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FINITE ELEMENT ANALYSIS OF THE THERMAL BEHAVIOUR OF SINGLE-DISC CLUTCHES DURING REPEATED ENGAGEMENTS

ANALIZA MES ZACHOWAŃ TERMICZNYCH SPRZĘGIEŁ JEDNOTARCZOWYCH PODCZAS POWTARZANYCH ZAŁĄCZEŃ

Key words:

thermal analysis, friction clutches, dry condition, FEM

Słowa kluczowe:

analiza termiczna, sprzęgła cierne, MES

Abstract

Most of the failures in the sliding systems occur due to the high thermal stresses, which generated at the interface between the contacting surfaces due to sliding between parts, such as friction clutches and brakes. In this paper, the thermal behaviour of a single-disc clutch is investigated. The surface temperatures of the friction clutch disc will be increased during repeated

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engagements, in some cases, will lead to premature failure of the clutch disc. In order to avoid this kind of failure, it the surface temperature should be calculated with high accuracy to know the maximum working temperature of the friction system. In this work, the temperature distributions are computed during four repeated engagements at regular intervals (5 s) for the same energy dissipation. Three-dimensional finite element models are used to simulate the typical friction clutch disc.

List of Symbols and Abbreviations

Symbol	Description
$P_{mech.\ input}$	The mechanical energy input to the clutch system
$P_{mech.\ output}$	The mechanical energy output from the clutch system
$P_{therm.\ output}$	Thermal energy losses
t	Time
t_s	Sliding time
T_i	The initial temperature of the clutch system
T_{max}	The maximum temperature of the clutch system
t_{max}	The time corresponding to the maximum temperature T_{max}
T_f	Final temperature of the clutch system at the end of clutch cycle
$Q_{gen.c}^{t+\Delta t}$,	The energy generated on the clutch disc
$Q_{conv.c}^{t+\Delta t}$	The heat transfer by convection of clutch disc
$\Delta Q_{Int.\ energ.c}$	The change in the internal energy of a clutch disc
V	The volume
c	The specific heat
ρ	Density of body
$Q_{I.\ int.\ energ.}$	Initial internal energy of the clutch disc
$[C]$	The heat capacity matrix
$[K]$	The heat conductivity matrix
$\{F\}$	Thermal load
V_{ele}	Volume of element
$T_{ele.\ (t)}$	Temperature of element at time (t)
$T_{ele.\ (t+\Delta t)}$	Temperature of element at time (t)

INTRODUCTION

Friction clutches and brakes are considered to be the most common type used in automotive application. Two or more surfaces are pressed together by a normal force to create a friction torque. The friction surfaces could be flat and perpendicular to the axis of rotation. **Figure 1** shows the parts of typical single-disc clutch system. In the design of friction clutches, it is very important to calculate the temperature distribution on the contact surfaces accurately during the slipping period when the clutch starts to engage. High thermal stresses in the contact surfaces produce several disadvantages, such as surface cracks and permanent distortions, and they are likely to lead the friction facings to failure, before the expected lifetime of the clutch. For these reasons, investigations of the effects of the repeated engagements on the thermal behaviour (maximum temperature) of the friction clutches are necessary.

Abdullah and Schlattmann **[L. 1–8]** investigated the temperature field and the energy dissipated of a dry friction clutch during the beginning of engagement, assuming uniform pressure and uniform wear conditions. They also studied the effect of contact pressure between the contact surface when varying with time on the temperature field and the internal energy of the clutch disc using two approaches: the heat partition ratio approach to compute the heat generated of each part individually, and the second approach that applies the total heat generated of the whole model using the contact model. Furthermore, they studied the effect of engagement time, sliding speed function, thermal load, and dimensionless disc radius (inner disc radius/outer disc radius) on the thermal behaviour of the friction clutch in the beginning of the engagement.

Yevtushenko et al **[L. 9]** investigated a transient thermal problem of three-elements (disc/pad/calliper) with time-dependencies. The effects of Biot number and the duration when the pressure is increasing (from zero at the initial moment of time to the nominal value at the moment of a stop) on the values of the temperature of the cast iron disc/metal ceramic and pad/steel calliper have been studied. The research results showed that the effect of Biot number will reduce the heat transfer through the contact surface.

Seo et al. **[L. 10]** suggested a thermal model to estimate the temperature distribution of a wet clutch in 4WD coupling to avoid the thermal failure of the clutch plate during the operating condition of the vehicle. The results of this model are validated with the experimental results of the actual 4WD vehicle under the limitation of torque 900 [N.m]. The theoretical results have shown acceptable agreement with the experimental results.

Adamowicz and Grzes **[L. 11]** studied the influence of convective heat transfer on the transient temperature distribution of a real disc brake. They used the finite element method to investigate an impact of the heat transfer coefficient on the amount of heat dissipation from the solid disc rotor of a disk brake. The results showed that, in terms of a single braking process for the specified dimensions and thermo-physical properties of materials, the amount of convection heat transfer does not allow significantly reducing the temperatures of the rotor. However, in the following release period, after the braking action when the velocity of the vehicle remains on the same level, a considerable decrease in temperature values has been observed.

Yevtushenko and Kuciej **[L. 12]** solved the transient heat conduction problem analytically and numerically. They computed the surface temperature of a tribosystem consisting of the pad sliding with time-dependent velocity on a disc surface. The two-dimensional finite element model was used to study the thermal stresses during the braking process and the effect of boundary conditions on the temperatures and thermal stresses of the tribosystem. They concluded that the value of thermal stresses decreases with a significant increase of braking time.

This paper highlights the importance of the effect of the repeated engagements on the thermal behaviour (the magnitude and distribution of surface temperature) of the friction facing of clutches, assuming the pressure is uniform between the contact surfaces.

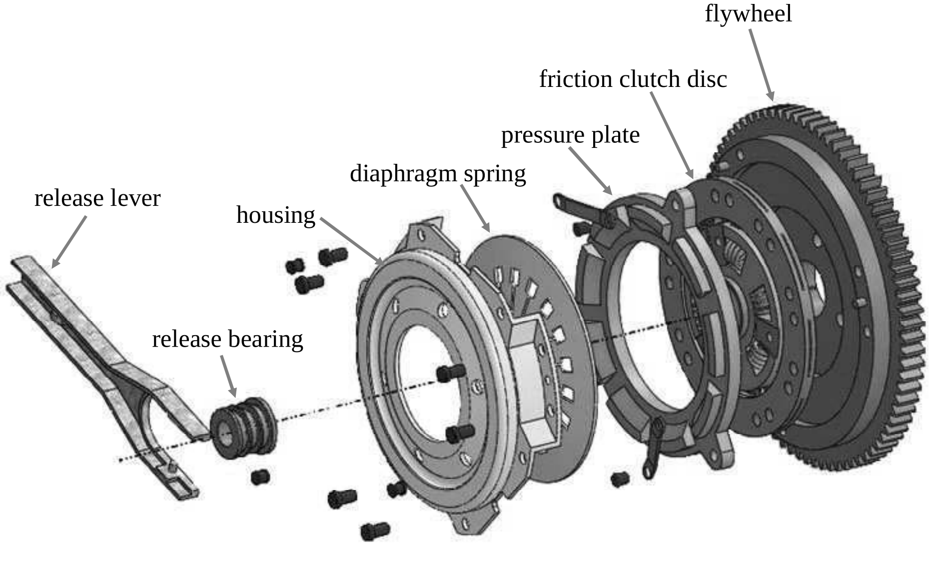

Fig. 1. Single-disc friction clutch
Rys. 1. Jednotarczowe sprzęgło tarciowe

ENERGY CONSIDERATIONS

Heat transfer is a kind of energy transportation, and there are basically three types of heat transfer mechanisms:
Conduction,
Convection, and
Radiation.

The time period of slipping in the clutch system is very short, due to this fact, the effect of radiation can be neglected **[L. 10]**. **Figure 2** shows the power flow in a typical system of a friction clutch, it is clear from this figure that there are two types of energy in the system, i.e. mechanical energy ($P_{mech.\ input}$ and $P_{mech.\ output}$) and thermal energy ($P_{therm..\ output}$). The difference between the mechanical power input and output represents the amount of power that turns into heat energy, and this energy will be the thermal load on the contact surfaces. Due to this thermal load, the temperature of the clutch system will increase. The maximum thermal load occurs during the slipping, and the time that the maximum thermal load occurs depends on the function of the pressure between contact surfaces with time.

Figure 3 illustrates the maximum temperature of the contact surfaces of a friction clutch as a function of time during a single engagement. This curve shows the temperature of contact surfaces during two phases; the first one represents the heating phase (slipping period) and the second one represents the cooling phase (full engagement). From this figure it is clear, at the beginning of engagement (t = 0), the temperature of the surfaces is equal to the initial temperature of the clutch system (T_i) and then the temperature will increase to maximum temperature T_{max} during the slipping period [$0 < t < t_s$] (the time t_{max} corresponding to the maximum temperature T_{max}, it depends on the type of function of the thermal load). Finally, the temperature will be decreased after slipping to the final temperature T_f, and the value of T_f depends on the convection factor and the time of the cooling phase.

Figure 4 demonstrates the input and output energies of the clutch disc during the period of time (ΔT). The equation of the energy balance of the frictional facing of the clutch when time changes from (t) to (t+Δt) is as follows:

$$Q_{conv.} = Q_{gen.c} - \Delta Q_{Int.energ.}\,; \quad t > 0 \; Q_{conv.} = Q_{gen.c} - \Delta Q_{Int.energ.}\,; \quad t > 0 \qquad (1)$$

where $Q_{gen.}$, $Q_{conv.}$ and $\Delta Q_{Int.\ energ.}$ are the energy generated during the slipping period, the heat transfer by convection and the change in the internal energy of the clutch disc, respectively. The change in the internal energy of the clutch disc is as follows:

$$\Delta Q_{Int.energ.} = Q_{t+\Delta t} - Q_t = \rho\ V\ c\left(T_{t+\Delta t} - T_t\right); \quad t > 0 \qquad (2)$$

Substituting equation (2) into equation (1) and re-arranging yields:

$$Q_{conv.} = Q_{gen.c} - \rho V c\left(T_{t+t\Delta} - T_t\right); \quad t > 0 \qquad (3)$$

The internal energy of the clutch disc at any time is

$$Q_{Int.energ.} = Q_{I.\mathrm{int}.energ.} + \Delta Q_{Int.energ.} = Q_{Int.energ.(t+\Delta t)} = \rho V c\, T_{t+t\Delta};\quad t > 0 \tag{4}$$

where $Q_{I.\ int.\ energ.}$ is the initial internal energy of the clutch disc.

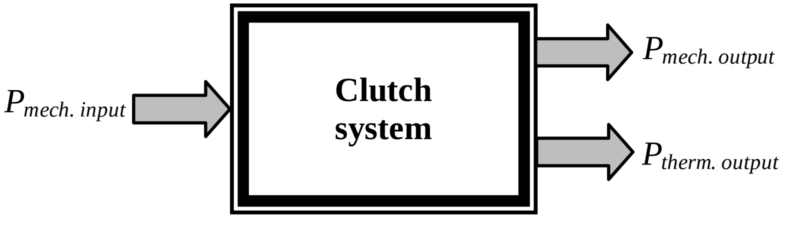

Fig. 2. The power flow in typical clutch system
Rys. 2. Przepływ strumienia mocy w typowym systemie sprzęgłowym

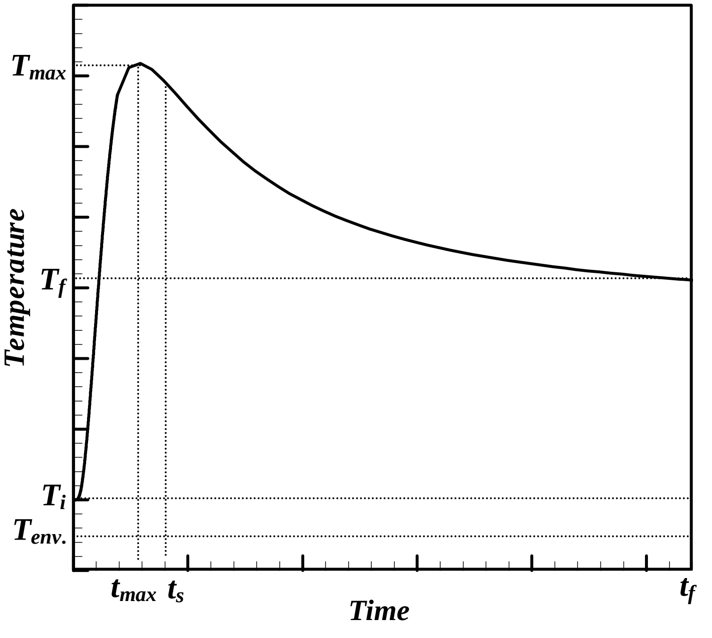

Fig. 3. Variation of the maximum temperature of a clutch system with time
Rys. 3. Przebieg maksymalnej temperatury w systemie sprzęgłowym w funkcji czasu

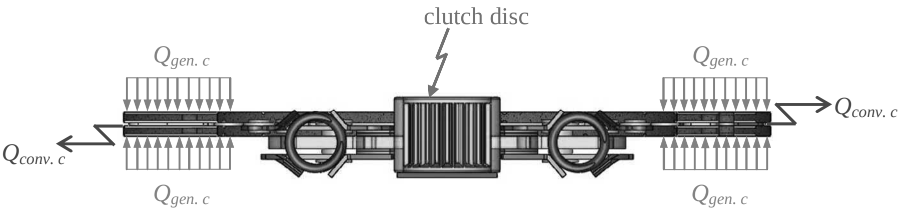

Fig. 4. Input and output thermal energies of a friction clutch disc
Rys. 4. Strumienie energii na wejściu i wyjściu dla tarczy sprzęgła ciernego

FINITE ELEMENT FORMULATIONS

Transient condition involved a time dependent function of the heat transfer analysis. During the transient condition, the temperature change in a unit volume of material is resisted by thermal mass that depends on the mass density ρ of the material and its specific heat c. The finite element formulation can be expressed as **[L. 13]**

$$[C]\{\dot{T}\}+[K]\{T\}=\{F\} \tag{5}$$

where $[C]$ is the specific heat matrix, $[K]$ the conductivity matrix, $\{T\}$ the vector of nodal temperatures, $\{\dot{T}\}$ is the derivative of temperature with time $(\partial \dot{T} = \partial T / \partial t)$, and $\{F\}$ is the applied heat flows. The total amount of heat transfer by the convection of the frictional facing of clutch when time changes from (t) to (t+Δt) is **[L. 1]:**

$$Q_{conv}=Q_{genc}-\sum_{i=1}^{No.ofelemnt}\rho V_{ele.}c\left(T_{ele(t+t\Delta)}-T_{ele(t)}\right);t>0 \tag{6}$$

where $V_{ele.}$, $T_{ele.\ (t)}$ and $T_{ele.\ (t+\Delta t)}$ are the volume of element, temperature of element at time (t) and the temperature of element at time (t+Δt), respectively. The total amount of the internal energy of the clutch disc at (t+Δt) is

$$Q_{Int.energ..}=\sum_{i=1}^{No.ofelemnt}\rho V_{ele.}c\ T_{ele.(t+t\Delta)};\ \ t>0 \tag{7}$$

In order to determine temperature distribution of this transient heat conduction problem, a fine mesh element was essential. Moreover, when the iterative method of the given problem is employed, then a relatively short time step is needed for the calculations. In the next step of the study, the Crank-Nicolson method was selected as an unconditionally stable scheme. In this paper, ANSYS13 software was used to investigate the transient behaviour of a dry friction clutch (single disc clutch). Uniform pressure between the contact surfaces is assumed in this analysis. In all computations for the friction clutch model, a homogeneous and isotropic material has been assumed and all parameters and materials properties are listed in Table. 1.

The heat transfer coefficient has been taken as 40.89 W/m^2 **[L. 2]** and is assumed to be constant over all exposed surfaces, and the slipping time is 0.4 s. **Figure 5** shows the three dimensional finite element models of the friction facing of clutch discs. The first model is the clutch without grooves and the second model is the clutch with grooves. These models of clutch discs are built using ANSYS software **[L. 14]**. The twenty-node thermal element (SOLID90) was used in this analysis. These elements have compatible temperature shapes and are well suited to model curved boundaries. A mesh sensitivity study was done to choose the optimum mesh from a computational accuracy point of view.

Table 1. Model parameters and material properties
Tabela 1. Parametry modelu i własności materiałowe

Parameters	Values
Inner disc radius of friction material and axial cushion , r_i [m]	0.06298
Outer disc radius of friction material and axial cushion , r_o [m]	0.08721
Thickness of friction material [m]	0.003
Thickness of the axial cushion [m]	0.0015
Inner disc radius of pressure plate [m]	0.05814
Outer disc radius of pressure plate [m]	0.09205
Thickness of the pressure plate [m]	0.00969
Inner disc radius of flywheel [m]	0.04845
Outer disc radius of flywheel [m]	0.0969
Thickness of the flywheel [m]	0.01938
Pressure, p [MPa]	1
Coefficient of friction, μ	0.3
Number of friction surfaces	2
Torque [Nm]	432
Maximum angular slipping speed, ω_o [rad/sec]	200

Conductivity of friction material, [W/mK]	0.6
Conductivity of pressure plate and flywheel, [W/mK]	42
Density of friction material, [kg/m^3]	1570
Density of pressure plate, flywheel and axial cushion, [kg/m^3]	7800
Specific heat of friction material, [J/kg K]	534
Specific heat of pressure plate, flywheel and axial cushion, [J/kg K]	450
Initial temperature, T_i [K]	300
Time step, Δt [s]	0.001
Number of engagements, n	4

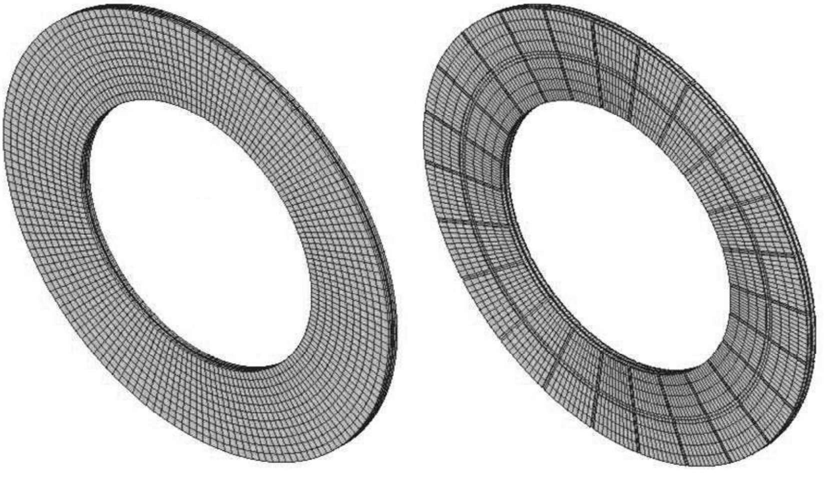

Fig. 5. Finite element models of friction facings of clutch disc (with and without grooves)
Rys. 5. Modele MES powierzchni tarcia tarczy sprzęgła (z rowkami i bez)

RESULTS AND DISCUSSIONS

In the practical application, the friction clutch makes repeated engagements, and the temperature fields (especially the maximum temperature) during these engagements are considered essential for the designer. The temperature distributions were computed during 4 repeated engagements at regular interval (5 s) for the same energy dissipation. In this work, the pressure was assumed uniform between contact surfaces.

The approach of this work was compared to the numerical results of Fu et al **[L. 15]** to find the maximum temperature ($T_{max.}$) at inner and outer radii of a friction clutch. Table 3 shows the current results with numerical results of Fu et al **[L. 15]** and the values of percentage difference with numerical results. In this table, the maximum difference does not exceed (1%). The data for the verification case are shown in **Table 2**.

Table 2. The parameters and material properties for verification case (Fu et al [L. 15])
Tabela 2. Parametry i własności materiałowe dla przypadku weryfikacji (Fu et al. [L. 15])

Properties	Friction material properties	Steel properties
Dimensional parameters	interior diameter, D1 = 252 mm, external diameter, D2 = 386mm; thickness, t2 = 5 mm	Thickness t1 = 10mm t3 = 5mm
Modulus of elasticity (MPa)	70	70 2×10^5
Thermal conductivity (W/mK)	0.25	48
Specific heat capacity (J/kg K)	1337.6	480
Density, (kg/m^3)	1300	7800

Table 3. The values of maximum temperature at inner and outer radius
Tabela 3. Wartości temperatury maksymalnej dla wewnętrznego i zewnętrznego promienia

	$T_{max.}$ at r_i [K]	$T_{max.}$ at r_o [K]
Present work	533.3	639.5
Fu et al [15]	536.4	642.6
% Difference	0.57	0.48

Figure 6 demonstrates the temperature distributions with a disc radius at the contact region between the flywheel and clutch disc at the first and third engagements. It can be seen from this figure that the maximum temperature is located approximately at r = 0.95 r_o, and the temperatures decrease at the regions near the inner and outer disc radii due to the effect of convection.

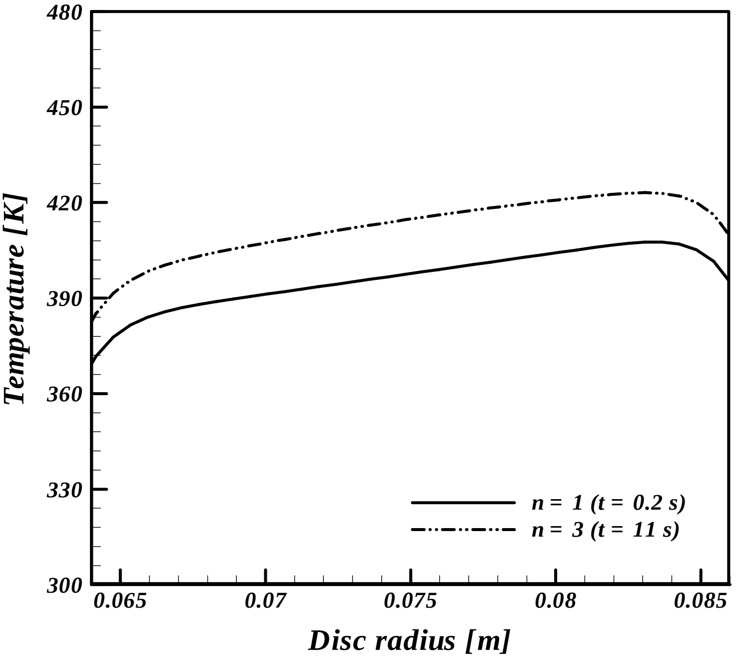

Fig. 6. Distribution of surface temperature with disc radius (flywheel side)
Rys. 6. Rozkład temperatury powierzchni w funkcji promienia tarczy (od strony koła zamachowego)

The temperature distributions in relation to disc radii at the contact region between the pressure plate and the clutch disc at different times are shown in **Figure 7**. This figure has the same behaviours as shown in **Figure 6**, but the range of the values of temperature are higher than those on the flywheel side. These results are because of the lower thermal capacity of the pressure plate compared with the flywheel.

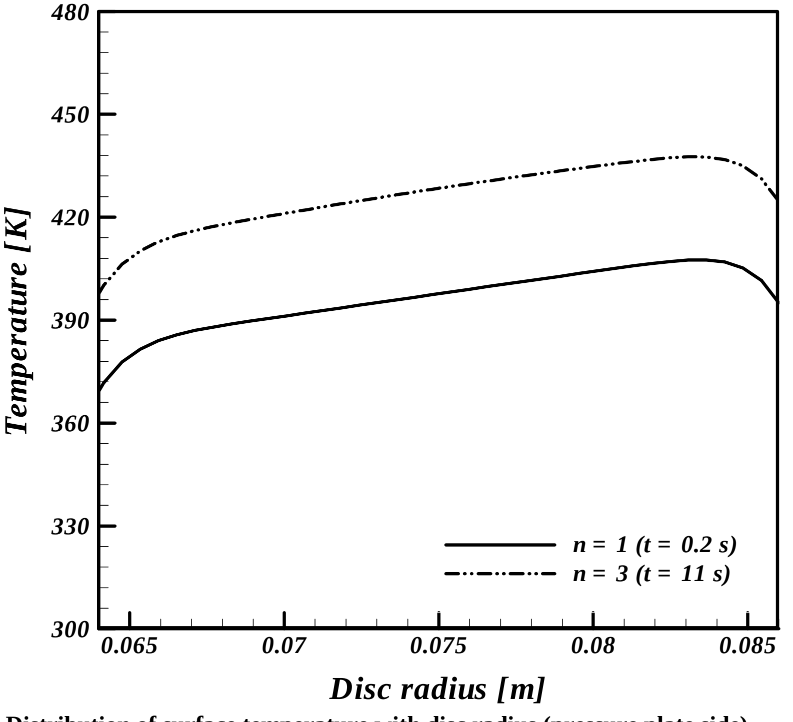

Fig. 7. Distribution of surface temperature with disc radius (pressure plate side)
Rys. 7. Rozkład temperatury powierzchni w funkcji promienia tarczy (od strony płyty dociskowej)

The temperature distributions of clutch elements during repeated engagements are shown in **Figure 8**. The maximum temperatures are located near the outer disc radius (r_o), and the minimum one occurs near the inner disc radius during all engagements. The temperature decreases in regions that are located at inner and outer disc radii of the clutch disc because of the effect of convection on these regions.

The variation of the maximum temperature during 4 engagements at the contact area between the pressure plate and clutch disc from one side and clutch disc and flywheel from the other side is shown in **Figures 9** and **10**. During all engagements, it can be observed that the values of the temperature increase with the number of engagements. The maximum temperatures reached after 4 engagements are found to be 421.7 K and 444 K, corresponding to the interface between the flywheel and clutch disc and the interface between the pressure plate and clutch disc, respectively.

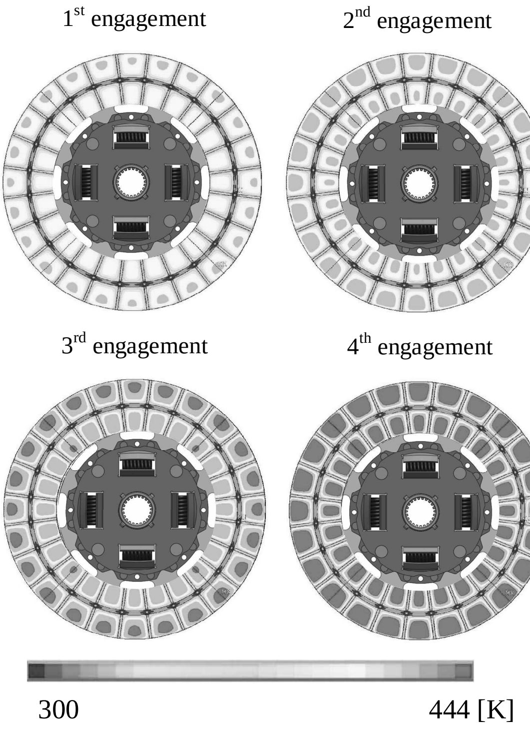

Fig. 8. Temperature distribution of friction clutch disc during four engagements
Rys. 8. Rozkład temperatury tarcia tarczy sprzęgła podczas czterech załączeń

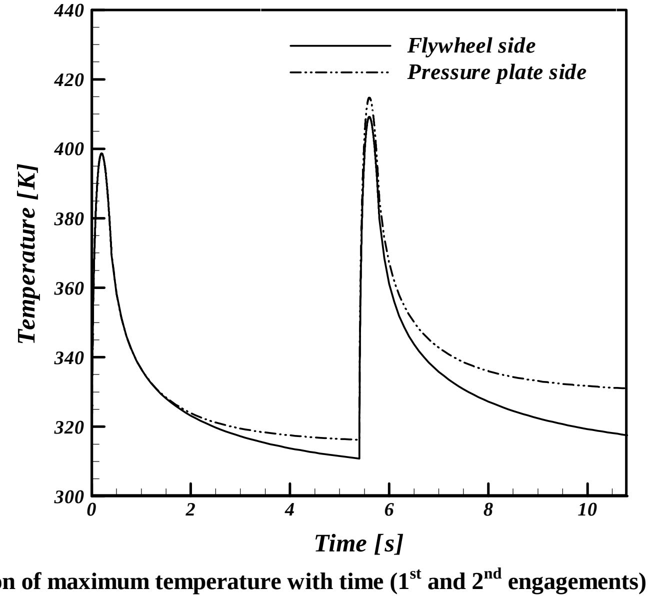

Fig. 9. Variation of maximum temperature with time (1^{st} and 2^{nd} engagements)
Rys. 9. Zmienność temperatury maksymalnej w funkcji czasu (1. i 2. załączenie)

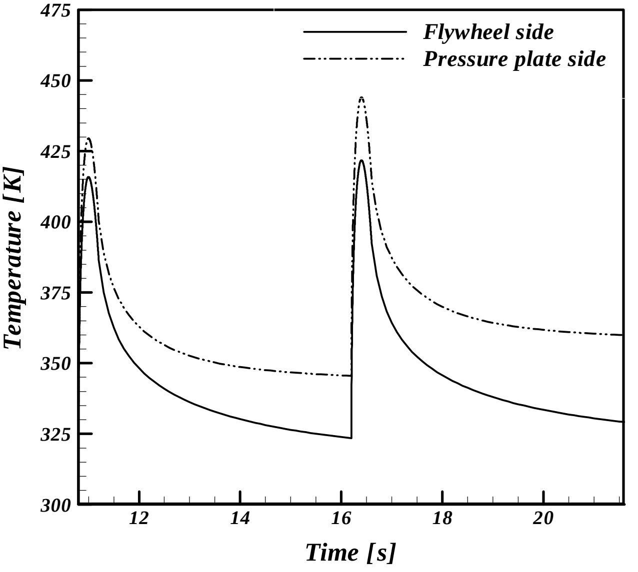

Fig. 10. Variation of maximum temperature with time (3rd and 4th engagements)
Rys. 10. Zmienność temperatury maksymalnej w funkcji czasu (3. i 4. załączenie)

CONCLUSIONS AND REMARKS

In this paper, the transient thermal analysis of a dry friction clutch system (friction faces have radial and circumferential grooves) during 4 repeated engagements based on a uniform pressure assumption was performed. A three-dimensional model was built to obtain the numerical simulation of friction clutch elements during the slipping period.

The surface temperature will increase rapidly when the number of engagements increases and, in some cases, the temperature exceeds the maximum limit of the working temperature of the sliding system, and this situation may lead to friction clutch failure before the expected lifetime. Therefore, the study of the temperature field of contact surfaces during repeated engagements operation is necessary to give an indication about the maximum temperature under this condition. The maximum temperature of a friction clutch occurs approximately in the middle of slipping time for all engagements.

The maximum temperatures that occur on the pressure plate and flywheel are approximately equal at the first engagement but, in the subsequent engagements, the maximum temperature of the pressure plate will be higher than the maximum temperature of flywheel and the difference between them will increase with increases of engagement's number. The reason for these differences in temperatures is the low thermal capacity of the pressure plate compared with the flywheel. To reduce the temperatures range of the pressure plate, the thickness of pressure plate needs to be increased, and this solution is considered very expensive from point of view of production. Increases in the quantity of heat transfer by convection is consider the other solution to reduce the temperature of a friction clutch system; therefore, it is important to select a suitable design for the clutch to increase the exposed area of the clutch (e.g., grooves) to increase the heat transfer to the environment.

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Streszczenie

Większość uszkodzeń systemów ślizgowych jest spowodowana wysokimi naprężeniami termicznymi generowanymi na powierzchniach styku, takich elementów jak części sprzęgieł i hamulców. W tej pracy zbadano zachowania cieplne jednotarczowego sprzęgła ciernego. Temperatury powierzchni tarczy sprzęgła wzrastają podczas powtarzanych załączeń, co w niektórych przypadkach prowadzi do przedwczesnych uszkodzeń tarcz sprzęgłowych. W celu uniknięcia tego rodzaju uszkodzeń temperatura powierzchni powinna być obliczona z wysoką dokładnością dla wyznaczenia maksymalnej temperatury pracy systemu tarciowego. W tej pracy wyznaczono rozkłady tempratury podczas czterech powtórzonych załączeń dla regularnych interwałów (5 s) przy identycznym rozpraszaniu energii. Do symulacji typowego sprzęgła ciernego zastosowano trójwymiarowe modele.