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Modeling and measurement methods for multi-source heat pumps

Péter Hermanucz[1], Gábor Géczi[2], István Barótfi[1]

[1] Institute of Technology, Hungarian University of Agriculture and Life Sciences, 2100 Gödöllő, Hungary
[2] Institute of Environmental Science, Hungarian University of Agriculture and Life Sciences, 2100 Gödöllő, Hungary

Original article

Abstract

Multi-source heat pumps are gaining prominence as energy prices rise and environmental efforts intensify. This type of heat pump system can be used for industrial processes as well as for heating. However, it is a challenge to properly model the two heat sources, possibly operating in parallel, to provide the appropriate input parameters for the design process. The aim of our work is to compile a system that is suitable for the parallel operation of two heat sources. We can compare the measured data with the results of the software used for modeling.

Keywords

- multi-source heat pump
- heat pump modelling

Authors contributions

A – Conceptualization
B – Methodology
C – Formal analysis
D – Software
E – Investigation
F – Data duration
G – Visualization
H – Writing – original draft preperation
I – Writing, reviewing & editing
J – Project administration
K – Funding acquisition

Corresponding author

Péter Hermanucz
e-mail: hermanucz.peter@uni-mate.hu
Hungarian University of Agriculture and Life Sciences
Institute of Technology
Páter K. u. 1.
2100 Gödöllő, Hungary

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Conflict of interest

None declared.

Introduction

Our goal is to introduce the parameters and the modifications needed to measure them in a heat pump system that has at least two parallel energy sources. Where the measurement process would be too complicated, the authors propose a solution to determine the given parameter by calculation. Given that it is currently no widely used, standardized measuring device for real-world refrigerant testing, the development of a suitable device for this purpose is an important part of the research.

The use of two heat source heat pumps is conceivable in the food industry, as during pasteurization processes heat demand and heat source [1] occur at temperature levels that can be met even simultaneously with the help of a well-designed heat pump system. The increasing use of heat pumps is also justified by environmental considerations: if the performance of local combustion plants can be reduced by commissioning a new heat pump system, the concentration of particulate matter [2], which is already causing problems, may decrease. An important area of application can also be the heat supply of buildings, where there is also an economically satisfactory heat demand with a heat pump system [3].

The economical operation of a heat pump is primarily feasible by using multiple heat sources, such as solar and ground heat [4]. Parallel utilization of energy resources has been previously investigated by [5] in the topic of renewable energy.

In the paper of Junfang and co-workers [6], the utilization of air and water heat sources using the same equipment. In the course of their work, the two heat sources are utilized simultaneously with a special heat exchanger, so primarily the so-called composite heat exchanger is the novelty.

In this case, therefore, the refrigeration circuit continues to operate with one evaporator, despite the fact that it utilizes two different heat sources. This, of course, can have advantages in defrost cycles (of which five different cases are examined by the author), but the essence of the procedure we propose is not covered here: No two evaporators are operating in parallel.

Han with a team [7] presents a computer simulation of a complex combined heat pump system with six modes. It describes in detail the components of the equipment, its possible modes of operation, and the mathematical relationships used for the simulation. It simulates the seasonal power factor, operating times of operating modes, the thermal balance of the soil, and operating costs over 10 years. A putative conventional geothermal heat pump system is complemented by a so-called multi-heat source heat pump suitable for the utilization of direct air thermal energy and geothermal energy through a direct or two-stage heat pump circuit. Given that it is suggested that the heat pump does not utilize the heat sources at the same time, but separately for each operating mode.

Some authors [8] demonstrate the utilization of air and water a system that is most similar to what we defined as a research goal. It uses an air-refrigerant and water-refrigerant evaporator in parallel operation. In this case, however, the heat exchangers used in parallel are connected to compressors used in parallel, so we can actually speak of heat pumps used in parallel.

Modeling specifically caloric characteristics is very rare in the international literature [9] presents such modeling system, however, uses a rather complex procedure and software. IMST-ART's well-known software also does not allow the modeling of two parallel evaporators. Therefore, in the following, we will model a specific setup of the system we describe using a simpler scientific software package, Solkane 7.0.

Materials and methods

In the course of our work, we have designed custom-built equipment to meet the following requirements:

- The condenser side should be a refrigerant-water heat exchanger.
- Does employ an air and a water evaporator.
- The water side performance of the aqueous evaporator and condenser can be measured.
- The effective electrical power absorbed by the compressor can be measured.
- Temperatures and pressures can be measured on the suction and discharge side of the compressor.
- The temperature and pressure of the refrigerant inlet and outlet side of the heat exchangers can be measured.

The design of the experimental equipment was performed according to the above aspects, the circuit diagram of the system is also shown in Figure 1.

The measuring devices connected to the equipment and their most important characteristics are described in Table 1. The operating characteristics of the system will be described later.

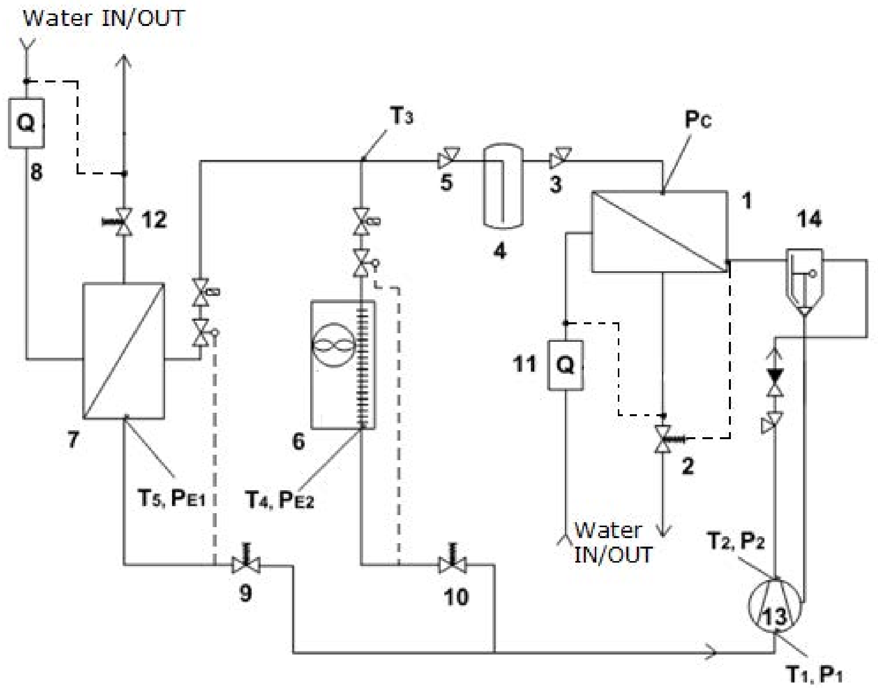

Figure 1. System Layout: 1 – custom built, coaxial condenser, 2 – condensing pressure regulator, 3/5 – shut-off valve, 4 – liquid receiver, 6 – air source evaporator, 7 – water source evaporator, 8 – evaporator side heat meter, 9/10 – evaporator pressure regulator, 11 – condenser side heat meter, 12 – water volume control valve, 13 – compressor, 14 – oil separator [10]

Table 1. The measuring devices and their accuracy

Parameter	Designation	Instrument	Accuracy
Presssure	P_1, P_3	Dixell XC440c with Honeywell transducer	±1%
	P_{E1}, P_{E2}, P_2	Refco analog pressure meter	Accuracy class: 1
Temperature	T_1, T_2, T_3, T_5, T_6	Dixell XR01cx with PTCprobe	±0,7°C
Electric power	W_{KOMP}	Everflourish EMT 707CTL	±1%
Thermal power	Q_k, Q_{E2}	Techem Compact V e. heat meter	Effective: ΔT > 0.2K

The following interventions took place during the conversion of the experimental equipment.

Construction of a special condenser

The condenser is designed to be suitable for measurement purposes and accordingly has refrigerant side pressure measurement points that are not commercially available types. The built-in condensing pressure regulator is also connected to this point, so it is suitable for setting and maintaining the exact value of the condensing pressure [11]. The volume of the heat exchanger on the water side was minimized, therefore the equipment already works with a stable condensing pressure after a short time during one setting. This was confirmed during the test measurements, it took a maximum of 60 seconds to reach the steady-state, and the values were maintained without oscillation.

Exploration of heat and pressure measurement locations

The equipment was basically for education purposes, so despite the fact that countless measurement locations were set up on it, some had to be changed. This primarily means measuring temperatures T4 and T5. I solved the problem by re-wiring two existing temperature sensors, so the measurement can be done at these two locations with the same probe and instrument as the others.

Results and Discussion

The amount of heat transferred by the condenser and the water-heated evaporator was measured on the water side employing heat meters. The stability, accuracy, and especially the resolution of the instruments make them suitable for accurate measurements. Instantaneous power values that are very important to the task are read at a resolution of 0.001 kW in 20 s intervals. Utilizing this feature, the test measurements were able to determine the time required for the system to reach a stable operation, which took approximately 600 s. Prior to the modifications, no stable operating conditions were achieved during the investigated period in Figure 2.

Only minor modifications such as replacing the bulb of the thermostatic expansion valve and minimal adjusting in the set values of the regulating valves for example at evaporation pressure regulator resulted in significant changes in the curves of thermal power emitted by the condenser. This process is shown in Figure 2.

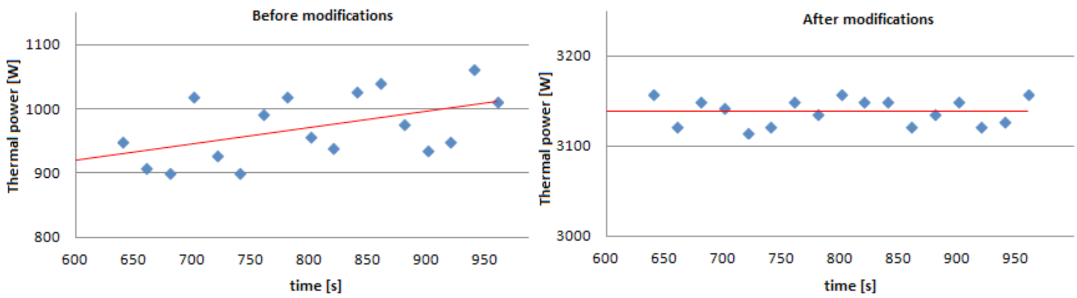

Figure 2. Measured thermal power of the condenser before- and after the modification

As Figure 2 shows the results of the test measurements, the measured power differs before and after the modification. The curve was fitted as a linear trend line, and although the changes in the measured power, the value of the statistical dispersion before the modifications was 48.6, after the modifications it dropped to 14.2.

Measuring the heat amount taken from the air heat source is difficult: the air velocity and temperature should be measured at several points on the evaporator front face, the specific gravity of air should be calculated and many measurement points should be evaluated in a single operating condition. The effect of water vapor condensing from the air would also be very difficult to consider. This would significantly complicate the measurement and would introduce uncertainty, so the power absorbed from the air heat source is calculated. For the calculations, it is essential to know the amount of heat released in the condenser and the enthalpy of the refrigerant at the inlet and outlet of the heat exchanger. This requires the values of T2, T3, and PC. Special software can be used to determine enthalpy values. For this purpose, we use the Solkane 7.0 database of Solvay Fluor. In order to calculate the power absorbed by the air-heated evaporator, the mass flow rate of the refrigerant fed to the heat exchanger (1) shall first be determined:

$$\dot{m}_{R2} = \dot{m}_R - \dot{m}_{R1} \quad [\mathrm{kg/h}] \tag{1}$$

where:

$$\dot{m}_R = \frac{\dot{Q_K}}{h_2 - h_3} \quad [\mathrm{kg/s}] \tag{2}$$

$$\dot{m}_{R1} = \frac{\dot{Q_{E1}}}{(h_5 - h_3)} \quad [\mathrm{kg/s}] \tag{3}$$

h_2, h_3, h_5 can be accurately determined based on the measured pressures and temperature, using software, Q_K and Q_{E1} the heat amount is known from the measurement, so the mass flow of refrigerant into the air evaporator can be calculated as (3), from which the power absorbed in the evaporator can be determined (4):

$$\dot{Q}_{E2} = \dot{m}_{R2} \cdot (h_6 - h_3) \text{ [kW]} \quad (4)$$

The most important test parameter of the refrigeration circuit is the Coefficient of Performance (COP), in addition to the pressure and temperature measured at the points characterizing the cycle. In our studies we used the easily determined indicated performance as the basis for the calculation:

$$COP = \frac{\dot{Q}_K}{W} \quad (5)$$

This is expected to result in higher values than usual in practice, so the results will be of limited use for such comparisons. On the other hand, COP calculated as (5) is independent of the type, internal efficiency, and drive mode of the compressor, so the results will be comparable even with a cycle implemented by a completely different compressor, thus facilitating the examination of different refrigerants [12].

Software modeling results

Test measurements were performed on the measuring system described above at a constant heat flow ratio. 50% of the heat absorbed by the heat pump came from the air-source evaporator and 50% came from the water-source evaporator. The test was run at different condensing temperatures. The heat amount of the evaporators and the condenser as well as the power absorbed were obtained as measurement results. Thus, the COP was determined using measurements.

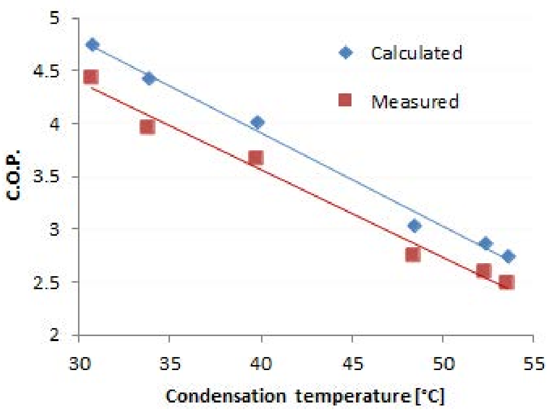

Figure 3. The measured and calculated COP as a function of condensation temperature

Solkane 7.0 software was used for modeling. The two heat sources were modeled as two separate cycles. The input parameters of the model were the evaporation and condensation temperatures and the evaporator heat flow, which were determined by measurement and calculation according to (4). The measurements obtained during the prototype tests are similar to the modeling results according to Equation (5). The results are shown in Figure 3.

The testing method of the correlation was the empirical correlation between COP measured and COP calculated is presented in Figure 4.

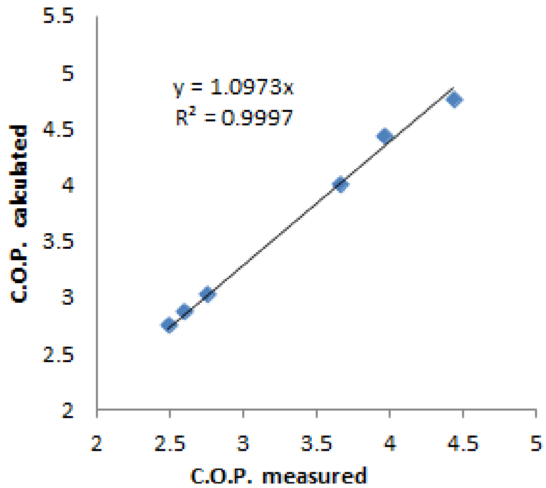

Figure 4. The correlation between measured and calculated COP

It can be observed in the figures that the measured and modeled values show a good correlation, the difference between them is mostly offset. This is probably due to the imperfect thermal insulation of the system and the existence of heat losses of unknown origin. Table 2 shows the measurement results:

Table 2. The measured and calculated parameters

Condensation temp. [°C]	30.7	33.8	39.8	48.4	52.3	53.6
$COP_{Measured}$	4.43	3.96	3.66	2.75	2.59	2.48
$COP_{Calculated}$	4.75	4.43	4.01	3.03	2.87	2.75
$T_{evap.1}$ [°C]	1.3	0.5	0.9	0.5	1.2	1.1
$T_{evap.2}$ [°C]	8.2	7.5	6.8	6.9	8.7	7.8
$Q_{evap.}$ [W]	1437	1340	1232	1034	929	840

In Table 2 the T_{evap1} value represents the evaporation temperature in the air-source evaporator (labeled as 7 on the system layout diagram, Figure 1), T_{evap2} represents the evaporation temperature of the water-source evaporator (labeled as 6 on the system layout diagram, Figure1), and Q_{evap} represents the heat power absorbed by the water-source evaporator. The measured values of all the temperature and pressure sensors and gauges are not presented, as they were used only for some calculations.

Conclusions

According to the test measurements, the modified experimental equipment is suitable for measuring, adjusting, and reproducing the energy characteristics that can be used to measure the thermodynamic effects of heat pumps utilizing two evaporators, therefore two different heat sources in parallel. The two evaporator connections used allow two heat sources to be used independently or even in parallel. With this solution, you can even optimize the cycle for certain refrigerants or applications. The results of the simple modeling procedure we used showed a good correlation with the measured values.

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