http://doi.org/10.35784/iapgos.2103

METHOD OF DETERMINING THE COP COEFFICIENT FOR A COOLING SYSTEM

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Abstract. *The basic metrological problem analyzed in this article is the experimental determination of the COP coefficient for a refrigerating system working in a stationary cooling chamber. This issue is because manufacturers of chillers provide the value of this coefficient in device technical data. In practice, after installing the refrigeration unit, the COP is often lower. The paper presents a method of balancing cooling energy for a complete refrigeration system.*

Keywords: coefficient of performance, cooling system, inverter compressor

SPOSÓB WYZNACZENIE WSPÓŁCZYNNIKA COP DLA UKŁADU CHŁODNICZEGO

Streszczenie. *Podstawowym problemem metrologicznym, który autorzy pragną przedstawić jest eksperymentalne wyznaczenie współczynnika COP dla układu chłodniczego, pracującego w stacjonarnej komorze chłodniczej. Zagadnienie to jest istotne, ponieważ producenci agregatów chłodniczych najczęściej podają współczynnik COP dla agregatu chłodniczego. W praktyce po zamontowaniu agregatu chłodniczego w komorze chłodniczej współczynnik COP jest niejednokrotnie niższy. Trudność z wyznaczeniem współczynnika COP w układzie chłodniczym polega na konieczności zbilansowania mocy chłodniczej dla całego układu wraz z komorą chłodniczą. W pracy przedstawiono sposób bilansowania energii chłodniczej dla kompletnego układu chłodniczego oraz opisano budowę stanowiska do wyznaczania współczynnika COP.*

Słowa kluczowe: współczynnik COP, układ chłodniczy, sprężarka inwertorowa

Introduction

The currently manufactured refrigeration systems commonly apply inverter compressors [2]. This solution is applied in systems with direct heat transfer of the cooled air (condensing chillers), as well as in systems comprising intermediate liquid (refrigerant aggregates) [1]. In practice, various types of compressor units are used, in which the heat of condensation can be transmitted to the environment or applied for generation of useful energy [3].

The principle of operation of a typical refrigeration system is shown in Figure 1. The refrigeration system consists of two heat exchangers (evaporator and condenser), an inverter compressor, and an expansion valve.

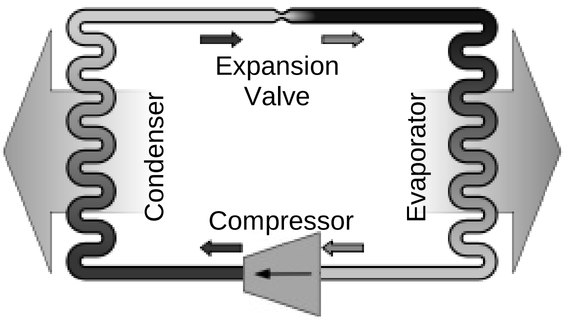

Fig. 1. Principle of operation of a typical refrigeration system

The main element in the operation of a refrigerator is a fluid, called a refrigerant or, occasionally, a working fluid. In a refrigerator, the refrigerant forms the "circuit" that allows heat transfer from the cooler region to the warmer region.

A compressor, using energy from the wall outlet, compresses the gas adiabatically. This raises its temperature and forces it into the condenser. As the compressed gas moves into the condenser coils, its temperature is higher than that of the surrounding air. Thus, heat is transferred out of the refrigerant and is taken away by the air. The refrigerant is condensed during this process, so it is in its liquid state as it moves on to the next stage. The condensed liquid is forced back into the refrigerator through an expansion valve. The active cooling agent is forced into evaporator coils where a sharp drop in pressure causes the refrigerant to revert to a gaseous state. During this process, the refrigerant absorbs a significant amount of ambient heat. This causes the cooling of the chamber with the evaporator. Finally, the "circuit" is closed; the refrigerant enters the compressor again.

The efficiency of the cooling units is greatly influenced by the selection of its individual components. The efficiency of the refrigeration unit is defined in terms of the COP (Coefficient of Performance). It is usually determined for several temperatures corresponding to the operating cycles of the compressor unit. Its value is the ratio of heating capacity (transferred in the condenser) to the use of electricity by the compressor:

$$\mathrm{COP} = \frac{Q_k}{P} \tag{1}$$

where: Q_k – heating efficiency of the condenser [W], P – the electrical power needed to achieve cooling capacity Q_k [W].

The use of a cooling capacity control system forms an indispensable element in refrigeration systems, the duration of products at a specified temperature is defined on its basis [4]. This is a necessary element of a refrigeration system [5], although it leads to the decrease of the value of COP. The paper presents an approach taken to determine the COP coefficient for a refrigeration system coupled with a dedicated control system. This solution provides a tool capable of determining the COP coefficient for the cooling system and not only for the chiller itself. This value more precisely corresponds to the actual cost of operating a refrigeration system.

1. Experimental setup

An experimental setup (Fig. 2) was developed and built for the purposes of the present study, the main element of which comprised a chiller manufactured by the Polish company FRIGIPOL, fed with an inverter compressor with a capacity of 480 W, and powered by 48V dc. As a result of implementing such a solution, it was possible to adjust the compressor capacity by varying the value of the input voltage. The experimental setup applied in the study also included a measuring chamber ($V = 12.41\ m^3$), which was in the form of an industrial container.

The container was placed in an empty space outdoors, as the present study envisaged an investigation of the performance of the investigated chiller in conditions similar to its natural operation. A tank filled with 0.75m3 of water was placed inside the container. This water tank acted as a thermal energy accumulator, as it simulated the material subjected to cooling. The setup was equipped with a data acquisition system that was developed in the LabView environment, which provided constant monitoring and recording of a number of physical parameters during the research.

In order to verify the correct operation of the laboratory stand, tests were carried out to cool the chamber interior under changing environmental conditions. For this purpose, several test cycles of cooling the inside of the chamber were carried out. Each time, the tests were started with a tank filled with water at a temperature of about 6°C. An example of the temperature measurement results for a compressor rotational speed of 3750 rpm is shown in Figure 3. The presented characteristics omit the results obtained from the T2 sensor placed inside the chamber, because the air temperature inside the chamber only slightly differed from the water temperature in the tank.

T4 temperature changes on the outside of the chamber are caused by the daily temperature changes that prevailed during the experiment. Along with these changes, the T1 temperature on the inner wall of chamber also changed. This proves that during the measurement, a significant part of the energy of the refrigeration system was lost due to conduction through the walls of the chamber. Uniform cooling (T3) of the water in the tank proves a properly selected capacity, which guarantees the stability of the process. As it results from the conducted research, the fluctuations in the outside temperature do not significantly affect the drop in water temperature. However, energy losses due to conduction through the chamber walls should be included in the power balance when calculating the COP.

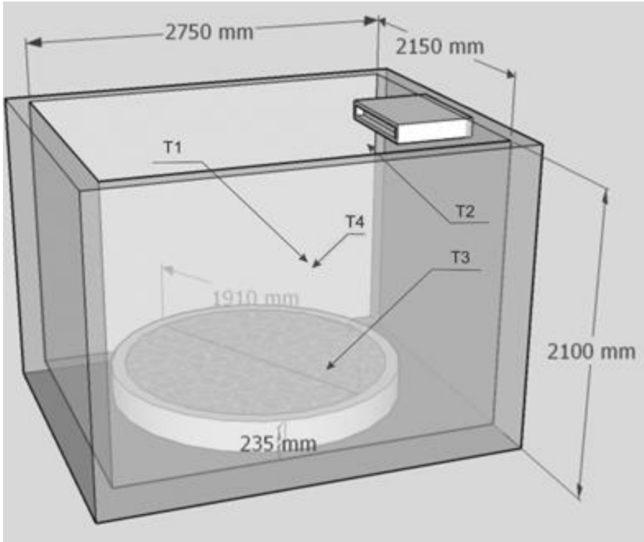

Fig. 2. Test stand

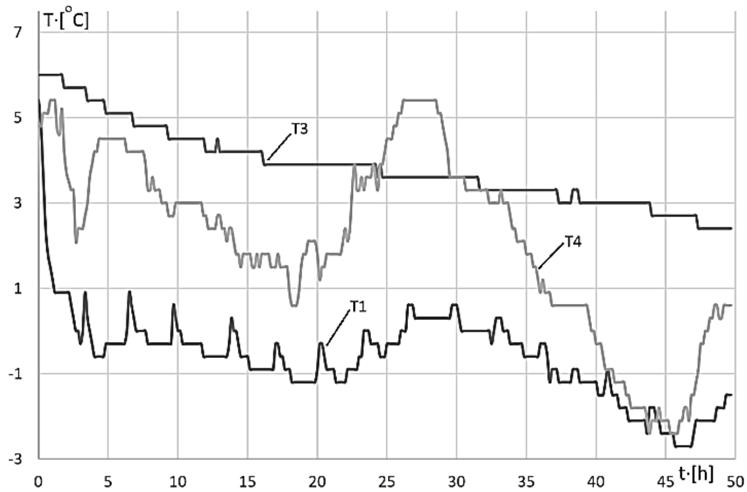

Fig. 3. Temperature characteristics of the cooling process

2. Laboratory determination of COP

For the purposes of determining the COP of a cooling system, it is necessary to know the power P_a that has been input from the refrigeration system. For this purpose, the power balance of the system was developed. The basic power sources are: thermal power delivered from the water tank P_w and thermal power received from the air inside the measuring chamber P_p. The heat conducted through the walls of the container cannot be underestimated Q. On this basis, the power balance equation was derived in the form:

$$P_a = P_w + P_p + Q \tag{2}$$

The values of power delivered from the water tank and the power derived from the air inside the chamber are calculated on the basis of the measurements from the relations in (4) and (5). The power transmitted through the walls of the cooling chamber is calculated on the basis of the conduction law on the basis of equation (6). The formula in (6) accounts for a heat transfer coefficient whose precise determination poses some difficulties. This is due to the design of the cooling chamber as insulation (thermal bridges) is not homogeneous on the entire surface of the chamber.

In addition, the heat supply by convection to the outer surface of the chamber is not the same in all cases. Further, variable weather conditions result in some differences in the heat transmission to the container surface, which results in a non-isotropic temperature distribution over the surface of the container. Curve T3 in Figure 4 represents the variation in water temperature during the cooling process. On the basis of this characteristic, it is possible to derive water cooling gradients for selected time intervals. The characteristics of changes in the temperature difference between sensors T1 and T4 forms the basis for calculating the heat transfer coefficient. Since the calculation of the power conducted by the walls of a chamber poses some difficulties and would normally require the application of a considerably more complex measurement system, the decision was made to calculate an averaged value of the heat transfer coefficient. A series of tests were carried out involving the determination of similar characteristics for various capacities of cooling by controlling the rotational speed of the compressor in the range from 3000 to 4400 rpm. On the basis of several characteristics, four time ranges were identified corresponding to occurrence of stable conditions.

In the case of compressors driven by DC inverters, the electrical power consumed by the compressor strongly depends on the rotational speed, which affects the cooling efficiency. The dependence of the consumed power on the rotational speed for the compressor used is shown in Figure 5.

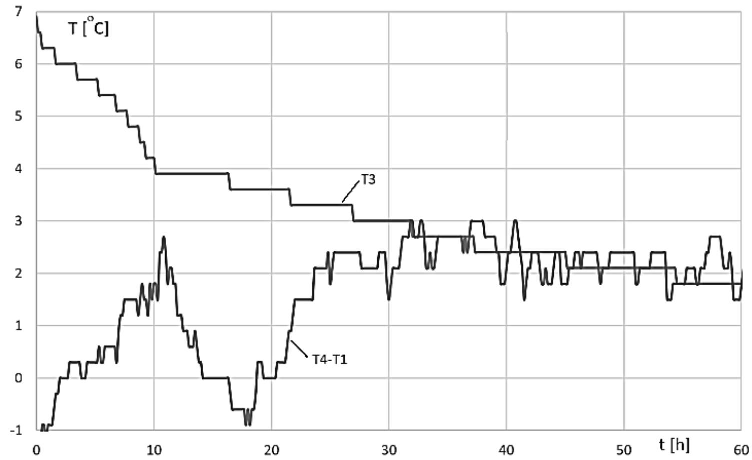

Fig. 4. Temperature characteristic of the cooling system

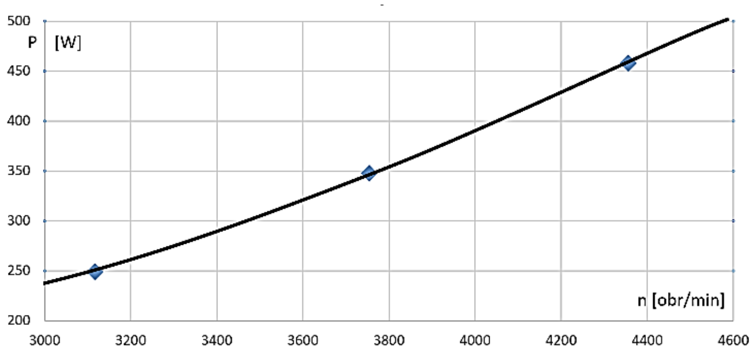

Fig. 5. Characteristics of changes in electric power consumption depending on the rotational speed of a compressor driven by a DC inverter motor

Table 1. Plunger material characteristics

Measurement range	Time	ΔT	P_W	P_p
	h	°C	W	W
1	42–55	2.1	0.52	0.16
2	30–35	8.1	1.2	0.94
3	14–20	2.4	0.3	0.02
4	41–46	0.3	0.3	0.24

The stabilization of the process is evidenced by the fact that the temperature intervals measured on the wall of the chamber do not change significantly for a few hours even in the conditions when an instant drop in the temperature of the cooled water occurs. As a result, temperature gradients for the cooled water were determined and the mean value of the differential temperature on the wall of the measuring chamber was recorded for these ranges. Consequently, on the basis of the water temperature gradients, the power delivered from both the water and air inside the chamber could be derived. A summary with the results of measurements and calculations is found in Tab. 1

In order to determine the averaged value of the heat transfer coefficient, in this study, it was assumed that the thermal heat transfer coefficient is constant and does not vary depending on the method of cooling. On the basis of this assumption, the following power balance equation can be developed:

$$P_{w1} + P_{p1} + Q_1 = P_{w2} + P_{p2} + Q_2 \qquad (3)$$

Subscripts 1 and 2 relate respectively to selected time intervals. Using the data summarized in Table 1, it is possible to develop six independent equations, which can be applied to calculate six values of the heat transfer coefficient by transforming the equation in (3).

The power that was extracted from the water and air could be calculated using the following formulae:

$$P_w = V_w \cdot \rho_w \cdot c_w \cdot \frac{dT_w}{dt} \qquad (4)$$

$$P_p = V_p \cdot \rho_p \cdot c_p \cdot \frac{dT_p}{dt} \qquad (5)$$

where: V_w, V_p – water and air density [m^3], ρ_w, ρ_p – water and air density [kg/m^3], c_w, c_p – specific heat of water and air [J/(kg·K)], T_w, T_p – water and air temperature [K], t – time [h].

Since the temperatures outside and inside the chamber were considerably different, it was necessary to determine the energy that is transferred through the chamber walls. The energy transferred through the chamber walls was derived on the basis of the Fourier law:

$$Q = \lambda \cdot A \cdot \Delta T_s \qquad (6)$$

where: A – surface area of the container wall, λ – heat transfer coefficient [W/(m^{2}K)], ΔT_s – temperature difference along the inside and outside interfacial surface of the container measured by the T1 and T4 sensors. The formula representing the heat transfer coefficient assumes the following form:

$$P_w = V_w \cdot \rho_w \cdot c_w \cdot \frac{dT_w}{dt} \qquad (7)$$

The results of calculations for various combinations of the measurement points are presented in Fig. 3. The majority of the results are found in the range that is close to λ = 0.89 W/(m^{2}K). The values corresponding to points 5 and 6 are not taken into account in the further calculations due to the fact that they were considered to be considerable errors. The reasons for their occurrence are associated with the small differences in the values of powers and temperatures adopted during the calculations of the heat transfer coefficient.

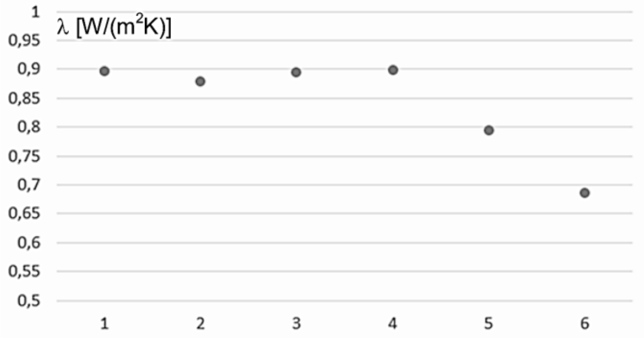

Fig. 6. Six values of heat transfer coefficient

The standard uncertainty was derived on the basis of the formula:

$$U_{\lambda}=\sqrt{\frac{(\bar{\lambda}-\lambda_i)^2}{N(n-1)}} \tag{8}$$

where: $\bar{\lambda}$ – the arithmetic mean derived from the value of the heat transfer coefficient, and λ_i – its particular component terms. The level of uncertainty after the results are extended to include the confidence level $p = 95\%$ was equal to $U_{\lambda} = 0.098$ W/(m^{2}K). This result can be considered as satisfactory and adequate for engineering applications.

The examples of the calculated results of COP for the refrigeration system used are given in Table 2. As we can see from these calculations, the COP coefficient for the entire analysed refrigeration system is slightly higher than one. In comparison to the values of this coefficient for refrigeration units given by manufacturers, the calculated value is about two times lower. The basic reason is associated with the process of determining COP in which manufacturers of chillers do not take into account the characteristics of the cooling system and only determine the thermal power of the condenser, which leads to a significant overestimation of the results.

Table 2. Results of calculated power

Compressor speed	Electric power input	Cooling power	COP
pmr	W	W	-
4356	458.2	617.6	1.35
3753	338.1	447.9	1.32
3118	249.6	333.5	1.34
4356	458.2	617.6	1.35

3. Conclusions

The work discusses the manner in which the COP coefficient of a refrigeration system can be feasibly determined. As was shown, the values of COP for the cooling system are significantly lower than the values of this coefficient determined for the chiller. The presented approach provides the means to assess the cooling parameters of a refrigeration system, and can offer much better insight in terms of assessing the efficiency of cooling equipment than just a comparison of the COP coefficients in commonly applied chillers. The use of some simplifications in determining the mean heat transfer coefficient does not lead to a significant level of error, which can be evidenced by the uncertainty level of 10% of the value calculated in the present study. Thus, the solution discussed in this paper can be applied in the study of complex refrigeration systems and in assessing the power consumption of these systems.

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otrzymano/received: 02.08.2020 *przyjęto do druku/accepted: 10.12.2020*