Convective heat transfer investigations at parts of a generator circuit breaker

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Abstract

Current carrying equipment for power engineering is getting smaller and more compact in order to meet customer demands. In spite of small dimensions it should be able to carry still growing currents, which lead to high current densities in the current carrying parts and therefore in high heating of these parts. Due to small dimensions the heat transfer from these parts is restricted too. To assure that the maximum temperature rise in equipment parts stays under the allowed temperature rise fixed in the standards, the temperatures in these parts should be calculated first (for instance with thermal networks).

The power losses produced in the current carrying parts are transferred through convection, radiation and conduction from these parts to the ambient. For better cooling effectiveness several heat sink types, for example on the parts of generator circuit breakers can be used. To improve the accuracy of convective heat transfer calculation with thermal networks, the n_1, c_1 factors for affinity function $Nu = c_1(GrPr)^{n_1}$ for natural convection, depending on the angle of the heat sink to the airflow, have been determined experimentally. Another point of thermal investigation is heat sink cooling with air, while the air is pre-warmed by other hot parts below the heat sink. This kind of interaction appears particularly in very compact devices and should therefore be investigated for a better accuracy of temperature calculation with thermal networks. Investigations have been carried out for different average temperatures of the heat plate, which was placed below the heat sink and for different heat sink positions.
Keywords: thermal networks, heat transfer coefficient, affinity function, natural convection.

1   Introduction

As a result of still increasing operating currents and decreasing dimensions of the current currying equipment, which results in high temperatures of the current path and the enclosure, a correct thermal design for this equipment is more important than ever. All temperatures along the current path should stay under temperature limits given in adequate standards (for generator circuit breaker [1]). For a generator circuit breaker operating at maximum ambient temperature $\vartheta_0 = 40°C$ the maximum allowable temperature is $\vartheta_{max} = 105°C$ at silver-coated contacts and $\vartheta_{max} = 70°C$ (in some cases 80°C) on the enclosure. Because of small dimensions and compact design of such devices the current densities in the current path are comparatively high and heat transfer from these device's parts to the ambient is restricted. In order to increase the heat transfer from these parts to the cooling medium, additional heat sinks have to be used. They can be located on the current path (fig. 1a) or on the enclosure (fig. 1b).
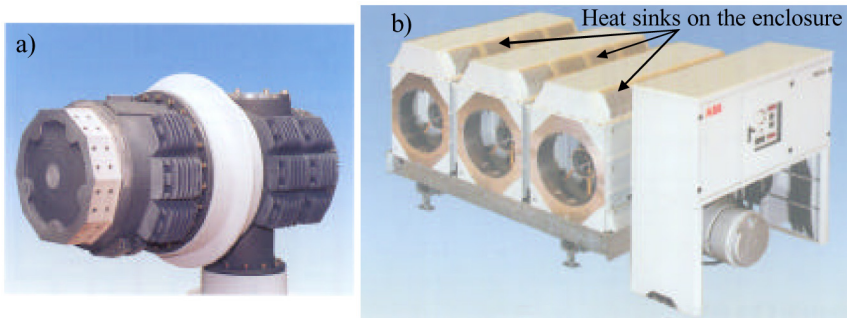

Figure 1:   Heat sinks located on: a) current path and on b) enclosure

To improve the current carrying capacity of generator circuit breaker or other current carrying devices and for accelerating the development process of new devices thermal network models are used to predict the behaviour of the device. These models allow temperature calculation of device's parts and examination of several changes in device's design as well. In order to use thermal network models for development and improvement of electrical devices fundamental research on convective heat transfer were carried out. The following article deals with experimental convective heat transfer investigations at parts of generator circuit breaker. A method will be shown how to determine the n_1, c_1 parameters for affinity function $Nu = c_1(GrPr)^{n_1}$ (free convection) for two heat sink types, depending on heat sink's position in the air flow.

2   Temperature calculation with thermal networks - fundamentals

The thermal network method, based on the analogy between electrical and thermal field, allows one to calculate the temperature distribution in electrical

devices. The thermal heat transfer between a device's parts and from these parts to the ambient can be calculated as well [2, 5]. For the temperature calculation the device's model has to be "imaginary" divided into n – "slices". Each slice (fig. 2) is represented by thermal sources (thermal power losses in current path and in an enclosure P_L and P_K), convective (R_{K1}, R_{K2}, R_{K3}), radiative (R_{S1}, R_{S2}) and conductive (R_L) resistances and for forced convection by the thermal resistance of the coolant stream $R_{\dot{V}}$. For dynamic temperature calculation the heat capacities C_L and C_K have to be included.
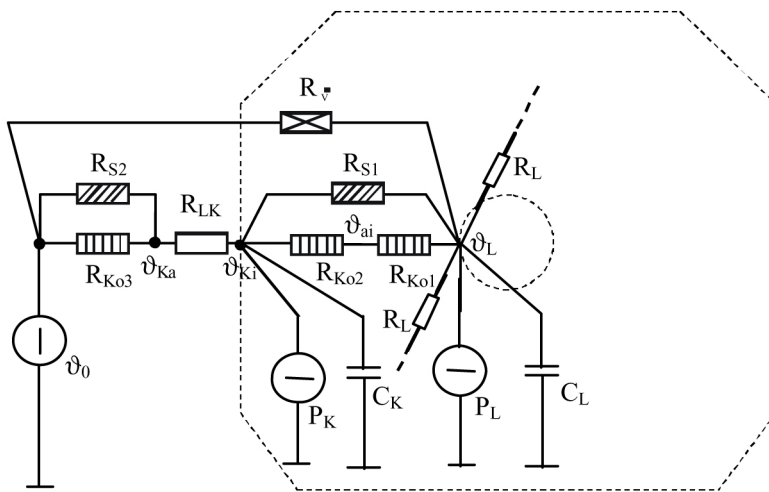

Figure 2:   Thermal network for one section (slice) of generator circuit breaker.

Thermal power losses produced in the current path with temperature ϑ_L are convoyed partially via convection (R_{Ko1}) to the interior air with a temperature ϑ_{ai} and from there to the internal side of the enclosure (R_{Ko2}) with ϑ_{Ki}. Another part of thermal power losses produced in current path is convoyed via radiation (R_{S1}) directly to the internal side of the enclosure and via conduction (R_L) across the current path to the neighbouring sections (layers). The power losses arising from the current path to the internal side of enclosure sum up with power losses produced in this enclosure (P_K) and are convoyed via conduction (R_{LK}) to the outer side of the enclosure. From there, the power losses are transferred via convection (R_{Ko3}) and radiation (R_{S2}) to the ambient air with the temperature ϑ_0.

In case of cooling via forced convection, the volume stream of cooling air ($R_{\dot{V}}$) convoys a major part of thermal power losses from current path directly to the ambient air.

Each element of the thermal network can be described with adequate equations (Table 1).

The accuracy of temperature calculations with thermal networks depends significantly on the accuracy of input parameters in equation (1) to (9) used for calculation. Assuming that the geometrical dimensions (l, d, l_w, A, O_{Ko}, O_S) are known, other parameters and physical constants depending on material, temperature and so on can be in most cases found in the literature. In some cases, especially for seldom-used geometries, or for common geometries under special conditions some parameters must be determined experimentally. In the following, experiments for determination of c_1, n_1 parameters at free convection (eqn. (4)) and c_2, n_2 parameters at forced convection (eqn. (5)) on heat sinks under special conditions will be described.

Table 1:   Equations describing thermal network elements.

Thermal network elements	Equations	
Power losses in current path $\mathbf{P_L}$ *and in enclosure* $\mathbf{P_K}$	$P_L = kI^2 \dfrac{\rho_{20}\, l}{A}\left[1 + \alpha_T\left(\vartheta - 20°C\right)\right]$	(1)
Thermal conduction resistance $\mathbf{R_L}$ *between parallel surfaces*	$R_L = \dfrac{d}{\lambda A}$	(2)
Convective resistance $\mathbf{R_{Ko}}$	$R_{Ko} = \dfrac{1}{\alpha_{Ko} O_{Ko}}$	(3)
Convective heat transfer coefficient $\mathbf{\alpha_{Ko}}$ *for free convection:*	$\alpha_{Ko} = \dfrac{\lambda}{l_W} Nu = \dfrac{\lambda}{l_W} c_1 \left(Gr\, \Pr\right)^{n_1}$	(4)
Convective heat transfer coefficient $\mathbf{\alpha_{Ko}}$ *for forced convection:*	$\alpha_{Ko} = \dfrac{\lambda}{l_W} Nu = \dfrac{\lambda}{l_W} c_2\, \mathrm{Re}^{n_2}$	(5)
Radiation resistance $\mathbf{R_S}$	$R_S = \dfrac{1}{\alpha_S O_S}$	(6)
Heat transfer coefficient of radiation $\mathbf{\alpha_S}$	$\alpha_S = \varepsilon_{12} C_S \dfrac{T_2^4 - T_1^4}{\vartheta_2 - \vartheta_1}$	(7)
Thermal resistance of the coolant stream by forced convection $\mathbf{R_{\dot{V}}}$	$R_{\dot{V}} = \dfrac{1}{2c_P \rho \dot{V}}$	(8)
Thermal Capacity $\mathbf{C_W}$	$C_W = c_W m$	(9)

3   Determination of c_1, n_1 parameters depending on heat sink's angle to horizontal position ($c_1 = f(\alpha_H)$, $n_1 = f(\alpha_H)$)

A heat sink on the current path or on the enclosure of a generator circuit breaker can be placed with different angles to the ascending, cooling air (fig. 1). This can result in different values of the convective heat transfer coefficient α_{Ko} and c_1, n_1 parameters for each heat sink position. To improve the accuracy of temperature calculation with thermal networks, experimental thermal investigations to determine these parameters on two different heat sinks were carried out. The

smaller heat sink (fig. 3a), hereinafter called "HS1", has a characteristic length $l_w = 107$ mm, a convective surface $O_{Ko} = 0{,}098$ m² and a radiating surface $O_S = 0{,}034$ m². The bigger one ("HS2" fig. 3b) has a characteristic length $l_w = 330$ mm and respectively $O_{Ko} = 0{,}399$ m² and $O_S = 0{,}09$ m². Both heat sinks were mounted on separate heat plates with controllable electric power. The power was measured with a power meter LMG 95. The heat plates were thermally insulated from the ambient with mineral wool, so that the major part of the supplied power was delivered to the ambient via the heat sink. The heat sink-, heat plate-, insulation- and ambient-temperatures were measured with thermocouples and all data were transferred to the PC data acquisition – system. The heat sinks angle to the horizontal position α_H was changed from 0° to 120° for HS1 and from 0° to 90° for HS2 (fig. 3c).
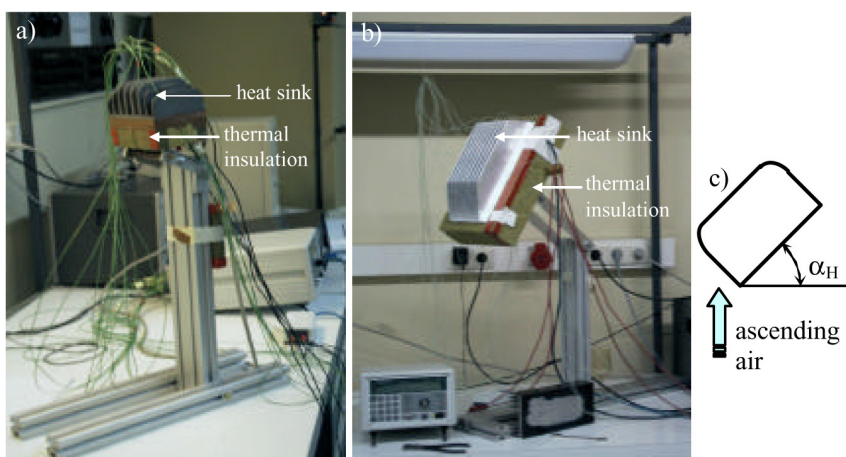

Figure 3:   Test arrangement for thermal investigations on heat sinks HS1 (a) and HS2 (b) by different angle αH to horizontal position (c)

The convective dissipation of heat P_{Ko} (fig. 4) can be calculated from:

$$P_{Ko} = P_{el} - (P_{Is} - P_S) \tag{10}$$

where P_{el} is the supplied electrical power, P_{Is} is the dissipated heat through the thermal insulation and P_S is the dissipated heat via radiation given by:

$$P_S = \varepsilon_{12} \, C_S O_S \, 10^{-8} (T_1^4 - T_2^4). \tag{11}$$

Resultant emissivity ε_{12} in equation (11) was determined by dint of a pyrometer to $\varepsilon_{12} = 0{,}96$ for HS1 and $\varepsilon_{12} = 0{,}3$ for HS2. C_S is the radiation capability of the "black body" ($C_s = 5{,}67$ Wm^{-2}K^{-4}) and T_1 and T_2 are absolute temperatures of the heat sink and the ambient respectively.

The heat leak through the thermal insulation P_{Is} was determined by calibration. For calibration each heat sink was replaced by an insulating plate and the temperature difference $\Delta\vartheta$ between the heat plate and the insulation was

measured at different supplied power values. This heat dissipated through the thermal insulation, for an ascertained temperature difference $\Delta\vartheta$, was then subtracted from the supplied power in experiments conducted with heat sinks (fig. 4).

After the calculation of P_S and the determination of P_{Is} the convective heat flow P_{Ko} could be calculated.

For known P_{Ko}, the Nußelt – number Nu in eqn. (4) given by:

$$Nu = c_1 (Gr\,\mathrm{Pr})^{n_1} \tag{12}$$

can also be calculated from:

$$Nu = \frac{P_{Ko} l_w}{O_{Ko} \Delta\vartheta \lambda} \tag{13}$$

For determination of c_1 and n_1 parameters in eqn.(4 and 6) the Grashof – number Gr must be known too. It is given by:
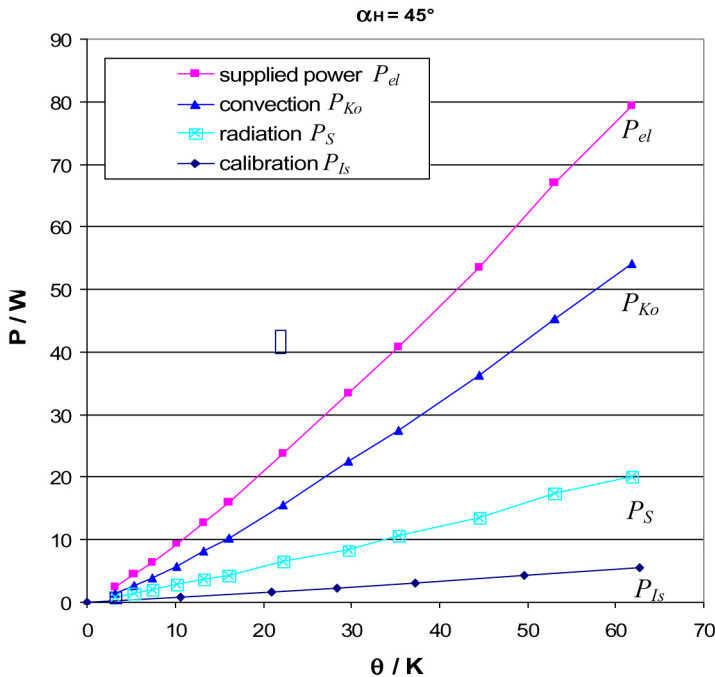
$$GrPr = k_s \Delta\vartheta l_w^3 \tag{14}$$

Figure 4:   Supplied electrical power P_{el}, convective dissipated heat P_{Ko}, via radiation dissipated heat P_S, and through the thermal insulation dissipated heat P_{Is} versus temperature difference between heat plate and outer side of the thermal insulation (HS1, angle to the horizontal position $\alpha H = 45°$).

Because of the two unknown parameters c_1 and n_1 in eqn. (12) this eqn. can't be calculated. By logarithm each side of eqn. (12) it can be rewritten as:

$$\log Nu = \log c_1 + n_1 \log (GrPr) \tag{15}$$

and then be solved graphically via double-logarithmic diagram [3, 4].

The substitution of log Nu by A, log (GrPr) by B and log c_1 by C results in a linear equation:

$$A = n_1 B + C \tag{16}$$

in which n_1 and C can be determined graphically. Putting the investigation results for free convection in a diagram where log Nu is the ordinate and log (GrPr) the abscissa, the results build a straight line so that the n_1, c_1 parameters can be determined.

In this way the determined parameter n_1 was approximately $n_1 = 0,33$ in the whole range of α_H angle. In affinity theory this value of α_H indicates a turbulent airflow and therefore it has been set up for further calculations for each α_H angle. With this value of n_1 parameter the c_1 parameter for both HS1 and HS2 was calculated (fig. 5) from:

$$c_1 = \frac{Nu}{(GrPr)^n} \tag{17}$$

In fig. 5 is clearly to see, that c_1 parameter and therefore convective heat transfer from the heat sink to the ambient depends for HS1 and HS2 on α_H angle. Both heat sinks have the best cooling performances for α_H in range between $\alpha_H = 60°$ and $\alpha_H = 90°$.
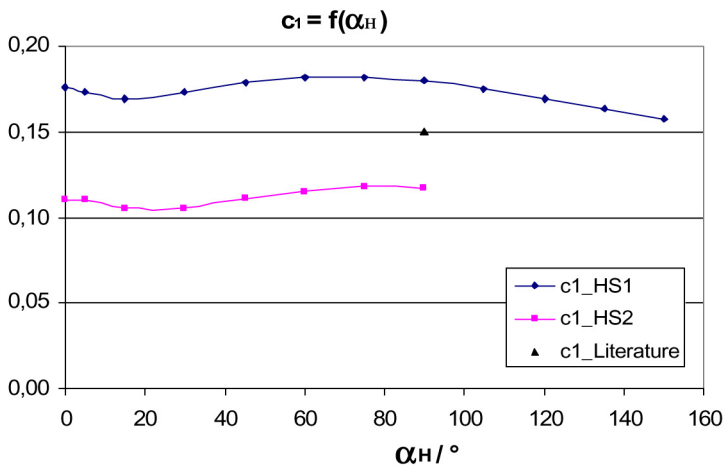

Figure 5:   Parameters c1 by n1 = 0,33 for different heat sink angle αH to the horizontally position.

Furthermore it can be seen, that c_1 parameter for HS1 achieves much higher values then c_1 parameter for HS2. The $c_1 = 0{,}15$ given in [3] for $\alpha_H = 90°$ lies between the values of the c_1 parameters for HS1 and HS2. This difference is caused by different characteristic length l_w for each heat sink. After air passes a certain critical length between fins, its temperature and therefore pressure drop increases. Thus decreases the heat sink cooling performance. It is therefore advantageous if the characteristic length l_w of the heat sink isn't too long.

4　Determination of c_1, n_1 parameters depending on heat sink's angle to horizontal position by heat sink cooling with pre-warmed air

In some cases heat sinks in current carrying devices are placed above other hot parts and therefore they are cooled with air that is warmer then ambient air. In order to estimate that effect on the heat sink's cooling performance thermal investigations have been performed. The test arrangements described above were modified so that 1000 mm x 250 mm stainless steel heat plates below each heat sink were integrated (fig. 6). Measurements analogous to point 3 for two different heat plate power levels (HP1: 40 W; HP2: 80 W) were carried out. In order to achieve uniform current distribution in the heat plates and therefore uniform temperature distribution in the heat plate DC current was used. The heat plate temperature rise was approximately 26 K for HP1 and 46 K for HP2 power level. Reference temperature was the ambient temperature ϑ_0.
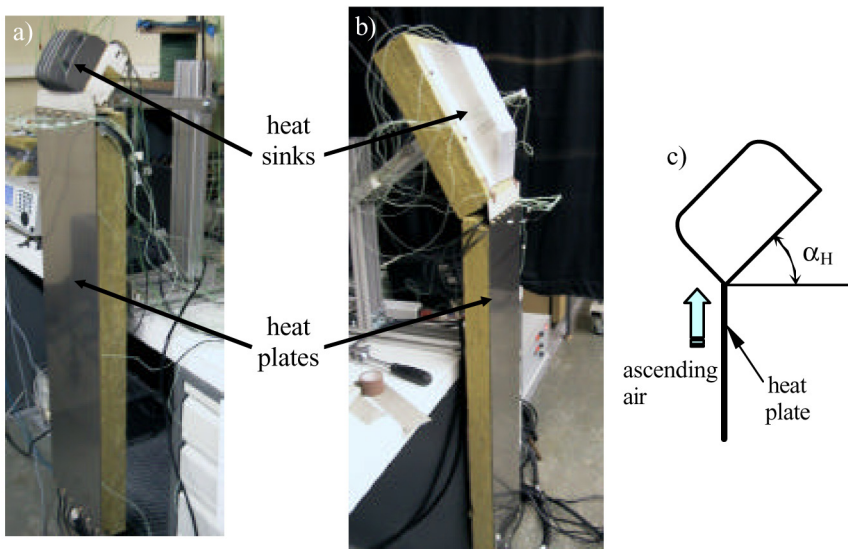

Figure 6:　Test arrangement for thermal investigations on heat sink HS1 (a) and HS2 (b) for different angle α_H to horizontal position by cooling with pre-warmed air.

For small α_H angles (0° to 30°) heat plates below heat sinks have only a little influence on c_1 factors for both HS1 and HS2 (fig. 7). In that case the major part of ascending warm air flows over the heat sink fins. For α_H equal or bigger than 60° the heat plates directly below the heat sink can even slightly increase the heat sink's cooling performance. The ascending air, which was pre-warmed by a hot plate, is warmer than the surrounding air and therefore can pass between cooling fins faster than cold air. That can also cause air turbulences by passing cooling fins, which improve heat transfer between heat sink and the ambient air. The increased temperature of the cooling air is compensated by a higher air velocity.

The heat plate temperatures examined in experiment were smaller than the heat sinks temperature for each power level. If the heat plate placed below the heat sink is too warm (approximately as warm as heat sink temperature or warmer) the cooling air can be too hot for efficiently cooling the heat sink even with higher air velocity. In such a case the cooling performance of the heat sink placed above heat plate can decrease.
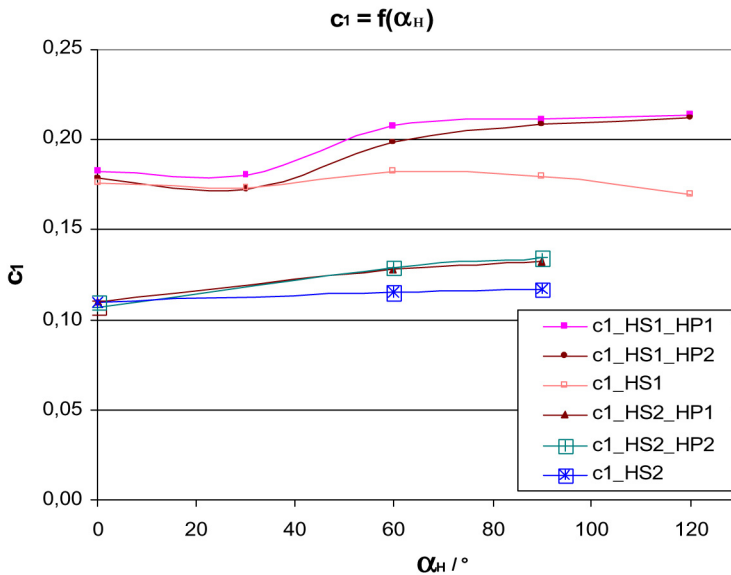

Figure 7: Parameter c1 for different heat sink angles αH to the horizontal position and different power levels of the heat plates below heat sinks.

5 Summary

In order to calculate the temperatures of current carrying devices with thermal networks accurately, thermal investigations by free convection on heat sinks,

used in such devices, were carried out. These investigations showed that in addition to the geometry and the dimensions of the heat sink its position to the airflow must be considered too. Further, the influence of the presence of other hot parts below the heat sink was investigated. The results for this experiment shown in fig. 7 indicate only a slight influence of such parts on the cooling performance of the heat sink. In case of heat plates placed below heat sinks, whose temperatures are lower than the heat sink temperature and with heat sink angles to horizontal position α_H equal or bigger than 60° the heat sinks cooling performance can even be slightly improved.

References

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