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Modelling of Thermal Field of Electromotor's Collector in Service.

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ABSTRACT

This article represents stages of developing a mathematical model of thermal field of direct current electromotors' collector in service. The article shows solution of a problem of heat exchange within the system of bodies via finite elements method with the help of software package. Here you can find model's parameters, thermal load and boundary conditions within the system of bodies, and rationale of selection of finite elements type. The article gives initial data for calculation and solves criterial equations describing heat transmission. The authors obtained data on distribution of thermal fields in nonstationary and stationary operating conditions. Calculations results show thermal field's distribution in a collector from the moment of start of work and can be used for improvement of heat exchange and construction of commutator motors. Research's results are coherent with experimental data obtained in industrial conditions.

Keywords: modelling, finite elements method, heat flow, heat exchange, direct current motor, collector, Reynolds number, Prandtl number.

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INTRODUCTION

Modelling of thermal state of node points of electrical machine in service is one of the most important tasks defining its work reliability. Collector is the most complicated node point, because it defines commutation conditions. Limit values of heating temperature is determined by temperature class of insulation, and exceeding them can make electromotor inoperative.

METHODS

For modelling collector's thermal state we use the finite elements method (FEM), which is very popular in these latter days for solving problems of this class [1-4].

As collector is an axisymmetric structure and its active part is composed of homomorphic commutator segments, let's take a collector's sector having one commutator segment and one insulating pad as a design contour (fig. 1).

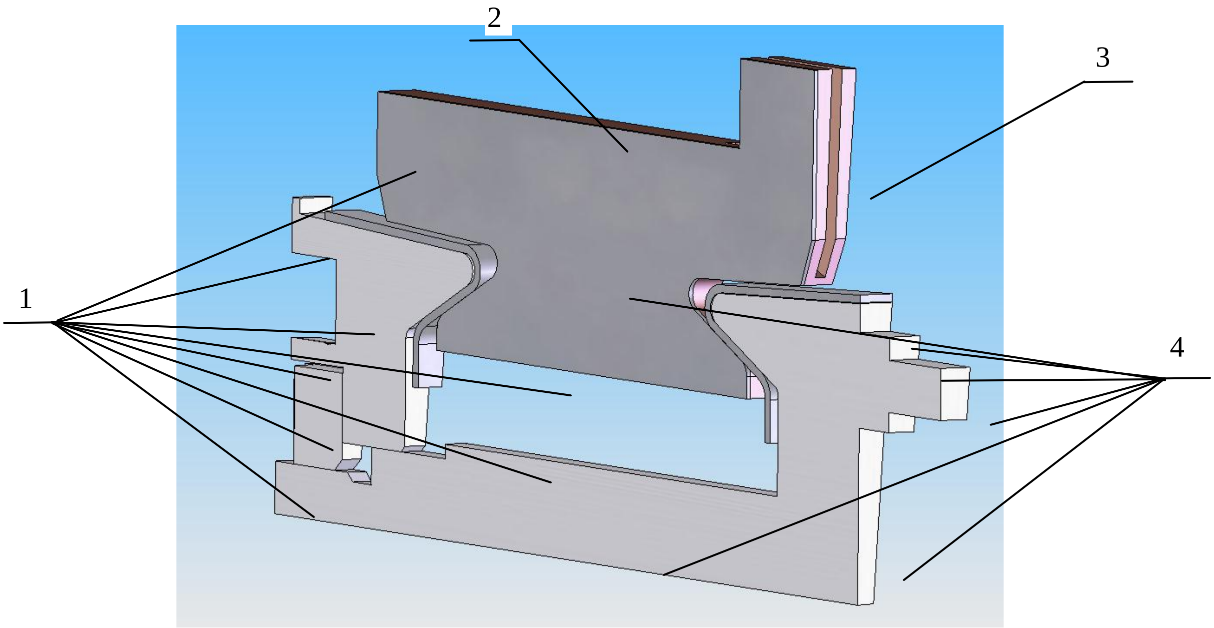

Figure 1: Collector's design contour: 1 – surfaces with convective heat exchange; 2 – surfaces with heat flow; 3 – internal heat source; 4 – thermally insulated surfaces.

We included several details into design contour, so let's admit an assumption that their interfaces have tight thermal contact. This is close to the reality, because, to provide integrity, so-called "arch action" force is in effect in collector, and that produces a pressure in all interfaces. We also admitted an assumption about absence of heat flow going between adjoining commutator segments on lateral faces through insulating pads, that is main heat flow goes along vertical axis through commutator segment. Accepted design contour allows to investigate different thermal state of individual segments, because during motor's work, as a rule, not the whole collector runs hot, but individual segments or groups of segments "burn up".

Initial data for calculation of temperature in the specified contour are following: structure of details, thermophysical properties of materials and heat flows affecting collector. Details structure is presented by coordinates of node points of finite elements composing design contour, thermophysical properties of materials are expressed by heat conduction coefficients, and heat flows impact is taken into account in boundary conditions [5, 6]. According to heat flows affecting collector, the chosen design contour includes boundary conditions of the 2nd type (density of heat flow going through surface) and boundary conditions of the 3rd type (presence of convective heat exchange).

The boundary condition of the 2nd type works for:

- Collector's contact surface, through which heat flow (arising due to brushes' frictions on collector and voltage drop) enters (fig. 3.1, pos. 2);
- Thermally insulated surfaces, that is surfaces with heat flow rate equaling to zero (fig. 1, pos. 4). As thermally insulated surfaces we accepted: lateral surfaces of commutator segments, contacting with insulating pads; surfaces of a front pressure-exerting cone and a commutator lug, insulated from armature winding. This assumption was made because of low thermal conductivity coefficient of insulation materials – two orders lower that copper thermal conductivity coefficient.

Strong interrelation of heat generation sources eliminates the possibility of considering collector in isolation from other node points of electrical machine. Therefore, when defining collector's temperature we should take into consideration impact of three main factors [7]:

- Environmental temperature, that is temperature of air inside the machine, which depends on value of motor's total loss.
- Value of intrinsic losses on collector due to voltage drop in brushes contacts and brushes frictions on collector's outer surface.
- Collector's thermal coupling with armature through armature winding sections contacts with commutator segments lugs.

MAIN PART

Temperature of air flowing around collector in case of ventilated motors can be accepted as such which is equal to temperature of incoming cooling air. When considering self cooled enclosed motors, with some approximation we can assume temperature of air around collector to be an average temperature of air inside motor:

$$\Delta\Theta_{в} = P_{\Sigma} /(\alpha_{в} S_{\text{motor}}), \qquad (1)$$

where [delta][theta]$_в$ is an average exceeding of temperature of air inside machine over outer cooling air temperature, S_{motor} is a conditional cooling surface of machine, [alpha]$_в$ – coefficient of convective heat exchange of outer cooling surface, P_{Σ} is total losses.

When defining motor's total losses we should consider all arising losses: fundamental and added ones.

Specific heat flow from collector's losses, which is relegated to collector's cooling surface, can be calculated by the following formula:

$$p_{coll} = (p_{c.b.} + p_{f.b.}) / S_{coll}, \qquad (2)$$

where $p_{c.b.}$ is electric losses in brushes' contacts; $p_{f.b.}$ is losses from brushes' frictions on collector; S_{coll} is conditional cooling surface of collector.

Electric losses in sliding contact should be expressed by the formula:

$$p_{c.b.} = \Delta U_b \cdot I_2, \qquad (3)$$

where [delta] U_b is voltage drop in brushes' contacts; I_2 is armature current.

Mechanical losses from brushes' frictions on collector should be expressed by the formula:

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$$p_{f.b.} = k \cdot S_{b\sum} V_{coll}, \tag{4}$$

where k is coefficient of brushes' frictions on collector; $S_{b\sum}$ is total area of brushes' contact; V_{coll} is collector's periphery speed.

Expression (2) is used during designing of electrical machines to find average exceeding of collector's temperature. Thereat one should make an assumption that all heat losses in brushes and collector contact are firstly absorbed by collector and then transferred into air inside machine via collector's surface.

For analysis of distribution of temperature inside collector let us transform the expression (2) taking into account part of heat flow passed by convection into the air flowing around the collector:

$$p_{\text{coll}} = (p_{\text{c.b.}} + p_{\text{f.b.}} - k_{\text{o}} \alpha_{\text{coll}} S_{\text{coll}} \Delta\Theta_{\text{coll}}) / S_{\text{coll}}, \tag{5}$$

where k_o is coefficient considering ratio of heat flow effect and convection; [alpha]$_к$ is coefficient of convective heat exchange of collector's outer surface; [delta][theta]$_к$ is exceeding of collector's outer surface temperature over ambient temperature.

Exceeding of collector's outer surface temperature over ambient temperature should be taken for established thermobalance.

Heat flow between armature and collector arises as a result of difference in temperatures of active elements – armature winding sections and commutator segments, having strong interrelation because of a high thermal conductivity coefficient of the material (copper). Thermal coupling is conducted with winding sections along the armature winding through commutator segments lugs joint surfaces (splined groove). In this regard let us assume collector's thermal coupling with armature to be a one-dimensional field, and for defining direction of motion of heat flow between armature and collector and also for estimating its rate, let us use values of average temperatures of collector copper and armature winding:

$$Q = (\Delta t_2' - \Delta t_{coll}') \lambda S / \delta, \tag{6}$$

where [delta] t_2' is average exceeding of collector copper temperature over temperature of air inside the machine; [delta] t_{coll}' is average exceeding of armature winding temperature over temperature of air inside the machine; [lambda] is thermal conductivity coefficient of armature winding material (copper); S is winding sections cross-section area; [[delta]] is distance between the commutator segment mid-point and the mid-point of armature section in axial direction.

Sign of the expression (6) characterizes heat flow direction. Sign "+" means heat movement from armature to collector, sign "-" – from collector to armature. This formula defines thermobalance between armature and collector, allowing to use initial data – heat flow rates of armature and collector, via values of their average temperatures.

Actually heat flow between armature and collector is not uniform. Part of heat is passed by the air flowing around armature and collector, also there is a heat flow from the details which are more heated to those which are less heated, that is from copper segments to steel cones in collector and from winding to body in armature. Presence of these two types is indirectly considered in definition of average temperatures of copper of armature and collector, in expression of which heat flows passed into air and contacting details are included.

Average exceeding of collector copper temperature over temperature of air inside the machine:

$$\Delta t'_{coll} = p_{coll} / \alpha_{coll}, \qquad (7)$$

where [alpha] $_{coll}$ is coefficient of convective heat exchange of collector's outer cooling surface.

During estimation of collector's thermal state, several expressions ((1), (5), (7)) include coefficient of convective heat exchange. Accuracy of the whole thermal calculation depends to a substantial degree on accuracy of its calculation. Coefficient of heat transmission defines speed of the process and other main characteristics [8, 9].

To estimate coefficient of heat transmission let us use criterial equations [10]. For the case of natural convection (heat transmission from the electromotor body outer surface)

$$Nu = C(Pr_0 \cdot Gr_0)^n, \qquad (8)$$

for the case of forced convection (heat exchange of rotating collector with the air inside the machine)

$$Nu = C \cdot Re_0^{\ m} \cdot Pr_0^{\ n} \cdot Gr_0^{\ p} (Pr_0 / Pr_s), \qquad (9)$$

$$Nu = \alpha \cdot l / \lambda, \qquad (10)$$

where Nu is a dimensionless group called Nusselt criterion; C, n, m, p are coefficients; Pr_0, Pr_s are Prandtl numbers correspondingly for cooling medium and cooling surface; Gr_0 is Grashof number for cooling medium; Re_0 is Reynolds criterion for cooling medium; α is value of heat exchange coefficient which is average for the exposed surface W/m^{2}ºC; l is characteristic dimension, m; [lambda] is medium's coefficient of heat conductivity W/mºC.

Collector is a body of rotation, so we should take its diameter as a characteristic dimension.

Reynolds criterion characterizes speed of medium's movement in relation to solid body:

$$Re = w \cdot l / \nu, \qquad (11)$$

where w is flow's speed, m/s; υ is medium's kinematic coefficient of viscosity, m^2/s.

Certain range of Reynolds criterion numbers defines the circumambient medium's movement mode - laminar or turbulent. In case of collector the circumambient medium's movement may be considered as turbulent one.

Prandtl number characterizes capability of heat to distribute in this medium.

$$Pr = \nu / \omega, \qquad (12)$$

where [omega] is medium's temperature conductivity.

Grashof number considers natural convection's impact inside the medium. In case of forced convection, arising during collector's rotating, exponent of power in a criterial equation for Grashof number is equal to zero, therefore, this number equals to one and it can be excluded from the equation (9). In case of natural convection Grashof number is estimated by the expression

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$$Gr = \beta \frac{gl^3}{\nu^2}\left(\Theta_s - \Theta_0\right), \quad (13)$$

where [beta] is medium's coefficient of volumetric expansion, 1/ºC; *g* is free fall acceleration, *g* = 9.81 m/s^2; [theta]$_s$, [theta]$_0$ are temperatures of cooling surface and cooling medium correspondingly, ºC.

In case of cross flow around cylindrical surfaces and turbulent nature of the flow the following exponents of power in the criterial equation (9) take place: *C* = 0.28; *m* = 0.6; *n* = 0.36. For conditions of natural convection - (8) *C* = 0.135; *n* = 0.33.

Thus, let's write expressions for estimation of heat-conduction coefficient during collector's rotating

$$\alpha = 0{,}28 \cdot Re_0^{\ 0,6} Pr_0^{\ 0,36} (Pr_0 / Pr_s)^{0,25} \lambda / l, \quad (14)$$

in case of natural convection of electromotor body

$$\alpha = 0{,}135 (Gr_0 \cdot Pr_0)^{0,33} \lambda / l. \quad (15)$$

For computational experiment we used finite-element packet COSMOS/Works, integrated into CAD-system of solid modeling SolidWorks.

Computational experiment includes following main stages:

- Creation of a solid model of design contour (fig. 1) via means of SolidWorks.
- Description of structure materials' properties. For thermal calculation we should set heat conductivity coefficient and volumetric heat capacity of the material. Materials employed may be isotropic and orthotropic. Possibility of setting orthotropic properties is very important, because insulating material consists of resin-impregnated pressed mica, therefore its properties differ through the length and breadth of fibres.
- Digitalization of computational domain, generation of finite elements grid. For solid bodies COSMOSWorks uses three-dimensional tetrahedral elements which can be of two types: linear (Draft quality mesh) and parabolical (High quality mesh), they are represented in fig. 2.

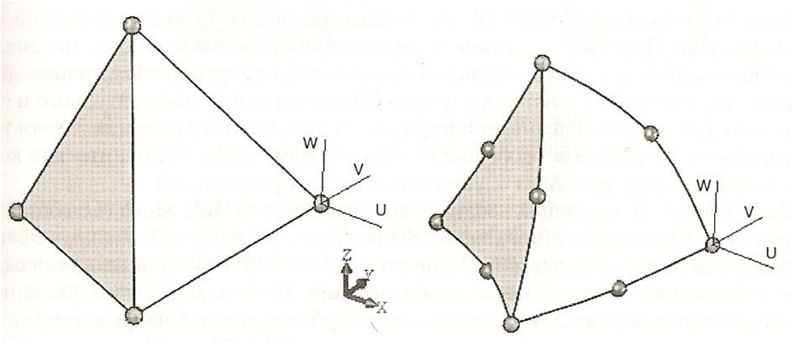

Figure 2: Solid finite elements

Linear elements have four node points in vertices. Such configuration provides linear variation of required parameter, in this case – temperature, within the elements volume. Parabolical elements, apart from node points in vertices, have node points in the middle of edges. The required parameter is described by quadratic polynomial. Edges and, consequently, planes can be curvilinear. This allows to reproduce a curvilinear geometrical pattern quite adequately. Taking into account these factors, let's use linear finite elements for development of design models: selection of correct boundary conditions, mode of interaction of details in the construction, parameters of grid seal. After estimating these factors, let's use higher order elements to obtain final results. This approach allows to save a lot of time for preparation of initial data [10].

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Figure 3: shows grid of finite elements of design model of the dirrect current excavating electromotor DPE-52 (ДПЭ-52).

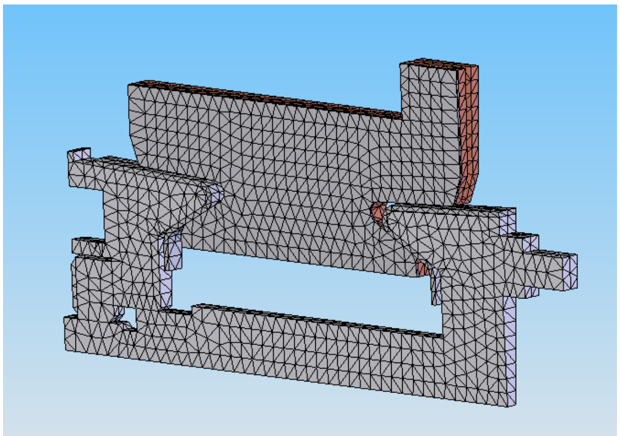

Figure 3: Grid of design model finite elements

- Application of loads and boundary conditions. Let's use expressions (1) – (15).
- Assignment of analysis type. COSMOSWorks provides stationary and nonstationary calculation of temperature. Stationary calculation should be used for estimation of collector's thermal field at the moment of thermobalance establishment. Nonstationary calculation is not less important. It allows to define collector's thermal field in transient conditions. Nonstationary thermal calculation does not consider changes of boundary conditions through time. If necessary, this limitation may be compensated via dividing of an assigned task into a range of subtasks connected with each other: thermal field, estimated in a previous task, is used as the initial datum for the following one. With the help of this stratagem one can imitate varying boundary conditions and loads, consequently changing them in different subtasks, at the same time within one subtask boundary conditions and loads are constant [11].

Computational experiment was carried out for a mining traction motor DPE-52. According to technical characteristics, heat flow parameter limits for a contact surface are shown in table 1. The electromotor under investigation can work both in blowing mode (with cooling) for a long time and in enclosed design (without cooling) up to 45 min. under nominal speed 1200/2200 rpm.

Table 1: Parameters of heat flow along the contact surface

№	Title	Mark	Value
1	Electric losses in contact	$p_{к.щ.}$	375 W
2	Mechanical losses in contact: min under rotation frequency 1200 rpm max under rotation frequency 2200 rpm	$p_{т.щ.}$	150 W 450 W
3	Part of total losses, passed by the flowing air min under rotation frequency 1200 rpm max under rotation frequency 2200 rpm		257 W 352 W
4	Total losses min under rotation frequency 1200 rpm max under rotation frequency 2200 rpm		268 W 473 W
5	Area of conditional cooling surface	$S_к$	0.07 m^2
6	Total specific heat flow going through contact surface: min under rotation frequency 1200 rpm max under rotation frequency 2200 rpm	$p_к$	3829 W/m^2 6757 W/m^2
7	Coefficient of convective heat exchange min under rotation frequency 1200 rpm max under rotation frequency 2200 rpm	$\alpha_к$	46 W/m^2 63 W/m^2

CONCLUSION

Fig. 4 and 5 show results of estimation of collector's thermal field in the established mode with cooling under different nominal speeds.

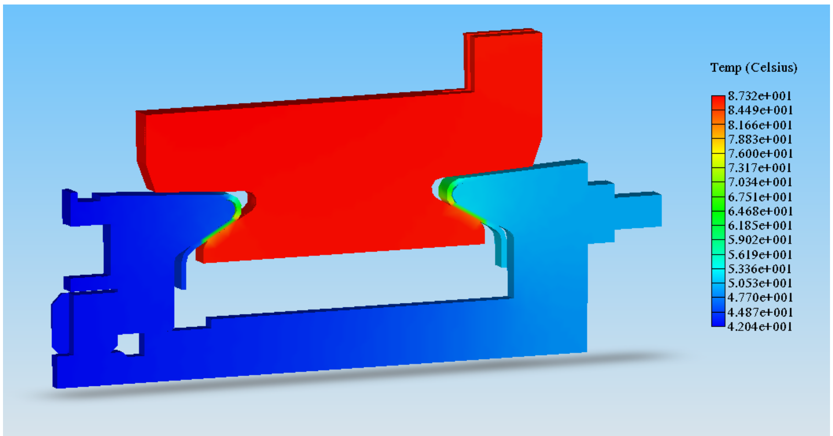

Figure 4: Collector's thermal field in the established mode under rotation frequency 1200 rpm

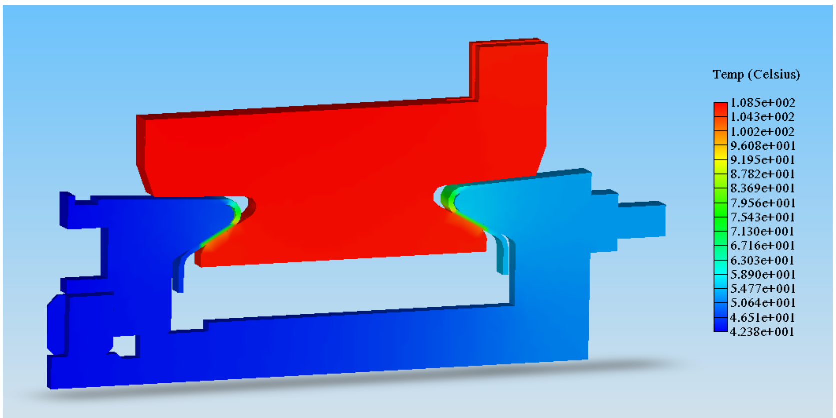

Figure 5: Collector's thermal field in the established mode under rotation frequency 2200 rpm

Results show that commutator's segment has the highest temperature: 87ºC and 108ºC correspondingly under 1200 and 2200 rpm. The higher rotation frequency intensifies heat flow going through the contact surface, therefore the segment heats up at a more intensive rate. At the same time cooling rate also increases due to increase of the coefficient of convective heat exchange, therefore pressure-bearing cones temperature in different modes of functioning varies unissentially. All collector's details temperature is in permissible range from the viewpoint of temperature class of insulation.

Results show (fig. 7) that after 45 minutes of functioning collector's temperature is close to the nominal established one. The further exploitation of the motor without cooling leads to its overheat. As well as during functioning with cooling, commutator segment has the highest temperature. Pressure bearing cones' heating nature changes to some degree. Left cone is more heated, because it is heated by convection with the internal air, which absorbs all heat losses of the motor functioning without forced cooling. Thermal fields for any stage and mode of the work may be created likewise, and that allows to rationalize construction solutions and technology concepts [12].

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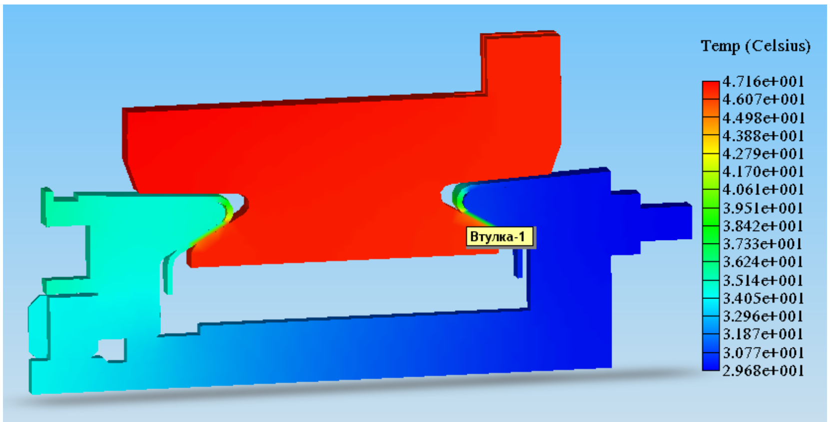
Figure 6, 7 show changes of collector's thermal field in transient condition during functioning without cooling under rotation frequency 1200 rpm: fig. 6 *a* – after 15, fig. 6 *b* – after 30, and fig.7 – after 45 minutes of working.

a



b

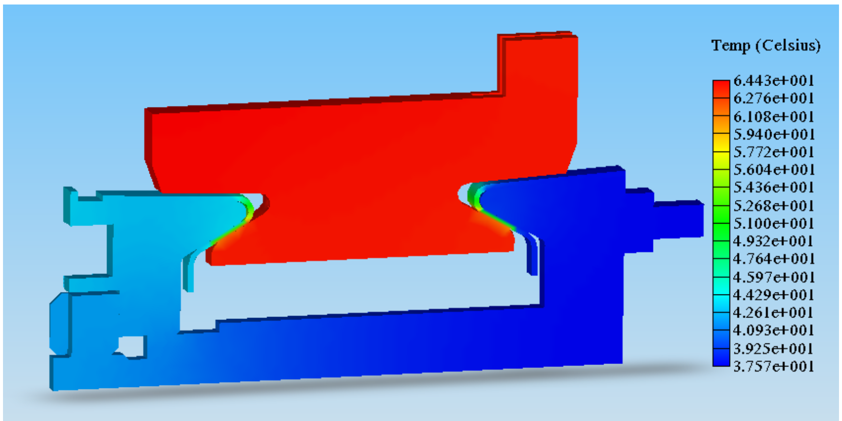


Figure 6: Collector's thermal field in transient condition under rotation frequency 1200 rpm: *a* – after 15 min; *b* – after 30 min

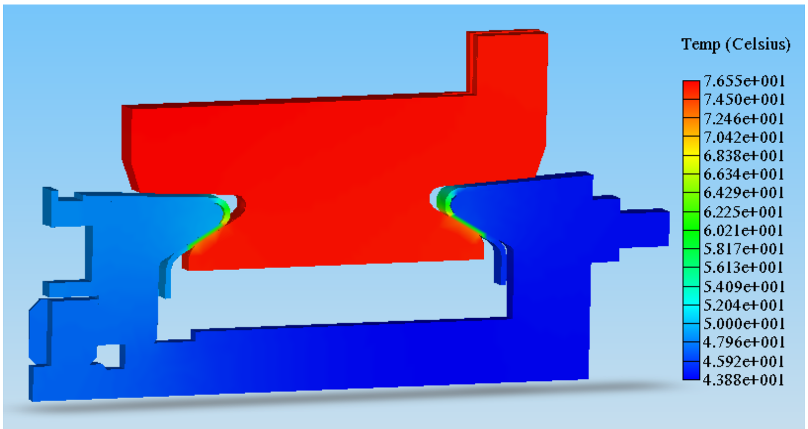


Figure 7: Collector's thermal field in transient condition under rotation frequency 1200 rpm after 45 min

RESUME

The developed model of collector's thermal state allows to model collector's thermal field, both in established thermobalance and in transient condition, taking into account heat flows functioning while in service in normal work mode and in modes of different overloads.

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