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Thermal and stress-strain state of friction pairs in ventilated disc brakes of lightweight vehicles

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Abstract

The work is dedicated to the thermal behavior and stress-strain state of ventilated disc brakes installed in the lightweight vehicles (scooters, electric bikes, ATVs, etc.) using ANSYS environment in various experiment modes. Modeling of the temperature distribution in the rotor (disc) and the corresponding brake pads is determined taking into account a number of factors and input parameters during the braking operation: the amount of rotation speed, the gap between the pads and the disc, the speed of load application, thermal expansion, etc. Numerical modeling of the transient thermal and the stress fields in the area of contact between the pads and the rotor is carried out by the method of sequential thermostructural connection of the intermediate calculation states of the brake model in the ANSYS Coupled Field Transient environment. For a comprehensive assessment of brake behavior, our research considers two load approaches: constant long-term (20 s) with an influence factor in the form of thermal expansion as a result of contact pair friction; linear load from the pads on the disc with a corresponding increase in pressure up to the moment when the rotation of the system is blocked. Our research presents an assessment of the rotor ventilation channels influence on the nature of the contact spot with the brake pads (open far-field contact, sliding contact, sticking contact, etc.). In addition, it is demonstrated that despite the linear increase in pads pressure on the rotor, the graphs of temperatures, volume (thermal expansion) and stresses are of parabolic character with a disproportionate increase in indicators. Such a result forces us to come to the conclusion that it is not possible to predict the behavior of the brakes based on the analysis during a short period of time of the experiment - conducting long-term analytical studies is extremely important in the case of brakes.

Key words: friction, brake disc, brake pads, thermal load, stress-strain state, heat flow, von Mises stress, contact pressure, thermal expansion

Introduction

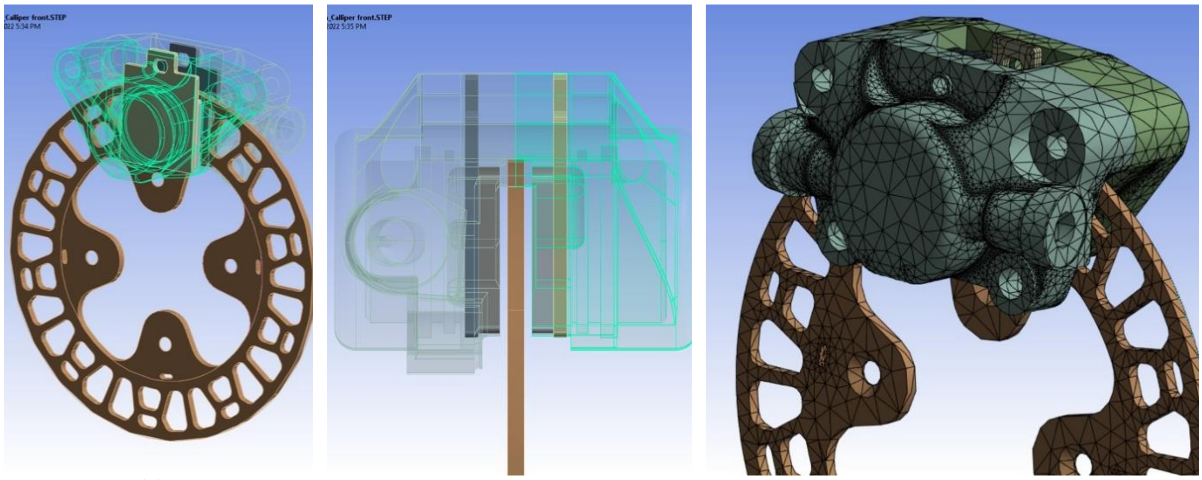
Scientific and technological progress has provided the industry with significant theoretical developments in the field of heat and mass transfer, for example, in spheres such as tribology or thermodynamics, which have developed over several decades with progress in many sectors: nuclear energy, aerospace and aviation, automotive, etc. Modeling of problems related to the phenomenon of heat or mechanical energy transfer in general and through friction pair contacts in particular, is of primary importance in the design of relevant units, for example, disc brakes of vehicles. Many authors raised such topics in their publications as: design and thermal analysis of disc brake for minimizing temperature [1]; effect of cross-drilled hole shape on crack of disc brake rotor [2]; thermal analysis of disc brakes using FEA [3-4]. In fact, it is not only about the development of new models of brakes, but also about the selection of optimal options for systems for existing vehicles, taking into account their class, type and operating conditions. As you know, brakes are a device that creates frictional resistance to move a system element (rotor) to stop further movement, so we can get acquainted with the modeling and analysis of FSAE car disc brake using FEM in [5] and discover the enhancement in design and thermal analysis of disc brake rotor in [6]. Brakes are a mechanism used to reduce the speed or stop the cycle of movement of a vehicle. Long-term use of the brake in a lightweight vehicle (bicycle and motorcycle) causes heating during the braking process [7-9], so that the rotor is deformed (jammed between the pads or breaks) due to high temperature and thermal expansion. Actually, the thermal analysis of disc brake is the topic of publication [7], which could be effectively supplemented by the study on crack initiation at small holes of one-piece brake discs in [9]. In the research [10] authors present the velocity and relative contact size effect on the thermal constriction resistance in sliding solids. The influence of the braking time on the soundness of ventilated disc brake systems is reflected in [11-12]. Topic of investigation of temperature and thermal stress in ventilated disc brake based on 3D thermomechanical coupling model raised in [13] is similar to our work and makes sense to be researched. Our goal is to proceed the analysis of the behavior of the system under conditions of long-term friction at constant pressure with a corresponding increase in temperature and volume of the model, as well as with a variable load in the system (from the hydraulic cylinder), which leads to blocking of the brakes with plastic disc deformations (determination of stresses according to Mises).

The purpose of the work

Formation of a methodology for analytical studies of thermal modes of the lightweight vehicles disc brakes operation as a result of friction pairs contact with variable and constant pressure in the ANSYS Coupled Field Transient software environment. Analysis of the influence on heat dissipation and stress distribution of such factors as: duration of braking, convection in the environment, geometry of the brake disc and pads, system actuation time.

Results of studies under constant load

A disc brake is a system consisting of a brake disc (rotor), brake pads and calipers actuated by a hydraulic cylinder. The brake disc rotates with the wheel and the pads mounted on the brake calipers clamp it to stop or slow the wheel (Fig. 1). Brake pads generate heat through friction, converting kinetic energy into heat to reduce the total kinetic energy of the vehicle. Thus, due to the thermal energy generated during the braking process, the temperature of the disc on the contact part increases and generates fatigue stresses accumulation, causing cracks or plastic deformations that reduce the service life of the disc. Usually, ventilated discs are used to improve the efficiency of heat dissipation, because they have channels for air circulation: the higher the rotational speed of the disc, the higher the centrifugal force, which contributes to the dissipation of heat.



(a) (b) (c)

Fig. 1. Solid disc brakes model: a) isometry; b) the gap between the disc and the pads in the initial state (1.5 mm); c) FE grid of the model

To select the optimal brake system according to the target vehicle, it is advisable to determine the required pressure [10-12] from the brake pads:

$$P = F_d / S\mu, \tag{1}$$

where: P - pressure between the disc and the pad; F_d - force acting on the disc; S - surface of the pad in contact with the disc; μ - coefficient of friction.
The actual value of the pressing force F_d can be found as follows:

$$F_d = {k \cdot (M \frac{v^2}{2})} \Big/ {2 \frac{r}{R} (vt - \frac{1}{2} \left(\frac{v}{t}\right) t^2)}, \tag{2}$$

where: k – load factor – 0.3 (corresponding to 30%); M – mass of the vehicle; r – brake disc radius; R – wheel radius; v – vehicle speed.

We apply the following values to the boundary conditions of the calculation: time of the experiment t = 20 s; coefficient of friction μ = 0.2 c; angular velocity w = 3.5 rad/s, which corresponds to a wheel speed of 200°/s. Wheel is rotated due to the hub with the 4 holes for mounting bolts (Fig. 1a), where are observed the highest meanings of stress (Fig. 2b); the movement of pad Δ is symmetrical and presented in steps (Table 1). The initial gap between pads and disc is 1.5 mm. Starting from 0.3 s and until the end of the experiment Δ=1.501 mm – thus, the full contact between friction pairs is simulated.

Table 1

Brake pads travel during the experiment (20 s)

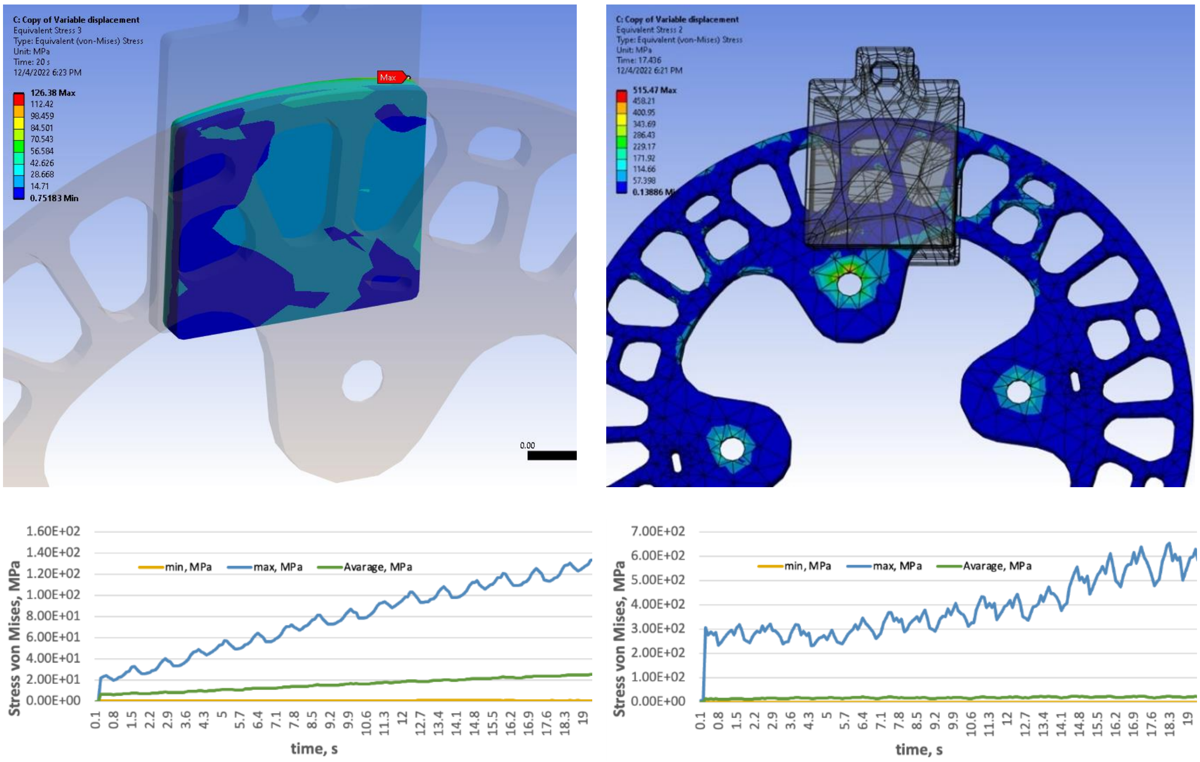
Time moment	0 s	0.1 s	0.2 s	0.3 s	20 s
Δ	0 mm	0.75 mm	1.5 mm	1.501 mm	1.501 mm

The FEM model consists of 101529 elements; applied material is Structural Steel (typical characteristics are embedded in Ansys); the number of time steps is 200 (duration of a step is 0.1 s); the total calculation time on the equipment (2 Intel Xeon processors 24 cores, RAM 48 Gb, NVIDIA GeForce 4Gb video) was 10 hours 42 min.

Let's analyze the stress maps of the brake pads and the ventilated disc (Fig. 2) - as we can see, there is a stress increase tendency while the experiment continues:

- the pad is pressed to the disc in 0.2 s and its stress increases from 8.3 MPa (caused by reactions from the rotational movement in the holes for mounting bolts attaching the disc to the hub) to 223 MPa (when Δ reaches 1.501 mm). Further, as the experiment progresses, the stress increases to a maximum of 647 MPa at a time of 18.2 s. The intermediate state of stress at the moment of time 17.4 s is presented in Fig. 2b (the curve is “max, MPa”). The curves "min, MPa" and "average, MPa" correspond to the minimally loaded locations of the body and the average value of loads for all its locations, respectively.

- the pad stress increases to 22 MPa in the first 0.2 s (the period of pressing against the disc) and then reaches up to 138 MPa at the time of 19.5 s (Fig. 2a).

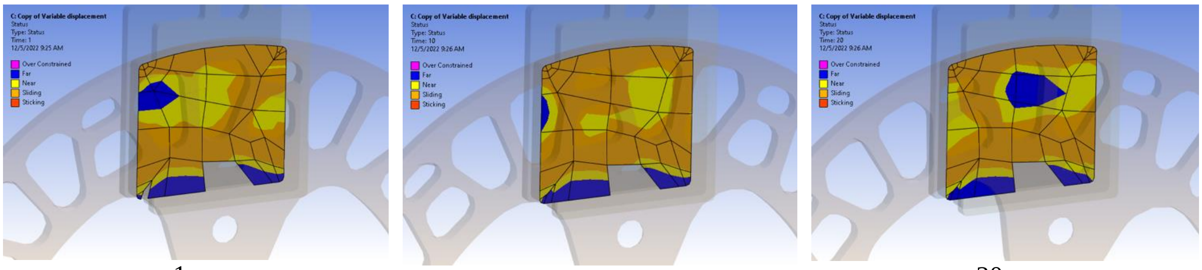


(a) (b)

Fig.2. Stress in the braking system at a constant load: a) brake pad and stress graph over time; b) ventilated disc and stress graph over time

What should be paid attention to: despite the constant value of the displacement of the pads (it is stable and equal 1.501 mm during the entire experiment lasting 20 s), the stress values fluctuate and increase. Why do we observe such processes? Let's consider the answers to both questions sequentially.

1) Stress fluctuations are explained by the uneven structure of the disc itself (holes in the structure for ventilation): as it rotates, the area of contact with the pad is constantly changing, and thus the pressure and stress change as well. Let's visually check the nature of the pad contact with the disc at different moments of time (Fig. 3) - the unevenness of the distribution is dictated by the channels in the disc that affect the contact spot: the position and movement of the contact element determines its condition relative to the target surface associated with it.



1 s 10 s 20 s

Fig.3. Analysis of brake pad contact area at different moments of time

ANSYS monitors each contact element and assigns a status:

- STAT = 0 Open far-field contact (open remote contact) – blue color;
- STAT = 1 Open near-field contact (open near field contact) – yellow color;
- STAT = 2 Sliding contact (sliding contact) – orange color;
- STAT = 3 Sticking contact (sticking contact) – red color.

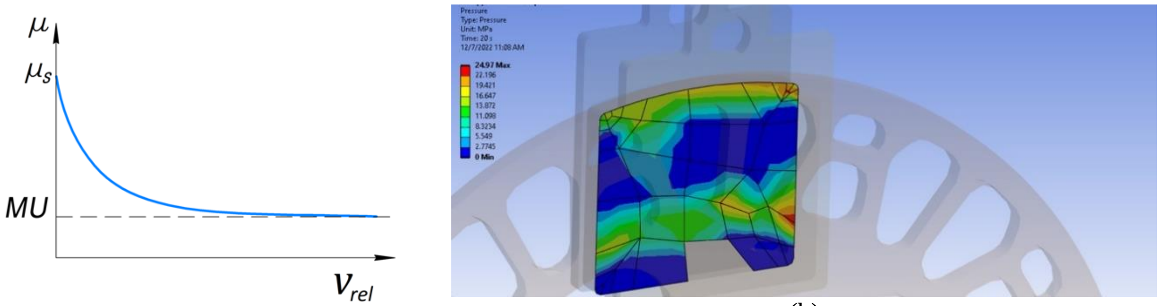
An element is considered to be in close contact if its integration points (Gauss points or nodal points) are within the code-calculated (or user-defined) distance to the corresponding target surface. This distance is called the pinball area. A pinball domain is a circle (in 2-D) or a sphere (in 3-D) centered around a Gauss point.

The friction coefficient may depend on the relative speed of the contacting surfaces. As a rule, the static coefficient of friction is higher than the dynamic one. ANSYS provides the following exponential friction damping model:

$$\mu = MU \cdot (1 + (FACT - 1)\exp(-DC \cdot v_{rel})), \tag{3}$$

where: μ – friction coefficient; MU - dynamic coefficient of friction (using the MP command in Ansys); $FACT$ - the ratio of static to dynamic friction coefficients (the minimum value is set by default 1.0); DC - damping coefficient (by default it is equal to 0 and has the unit of dimension time/length), so time has a certain value in static analysis); v_{rel} - slip velocity calculated by ANSYS. "Friction Decay" shows an exponential decay curve (Fig. 4a), where the static coefficient of friction is defined as:

$$\mu_s = MU \cdot FACT \tag{4}$$



(a) (b)

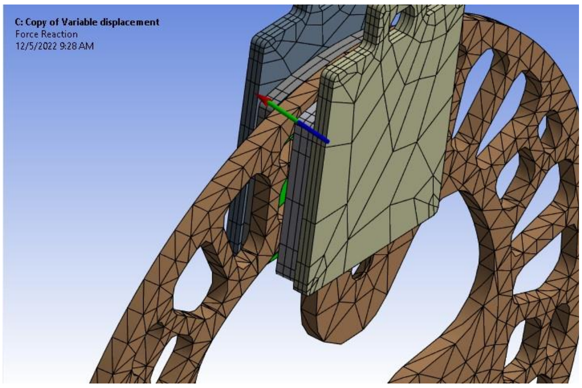
Fig.4. Research of friction: a) exponential curve of friction damping; b) pressure map on the pad surface at the time of 20 s

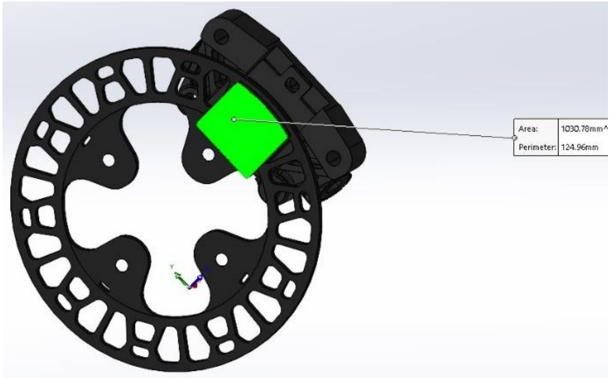
The damping coefficient can be determined if the static and dynamic coefficients of friction and at least one data point are known (μ_1; v_{rel1}). The equation to describe friction damping can be written as follows:

$$DC = -\frac{1}{v_{rel1}} \cdot \ln\left(\frac{\mu_1 - MU}{MU(FACT - 1)}\right) \tag{5}$$

If no damping factor is specified in the simulation process, and FACT is greater than 1.0, then the friction coefficient will suddenly change from static to dynamic value as soon as the contact reaches the sliding state. It should be noted that such behavior is strongly not recommended, since the gap can lead to convergence difficulties when solving the problem [7-8].

2) Why does the value of stresses in the disc and brake pads increase, if they remain stationary and do not increase the external load from the hydraulic cylinder? By the way, what is the maximum pressure value recorded during the experiment (Fig. 5)?



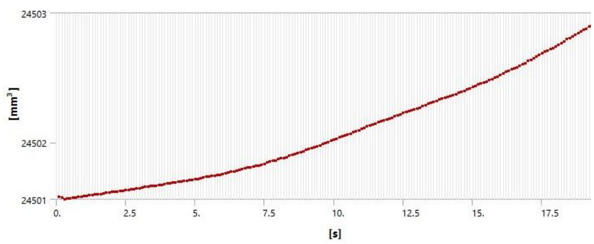


(a) (b)

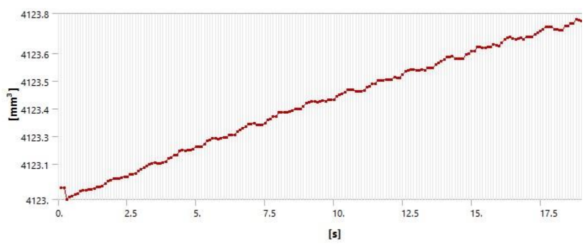
Fig. 5. Determination of pressure in the disc brake system: a) FEM model with load vector; b) determination of the brake pad area (SolidWorks environment)

We have measured the maximum value of the load (Fig. 5a) during the experiment in the Ansys environment: 7968.5 N at the time of 18.2 s. The pad area is 1030.78 mm^2 (Fig. 5b), which corresponds to a pressure of 7.73 MPa. It's possible to observe a similar value on the graph (Fig. 6d - orange color), which shows the average pressure value over the pad area. However, taking into account that the contact area varies, as shown in Fig. 3, and can occupy up to 35-40% of the pad area due to the ventilation holes at certain moments of time, the pressure value increases up to 20 MPa. This is a typical value for disc brakes in automotive and two-wheeled vehicles. Therefore, our experiments with the applied boundary conditions are approaching to the natural tests.

The reason of the stress increase is the thermal expansion of the disc and pads (increase in volume) as a result of heating (Fig. 6a, b) and internal energy growth (Fig. 6c), which leads to a decrease in the gaps between disc and pads with the appropriate pressure rise (Fig. 6d). It should be noted that the increase in the volume of the pad is relatively linear over time, but the disc expands according to a geometric progression - in fact, this already prompts the idea of the feasibility of scientific research on ventilation holes in the structure of the disc, the selection of their optimal configuration, etc.



(a)



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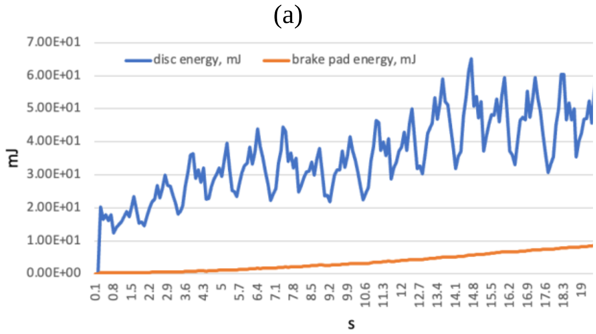
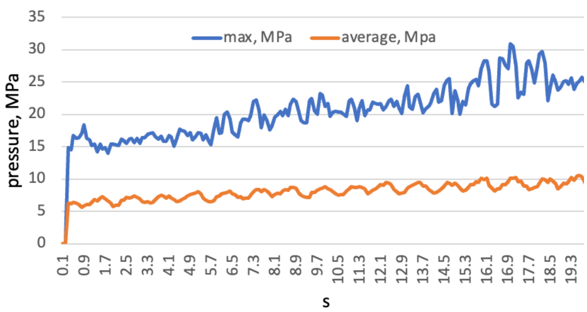


(c)



(d)

Fig.6. Thermal state analysis of disc brakes: a) volume of the disc; b) volume of pads; c) growth of energy over the friction; d) pressure on the pad surface (average in area and maximum in locations)

Fig. 7 shows temperature maps of the disc at certain moments of time. Thus, the value of the disc temperature during the experiment lasting 20 seconds reached 34.87°C. It should be understood that the following boundary conditions were applied as a part of our research: temperature $T(x,y,z)$ = 22°C at time t = 0 s and zero value of convection (please note that the simulation of moving air masses assumes 5 W/m^{2}C in a static position and around 25 W/m^{2}C - in dynamics) to obtain clean results of body heating and heat flux (fig. 8a,c). The value of the pad's temperature reached - 35.04°C, which is shown on the graph of both elements heating (disc and pads) -

Fig. 8b. It's quite exciting to observe how close are both graphs (Fig.8b) – heat transfer from pads to rotors could be visually observed by the temperature equalizing between both units at any time moment.

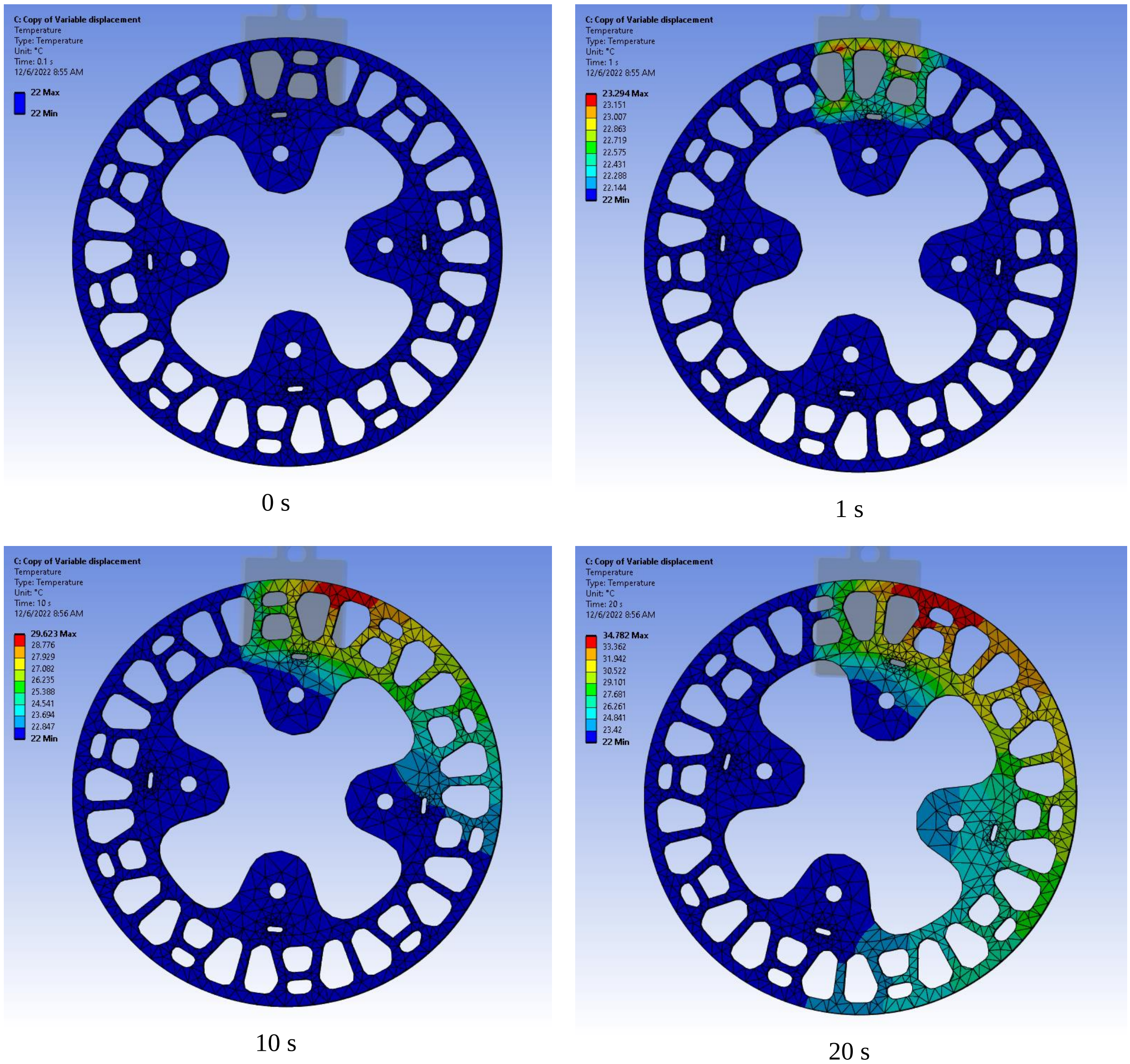


Fig. 7. Temperature maps of the brake disc at different times of the experiment

Let's turn to the theory of the thermal state description of the body - the first law of thermodynamics, which shows on the thermal energy saving [13]:

$$C_p\left(\frac{\partial T}{\partial t}+\{v\}^T\{L\}T\right)+\{L\}^T\{Q\}=p \tag{6}$$

In our calculated case, there is no internal pressure source ($p = 0$), and therefore equation (6) will be written as follows:

$$\rho C_p\left(\frac{\partial T}{\partial t}+\{v\}^T\{L\}T\right)+\{L\}^T\{Q\}=0, \tag{7}$$

where:

$$\{L\}=\begin{Bmatrix}\frac{\partial}{\partial x}\\ \frac{\partial}{\partial y}\\ \frac{\partial}{\partial z}\end{Bmatrix}, \quad \{v\}=\begin{Bmatrix}v_x\\ v_y\\ v_z\end{Bmatrix}, \tag{8}$$

where: $\{L\}$ – vector operator, $\{v\}$ – vector speed of the vehicle.
Let's write Fourier's law (7) in matrix form:

$$\{Q\} = -[K]\{L\}T, \tag{9}$$

where: $[K]$ – matrix with the corresponding coefficients K_{xx}, K_{yy}, K_{zz} by axles X, Y, Z, which are equal in all directions for isotropic materials: $K_{xx} = K_{yy} = K_{zz}$ [13]:

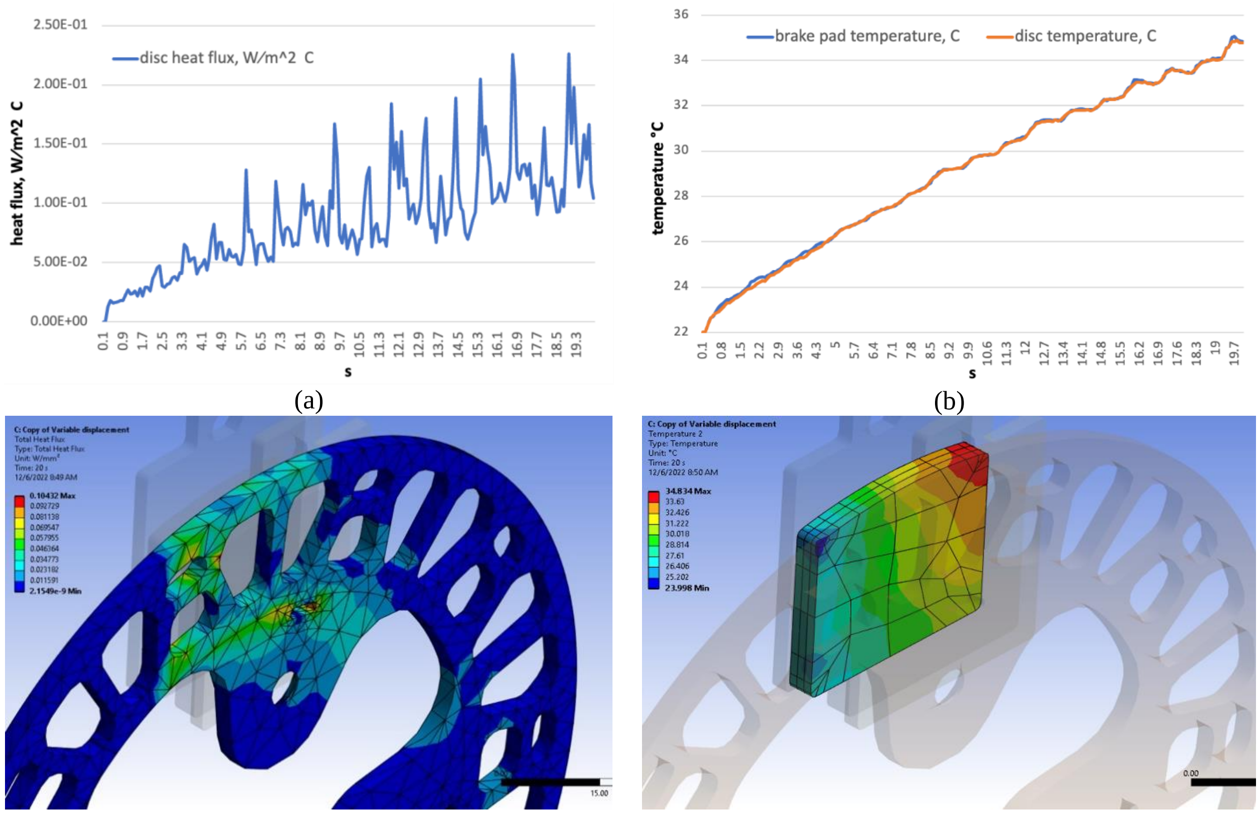
$$[K] = \begin{bmatrix} K_{xx} & 0 & 0 \\ 0 & K_{yy} & 0 \\ 0 & 0 & K_{zz} \end{bmatrix} \tag{10}$$

When combining equations (7) and (9), we get the following expression:

$$\rho C_p \left(\frac{\partial T}{\partial t} + \{v\}^T \{L\} T \right) + \{L\}^T ([K]\{L\}T) \tag{11}$$

Let's rewrite (11) in the following form:

$$\rho C_p \left(\frac{\partial T}{\partial t} + v_x \frac{\partial T}{\partial t} + v_y \frac{\partial T}{\partial t} + v_z \frac{\partial T}{\partial t} \right) = \frac{\partial}{\partial x}\left(k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y}\left(k_y \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z}\left(k_z \frac{\partial T}{\partial z} \right) \tag{12}$$



(a) (b)

(c) (d)

Fig. 8. Heat load of brakes: a) heat flow of the disc; b) temperature of the disc and pad; c) heat flow of the disc at 20 s; d) temperature of the pad at 20 s

In general, the typical boundary conditions of thermal calculation can be attributed to [13]:

- surface temperature: $SURF_T$: $T = T^*$;
- thermal dissipation on the surface: $SURF_Q$: $\{Q\}^T\{n\} = -Q^*$;
- convection on the surface: $SURF_C$: $\{Q\}^T\{n\} = h(T_p - T_f)$,

where: $SURF_T$, $SURF_Q$, $SURF_C$ – surface temperature, flow and convection; T^* - the temperature given at the surface; Q^* - the heat flux given at the surface; T_p – surface body temperature; T_f - environment temperature; h - coefficient of convective heat transfer.

In turn, the thermal expansion presented in the graph (Fig. 6a) can be described by the following conditions, which are relevant for the behavior of solid bodies:

- thermal coefficient of volumetric expansion (measured in inverse degrees Kelvin, K^{-1}):

$$\alpha = \frac{1}{V}\left(\frac{\partial V}{\partial t} \right) \tag{13}$$

If the volume expansion coefficient changes significantly with temperature, then the equations must be integrated:

$$\frac{\Delta V}{V} = \int_{T_0}^{T_0+50} \alpha(T)dT \tag{14}$$

- thermal expansion of the area of a solid body:

$$\Delta S = 2\alpha S_1 \Delta t, \tag{15}$$

where: T_0 - initial temperature; ΔS – area change (for example, brake pads or disc); S_1 – starting area; Δt – temperature change.

Results of studies under variable load

We have previously considered the behavior of the brake system consisting of a disc and pads under a constant external load (pressure) from hydraulic cylinder on them: the static position of the pads during the entire experiment lasting 20 s and constant travel (Δ=1.501 mm). How will the system show itself if we increase the pressure in the hydraulic cylinder and set the pads movement according to the linear law (Table 2)?

Table 2

Dynamics of brake pads movement during the experiment (20 s)

Moment of time	0 s	0.1 s	0.2 s	0.3 s	1 s	5 s	10 s
Δ	0 mm	0.75 mm	1.5 mm	1.501 mm	1.508 mm	1.548 mm	1.598 mm

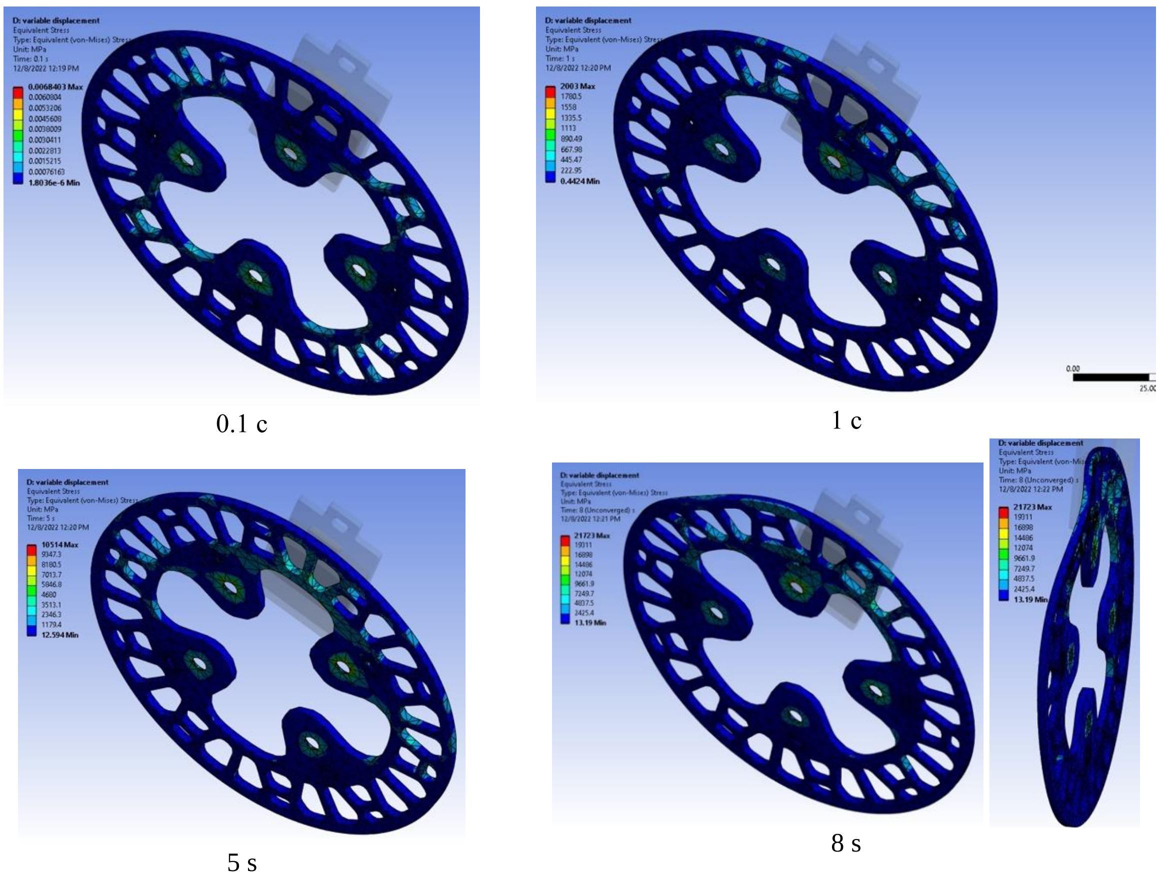
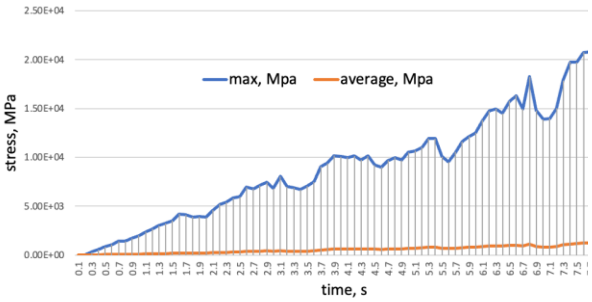
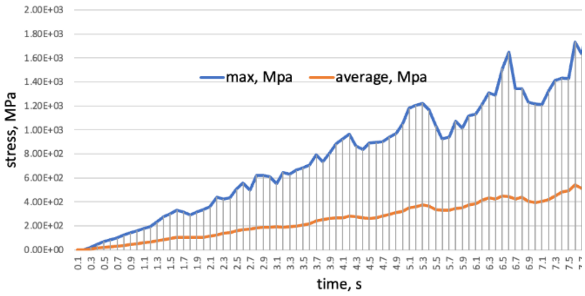


Fig. 9. Stress-deformable state of the brake rotor

This setting of boundary conditions leads to jamming of the brakes, because two factors come into play: the increase in pressure from the side of the pads; the thermal expansion of pads together with the disc. The stress map at the critical moment is presented in Fig. 9 - the plastic deformation of the disc is visually observed as a result of an attempt at inertial scrolling. As you can see, the experiment stopped at the 8th second - the further process is a static state of the system and does not require an assessment of its behavior (the disc cools down to the initial 22°C). Rotor and pad stress trends are demonstrated on Fig.10 – fluctuations on the graph mean the ventilation channels influence on the stress meaning. Additional plasticity of the material during the long-term friction is provided by an increase in temperature as a result of pressure growth - the disc model received 32-37% higher temperature values compared to the previous experiment (unchanged travel of pads Δ=1.501 mm). Such results lead to the opinion of the necessity to arrange not only the structural optimization of the disc (ventilation channels), but also force to think about the relevance of using heat-resistant materials for the production of brakes: ceramics, which is a standard point in the premium segment of cars, sports cars, etc.



(a)



(b)

Fig. 10. Brake stress under the variable load: a) disc stress graph; b) pads stress graph

Conclusions

1. The results of thermal behavior and stress-strain state of ventilated disc brakes presented in the work using the ANSYS Coupled Field Transient calculation environment are of a practical nature not only from the point of view of designing new vehicles with the appropriate selection of the optimal brake configuration for them, but also the optimization of existing structures. The research provides such valuable data as: temperature distribution along the rotor and pads during the friction process; heat dissipation, cooling and ventilation activities; selection of suitable materials for the production of friction pairs; creating an optimal configuration of the disc ventilation holes; determination of the required pressure in the hydraulic cylinders, taking into account the mass of the vehicle and the conditions of its operation (speed, convection of the medium, etc.).

2. The results obtained in the conditions of a pads static position during the entire experiment lasting 20 s with their constant travel (Δ=1.501 mm) allow us to quantitatively assess the influence of thermal expansion on the key performance indicators of the brakes as a result of friction (heating from 22°C to 35.04°C). This approach provides an understanding of the necessity to remove heat and ventilate the brakes, because the trends presented in the graphs indicate an exponential rather than a non-linear increase of the disc volume during heating, and suggest the inevitability of jamming / burning of the brakes (depending on the degree of vehicle movement inertia) with prolonged contact of friction pairs.

3. The use of the ANSYS Coupled Field environment in conjunction with the boundary conditions proposed in the work allows you to form your own effective brake modeling methods, which is especially useful in the conditions of small design studios and workshops, which, in fact, are often involved in the production and design of lightweight vehicles: motorcycles, e-bikes, ATVs, scooters, buggies, etc.

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Голенко К.Е., Диха О.В., Падгурскас Ю., Бабак О.П. Термічний та напружено-деформований стан пар тертя вентильованих дискових гальм легких транспортних засобів

Робота представляє собою дослідження теплової поведінки та напружено-деформованого стану вентильованих дискових гальм легких транспортних засобів (скутерів, електробайків, квадроциклів, тощо) за допомогою розрахункового середовища ANSYS в різних режимах випробувань. Моделювання розподілу температури в роторі (диску) і відповідних гальмівних колодках визначається з урахуванням ряду факторів і вхідних параметрів під час операції гальмування: величини швидкості обертання, зазору між колодками і диском, швидкості прикладення навантаження, теплового розширення та ін. Чисельне моделювання перехідного теплового поля та поля напружень в області контакту колодок та диску здійснюється методом послідовного термоструктурного зв’язку проміжних розрахункових станів моделі гальм у середовищі ANSYS Coupled Field Transient. Для комплексної оцінки поведінки гальм в публікації розглядаються два підходи навантажень: стале (тривалістю 20 с) з фактором впливу у вигляді температурного розширення в результаті тертя контактних пар; лінійне навантаження з боку колодок на диск з відповідним зростанням тиску аж до моменту блокування обертання системи. Також дослідження включає в себе оцінку впливу вентиляційних каналів ротора на характер плями контакту з гальмівними колодками (відкритий дальній контакт, контакт ковзання, залипання тощо). Крім того, показано, що незважаючи на лінійне зростання тиску колодок на ротор, графіки температур, об’єму (теплового розширення) і напружень мають параболічний характер із непропорційним зростанням показників. Такий результат змушує прийти до висновку, що неможливо передбачити поведінку гальм на основі аналізу короткого проміжку часу експерименту - проведення довгострокових аналітичних досліджень є надзвичайно важливим у випадку гальм.

Ключові слова: тертя, гальмівний диск, гальмівні колодки, теплове навантаження, напружено-деформований стан, тепловий потік, напруження фон Мізеса, контактний тиск, теплове розширення.