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THE EFFECT OF CONSTRAINT CONDITIONS ON THE CONTACT PRESSURE DISTRIBUTION OF MULTI-DISC FRICTION CLUTCHES

WPŁYW WARUNKÓW UTWIERDZENIA NA ROZKŁAD NACISKÓW STYKOWYCH W WIELOTARCZOWYCH SPRZĘGŁACH CIERNYCH

Key words:

dry friction, Multi-disc clutch, FEM, Contact analysis

Słowa kluczowe:

tarcie suche, MES, analiza styku

Summary

High thermal stresses generated between the contact surfaces in a clutch system, e.g., multi-disc friction clutch (back plate, clutch discs, separators, and piston), due to the friction heat generated during the slipping period are considered to be one of the main reasons for friction clutch failure. The finite element method has been used to study the contact pressure distribution of a multi-disc clutch (working in dry conditions) when all parts are closing together and the clutch

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system starts to engage. The augmented Lagrange algorithm is used to obtain the contact pressure distribution of the friction surfaces under different restriction conditions. Analysis has been completed using the developed axisymmetric model that is used to simulate the multi-disc clutch system.

INTRODUCTION

Friction clutches are considered to be the most important and common type used in automotive applications. 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. Multi-disc clutches are used for high torque capacity, but for this type, the heat transfer will be limited, and it is too difficult to cool it. Therefore, it is appropriate for high-load and low-speed applications. For high-speed dynamic loads, it is better to use fewer friction surfaces **[L. 1]**. **Figure 1** shows the main parts of the typical multi-disc friction clutch system.

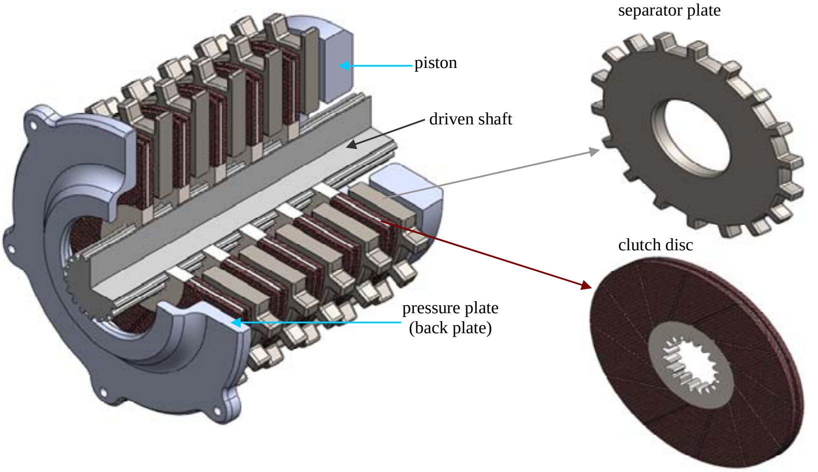

Fig. 1. Main parts of a multi-disc clutch system
Rys. 1. Główne elementy wielotarczowego systemu sprzęgłowego

The analysis of the contact problem is considered an essential issue for mechanical designers in different fields of engineering. The contact phenomenon occurs in most machines due to using the contact to transmit the forces between parts in any mechanical system. The length and time scale of the contact phenomenon cover a wide range, from nanoscale to the macroscopic level, and from hypervelocity impact to quasi-static contact interaction. Generally, the machines consist of many parts and joints, such as pins and bolts that are used to connect them. Therefore, one should deeply investigate the effect of the variables and parameters on the contact problem in these parts. The contact mechanics is used to develop the production of machines in different fields, such as automotive clutches and brakes, tires of cars, bearings, splines, gears, seals, etc. The mechanical designing process mainly depends on the understanding of the contact phenomenon, with the aim of increasing the lifecycle and decreasing the wear rate of contact bodies.

Al-Shabibi and Barber **[L. 2]** investigated the thermomechanical behaviour of the sliding systems using a reduced order model approximation, which is described by one or more dominant perturbation or eigenfunctions. They built a mathematical model of the sliding system with a modest number of degrees of freedom to obtain the temperature field and the contact pressure distribution. They showed that the reduced order models have very good approximations during the first part of the automotive brake or clutch engagement, when the sliding speed is higher than the critical sliding speed. At the last period of the engagement when the sliding speed is sub-critical, the eigenvalues become clustered and substantially more terms are needed to obtain accurate solutions. Later on, Al-Shabibi and Barber **[L. 3]** studied the transient non-homogeneous thermoelastic contact problem including the frictional heating. They divided the problem into two parts: the homogenous and the particular systems. The solutions of both systems are superposed to provide the degree of freedom needed to satisfy the initial and boundary conditions of the problem. Their approximation showed that the solution could be applied from low to medium range of the operating sliding speed of clutch. When the operating sliding speed is high, the results showed negative values of the contact pressure, which indicates a separation. Furthermore, they investigated the effect of varying the sliding speed on the thermoelastic behaviour of a clutch. The results of their approximation were validated against the direct finite element simulation.

Shahzamanian et al. **[L. 4]** investigated the thermoelastic contact problem of functionally graded (*FG*) rotating brake disk with a heat source due to the contact friction using the finite element method. In their work, the material properties of disk are assumed to be represented by power-law distributions in the radial direction. The inner and outer surfaces are considered metal and ceramic, respectively. Pure material is considered for the brake pad. Coulomb contact friction is assumed as the heat source. The results showed that the maximum value of radial displacement in mounted *FG* brake disk is not at the outer surface. They found that all areas between pad and brake disk is in full-contact status when the ratio of pad thickness to the brake disk thickness is larger than 0.66.

In this paper, the finite element model has been developed to compute the contact pressure distribution of a multi-disc clutches (back plate, clutch discs, separators and piston) when all parts of the clutch system closing together and clutch starts to engage. This work presents the effect of restriction conditions on the contact pressure distribution of multi-disc clutches.

FINITE ELEMENT ANALYSIS

This section presented the steps to simulate the contact elements of a friction clutch using ANSYS software. The first step in this analysis is the modelling; due to the symmetry in the geometry (frictional facing without grooves) and boundary conditions of the friction clutch (take into the consideration the effect of the applied pressure), an axisymmetric finite element model was developed **[L. 5]** to represent the elastic problem in multi-disc clutches. The elastic model of a multi-disc clutch system with boundary conditions is shown in **Figure 2**.

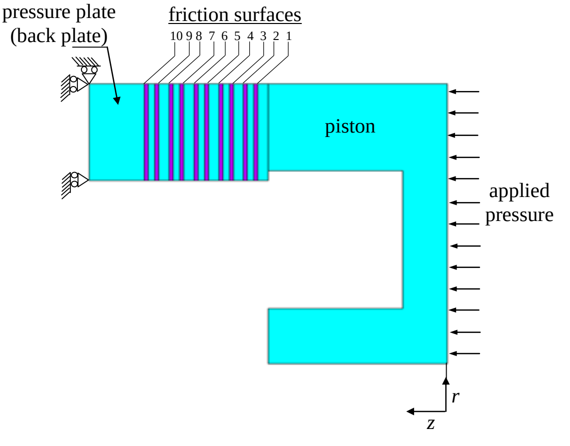

Fig. 2. Contact model with boundary conditions of a multi-disc clutch system
Rys. 2. Model styku dla wielotarczowego systemu sprzęgłowego z warunkami brzegowymi

Three basic types of contact are used in general mechanical applications. They are single contact, node-to-surface contact, and surface-to-surface contact. Surface-to-surface contact is the most common type of contact used for bodies that have arbitrary shapes with relative large contact areas. This type of contact is the most efficient for bodies that have large values of relative sliding, such as a block sliding on plane or a sphere sliding within a groove **[L. 6]**. Surface-to-surface contact is the type of contact assumed in this work. Five algorithms are used for surface-to-surface contact, and they are as follows:

(a) Penalty method: This algorithm used a constant „spring" to establish the relationship between the two contact surfaces as shown in **Figure 3**. The contact force (pressure) between two contact bodies can be written as follows **[L. 6]**:

$$F_n = k_n x_p \tag{1}$$

where F_n is the contact force, k_n is the contact stiffness, and x_p is the distance between two existing nodes or separate contact bodies (penetration or gap).

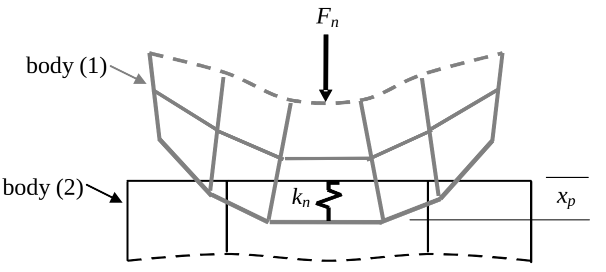

Fig. 3. Contact stiffness between two contacting bodies
Rys. 3. Sztywność stykowa dla dwu kontaktujących się ciał

(b) Augmented Lagrange (default): This algorithm is an iterative penalty method. The constant traction (pressure and frictional stresses) are augmented during equilibrium iterations so that the final penetration is lower than the allowable tolerance that is assumed in the numerical solution. This method usually leads to better conditioning, and it is less sensitive to the magnitude of the constant stiffness. The contact force (pressure) between two contact bodies is

$$F_n = k_n x_p + \lambda_L \tag{2}$$

where λ_L is the Lagrange multiplier component.

(c) Lagrange multiplier on normal contact and penalty on tangential contact: This method is applied on the constant normal and penalty method (tangential contact stiffness) on the frictional plane. This method enforces zero penetration and allows a small amount of slip for the sticking contact condition. It requires chattering control parameters, as well as the maximum allowable elastic slip parameter.

(d) Pure Lagrange multiplier on normal contact and tangential contact: This method enforces zero penetration when contact is closed and “zero slip” when sticking contact occurs. This algorithm does not require contact stiffness. Instead, it requires chattering control parameters. This method adds contact traction to the model as additional degrees of freedom and requires additional iterations to the stabilize contact conditions. It often increases the computational cost compared to the augmented Lagrangian method.

(e) Internal multipoint constraint: This method is used in conjunction with bonded contact, and there is no separation contact to model several types of contact assemblies and kinematic constraints.

The stiffness relationship between the contact and the target surfaces will decide the amount of the penetration. Higher values of contact stiffness will decrease the amount of penetration, but they can lead to the ill conditioning of the global stiffness matrix and convergence difficulties. Lower values of contact stiffness can lead to certain amount of penetration, and then they can decrease the difficulties in the convergence of the solution.

The contact stiffness of an element of area A_e is calculated using the following formula **[L. 7]**:

$$FKN_i = \int \{N_i\} (e) \{N_i\}^T \, dA_e \tag{3}$$

where N_i is the shape function and e is the elastic restraining stiffness, and it's dependent on the material properties. In ANSYS software, the default value of the contact stiffness factor *FKN* is 1, which is appropriate for bulk deformation. If bending deformation dominates the solution, a smaller value of *FKN* (*FKN* = 0.1) is recommended **[L. 6]**. Augmented Lagrange is the algorithm that is used to find the solution of the contact problem in multi-disc clutches in this paper.

In all computations of the friction clutch model, a homogeneous and isotropic material has been assumed, anall parameters and materials properties are listed in **Table 1**, and **Figure 4** shows the axisymmetric finite element model of multi-disc clutch.

Table 1. Properties of materials and operations parameters
Tabela 1. Własności materiałów i parametry operacyjne

Parameters	Values
Inner disc radius of friction clutch disc, r_i [m]	0.052
Outer disc radius of friction clutch disc, r_o [m]	0.067
Thickness of clutch disc including the friction facing, t_t [m]	0.00193
Thickness of friction facing, t_c [m]	0.00053
Inner disc radius of pressure plate, r_{ip} [m]	0.052
Outer disc radius of pressure plate, r_{op} [m]	0.067
Thickness of pressure plate, t_p [m]	0.0074
Inner radii of piston, r_{if1} and r_{if2} [m]	0.0235 0.0535
Outer radii of piston, r_{of1} and r_{of2} [m]	0.032 0.067
Thicknesses of piston, t_{f1} and t_{f2} [m]	0.006 0.024
Applied pressure, p_a [MPa]	1
Coefficient of friction, μ [1]	0.2
Number of friction clutch disc, n_p [1]	5
Young's modulus of friction material, E_c [GPa]	0.3
Young's modulus of pressure plate, plate separator, piston and clutch plate. E_p, E_s, E_f, and E_{cp} [GPa]	200
Poisson's ratio of friction material, pressure plate, plate separator, piston and clutch plate. ν_c, ν_p, ν_s, ν_f, and ν_{cp}	0.25

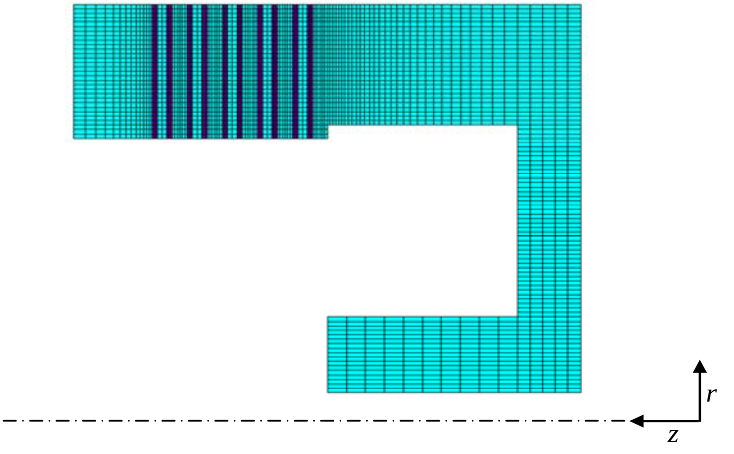

Fig. 4. Finite element model of a multi-disc clutch (no. of elements = 4590)
Rys. 4. Modelowanie elementów skończonych dla sprzęgła wielotarczowego (liczba elementów = 4590)

RESULTS AND DISCUSSIONS

The effects of boundary conditions (restricting) on the contact pressure distribution of the contact surfaces of a multi-disc clutch were investigated. Four different cases of boundary conditions are assumed in this paper. The boundary condition of the first case of a multi-disc clutch is shown in **Figure 2**, the cases (2, 3 and 4) are restricted with (l = 10, 20 and 50%) of the backside of back plate (l_m) respectively, as shown in **Figure 5**.

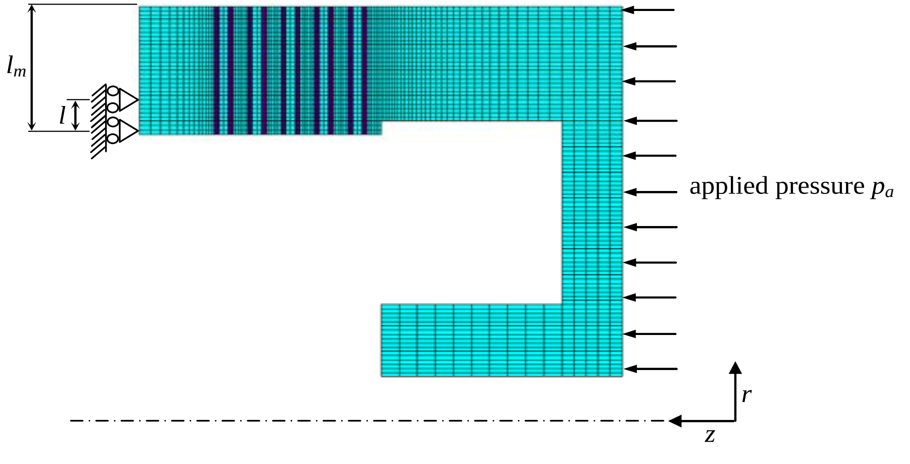

Fig. 5. Finite element model of a multi-disc clutch system (cases 2, 3 and 4)
Rys. 5. Modelowanie elementów skończonych dla wielotarczowego systemu sprzęgłowego (przypadki 2, 3 i 4)

Figures 6–8 show the normalized contact pressure distribution with disc radius of the selected contact surfaces (*f.s* = 1, 5 and 10) of a multi-disc clutch. It can be noticed that the maximum contact pressure occurs at the 10^{th} surfaces. Generally, it can be noticed that the contact pressure will be more uniform when using two points of support, and this uniformity of the contact pressure will decrease when decreasing the restriction of the back plate (or flywheel in a single-disc clutch).

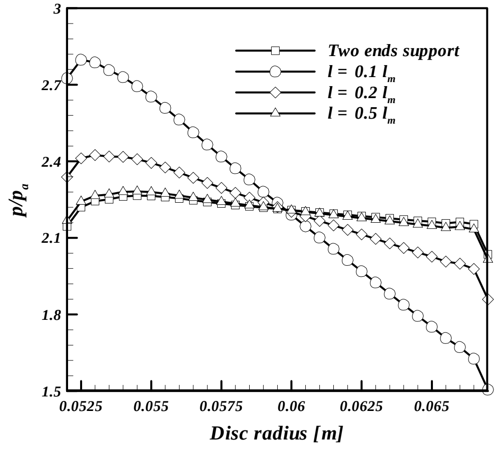

Fig. 6. Variation of the normalized contact pressure with disc radius applying different boundary conditions (f.s = 1)

Rys. 6. Zmienność unormowanej wartości nacisków stykowych w funkcji promienia tarczy dla różnych warunków brzegowych (f.s = 1)

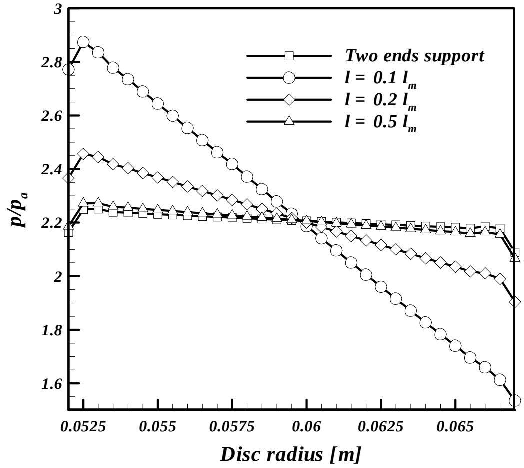

Fig. 7. Variation of the normalized contact pressure with disc radius applying different boundary conditions (f.s = 5)

Rys. 7. Zmienność unormowanej wartości nacisków stykowych w funkcji promienia tarczy dla różnych warunków brzegowych (f.s = 5)

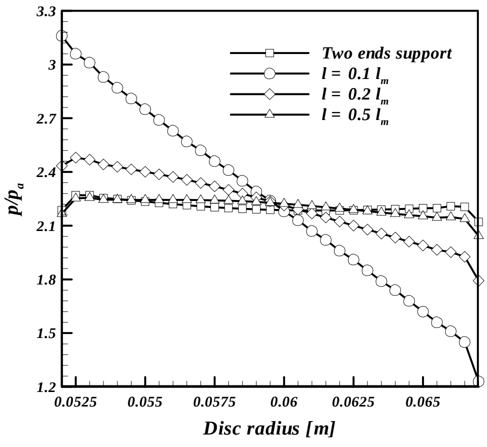

Fig. 8. Variation of the normalized contact pressure with disc radius applying different boundary conditions (f.s = 10)

Rys. 8. Zmienność unormowanej wartości nacisków stykowych w funkcji promienia tarczy dla różnych warunków brzegowych (f.s = 10)

It can be observed from these figures that the maximum contact pressure occurred near the inner disc radius and the minimum values of the contact pressure occurred near the outer disc radius for all contact surfaces. The percentage of increase in values of contact pressure of the second case (restricted 10% of the backside of back plate) when the boundary condition is applied, instead of the first case (two ends support) are found to be 25% and 44.9%, corresponding to the first friction surface and the tenth friction surface, respectively.

CONCLUSIONS AND REMARKS

In this research work, the contact problem of multi-disc friction clutches has been investigated using the finite element method. Axisymmetric models are developed to simulate the friction clutch system.

The boundary conditions (restricting condition) play important roles in specifying the magnitude and the distribution of the contact pressure. This effect will be more significant during the sliding period due to the friction heat generated between the contact surfaces.

The permanent deformations and thermal cracks on the contact surfaces of a clutch, if taken into consideration in the numerical model, will change the contact pressure distribution and the actual contact area will also change under these circumstances. These disadvantages will focus the contact pressure on a small region compared to the nominal contact area. In addition, the damaged or incorrectly machined back plate (such as a large deformation, thermal cracks, etc.) causes many problems. One of them focuses the contact pressure on small regions of the nominal frictional interface (e.g. bands and/or spots). Owing to this problem, it is essential when fitting a new clutch to a vehicle to ensure that the back plate is in perfect condition, in order to prevent any possible premature damage in the clutch system.

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Streszczenie

Wysokie naprężenia cieplne generowane na powierzchniach styku w systemach sprzęgających, np. dla wielotarczowego sprzęgła ciernego (płyta dociskowa, tarcze sprzęgłowe, separatory, tłok), spowodowane generowaniem ciepła tarcia w wyniku poślizgu, zostało uznane za podstawową przyczynę uszkodzeń sprzęgieł ciernych. Do zbadania rozkładu nacisków w sprzęgle wielotarczowym pracującym w warunkach tarcia suchego – podczas gdy jego elementy zbliżają się do siebie i następuje włączenie systemu sprzęgłowego – zastosowano metodę elementu skończonego. Rozszerzony algorytm Lagrange'a został użyty w celu wyznaczenia rozkładu nacisków kontaktowych na powierzchniach tarcia dla różnych warunków utwierdzenia. Analiza została przeprowadzona z wykorzystaniem opracowanego modelu osiowosymetrycznego dla symulacji wielotarczowego systemu sprzęgłowego.