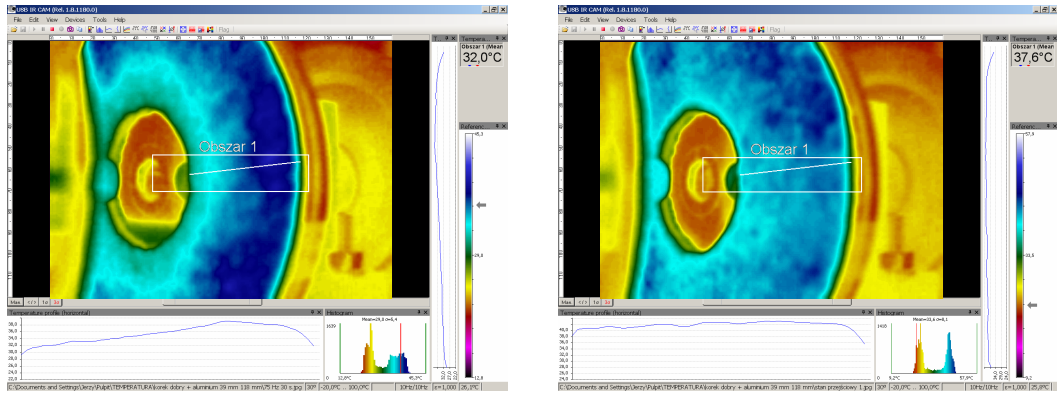


Figure 5: Surface temperature distributions and profiles on the friction surface for the uniform (on the left) and the steady non-uniform (on the right) pressure distribution.

5 Conclusions

The work presented in this paper is devoted to investigations of tribological phenomena and processes occurring on the contact surface of linings of a mechanical friction clutch. The results concern both the processes of wear of the friction linings and the processes of heat generation and propagation in the friction clutch as a result of friction. The goal of the experimental investigations carried out in the range of the work was confirmation of the proposed mathematical models describing tribological processes occurring in the mechanical friction clutch. The processes of wear of the clutch friction lining material have been verified experimentally. The results were compared with the numerical computations obtained for the linear wear model only. However, it was shown that decreasing the moment of the friction force transmitted by the clutch at a constant force pressing the discs, or changes of the contact pressure distribution occur in accordance with the proposed mathematical model. The performed simple experimental qualitative verification of the model describing thermal phenomena in a friction clutch indicates relatively good qualitative conformity of the numerical solutions with the results of the experimental studies. In the result of the conducted

numerical analysis for a wider range of parameter changes it was possible to determine the non-uniform contact pressure distributions as well as the wear of particular clutch linings on the surface of contacting friction materials. It allows a better understanding of wear mechanisms of clutch linings, which may be used *e.g.* for strength analysis of these systems. Moreover, consideration of the contact pressure distribution changes allows more accurate determination of the friction torque transmitted by the clutch. Concurrent modelling of friction phenomena, wear processes and thermal processes in a clutch allows a more accurate determination of the friction torque, which makes it possible to more accurately predict the dynamics of all power transmission system which includes a clutch considered in the work. Finally it has been shown, that obtained and presented above numerical simulations coincide with data obtained experimentally.

Acknowledgments

The work has been supported by the Ministry of Science and Higher Education of Poland under the grant No. 0040/B/T02/2010/38 for years 2010-2012 and MASTER Program of the Foundation for Polish Science for years 2010-2012.

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