Computer simulation of heat transfer processes in aircraft landing gear braking mechanisms

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Abstract. Landing gear brakes – multi-disc brakes, as the responsible mechanism of the aircraft, must have the lowest possible weight and dimensions, high performance, increased service life and reduced life cycle cost. Solving these problems will reduce the design and development cycle of multi-disc chassis brakes. To perform the calculations, a computer mathematical model based on the use of the finite element method is proposed, replacing the classical version of the calculation based on currently used mathematical models, taking into account the heating of the aircraft landing gear braking mechanism, which will increase the speed, accuracy of calculations and reduce their labor intensity. We have developed a computer mathematical model of heat transfer processes occurring in the friction elements of multi-disc landing gear braking systems at the second stage of landing using the COMSOL Multiphysics program. Based on the calculations performed using this model, it is possible to take into account the influence of various system parameters such as the number of friction disks, their diameter and thickness, the medium in the interstitial space, the size of the solver grid on the maximum heating temperature of the brake mechanism, the cooling rate, the calculation time and the calculation error. On the basis of the obtained experimental data, we can conclude about the preferable construction of the brake mechanism. For example, it is possible to produce friction plates, steel with application to the surface of the composite friction material, allowing it to maintain the necessary friction and thus reduce the dimensions and maximum temperature, wet-disc brakes. The results of this work can be applied not only in the aviation industry, but can also be used to calculate the mechanisms of multi-disc oil-loaded brakes, which are used in agricultural, forestry, construction, quarry and other special equipment. In future work, the developed model will allow carrying out calculations taking into account the changes of material properties depending on temperature, refining the geometry of the brake mechanism, taking into account periods of reverse thrust engines and aerodynamic brakes.

1. Introduction
The design of the chassis of the modern mainline aircraft is subject to the mandatory requirements imposed by the aviation regulations AR25 [1] (paragraph 25.735. "Brakes and braking systems"), which are fundamental when creating it and are the main source data for its mathematical modeling.

Disc brakes of the aircraft landing gear are the most important systems that ensure the safe completion of the flight and are a complex set of mechanical devices connected by the necessary connections [2-7]. This complex should allow you to optimize the braking process in conditions of changing a large number of different parameters. Consideration of the aircraft braking process during the post-landing run is fundamental in the design, development, and improvement of brakes and braking...
systems. Based on the analysis of physical processes occurring at the time of wheel braking, it is possible to determine the main design parameters of the friction discs of brakes, increase their energy consumption, which will reduce the braking distance, braking time, etc.

In the design of the landing gear wheels of the modern aircraft, disc brakes are used to reduce the length of the run when landing, which consume up to 95% of the kinetic energy [2]. Rotating and friction discs of such brakes must be made of materials with increased friction properties to increase energy consumption. The temperature on the surface of the friction elements during heavy braking can exceed 1000°C, which imposes restrictions on the operating modes of the aircraft. Therefore, the design and selection of disk materials is a responsible task, because they must provide heat removal from the friction surfaces, which would not lead to overheating of the entire structure during prolonged braking.

Mathematical modeling aims to provide opportunities to study the influence of various parameters of the chassis design on the braking process, the choice of its rational design. And it minimizes the braking distance on the runway when designing and certifying the parts for the compliance with AP25 standards during normal operation and design failures.

A model was developed, considering the processes of heat transfer in the friction elements of disc brakes at the second stage of landing.

The work was performed in the universal numerical simulation environment COMSOL Multiphysics, designed for modeling and calculating scientific and engineering problems based on partial differential equations (PDE) by the finite element method (FEM). PDE coefficients are set as physical properties and conditions, and their conversion to coefficients of mathematical equations occurs automatically. This program also supports loading the already created construction geometry from all common programs, such as SIEMENS NX, CATIA, SolidWorks, and AutoCAD.

The presented mathematical model allows studying the braking process when various parameters of its elements change within a wide range, as well as the parameters of the aircraft itself, the runway condition, etc.

2. Analysis of current research and publications

To describe the processes of heat transfer at the interface of solids, we can derive an equation from the localized heat balance equation written in the form for a spatial frame:

$$\rho \frac{\partial E}{\partial t} + \rho \dot{u} \nabla E + \nabla (q + q_r) = \sigma: D + Q.$$  

(1)

It shows that changes in internal energy over time are balanced by internal energy convection, thermal conductivity, radiation, mechanical stress dissipation, and additional bulk heat sources. In the entry for a solid, the right-hand side is thermoelastic damping and accounts for thermoelastic effects in solids [8].

By means of transformations, we can derive the equation we are looking for that describes the processes of heat transfer at the interface of solids:

$$\rho C_p \left( \frac{\partial T}{\partial t} + \dot{u}_{trans} \nabla T \right) + \nabla (q + q_r) = -\alpha T \frac{\partial S}{\partial t} + Q,$$

or

$$\rho C_p \frac{\partial T}{\partial t} + \rho C_p \dot{u}_{trans} \nabla T + \nabla q = Q_{ted} + Q,$$

(2)

(3)

where \(\rho\) – density [kg/m³]; \(C_p\) – specific heat at constant pressure [J/(kg·K)]; \(T\) – absolute temperature [K]; \(\dot{u}_{trans}\) – translational velocity vector [m/s]; \(q = -k \nabla T\) – heat flow due to the conduction effects [W/m²]; \(k\) – thermal conductivity [W/(m·K)]; \(q_r\) – heat flow due to radiation effects [W/m²]; \(\alpha\) – coefficient of thermal expansion [1/K]; \(S\) – second Piol–Kirchhoff stress tensor [Pa]; \(Q\) – constant that characterizes additional heat sources [W/m³].

Thermoelastic damping that takes into account thermoelastic effects in solids:

$$-\alpha T \frac{\partial S}{\partial t} = Q_{ted}.$$  

(4)

Or, by default, the boundary condition for all heat transfer interfaces is written as:

$$-nq = 0.$$  

(5)
This boundary condition means that there is no heat flow across the boundary, i.e. the temperature gradient across the boundary is zero. For this to be true, the temperature on one side of the border must be equal to the temperature on the other side. Since there is no temperature difference at the border, heat cannot be transferred through it. As a default boundary condition, it can only be applied to external borders; but when added manually, it can also be applied to internal borders.

In our model, we do not consider the landing gear braking mechanism as a thermally insulated system, but due to the presence of air blowing the brake, with a speed in the first approximation equal to the speed of the aircraft, heat exchange with the environment occurs due to external forced convection. To do this, we redefined the boundary condition for all heat transfer interfaces and wrote it as:

\[
-nq = h(T_{ext} - T)
\]

\[
h = \begin{cases} 
2k0.3387\sqrt{Pr}/Re_L & \text{if } Re_L \leq 5 \cdot 10^5 \\
2kL \left[ 1 + \frac{0.0468}{Pr} \right]^{\frac{1}{2}} & \text{otherwise}
\end{cases}
\]

where \(T_{ext}\) – external temperature [K]; \(Pr = \frac{\mu C_p}{k}\) – Prandtl number; \(Re_L = \frac{\rho u d}{\mu}\) – Reynolds number; \(k\) – air thermal conductivity [W/(m·K)]; \(L\) – characteristic size, in our case – the radius of the brake disc [m]; \(\mu\) – dynamic viscosity [Pa·s]; \(\vartheta\) – speed of the plane [m/s]; \(C_p\) – specific heat at constant pressure [J/(kg·K)]; \(\rho\) – air density [kg/m³]; \(h\) – average heat transfer coefficient for the plate under external forced convection.

In addition to forced convective heat exchange, it is necessary to take into account that bodies whose temperature exceeds the environment have heat exchange by radiation. To account for this in our mathematical model, we redefined the boundary conditions for all external surfaces and wrote it as:

\[
-nq = \varepsilon \sigma (T_{amb}^4 - T^4)
\]

where \(T_{amb}\) – ambient temperature; \(\sigma = 5.670367(13) \cdot 10^{-18}\) – Stefan–Boltzmann constant [W/(m²·K⁴)]; \(\varepsilon\) – degree of blackness of the body.

It is worth noting that redefining the boundary conditions for the external boundaries of the body under study in order to account for heat exchange caused by the effects of forced convection and radiation do not contradict each other and the presence of one condition does not cancel the other. That is, the presence of convective heat exchange does not exclude the presence of heat exchange by radiation.

When the disks of the aircraft landing gear brake system are brought into contact, due to their design, i.e. the alternation of fixed friction disks that play the role of brake pads with rotating disks that are analogs of the rotor, the kinetic energy of the aircraft movement is converted into thermal energy of friction. To take into account the presence of thermal contact, i.e. heat exchange between disks, in our mathematical model, we need to assign boundary conditions for the contact surfaces in the form:

\[
\begin{align*}
-n_d(-k_d \nabla T_d) &= -h(T_u - T_d) + rQ_b \\
-n_u(-k_u \nabla T_u) &= -h(T_u - T_d) + (1-r)Q_u
\end{align*}
\]

where \(u\) – index denoting the upper side of the thermal contact surface; \(d\) – index denoting the lower side of the thermal contact surface.

These equations characterize the dependence of heat fluxes at the upper and lower boundaries on the temperature difference, and also have a correlation estimate of the joint conductivity on two contacting surfaces:

\[
h = h_u + h_g + h_r.
\]

The joint conductivity \(h\) consists of three parts: the conductivity of the narrowing \(h_u\) from the contact spots, the conductivity of the gap \(h_g\) due to the presence of an external environment (air) in the interstitial space, and the radiation conductivity \(h_r\), since at the microscopic level, the contact is made
with a finite number of points.

Microscopic surface asperities (irregularities) are characterized by their average height $\sigma_{\text{asp},u}$ and $\sigma_{\text{asp},d}$ and the average slope $m_{\text{asp},u}$ and $m_{\text{asp},d}$ for the upper and lower surfaces, respectively. To make further calculations, it is necessary to introduce the concepts of root-mean-square values $\sigma_{\text{asp}}$ and $m_{\text{asp}}$, which can be calculated using the following formulas, respectively:

$$\sigma_{\text{asp}} = \sqrt{\frac{\sigma_{\text{asp},u}^2 + \sigma_{\text{asp},d}^2}{2}}$$

$$m_{\text{asp}} = \sqrt{\frac{m_{\text{asp},u}^2 + m_{\text{asp},d}^2}{2}}$$

The constriction conductivity was calculated using the Cooper–Mikich–Jovanovich correlation, which is valid for isotropic rough surfaces and was formulated using a model that assumes plastic deformation of surface roughnesses [9]. However, this model does not calculate or store plastic deformations of asperites. This means that, despite the fact that plastic deformation of asperites (irregularities) is assumed, this contact model has no memory. For example, if the load is applied twice, the thermal contact is identical in both cases. The Cooper–Miki–Jovanovich correlation says that the constriction conductivity depends on the root-mean-square values of asperites and the pressure load at the contact point:

$$h_c = 1.25k_{\text{contact}} m_{\text{asp}} \left( \frac{p}{H_c} \right)^{0.95}$$

where $H_c$ — microhardness of the softer material; $p$ — contact pressure; $k_{\text{contact}} = \frac{2k_u k_d}{k_u + k_d}$ — average harmonic value of contact conductivities.

If $k_u$ or $k_d$ (contact conductivities of the upper and lower surfaces, respectively) is not isotropic, they are replaced with their normal conductivities.

The relative pressure $p/H_c$ can be estimated either by specifying the $H_c$ value directly or by using the following relations:

$$\frac{p}{H_c} = \left( \frac{p}{c_1 \left( 1.62 \sigma_{\text{asp}} m_{\text{asp}} \right)^{c_2}} \right)^{(1+0.071c_2)^{-1}}$$

The coefficients $c_1$ and $c_2$ are the Vickers correlation coefficient and the size index, respectively, and $\sigma_0$ is 1 µm. For materials with Brinell hardness between 1.3 and 7.6 GPa the Vickers correlation coefficients can be calculated from the following relations:

$$\frac{c_1}{H_0} = 4 - 5.77 \frac{H_B}{H_0} + 4 \left( \frac{H_B}{H_0} \right)^2 - 0.61 \left( \frac{H_B}{H_0} \right)^3$$

$$c_2 = -0.37 + 0.422 \frac{H_B}{c_1}$$

where $H_B$ — Brinell hardness; $H_0 = 3.178$ GPa.

It is worth noting that if we were to calculate the rough surface and assume the presence of elastic deformations of surface roughnesses, then it would be necessary to use the Mikich correlation to determine the constriction conductivity:

$$h_c = 1.54k_{\text{contact}} m_{\text{asp}} \left( \frac{\sqrt{2}p}{mE_{\text{contact}}} \right)^{0.94}$$

Here $E_{\text{contact}}$ is an effective young’s modulus for a contact satisfying the relation:

$$\frac{1}{E_{\text{contact}}} = \frac{1 - v_u^2}{E_u} + \frac{1 - v_d^2}{E_d}$$
where \( E_u \) and \( E_d \) – Young’s modules of two contacting surfaces; \( \nu_u \) and \( \nu_d \) – Poisson coefficients.

The gap conductivity must be taken into account if the medium (liquid) in the interstitial space has a high temperature conductivity or there is a large contact pressure. The correlation of the gas gap of parallel plates implies that the liquid in the interstitial space is a gas (air) and defines \( h_g \) as:

\[
h_g = \frac{k_g}{Y + M_g}
\]

where \( k_g \) – gas conductivity; \( Y \) – average thickness of the separation; \( M_g \) – gas parameter, equal to:

\[
M_g = \alpha \beta \Lambda, \quad \Lambda = \frac{k_B T_g}{\sqrt{2\pi D^2 p_g}} \tag{20,21}
\]

where \( \alpha \) – local thermal contact parameter; \( \beta \) – gas parameter (1.7 for air); \( \Lambda \) – average path length of a gas molecule; \( k_B \) – Boltzmann’s constant; \( D \) – average diameter of the gas particle; \( p_g \) – gas pressure; \( T_g \) – temperature gap is equal to:

\[
T_g = \frac{T_u + T_d}{2}. \tag{22}
\]

The average separation thickness \( Y \) is a function of the contact pressure \( p \). For small values of \( p \) (about 0 Pa) \( Y \) goes to infinity, because no contact occurs. For large values of \( p \) is greater than \( H_0/2 \) in the Cooper–Mikich–Jovanovich model and greater than \( H_0/4 \) in the elastic Mikich model – \( Y \) decreases to zero, this means that the contact is considered perfect.

At high temperatures above 600 degrees Celsius, radiation conductivity must be taken into account. The gray-diffusion parallel plate model defines \( h_c \) as:

\[
h_c = \frac{\epsilon_u \epsilon_d}{\epsilon_u + \epsilon_d - \epsilon_u \epsilon_d} \sigma(T_u^3 + T_u^2 T_d + T_u T_d^2 + T_d^3) \tag{23}
\]

or

\[
h_r(T_u - T_d) = \frac{\epsilon_u \epsilon_d}{\epsilon_u + \epsilon_d - \epsilon_u \epsilon_d} \sigma(T_u^4 - T_d^4) \tag{24}
\]

\[
h_r(T_d - T_u) = \frac{\epsilon_u \epsilon_d}{\epsilon_u + \epsilon_d - \epsilon_u \epsilon_d} \sigma(T_d^4 - T_u^4). \tag{25}
\]

To describe thermal friction in our mathematical model, we divided the heat of friction \( Q_b \) into \( rQ_b \) and \( (1 - r)Q_b \) in the heat contact zone. If both bodies are identical, then \( r \) and \( (1 - r) \) are 0.5, so that the heat of friction is evenly distributed between the two surfaces. In general, when both bodies are made of different materials, the partition coefficient may not be equal to 0.5. the value of \( r \) and \( (1 - r) \) is determined from the Sharron ratio:

\[
r = \frac{1}{1 + \xi_d}, \quad \xi_d = \left(\frac{\rho_u C_p u k_u}{\rho_d C_p d k_d}\right)^{0.5} \tag{26,27}
\]

\[
(1 - r) = \frac{1}{1 + \xi_u}, \quad \xi_u = \left(\frac{\rho_d C_p d k_d}{\rho_u C_p u k_u}\right)^{0.5}. \tag{28,29}
\]

If \( k_u \) or \( k_d \) (contact conductivities of the upper and lower surfaces, respectively) are anisotropic, they are replaced with their normal conductivity values.

3. Mathematical modelling
To test the performance of the mathematical model, we created a simplified geometry of the brake discs of the main landing gear struts in the SIEMENS NX 10 program. After that, we loaded it into the COMSOL Multiphysic program, pre-configuring the 3D space dimension, physics-thermal conductivity in solids, and the time-dependent solver. Next, we set the following properties of materials for rotating and friction disks: density, thermal conductivity, heat capacity at constant pressure, and the degree of blackness of the body. To describe the model, it is necessary to specify the following system parameters: the speed of the aircraft to braking, acceleration, deceleration, radius of wheel landing gear, the weight of the aircraft on landing, the ambient temperature, time of braking and the friction disks. The next step is to set the speed functions of the aircraft, considering the deceleration to be equidistant, and the speed
of the aircraft before and after deceleration to be constant. At the same time, we set the existence of the second derivative and the size of the smoothing zone to 0.2. This is necessary for the existence of the acceleration function, which we constructed as a derivative of the velocity function.

After that, we set the motion function of the rotating brake discs of the landing gear, redefined the boundary conditions by setting the microhardness, pressure in the contact zone, and heat release intensity, taking into account that not all the kinetic energy of the aircraft's movement is absorbed by the landing gear braking system. The last step in setting up a mathematical model is to set a grid, given that the smaller the grid elements, the more accurate the calculations are, but the more time they require. For example, all other things being equal, the solution time for the coarser grid sizes was 3 minutes, and for the extremely fine sizes it was 4 hours, 17 minutes, and 5 seconds.

Figure 1. Automatically generated grids with settings: a – extremaly fine; b – coarser.

Figure 2. Graphs of changes in the maximum heating temperature over time depending on the solver grid settings

4. Results and Discussion
The proposed model considers the processes of heat transfer in the friction brakes at the second stage of aircraft landing. Also, by changing various parameters, we qualitatively tracked their impact on the
system [10, 11]. Analyzing the obtained graphs of changes in the maximum temperature from time to time of the disks of the braking system of the aircraft landing gear, we can draw the following conclusions.

When choosing a grid solver, we need to find a compromise between the desired accuracy of calculations and the available time that we can or want to spend on calculations.

It is quite clear that as the number of friction disks increases, the maximum heating temperature decreases, but with each new disk added, the effect that it has on the overall decrease in the heating temperature becomes smaller. Therefore, when designing, it is necessary to find a balance between the allowed maximum heating temperature and the dimensions/weight of the aircraft chassis.

Reducing the diameter of the brake discs increased the maximum temperature by 49.7%, while increasing the diameter by 20% reduced the maximum temperature by only 27.4%. Therefore, it is not always effective to go by increasing the diameter of the brake discs to reduce the temperature – the more we increase the diameter, the more the dimensions/weights of the chassis increase and the less the maximum temperature decreases. You need to find a balance between these characteristics.

When the thickness of the friction disks is reduced 2 times, the maximum temperature value increases slightly, but when the thickness of the movable disks is reduced by the same number of times, the

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**Figure 3.** Graphs: a – changes in the maximum heating temperature over time depending on the number of friction discs; b – dependences of the maximum heating temperature on the number of friction discs

**Figure 4.** Graphs of changes in the maximum heating temperature of the chassis brake system disks depending on: a – diameter of the disks; b – thickness of the movable and friction disks
maximum temperature increases significantly (Fig. 2). Due to the fact that mobile disks are made of steel heat capacity and thermal conductivity at constant pressure which is much higher than that of cast iron, from which the friction discs are made. If you use composite materials, replacing cast-iron discs with steel ones that will be coated with friction composite material, we can reduce the overall weight of the structure and reduce the maximum heating temperature of the brake discs.

The environment in which the brake mechanism is located has a significant impact on the maximum heating temperature and cooling rate. Provided that all other parameters are constant, and only the environmental parameters change, the maximum heating temperature of the brake mechanism in water will be lower, and cooling will go faster. When braking in transformer oil, the maximum temperature is also lower than that in air, and cooling is faster, but still not comparable to the indicators in water. This has found application, for example, in motor graders. In their balancing cart, multi-disc oil-loaded brakes are installed on the shaft connecting the driven gear of the on-board gearbox to the wheel hub.

5. Conclusion
Our mathematical model allows us to optimize the dimensions and weight of the braking mechanism, select materials so that when braking, the maximum heating temperature is in the range of acceptable values, which prevents rapid wear of the structure and increases the life cycle of the aircraft.

Further refinement of our model will be due to the fact that part of the kinetic energy at the landing stage of the aircraft is absorbed, for example, by the engine thrust reversal or brake pads, and, as a result, part of the kinetic energy absorbed by the brake is not constant and changes during braking. When taking into account the period of operation of the engine thrust reverse and brake flaps, it is necessary to specify the functions of acceleration of braking and speed of the aircraft. The geometry of the braking mechanism will be improved, taking into account the use of composite materials in the design, the fuel consumption during the braking process. This leads to changes in the mass of the aircraft, as well as the properties of materials such as thermal conductivity, heat capacity at constant pressure, microhardness, coefficient of friction, density, and the degree of blackness of the body, which change depending on temperature.

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