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Thermal Analysis of Radiator Under Natural and Forced Convection Conditions Using Numerical Simulation and Thermography

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Based on the physical model in COMSOL Multiphysics package, a 3D model of a radiator has been developed. Using the finite element method, a number of numerical simulations were carried out. The results of the temperature distribution on the radiator surface were obtained. Numerical methods are subject to some errors, so it is important to validate the results. In order to verify the correctness of the numerical model, a test stand equipped with a radiator, a heating resistor, a fan and a FLIR SC7600 thermal imaging camera was constructed. The heating resistor allowed the regulation of the heat flux by changing the DC voltage. Thermograms were obtained showing the temperature distribution on the radiator surface. It has been found out that it is possible to use thermovision in laboratory tests to validate numerical models. The validation of the numerical results was carried out by comparing the results of the numerical simulations with temperature measurements in the real conditions. Therefore, both techniques can be methodically used when designing new equipment.

1. Introduction

Convective heat transfer can take place in conditions of natural or forced convection. Natural convection occurs when the fluid movement (in the case of the air in question) is caused by the difference in density (due to the difference in temperature). Forced convection occurs when the fluid movement takes place under the influence of external forces in the form of a pressure difference created by, among others, fan. Determination of the value of heat transfer coefficient under natural and forced convection conditions is a complex issue. This issue can be solved through experimental or analytical means. It is also important to use numerical simulations. Wang et al. (2017) performed numerical simulations in the field of convective heat transfer for computer components. In this case, the results of numerical simulations have been confirmed with experimental results. The presented method of conducting scientific research, based on numerical simulations, can be used in optimization issues. Examination of the heat transfer conditions and then energy losses is important particularly in the efforts to reduce energy consumption by machines. It matters in to ensuring the healthy functioning of the world economies (Yong et al., 2016).

The aim of the authors of this paper was to ensure optimal cooling conditions for computer components. Regarding measurement techniques, thermovision can be used in experimental studies. The issues presented in this article concern the continuation of works on the use of thermal imaging carried out by Grabowski et al. (2016). This paper presents the results of tests that concern the heat exchange in conditions of free convection for the heat radiator. The authors validated the results of numerical simulations using thermovision for three heat flux values. This article is an extension of the research works presented.

Thermovision is a tool for checking the correctness of the numerical model. The numerical model of the radiator was made in the COMSOL Multiphysics package. This package has a wide range of practical applications. Despite the significant possibilities of numerical simulation software, the critical issue is the critical evaluation of the obtained results. In order to assess the accuracy and level of reliability of numerical simulations, they are verified and validated. In order to validate the numerical solution, a research stand was performed on which

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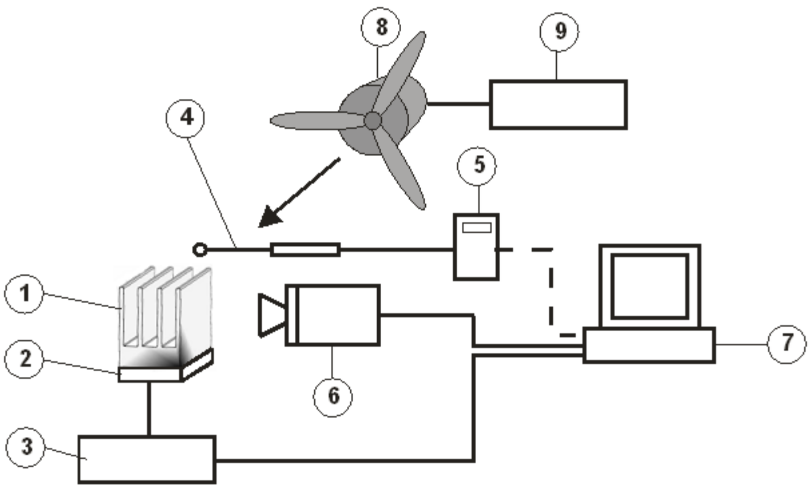
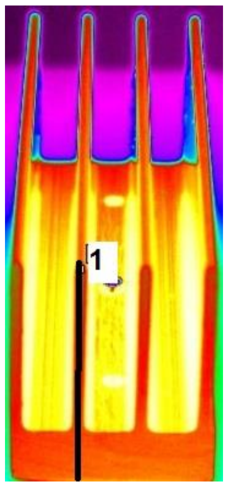
experiments using thermovision were carried out. The main elements of the stand were: radiator, fan and thermal imaging camera.

2. Thermographic investigation

Figure 1 shows scheme of stand and an example thermogram of radiator. Thermal power was provided by a heating resistor located under base of radiator. The length and width of the base of radiator is 70 mm and 26.6 mm, respectively. Radiator height is 33 mm, it has four fins 25.5 mm high. Regulation of the supplied thermal power was carried out using a high-current power supply. The power supply was controlled by a computer through a control program written in the LabView environment. VelociCheck Air Velocity Meters 8330 were used to determine the air velocity under forced convection conditions. The temperature changes were observed with the FLIR SC7000 camera, for which the NETD (Noise Equivalent Temperature Difference) parameter is 20 mK. The signal from the camera was sent to a PC in which thermograms were obtained for the tested radiator under specific heat exchange conditions. The emissivity of the radiator surface was determined experimentally by comparative measurement using a surface with a constant, known emissivity. Comparative measurement for the surface of the paint-covered radiator allowed to estimate the emissivity value at 0.96. Research using a thermal imaging camera was carried out at an ambient temperature of 22 ^{o}C. The obtained thermograms allowed to determine the actual temperature values on the radiator surface. The set of temperature values was used to determine the average temperature in specific places. Line 1 on the thermogram indicates the place from which data was collected. These data correspond to the temperature values in the direction along the height of radiator, i.e. from the top to the base.



(a)



(b)

Figure 1: (a) Scheme of the measurement stand. Markings: 1 Radiator, 2 Heater, 3 High current amplifier, 4 Probe, 5 Anemometer, 6 Thermovision camera, 7 PC, 8 Fan, 9 Fan power supply. The dashed line - manual transfer of information. (b) Thermogram of the radiator

The heat power values for which the tests were performed are shown in Table 1. They correspond to two cases for natural convection and two for forced convection. The velocity of the flowing air was measured at the inlet to the intercostal channels. The speed value was the same for each channel.

Table 1: Heat power values

Case	1	2	3	4
Thermal power, *Q* [W]	2.60	10.22	2.60	10.22
Air flow velocity, v [m/s]	0	0	1.05	1.05

Figure 2 presents the obtained data in the form of graphs.

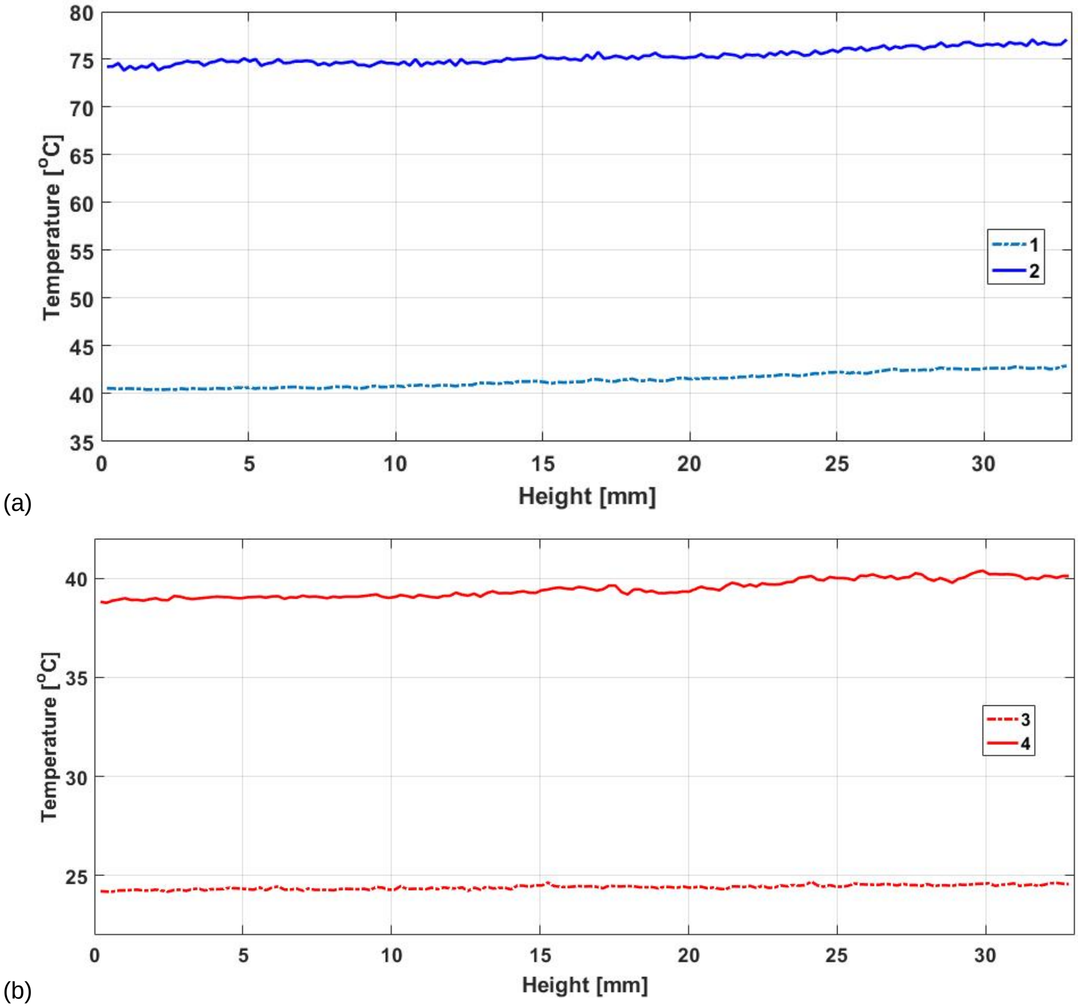


Figure 2: (a) Temperature distribution for natural convection. (b)Temperature distribution for forced convection

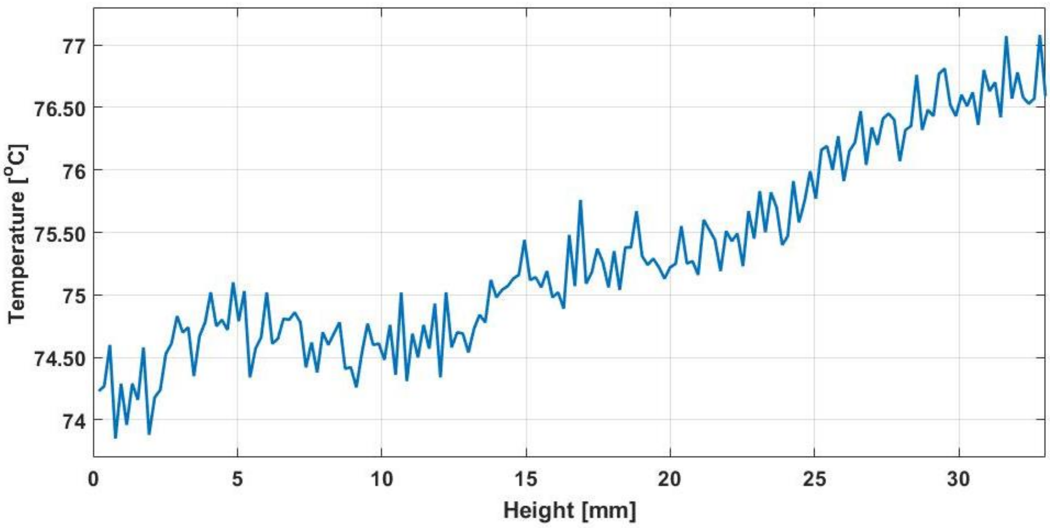


Figure 3: Temperature distribution for natural convection (case 2)

The camera used for measurements has a high resolution, which allows detailed data analysis for each case examined. Figures 3 and 4 show the temperature distribution for Cases 2 and 3 using the exact scale.
The obtained temperature data allowed to determine the value of heat transfer coefficient from the radiator surface to the environment. The values of these coefficients were necessary to carry out numerical simulations. They were used to define appropriate boundary conditions. The measured temperature values for the fins then allowed to validate the results obtained from numerical simulations. Validation concerns the study of the correctness of the model in the light of experimental research. The usefulness of thermovision in the validation process is presented on the example of the vacuum pump unit by Wernik (2017).

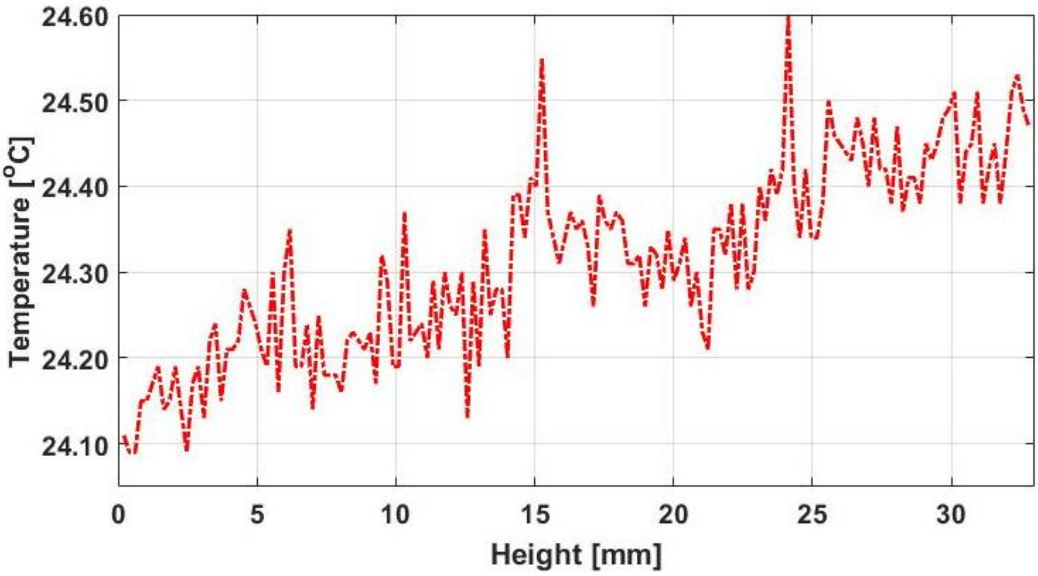


Figure 4: Temperature distribution for forced convection (case 3)

3. Determination of heat transfer coefficients

The determination of correct heat transfer coefficient is an easy task by no means, and researchers still continue to study those issues (Chow et al., 2015). The values of heat transfer coefficients can be determined analytically, depending on whether the conditions of natural or forced convection prevail.

3.1 Natural convection

The heat transfer is conditioned by gravitational air movement near the fins. There are many experimental methods that have been developed for a long time to allow the observation of the air temperature distribution near the fins. The air temperature decreases as the distance from the fin increases, while the speed increases. The general dependence on the heat transfer coefficient under free convection conditions is presented in the form:

$$h=f(\lambda, \eta, c_p, \Delta T, d, \rho, \beta, g) \tag{1}$$

where λ is thermal conductivity of air, η is air viscosity, c_p is specific heat, ΔT is temperature difference, d is characteristic dimension, ρ is density, β is coefficient of volumetric expansion, g is gravity of Earth.

The non-dimensional form of the h coefficient is the Nusselt number:

$$Nu = \frac{hd}{\lambda} \tag{2}$$

In order to determine the heat transfer coefficient, a dependence was assumed Churchil and Chu (1975):

$$Nu = a_0 + a_1 (GrPr)^m [1 + (a_2/Pr)^n]^{mr} \quad \text{for } GrPr \leq 10^9 \tag{3}$$

where Gr and Pr are dimensionless numbers of similarity, while constants a_0, a_1, a_2 and exponents m, n, r are determined on the basis of experience.
The values designated, among others using thermovision, heat transfer coefficients for selected investigated cases, were presented in Table 2.

Table 2: Values of heat transfer coefficients for natural convection

Case	1	2
Heat transfer coefficient, h [W/(m^{2}K)]	8.41	10.59

3.2 Forced convection

In the case of forced convection, the formula of Heiles can be used, which is presented by Staton and Cavagnino (2008):

$$h = \frac{\rho c_p D v}{4L}(1 - e^{-n}) \tag{4}$$

where parameter n is calculated from the formula:

$$n = 0.1448 \frac{L^{0.946}}{D^{1.16}} \left(\frac{\lambda}{\rho c_p v}\right)^{0.214} \tag{5}$$

where ρ is material density, c_p is heat capacity at constant pressure, v is inlet air velocity in the fin channels, L is axial length of cooling fins, D is hydraulic diameter (four times the channel area divided by the channel perimeter, including the open side).
The calculated value of the heat transfer coefficient for cases 3 and 4 is 17.38 W/m^{2}K. This is due to the fact that the value of the coefficient depends on the velocity of the air flowing. This value was the same in both cases.

4. Numerical simulations

The COMSOL Multiphysics package was used for the simulation. The numerical model contained 15957 tetragonal elements. The following boundary conditions have been adopted:
- heat transfer coefficient for aluminum,
- ambient temperature,
- heat flow flux density from the base of radiator for four cases,
- values of heat coefficients in accordance with the calculations presented above.

A series of numerical simulations were performed, changing the value of the density of the heat stream being conducted from the base of radiator. An exemplary simulation result for case 3 is shown in Figure 5. The average temperature along the fin height was 24.4 ^{o}C.

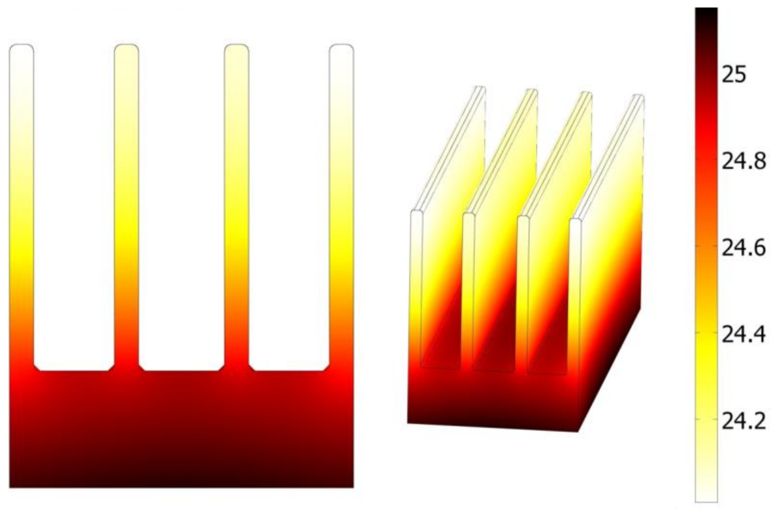


Figure 5: The temperature distribution at the radiator, case 3

It should be borne in mind that errors are the sum of errors in modelling and errors in numerical simulations. Comparisons between the results of numerical simulation and experimental results are made by analysing the so-called comparison error. The comparison error is defined as the difference of the value obtained in the experiment and the value obtained in the numerical simulation. The validation of the numerical simulation is achieved if the comparison error value is less than its uncertainty. At the same time, it is assumed that for the purposes of validation, the error should be less than 10 %. Error values, for the investigated cases, are in the range of 5-9 % which according to the literature of the subject allows to regard the obtained simulation results as satisfactory.

5. Conclusions

The purpose of the presented research was to validate the results obtained from numerical simulations using thermography. The thermovision technique for validation of numerical codes has been demonstrated. The tests were carried out for selected cases of heat exchange for natural and forced convection. Experimental measurements and numerical simulations for four thermal power values were made. Verified and validated numerical models allow to extend the scope of simulation. The increase in the heat transfer coefficient value for the radiator can be obtained by using forced convection. In this case, a well-adjusted fan can even reduce the thermal resistance of the cooling system in which the radiator is used by several times. In the analytical calculation of the heat transfer coefficient it is convenient to use the formula proposed by Heiles. Exceeding the velocity of flowing air in intercostal trunks above 7 m/s results in noise, the volume of which is noticeable. Therefore, in further studies on the influence of air velocity on the value of the heat transfer coefficient, the value of 7 m/s will be accepted as the maximum. This study presents a method for determination heat transfer coefficient for radiator under forced convection conditions. Which is why it broadens the state-of-the-art in the scope of investigations of small radiators and make it possible to quickly estimate heat transfer conditions.

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