TEKA Kom. Mot. Energ. Roln. – OL PAN, 2008, 8, 99–105

ENERGY EFFICIENCY OF SMALL, COMPRESSOR ASSISTED AIR-WATER TYPE HEAT PUMPS

Jarosław Knaga

Agricultural University in Cracow, Departament of Power Engineering

Summary. The paper presents the results of testing into the OW 150 small, compressor-assisted air-water type heat pump, dedicated for preparing utility hot-water. The detailed analysis performed into the results of tests has shown that the energy efficiency factors change within the range 1.4÷2.6 in the course of water heating cycle in the vessel. The tests performed under operation conditions revealed that this factor is 40% lower than the factor as determined under laboratory testing.

Key words: energy, heat pump, source of heat, energy efficiency factor, utility hot-water.

INTRODUCTION

In the last few decades, the constant, growing trend of electricity prices has been seen in highly developed countries and developing countries, which is also noticeable in Poland. The fact that electricity is more and more expensive becomes the driving force for the development of alternative energy equipment. One of the basic appliances allowing for a more efficient use of energy is the heat pump. The heat pump market had already developed in the previous century, where, only in Japan, two million of heat pumps were sold in 1984 [Lewandowski 1996, Wołoszyn 1991]. Currently, the annual production of heat pumps all over the word amounts to several million. In Western Europe only, 154 companies deal with their production or sales [Kubski in. 1994, SeCes-Pol 1998, Zimny in. 1997]. The number of these companies has been growing also under the conditions of the Polish market [Lewandowski 2006].

The companies manufacturing, inter alia, compressor-assisted heat pumps perform research into these appliances in the laboratories, whose objective is mostly the optimization of subassemblies, as well as the very conditions of operation. These tests are frequently performed under ideal conditions (the simulation of upper and lower source of heat) and under steady operating parameters [Knaga 2007, Rubik 1996, Zimny in. 1998]. The energy efficiency factor of the compressor-assisted heat pump is frequently oversized under such conditions and therefore may not be transferred to the conditions of actual operation. Hence, the need for performing testing into performance may be seen, which will allow for a determination of the actual energy efficiency for such type of appliances.

OBJECTIVE

The objective of the report has involved the performance of operating tests on an example of a small, compressor-assisted heat pump of air-water type, manufactured by Eda Poniatowa. The obtained results of tests served the elaboration of operating performance characteristics, describing the parameters of the bottom and upper source of heat. This has given the grounds for elaborating the model describing the energy efficiency of this heat pump.

SUBJECT MATTER AND METHODOLOGY OF TESTING

The subject matter of tests has been the OW 150 compressor-assisted heat pump of air-water type, with the rated power of compressor 320 W. This pump is equipped with the utility hot-water vessel with volume 150 dm3, where the exchanger of the upper source of heat is located. The lower source of heat is the air-refrigerant exchanger, whose refrigerant circulates in the sealed system. The compressor-assisted pump, including vessel and the exchangers, makes up the integrated, sealed structure of water heater, whose entire specification has been provided in Table 1.

Table 1. Compressor-assisted heat pump specification

Parameter	value
Water vessel volume [dm^3]	150
Usable water vessel volume [dm^3]	129,5
Utility hot water temperature control range [^{O}C]	30÷65
Water temperature obtained from heat pump operation [^{O}C]	55
Heating capacity of heat pump [W]	1000
Electric heater capacity [W]	1500

Operation testing of the small, compressor-assisted heat pump was performed at the laboratory stand in the Agricultural Energy Faculty, Agricultural Engineering Division of the Agricultural Academy, Cracow.

The measuring system was based on the measurement of: temperature of the upper source of heat, temperature of the air flux flowing through the exchanger of the bottom source of heat, and the electricity capacity (Fig.1). The measurements of temperature at the upper source of heat were performed in the sealed vessel (utility hot-water) in the upper and bottom zone, while the temperature at the bottom source of heat was measured downstream and upstream the heat exchanger.

Under the measurement of electricity capacity, the capacity consumed by the compressor and the power of circulation pump and the fan forcing the flow of air through the bottom exchanger was measured. All the measurements were recorded on PC, following averaging to one minute of direct observations from the basic sampling with the frequency of 10 Hz. Such a method of recording the measured parameters was allowed by the application written in DasyLab 6.0 software, supporting the PCL 818L measuring card.

The measurements of the parameters of the bottom and upper source of heat, as well as electric qualities, were performed by means of laboratory instruments, and the entire specifications have been provided in Table 2.

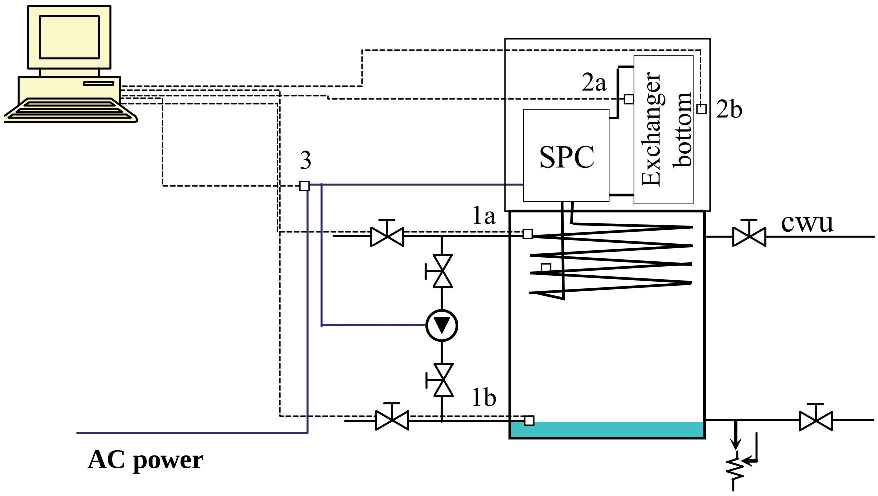

Fig. 1. Scheme of the measuring system of the compressor-assisted heat pump; 1a, 1b − in the lower, upper zone of the vessel, 2a, 2b − temperature detector before and after the bottom exchanger, 3 − active power converter

Table 2. Specifications of the measuring system

Designation	Measurement	Type of instrument
1a, 1b	Temperature at the vessel	Temperature detector PT 100, range; -30÷110 [şC], accurate; 0,01 [şC]
2a, 2b	Temperature before and after the heat exchanger of bottom source of heat	Temperature detector PT 100, range; -30÷110 [şC], accurate; 0,01 [şC]
3	AC power	Power converter PP83, range; 0÷5 [kW], accurate; 1 [W]

The tests were performed under the conditions of actual operation, making multiple repetitions of the water heating cycle in the vessel. The utility hot-water heating cycle was adopted to be in the range from the temperature of the network water and to the point where additional heater was activated.

The heating capacity of the upper source of heat of the small, compressor-assisted heat pump was determined by means of the direct method, on the grounds of an increase of the average water temperature in the sealed vessel with the known volume. The basic calculations were made in the spreadsheet, while in the Statistica software, the models were developed to verify hypotheses at the significance level α = 0,05.

THE RESULTS OF TESTS

The analysis of the test results was commenced from drawing up time characteristics of specific variables recorded during the measurements. On the grounds of these characteristics it was found that the one-minute periods of averaging the parameters sampled are too short as a result of big inertia of processes occurring in the compressor assisted heat pump subject to analysis. Hence, the period of 10 minutes for averaging the parameters recorded was adopted for further analyses.

Then, from among more than a dozen entire cycles of water heating in the vessel only by means of the heat pump, three representative samples were selected for a further, detailed analysis. The selected samples (Fig. 2) ensure the repeatability of the process, and at the same time allow for an estimation of the basic statistics.

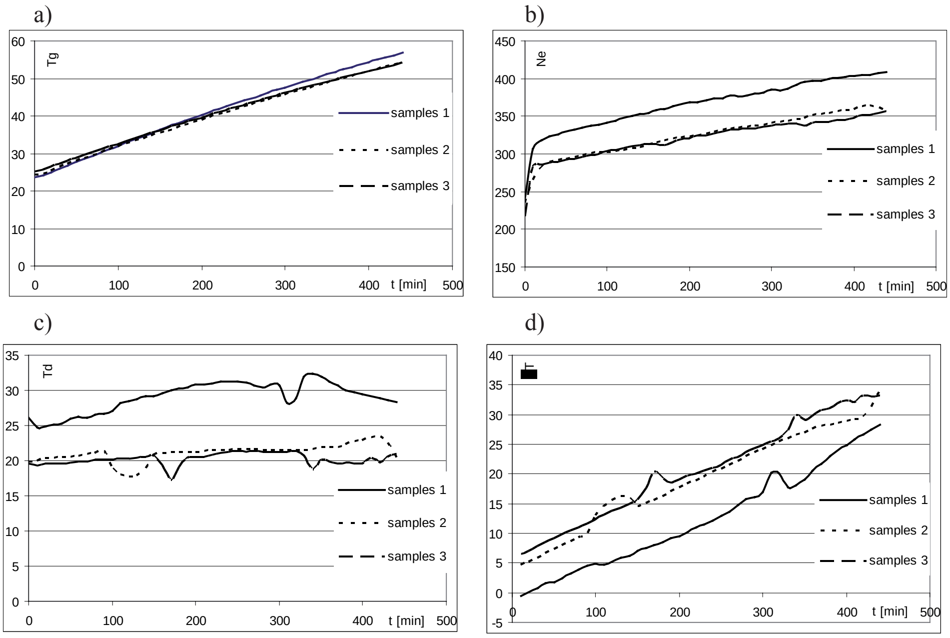

Fig. 2. Time characteristics: a) temperatures of upper source of heat, b) electric power, c) temperature of lower source of heat, d) temperature difference between the upper and lower source of heat

On the grounds of the performed analysis (Fig. 2), the first sample differs from the second and third one. This difference arises out of the larger temperature of the bottom source of heat, which causes an increase of demand for the electric power consumed by the compressor by 14%, with the slightly larger temperature of the upper source of heat (4,5%). Subsequently, for the previously adopted period of averaging, the rate of energy given back by the heat pump to the water in the vessel was calculated in accordance to the relation 1;

$$Q_i = V \cdot \rho \cdot c_p \cdot (T_j - T_i)\,, \tag{1}$$

where: Q_i – energy in the i–th period of averaging,
V – vessel volume,
ρ – refrigerant density,
c_p – specific heat of refrigerant,
T_i ,T_j – temperature of upper source in the subsequent intervals of averaging.

Then, pursuant to the definition of the energy efficiency factor of the heat pump, its value was arrived at with the inclusion of the total demand for electricity through the tested appliance. The obtained results have been presented in the chart (Fig. 3), which shows that the factor (COP) changes from 1,4 at the end of the cycle to 2,6.

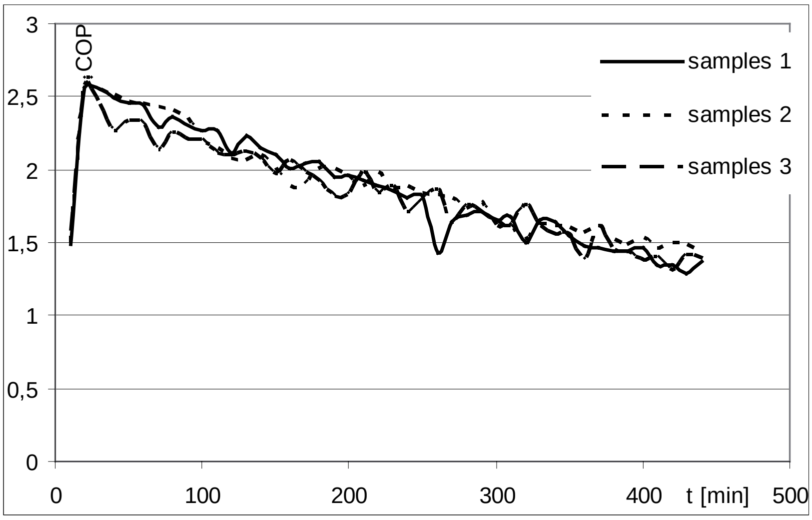

Fig. 3. Time characteristics of the energy efficiency factor (COP) of the compressor-assisted heat pump over the period of the heating cycle

At the beginning of the conduct (Fig. 3) the startup phase is explicitly seen, reflected with the significant lowering of (COP), and arising from the unsteady states – having the transient character. In the further part of conduct (COP), the differences are not seen for the samples presented in spite of the existing deviations in the conduct of the temperature of the bottom source of heat and the consumption of active power (Fig. 2), which may show that the selection of one of exchangers is improper.

After rejection of the observations in the unsteady states and performing the statistical analysis within the basic range, the factors having a significant impact on the value of the energy efficiency factor of the heat pump were established. The most essential factor having impact on the (COP) is the temperature of the upper source of heat. Significantly slighter reaction, but also significant, is caused by the temperature of the bottom source of heat and the consumed electricity. The impact of these factors is well described by the linear regression equations (2, 3) in the double variable system, which were determined at the same factor of determinance R^2=0,96:

$$COP = -0{,}04 \cdot T_g + 0{,}0047 \cdot T_d + 3{,}4\ , \tag{2}$$

$$COP = -0{,}028 \cdot \Delta t - 0{,}0048 \cdot N_e + 4{,}07\ , \tag{3}$$

where: *COP* – energy efficiency factor,
T_g ,T_d – respectively the temperature of the upper and bottom source of heat,
$\Delta t = (T_g - T_d)$ – temperature difference,
N_e – total power consumed by the water heater with the heat pump.

The relations (2, 3) arrived at are inter-complementary. The relation 2 exerts a particularly big significance from the point of practical use of this type of heat pump in the aspect of economic analysis. By contrast, the model 3 allows relatively easy verification of the technical condition of the pump through comparing the efficiency factor after the previous measurement of the active power.

The relation 2 shows that the temperature of upper source of heat has a different impact on the energy efficiency factor than the temperature of the bottom source. A small sensitivity of the system to the temperature of the bottom source (some 10 times lower than the upper source of heat) is just desired from the viewpoint of practical application of heat pumps. However, it should be noticed that the inception operating tests were performed with the relatively high temperatures of the bottom source of heat and small humidity of air (summertime).

SUMMARY

On the grounds of performed calculations and the conducted analysis, it may be declared that the energy efficiency factor of a small compressor-assisted heat pump of air-water type changes within the interval 1,4÷2,6 (at the approximate value 1,85) within the period of water heating in the vessel to the temperature of 55 ^{O}C. This factor is 40% lower than the efficiency factor of the compressor-assisted heat pumps of water-water type [Knaga 2005, Zimny in. 1998]. This is a large difference and may be accounted for with the method of tests performance (operating or laboratory – steady conditions) or with another type of bottom source, as the thermodynamic cycle is implemented by the compressor aggregate of a similar type. In order to rule out the above-mentioned cases, the laboratory testing of this type of heat pump should be performed, which will allow to learn the proper grounds (e.g. inappropriately selected heat exchanger, too large volume of the refrigerant) of the small energy efficiency.

The elaborated models may be suitable for:

a) economic analysis of the practical use of this type of heat pump (relation 2),
b) technical diagnostics (relation 3).

REFERENCES

Kubski P., Lewandowski W.M., Buzuk M. 1994: Zastosowanie pomp ciepła w zintegrowanym systemie energetycznym oczyszczalni ścieków. Technika Chłodnicza i Klimatyzacja, nr 6, s 210-213.

Knaga J. 2007: Zależność efektywności energetycznej sprężarkowej pompy ciepła od czynników eksploatacyjnych, Problemy Inżynierii Rolniczej, 2 (56), Warszawa s. 113–118.

Knaga J. 2005: Efektywność pompy ciepła ze spiralną sprężarką, Inżynieria Rolnicza 2005

Lewandowski M.W. 1996: Konwencjonalne i odnawialne źródła energii. Zeszyty Zielonej Akademii. Gdańsk, Wydawnictwo Okręgu Wschodnio-Pomorskiego Polskiego Klubu Ekologicznego.

Lewandowski W.M. 2006: Proekologiczne odnawialne źródła energii, Wydanie trzecie zmienione, Wydawnictwa Naukowo–Techniczne Warszawa.

Rubik M. 1996: Pompy ciepła, Poradnik. Branżowy Ośrodek Informacji Naukowej, Technicznej i Ekonomicznej, „Inastal" Warszawa.

SeCes-Pol 1998: Cetus – Polskie pompy ciepła, Typoszereg pomp, Gdańsk.

Wołoszyn M. 1991: Wykorzystanie energii słonecznej w budownictwie jednorodzinnym. Centralny Ośrodek Informacji Budownictwa, Warszawa.
Zimny J., Knaga J., Kempkiewicz K. 1998: Wpływ wybranych parametrów pompy ciepła na wydajność cieplną skraplacza, Chemia i Inżynieria Ekologiczna nr 4/98, Opole.
Zimny J., Knaga J., Kempkiewicz K., Kowalski G. 1997: Pompy ciepła stosowane w technice rolniczej, Rynek Instalacyjny, 12/1997, str. 14-16.