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Experimental Study on the Control of the Positions of the Cycle Isotherms of the Heat Pump in the Air Handling Unit

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***Abstract* – The use of air-to-air heat pumps in modern air handling units is becoming more popular for building heating and cooling supply. However, the constantly changing state of outdoor air (temperature, humidity) means that energy conversion modes of heat pumps need to be flexible and able to efficiently respond to ambient air parameters. The ability to adapt to changes in ambient air conditions in an efficient and rational manner is a crucial factor to consider when selecting the operating mode and control of energy transformers. Modern commercial heat pumps typically have two control degrees of freedom, which include a variable-speed compressor and an electronic expansion valve. This combination of control components has previously helped to improve the seasonal efficiency of heat pumps. However, for a long time, focus was placed mainly on refining electronic control tools, without considering how changes in the thermodynamic cycle of heat pumps are correlated with energy demands for air preparation that correspond to outdoor temperatures. The authors of the article propose introducing an additional component into traditional heat pump control measures to provide a third degree of freedom for the cycle control. The experiments were conducted to assess the impact of the heat pump system volume on its thermodynamic operating cycle and it was found that the system volume parameter has the potential for regulation capabilities in controlling the operation of a heat pump. This technological solution has the potential to increase efficiency indicators of the air preparation process and warrants further development and study.**

***Keywords* – Air handling unit; control; experimental study; heat pump; isotherms; vapor compression cycle.**

1. INTRODUCTION

According to [1], 30 % of all final energy consumption in the world is in the buildings sector. Energy consumption is increasing every year. Analytical forecasts indicate that if energy efficiency is not increased in the building sector, in 2050, energy consumption may increase by 50 % [2]. Therefore, energy efficiency is a strategic priority of the EU, encouraging the replacement of inefficient energy production methods with more efficient ones [3].

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The main consumers in buildings are heating, ventilation and air conditioning (HVAC) systems, which consume 50 % of the European Union's (EU) energy consumption, much of which is lost through inefficient use [4]. Namely, HVAC systems require new solutions that open up opportunities to create more efficient products that directly contribute to the implementation of the EU's strategic goals - reducing energy consumption, using renewable energy sources, and increasing energy efficiency.

The policy of reducing energy consumption led to the growth of requirements for the thermophysical properties of external structures of new and renovated buildings and their tightness. These measures gave a positive result and reduced energy costs for heating systems, reducing the required thermal load of buildings. However, these measures in turn had a negative impact on indoor air quality [5]–[7]. For these reasons, the relevance of mechanical ventilation systems in new and renovated buildings has increased. Because it is they that allow to ensure the necessary circulation of fresh air indoors without reducing tightness and thermal comfort in cold climate buildings.

Ventilation is becoming the most important component of a building's energy balance [8], [9], reaching 60 % of the total annual energy demand [10]. Trends show that the energy demand for space heating will be regulated by ventilation heat losses in the future [11]. Therefore, in order to optimize energy consumption and hygiene conditions, forced ventilation systems with heat recovery equipment will be used and will not lose their relevance in the future.

According to the conclusions of the International Energy Agency (IEA), the transition from fossil fuels to electric heat pumps will significantly contribute to the decarbonization of building heating. According to the Announced Pledges Scenario (APS), the total CO_2 emissions related to space and water heating worldwide, including indirect emissions from electricity generation, will decrease by more than 1.2 gigatons (Gt) or over a quarter by 2030. The shift from fossil fuels to electric heat pumps accounts for approximately 500 million tons (Mt) or almost 40% of this reduction [1].

Today, heat pump technologies do not stand still and are constantly improving, revealing new and even broader application possibilities and more energy efficient operation. The use of heat pumps for building heating systems is not new, but their application to ventilation systems is less common [12]. Recently, the situation has been changing, with more and more manufacturers of air handling units (e.g. *FläktGroup, Mandik, Dantherm A/S, Komfovent, VENTS*) offering ventilation units with integrated heat pumps [13].

An exhaust air heat pump is an effective way to reduce the energy costs of outdoor air supply [14]. The efficiency of an air source heat pump is contingent upon the outdoor air temperature, with lower temperatures leading to a diminished coefficient of performance (COP) [15], [16]. During the cold season, the air extracted from the building by the ventilation system usually has higher parameters than the air outside. Therefore, the use of the extracted air flow as a heat source creates favorable conditions for the efficiency of the heat pump. In this case, due to the increased temperature of the heat source (compared to the ambient air), the efficiency of the HP increases, as well as the general indicators of the efficiency of the air handling unit (AHU), as the amount of heat recovered from the exhaust air flow increases.

One of the criteria for reducing the energy demand of buildings by upgrading their HVAC systems is the seasonal coefficient of performance ($COP_{HP.SEZ}$) of these systems. In this case, their integral efficiency for the entire period of use (e.g. heating season) depends both on high nominal, instantaneous efficiency indicators and on the ability to maintain them in a wide range of energy demand variation [17]. The longest part of the season is occupied by the average long-term (only a few or a dozen degrees below room temperature) and constantly changing ambient temperature. This determines the seasonal COP level of the heat pump.

Therefore, to properly manage the operation cycle of the heat pump of the ventilation system with changing environmental air parameters, it is necessary to know the influence of control measures on it and how the performance characteristics depend on them.

The traditional means of controlling the operation mode of the heat pump are the compressor and the expansion valve and the control combination of these components. Adjustment of compressor work/power in heat pumps can be performed in several modes: ON / OFF mode; cascade operation of on/off compressors of the same or different power; modulating operating mode (variable speed compressor). A combination of a variable speed compressor and an expansion valve with variable flow capacity is increasingly common in modern heat pumps. This solution once enabled increased seasonal energy efficiency [18], [19]. These are further developed to find solutions to increase the $COP_{HP\ SEZ}$. Modulating regulation (even regulation) is the most appropriate control mode for the heat pump compressor integrated into the ventilation system. Since there is no heat accumulation in this type of system, the heat pump for the AHU must immediately react to the constantly changing outdoor air temperature.

The objective of the work is to experimentally verify the possibility of controlling the isotherms of an air handling unit's heat pump cycle using traditional control tools (variable speed compressor and expansion valve) according to the variation dependencies obtained in the performed parametric analysis. And also to evaluate the possibility of controlling the heat pump operating mode using an additional innovative control tool (system volume change).

2. METHODS AND METHODOLOGY

2.1. Analytical part

In this article, the energy transformation processes in the heat pump (HP) of the air handling unit (AHU) and the operation of the system are examined. A diagram of such a ventilation unit is shown in Figure 1. The components of the HP are a condenser (CN), evaporator (EV), compressor (CM) and expansion valve (EEV). Together with two fans (supply – F_s and exhaust – F_e) and the ventilation heat recovery heat exchanger (HRE), an AHU is formed. The interaction parameters of these HP components under the characteristic operating conditions of the AHU are analyzed in this work. In the context of this study, the characteristic operating conditions are understood as the states of the HP's refrigerant and the outdoor and ventilated room, which are primarily characterized by temperatures.

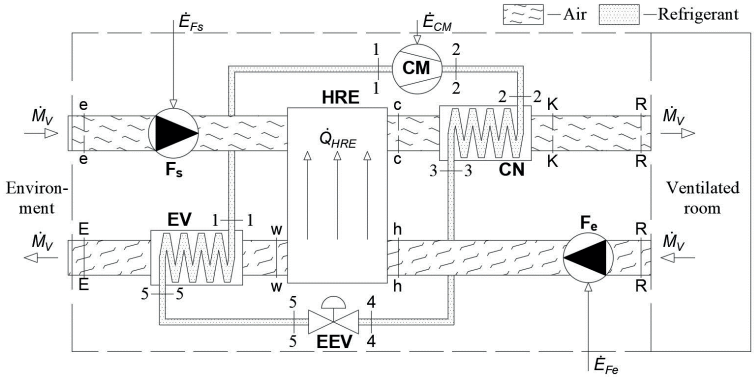


Fig. 1. The schematic diagram of the analyzed heat pump integrated into the ventilation unit.

The thermodynamic processes (primarily heat dissipation and heat transfer) of the heat exchangers (HRE, CN, EV) of the examined air preparation unit are considered to be similar. The energy balance equations of these heat exchangers are presented as Eqs. (1)–(3).

Energy balance equation for the condenser (CN):

$$\dot{Q}_{\mathrm{CN}} = \dot{M}_{\mathrm{f}}(h_2 - h_4) = \dot{M}_{\mathrm{V}}(h_{\mathrm{K}} - h_{\mathrm{c}}) = \dot{M}_{\mathrm{V}} c_{\mathrm{pa}}(T_{\mathrm{K}} - T_{\mathrm{c}}), \tag{1}$$

where

$\dot{Q}_{\mathrm{CN}}$ heat flow of condencer;

$\dot{M}_{\mathrm{f}}$ heat pump refrigerant flow rate;

$\dot{M}_{\mathrm{V}}$ air flow rate;

c_{pa} isobaric specific heat of air;

h_i and T_x enthalpy and temperature at the corresponding point within the AHU (Fig. 1) during the operation cycle of the HP. The typical operating cycle of a heat pump in a *p-h* diagram is shown in Fig. 2.

Refrigerant R410A is used in this work. From a numerical point of view, the properties of the selected refrigerant are important, but it is accepted that their influence on AHU processes is not exceptional compared to other cases of the reverse vapor compression cycle.

Energy balance equation for the evaporator (EV):

$$\dot{Q}_{\mathrm{EV}} = \dot{M}_{\mathrm{f}}(h_1 - h_5) = \dot{M}_{\mathrm{V}}(h_{\mathrm{w}} - h_{\mathrm{E}}) = \dot{M}_{\mathrm{V}} c_{\mathrm{pa}}(T_{\mathrm{w}} - T_{\mathrm{E}}), \tag{2}$$

Energy balance equation for the heat recovery heat exchanger (HRE):

$$\dot{Q}_{\mathrm{HRE}} = \dot{M}_{\mathrm{V}}(h_{\mathrm{h}} - h_{\mathrm{w}}) = \dot{M}_{\mathrm{V}}(h_{\mathrm{c}} - h_{\mathrm{e}}) \text{ or } \dot{Q}_{\mathrm{HRE}} = \dot{M}_{\mathrm{V}} c_{\mathrm{pa}}(T_{\mathrm{h}} - T_{\mathrm{w}}) = \dot{M}_{\mathrm{V}} c_{\mathrm{pa}}(T_{\mathrm{c}} - T_{\mathrm{e}}) \tag{3}$$

It is the combination of these equations Eqs. (1)–(3) for the energy balance of heat exchangers that reflects the thermodynamic operation of the air handling unit.

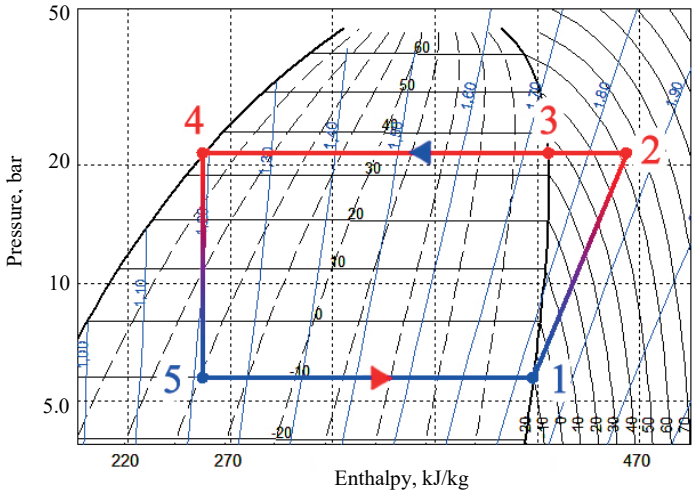


Fig. 2. Typical heat pump operation cycle (vapor compression) on the *p-h* diagram.

Heat transfer processes are described by similar equations for both CN and EV at their respective specific heat flows $(Ak)_{CN}$ and $(Ak)_{EV}$. Heat flow between air and refrigerant in CN Eq. (4) and EV Eq. (5):

$$\dot{Q}_{CN} = (Ak)_{CN} \Delta T_{CN}, \tag{4}$$

$$\dot{Q}_{EV} = (Ak)_{EV} \Delta T_{EV}, \tag{5}$$

where ΔT_i is temperature difference of heat transfer, K; $(Ak)_x$ – specific heat flow rate of heat exchanger, W/K.

The dependences of the isotherms and temperature difference of heat transfer are related by the equations of energy balance and heat transfer. The resulting condenser and evaporator isotherm temperature equations Eq. (6) and Eq. (7) are shown below.

$$T_{izot.CN} = \frac{T_c + T_K}{2} + \Delta T_{CN} = \frac{T_c + T_K}{2} + \frac{\dot{M}_V c_{pa} (T_K - T_c)}{(Ak)_{CN}}, \tag{6}$$

$$T_{izot.EV} = \frac{T_w + T_E}{2} - \Delta T_{EV} = \frac{T_w + T_E}{2} - \frac{\dot{M}_V c_{pa} (T_w - T_E)}{(Ak)_{EV}}, \tag{7}$$

where $T_{izot.CN}$ – condensation isotherm temperature, $T_{izot.EV}$ – evaporation isotherm temperature, T_x – temperature at the corresponding point within the AHU (Fig. 1) during the operation cycle of the HP.

A more detailed description of the calculation methodology, the processes of heat transfer in the condenser and evaporator of the HP, depending on changes in the parameters of the outdoor air and refrigerant, have been presented earlier in other studies on this topic [13], [20], [21].

The following assumptions were made in the study:

1. The effect of the degree of superheating and subcooling of the refrigerant is not considered to be significant. They are assumed to be zero.
2. The system is at steady state. Losses of pressure, heat, kinetic and potential energy are neglected.
3. Indoor and outdoor air is dry, and the mass flow rate of supply and exhaust air is the same and constant ($\dot{M}_V$).
4. AHU is well heat-insulated (adiabatic), air and refrigerant receive or release heat only through heat exchangers (CN, EV, HRE) or fans (F_e and F_s).
5. The power of F_e and F_s is selected according to the recommended specific fan power (SFP) index, expressed in W/m^3/s.
6. There is no heat input or other heat source in the ventilated room.
7. It is assumed that the changes in air temperatures influenced by F_e and F_s are estimated by equating that all of their power is used to heat the air.
8. The specific heat flow of heat exchangers is considered to be a constant value independent of changes in heat carrier temperatures.
9. The evaporation temperature is assumed to be 5 °C lower than the temperature of the air extracted from the heat recovery heat exchanger.

The main characteristics of the air handling unit (AHU) with an integrated heat pump (HP) were selected for analysis: supplied and exhaust air flow 430 m^3/h; AHU supply temperature (room temperature) 20 °C; ambient (outdoor) air temperature range −20...15 °C; specific fan

power (SFP) 150 W/m^3/s according to [21]; condenser and evaporator specific heat flows $(Ak)_{CN}$ = 170 W/K and $(Ak)_{EV}$ = 150 W/K; temperature efficiency of heat exchanger 0.8; the isentropic efficiency coefficient of the heat pump compressor is equal to 0.8.

2.2. Experimental part

The schematic diagram of the experimental setup is shown in Fig. 3.

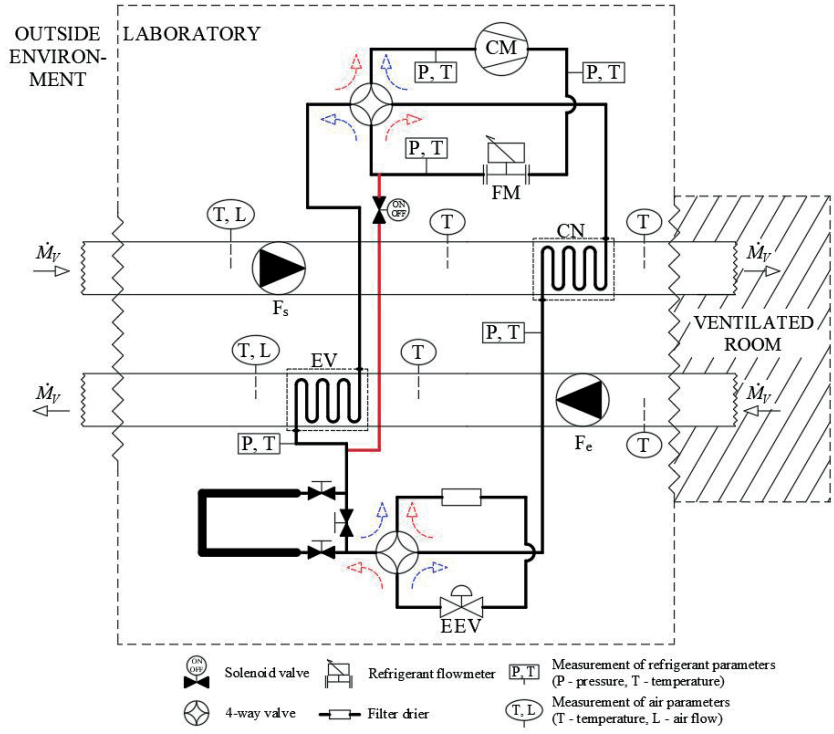


Fig. 3. The schematic diagram of the experimental setup.

The experimental bench can be divided into two main parts: the ventilation supply/exhaust duct system that serves the ventilated room, and the heat pump system, which heats the air supplied to the room by recovering thermal energy from the exhaust air flow. Unlike the scheme of the theoretical model, there is no heat recovery heat exchanger in the experimental bench. Air circulation is provided by supply (F_s) and exhaust (F_e) fans. The average air flow rates supplied and removed during the experiments were equal to 430 m^3/h. The heat pump circuit, apart from the main components (variable speed compressor, electronic expansion valve, condenser, and evaporator), has a hot steam line for defrosting the evaporator and two four-way valves, allowing the heat pump to be used in cooling mode, but within the scope of this work, only heating mode is considered. Also, between the electronic expansion valve (EEV) and the evaporator (EV), a heat pump system volume change device is installed, enabling to increase the volume of the system's active operation circuit. A more detailed diagram of this device is shown in Fig. 4.

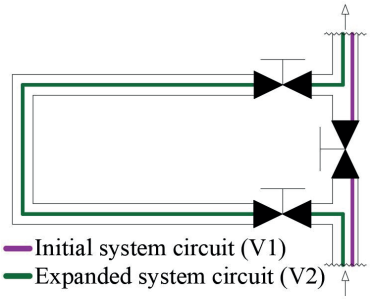


Fig. 4. Element for increasing the volume of the active operation circuit of the heat pump.

The element for increasing the volume of the heat pump's active operation circuit consists of a system of pipes and closing valves, which allows to increase the internal volume of the heat pump circuit by 26.4 % (+1.26 dm^3), compared to the initial baseline heat pump circuit variant, which has a volume of V_1 = 4.76 dm^3.

During the experiment, the pressures, temperatures, and flow rates of the refrigerant circulating in the circuit of the heat pump are measured at 10-second intervals. Also, supply/exhaust air volumes are determined by measuring the pressure difference before and after the diaphragm, and air temperatures are also measured. The measurement points for these parameters are indicated in the schematic diagram of the experimental setup shown in Fig. 3.

3. RESULTS

3.1. Analytical part

Thermodynamic parametric analysis was carried out showing the theoretical dependence of the change in the condensation and evaporation temperatures of the heat pump on the heating power required by the AHU (Fig. 5). This dependence sets limits on the control of the heat pump.

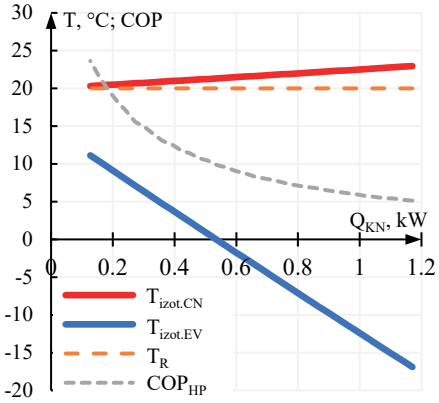


Fig. 5. Dependence of condensation and evaporation temperatures and COP on heating power of HP.

From Fig. 5, as the heating demand decreases, the condensation temperature ($T_{izot.CN}$) decreases steadily, while the evaporation temperature ($T_{izot.EV}$) increases. The distance

between the isotherms of the heat pump cycle decreases, which, depending on the thermodynamic properties of the refrigerant, means that the pressure difference between the high- and low-pressure zones of the cycle also decreases. T_R here is the required temperature of the air supplied to the ventilated room.

By controlling the heat pump in this mode, it is possible to achieve theoretically the highest COP, given the set characteristics and parameters of the device.

3.2. Experimental part

One of the main tasks of the experiment is to verify the possibility of controlling the isotherms of the HP according to the trends obtained in the parametric analysis. To achieve this goal, tests of an experimental HP integrated into the ventilation system were carried out. During the tests, the HP was controlled by traditional cycle control measures – a variable speed compressor (CM) and an electronic expansion valve (EEV). The operation of the heat pump was tested with different combinations of these control parameters. The compressor rotation speeds tested were 1800, 1900, 2100 and 2220 RPM, and the expansion valve openings tested were 30, 50 and 70 %. Which result in a total of 12 combinations. The overall results obtained are shown in Fig. 6. The isotherm temperature points are the 2-minute averages of a steady-state operation.

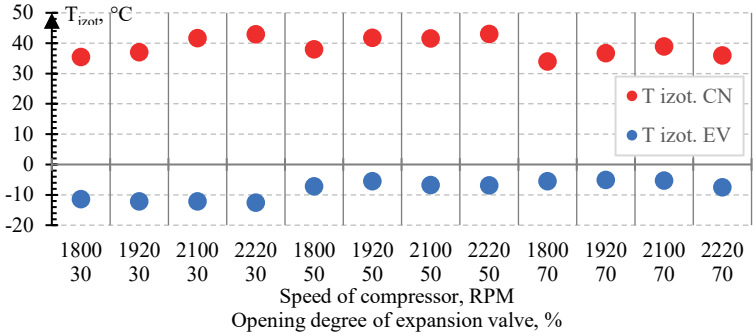


Fig. 6. Condensation and evaporation isotherm temperatures at different compressor speeds and expansion valve opening combinations (at system volume V_1).

The next step involved increasing the volume of the heat pump system from $V_1 = 4.76$ dm^3 to $V_2 = 6.02$ dm^3 and conducting tests with the same combinations of control measures. The overall results obtained are shown in Fig. 7.

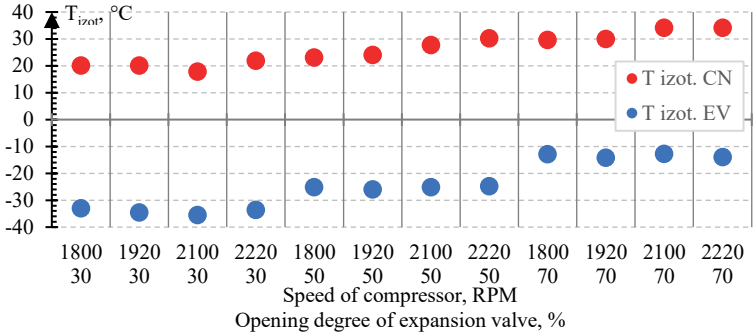


Fig. 7. Condensation and evaporation isotherm temperatures at different compressor speed and expansion valve opening combinations (system volume V_2).

From Fig. 6 and Fig. 7, it can be seen that due to the increase in the heat pump system circuit volume, the temperatures of condensation and evaporation isotherms decreased, on average by 12.9 °C and 16.1 °C, respectively. In addition, the range of condensation temperature variation widened by 81 %, from 9.0 °C to 16.3 °C. The range of evaporation temperature variation also increased almost 2 times, from 7.4 °C to 22.8 °C (208 %).

From the results of the first stage of the experiments (when the system volume is V_1), it was possible to determine that by increasing the compressor speed and simultaneously reducing the opening degree of expansion valve, it is possible to form a control mode similar to the temperature change trend of the isotherms of a heat pump obtained in a parametric analysis (Fig. 8(a)).

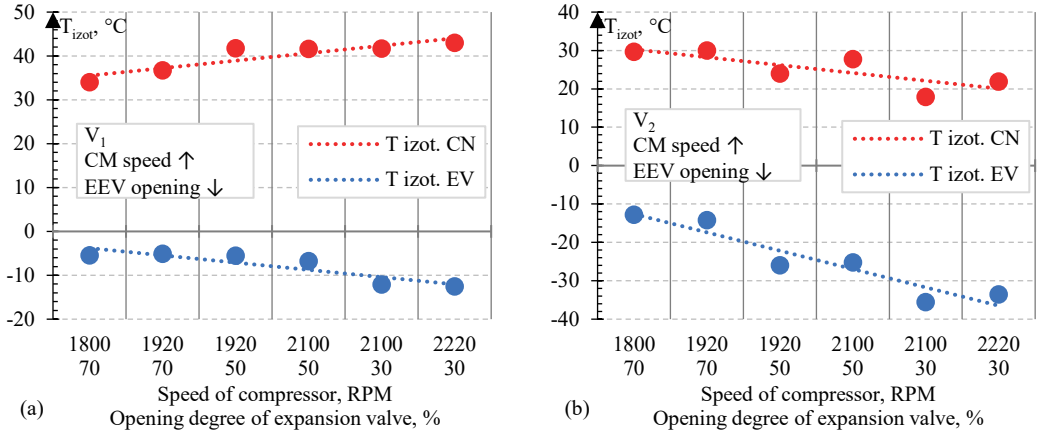


Fig. 8. Condensation and evaporation isotherm temperatures as expansion valve permeability decreases and compressor speed increases: (a) System volume V_1; (b) System volume V_2.

In Fig. 8(a), we see that as the condensation temperature increases, the evaporation temperature decreases, which corresponds with the main trend obtained in the analytical part.

After increasing the volume of the heat pump system from V_1 to V_2, the experimental test was repeated with the device operating under the same operating modes as in the initial case variant. The results are presented in Fig. 8(b).

From Fig. 8(b), we can see that the evaporation isotherm temperatures decreased by an average of 17 °C and became more responsive to changes in control parameters compared to the heat pump system with a volume of V_1. However, their main trend has not changed, as the speed of the compressor increases and the permeability of the expansion valve decreases, the evaporation temperature decreases. The trend of the condensation isotherm temperatures changed in the opposite direction, decreasing instead of increasing. The condensation temperatures decreased on average by 15 °C.

Also, a control variant of the heat pump was tested, forming it in the opposite way to the variant shown in Fig. 8, i.e. by increasing the expansion valve flow rate together with the compressor speed. The results of this operating mode are shown in Fig. 9.

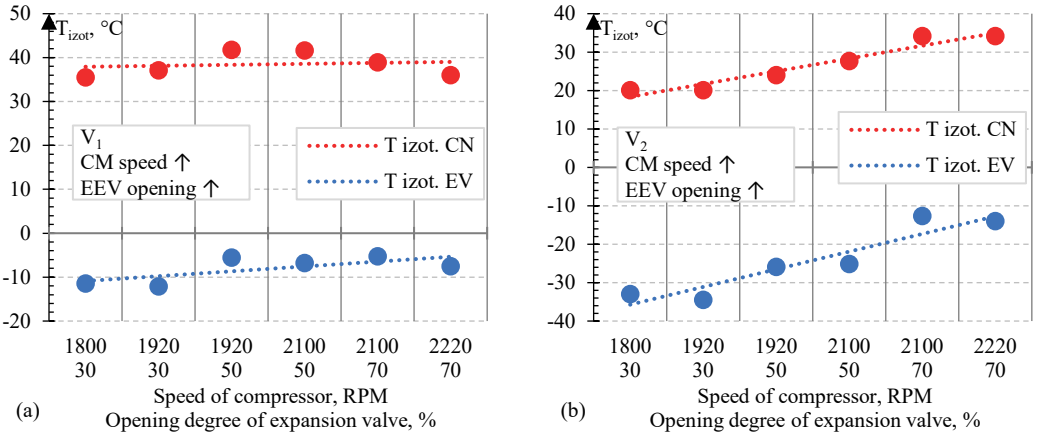


Fig. 9. Condensation and evaporation isotherm temperatures as expansion valve permeability increases and compressor speed increases: (a) System volume V_1; (b) System volume V_2.

In Fig. 9, due to the increase in the volume of the heat pump circuit, the overall level of condensation and evaporation temperatures decreased, on average, by 12 °C and 16 °C, respectively. In addition, increasing the system volume did not substantially change the trend of the overall isothermal temperature change within the range of this control variant. As the expansion valve opening increases along with the compressor speed, the condensation and evaporation temperatures increase. However, increasing the system volume made the isothermal temperatures more sensitive to changes in control parameters.

From Fig. 8 and Fig. 9, we can see that changing the volume of the heat pump circuit induced a different degree of behavior in the system. Therefore, we can say that volume change added a third degree of control, and further studies will expand the volume increase to identify a wider influence on the process. In this work, the possibilities of using this volume change potential are not examined, the aim was to assess whether it is possible to change the position of the isotherms of the heat pump cycle.

4. Conclusion

The performed parametric analysis determined the theoretical dependence of the heat pump's condensation and evaporation isotherm temperature changes on the required heating capacity of the AHU. As the heating demand decreases, the condensation temperature decreases uniformly while the evaporation temperature increases steadily. This dependence sets the direction of the control of the heat pump.

It was experimentally determined that using traditional control measures (variable speed compressor and expansion valve), it is possible to form a control regime close to the trend of the heat pump's isothermal temperature change obtained in the parametric analysis.

Also, during the experimental tests, it was found that changing the volume of the heat pump circuit of the air handling unit caused a different degree of behavior in the system. Therefore, we can say that changing the volume added a third degree of control, which expands the possibilities of controlling the positions of the isotherms. And further studies will expand the volume increase to determine a wider influence on the process.

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