Jerzy ZGRAJA

Lodz University of Technology, Institute of Applied Computer Science

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Heat calculations for water-cooled radiators of the inverter for induction heating

Abstract. *Induction heating inverters are generally equipped with water-cooling systems. This results from considerable, of at least several kilowatt, power of these devices as well as from the easy availability of water-cooling used commonly, for example, for cooling the inductor. The paper presents a method for the calculation of water cooling of radiators based on the use of the finite element method for the analysis of heat transfer in the radiator, taking into account convection heating of water flowing in the cooling channel, which leads to a change in heat transfer conditions.*

Streszczenie. *Falowniki do nagrzewania indukcyjnego są zazwyczaj wyposażone w systemy chłodzenia wodnego. Wynika to zarówno ze znacznych, co najmniej kilkunasto kilowatowych, mocy tych urządzeń jak również z faktu dostępności chłodzenia wodnego stosowanego powszechnie np. do chłodzenia wzbudnika. W pracy przedstawiono metodę obliczania chłodzenia wodnego radiatorów opartą o wykorzystanie metody elementów skończonych do analizy przewodzenia ciepła w radiatorze z uwzględnieniem konwekcyjnego podgrzewania przepływającej w kanale chłodzącym wody, prowadzącego do zmiany warunków wymiany ciepła.* (***Obliczenia cieplne chłodzonych wodą radiatorów falownika do nagrzewania indukcyjnego***).

Keywords: induction heating, inverter, cooling systems.
Słowa kluczowe: nagrzewanie indukcyjne, falownik, chłodzenie wodne.

Introduction

At present, semiconductor resonant inverters are a predominant group of power sources in industrial induction heating technologies. The use of IGBT and MOSFET transistors allows one to apply sources of both high power exceeding 1MW and of high frequencies of over 500 kHz [1], [2]. Limiting the size of power sources calls for the application of effective cooling systems, which – even for sources of relatively low power – leads to the common use of water-cooling [1]. This refers to the cooling of the inductors, where the greatest concentration of power losses usually occurs, as well as to other elements such as transformers, chokes, capacitors, busbars and power electronic high current elements, i.e. transistors and diodes. When electro-insulated units are used, many transistors and diodes can be placed on one radiator, which leads to the high concentration of power losses, even in the case of water-cooled radiators. With the growing miniaturisation of devices there is a need for increasingly accurate calculations for heat radiators as early as at a stage of their design. The present paper presents the method used and the results obtained for such calculations based on the combination of the numerical finite element method for heat transfer in a radiator and – based on the similarity theory – determination of both coefficients of the convection capture of heat in cooling channels with regard to the effect of heating of the flowing cooling water. The values of power losses in the units mounted on the radiator as well as transfer resistances in the unit-radiator were applied arbitrarily on the basis of catalogue parameters and can easily be modified. This type of approach constitutes the alternative for numerical calculations combining the analysis of heat transfer and computational fluid dynamics. Given that the aim of the radiator heat calculations is not the calculation of temperature distribution in the cooling water and that the semiconductor elements are at a relatively considerable distance from the cooling water surface (which reduces the effect of local values of the convective heat transfer coefficient on the temperature value in the vicinity of the semiconductor element), calculation limitations of the method being applied do not appear particularly significant when it comes to the replacement of local values of the convective heat transfer coefficient with substitute averaged values. However, it is clear that the problem of accuracy of calculation of equivalent heat transfer coefficients using the similarity theory remains.

Calculation model for heat transfer in the radiator

A radiator in the form of a water-cooled aluminium block, having dimensions of 460x310x25 mm shown in Fig. 1, to with a six 300A IGBT bridges in the form of electro-insulated modules, was analysed.

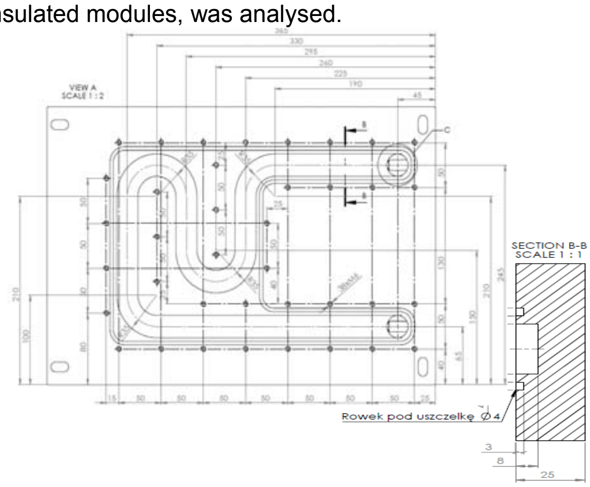

Fig.1. Water-cooled radiator, the water channel view.

Calculations of the temperature field distribution in the radiator were based on the analysis of the Fourier-Kirchhoff equation:

$$div[(-k)grad\vartheta]+\rho c\frac{\partial\vartheta}{\partial t}=p_v \tag{1}$$

where: λ – the specific thermal conductivity of the radiator material, ρ – the mass density, c – the specific heat, p_V – volume density of the heat flux of sources,
taking into account the third boundary condition and the initial condition in the form of an isothermal field of the temperature value equal to the inlet water temperature.
For the radiator under consideration a discrete 3D model shown in Fig. 2a was built using the commercial Flux® program.

To take into consideration a change in the temperature of the cooling water along the channel in the calculations, the channel was divided into ten isothermal zones, Fig.2.b, whereby the isothermality referred only to the water temperature field and not to the walls of the radiator water channel.

a)

b)

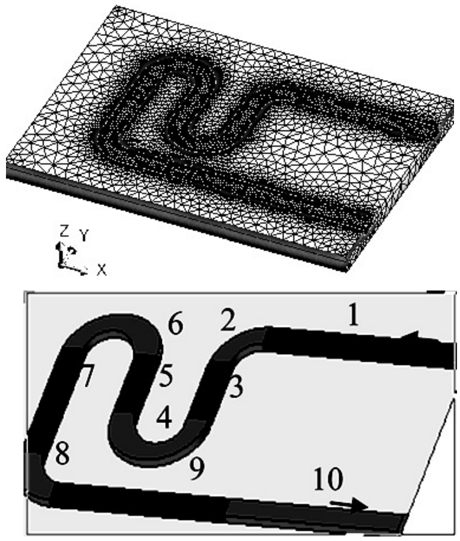

Fig.2. The 3D calculation model of a radiator: a) discrete model; b) division of the channel into ten zones with different temperatures of the cooling water.

Both for the outer radiator surface (excluding the surface of contact with the IGBT module) and for the inner surface of the water channel the third boundary condition of heat exchange described by Newton's law was assumed:

(2) $$q=\alpha_k(\vartheta_F-\vartheta_o)$$

where: α_k – the convective heat transfer coefficient, ϑ_F – the temperature of the radiator surface, ϑ_o – the temperature of the surrounding fluid.

The radiation heat exchange was neglected in the calculations, which is fully justified for the channels, whereas for the outer surface a calculation error of the radiator temperature distribution is strongly limited in this case, both due to a small value of aluminium emissivity and a decisive effect of radiator-water heat exchange.

Convective heat transfer coefficient in the channels

The convective heat transfer is strongly dependent on both the flow type and the fluid flow character as well as the geometric configuration of the solid body washed by fluid [3]. In this context, when the similarity theory and dimensional (criteria) analysis are applied, the convective heat exchange between the inner surface of the radiator cooling channel and the water flowing in it should be treated as a specific exchange with the forced flow in the closed system. For this type of heat exchange, the critical value of the Reynolds criterion Re_{kr} is, acc. to [3] Re_{kr} =2000, while acc. to [4] for channels of circular section and rectangular cross section is Re_{kr} =2300. This means that for convective heat exchange described by the number $Re< Re_{kr}$, the flow motion in the channel is of laminar character. Knowing Re_{kr}, one can determine the critical speed w_{kr} of a fluid of kinematic viscosity ν below which the laminar motion occurs [3]:

(3) $$w_{kr}=\mathrm{Re}_{kr}\frac{\nu}{l}$$

where: l – the linear characteristic dimension.

For a circular section of a diameter d, the linear characteristic dimension l=d, whereas for a channel of a different cross-section, it is:

(4) $$l=\frac{4S_k}{Ob}$$

where: S_k – the channel cross-section value, Ob – the perimeter of the channel cross-section value.

For a fully turbulent character of the fluid motion, one can obtain cases for which [4] Re>3000, although in practice, the certainty of obtaining of such a state is possible with Re>10000 [3], [4]. The numerical values given above show that the considerable discrepancies in defining a turbulent state on the basis of a criteria analysis, which can be the cause of calculation error for this type of flows.

In the case of the radiator under consideration and the occurring power losses, we mainly deal with a laminar flow, which limits the level of the calculation error expected. For laminar motion acc. to [5], the Nusselt number for a flow in channels of circular section of a diameter D, taking into consideration turbulences in the initial section of a channel of a length L, can be determined from the relationship:

(5) $$Nu=3{,}66+\frac{0{,}065\cdot\frac{D}{L}\cdot Re\cdot Pr}{1+0{,}04\cdot(\frac{D}{L}\cdot Re\cdot Pr)^{2/3}}$$

where: Re and Pr – the Reynolds and Prandtl numbers calculated for the calculation temperature equal to the arithmetic mean of the temperatures of the fluid and the channel wall.

For channels of a rectangular cross-section, this relationship is transformed [4] into the form:

(6) $$Nu=7{,}49-17{,}02\cdot\frac{H}{W}+22{,}43\cdot(\frac{H}{W})^2-9{,}94\cdot(\frac{H}{W})^3+ \\ +\frac{0{,}065\cdot\frac{l}{L}\cdot Re\cdot Pr}{1+0{,}04\cdot(\frac{l}{L}\cdot Re\cdot Pr)^{2/3}}$$

where: H, W – the smaller and bigger linear dimensions of the rectangular cross section of the channel.

At the same time, for the same character of a laminar flow acc. to [3], the Nusselt number in channels of any cross section, taking into consideration the turbulences in the initial section of the channel of a length L, can be determined from the relationship:

for $\mathrm{Re}_f\cdot\mathrm{Pr}_f\cdot\frac{l}{L}\le 13$

(7) $$Nu_f=(0{,}5\cdot\mathrm{Re}_f\cdot\mathrm{Pr}_f\cdot\frac{l}{L})^{0{,}79+0{,}17\cdot\log\frac{\nu_f}{\nu_F}}$$

and for $\mathrm{Re}_f\cdot\mathrm{Pr}_f\cdot\frac{l}{L}>13$

(8) $$Nu_f=(6{,}43\cdot\mathrm{Re}_f\cdot\mathrm{Pr}_f\cdot\frac{l}{L})^{\frac{1}{3}}(\frac{\nu_f}{\nu_F})^{0{,}14}$$

where – the subscript f denotes the value determined for the cooling water temperature, while the subscript F denotes the value determined for the channel surface temperature.

Using the Nusselt number, the value of the convective heat transfer α_k can easily be calculated from the relationship [3]:

(9) $$\alpha_k=\frac{Nu\cdot\lambda}{l}$$

where: λ – the specific thermal conductivity of the fluid (cooling water).

Table 1, (flow Q=0.5*10^{-4} [m^3/s] i.e. 3 [l/min], water temperature 15°C, channel temperature 40°C, L=1m), by way of example, presents a comparison of the following: the values $\alpha_{k[3]}$ and $\alpha_{k[4]}$ of the convective heat transfer coefficient determined using the Nusselt number, calculated on the basis of the literature, [3] (formulae (7), (8)) and [4] (formula (6)), respectively, for a channel of the rectangular cross section.

Table 1. Values of the convective heat transfer coefficient in the channel.

H [mm]	20	15	10	5
W [mm]	25	25	25	25
H/W [-]	0.8	0.6	0.4	0.2
w [m/s]	0.10	0.13	0.20	0.40
Re[-]	1850	2080	2370	2770
$\alpha_{k[3]}$ [W/m^2/K]	377	438	549	828
$\alpha_{k[4]}$ [W/m$^{2/}$K]	290	339	439	743
$\varepsilon = \frac{\lvert \alpha_{k[3]} - \alpha_{k[4]} \rvert}{\alpha_{k[4]}}$ [-]	0.30	0.29	0.25	0.11

Table 2 presents a comparison as above, prepared for a considerably shorter channel, of a length L=0.2m.

The results from Table 1 and 2 show approximately a 10-30% discrepancy in the values of the convective heat transfer α_k calculated using relationships from [3] or [4]. They also show a very large effect of "initial" phenomena in the channel (leading to local turbulences), which for a relatively short channels leads to a significant increase in the coefficient α_k. This also causes considerable differences in the values of α_k along the channel i.e. significantly greater values in the initial, inlet, section.

Table 2. Convective heat transfer coefficients in the channel, L=0.2m

H [mm]	20	15	10	5
W [mm]	25	25	25	25
H/W [-]	0.8	0.6	0.4	0.2
w [m/s]	0.10	0.13	0.20	0.40
Re[-]	1850	2080	2370	2770
$\alpha_{k[3]}$ [W/m^2/K]	641	747	936	1411
$\alpha_{k[4]}$ [W/m$^{2/}$K]	533	622	794	1276
$\varepsilon = \frac{\lvert \alpha_{k[3]} - \alpha_{k[4]} \rvert}{\alpha_{k[4]}}$ [-]	0.20	0.20	0.18	0.11

Execution of thermal calculations by means of the Flux® software, simulation calculation results

Simulation calculations of the temperature distribution in the radiator were executed by means of a commercial Flux® software [6], used for field FEM calculations of electromagnetic and thermal problems in the branch of heat conduction. When determining the values of the convective heat transfer, used in simulation calculations, relationships given in [4] were used, which usually lead to obtaining lower values of the coefficient than when using those used in [3], which – in the radiator under consideration – leads to higher values of the temperature of the elements cooled. Moreover, in simulation calculations, the problem of consideration of cooling water temperature change and the convective heat transfer coefficient along the radiator channel had to be solved.

In the model being considered, 10 zones of water temperature changes were assumed, as presented in Fig.2. The temperature ϑ_n in the n-th (in the direction of flow) cooling water zone is the result of energetic balance of the thermal flux $P_{i,\ convective\_losses}$ picked by means of convection from the radiator in each i-th (i=1÷n) zone and the accumulated flux $P_{a,n} = \sum_{i=1}^{n} P_{a,i}$ in the flowing cooling-water at a flow Θ and a temperature ϑ_{inlet} at the inlet, acc. to the relationship:

$$P_{a,n} = m_{water} c_{water} \Delta\vartheta_n = \Theta \rho_{water} c_{water} (\vartheta_n - \vartheta_{inlet}) \tag{10}$$

wherein for n-th zone $n \in (1,10)$:

$$\Delta\vartheta_n = \vartheta_n - \vartheta_{inlet} = \frac{\sum_{i=1}^{n} P_{i,convective\_losses}}{\Theta \rho_{water} c_{water}} \rightarrow \tag{11}$$

$$\rightarrow \vartheta_n = \frac{\sum_{i=1}^{n} P_{i,convective\_losses}}{\Theta \rho_{water} c_{water}} + \vartheta_{inlet}$$

where:

$$P_{i,convective\_losses} = \alpha_{k,i} (\vartheta_{radiator} - \vartheta_i) \quad for \ i = 1 \div n \tag{12}$$

With the approach presented above, the thermal flux intercepted by means of convection in particular zones depends on the water temperature and convective heat transfer coefficient of the zone. However, the water temperature in the zone is, at the same time, dependent on this flux, while the convective heat transfer coefficient depends on the water temperature. This interdependence makes calculations complicated, which leads to executing thermal calculations of the transient state with a shift by a time step of the effect of convective losses upon the water temperature and the convective heat transfer coefficient. However, with slowly-varying of the radiator temperature, this does not require to radically decrease the time step of calculations.

With regard to heat sources, the calculation model considered was reduced to – arranged on the outer radiator surface (on the side intended for mounting semiconductor elements) – the regions representing the radiator-semiconductor module transition zone (silicone paste). Material parameters of the regions were selected so that their thermal resistances perpendicular to the radiator corresponded to the catalogue values of the casing-module transition resistance. To the outer surfaces of these regions (which, taking physical aspect into account, represent outer surfaces of the modules) thermal fluxes equal to the values of the assumed power losses in the modules were assigned.

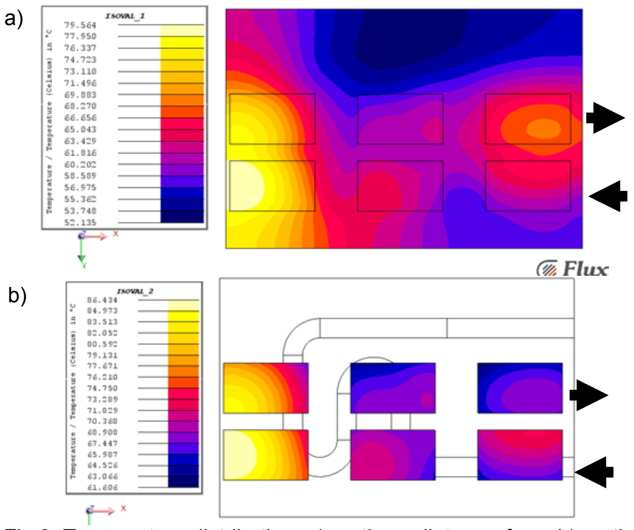

Fig.3. Temperature distribution: a) on the radiator surface, b) on the surface of casings of the IGBT modules cooled.

Taking into consideration both the cooling by means of the flowing water and the natural air cooling, in Fig.3a the temperature distribution on the radiator surface is shown for an exemplary arrangement of six IGBT SKM 200GB125D half-bridge modules, with 200 watt losses in each of the modules and the flow $\Theta \approx 3$ l/min of the cooling water of a temperature ϑ_{inlet}=18°C at the inlet, for the thermal state after 0.5h of the system operation. In Fig.3b the distribution of the surface of the casing of the modules being cooled, with the assumed catalogue value of the thermal resistance transition of the casing-radiator $R_{t,cs}$= 0.038 K/W, and the temperature distribution on the surface of the casing of the modules cooled are presented. For the assumed spatial arrangement of modules on the radiator, one can observe significant, reaching as much as 20°C, temperature differences on the surfaces of casings of the modules cooled.

Fig.4 how, in the system under consideration, both the value of the convective heat transfer coefficient and the value of the water temperature along the channel, i.e. for its successive zones (Fig.2) are changing.

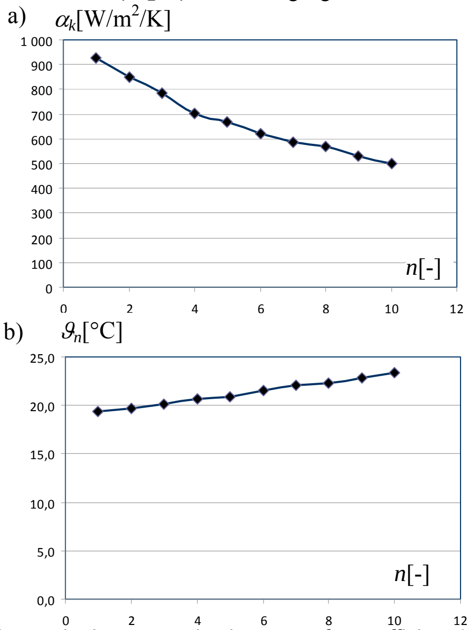

Fig.4. Change in the convective heat transfer coefficient (a) and the water temperature (b) along the channel length (for successive channel zones).

As can be seen in Fig.4, in particular, the value of the convective heat transfer coefficient is subject to a very large, nearly two-fold change. This is mainly caused by a change in the character of the fluid motion, which becomes increasingly laminar when it recedes from the water inlet zone. Taking into consideration the above-mentioned (10÷30)% differences in the values of convective heat transfer coefficients calculated acc. to the relationships from [3] and [4], the temperature sensitivity on the surface of casings of the semiconductor modules for a 20% increase of the calculated convective heat transfer coefficients was examined (which can, in a first approximation, correspond to the calculations using relationships from [3]).

In Fig.5 temperature distributions both on the radiator surface and on the surfaces representing the module casings are shown, obtained for the set up presented above, but when values of the convective heat transfer increased by 20% are assumed. The comparison of the distributions in Fig.3 and Fig.5 shows that this 20% increase leads only to approx. 6°C decrease in the maximum temperature on the casing of the modules cooled.

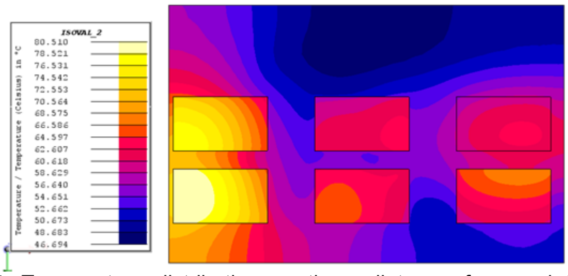

Fig.5. Temperature distribution on the radiator surface and the surfaces representing casings of the IGBT modules, calculated for the convective heat transfer coefficients increased by 20%.

Conclusions

The simulation calculations presented above demonstrate that with the use of the software for field heat transfer calculations, such as the Flux program, it is also possible to effectively analyse more complex problems, including convective heat transfer in closed systems. By taking into consideration the effect of the heat picked on the temperature of the cooling agent (water), it becomes possible to consider a change in the conditions of heat transfer along the cooling channel more effectively, with particular focus on the change in the convective heat transfer coefficient that depends to a large extent on the effect of water temperature upon its material parameters. This type of approach combining numerical calculations of 3D heat transfer and, based on the similarity theory, an analysis of convective heat transfer allows for fast and relatively accurate calculation of water-cooling systems in power electronic devices. An improvement in the accuracy of such calculations can possibly be found in the scaling of the calculated convective heat transfer coefficients based on the standardised thermographic measurements of temperature distribution for than exemplary radiator. Such analyses are envisaged at a further stage of the research conducted.

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***Autor**: dr hab. inż. Jerzy Zgraja prof. P.Ł., Politechnika Łódzka, Instytut Informatyki Stosowanej, al. Politechniki 11, 90-924 Łódź, E-mail: jzgraja@p.lodz.pl*

REFERENCES

[1] E. J. Dede, J. Jordan, V. Esteve, 'State-of-the Art and Future Trends in Transistorised Inverters for Induction Heating Applications'. Proc. of 5th IEEE International Caracas Conference on Devices, Circuits and Systems (2004), pp. 204-210

[2] M. Cesano, S. Etflandrin., G. Griffero, L. Natale, 'Design Optimization of a High Frequency Power for Induction Heating Application Supply'. International Symposium on Heating by Electromagnetic Sources HES-10, (2010), Padua, Italy, pp.419-424

[3] M. Hering –Termokinetyka dla elektryków. WNT, Warszawa, (1980)

[4] D. Staton, Motor-CAD v8.1 Manual, (2014)

[5] A.F. Mills, 'Heat Transfer', Prentice Hall, (1999)

[6] Cedrat, 'Flux User's Guide' v.12, (2015)