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RESEARCH

Stresses and Deformations Analysis of a Dry Friction Clutch System

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A B S T R A C T

The friction clutch is considered the essential element in the torque transmission process. In this paper, the finite element method is used to study the stresses and deformations for clutch system (pressure plate, clutch disc and flywheel) due to the contact pressure of diaphragm spring and the centrifugal force during the full engagement of clutch disc (assuming no slipping between contact surfaces). The investigation covers the effect of the contact stiffness factor FKN on the pressure distribution between contact surfaces, stresses and deformations. The penalty and Augmented Lagrangian algorithms have been used to obtain the pressure distribution between contact surfaces. ANSYS13 software has been used to perform the numerical calculation in this paper.

1. INTRODUCTION

A clutch is a very important machine element which plays a main role in the transmission of power (and eventually motion) from one component (the driving part of the machine) to another (the driven part). A common and well known application for the clutch is in automotive vehicles where it is used to connect the engine and the gearbox. Furthermore, the clutch is used also extensively in production machinery of all types. When the friction clutch begins to engage, slipping occurs between the contact surfaces (pressure plate, clutch disc and flywheel) and due to this slipping, heat energy will be generated in the interfaces friction surfaces. At high relative sliding velocity, high quantity of frictional heat is generated which lead to high temperature rise on the clutch disc surfaces and hence thermo-mechanical problems such as thermal deformations and thermo-elastic instability can occur. This in turn, can lead to thermal cracking and high rate of wear. The pressure distribution is essential factor effect on the performance of the friction clutch because of the heat generated between contact surfaces during the slipping period dependent on the pressure distribution.

Al-Shabibi and Barber [1] used the finite element method to find the transient solution of the temperature field and contact pressure distribution between two sliding disks. Two dimensional axisymmetric FE model used to explore an alternative method based on an eigenfunction expansion and a particular solution that can be used to solve the thermoelastic contact problem with frictional heating. Both constant and varying sliding speed is considered in this analysis. Results of the direct finite element simulation have been obtained using the commercial package ABAQUS. The results from the approximate solution show a good agreement with the results from the direct finite element simulation.

Lee et al. [2] used finite element method to study the effect of thermo-mechanical loads on the pressure plate and the hub plate of the friction clutch system. Three types of loads are taking into consideration the thermal load due to the slipping occurs at the beginning of engagement, the contact pressure of diaphragm spring and the centrifugal force due to the rotation. Two and three dimensional finite element models were performed to obtain the temperature distributions and the stresses. The results show the significant effect of the thermal load on the temperatures and stresses; therefore it is desirable to increase the thickness of the pressure plate as much as possible to increase the thermal capacity of the pressure plate to reduce the thermal stresses. High stress intensity value occurs around the fillet region of the window in the hub plate.

Shahzamanian et al. [3] used numerical simulation to study the transient and contact analysis of functionally graded (FG) brake disk. The material properties vary in the radial direction from full-metal at the inner radius to that of full-ceramic at the outer radius. The coulomb contact friction is considered between the pad and the brake disk.

Two-dimensional finite element model used in the work to obtains the pressure distribution, total stresses, pad penetration, friction stresses, heat flux and temperature during the contact for different values of the contact stiffness factor. It was found, that the contact pressure and contact total stress increase when the contact stiffness factor increases and the gradation of the metal–ceramic has significant effect on the thermo-mechanical response of FG brake disks. Also, it can be concluded when the thickness of the pad increases the contact status between pad and disc changes from sticking to contact and then to near contact.

Abdullah and Schlattmann [4-8] investigated the temperature field and the energy dissipated of dry friction clutch during a single and repeated engagement under uniform pressure and uniform wear conditions. They also studied the effect of pressure between contact surface when varying with time on the temperature field and the internal energy of clutch disc using two approaches heat partition ratio approach to compute the heat generated for each part individually whereas the second applies the total heat generated for the whole model using contact model. Furthermore, they studied the effect of engagement time and sliding velocity function, thermal load and dimensionless disc radius (inner disc radius/outer disc radius) on the thermal behavior of the friction clutch in the beginning of engagement.

In this paper the finite element method used to study the contact pressure and stresses during the full engagement period of the clutches using different contact algorithms. Moreover, sensitivity study for the contact pressure is presented to indicate the importance of the contact stiffness between contact surfaces.

2. FUNDAMENTAL PRINCIPLES

The main system of the friction clutch consists of pressure plate, clutch disc and flywheel as shown in Fig. 1.

When the clutch starts to engage the slipping will occur between contact surfaces due to the difference in the velocities between them (slipping period), after this period all contacts parts are rotating at the same velocity without slipping (full engagement period). A high amount of the kinetic energy converted into heat energy at interfaces according to the first law of thermodynamics during the slipping period and the heat generated between contact surfaces will be dissipated by conduction between friction clutch components and by convection to environment, in addition to the thermal effect due to the slipping there is other load condition which is the pressure contact between contact surfaces. In the second period, there are three types of load conditions the temperature distribution from the last period (slipping period), the pressure between contact surfaces due to the axial force of diaphragm spring and the centrifugal force due to the rotation of the contacts parts. Figure 2 shows the load conditions during the engagement cycle of the clutch, where t_s is the slipping time and T is the transmitted torque by clutch.

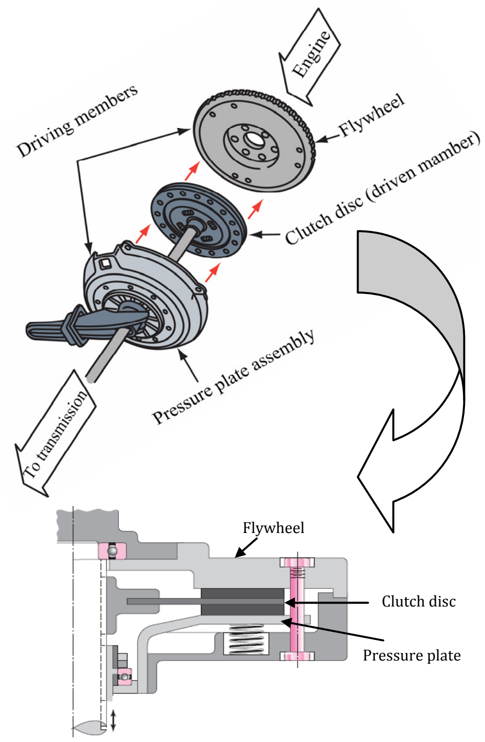

Fig. 1. The main parts of clutch system.

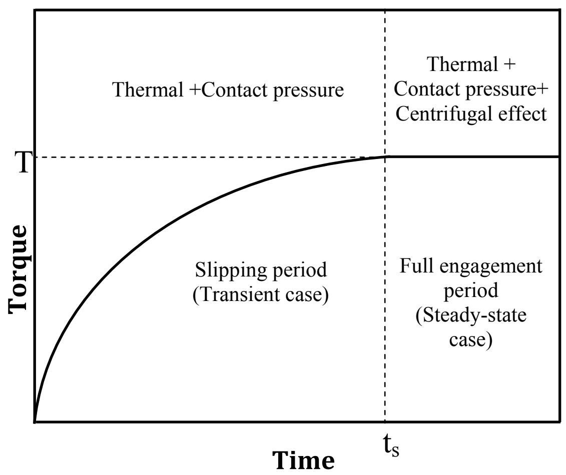

Fig. 2. The load conditions during the engagement cycle of the clutch

When the clutch starts to engage the slipping will occur between contact surfaces due to the difference in the velocities between them (slipping period), after this period all contacts parts are rotating at the same velocity without slipping (full engagement period). A high amount of the kinetic energy converted into heat energy at interfaces according to the first law of thermodynamics during the slipping period and the heat generated between contact surfaces will be dissipated by conduction between friction clutch components and by convection to environment, in addition to the thermal effect due to the slipping there is other load condition which is the pressure contact between contact surfaces. In the second period, there are three types of load conditions the temperature distribution from the last period (slipping period), the pressure between contact surfaces due to the axial force of diaphragm spring and the centrifugal force due to the rotation of the contacts parts. Figure 2 shows the load conditions during the engagement cycle of the clutch, where t_s is the slipping time and T is the transmitted torque by clutch.

3. FINITE ELEMENT FORMULATION

This section presented the steps to simulate the contact elements of friction clutch using ANSYS software. Moreover it gives more details about the types of contacts and algorithms which are used in this software.

The first step in this analysis is the modelling; due to the symmetry in the geometry (frictional lining without grooves) and boundary conditions of the friction clutch (take into the consideration the effect of the pressure and centrifugal force loads, and neglected the effect of thermal load due to the slipping), two-dimensional axisymmetric FEM can be used to represent the contact between the clutch elements during the steady-state period as shown in Fig. 3.

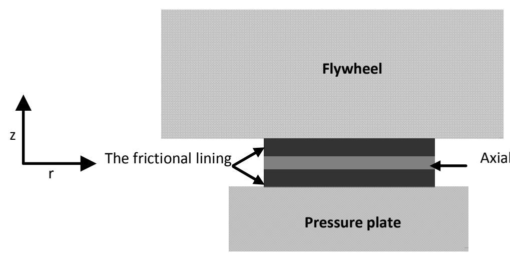

Fig. 3. The Contact model for clutch system.

There are three basic types of contact used in Ansys software single contact, node-to-surface contact and surface-to-surface contact. Surface-to-surface contact is the most commonly type of contact used for bodies that have arbitrary shapes with relative large contact areas. This type of contact is most efficient for bodies that experience large values of relative sliding such as block sliding on plane or sphere sliding within groove [9]. Surface-to-surface contact is the type of contact assumed in this analysis because of the large areas of clutch elements in contact.

In this work, it has been assumed two types of load conditions effects on the clutch system during the steady-state period (full engagement period) the contact pressure between clutch elements due to the axial force by diaphragm spring and the centrifugal force due to the rotation.

The elements used for contact model are:

- "Plan13" used for all elements of the clutch (flywheel, clutch disc and pressure plate).
- "Conta172" used for contact surfaces that are the upper and lower surfaces of clutch disc.
- "Targe169" used for the target surfaces that are the lower surface of the flywheel and the upper surface of the pressure plate.

Figure 4 shows the details about schematic for all elements that has been used in this analysis.

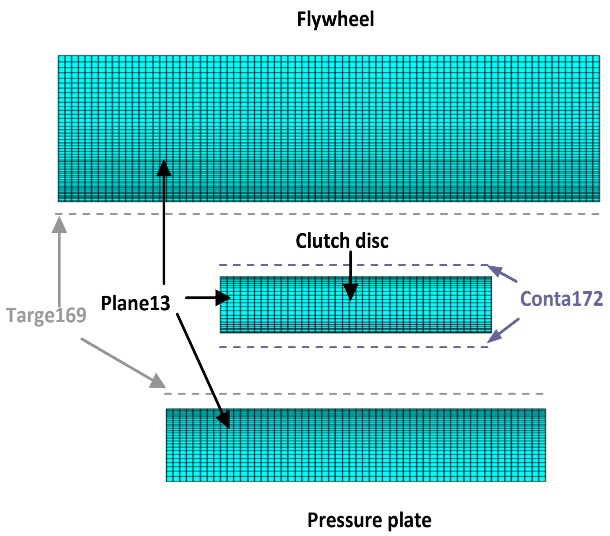

Fig. 4. Schematic elements used for the friction clutch elements.

The stiffness relationship between contact and target surfaces will decide the amount of the penetration. Higher values of contact stiffness will decrease the amount of penetration, but can lead to ill-conditioning of the global stiffness matrix and convergence difficulties. Lower values of contact stiffness can lead to certain amount of penetration and low enough to facilitate convergence of the solution. The contact stiffness for an element of area A is calculated using the following formula [10]:

$$F_{kn} = \int \{f_i\} (e) \{f_i\}^T \, dA \tag{1}$$

The default value of the contact stiffness factor FKN is 1, and it is appropriate for bulk deformation. If bending deformation dominates the solution, a smaller value of KKN = 0.1 is recommended.

There are five algorithms used for surface-to-surface contact type are:

- Penalty method: this algorithm used constant "spring" to establish the relationship between the two contact surfaces (Fig. 5). The contact force (pressure) between two contact bodies can be written as follows:

$$F_n = k_n \, x_p \tag{2}$$

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).

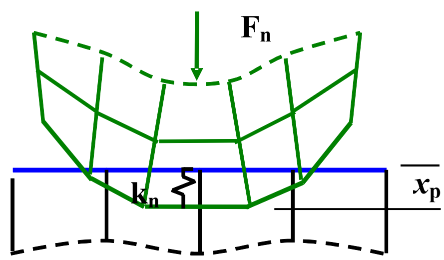

Fig. 5. The contact stiffness between two contact bodies.

- 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 small than the allowable tolerance. This method usually leads to better conditioning and is less sensitive to the magnitude of the constant stiffness. The contact force (pressure) between two contact bodies is:

$$F_n = k_n \, x_p + \lambda \tag{3}$$

where λ is the Lagrange multiplier component.

- Lagrange multiplier on contact normal and penalty on tangent: this method applied on the constant normal and penalty method (tangential contact stiffness) on the frictional plane. This method enforces zero penetration and allows small amount of slip for the sticking contact condition. It requires chattering control parameters, as well as the maximum allowable elastic slip parameter.
- Pure Lagrange multiplier on contact normal and tangent: 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 increase the computational cost compared to the augmented lagrangian method.
- Internal multipoint constraint: this method used in conjunction with bonded contact and no separation contact to model several types of contact assemblies and kinematic constraints.

The axisymmetric finite element model of the friction clutch system with boundary conditions is shown in Fig. 6. A mesh sensitivity study was done to choose the optimum mesh from computational accuracy point of view.

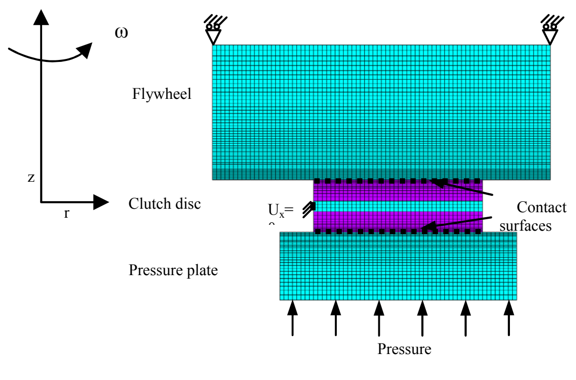

Fig. 6. FE models with the boundary conditions.

The full Newton-Raphson with unsymmetric matrices of elements is used in this analysis assuming a large-deflection effect. In all computations for the friction clutch model, it has been assumed a homogeneous and isotropic material and all parameters and materials properties are listed in Table. 1.

In this analysis also assuming there are no cracks in the contact surfaces and the actual contact area is equal to the nominal contact area.

Table 1. The properties of materials and operations.

Parameters	Values
Inner radius of friction material & axial cushion, r_i [m]	0.06298
Outer radius of friction material & axial cushion, r_o [m]	0.08721
Thickness of friction material [m], t_l	0.003
Thickness of the axial cushion [m], $t_{axi.}$	0.0015
Inner radius of pressure plate [m], r_{ip}	0.05814
Outer radius of pressure plate [m], r_{op}	0.09205
Thickness of the pressure plate [m], t_p	0.00969
Inner radius of flywheel [m], r_{if}	0.04845
Outer radius of flywheel [m], r_{of}	0.0969
Thickness of the flywheel [m], t_f	0.01938
pressure, p [MPa]	1
Coefficient of friction, μ	0.2
Number of friction surfaces, n	2
Torque [Nm], T	432
Maximum angular slipping speed, ω_o [rad/sec]	200
Young’s modulus for friction material, E_l [GPa]	0.30
Young’s modulus for pressure plate, flywheel & axial cushion, (E_p, E_f, and E_{axi}), [Gpa]	125
Poisson’s ratio for friction material,	0.25
Poisson’s ratio for pressure plate, flywheel & axial cushion	0.25
Density for friction material, (kg/m^3), ρ_l	2000
Density for pressure plate, flywheel & axial cushion, (kg/m^3), (ρ_p, ρ_f, and ρ_{axi})	7800

4. RESULTS AND DISCUSSIONS

Series of computations have been carried out using ANSYS13 software to study the contact pressure and stresses between contact surfaces of clutch (pressure plate, clutch disc and flywheel) during a full engagement period using different algorithms and contact stiffness factor values.

The variation of the contact pressure with disc radius for both sides of clutch disc (flywheel side and pressure plate side) using penalty and augmented algorithms (FKN = 1) is shown in Figs. 7 and 8. From these figures, it can be seen that the identical results when using penalty and augmented (default) methods and approximately the same behaviour of contact pressure for both sides of clutch disc. The maximum contact pressure values in the flywheel side and pressure plate side are found to be 1.491 MPa and 1.524 MPa, respectively. The maximum and minimum contact pressure values occur at outer disc radius r_o and near inner radius (1.01 r_i) for both cases, respectively.

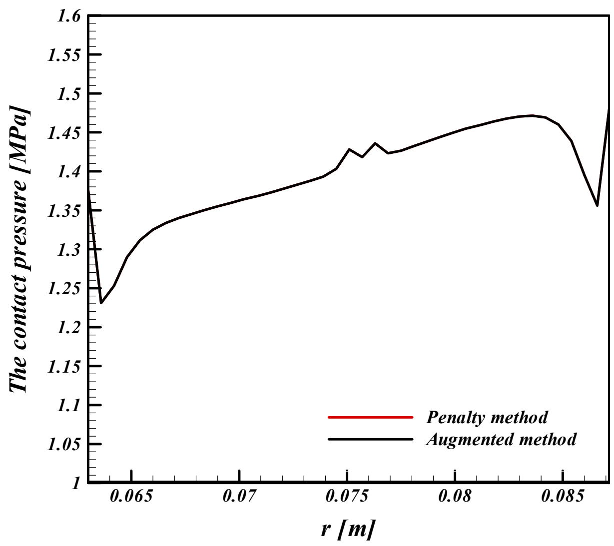

Fig. 7. The variation of contact pressure with disc radius (flywheel / clutch disc).

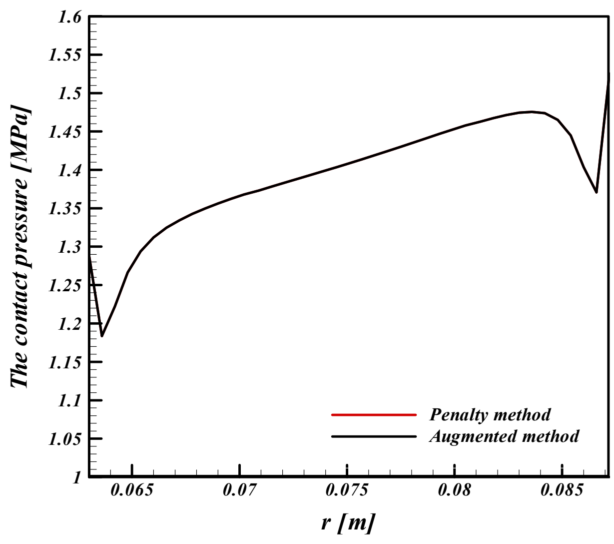

Fig. 8. The variation of contact pressure with disc radius (pressure plate / clutch disc).

Figures 9 and 10 show the variation of total contact stresses with disc radius for both sides of clutch disc. It can be seen, that the total contact stresses have the same behaviour of the contact pressure.

Figures 11 and 12 demonstrate the variation of total displacement of clutch surfaces with disc radius. It's clear the values of total deformations of clutch disc (pressure plate side) are higher than the displacements values at the flywheel side. The maximum values of total deformation in the clutch disc at flywheel and pressure plate sides are found to be 4.6529 E^{-6} m and 2.84 E^{-5} m, respectively.

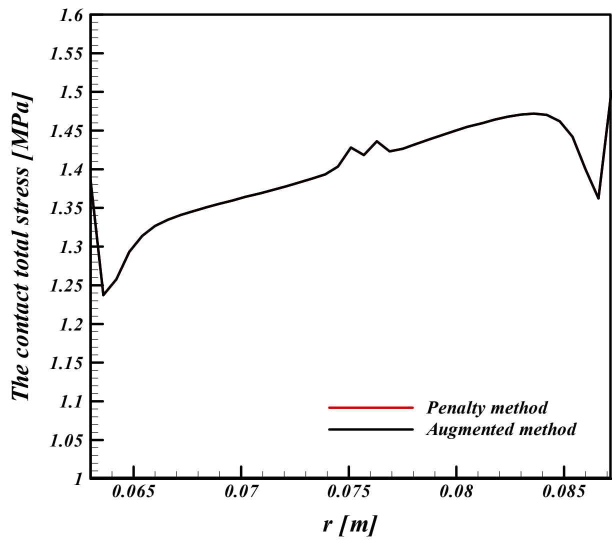

Fig. 9. The variation of total contact stress with disc radius (flywheel / clutch disc).

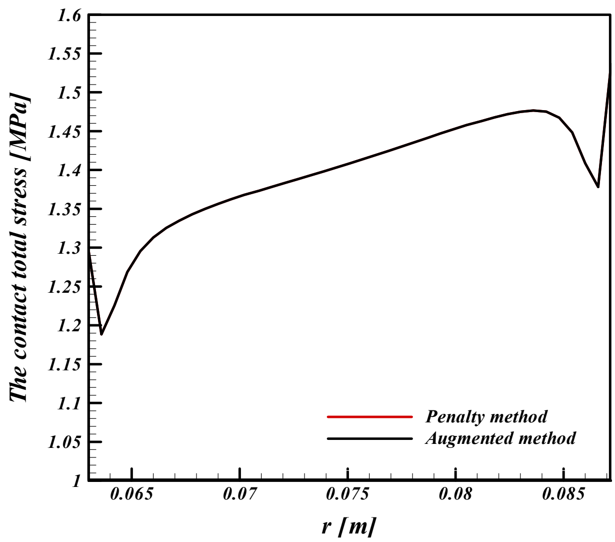

Fig. 10. The variation of total contact stress with disc radius (pressure plate / clutch disc).

The variation of the contact pressure for using different algorithms and different values of FKN along the radial direction at contact area of clutch disc with flywheel is shown in Figs. 13 and 14. It can be noted for both cases (when using penalty and augmented method), that the values of contact pressure increases when FKN increases. The percentage increasing in contact pressure when FKN change from 0.01 to 10 is found to be 19.5 % and 17.9 % corresponding to penalty and augmented methods, respectively.

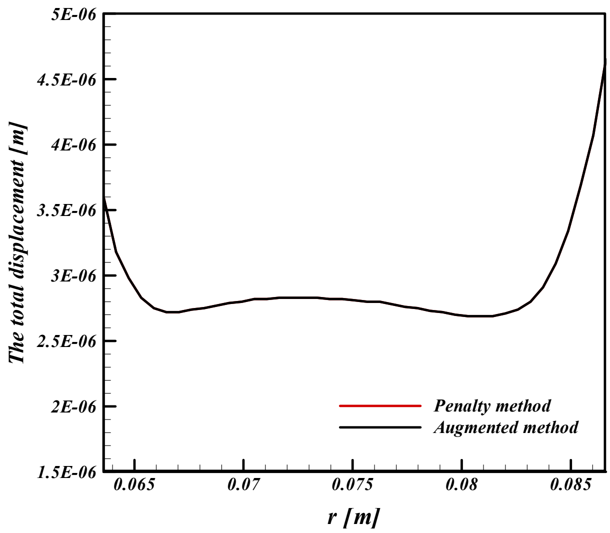

Fig. 11. The variation of total displacement with disc radius (flywheel/clutch disc).

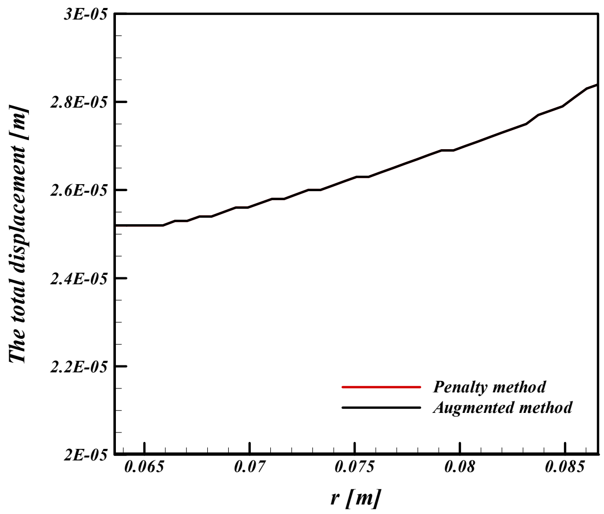

Fig. 12. The variation of total displacement with disc radius (pressure plate/clutch disc).

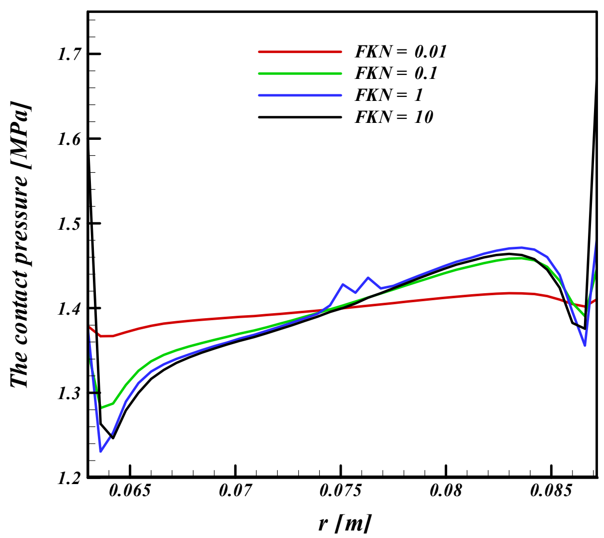

Fig. 13. The variation of contact pressure with disc radius using Penalty method (flywheel/clutch disc).

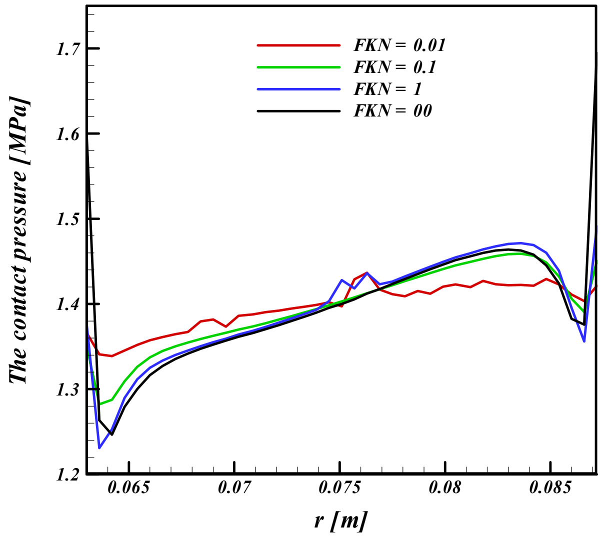

Fig. 14. The variation of contact pressure with disc radius using augmented Lagrange algorithm (flywheel/clutch disc).

5. CONCLUSIONS AND REMARKS

The variations of the contact pressure, total contact stress and total displacements of the friction clutch using different contact algorithms and different values of FKN are investigated. Two-dimensional axisymmetric finite element model for the contact elements of clutch were conducted to obtain the numerical results.

The present work presents a simplified model of clutch to determine the contact pressure between contact surfaces during a full engagement period.

The conclusions obtained from the present analysis are summarized as follows:

1. The value of FKN is very important and effective on the values of contact pressure, the contact pressure is directly proportional to FKN for both contact methods (penalty and augmented).
2. The penalty method has sensitivity for FKN more than the augmented method.
3. The maximum and minimum values of contact pressure and total contact stress occur at outer disc radius and inner disc radius, respectively.

The permanent deformations and thermal cracks on the contact surfaces of clutch if taken into consideration will affect the contact pressure distribution and the actual contact area will change. These disadvantages will focus the contact pressure on small region compared with the nominal contact area.

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