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Technical note

PORTABLE HEAT PUMP TESTING DEVICE

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The aim of this paper is to present the design and working principle of a portable testing device for heat pumps in the energy recirculation system. The presented test stand can be used for any refrigerating/reverse flow cycle device to calculate the device energy balance. The equipment is made of two portable containers of the capacity of 250 liters to simulate the air heat source and ground heat source with a system of temperature stabilization, compressor heat pump of the coefficient of performance (COP) of = *4.3*, a failsafe system and a control and measurement system.

Key words: heat pump, renewable energy, coefficient of performance (COP).

1. Introduction

Currently, the main source of heat is the energy from the combustion of fossil fuels such as hard coal, brown coal, natural gas and crude oil. As statistics indicate, the most widespread fuel in Poland is hard coal that constitutes approximately *75*% of the production of heat energy. Another *5*% of the energy comes from the combustion of biomass that emits to the atmosphere significant amounts of dust and volatile fractions. The average consumption of hard coal (according to GUS- National Office for Statistics for 2011) in the housing sector it amounts to approximately 11.5 million tons, which corresponds to *13*% of the total consumption. During the combustion of one ton of coal in a boiler of the power output of *50 kW 7kg* of dust is emitted to the atmosphere (Lachman, 2012). The GUS data for 2011 indicate that the amount of coal used in the housing sector in the area of Wielkopolska was 1 million tons (*9*%), which classifies this province on the third position nationwide. There were only two provinces before Wielkopolska: Mazowsze and Śląsk, in which this consumption amounts to 1.5 million tons (*13*%). From the data of the Institute of Environment Protection, in the years 2005-2009 the emission of dust in the housing sector was *45*% of the total emission for Poland. On 7 December 2010 the Council of Ministers adopted the 'National Plan of Actions in the Area of Energy and Renewable Energy Resource' that was presented to the European Commission. In the plan submitted by the Minister of Economy, it is forecasted that Poland will have reached a renewable energy share of *15.5*% in the gross energy consumption by the year 2020. Due to the tightening of the exhaust emissions regulations and standards, modern, clean and unconventional energy sources must be used in order to reduce harmful emissions.

In order to reduce the harmful emissions to the atmosphere, renewable energy sources are applied. The most common machine and devices using the potential of renewable energy sources are: solar collectors, wind turbines, photovoltaic cells and heat pumps. The history of heat pumps dates back to 1824 when Carnot first published its working principle. In the years that followed, the development of heat pumps was dynamic. In 1852 detailed theoretical fundamentals of heat pumps were announced by Lord Kelvin who proved that refrigerating machines may also be used for heating. He also proved that for heating with heat

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pumps less primary energy is required than for direct heating as the thermal energy is partly taken from the surroundings (air, water, ground), which results in energy savings.

Currently, the manufacturers of heat pumps race to improve these systems. The efficiency of a heat pump is defined by the coefficient of performance (COP). The value of this coefficient can be assessed based on the comparative Carnot cycle, from which it follows that the system's highest efficiency is directly related to the temperature of the ground and air heat source. In theoretical considerations, the maximum value of the coefficient of performance (COP) can reach 9. In practice, however, this value is unattainable. Great divergences in defining the efficiency of heat pumps led to a situation that today, in order to determine the efficiency of the pump, we need to use a procedure contained in a prescribed standard. Despite the existing procedure the COP level is still unreliable and strays from the PN-EN 14511 standard applicable in Poland and Europe. The most frequent mistakes are: providing COP according to an old PN-EN 255 standard, no information on the parameters of the ground and air heat source, non-inclusion of the electrical power of the circulation pumps and other devices, without which the operation of the system is impossible.

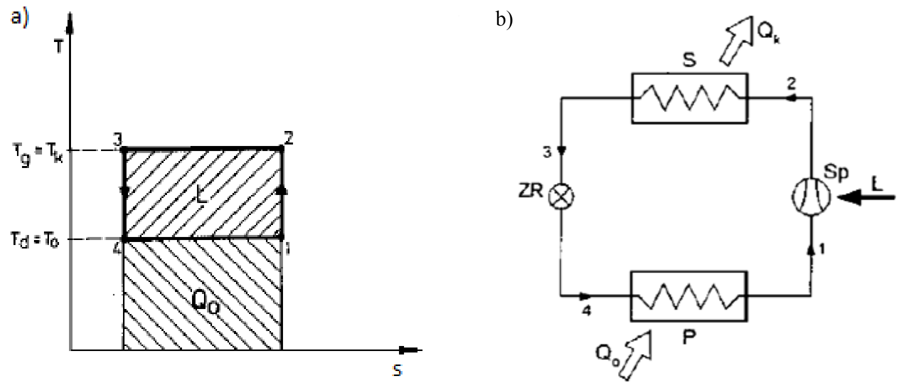

Fig.1. Schematic diagrams: a) Carnot cycle, b) compressor heat pump (Williams *et al*., 1999-2001).

$$\mathrm{COP} = \frac{T_g}{T_g - T_d}\,. \tag{1.1}$$

The presented Eq.(1.1) shows the method of determination of the machine efficiency (referred to as Carnot efficiency) allowing determination of the maximum attainable result with the known temperatures of the ground and air heat sources. This means that for comparison with an actual machine we should assume the Carnot efficiency to be *100*%. When the temperatures are known the theoretical value of COP, compared to the actual heat pump cycle referred to as the cycle energy efficiency ratio, falls in the range from *0.35* to *0.55*.

2. Research

The portable test stand for calculating the energy balance is composed of two 250-liter tanks simulating the ground and air heat source (Fig.2.). The tanks have two pipe coils placed in the upper and lower part of the tanks of the heat exchange area of *1.08m-2*. In the upper and lower part of the tanks there are pipes connected with each other through a circulation pump. From the lower part of the tank, cold water is taken by the circulation pump and pushed into the upper part. Such an exchange of the working medium between the lower and the upper part of the tank prevents thermal stratification inside the tank. The lower coil of the left tank (ground heat source) is connected with the evaporator of the compressor heat pump cycle. The right tank is the air heat source. Figure 2 presents the hydraulic and measurement schematics of the portable device. The lower pipe coil of the tank is connected to a condenser in which the heat is collected from the gas fraction of the R410A refrigerant. In order to ensure the longest operation of the system a temperature stabilization system was applied. The upper pipe coils of the tanks were connected with a working medium circulation forced with a circulation pump. Hence, the energy from the air heat source can be transported to the ground heat source. The amount of transferred energy is controlled by the measurement of the temperature and adjustable flow of the medium. Due to high temperature differences in the cycle, different cycle media were selected. In the hydraulic part, where the temperature drops below 0^oC, a mixture of water and propylene glycol was applied. The concentration of glycol was adjusted to obtain the temperature below the minimum specified by the compressor manufacturer. In the control board, there are voltage and current signal outputs of each of the measured quantities, which enables signal recording on a computer via a measurement card.

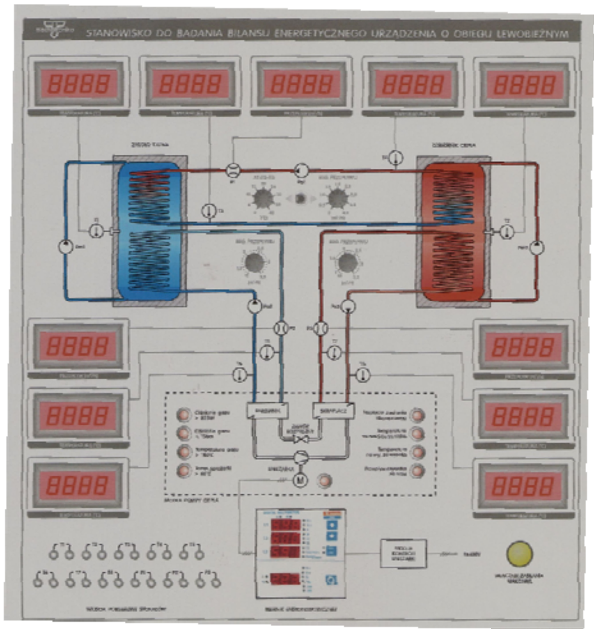

Fig.2. Hydraulic and measurement schematics of the tests stand for calculating the energy balance of heat pump cycles.

The measured values present the characteristics of the operation of the entire heat pump energy balance calculation stand and its individual components, i.e., the heat pump cycle (refrigerating cycle) with the scroll compressor.

3. Control and measurement

The test stand for energy balance calculation of the heat pump (refrigerating) cycles was equipped with temperature and pressure sensors and mass flow meters. The measurements were recorded with the ADAM measurement modules (manufactured by Advatech). The voltage signal was taken from the front pane of the control board in Fig.2. Through an electrical wiring the measurement card was connected with the heat pump and then the card was connected to the computer for data recording. The information on the current temperature values, mass flow of the medium and electric power consumption were recorded with a 1 second resolution. The tests were performed for the scenario of the air heat source reaching a stable temperature. This was a scenario when the heat pump reached the maximum coefficient of performance.

4. Measurements and results

The measurements were performed for several settings of the mass flow of the working medium. The mass flow was adjusted to the working medium flowing through the ground heat source, the air heat source and the temperature stabilization system. The scroll compressor by Copeland of the parameters specified in table 1 was also a subject of an analysis.

The results of the performed tests are shown in the graphs presenting the course of changes of the values characteristic of the heat pump operation. In the measurements the authors recorded the energy consumed for the drive of the circulation pumps necessary for the proper operation of the test stand.

Table 1. Parameters of the tested scroll compressor (Zalewski, 2001).

Rated heat power output of the heat pump [kW]	*9.7*
Refrigerating capacity [kW]	*7.5*
Coefficient of performance (COP) at heating operation	*4.3*
Electrical energy consumption [kW]	*2.2*
Minimum temperature of the ground heat source [^{o}C]	*–5*
Maximum temperature of the air heat source [^{o}C]	*60*

A series of measurement was carried out of the parameters: $\dot{m}_d = 0.33\frac{kg}{s}$, $\dot{m}_g = 0.36\frac{kg}{s}$, the flow through the temperature stabilization system was zero.

Figure 3 presents the two most important parameters describing the operation of the heat pump cycle as a function of temperature of the ground heat source. The minimum temperature of the ground heat source obtained during the tests was $-5.85^{o}C$ and the maximum temperature of the air heat source oscillated around $59^{o}C$. The characteristics of the refrigerating capacity presented in Fig.3 are inclined at a greater angle than the characteristics of the heating capacity. The greatest difference in the capacities occurs in the area of low temperatures of the ground heat source. As the temperature of the ground heat source increases the refrigerating capacity increments faster, compared to the heating capacity.

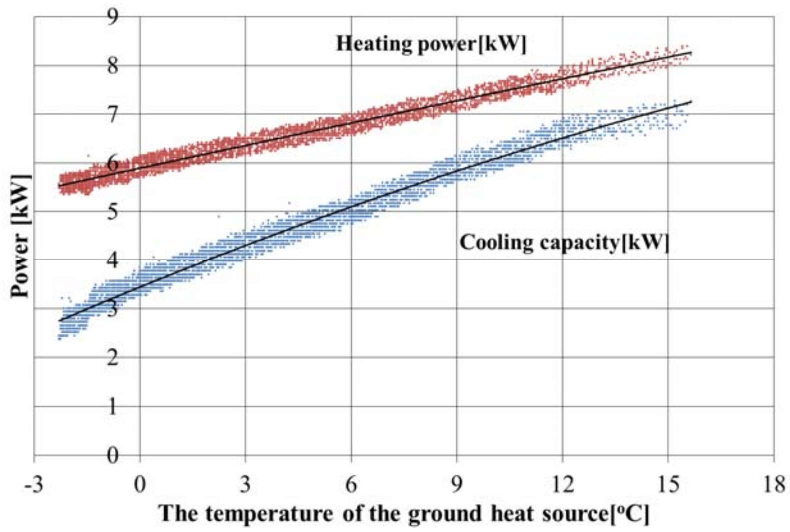

Fig.3. The dependence of the heating and refrigerating capacities on the temperature of the ground heat source.

The differences between the two curves is the energy that must be supplied to the systems to fulfill the cycle. As results from the obtained measurements this capacity is reduced as the temperature of the ground heat source increases. This is directly translated into the efficiency of the cycle that is the greater the less energy is supplied from the outside. Figure 4 presents the relation of the air heat and ground heat source temperature difference, the value of the COP for the actual heat pump cycle and the theoretically determined COP for the comparative Carnot cycle (Viessmann). The most frequent mistake when determining the COP is failure to provide the parameters of the ground heart source. The value of COP is heavily dependent on this temperature. At a constant inlet temperature (35^oC), the value of COP may grow beyond 6. The comparison of the theoretical heat pump efficiency with the actual measured efficiency is shown in Fig.5. As the temperature difference grows between the ground and the air heat source the difference between the theoretical and the actual cycle shown in Fig.4 decreases. The machine approximates *100* % efficiency according to the generally accepted principles of increasing the cycle efficiency by increasing the temperature of the air heat source and decreasing the temperature of the ground heat source.

For the results shown in Fig.5, the fact that the COP is linearly dependent on the temperature of the ground heat source is quite significant. This allows a creation of a simple algorithm estimating the machine efficiency depending on instantaneous external conditions.

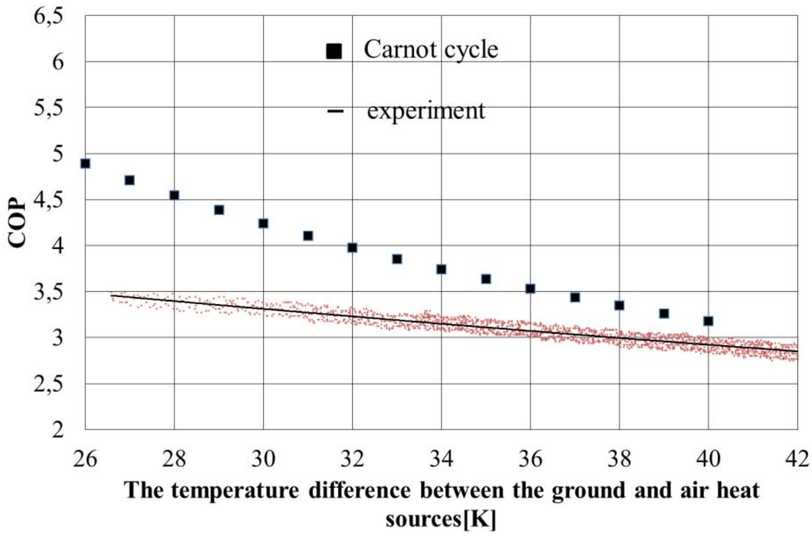

Fig.4. Dependence of COP on the temperature difference between the ground and air heat sources.

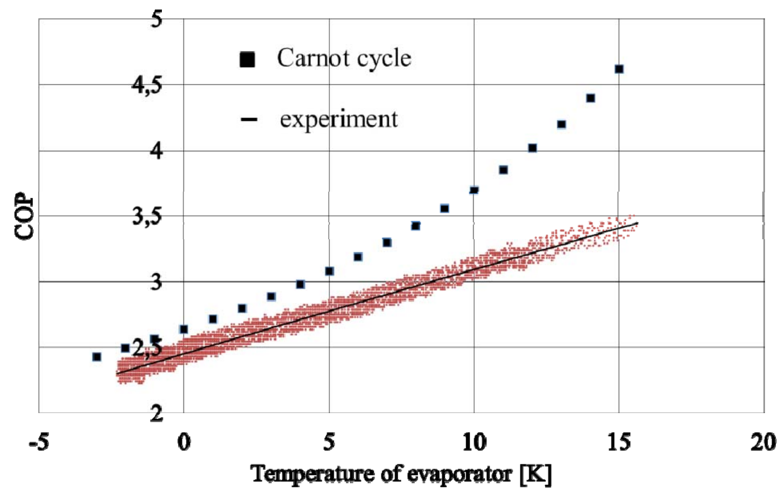

Fig.5. Dependent of COP on the temperature of the ground heat source.

5. Summary

For the measurement, the authors used a compressor heat pump of the COP= *4.3* (as given in the technical specifications). The results of the conducted measurements can be compared with the manufacturer specifications for the parameters B2/W45, for which the heat pump attains COP= *3.5*.

The portable test stand for calculating the energy balance of refrigerating cycles takes into account the power consumed by the circulation pumps that negatively influence the COP. The applied system of temperature stabilization allowed an extended recording of the heat pump operating parameters. The authors found it problematic to maintain constant parameters at the intake to the air heat source. The measurement stand is a fully portable unit and allows installing different heating systems in the refrigerating cycle to determine its characteristics at different parameters of the air heat and ground heat sources.

Nomenclature

L – mechanical energy, *W*
$\dot{m}_d$ – mass flow rate of evaporator, *kg/s*
$\dot{m}_g$ – mass flow rate of condenser, *kg/s*
P – evaporator
Q_o – heat exchange in evaporator, *W*
Q_k – heat exchange in condenser, *W*
S – condenser
TSP – Total Suspended Particulates, *g/Mg*
T_d – temperature of evaporator, *K*
T_g – temperature of condenser, *K*

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