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Investigation of Heat Loss from the Housing of Compressor

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Based on the physical model in COMSOL Multiphysics package, a 3D model of the compressor has been developed. Heat is generated mainly by the electric motor and is discharged the air that surrounds the device through free convection from the housing and radiator.

The experimental method (laboratory test) consisted of temperature measurements of the compressor components with the FLIR SC7000 thermal imaging camera. Heat losses were calculated using literature derived correlations. Heat transfer coefficients were determined by through the experimental study. The determination of convective heat transfer coefficients was of particular importance in defining the appropriate boundary conditions for the simulations.

The results of the numerical simulations were compared with the results of the thermovision analyses. The use of thermovision analyses allows for validation of the simulation results. In the result, heat losses were determined.

1. Introduction

Analysing the heat transfer that takes place in convection conditions, it should be noted that it is necessary to consider the equations of fluid motion. For this reason, convection analysis is much more difficult than in the case of heat conduction alone. In the case of free convection, the cause of fluid movement is the difference in density that arises due to the difference in fluid and wall temperature. One of the most difficult tasks is to determine the value of heat transfer coefficient. This is done:

- theoretically (in a few cases of laminar flow),
- experimentally (by building appropriate models),
- by experimental and computational method.

If the experimental method is used, the temperature distribution for the actual test object should be determined. Then one can determine the heat transfer coefficient from dependence:

$$h = -\frac{k}{\Delta T}\left(\frac{\partial T}{\partial n}\right) \tag{1}$$

where h is convective heat transfer coefficient, k denotes thermal conductivity, ΔT is difference in average wall and air temperatures, $\frac{\partial T}{\partial n}$ is temperature derivative along the direction normal to isothermal surface.

A numerical simulation can be used in a methodical approach that uses real-world and analytical calculations. In scientific research, numerical methods are widely used in confirming the proposed theoretical models. In many cases, numerical simulations can replace experiments, but they can also complement the studies. Numerical simulations are a convenient tool for studying heat transfer in various industrial applications. Mikulčić et al. (2017) presented the use of simulation in the cement industry. Wang et al. (2017) performed numerical simulations in the field of convective heat transfer for computer components. It shows a wide range of applications of numerical simulations in various fields. In the tests of compressors and pneumatic systems, various methods are used, as presented by Du et al. (2017).

The main purpose of this article is to present the method for determining heat losses by determining the value of convective heat transfer coefficients from the housing of operating compressor to the environment. Empirical dependencies, thermography and numerical simulations were used for the research. The tests were carried out

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on a compressor in which heat is mainly generated by an electric motor and discharged into the air surrounding the device through convection from the body and radiator. The studied phenomenon was conduction of heat through the compressor housing, including a radiator with straight fins placed on it. An experimental station was built to allow the use of thermovision. The usefulness of thermovision in the validation process is presented on the example of the vacuum pump unit by Wernik (2017). Thermovision is a very useful measuring technique to determine the heat transfer conditions. Based on the actual object, a 3-D model of the compressor was made on which numerical simulations were carried out. Finite element method (FEM) was used as one of the basic tools of computer-aided research and engineering analysis, with a very wide range of applications. It should be borne in mind that the results obtained using FEM are always subject to a certain error, hence it is essential to verify the results. Examples of the use of FEM in the field of heat transfer can be found in the work presented by Liu and Quek (2014).

Finally, examination of the heat 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).

2. Research object and stand

In Fig. 1 the general appearance of the test stand with the tested compressor was presented. Temperature changes on the housing surface were observed using 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 at specific heat transfer conditions.

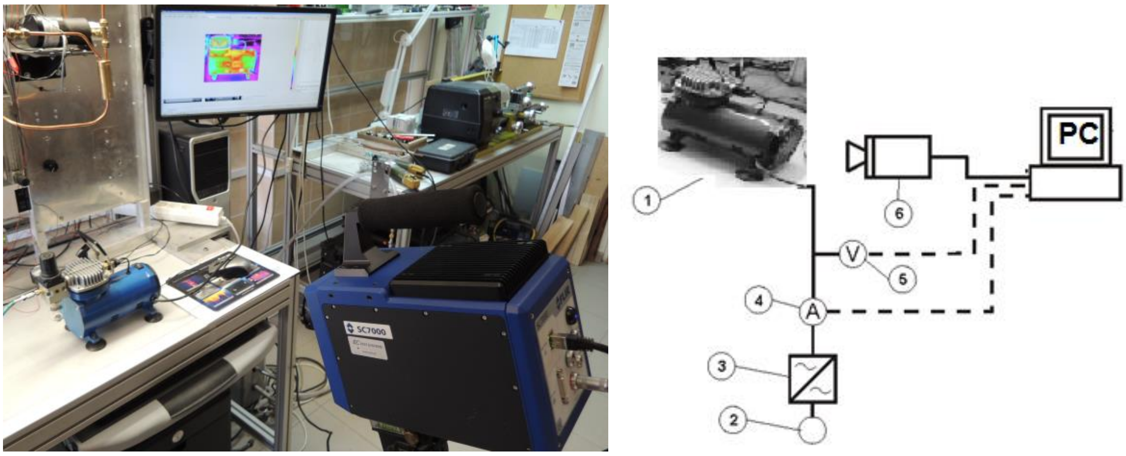

Figure 1: Scheme of the measurement stand. Markings: 1. Compressor, 2. AC power, 3. Inverter, 4. Amperemeter, 5. Voltameter, 6. Thermovision camera. The dashed line represents manual transfer of information.

Basic compressor operation parameters are:

- maximum compressor discharge pressure 0.4 MPa,
- power of the compressor's electric motor 140 W.

The first stage of experimental determination of compressor heat losses was to conduct a laboratory test. The test consisted in starting the compressor and leaving it in operation until the conditions for heat transfer were established. The heat generated in the compressor was given to the surrounding air by means of free convection. The compressor operated for 30 minutes. Then measurements were taken with a thermal camera to determine the temperature distribution for individual compressor elements. Measurements of two components of the compressor were made: housing and radiator. The compressor housing receives heat from the internal electric motor. The radiator is designed to remove heat generated during air compression.

3. Thermographic investigation

Tests using a thermal imaging camera were made at an ambient temperature of 20 °C. The housing is made of aluminum covered with blue paint. The emissivity of such an element is 0.8. The radiator was made as an aluminum cast and its outer surface is matt. The emissivity of the radiator surface is 0.97. Selected thermograms are shown in Figure 2. The average temperature of the body is 49.19 °C, while of the radiator 38.63 °C.

Figure 2: Thermogram of the compressor: a) side view b) upper view

Thermographs obtained in the tests allow to calculate compressor heat losses based on literature dependencies. It can be assumed that the compressor body is a hollow cylinder with a length of 180 mm and a diameter of 100 mm. To calculate the convective heat transfer coefficient, the value of the Nusselt number should be determined. Dependence on it is conditioned by the nature of the fluid flow, which can be known by the Rayleigh number. To do this, it is necessary to determine the Grashof and Prandtl numbers, the characteristic dimension in this case being the diameter of the body d. In this case, the Grashof number is 6.16·106, while the number Rayleigh 3.91·106. In this case, the Nusselt number relation due to Michiejew (Wisniewski and Wisniewski, 1994) have been chosen:

$$Nu = CRa^c \tag{2}$$

where the *C* and *c* coefficients depend on the value of Rayleigh number and are 0.54 and 0.25, respectively
Knowing the averaged value of the Nusselt number, equal to 24.01, one can calculate the convective heat transfer coefficient *h:*

$$h = (Nu\,k)/d \tag{3}$$

where *k* is thermal conductivity of air.
The calculated convective heat transfer coefficient for the housing is 6.79 W/(m^{2}K).
Determining the convective heat transfer coefficient for the radiator is a more difficult issue. This is due to the more complex construction of the radiator. It is also not a typical radiator, which would consist of a base on which fins of the same shape and dimensions are placed. On the basis of the radiator there are 11 fins, the

shape of which is to be taken as a rectangular cross-section. The fins have an identical small thickness of 3 mm (thus can be negligible), however, they have different lengths w and height L. Therefore, a mean fin length of 67 mm was used for further calculations. The average fin height can be determined from the dependence:

$$L = A_f/(11w2) \tag{4}$$

where A_f means the total surface area of the fins (calculated for both sides of fin).

The determined average fin height is 19 mm. Knowing the aforementioned dimensions, one can proceed to further determination of convective heat transfer coefficients: h_f for fins and h_b for the radiator base. Fins should be treated as vertical plates located next to each other. A certain simplification has been applied in the calculation, which considers a single vertical plate submerged in a fluid, with a height equal to the average fin height and the fin thickness. Simplification will not significantly affect the results because the radiator is small, and the fins are short. As with calculations for the compressor housing, determine the Nusselt number, depending on Rayleigh numbers. The parameters of the heat receiving fluid are identical to those in the heat transfer considerations between the housing and the air. The characteristic dimension is L. The Grashof number is 2.74×10^4, Rayleigh number is 1.7×10^4. For a vertical board, one can use the dependence on the Nusselt number in the form:

$$Nu = 0.63Ra^{1/4} \tag{5}$$

The calculated Nusselt number is 11.41. Using the relationship (3) and adopting L as the characteristic dimension, the convective heat transfer coefficient for the fins was determined. Its value amounted to 7.19 W/(m^{2}K).

The same can be done by determining the convective heat transfer coefficient for the radiator base. The average length of the side of the radiator base b is 79 mm and is the characteristic dimension in this case. The assigned Grashof and Rayleigh numbers of are respectively 1.97×10^6, 1.25×10^6. In order determine the Nusselt number, one can use the McAdams dependency for a horizontal square plate heated from the bottom:

$$Nu = 0.27Ra^{1/4} \tag{6}$$

The designated Nusselt number is 9.02. Using the relationship (3) and adopting b as a characteristic dimension, the convective heat transfer coefficient for the fins was determined. Its value amounted to 3.23 W/(m^{2}K).

Knowing the value of convective heat transfer coefficients for the fins and radiator base, one can proceed to determine the convective heat transfer coefficient for the entire radiator h_r from the dependence:

$$h_r = \eta h_f \frac{A_f}{A_d} + h_b \frac{A_i}{A_d} \tag{7}$$

where η means the efficiency of the fin, A_i means the surface area between the fins, A_d means the expanded surface area (this is the sum of the surface area between the fins and the surface area of the outer fins), $A_d=A_i+A_f$.

The fin efficiency present in the Eq(7) can be determined by several methods - either analytically or read from the tables. The calculated efficiency of the fin is 0.99. Knowing all the necessary data, one can determine the convective heat transfer coefficient to the air from the radiator from the Eq(7). In the described case h_r =5.84 W/(m^{2}K).

Taking into account the determined convective heat transfer coefficients for the radiator and housing, one can calculate the total amount of heat lost to the environment. The results are presented in Table 1.

Table 1: The amount of heat lost to the environment

Part	Housing	Radiator
Heat flow [W]	11.19	2.06
Convective heat transfer coefficient [W/(m^2 K)]	6.79	5.84
Average temperature [K]	322.34	311.78

4. Numerical simulations

In order to determine the lost heat flow conducted through the casing and radiator of the compressor, an approach was developed which compares the results of measurements taken at the stand with numerical simulations. The COMSOL Multiphysics package using FEM was used for numerical calculations. The governing equation can be presented as:

$$\rho C_p \frac{\partial \mathrm{T}}{\partial \mathrm{t}} - \nabla \cdot (k \nabla T) = Q \tag{8}$$

where ρ denotes material density, C_p means heat capacity at constant pressure, Q means heat generation in the source or heat consumption in the sink.

The geometry of the numerical compressor model was made using the CAD tools of the COMSOL Multiphysics program. In simulations as limit conditions, the following values were adopted: values of convective heat coefficients determined by means of thermovision. All outer surfaces of the casing in contact with the surrounding air are cooled by means of free convection. It can be reflected by a boundary condition:

$$-(k \frac{\partial T}{\partial n}) = h(T_{\mathrm{inf}} - T_{\mathrm{amb}}) \tag{9}$$

where T_{inf} is surface temperature, T_{amb} is ambient temperature.

Figure 3 illustrates the development of the numerical model of the compressor.

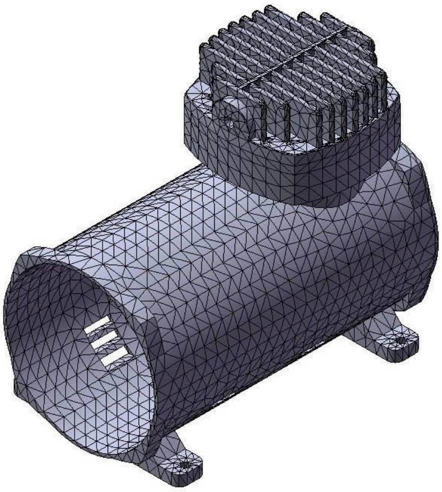

Figure 3: 3-D model of compressor

As the surface to which heat is applied, the inner surface of the housing was selected. The heat flow was taken from Table 1. In Figure 4, the temperature distribution on the external surfaces of the compressor representing the results of the numerical simulation is shown. The temperature distribution obtained numerically, in comparison to the thermovision test, has a higher temperature value for the housing and similar for the radiator.

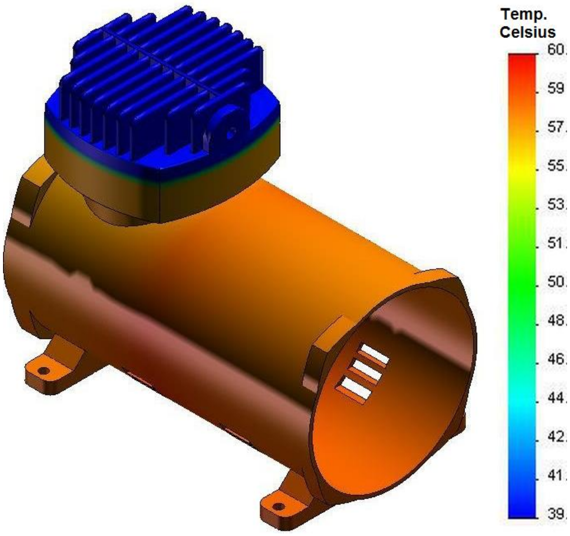

Figure 4: The temperature distribution obtained numerically

5. Conclusions

The aim of the presented research was to examine the heat transfer conditions during the compressor operation. The temperature distribution for the compressor housing and radiator was determined by means of thermovision. Using the obtained temperature results, the values of convective heat transfer coefficients were determined (based on literature correlations). As a result, it allowed to determine heat losses. During the tests, the power absorbed by the compressor was also measured, which allowed to determine that 12.6 % of the power supplied to the engine is a loss in the form of heat discharged to the outside.

The thermal imaging test is also a convenient tool for checking the results obtained during numerical simulations. Preliminary numerical simulations were carried out using the COMSOL Multiphysics computer package. The convective heat transfer coefficient determined using thermovision were used as accepted boundary conditions. At the further stage of testing, simulations should be carried out, eliminating some simplifying assumptions and validating the numerical model of the compressor. In the numerical model the piston, the motor and the layer of lacquer, which covers the compressor housing, have not been taken into account. The numerical model has been also simplified by applying heat flux on the internal surface of the compressor. The flux modeled internal heat coming from thermodynamic gas conversions and compression work.

Nomenclature

A_f	total area of the fins, m^2
A_i	area between the fins, m^2
C_p	heat capacity at constant pressure, J/(kg K)
d	characteristic dimension, m
h	convective heat transfer coefficient, W/(m^2 K)
k	thermal conductivity, W/(m K)
L	height of the fin, m
Q	heat generation/consumption in a source/sink, W
T	temperature, K
T_{amb}	ambient temperature, K
T_{inf}	external surface temperature, K
w	length of the fin, m
Greek symbols	
ρ	density, kg/m^3
η	efficiency of the fin

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