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BRAKE SYSTEM STUDIES USING NUMERICAL METHODS

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The paper presents the examples of numerical simulations of braking systems performed in order to determine the parameters of their work. Using a typical finite element and meshless numerical methods, dynamic analyses of the brake were performed paying particular attention on the phenomenon of thermo-mechanical coupling occurring in the process of braking, as well as wear processes occurring on the surface of the linings. The article presents sequence of research steps, including hybrid modelling of braking system.

***Keywords**: brakes, FEM modelling, SPH modelling, thermomechanics, friction.*

W pracy przedstawiono przykłady symulacji numerycznych układów hamulcowych wykonanych w celu wyznaczenia parametrów ich pracy. Stosując siatkowe i bezsiatkowe metody numeryczne przeprowadzono dynamiczne analizy pracy hamulca ze zwróceniem szczególnej uwagi na zjawiska sprzężone (cieplno – mechaniczne) występujące w procesie hamowania oraz procesy zużycia zachodzące na powierzchni okładziny ciernej. W artykule przedstawiono kolejne kroki badań obejmujące modelowanie hybrydowe układu hamulcowego.

***Słowa kluczowe**: hamulce, modelowanie MES, modelowanie SPH, termomechanika, tarcie.*

1. Introduction

This paper discusses the issues associated with the operation of the braking system, which is one of the most important safety systems in a car [23]. As a result of braking friction surfaces of the lining are heated to the temperature up to 500°C [14, 15, 19]. Such thermal conditions and other phenomena accompanying the process of braking cause a constant grow of requirements for the properties of materials used for brake linings.

On the other hand, increasing availability of appropriate tools causes that, as in many other areas, numerical analyses become increasingly important in the brakes design process. Nevertheless, the nature of the physical processes associated with braking makes their correct modelling (using the most popular analytical tools based on the finite element method) virtually impossible [20] at the moment. The above statement seems to be surprising, but only to the point where we will realize what should be included in a numerical modelling of the brake process, such as:

- nonlinearities associated with large rotation,
- nonlinearities of boundary conditions (contact, friction),
- non-stationary nature of the process,
- thermo-mechanical coupling,
- wear of linings required to model change in geometry of friction pair.

For this reason, in the numerical studies of braking systems, various simplifications are usually used. For example, the geometry of brake is simplified to axisymmetric representation [24] or system is analysed in the plain strain [16].

In the case of temperature field analysis, many authors define stationary or non-stationary heat sources and ignore the fact that heat is generated by friction [2, 4, 8, 21]. In other words: thermo-mechanical coupling is omitted in investigations.

Wear of the friction pair is most commonly investigated using tools developed by tribology. From the point of view of this work these methods can be divided into those that use semi-empirical equations describing surface wear [3, 12, 17], and those in which models describe some aspects of the phenomena occurring at the micro level, i.e. the level of the surface layer of components. Review of publications representing the latter approach can be found in [26].

An interesting attempt to implement a tribological approach to macro models is presented in [18], where a procedure for random generation of irregularities appearing on the working surface of the brake pad was described. In [25] attempts to assess the lining wear in the macroscopic model, however the problem was brought to the analysis of the sole brake pad running in stationary conditions. Popular linear Archard model was used to describe the wear process. A more accurate model of the whole brake is described in [1], although thermal effects were also omitted and geometry changes resulting from the wear were modelled based on experimental measurements by the arbitrary movement of nodes in the direction normal to the contact surface.

Above examples show the difficulties that arise when trying to take into account the microscopic processes accompanying wear in the macroscopic models of whole components. In the case of brake, this inconvenience manifests itself when we will try to include a sufficiently accurate description of the surface friction geometry in the assembly model [17, 20]. One way to solve this problem is the implementation of meshless methods. Here analysed object is represented by a set of points (particles) with a finite size which can form any shape [22]. Although known for about 30 years, these methods become popular in the analysis of tribological processes quite recently [11, 20]. Authors, as part of this trend, also applied this meshless approach to simulate wear of friction surface.

Presented paper discusses numerical simulations of braking process conducted using LS-Dyna explicit code. Obtained results for brake with two pistons and for modified set-up with additional third piston are compared. Due to the fact that a number of thermo-mechanical phenomena begin in micro scale, it was decided to carry out the analysis of contact surfaces in a microscopic scale including the wear

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl*** process using the aforementioned meshless method, more particular Smooth Particle Hydrodynamics (SPH) technique.

In the earlier work authors presented results of a drum brake analyses [10]. Numerical models presented there enabled the development of modelling procedure that allows to estimate the efficiency of brakes and to predict the most exposed areas to wear (Fig. 1)

Obtained results showed that algorithms of conversion of work done by friction forces into heat are working properly. This was one of the most important elements necessary for proper numerical description of brake. It should be pointed out, though, that the algorithms are very sensitive to the parameters describing thermal and mechanical properties of brake materials. Another important factor strongly affecting results is a discretisation of contact areas of interacting bodies. This translates into accurate values of contact forces, which is a key factor for the proper definition of heat flux generated by friction.

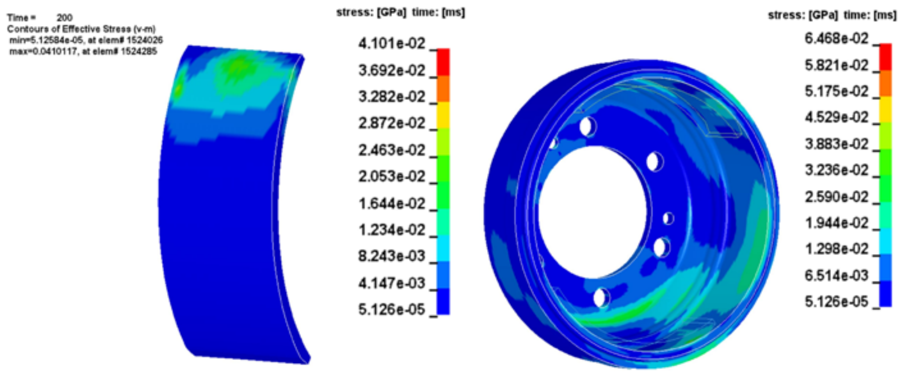

Fig. 1. Stress distribution in the brake lining and drum at the specific moment of time [10]

2. Description of numerical methods used in brakes studies

In order to model disc brakes mentioned earlier, commercial software LS-Dyna was used. This program utilizes method of direct integration of complete dynamic equation of motion using slightly modified central difference scheme. In this approach velocity and acceleration of a point are described as follows [13]:

$$\dot{x}_{n+\frac{1}{2}} = \frac{1}{\Delta t_{n+\frac{1}{2}}}\left[x_{n+1} - x_n\right] \tag{1}$$

$$\ddot{x}_n = \frac{1}{\Delta t_n}\left[\dot{x}_{n+\frac{1}{2}} - \dot{x}_{n-\frac{1}{2}}\right] \tag{2}$$

Matrix equation of motion for nonlinear case is described as follows [13]:

$$\mathrm{M}\ddot{x}_n = \mathrm{F}_n^{ext} - \mathrm{F}_n^{\mathrm{int}} - \mathrm{C}\dot{x}_n \tag{3}$$

where: M – global stiffness matrix, C – global damping matrix, F_n^{ext} – external forces (vector) $\mathrm{F}_n^{\mathrm{int}}$ – internal forces (vector).

As it was mentioned before, in each numerical model, contact conditions between lining and drum or disk were defined. It was assumed that friction coefficient in the lining-drum (disk) pair has constant value of $\mu = 0.4$ (chosen based on experimental data [14, 15]). Contact algorithm was based on penalty function approach [9,13]. Based on calculated friction forces, heat generation was computed using the following formula:

$$F_f \frac{dS}{dt} = mc_p \frac{dT}{dt} \tag{4}$$

where: F_f – friction force, S – distance of braking, t – time, m – mass, c_p – specific heat, T – temperature.

Due to the fact that braking time covered by analyses was short enough no convection and radiation were included in performed simulations.

3. Numerical model of disc brake

After studies on the brake drum, authors decided to conduct, in addition to the previous tests, analyses on disc brake setup. Obtained results gave a broader view of the phenomena occurring during the braking process and the possibility to support the design process of brakes by numerical analysis. Figure 2 shows the CAD model of discussed brake system.

At a later stage of investigation FE model was developed based on the CAD geometry.

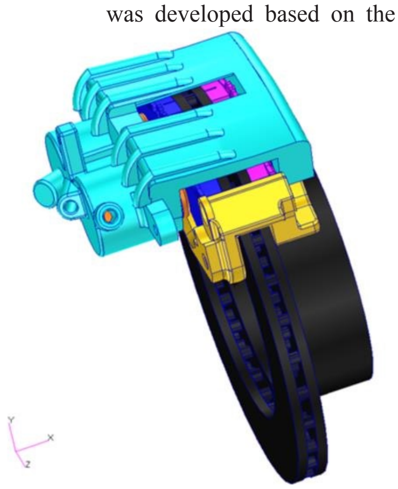

Fig. 2. CAD model of brake system with dual-piston clamping system [5]

Figure 3 shows the model of brake after discretization process. Due to the complex shape of the disc, it was modelled using tetragonal elements (TET4), while pads and lining were modelled using hexagonal solid elements (HEX8). Running belt test bench was modelled as a concentrated mass (with proper moments of inertia defined) connected to the disc via beam elements. The entire model consisted of approximately 150500 elements and 40300 nodes.

3.1. Initial and boundary conditions

Solution to the problem – involving solution of differential equations – requires initial and boundary conditions, i.e. adoption of appropriate loads (e.g. power, torque force, pressure) and constraints of FE model by taking back the degrees of freedom in the selected

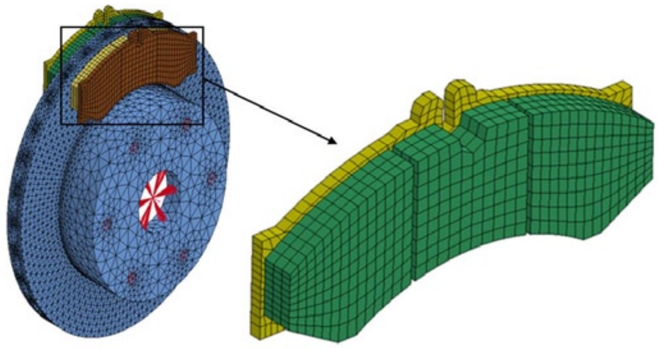

Fig. 3. Numerical model of the disc-brake system [5]

nodes. In addition, the solution must include constitutive equations with stress and displacement equilibrium conditions.

Each part of FE model was given the same material properties as in the drum brake model [10]. The disc was supported in the mounting holes (Fig. 4), which was an approximation of the actual conditions of restraint.

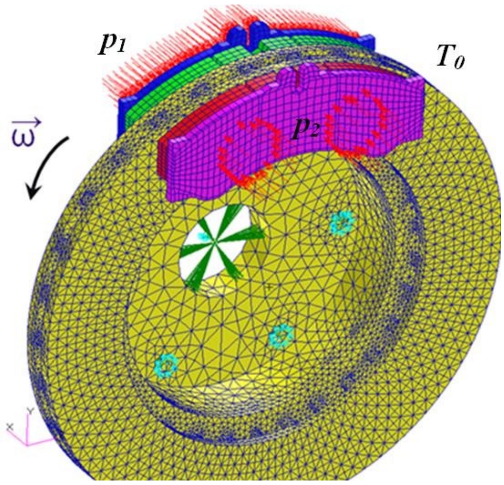

Fig. 4. Initial boundary conditions in the numerical model of disc brake system [5]

In areas of pistons and clamp interaction, pressure corresponding to this prevailing in the hydraulic system on the stand was applied to the pads (p_1 = 6.238 MPa at the calliper side and p_2 = 25.858 MPa at the pistons side). Also, the initial rotational velocity ω_0 = 42.0 rad/s was applied to the disc and concentrated mass. Moreover, initial temperature T_0 = 20° (293 K) was prescribed to all nodes in the model.

3.2. Results of two-piston brake numerical analyses

The figures below shows the results of the two piston brake. Figure 5 shows distribution of the normal stress in lining: a) pushed against calliper, b) pushed against pistons, for a selected time t = 0.4 s.

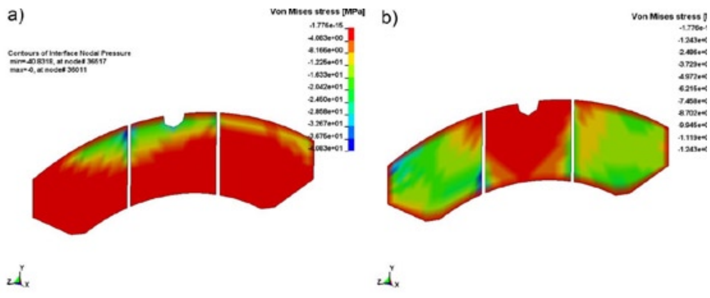

Fig. 5. Normal stresses on the brake pad surfaces at t=0.4 s, a) pressed using clamp, b) pressed using pistons

Above figures shows uneven work conditions of inner and outer lining. Based on this it can be concluded that this visible inequality can be resulted from the badly selected stiffness and geometry of the brake calliper. Thus, it was decided to modify the brake system.

3.3. Model of improved brake version

Based on the conducted research, it was decided that from the efficiency and economy point of view, the best direction of the brake design modification will be to apply additional pistons, keeping – if possible – outer dimensions of brake intact. The outcome was a three pistons brake, with altered calliper geometry, utilizing the same disc and the brake pads.

From the point of view of FEM modelling, the proposed design modifications were very small. The use of the same disc and pads allowed to use the same finite element mesh with initial conditions and constrains unchanged. The only change, compared to the original mode, was a different method of application of the pressure acting on the structure (Fig. 6). Applied pressure values were as follows: p_1=55.954 MPa outside of the calliper, p_2=34.289 MPa inside of the calliper, p_3=22.408 MPa external pistons and p_4=21.279 MPa middle piston.

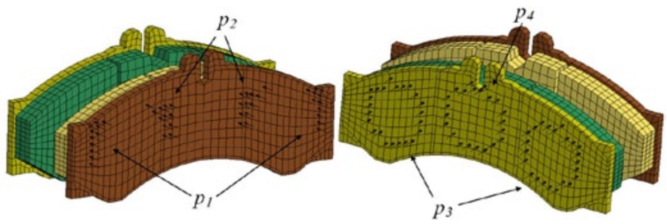

Fig. 6. Forces distribution in the triple-piston brake system

3.4. Comparative analysis

Applied modifications resulted in higher stresses on the surface of the pads. In order to compare working conditions of the structure before and after the modification, graphs of temperature versus time for both layouts (Fig. 7) and change of the rotational speed of the disc in time were prepared (Fig. 8). As expected, increasing the number of pistons resulted in increased pads downforce acting on disc. As a result, temperature raised and the brake disc deceleration increased.

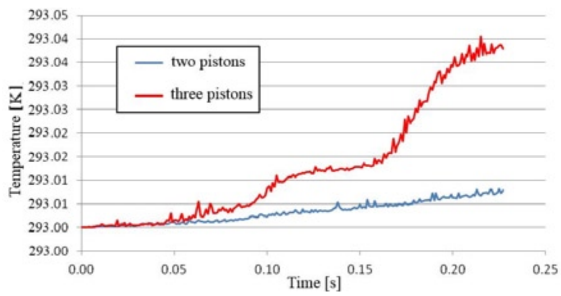

Fig. 7. Comparison graph of temperature versus time for the both cases

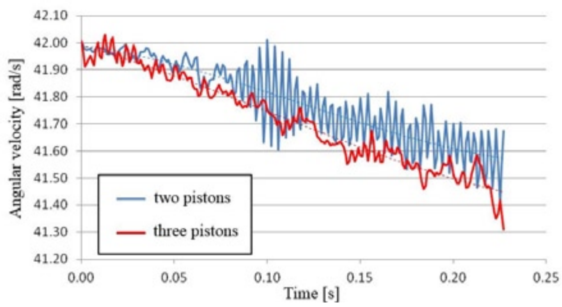

Fig. 8. Comparison graph of angular velocity versus time for the both cases

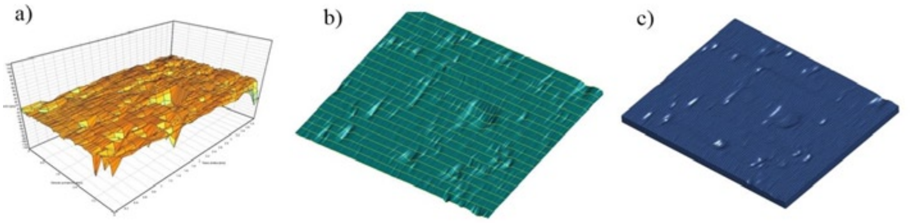

Fig. 9. Subsequent stages of the microscopic model development process; a) profilometer model, b) geometrical model, c) FE model

4. Microscopic model of pad lining

All the processes occurring on the surface lining, disc or drum, have their origin in micro areas of contacting bodies [7]. Therefore, authors decided to develop a numerical model of lining surface, in which surface roughness and the traces of destruction were mapped in the microscopic scale. This decision resulted also from the fact that such approach is difficult to implement in case of a global model of the braking system, where it is virtually impossible to introduce a fine mesh correctly reproducing roughness of a lining surface. The proposed microscopic scale modelling has enabled a more accurate numerical representation of the impact of surface roughness on the phenomena occurring on the surface of the friction lining. Figure 9 shows the successive stages of developin the numerical model of lining in microscopic scale. Whole process is described in more detail in [7].

Developed micro-sample had dimensions of 4 mm x 4 mm x 0.25 mm. It was modelled with 25600 solid hexagonal elements (HEX) with 32805 nodes. Both lining sample and counter sample were given material properties taken from literature. In order to simulate a braking process the linear velocity $v = 19.79$ m/s was applied to all lining sample nodes and pressure $p = 0.587$ MPa (Fig. 10) was applied on its upper surface. Initial and boundary conditions directly corresponded to conditions encountered during the experiment, which have been widely described in [6].

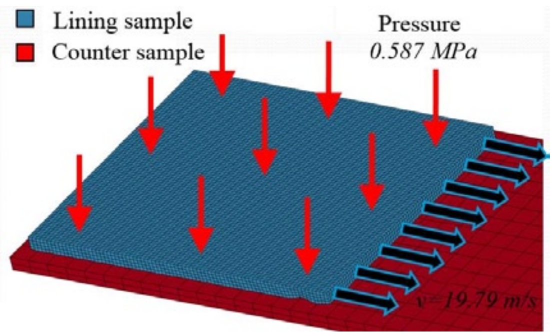

Fig. 10. Initial boundary conditions in the microscopic model [6]

4.1. Analysis results

From the carried out dynamic numerical analyses obtaining temperature distribution for sample as well as for counter-sample was obtained. In Fig. 11 temperature result for a given time $t = 0.15$ s is presented, while changes of temperature as a function of time for the highest peak and entire model of the sample is shown in Fig. 12.

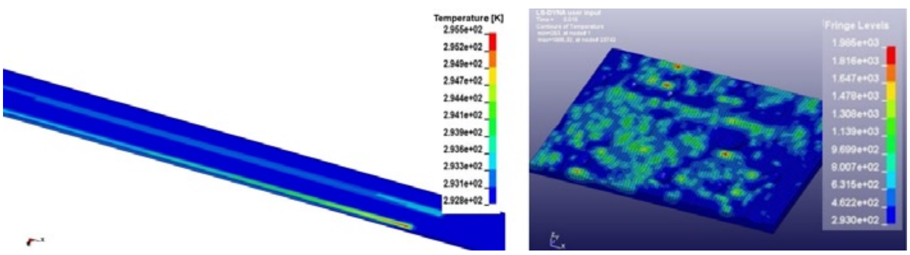

Fig. 11. Temperature distribution on the counter-surface and microscopic lining surface at the time t=0.15s

Foregoing figures show that the maximum temperature is generated on the roughness peaks, which in combination with the conditions occurring during the braking process causes the grinding of these vertices, the propagation of wear and hot spots generation [7]. Taking all above into consideration, authors decided to reproduce the wear process of samples using SPH method in micro-scale.

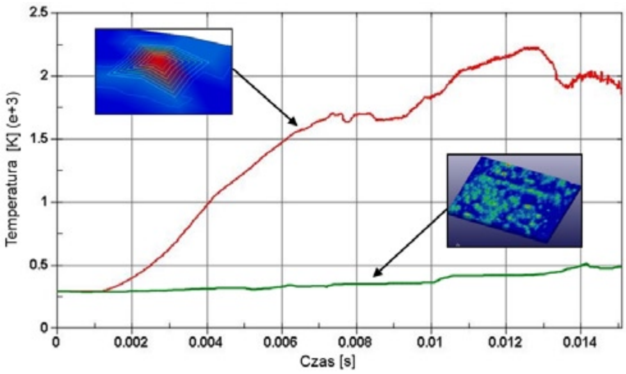

Fig. 12. Temperature versus time graph for the whole microscopic model and for the highest roughness peak

4.2. Modelling of lining wear using SPH

Smooth Particle Hydrodynamics method (SPH) is a meshless method especially useful tin simulation where large deformation of the material takes place, i.e., crash tests and fluid flow. It was developed in order to avoid significant reduction of accuracy generated by finite element mesh at large deformation. The main advantage of the SPH method is the lack of a mesh connecting the nodes.

The basis of this method is interpolation. Distributions of physical parameters replaced by the corresponding estimates at a given interpolation kernel [13] are given by:

$$\prod^k f(x) = \int f(y) W(x-y,h) dy \tag{5}$$

where: W is approximation kernel.

Approximation kernel W has the following form:

$$W(x,h) = \frac{1}{h(x)^d}\theta(x) \tag{6}$$

where: d – number of space dimentions, h – smoothing lenght, defining distance at which particle can interact with other particles.

Finite elements of sample model were replaced by the hydrodynamics particles. Initial and boundary conditions were identical as for the FEM model in a microscopic scale. The one and a major difference between those two lied in the strain based eroding criterion allowing the pieces of material (SPH particles) to detach from the lining model. Figure 13 shows the results of

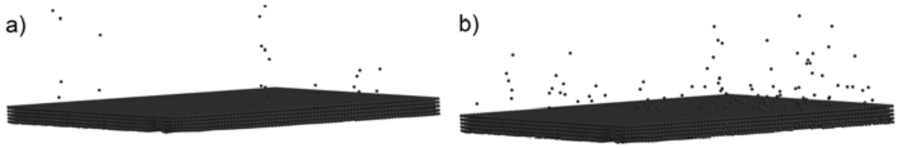

Fig. 13. SPH lining model erosion for the two chosen moment of time, a) t = 0.08 s, b) t = 0.2 s

the numerical analysis involving a process of failure of the sample for two specific moments of time.

Based on obtained results it can be seen, that the proposed implementation of algorithm of linings material failure works properly. It is difficult to clearly determine whether it fully reflect the real mechanics of material damage, but from numerical point of view, the procedure seems to work properly. To have a complete picture of the wear process nature used in the study, its exact characteristics and parameters should be measured and acquired. This would allow for more exact simulation of wear process. Despite the lack of such data, due to the promising effects of the application of meshless methods, authors decided to investigate and develop this approach much more thoroughly. At the moment researches on hybrid models involving a combination of FEM modelling techniques SPH are conducted. In Figures 14 and 15 an initial computational model is presented where pads are modelled using SPH particles whereas the simplified brake disc is modelled using solid elements.

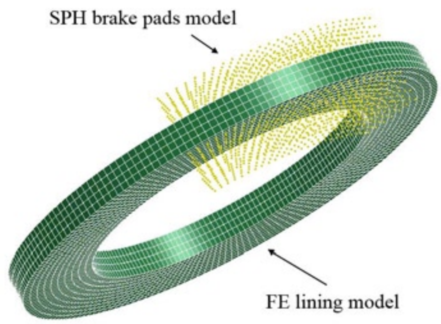

Fig. 14. Hybrid model (SPH+FEM) of the brake system

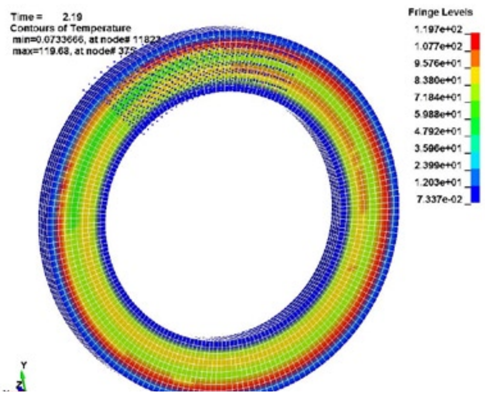

Fig. 15. Hybrid model of the brake system – temperature [K] distribution at the specific moment of time

5. Conclusions

This paper presents the results of the investigations covering a broad phenomenon, which is certainly the braking process. After analysing the results it can be stated, that today, with the help of numerical methods, it is possible to perform detailed analysis of the brake during its operation. The main problems, however, are long calculation runtimes and hardware resources requirements. Also, the need for obtaining the full characteristics of the material as a function of temperature, which have a significant impact on the results, is an additional factor that hinders the use of numerical simulations in the brake design process. Another limitation is associated with FEM discretization, which should be detailed enough to include surface roughness and at the same time should give acceptable calculation time. Therefore, authors decided to conduct separate studies on macroscopic objects (drum brakes and disc) and microscopic (sample friction lining). The simulations results showed that the macroscopic approach allows for conclusions of the general nature of the brake (like the uniformity of pad work or comparative analyses), but is not enough detailed to predict the wear of pads, which – for example – prevents to predict life of the brakes.

The proposed concept of numerical description of the surface layer wear will be further modified and investigated, in order to find optimal parameters for modelling the process of wear. Authors hope that with combining the SPH method and FEM modelling it will be possible to simulate the braking process using numerical model of the whole brake and also to include such details as surface roughness in the calculations. At the moment, performing such analysis using the classical FEM and available hardware is practically impossible.

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