UDC: 621.82:620.17(047.3)=20
COSATI: 13-07, 09-02

# Thermal stresses of the motor vehicle clutch and brake friction discs

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**During the friction discs work, great amount of heat is produced on the friction disc surfaces. This leads to thermal loads which, exceeded limited values, can cause serious problems during disk work.**

**A research on thermal stresses on the frictional surfaces, arising under thermal load, presents a very complex task. The whole complexity of theoretical research of this problem based on existing literature notes is shown and some of the results obtained by solving complex mathematical equations are presented in the first part of this paper. The second part presents the results of friction disc testings on the test stand, based on which the thermal stress valuation is given. The possibility of thermal stress research by using modern program packs, such as Pro/Engineer, is also shown. It enables the simulation of the friction disc sliding process and the creation of the complete picture of the thermal stress level.**

*Key words:* **friction discs, thermoelastic instability, thermal stresses, theoretical analysis, experimental tests.**

## Introduction

FRICTION discs are used as clutches and brakes during the gearshift, clutches and brakes in turning mechanisms and working brakes, in transmission of tracked vehicles with planetary gearboxes. They perform various functions but work in the same way – they use friction force.

Friction elements are under a large amount of stress during their work. High rates of frictional heat generation in clutches and brakes result in high local temperatures and thermal stresses. They can increase more if the contact pressure on the nominal contact area is non-uniform. The initial pressure distribution is modified by thermal deformations. It is known that extremely non-uniform distributions may be produced in that way. The mechanism of this phenomenon is related to the nature of frictional heating. The generated frictional heat flux is proportional to the contact pressure increase due to the higher local thermal expansion.

The frictionally induced thermoelastic process may become unstable in some conditions and it is known as the thermoelastic instability (TEI). TEI is responsible for the generation of hot spots on the contact surfaces, which represents one of the major practical problems in high-speed clutches and brakes [1].

The feedback mechanism responsible for TEI is illustrated by circular diagram in Fig.1. Frictional heating during braking causes thermoelastic distortions which in return modify the contact pressure distribution $p(x,y,t)$ and hence the distribution of frictional heating $q(x,y,t)$. The connection between mechanical and thermal problems is introduced by the energy balance relation:

$$q(x,y,t)=\mu Vp(x,y,t) \tag{1}$$

where: $\mu$ is the friction coefficient and $V$ is the sliding speed. As seen in Fig.1, the product $\mu V$ is the feedback process gain. Therefore there will be a critical speed $V_{cr}$ above which every given sliding system will be unstable. Above the critical speed, non-uniform perturbations in the temperature field will increase, leading to a characteristic pattern of hot spots or bands on the brake discs.

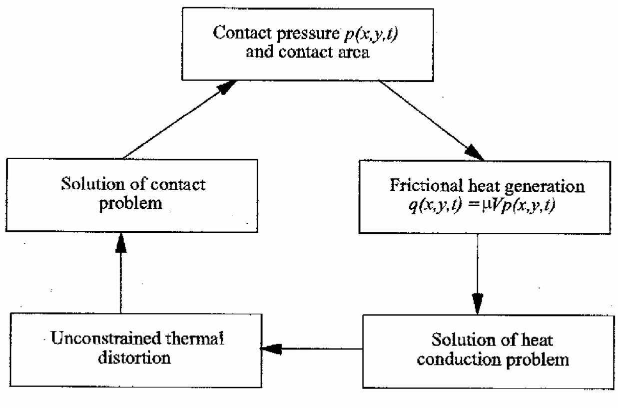


**Figure 1.** Feedback mechanism of TEI

## Theoretical study of thermal stresses on friction discs

To make a theoretical study, including the creation of mathematical model that describe the system, it is necessary to consider the whole problem which involves geometry, boundary conditions and material used. Some assumptions about the system behavior are made and some simplifications are introduced, in order to simplify the solving of mathematical equations during this process.

Typical multidisc wet clutches embody alternately as-

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sembled discs of two types. Those are discs with a steel core, lined with friction material layers, and discs made entirely of steel. Friction material layers may be made of metalceramics, sintered bronze, semimetal and nonmetal materials.

Fig.2 [3] shows a cross section of two adjacent clutches of the pack containing a larger number of discs. Thermal properties of the friction material differ considerably from those of the steel so it has to be taken into account. In the study it is assumed that the friction material is metalceramic because this is the material used in clutches and brakes employed in the transmission of tracked vehicles. The heat conduction problem will be regarded as axisymmetric, since the friction discs and the density of the heat flux generated on the friction surfaces have approximately an axial symmetry.

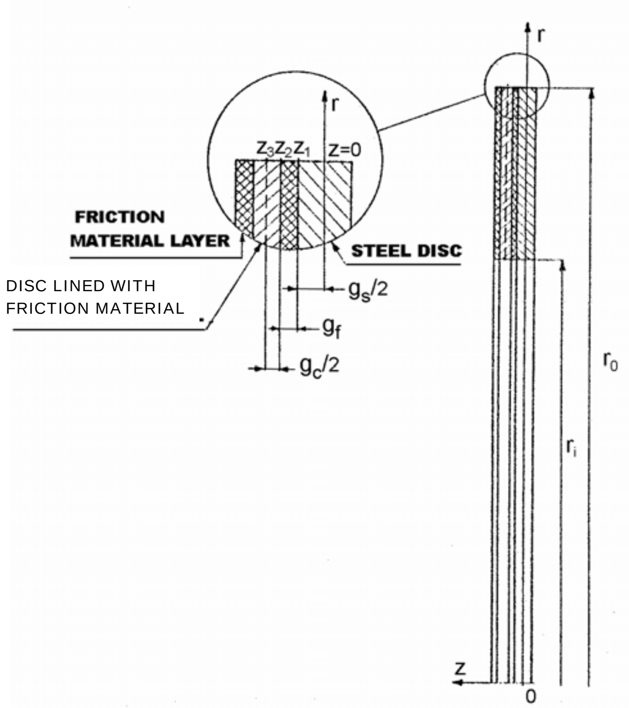



**Figure 2.** Geometry considered

Heat flux distributions on two friction surfaces of the same disc are similar. The temperature distribution on each of the discs will therefore be approximately symmetric in relation to the plane crossing the disc centre and perpendicular to the disc axis. For this reason it is sufficient to determine temperature distribution on a half of both adjacent discs. For the purpose of this study [1], it is assumed that the heat flow rate *q(t,r)* generated on the friction surfaces is known. However, it is necessary to determine the heat flux distribution *q(t,r)* by the radius, using eq.(1). It is also adopted that the friction coefficient $\mu$ does not depend of radius and it is constant. This is adopted because the friction surfaces have been lubricated by oil that flows through grooves on the friction material layer. It is assumed that the normal pressure p on the friction surface is uniformly distributed over the radius *p(r)=const*. Under these assumptions, the density of heat flux *q(t,r)* generated on the friction surface at each moment t is proportional to *r*.

The oil flowing through the grooves on the friction material surface absorbs a part of the heat being generated. However, this effect is of secondary importance for cooling of friction surfaces because of relatively small oil flow intensity, and it will therefore be neglected. Although small, cooling of cylindrical surfaces $r=r_i$ and $r=r_o$ will be taken into account because it influences local temperatures on the disc circumference and thus the local thermal stresses.

Fig.3 [3] illustrates diagrammatically the problem considered. Boundary conditions on the friction surface $z = z_1$ are not explicitly given since the proportions in which the heat flux *q(t,r)* is distributed on the discs concerned are unknown. It is very simular to the heat flux $q_3(t,r)$ on the boundary between the metalceramic layer and disc steel core. Adiabatic boundary conditions resulting from the symmetry of temperature fields are used in the center of each disc cross section.

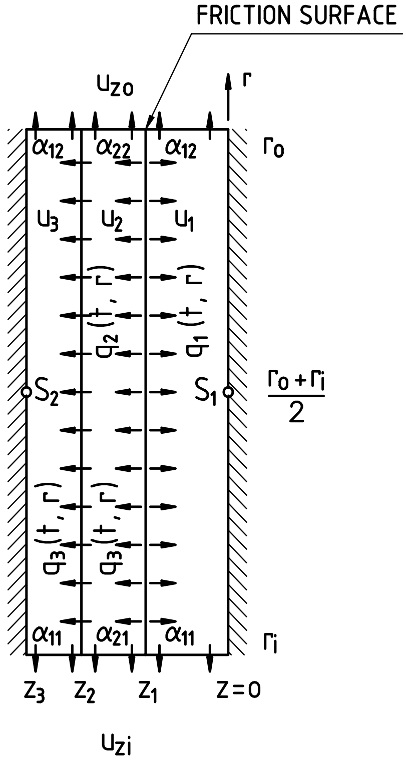



**Figure 3.** Scheme of boundary conditions

The heat conduction equations are [3]

$$\frac{\partial^2 u_1}{\partial r^2}+\frac{1}{r}\frac{\partial u_1}{\partial r}+\frac{\partial^2 u_1}{\partial z^2}=\frac{1}{a_1}\frac{\partial u_1}{\partial t} \tag{2}$$

$$\frac{\partial^2 u_2}{\partial r^2}+\frac{1}{r}\frac{\partial u_2}{\partial r}+\frac{\partial^2 u_2}{\partial z^2}=\frac{1}{a_2}\frac{\partial u_2}{\partial t} \tag{3}$$

$$\frac{\partial^2 u_3}{\partial r^2}+\frac{1}{r}\frac{\partial u_3}{\partial r}+\frac{\partial^2 u_3}{\partial z^2}=\frac{1}{a_3}\frac{\partial u_3}{\partial t} \tag{4}$$

The boundary and initial conditions for eq.(2) are as follows

$$\left(\frac{\partial u_1}{\partial r}-\frac{\alpha_{11}}{\lambda_1}u_1\right)_{r=r_1}=-\frac{\alpha_{11}}{\lambda_1}u_{zi} \tag{5}$$

$$\left(\frac{\partial u_1}{\partial r}-\frac{\alpha_{12}}{\lambda_1}u_1\right)_{r=r_0}=\frac{\alpha_{21}}{\lambda_1}u_{zo} \tag{6}$$

$$\left(\frac{\partial u_1}{\partial z}\right)_{z=0}=0 \tag{7}$$

$$\left(\frac{\partial u_1}{\partial r}\right)_{z=z_1} = -\frac{q_1(t,r)}{\lambda_1} \tag{8}$$

$$\{u_1(t,r,z)\}_{t=0} = f_1(r,z) \tag{9}$$

where

$a$ (m$^2$/s) - thermal diffusibility

$\lambda$ (W/mK) - thermal conductivity

$u$ - temperature field.

The boundary and initial conditions for eq.(3) and (4) are given in the same way [3].

In all given equations three unknown functions $q_1(t,r)$, $q_2(t,r)$ and $q_3(t,r)$ occur. The boundary conditions should therefore be complemented by three additional conditions:

$$q_1(t,r)+q_2(t,r) = q(t,r),\quad u_1(z_1)=u_2(z_1) \quad \text{and } u_2(z_2) = u_3(z_2) \tag{10}$$

where $q$ is a function given by eq.(1).

These equations are based on the Fourieov linear differential equation of second order [5]. Solving of this equation presents a big problem, and this can be done just in elementary cases. Oppositely, computer programs are used for solving it. A lot of data about thermal stresses, which can be obtained by solving these equations, can be found in literature.

*Calculation of thermal stresses*

It was already mentioned that the temperature field in the disc is axisymmetric and symmetric with respect to the plane $z=0$. The symmetry of the temperature distribution in the clutch disc produces an appropriate symmetry for both displacements and stresses. From the symmetry of the displacements with respect to the plane $z=0$ the displacements towards the z axis of the points situated in this plane are of zero value: $\{w(r,z)\}_{z=0}=0$. In this part, the results of calculation of thermal stresses, based on the previously given equations, were shown.

For a theoretical estimation, it was necessary to adopt physical and geometrical characteristic of the observed system, i.e. coefficients of thermal diffusivity and conductivity for the metal and the friction materials $a_s$, $a_f$, $\lambda_s$ and $\lambda_f$, heat transfer coefficients $\alpha_{11}$, $\alpha_{21}$, $\alpha_{12}$ and $\alpha_{22}$, geometric characteristis of the system $r_o$, $r_i$, $g_s/2$, $g_c/2$ and $g_f$, modulus of elasticity of the steel and the friction material $E_s$ and $E_f$ and Poisson's ratio $\upsilon_s$ and $\upsilon_f$. In the calculations, the initial temperature of the discs was assumed to be 80$^0$C [3].

The calculated relationships between the time and the temperatures growth on the characteristic points of the discs were shown in Fig.4, curve 2 -point S1 and curve 3 - point S2 (S1, S2- see Fig.3) [3], while curve 1 present maximum calculated temperature on friction surface. As it can be seen, the maximum values of the friction surface temperature, amounting to 248$^0$C, occurred towards the end of the engagement process at the moment $t$=0.25s. From the adopted model it follows therefore that, if the next successive clutch engagement takes place before the discs have completely cooled, the temperature gradients close to those given above will occur in the discs, i.e. similar values will have thermal stresses (with higher mean temperatures).

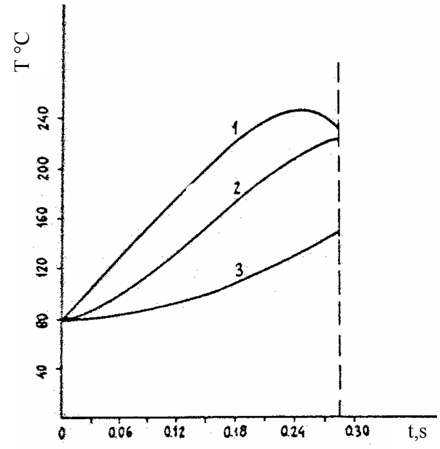



**Figure 4.** Temperature increases in the characteristic points

The temperature growth on the friction surfaces under generated heat during functioning provoques stresses inside the discs. These stresses can be also calculated.

From the symmetry of temperature fields, it follows that the distribution of the calculated stresses is also axisymmetric. There are three components of stresses: $\sigma_\theta$, $\sigma_r$ and $\sigma_z$. The calculated values [3] showed that the components $\sigma_\theta$, and $\sigma_r$ and are approximately 30 times higher than $\sigma_z$. This is the reason why the analysis of the triaxial stress state can be confined to the investigation of the components $\sigma_\theta$ and $\sigma_r$. The distributions of the calculated temperatures and stresses in the disc axial cross section, in a moment when these values reach their maximum, are shown in Fig.5. The calculated maximum values of the normal stresses reached about 129 MPa. The calculated stresses should not exceed the yield stresses. The calculated stresses in this case were far below the yield stresses. The experimens demonstrate that disc failure occurred only after a large number of engagements (from several thousands to several tens of thousands).

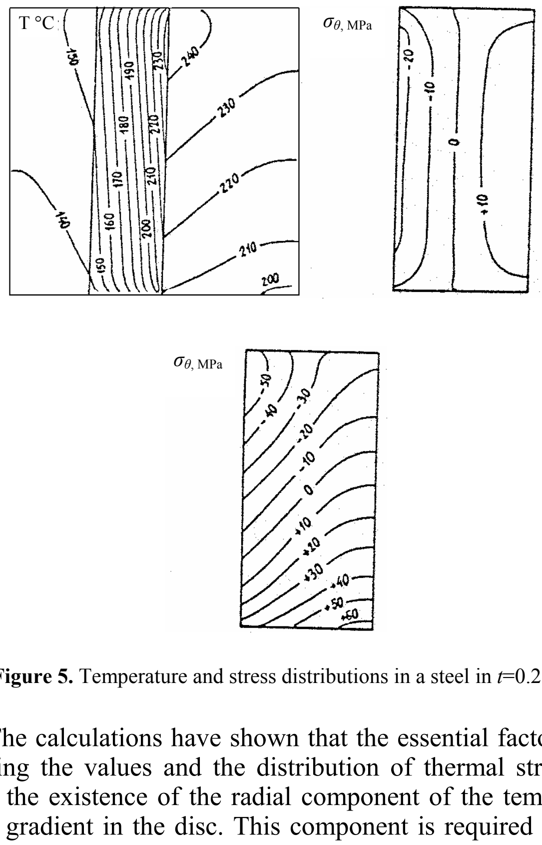



**Figure 5.** Temperature and stress distributions in a steel in $t$=0.25s

The calculations have shown that the essential factor affecting the values and the distribution of thermal stresses was the existence of the radial component of the temperature gradient in the disc. This component is required to be as small as possible. Therefore, an attempt should be also made to ensure that the heat flux generated on the friction surface is uniform throughout its entire width *(q(r)*=const*)*. Since $q=\mu \cdot p \cdot v$, this condition can be obtained by changing the distribution *p(r),* and by selecting properly the shape and dimensions of the grooves on the friction surfaces.

Fig.6 shows the maximum calculated temperatures and stresses on a disc made entirely of steel in the moment $t$=0.25 s [4]. Opposite to the previous calculation (Fig.5), in this case the pressure distribution was adopted to be non-uniform.

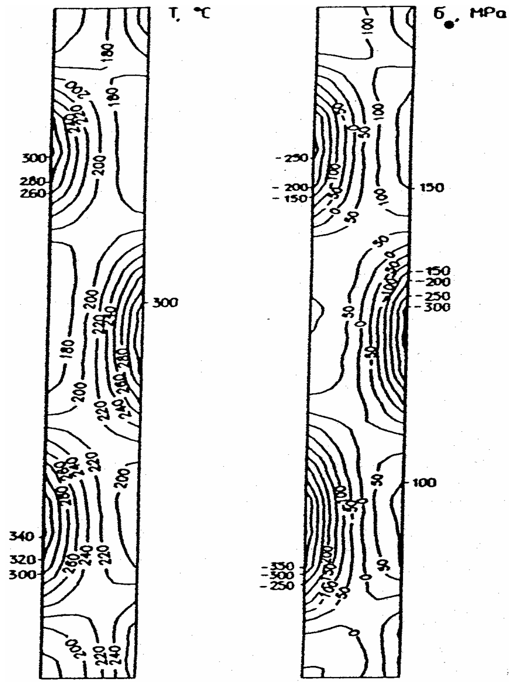



**Figure 6.** Calculated temperatures and stresses in the moment $t$=0.25 s

The non-uniformity of the pressure distribution can result from various factors:

- Design and material factors can make the initial distribution of pressure (under isothermal conditions, for example) non-uniform,
- The pressure distribution can become non-uniform during the engagement time due to the thermal deformation (even when the initial distribution was uniform).

## Experimental tests of thermal stresses

Besides the theoretical study, presented in the previous chapter, we carried out experimental tests on the stand PSS--51 for the completition of the thermal stress analysis.

This test stand is used for testing functional characteristics, i.e. performances and reliability of oil immersed friction discs. For this purpose there is a standardized procedure that defines test conditions. Fig.7 shows the scheme of the stand PSS-51.

The test stand operates as follows: the electromotor (1), through the belt drive (2), drives the hollow shaft with the accumulator of kinetic energy (3). By switching on the clutch (4), the elements with mass (6) start to rotate. When the aimed rotational speed is achieved, the clutch is switched off, and by switching on the brake (5), the mass with rotational elements is stopped.

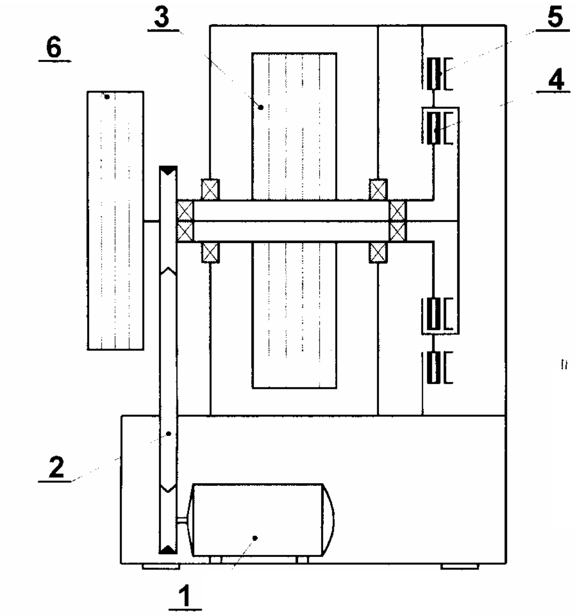



**Figure 7.** Scheme of test stand PSS-51

This is an inertia-type stand, and operates by of repeating standardized cycles to a desired engagement number.

The values that characterize testing conditions are:

- Moment of inertia of mass
- Activation pressure of clutches and brakes
- Nominal initial temperature on the friction brake
- Temperature of oil at the entrance and the exit
- Oil flow and pressure for the lubrication of discs
- Duration of the basic cycle.

The values that are measured or calculated during testing are:

- Activation pressure
- Initial rotational speed
- Moments of clutches and brakes,
- Friction coefficients of clutches and brakes
- Resulting sliding time
- Temperatures of friction surfaces of clutches and brakes
- Friction energy.

*Results of testing*

For the analysis of thermal stresses on the friction discs, the standardized procedure of testing was used. However, for the completion of these results, some additional tests were also done, but with some changes in test stand operation, in order to determine their influence on the thermal stress level. The test stand has a reduced ability in changing testing conditions, so it is possible to change just a few parameters, i.e. lubrication type of friction surfaces, quantity of oil in the test stand housing and activation pressure of clutches and brakes. Fig.8 and 9 show measured and calculated values for standardized conditions and with changed parameters in the test stand operation. As it can be seen, the changes in testing conditions caused changes in outcome values. However, it could be seen that temperatures on friction surfaces did not exceed $100^0$C. The obtained temperatures were so low that they could not affect the operation of these elements, and they were not appropriate for the analysis of thermal stresses.

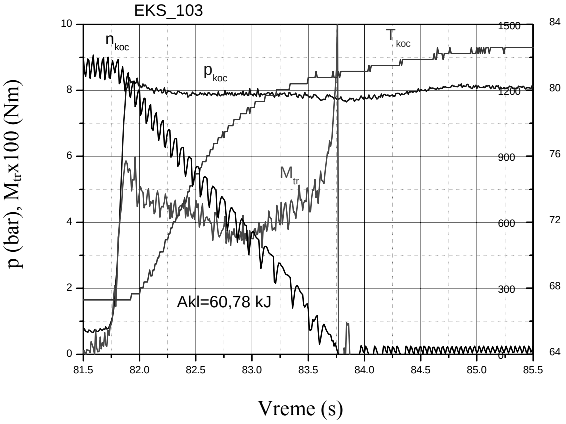



**Figure 8.** Characteristic parameters during brake engagement (standardized cycle)

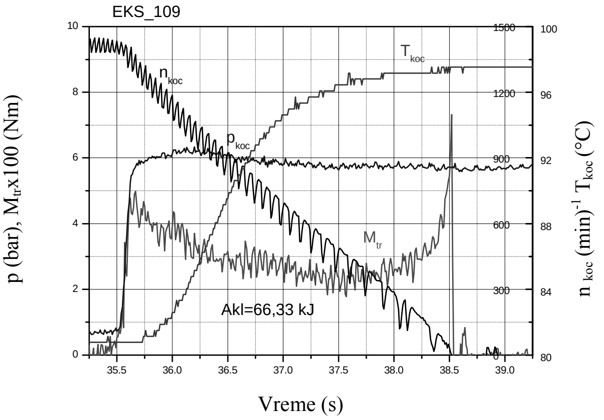



**Figure 9.** Characteristic parameters during brake engagement (decreased pressure activation and oil quantity for 80%)

The obtained results in both cases, (the standardized one and the one with changes in the test stand conditions), showed that loads imposed on friction elements were far below loads during real operation. During real operation these elements operate under conditions of boundary lubrication, and the obtained results show that in these analyses the friction discs were immersed in oil. This fact also verifies the calculated friction coefficients shown in Fig.10 and 11.

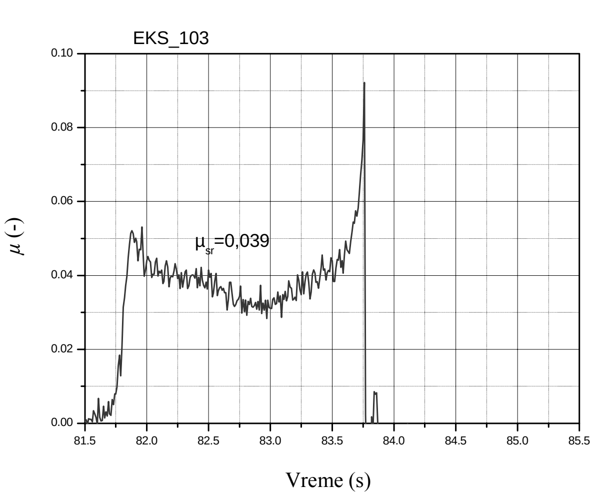



**Figure 10.** Friction cofficients during brake engagement (standardized cycle)

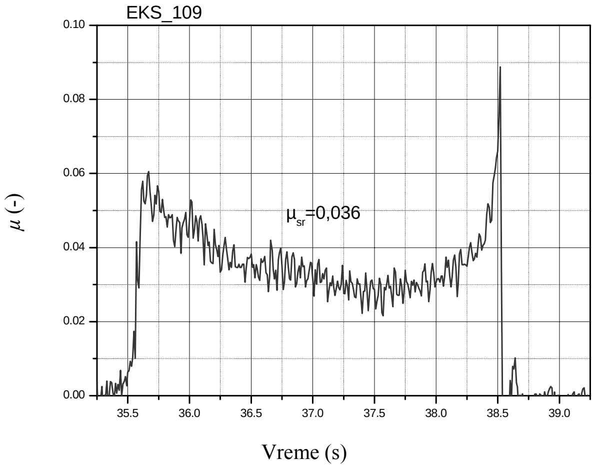



**Figure 11.** Friction coefficients during brake engagement (decreased activation pressure and oil quantity for 80%)

To make the operating conditions of the test stand more severe, it is necessary to change the way of lubricating friction discs in future experiments, because they have the greatest influence on the thermal stress level. The performed changes on the test stand during the additional tests showed some, but not adequate influence in connection with the increase of thermal stresses. In order to get conditions that fit boundary lubrication and thus increase thermal stresses, it was necessary to introduce more considerable changes on the test stand. The obtained results, with significant changes in test stand operation, could be a good basis for a more complete analysis of thermal stresses.

## New methods for the thermal stress analysis

Besides the previously specified methods for determination of thermal stresses in friction elements, new methods are used more and more nowadays. A lot of useful results can be obtained, by a theoretical analysis but in that case it is necessary to solve a very complex system of mathematical equations, by using computer programs. On the other hand, experimental tests can also give useful results, but just in a case when appropriate testing conditions are obtained. In order to achieve this, it is sometimes necessary to spend a lot of time and money.

New methods that solve previously mentioned problems refer to the application of complex program packages for computer simulation of a physical process as a whole and automatical computation of some values that characterize the analyzed phenomena. One of the programs that enable this analysis is Pro/Mechanica. In further research work it is planned to analyze thermal stresses of friction discs in Pro/Engineer. For this purpose it is necessary to make a virtual model of a friction disc. This includes making a 3D disc model of real dimensions, entering physical characteristics of the used material into the model, entering boundary conditions and loads it is exposed to during operating. Load levels are taken from experimental tests, and the method of entering load into the model depends on various shapes of load distribution on the friction surface. Depending on these two factors, different data are obtained. The obtained results can be used for the analysis of thermal stresses.

The advantage of Pro/Mechanica and similar program packages is that they form finite elements by themselves, and calculate stresses automatically on the basis of entery values, without additional writing of complex mathematical equations that describe the analyzed phenomenon. By using these programs time and money needed for research are significantly reduced.

*An example of estimation of thermal stresses of friction discs by using Pro/Engineer and Pro/Mechanica*

Here is presented one characteristic example of using the Pro/Engineer and Pro/Mechanica program packages. It concerns thermal stresses determination of discs in disc brakes. This problem is taken into account because thermal load, (as in the case of multiplate friction discs) can cause big problems to discs during their operation (hot spots, splits, etc.). The disc brake operates in the conditions of dry friction, while multiplate friction discs work in the conditions of boundary lubrication. This fact is taken into account by adopting an appropriate friction coefficient.

The Pro/Engineer and Pro/Mechanica Program packages are used for research of thermal stresses on the friction surfaces of disc brakes. Disc since overheating may be very unfavorable for its operation, it is taken as a subject of consideration. In order to run an analysis, it was necessary to make a virtual model, and the whole process of making a model is given in sequel. Some of the results of these analyses are presented as well.

A three dimensional model of a disc brake, used in this analysis, is given in Fig.12. After designing a 3D model of a disc brake, in the process of virtual model design, it is necessary to define material properties. The Pro Mechanical implies a so-called Library, which contains data of physical characteristics of materials mostly used in construction of mechanical elements. The mask presented in Fig.13. is used to introduce necessary changes in data and material properties.

After the definition of material properties, and all physical characteristics connected with it, the next step in the process of virtual model design is a definition of mechanical and heat boundary conditions.

Defining mechanical boundary conditions is connected with a necessity to compute mechanical loads on discs as a result of braking force action. It was adopted (because of large rigidity of the hub), that displacements towards the axis of discs are zero. The occuring error is thus believed to be of no significance.

Heat boundary conditions involve defining conditions under which heat delivered during braking is distributed over the model. There are three laws that define methods of heat distribution over a surface: heat conduction, convection and radiation.

To define conduction, it is necessary to define the coefficient of thermal conductivity λ of the data entered into the mask in Fig.13.

Heat convection presents a part of heat delivered to the atmosphere. It is defined by heat transfer coefficients. In case of the disc of the disc brake, it is necessary to analyze a part of heat delivered into the atmosphere with respect to the construction of the whole disc and aptitude to heat convection. The disc with the marks of adopted heat coefficients on particular surfaces is presented in Fig. 14, and the suggested values of these marks are specified in Table 1.

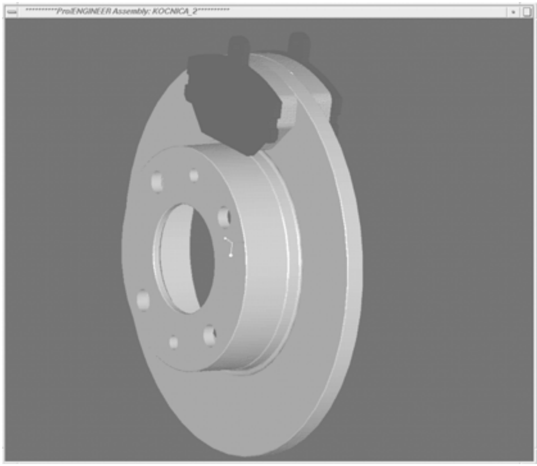


**Figure 12.** Digital model of a disc brake

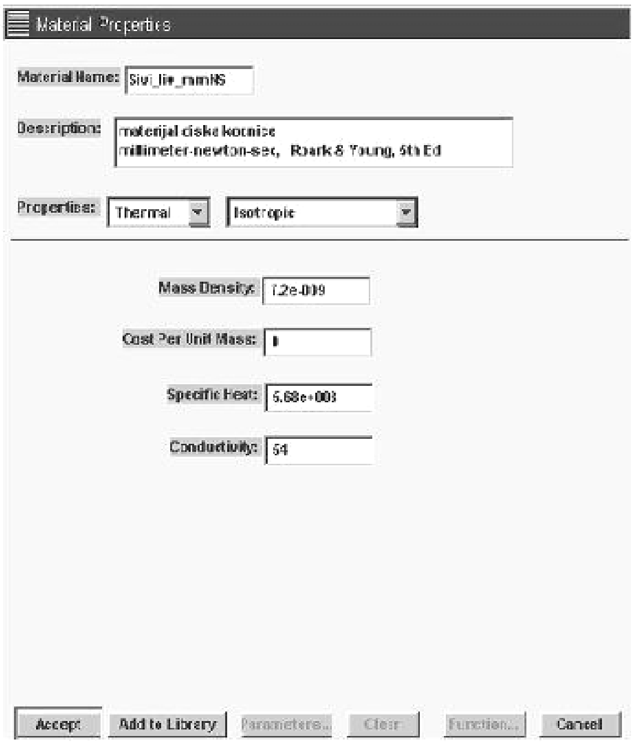



**Figure 13.** Masc for material properties introduction

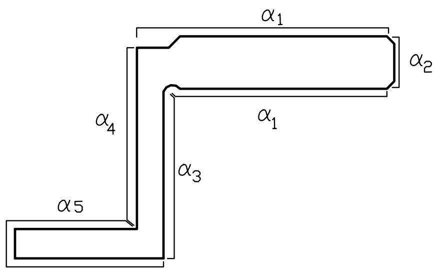



**Figure 14.** Heat transfer coefficients

**Table 1.** Adopted values of heat transfer coefficients

| | Rotational speed (rad/s) | | |
|---|---|---|---|
| $\alpha_i$ | 0 | 50.9 | 101.9 |
| $\alpha_2$ | 6 | 19 | 27 |
| $\alpha_2$ | 7 | 19 | 31 |
| $\alpha_3$ | 7 | 17 | 27 |
| $\alpha_4$ | 14 | 14 | 14 |
| $\alpha_5$ | 280 | 280 | 280 |

The third law of heat distribution concerning radiation was not taken into account, since average temperatures in discs did not exceed $300^0$C. There is a suggestion, in appropriate literature [5], that in the cases of temperatures below $600^0$C this factor can be neglected. Defining loads is the next step.

During braking, motor vehicle velocity, and hence, kinetic energy, decreases. It is assumed that all the quantity of energy is transformed into heat, and 91-99% of produced heat is delivered to the disc [7]. In this analysis it is adopted that 98% of the formed heat is delivered to the model. In order to simulate the contact between the disc and the pad, contact regions are made on the disc surface. It is adopted that the contact region is of a circular shape, the value of which is the same as the quantity of the active contact surface (with taking into account the surface roughness). Heat is delivered over the region into the model [8]. The quantity of heat delivered to the model for one period is the same as the difference of kinetic energy decreased for the adopted value (98%), for that period.

After all the processes of model defining, finite elements must be formed. The program packages make them, automatically. If a generated number of finite elements is too large, the analysis can be very slow, so it demands the best computer performance. In that case, it is recommended to form finite elements manually.

The part of the disc with contact regions and generated finite elements is presented in Fig.15. With the completition of the whole process, the model is finished and thermal analyses can start.

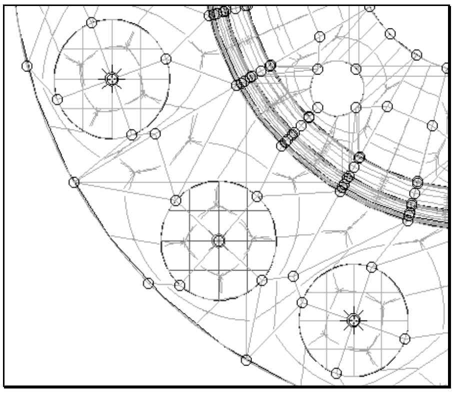


**Figure 15.** Part of the disc with contact surfaces and finite elements

The results of the calculated thermal stresses in some moment of the braking cycle are presented in the next pictures.

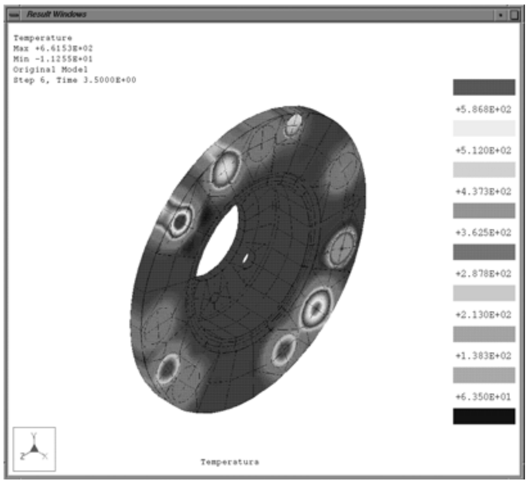


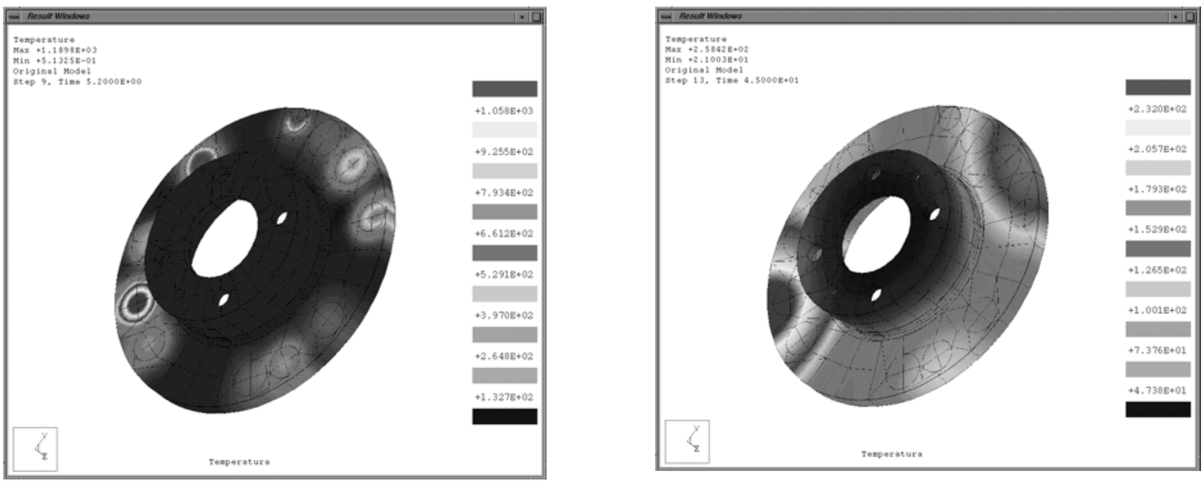

**Figure 16.** Temperature field in the moment $t$=3.5 s

The temperature field of the disc in the moment t=3.5s is given in Fig.16. This moment is chosen since it belongs to the middle period of the braking process that lasts for 5.2s, and shows graduality of delivering heat to the disc. The highest temperatures are on these parts to which temperature is distributed directly. The temperatures in these regions achieve a several times higher level than in the rest of the disc. Heat is delivered to particular contact surfaces, while in the rest of the disc it is distributed by conduction. The temperature field of the disc is presented in Fig.17a), (moment of finishing the braking process), and Fig.17b) presents the temperatures in the disc after cooling, (the end of the first braking cycle, directly before the next braking cycle).

Under the influence of temperature growth inside the disc there are stresses appearing that the Pro/Mechanica calculates automatically. The calculated values of the stresses and displacements for the moment t=5.2s are presented in Fig.18a) and 18b), and the stresses and displacements in the moment t=45s are in Fig.19a) and 19b).

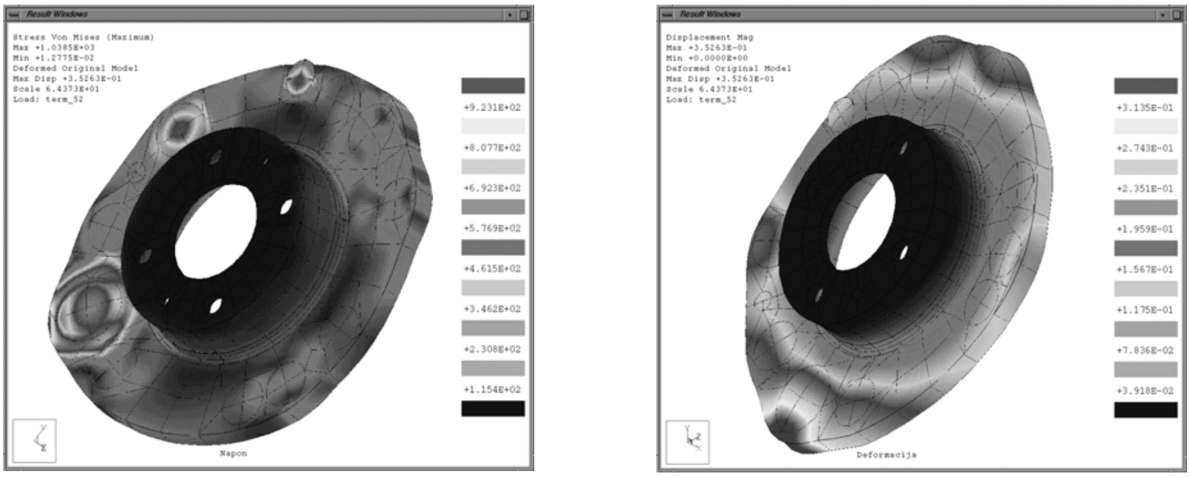
## Conclusions

Thermal stresses can cause considerable problems during the operation of friction elements. Their theoretical analy-



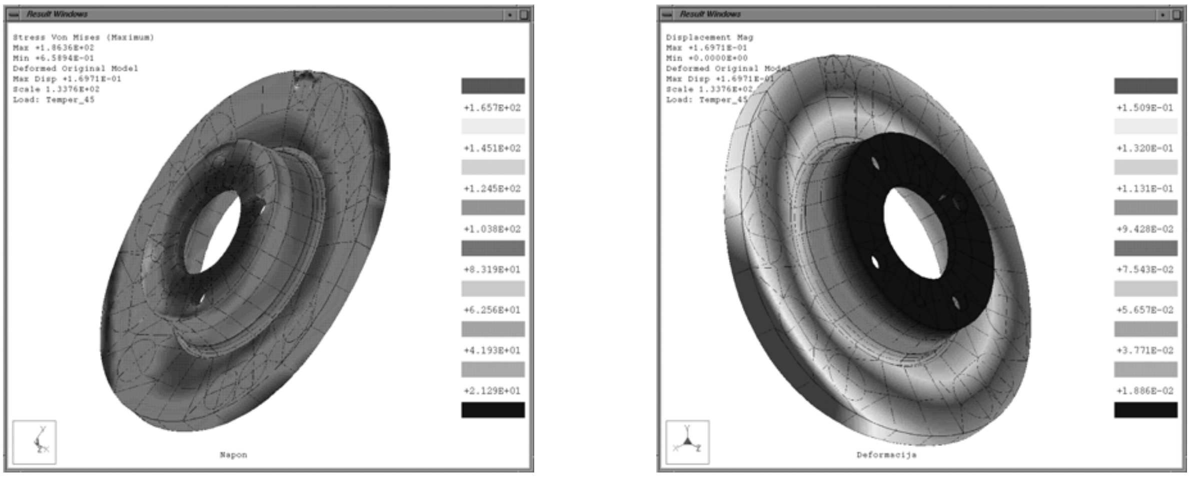

a) b)

**Figure 17.** Temperature fields in the moments $t$=5.2 and $t$=45 s respectively



a) b)

**Figure 18.** Stresses and displacements in the moment $t$=5.2 s



a) b)

**Figure 19.** Stresses and displacements in the moment $t$=45 s

ses implies forming mathematical equations. A system of these equations is often very complex, so they can only be solved by using appropriate computer programs. The results obtained from such calculations present a good basis for the analysis of thermal stresses and the analysis of the influence of some factors on their level.

On the other hand, experimental tests have shown that, by changing the operation conditions of the test stand, influence of these factors on the outcome parameters could be observed. However, in order to get the results which could serve as a basis for the thermal stress analysis, it is necessary to change operating conditions on the test stand and make them similar to those that appear in real exploitation of friction discs.

In the first and the second case, it is necessary to spend a lot of time and money to get appropriate results.

Therefore, program packages capable of simulating some physical processes, without spending time on equations that describe a system or expensive experimental tests, are increasingly in use nowadays. Time and money necessary for research have been thus reduced, and obtained results can be very useful for complete analyses of some phenomena such as thermal stresses. In order to get a clearer picture of the whole analysis process, we have shown here one characteristic example of using the program packages Pro/Engineer and Pro/Mechanica for the calculation of thermal stresses on friction discs of disc brakes.

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Received: 24.7.2003

# Termička opterećenja frikcionih diskova spojnica i kočnica motornih vozila

**Pri radu frikcionih sklopova, velika količina toplote se stvara na površini trenja frikcionih lamela. Ovo dovodi do pojave termičkih opterećenja koja, kada pređu dozvoljene vrednosti, mogu da prouzrokuju ozbiljne probleme u njihovom radu. Istraživanje termičkih napona koji nastaju kao posledica termičkih opterećenja predstavlja veoma složen zadatak. U radu se, najpre, na osnovu dostupne literature ukazuje na svu složenost teorijskog izučavanja ovog problema i prezentuju neki od rezultata koji su dobijeni rešavanjem složenih matematičkih izraza. U drugom delu ovog rada predstavljeni su rezultati ispitivanja frikcionih lamela na inercionom probnom stolu, na osnovu kojih se daje ocena o njihovom termičkom opterećenju. Na kraju, ukazano je na mogućnost izučavanja termičkih opterećenja i primenom savremenih programskih paketa kao što je Pro/Engineer, koji omogućava simulaciju procesa klizanja frikcionih sklopova i dobijanje potpune slike o nivou termičkih opterećenja.**

*Ključne reči:* **višelamelasti frikcioni sklopovi, termoelastična nestabilnost, termička opterećenja, teorijska analiza, eksperimentalna ispitivanja.**

# Contraintes thermiques sur les plateaux de friction chez les embrayages et freins de vehicules automobiles

**Une grande quantité de chaleur se produit sur la surface de frottement des plateaux de friction pendant le fonctionnement des assemblages de friction, ce qui conduit aux contraintes thermiques qui, en surpassant les valeurs limites, peuvent provoquer de problèmes sérieux pendant le fonctionnement. La recherche sur telles contraintes thermiques est très complexe. La première partie de ce papier met l'accent sur la complexité de la recherche téorique sur ce problème et présente quelques résultats obtenus à l'aide des expressions mathématiques complexes. La seconde partie contient les résultats des essais faits sur le banc d'essai inertiel – l'estimation de la contrainte thermique sur les plateaux de friction est fondée sur ces résultats. Finallement, on a démontré la possibilité d'étudier les contraintes thermiques en utilisant les progiciels modernes comme le Pro/Engineer qui facilite la simulation du processus de glissement des assemblages de friction et la création d'une idée complète sur le niveaux de contraintes thermiques.**

*Mots-clés*: **plateaux de friction, instabilité thermoélastique, contraintes thermiques, analyse téorique, essais expérimentales.**