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# The temperatures distributions of a single-disc clutches using heat partitioning and total heat generated approaches

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A R T I C L E I N F O

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A B S T R A C T

An accurate estimation of temperature distribution is considered necessary to avoid the premature failure of friction clutches. In this work, different approaches were used to compute the surface temperatures of the friction clutch disc. The results presented the maximum surface temperature when the contact occurs between the rubbing surfaces during a single engagement and repeated engagements. Two approaches were used to simulate the thermal models of the automotive clutches to obtain the temperature field are heat partitioning approach and total heat generated approach. The analysis was conducted using developed axisymmetric finite element models to study the thermal behavior of the friction clutches during multi-engagements. The comparison was made between the temperature distributions based on the proposed approaches to show the accuracy of each approach. It was found that the heat partitioning approach was not accurate to investigate the thermal problem of the friction clutch during the multi-engagements.

## 1. Introduction

Friction clutches are repeatedly subjected to a considerable amount of frictional heat generation on the contact surfaces. This kind of heat generation happened due to the existing slipping between contacting parts of the friction clutch (in a single-disc clutch e.g. pressure plate, clutch disc and flywheel). High surface temperatures appeared as a result of the frictional heat generated when the friction clutch started to work. In some case the maximum temperatures exceed the allowable working temperature; consequently, many drawbacks appear on the surfaces of contacting parts such as plastic deformations, fast wear and surface cracks. When the friction clutch works under these circumstances, the early failure may occur in the contacting area of friction clutch elements.

Abdullah et al. [1–12] developed different numerical approaches to investigate the distribution of the temperature and the dissipated heat of the dry friction clutches which appeared due to the relative motion between the contacting elements. The thermo-elastic behaviors of the single and multi friction plates under varies operational conditions were studied intensively. The new mathematical model of a clutch disc to determine the quantities of storied energies at any time of the engagement period was presented. Furthermore, the surface roughness of the friction surfaces was measured experimentally and then used to simulate the numerical model of clutch disc to solve the thermoelastic problem.

Senatore et al. [13–15] investigated experimentally the effect of the slipping and contact pressure on the response of automotive clutches and brakes using different types of friction materials. The results presented the surface temperature field and the amount of

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the energy dissipation during the sipping time. It was assumed that the heat generated was a function of the wall temperature. The analysis was built based on Dirichlet and Neumann boundary conditions. The results proved that the contact pressure and the slipping speed are considered the most effective factors on the life time of the automotive clutches and brakes.

Ayala et al. [16] carried out the stability of a transient thermoelastic of the sliding systems. The effect of the intermittent contact on the thermoelastic problem of the sliding systems was studied. They showed that the frictional heat input is important at lower Fourier numbers. Under this condition, the critical speed is an inverse linear function of the duration of the contact time. In this case, the higher Fourier numbers was used to obtain the lower critical speeds.

Cho and Ahn [17] studied theoretically and experimentally the effect of the operating boundary conditions on the chattering of the automotive disc brakes. It was used a brake dynamometer to determine the disc's temperature during chattering. Fast Fourier Transformation with finite element methods was used to examine the thermoelastic behavior of three-dimensional brake system. The results showed that the phenomenon of the transient thermoelastic instability and the chattering in the brake discs have a significant affect on bulk temperature.

Al-Shabibi and Barber [18,19] investigated the thermomechanical behavior of the sliding systems using a reduced order model approximation which is described by one or more dominant perturbation or eigenfunctions. The mathematical model was built of the sliding system with a modest degree of freedom to obtain the contact pressure and the temperature distributions. The results were proved that a reduced order models have very good approximations in the early period of the automotive brake or clutch engagement when the sliding speed is above the critical sliding speed of the system.

This work highlights a fundamental point to adopt the accurate mathematical model in the design of the dry friction clutches. Two mathematical models have been developed based on the heat partitioning and the total heat generated approaches to computing the surface temperatures of the friction clutch during the sliding and full engagement phases. The finite element method was applied to conduct the thermal analysis of a friction clutch during multiple engagements in this research paper. The selected friction clutch consists of a single plate with two active frictional faces.

The short overview of the approaches which used in this work to simulate the thermal models of the automotive clutches to obtain the temperature field is:

- Heat partitioning approach: modeling the clutch system parts individually based on the heat partition factor to determine the amount of heat which enters into to each part of the clutch system.
- Total heat generated approach: modeling the clutch system parts together (whole model of the clutch system) and apply the total heat generated at the interface between the contact parts.

In this analysis, the effect of convection is considered for both approaches.

## 2. History of heat partition factor

The slipping between contacting surfaces will occur at the beginning of the clutch engagement until the driven shaft have the same speed as the driving shaft. The heat will generate from the rubbing surfaces due to the power dissipation by the frictional slip between the pressure plate and flywheel from one side and the friction clutch disc from the other side. The heat generated which appeared at the boundary between the contacting surfaces will be divided unequally into the contact elements of the clutch system. The amounts of these energies (heats) are depending on the material properties of each contacting element. The heat partition ratio (γ) represented the factor that specifies the amount of heat which enters into each body of two bodies system.

In the case when there are two bodies were in contact, when the first body (1) slides over the second body (2). The heat was generated due to friction, and the amount of the total heat generated at any time is $q_T$ (Fig. 1). The amount of the heat flux which enters into the body (1) is

$$q_1 = q_T \gamma \tag{1}$$

where $q_T$ is the sum of heat fluxes ($q_T = q_1 + q_2$). The amount of the heat flux which enters into the body (2) is

$$q_2 = q_T (1 - \gamma) \tag{2}$$

In the clutch system the amount of heat generated is,

$$q = \mu \, p \, V_s \tag{3}$$

where $\mu$, $p$ and $V_s$ are the friction coefficient, the contact pressure and the sliding speed, respectively.

Blok [20] presented the heat partition term of two bodies system in contact. Two models were selected to achieve sliding problem between two bodies this study. The first one was roughness surface with a square cross section ($a \times a$) and the second one is a circular (radius $= a$) contact with a semi-space surface. It was concluded that the contact area ratio (actual contact area divide by nominal contact area) was very significant. It was assumed that the heat was generated at the contact surfaces; the development of the heat expansion was just in the perpendicular direction to the contacting surfaces. Constant heat flux intensity at any time was applied. It was applied a heat flux ($q_1$) to heat up the rough semi-space surface, and the heat flux ($q_2$) was applied for heating the small zone of the semi-space surface. The heat partition factor expression when the sliding speed range was low ($V_s \leq 8\, k_2/25a$ or $P_e \leq 0.32$) is

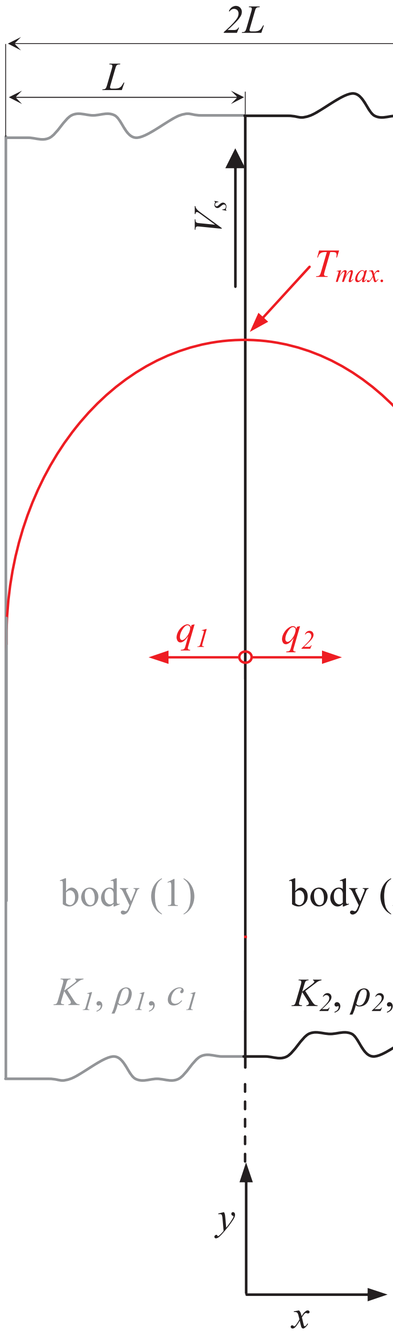


**Fig. 1.** Intensity and flow of heat generated at the rubbing surfaces.

$$\gamma = \frac{K_1}{K_1 + K_2} \tag{4}$$

where $P_e$, $k$ and $K$ are the Peclet number ($P_e = V_s\, a/k_2$), thermal diffusivity and thermal conductivity, respectively. The indices 1 and 2 in the Eq. (4) indicated to the first body and the second body, respectively. It was found that the maximum surface temperature of a rod model close to the edge opposed to the direction of relative motion. In this case, the range of sliding speed was high ($V_s > 8\ k_2/a$ or $P_e > 8$). It was assumed that the maximum temperatures were equal in both contact surfaces (cylinder and semi-space) when the sliding speed was high ($V_s > 40\ k_2/a$ or $P_e \geq 40$). The factor of heat partition under these conditions is [20]

$$\gamma = \frac{K_1}{K_1 + K_2\sqrt{\pi\ Pe/16}} \cong \frac{K_1}{K_1 + 0.44\, K_2\sqrt{Pe}} \tag{5}$$

Jaeger [21] determined the equation which used to estimate the factor of heat partition of two bodies system in contact with sliding. The first one was a semi-infinite rod with rectangular or quadrate cross-sectional area contacts with a semi-space surface. The heat partition factor (γ) was defined based on the mean temperature. The heat partition factor expression when the cross-sectional area of the rod was quadrate ($a \times a$) and the external surface was insulated, and rod slides on the surface of semi-space with a constant high sliding speed ($Pe > 20$) is,

$$\gamma = \frac{1.25\ K_1}{1.25\ \ K_1 + K_2\ \sqrt{Pe/2}}\ \cong\ \frac{K_1}{K_1 +\ 0.56\ \ K_2\ \sqrt{Pe}} \tag{6}$$

It was explained the case when two heat sources on the semi-spaces were found; the first source was the heat generated due to the friction, while the second heat source was the initial existing heating in the rod. In the meantime, the temperatures of the bar's tip were increases because of the frictional effect and were cooled due to the other surfaces of the semi-spaces. Owing to high sliding speed, a high amount of the heat will be absorbed by the semi-space. It was improved Eq. (6) taken into consideration the effect of the convection. The convective heat transfer coefficient ($h$) existing between the exposed area of a rod and the surrounding was assumed a constant value. The heat partition factor under these assumptions is,

$$\gamma = \frac{\sqrt{K_1}}{\sqrt{K_1}\ +\ 0.67\ K_2\ \sqrt{Pe/Bi}} \tag{7}$$

Where $B_i$ is the Biot number ($B_i\ =\ h\ a/K_2$). The amount of heat entered to the rod increases when the heat transfers coefficient increase too.

The most famous formula was used to find the factor of the heat partition in the friction clutches is Charron's formula [22]. This formula states the following,

$$\gamma = \frac{\sqrt{K_1\ \rho_1\ c_1}}{\sqrt{K_1\ \rho_1\ c_1}\ +\ \sqrt{K_2\ \rho_2\ c_2}}\ =\ \frac{K_1 \sqrt{k_2}}{K_1 \sqrt{k_2}\ +\ K_2 \sqrt{k_1}}\ =\ \frac{1}{1\ +\ \varepsilon} \tag{8}$$

The thermal activity is [23]

$$\varepsilon = \frac{K_1\ \sqrt{k_2}}{K_2\ \sqrt{k_1}} \tag{9}$$

Eq. (8) is used to calculate the amount of heat which enters to the clutch system parts (flywheel, presser plate and the clutch disc).

Based on the analytical solution of the thermal problem due to friction, it can be obtained the heat partition factor. When the contact occurred between two rods semi-spaces insulated (lateral surfaces), when the initial temperatures were equal for both bodies $T_i$ ($T_i\ =$ constant). The amount of heat enters into the first body is $q_1$ ($q_1\ =\ \gamma\ q_T\ =$ constant), and the amount of heat enter into the second body is $q_2$ ($q_2\ =\ \gamma\ q_T\ =$ constant). The temperature distributions in the semi-spaces are [24]

$$T_1\,(z,\ \ t) =\ \ T_i\ +\ \frac{2\,\gamma\ q_T\ \sqrt{k_1\ t}}{K_1}\ ierfc, \qquad 0 \le\ \ z \le\ \ \infty,\ \ \ t\ \ge\ \ 0 \tag{10}$$

$$T_2\,(z,\ \ t) =\ \ T_i\ +\ \frac{2\,(1-\gamma)\,q_T\ \sqrt{k_2\ t}}{K_2}\ ierfc\left(\frac{z}{2\ \sqrt{k_2\ t}}\right), \qquad 0 \le\ \ z \le\ \ \infty,\ \ \ t\ \ge\ \ 0 \tag{11}$$

where $z$ and $t$ are the spatial coordinate and the time. Applying Eqs. (10) and (11) to find the temperature at the contact surfaces ($z\ =\ 0$).

$$T_1\,(+0,\ \ t) =\ \ T_i\ +\ \frac{2\,\gamma\ q_T\ \sqrt{k_1\ t}}{K_1} \tag{12}$$

$$T_2\,(-0,\ \ t) =\ \ T_i\ +\ \frac{2\,(1-\gamma)\,q_T\ \sqrt{k_2\ t}}{K_2} \tag{13}$$

In this case, the thermal conductivity at the interface assumed perfect ($T_1\ =\ T_2$). It can be obtained the form of heat partition factor when compare between Eqs. (12) and (13), and then Charron's formula can be obtained too. This formula can be applied for a tribosystem that consists of a strip (plane-parallel) slide over a surface (semi-spaces) with a constant sliding speed. In case when the upper layer of the strip was insulated, the rate of heat from the contacting area to the strip was $q_1$ and the rate of heat which enters into the semi-space is $q_2$ [25,26],

$$q_1(t) =\ \frac{q_T}{(1\,+\,\varepsilon)}\ \ \sum_{n=0}^{\infty} \Lambda^n\ \left[erfc\left(\frac{n}{\sqrt{\tau}}\right) -\ erfc\left(\frac{n+1}{\sqrt{\tau}}\right)\right],\ \ t \ge\ \ 0 \tag{14}$$

$$q_2(t)\ \ =\ \frac{q_T\ \varepsilon}{2\,(1\,+\,\varepsilon)}\ \ \sum_{n=0}^{\infty} \Lambda^n\ \left[erfc\left(\frac{n}{\sqrt{\tau}}\right) +\ erfc\left(\frac{n+1}{\sqrt{\tau}}\right)\right],\ \ t \ge\ \ 0 \tag{15}$$

where

$$\Lambda = \begin{cases} (-1)^n\,|\lambda_q|, & -1 < \lambda_q \le 0 \\ \lambda_q^{\,n}, & 0 \le \lambda_q \le 1 \end{cases} \tag{16}$$

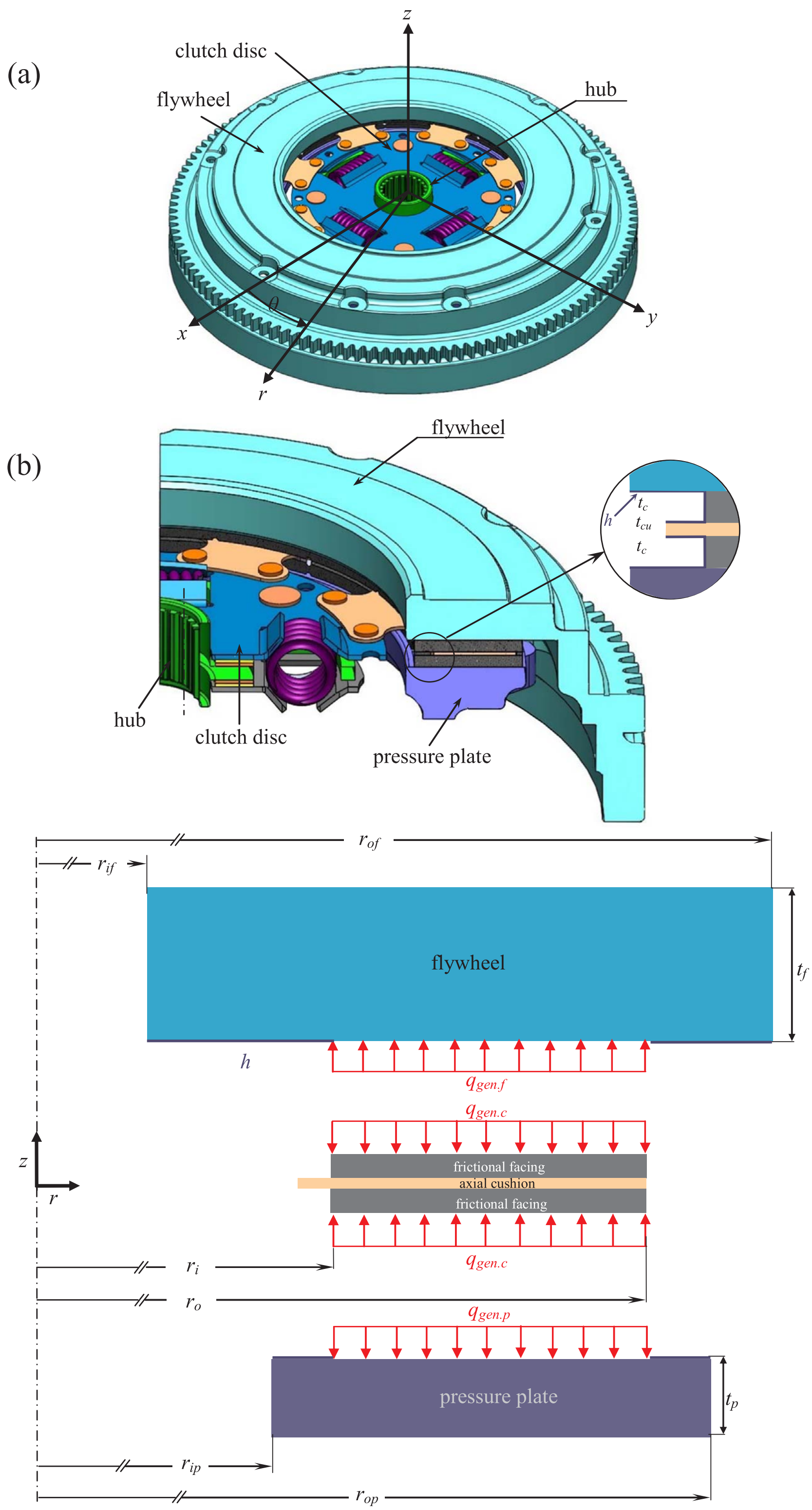


**Fig. 2.** (a) 3D model of the friction clutch assembly (single plate). (b) Axisymmetric model of the friction clutch (single plate).

$$\lambda_q = \frac{1-\varepsilon}{1+\varepsilon} \tag{17}$$

$$\tau = \frac{k_1 t}{d^2} \tag{18}$$

where $q_T$ and $d$ are the total power due to friction and the strip's thickness, respectively. By summing the Eq. (14) to Eq. (15) will obtain $q_T$, which proves the validity of Eqs. (14 and 15).

$$\begin{aligned} q_1(t) + q_2(t) &= q_T \sum_{n=0}^{\infty} \Lambda^n \; erfc\left(\frac{n}{\sqrt{\tau}}\right) - q_T \sum_{n=0}^{\infty} \Lambda^{n+1} \; erfc\left(\frac{n+1}{\sqrt{\tau}}\right) \\ &= q_T \; erfc(0) + q_T \sum_{n=0}^{\infty} \Lambda^n \; erfc\left(\frac{n}{\sqrt{\tau}}\right) - q_T \sum_{n=0}^{\infty} \Lambda^n \; erfc\left(\frac{n}{\sqrt{\tau}}\right) = q_T, \quad t \ge 0 \end{aligned} \tag{19}$$

In case, when separate both sides of Eq. (14) by $q_t$ and then find the limit when $\tau \to \infty$, it was obtained Charron's formula [Eq. (8)] also. Charron's formula is recommended by a considerate number of researchers to find the factor of heat partition which uses to find the solution of thermal problem that appeared in the automotive clutches and brakes [27–32] because of the results are obtained using this formula have a high accuracy in comparison to other formulas. In this work, Charron's expression is applied to find the factor of heat partition which specifies the share of each contact element of a friction clutch from the total heat generated at the interface during the slipping stage.

## 3. Mathematical model

Fig. 2(a) demonstrated the 3D model of the assembly of friction clutch (single plate) when all elements pressed together. Appling the equation of heat conduction (cylindrical coordinate) on a friction clutch was considered the beginning step to compute the temperature distribution, as follows [33]

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} = \frac{1}{k}\frac{\partial T}{\partial t}; \qquad r_i \le r \le r_o,\; 0 \le \theta \le 2\pi,\; 0 \le z \le \delta,\; t > 0 \tag{20}$$

where $k$, $r_i$, $r_o$ and $\delta$ denote the thermal diffusivity ($k = K/\rho\, c$), the internal radius of friction plate, the external radius of friction plate and the sum of the contact parts thicknesses, respectively. It can be represented the heat conduction equation based on the radial and axial coordinates only as shown in Fig. 2(b), because of there is no change for the heat flow in a circumferential path. Therefore, the equation of heat conduction reduces to:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = \frac{1}{k}\frac{\partial T}{\partial t}; \qquad r_i \le r \le r_o,\; 0 \le z \le \delta,\; t > 0 \tag{21}$$

Attributable to the existing symmetry in an axial direction ($z$-axis) of geometry and conditions of a fiction clutch, it can be simulated just the half part of the cutch plate to minimize the required time for computations. The initial and boundary conditions of the upper half of ungrooving clutch plate are [Fig. 2(b)]:

$$K_{cu}\frac{\partial T}{\partial r}|_{r=r_0} = h\,[T(r_o, z, t) - T_a], \qquad 0 \le \theta \le 2\pi,\;\; 0 \le z \le t_{cu}/2, \quad t \ge 0 \tag{22}$$

where $T_a$ and $h$ are the ambient temperature and the coefficient of heat transfer by convection, respectively.

$$K_c\frac{\partial T}{\partial r}|_{r=r_0} = h\,[T(r_o, z, t) - T_a], \qquad 0 \le \theta \le 2\pi,\;\; (t_{cu}/2) \le z \le (t_{cu}/2) + t_c, \quad t \ge 0 \tag{23}$$

$$K_c\frac{\partial T}{\partial z}|_{z=(t_{cu}/2)+t_c} = q_c(r, t), \qquad r_i \le r \le r_o,\;\; 0 \le \theta \le 2\pi,\;\; 0 \le t \le t_s \tag{24}$$

$$K_c\frac{\partial T}{\partial r}|_{r=r_i} = h\,[T(r_i, z, t) - T_a]\,, \qquad 0 \le \theta \le 2\,\pi,\;\; (t_{cu}/2) \le z \le\,, (t_{cu}/2) + t_c, \quad t \ge 0 \tag{25}$$

The temperature at the initial condition is,

$$T(r, \theta, z, 0) = T_i, \qquad r_i \le r \le r_o,\;\; 0 \le \theta \le 2\pi,\;\; 0 \le z \le t_c + (t_{cu}/2) \tag{26}$$

$$\frac{\partial T}{\partial r}|_{r=r_i} = 0; \qquad 0 \le z \le (t_{cu}/2), \quad t \ge 0 \tag{27}$$

$$\frac{\partial T}{\partial z}|_{z=0} = 0; \qquad r_i \le r \le r_o,\;\; 0 \le \theta \le 2\pi, \quad t \ge 0 \tag{28}$$

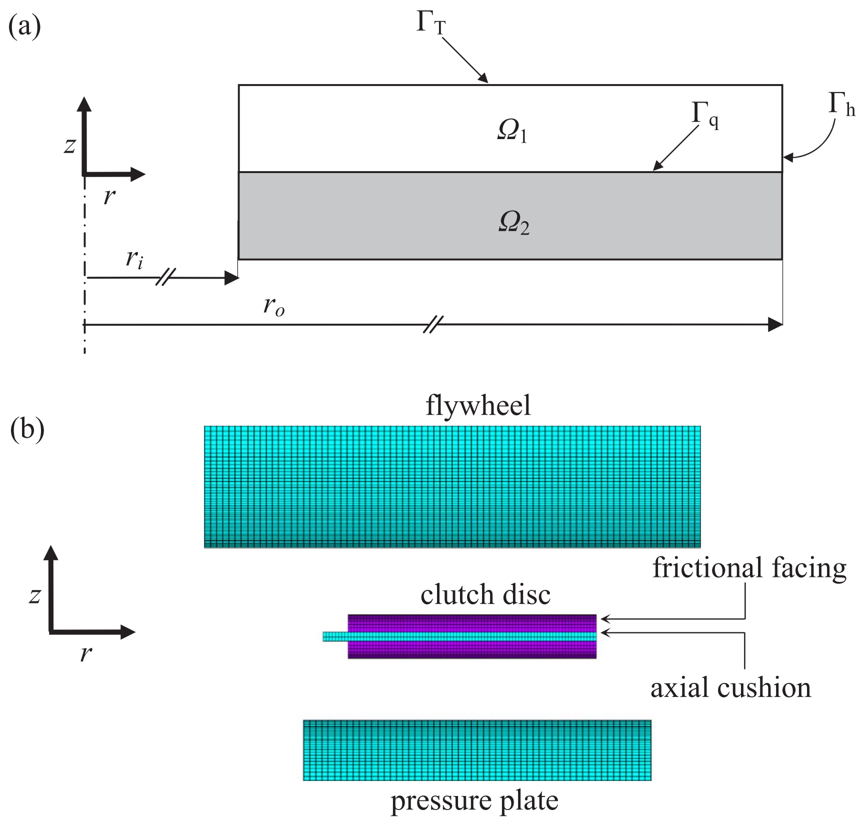


**Fig. 3.** (a) The sliding operation of two discs in contact, and (b) axisymmetric FE model of a single-disc friction clutch.

## 4. Finite element formulation

Two discs system $\Omega_1$ and $\Omega_2$ illustrated in Fig. 3(a). Owing to the symmetry, It can be represented the two discs system using axisymmetric model. It was assumed there is a relative motion between these discs. Based on these assumptions, the conditions of this model are:

$$T = T_p \quad on \quad \Gamma_T \tag{29}$$

$$q = -h\,(T - T_a) \;\; on \; \Gamma_h \tag{30}$$

$$q = q_{cu} \;\; on \; \Gamma_q \tag{31}$$

$$T = T_i\,, \;\; at \quad t = 0 \tag{32}$$

where $\Gamma_T$, $\Gamma_h$ and $\Gamma_q$ are the existing boundaries of the two discs system. $T_p$ denotes the temperature's prescribed. The approximated temperature can be defined as [34]

$$T\,(r,\,z,\,t) = \sum_{i=1}^{n} N_i(r,\,z)\,T_i(t) \tag{33}$$

where $N_i$, $T_i\,(t)$ and $n$ are the shape function, the nodal temperatures depend on time and the nodes number per element, respectively. Applying the Galerkin's method on Eq. (21) gives [34]:

$$\int_\Omega K\,N_i\left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} - \rho\,C\frac{\partial T}{\partial t}\right] d\,\Omega = 0 \tag{34}$$

Performing the integration of Eq. (34) gives the following,

$$-\int_\Omega K\left[\frac{\partial N_i}{\partial r}\frac{\partial T}{\partial r} + \frac{\partial N_i}{\partial z}\frac{\partial T}{\partial z} - \frac{N_i}{r}\frac{\partial T}{\partial r} + N_i\rho\,C\frac{\partial T}{\partial t}\right] d\Omega + \int_\Gamma K\,N_i\frac{\partial T}{\partial r}l\;d\,\Gamma + \int_\Gamma K\,N_i\frac{\partial T}{\partial z}n\;d\,\Gamma = 0 \tag{35}$$

The integral form of the boundary conditions is,

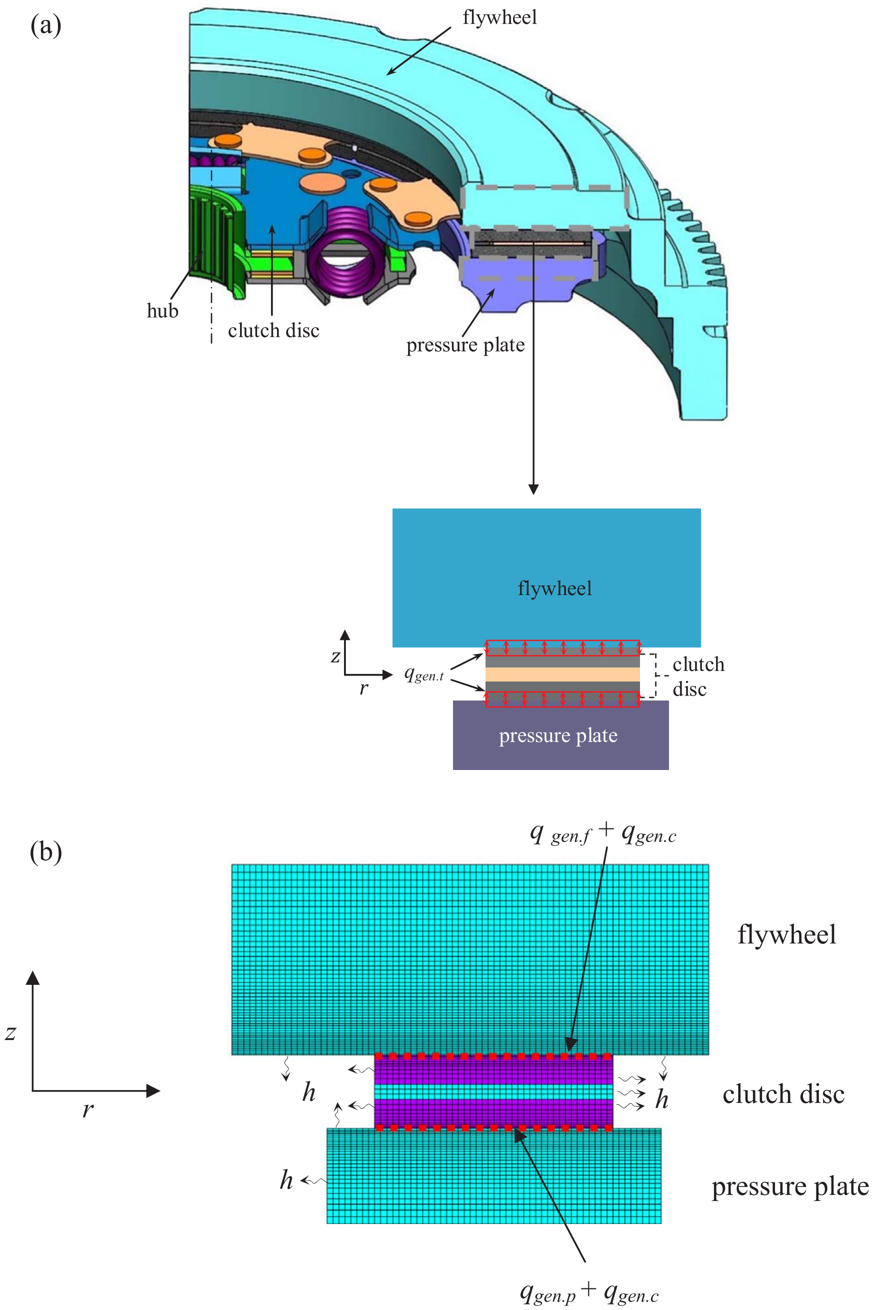


**Fig. 4.** (a) Axisymmetric model of the friction clutch with load condition (uniform wear), and (b) FE model with boundary conditions (No. of elements = 5378).

$$\int_{\Gamma} K\, N_i\, \frac{\partial T}{\partial r}\, l\, d\,\Gamma + \int_{\Gamma} K\, N_i\, \frac{\partial T}{\partial z}\, n\, d\,\Gamma = -\int_{\Gamma_q} N_i\, q\, d\,\Gamma_q - \int_{\Gamma_h} N_i\, h\, (T - T_a)\, d\,\Gamma_h \tag{36}$$

After substituting Eq. (32) and Eq. (29) to Eq. (31), it can be got the following equation:

$$-\int_{\Omega} K\left[\frac{\partial N_i}{\partial r}\frac{\partial N_j}{\partial r} + \frac{\partial N_i}{\partial z}\frac{\partial N_j}{\partial z} - \frac{N_i}{r}\frac{\partial N_j}{\partial r}\right] T_j\, d\,\Omega - \int_{\Omega} \rho\, C\, N_i \frac{\partial N_j}{\partial t} T_j\, d\,\Omega - \int_{\Gamma_q} N_i\, q\, d\,\Gamma_q - \int_{\Gamma_h} N_i\, h\, (T - T_a)\, d\,\Gamma_h = 0 \tag{37}$$

$i$ and $j$ denote the nodes. The matrix form of Eq. (37) is,

$$[C]\left\{\frac{\partial T}{\partial t}\right\} + [K]\{T\} = \{R\} \tag{38}$$

**Table 1**
The properties of materials and operation parameters [1].

| Parameters | Values |
|---|---|
| Inner radius of the friction plate and axial cushion, $r_i$ [mm] | 62.98 |
| Outer radius of the friction plate and axial cushion, $r_o$ [mm] | 87.21 |
| Thickness of the friction plate, $t_c$ [mm] | 3 |
| Thickness of the axial cushion, $t_{cu}$ [mm] | 1.5 |
| Inner radius of the pressure plate, $r_{ip}$ [mm] | 58.14 |
| Outer radius of the pressure plate, $r_{op}$ [mm] | 92.05 |
| Thickness of the pressure plate, $t_p$ [mm] | 9.69 |
| Inner radius of the flywheel, $r_{if}$ [mm] | 48.45 |
| Outer radius of the flywheel, $r_{of}$ [mm] | 96.9 |
| Thickness of the flywheel, $t_f$ [mm] | 19.38 |
| Maximum applied pressure, $p_{max.}$ [MPa] | 1 |
| Coefficient of friction, $\mu$ [1] | 0.3 |
| Number of friction surfaces, $n_p$ [1] | 2 |
| Torque, $T$ [Nm] | 432 |
| Initial angular slipping speed, $\omega_o$ [rad/sec] | 200 |
| Conductivity of the friction material, $K_c$ [W/mK] | 0.6 |
| Conductivity of pressure plate, flywheel and axial cushion. $K_p$, $K_f$ and $K_{cu}$ [W/mK] | 42 |
| Density of the friction material, $\rho_c$ [kg/m$^3$] | 1570 |
| Density of pressure plate, flywheel and axial cushion. $\rho_p$, $\rho_f$ and $\rho_{cu}$ [kg/m$^3$] | 7800 |
| Specific heat of the friction material, $c_c$ [J/kg K] | 534 |
| Specific heat of pressure plate, flywheel and axial cushions. $c_p$, $c_f$ and $c_{cu}$ [J/kgK] | 450 |
| Initial temperature, $T_i$ [K] | 300 |
| Number of engagements, $n$ [1] | 10 |

where $[C]$ and $[K]$ are the matrices of the heat capacity and the heat conductivity. $\{R\}$ represents the applied thermal load in the model. The other form of the above equation is,

$$C_{ij}\,\frac{\partial T_j}{\partial t} + K_{ij}\,T_j = R_j \tag{39}$$

where

$$C_{ij} = \int_\Omega \rho\, C\, N_i\, N_j\, d\Omega$$

$$K_{ij} = \int_\Omega K\left(\begin{array}{l}\frac{\partial N_i}{\partial r}\frac{\partial N_j}{\partial r}\{T_j\} + \frac{\partial N_i}{\partial z}\frac{\partial N_j}{\partial z}\{T_j\}\\ -\frac{N_i}{r}\frac{\partial N_j}{\partial r}\{T_j\}\end{array}\right) d\,\Omega \;+\; \int_\Gamma h\, N_i\, N_j\, d\,\Gamma$$

$$R_i = -\int_{\Gamma_q} q\, N_i\, d\,\Gamma_q + \int_{\Gamma_h} N_i\, h\, T_a\, d\,\Gamma_h$$

Or it can be written in the following forms,

$$[C] = \int_\Omega \rho\, C\, [N]^T\, [N]\, d\Omega$$

$$[K] = \int_\Omega K\, [B]^T\, [D]\, [B]\, d\Omega + \int_\Gamma h\, [N]^T\, [N]\, d\Gamma$$

There is a considerable effect of quality of the mesh on the degree of accuracy which required in the obtained results. In the other hand, the size of the time step was considered important to ensure that can be obtained reliable results. It was considered the suitable mesh is a fundamental issue to obtain the accurate results. Subsequently, it is important to discover the relationship between the biggest element's size in the direction of the highest gradient in the temperature and smallest period step to enhance the result's precision. The minimum size of time step is [28],

$$\Delta\, t = \frac{l_e^2}{4k} \tag{40}$$

where $l_e$ is the element‘s length in the direction of the highest gradient in the temperature. Finite element model (axisymmetric) was used to simulate the friction clutch as shown in Fig. 3(b). In this analysis, Crank-Nicolson algorithm was applied to conduct the thermal analysis in this work.

## 5. Total heat generation approach

This section presents the second approach which utilized to calculate the distribution of the temperature of the friction clutch (single plate) during the complete engagement cycle (slipping and full engagement periods). The complete model of the friction

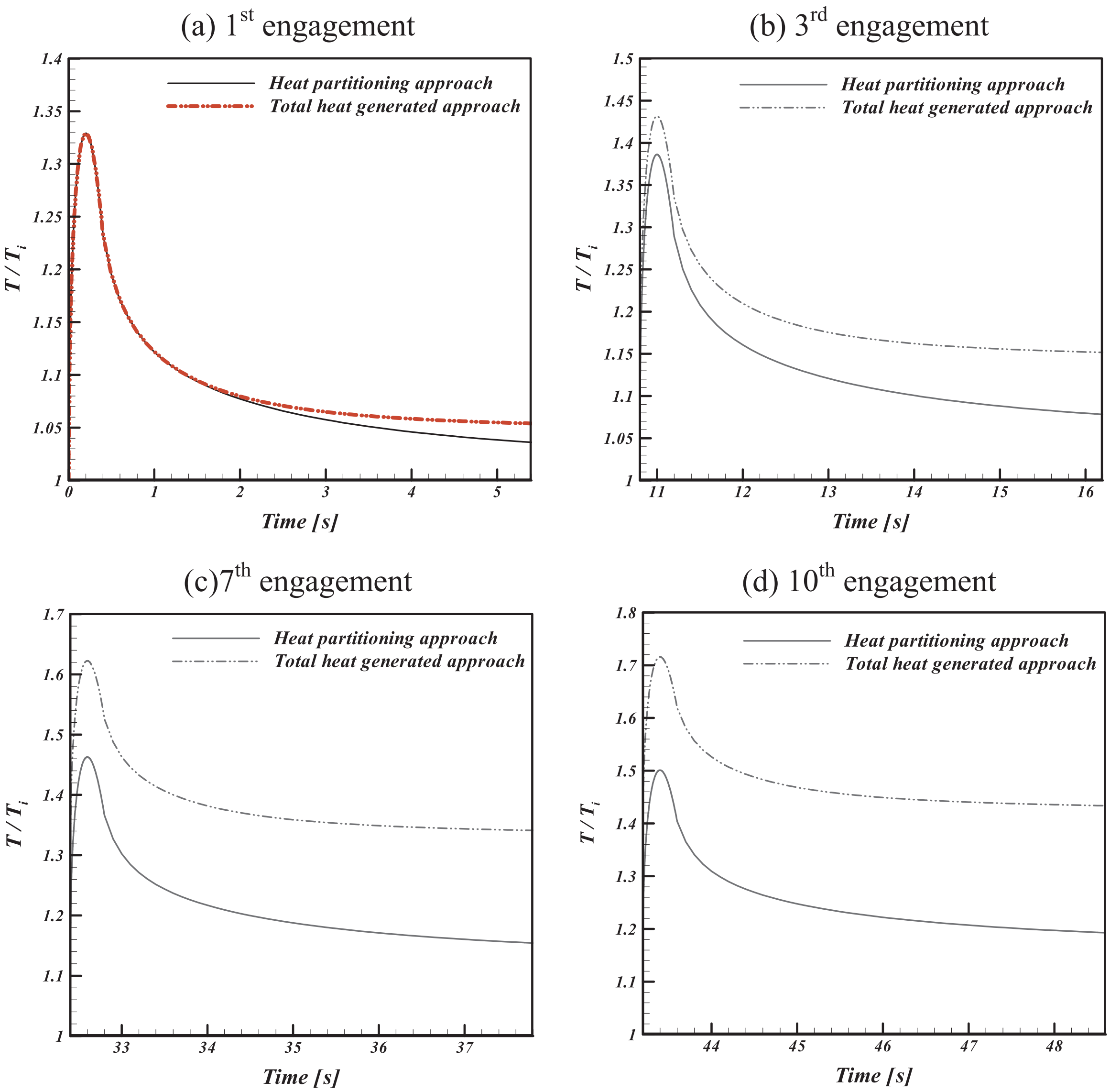


**Fig. 5.** Maximum surface temperatures during the different engagements.

clutch including main components (pressure plate, friction clutch disc, and flywheel) was simulated. It was examined the total amount of heat generated that appeared at the interface between the clutch elements as shown in Fig. 4(a). In this model, there is no more need to use the heat partition ratio. Fig. 4(a) shows the thermal model of a single disc clutch based on uniform wear assumption. The same assumption of the heat convection in the first approach was considered in this approach. Table 1 lists all properties of materials and operational parameters were used for both approaches. Axisymmetric finite element model of the friction clutch included all boundary condition is demonstrated in Fig. 4(b).

## 6. Results and discussions

In this section, the surface temperature distributions of the friction clutch plate during an individual and multiple engagements have been investigated deeply using finite element method. The simulation works consist of two parts; in the first part the heat partitioning approach was used, and in the second part the total heat generated approach was used. The fields of the temperature of the friction plate were computed during 10 engagements at a regular interval (5 s) for the same energy dissipation. It was assumed for all cases a uniform wear between all the contacting surfaces. All the results were focused on the surface of the friction plate which contacted with the pressure plate, because of the maximum temperatures occurred there. The reason for these results is the poor thermal capacity of the pressure plate in comparison with the flywheel.

Fig. 5 demonstrates the maximum normalized surface temperature during selected 10 engagements at the surface of friction plate.

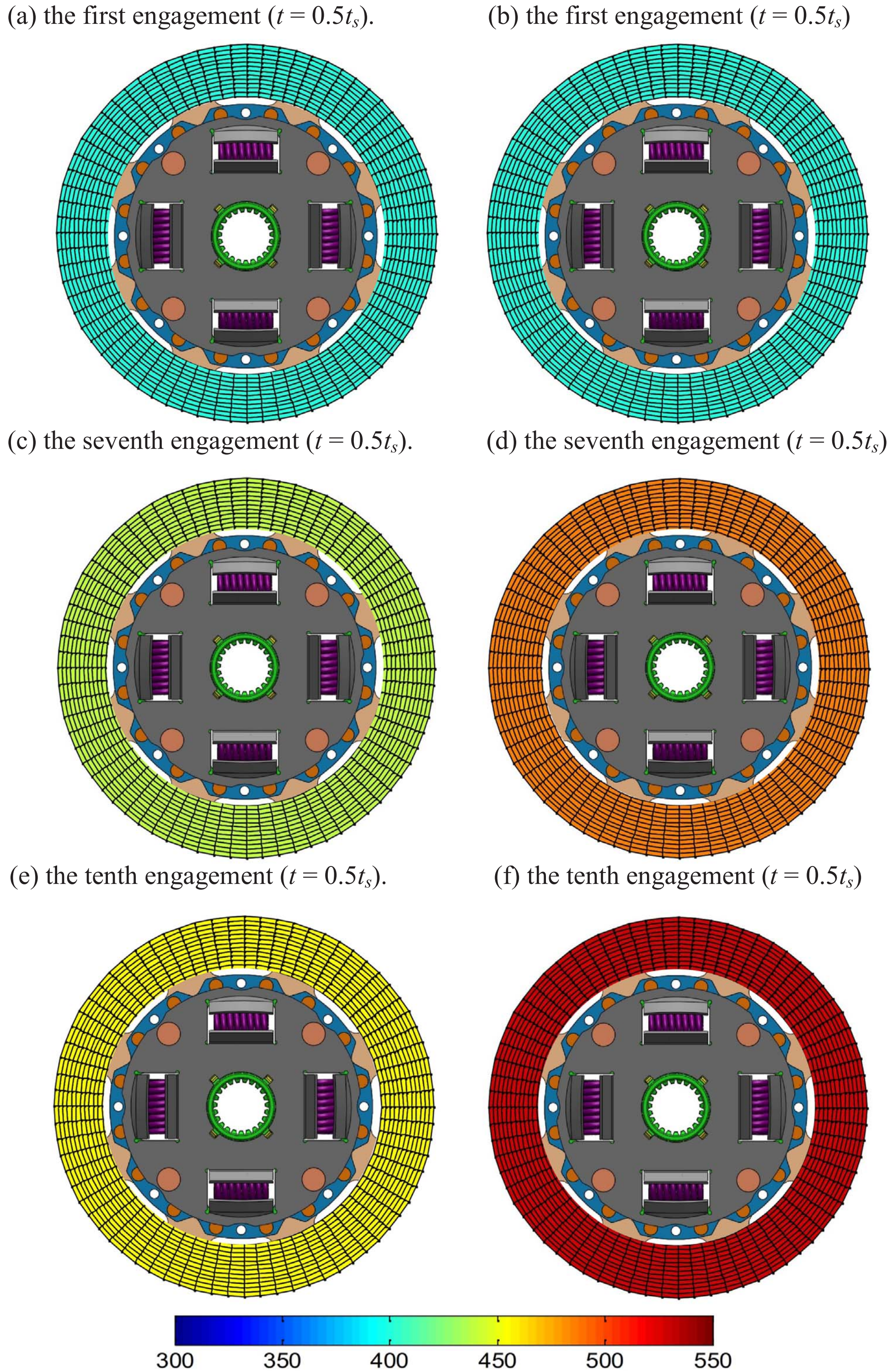


**Fig. 6.** Surface temperature [K] distribution of the friction plate during multi engagements plate. Heat partitioning approach (a, c & e); Total heat generated approach (b, d & f)], ($t_s$ is sliding period).

It can be noticed at 1st engagement that the highest temperatures (maximum values) which obtained based on both approaches were approximately equal (the difference was less than 1%). Later, the difference between them grows with an increase in the number of engagements. The percentages of difference between these approaches are found to be 1.3%, 3.1% and 13.7%, corresponding to the second, third and tenth engagements, respectively. The reason for these results because of the heat partitioning approach don't take into consideration the thermal capacity of the clutch elements (flywheel and pressure plate), while the second approach (total heat generated approach) takes into account this effect. The existing error in the values of temperature that obtained based on the first approach (the heat partitioning approach) leads the automotive engineers to calculate the temperature field inaccurately. Consequently, the estimation of the life cycles of the friction clutches is inaccurate also. On the other hand, the results obtained based on total heat generated are more accurate and reliable.

The surface temperature distribution of the friction plate during multi engagements using the heat partitioning and the total heat generated approaches as shown in Fig. 6, where $t_s$ is the sliding period. All results were selected at the middle time of sliding phase, because of the highest temperatures occurred at this instant for both cases and for engagements. The distributions of temperatures were uniform over the contacting area for all cases due to the applied a uniform thermal load (uniform wear assumption). It can be observed that the difference between the calculated temperatures depending on the approaches mentioned growing number of engagements (the peak difference occurred at the 10th engagement)

## 7. Conclusions and remarks

In order to evaluate the temperature distribution of the friction clutches during the heating period, it should be to minimize the existing error in the calculations process. The applied hypotheses to build the mathematical model have to be valid and realistic. The transient thermal problem of the friction clutch (single plate) during multi engagements has been solved using the heat partitioning and the total heat generated approaches. Axisymmetric finite element models were enhanced to find the numerical solution of the thermal problem of the friction clutches under operational conditions. The results show that the difference in the results of temperatures which obtained from both approaches was increased with a number of engagements due to the error exciting in the heat partitioning approach. The reason of this error is the heat partitioning approach didn't take the effect of the thermal capacity of the clutch parts in the calculation, and it's found the temperature field of each part of clutch separately, while the second approach (total heat generated approach) finds the temperature field for all systems in the same model. Therefore, it's recommended to use the total heat generated approach to evaluating the thermal behavior of the friction clutches during the sliding period (especially for multi engagements).

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