Research to estimate energy efficiency of a ventilation and air conditioning heat pump system inside a production premise with ventilation air recovery

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Abstract   The research provides a thermodynamic analysis of the theoretical model of a ventilation and air conditioning heat pump system with the ventilation air cold energy recovery depending on outside air parameters, the recovery efficiency and characteristics of a premise. A confectionery production workshop was taken as a prototype where technological conditions (temperature and humidity) must be maintained during the warm season. Calculations using the method of successive approximations to estimate air parameters at system's nodal points were conducted. It allowed to determine theoretical refrigeration efficiency of the studied system and proved advantages of heat recuperation for smaller energy consumption. The model can be applied for design of heating, ventilation, and air conditioning units which work as a heat pump. The studied system has the highest energy efficiency in the area of relatively low environment temperatures and relative humidity which is suitable for countries with temperate continental climates characterized by low relative humidity.

Keywords: Air conditioning; Heat pump system; Refrigeration efficiency; Recuperation

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Nomenclature

d	–	moisture content, $\text{g/kg}_{d.a.}$
G	–	mass flow rate, kg/s
h	–	enthalpy, $\text{kJ/kg}_{d.a.}$
K	–	the conversion factor which depends on external and internal heat flows
K_0	–	a variable ratio of fresh supply air for ventilation purposes
$K_{a.ex}$	–	air exchange rate, 1/h
L_c	–	compressor load, kW
t	–	temperature, °C
$Q_{a.c}$	–	required heat load for air conditioning during the warm season, kW
V_{prem}	–	volume of a premise, m^3

Greek symbols

Δ	–	difference
ε	–	refrigeration coefficient
η	–	recuperation coefficient, %
φ	–	relative humidity, %

Subscripts

0	–	outside air
1	–	supply air
1'	–	air for ventilation demands only
2	–	exhaust air
c	–	condenser
d.a	–	dry air
ev	–	evaporator
pc	–	precooled
ph	–	preheated
r	–	recuperation
tot	–	total
vent	–	ventilation

Acronyms

COP	–	coefficient of performance
HP	–	heat pump
HPS	–	heat pump system
HVAC	–	heating, ventilation, and air conditioning

1   Introduction

In 2018 world coal consumption increased by 2.5% [1]. Faced with problems of fossil fuel depletion and environmental pollution, researches are being carried out in two directions: development of new energy sources, especially

renewable ones, and increase in energy efficiency of heating, ventilation, and air conditioning (HVAC) systems to reduce fossil fuel consumption and pollution caused by its use [2].

Systems based on air heat pumps (HP) are widely used for ventilation and air conditioning of residential, commercial and industrial premises, which require the maintenance of set temperature and humidity conditions [3]. Such heat pump systems (HPS) are highly efficient and, when properly designed, reduce primary energy input requirements compared to traditional systems [4].

A special approach is required to use HPS for ventilation and air conditioning effectively, since the external clean air dried after the HP evaporator must be heated to a set temperature at the entrance to a premise [5]. This process can be organized in different ways [6]. A thermodynamic analysis of HPS was performed in the following study [7]. According to the proposed HPS design the supply air parameters at the entrance to a premise are achieved due to mixing of cold air flow after the HP evaporator and hot air flow after the HP condenser inside a mixing chamber. The energy efficiency of such HPS is increased because of the exhaust air partial recirculation, i.e. due to the partial utilization of generated cold within the system [8]. However, there exists a fundamental possibility to further improve the energy efficiency of such HPS by recuperation of the cold exhaust air to preliminarily cool down the fresh supply air [9]. The expected effect of the cold recovery is the reduction of the HP evaporator load, and, as a consequence, the decrease of the compressor load and the increase in the total energy efficiency of the HPS [10]. However, there is a limiting factor when heating the exhaust air inside the recuperator before it enters the HP condenser, as this can increase the air temperature after the condenser above critical values (about 70°C) which may result in failure of the HPS equipment [11]. Therefore, a special study is required to verify the feasibility of a recuperator application for the cold energy recovery inside a ventilation and air conditioning HPS.

As a result of the study HPS energy efficiency and operating modes during the warm season were estimated, depending on the outside air parameters, the recovery process efficiency and the specification of premises where ventilation and air conditioning are required. The HPS theoretical model was developed and its thermodynamic analysis to define air parameters in nodal points of the system and conditions to maintain supply air parameters at the entrance to a premise was carried out.

2   Objectives

The aims of the paper are to estimate the energy efficiency of a ventilation and air conditioning heat pump system (HPS) with previous cold energy recovery of the exhaust air [12] and a variable ratio of fresh supply air depending on ambient parameters, and to determine the preferable climatic conditions for the most efficient application of a novel HPS with preliminary utilization of generated cold.

3   Description of the ventilation and air conditioning heat pump system and the ventilation air workflow process

A schematic diagram of ventilation and air conditioning HPS with exhaust air cold energy recovery and with a variable ratio of fresh supply air is shown in Fig. 1. The principle of system's operation is as follows: fresh supply air with temperature t_0, moisture content d_0 and mass flow rate G_0 enters the recuperator, where it is cooled at a constant moisture content to temper-
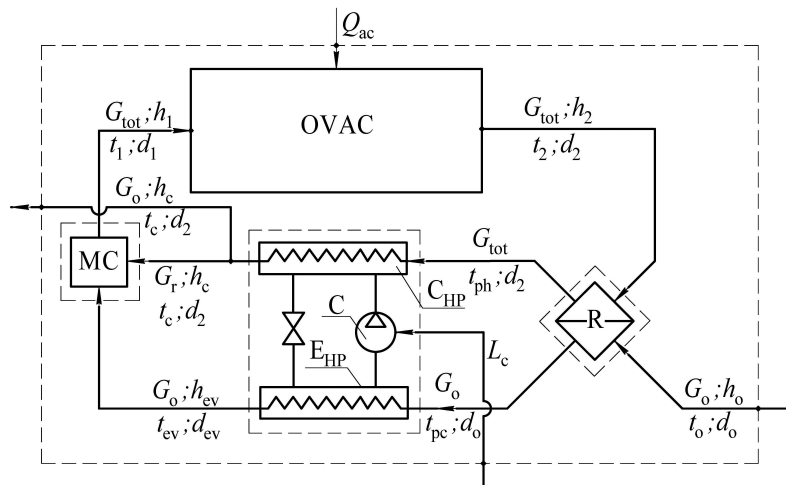

Figure 1: A general ventilation and air conditioning heat pump system inside a premise with exhaust air cold energy recovery: CHP – a heat pump condenser; EHP – a heat pump evaporator; C – a compressor; MC – a mixing chamber; R – a recuperator; OVAC – an object of ventilation and air conditioning.

ature t_{pc} by cold ventilation emissions from the premise with temperature t_2, moisture content d_2 and mass flow rate G_{tot}. The cooled supply air flows to the HP evaporator, where it is further cooled with a partial water vapor condensation to a temperature t_{ev} and a moisture content d_{ev}. The preheated exhaust air after the recuperator at a temperature t_{ph} and a constant moisture content enters the HP condenser, where it is further heated and at the outlet has a temperature t_c with a constant moisture content d_2. The heated exhaust air is then divided into two flows: one flow with a mass flow rate G_0 is discharged into the environment while the second one is fed to the mixing chamber, where it mixes with the cooled supply air after the HP evaporator. The resulting mixture of air (t_1, d_1, G_{tot}) is then supplied to the production premise for ventilation and air conditioning purposes (to compensate for internal and external heat flows).

The workflow process representing the change of states of different air flows within the ventilation and air conditioning system is shown in Fig. 2 using the *h-d* diagram. Point *O* corresponds to parameters of the environment $(t_o,\ h_o)$ [13]. Point *2* is plotted according to the required temperature t_2 and relative humidity φ_2 of air inside a production premise, which depend on the premise technological purpose [14]. Enthalpy h_2 is determined from the *h-d* diagram. The value of the supply air overheating Δt_{vent} is chosen according to building standards [14] for this production facility. Point *1′*, which represents the supply air state for ventilation purposes only in order to remove excess moisture, is found at an intersection of lines $h_2 = \text{const}$ and $t_{1′} = t_2 + \Delta t_{vent}$. The *1′-1* transition depicts additional air supercooling to meet air conditioning needs. The air temperature at point *1* on the line $d_{1′} = \text{const}$ can be estimated based on the relationship between the temperature difference corresponding to the supply air supercooling and the temperature difference between air outside and inside the production premise, i.e. $\Delta t = t_{1′} - t_1 = K(t_0 - t_2)$. The value of conversion factor K depends on external and internal heat flows. The exhaust air with parameters at point *2* is heated in the recuperator to a temperature t_{ph} at a constant moisture content d_2. Then this flow enters the HP condenser, where it is further heated to a temperature t_c at a constant moisture content. One part of the flow is discharged into the environment, while the other one is mixed with cooled supply air after it passes through the recuperator and then the HP evaporator (points pc and *Ev*, respectively). The resulting air mixture at point *1* with moisture content $d_{1′} = d_1$ and temperature t_1 is then supplied to the premise to cover demands for ventilation and air conditioning. The working process *1–2* depicts the change of air state inside the premise.
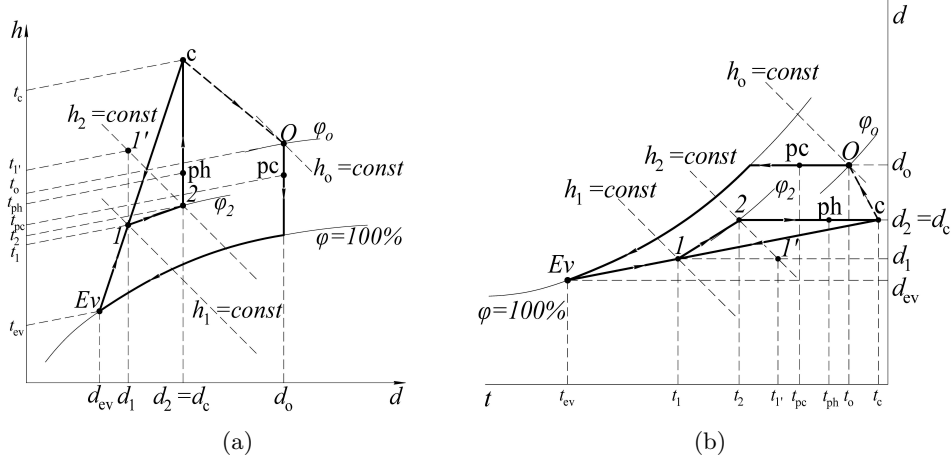
Figure 2: The workflow process in the *h-d* diagram (a) and psychrometric chart (b) showing the change of air state inside the ventilation and air conditioning system.

4   Estimation of the supply air temperature at the entrance to the premise when air conditioning is required

According to the method described in [7], the relationship between the supply air supercooling to cover demands for air conditioning and the temperature difference between air outside and inside the production premise can be written as

$$\Delta t = t_{1'} - t_1 = K\left(t_0 - t_2\right), \tag{1}$$

where K is the conversion factor which depends on external and internal heat flows.

The value of K is determined as follows:

$$K = K_1 + K_2, \tag{2}$$

$$K_1 = \frac{3.6\bar{K}F}{\rho_\mathrm{a} c_\mathrm{p} K_\mathrm{a.ex} V_\mathrm{prem}}, \tag{3}$$

$$K_2 = \frac{Q_\mathrm{in}}{G_\mathrm{tot} c_\mathrm{p}(t_0 - t_2)}, \tag{4}$$

where: K_1 – coefficient that depends on the geometric and thermophysical characteristics of the building and the air exchange rate of the air conditioning object; K_2 – coefficient that depends on internal heat flows;

$\bar{K}$ – average heat transfer coefficient of the premise enclosing structures, $\mathrm{W/(m^2\,{}^\circ C)}$; F – area of enclosing structures, which external heat enters the premise through, $\mathrm{m^2}$; ρ_a – air density, $\mathrm{kg/m^3}$; c_p– specific heat capacity of air, $\mathrm{J/(kg\,{}^\circ C)}$; $K_\mathrm{a.ex}$ – air exchange rate, $\mathrm{1/h}$; V_prem – volume of a premise, $\mathrm{m^3}$; Q_in – internal heat generation, W.

Thus, the conversion factor, K (Eq. (1)) can be predetermined through coefficients K_1 and K_2. The supply air temperature at the entrance to the premise t_1 can be determined from Eq. (1).

5   The thermodynamic analysis of the studied heat pump system

Thermodynamic analysis of the heat pump system provides preliminary estimation of its thermodynamic condition at the accepted initial parameters of air indoors and depending on variable parameters of external environment (air temperature and moisture content). The state of the HPS depends on unknown parameters of air at system's nodal points [15]. These parameters are the ratio of fresh supply air fed to the HP evaporator, air temperatures after the recuperator, after the HP evaporator and condenser, as well as the refrigeration coefficient value of the entire system. Thermal and material balance equations of system's individual elements and the system as a whole allow determining relationships for these parameters.

Temperatures of air flows after the recuperator can be determined from the expression to evaluate its efficiency [16]

$$\eta_\mathrm{r} = \frac{t_\mathrm{ph} - t_2}{t_0 - t_2} \qquad \text{or} \qquad \eta_\mathrm{r} = \frac{t_0 - t_\mathrm{pc}}{t_0 - t_2} \,, \tag{5}$$

where: η_r – recuperator efficiency ratio; t_ph, t_pc – temperatures of preheated exhaust and precooled outside air respectively, $^\circ$C.

From (5) expressions for temperatures t_ph and t_pc have the following form:

$$\begin{aligned} t_\mathrm{ph} &= t_2 + \eta_\mathrm{r}\left(t_0 - t_2\right), \\ t_\mathrm{pc} &= t_0 - \eta_\mathrm{r}\left(t_0 - t_2\right). \end{aligned} \tag{6}$$

Enthalpies h_ph and h_pc are determined from the h-d diagram at intersection points of respective temperature and moisture content curves for exhaust ($t_\mathrm{ph} \times d_2$) and supply ($t_\mathrm{pc} \times d_0$) air flows.

The enthalpy of fresh supply air after the HP evaporator can be determined from the heat balance equation for the mixing chamber, which has the form

$$G_0 h_{ev} + G_r h_c = G_{tot} h_1 \,, \tag{7}$$

where: G_0, G_r, G_{tot} – fresh supply, the ratio of heated exhaust and total mass air flows respectively, kg/s; h_{ev}, h_c, h_1, kJ/kg$_{d.a.}$ – enthalpies of corresponding air flows.

Dividing left and right parts of Eq. (7) by the value G_{tot}, one obtains

$$K_0 h_{ev} + (1 - K_0) h_c = h_1 \,, \tag{8}$$

where K_0 is a variable ratio of fresh supply air for ventilation purposes. Then the enthalpy h_{ev} determined from Eq. (8) is written as

$$h_{ev} = \frac{h_1 - (1 - K_0) h_c}{K_0} \,. \tag{9}$$

The moisture content of fresh supply air after the HP evaporator can be expressed from the material balance equation of moisture content for the whole system, which has the form

$$G_{tot}(d_2 - d_1) + G_0 d_0 = G_0 d_2 + G_0(d_0 - d_{ev}). \tag{10}$$

Dividing left and right parts of Eq. (10) by the value G_{tot}, after a series of transformations the following relationship is obtained

$$d_{ev} = d_2 - \frac{d_2 - d_1}{K_0} \,. \tag{11}$$

The temperature at the outlet of the HP evaporator, t_{ev}, can be determined using the *h-d* diagram for moist air on the line at the point of its intersection with h_{ev} or d_{ev}.

The ratio of fresh supply air for ventilation purposes can be determined from the energy balance equation of the HP

$$Q_c = Q_{ev} + L_c \,. \tag{12}$$

Components of Eq. (12) are defined as follows:

i) the heat flow taken from the HP condenser

$$Q_c = G_{tot}(h_c - h_{ph}); \tag{13}$$

ii) the heat flow supplied to the HP evaporator

$$Q_{\mathrm{ev}} = G_0(h_{\mathrm{pc}} - h_{\mathrm{ev}}); \tag{14}$$

iii) the expenditure of the external electrical energy for the HP compressor

$$L_{\mathrm{c}} = Q_{\mathrm{ev}}/\varepsilon_{\mathrm{HP}}; \tag{15}$$

where $\varepsilon_{\mathrm{HP}}$ is a refrigeration coefficient of the HP (COP – coefficient of performance).

From Eq. (12), taking into account relationships (13)–(15), after some transformations, the expression to determine a variable ratio of fresh supply air takes form

$$K_0 = \frac{h_{\mathrm{c}} - h_{\mathrm{ph}}}{h_{\mathrm{pc}} - h_{\mathrm{ev}}} \frac{\varepsilon_{\mathrm{HP}}}{\varepsilon_{\mathrm{HP}} + 1} \,. \tag{16}$$

The HP efficiency, which works as a refrigeration machine in air conditioning mode, is estimated in this case by the refrigeration coefficient (COP)

$$\varepsilon_{\mathrm{HP}} = \frac{1}{\dfrac{273.15 + t_{\mathrm{c}} + \Delta t_{\mathrm{c}}}{273.15 + t_{\mathrm{ev}} - \Delta t_{\mathrm{ev}}} - 1}, \tag{17}$$

where: t_{ev} – the temperature of fresh supply air at the outlet of the HP evaporator, $^\circ$C; t_{c} – the temperature of the exhaust air at the outlet of the HP condenser,$^\circ$C; Δt_{ev} – the temperature difference between flows of the supply air and the refrigerant at the outlet of the HP evaporator, $^\circ$C; Δt_{c} – the temperature difference between flows of the refrigerant and the exhaust air at the outlet of the HP condenser, $^\circ$C.

Different studies provide numerical values for temperature differences in the condenser and the evaporator of air-to-air heat pumps. According to [11] the following can be taken for the condenser and the evaporator of a HP $\Delta t_{\mathrm{c}} = \Delta t_{\mathrm{ev}} = 10^\circ$C.

The exhaust air enthalpy at the outlet of the HP condenser is determined from the energy balance equation for the entire system

$$G_0 h_0 + L_{\mathrm{c}} + Q_{\mathrm{a.c}} = G_0 h_{\mathrm{c}} + Q_d \,, \tag{18}$$

where Q_d is the heat flow extracted out of the system with drainage of moisture condensation from air passing through the HP evaporator (due to low values of this parameter it is neglected in the current analysis), W.

Dividing left and right parts of Eq. (18) by the value G_{tot} and taking into account the Eq. (1) for $Q_{\text{a.c}}$ after a series of transformations the following formula is obtained

$$h_{\text{c}} = h_0 + (h_{\text{pc}} - h_{\text{ev}})\frac{1}{\varepsilon_{\text{HP}}} + \frac{c_{\text{p}}K(t_0 - t_2)}{K_0}\,. \tag{19}$$

The temperature at the outlet of the HP condenser t_{c} can be determined using the h-d diagram for moist air at the intersection point of h_{c} with the moisture content inside the premise d_2.

The enthalpy h_1 can be determined with sufficient accuracy by the following calculation formula for moist air [13]

$$h_1 = 1.005t_1 + (2500 + 1.8t_1)d_1\,. \tag{20}$$

The HP refrigeration coefficient, as well as the COP, depends only on operating conditions of the HP, i.e. temperature levels of heat transfer processes taking place in the HP evaporator and condenser. Therefore, in order to characterize the efficiency of the entire ventilation and air conditioning HPS it is more appropriate to use a comprehensive indicator (the refrigeration coefficient of the entire system), which can be represented as

$$\varepsilon_{\text{HPS}} = \frac{Q_{\text{a.c}}}{L_{\text{c}}}\,, \tag{21}$$

where: $Q_{\text{a.c}}$ – refrigeration load produced at the entrance to the production premise and it is defined as, kW,

$$Q_{\text{a.c}} = G_{\text{tot}}(h_0 - h_1). \tag{22}$$

Taking into account Eqs. (22) and (14), (15), the expression (21) can be written as

$$\varepsilon_{\text{HPS}} = \frac{\varepsilon_{\text{HP}}(h_0 - h_1)}{K_0(h_{\text{pc}} - h_{\text{ev}})}\,. \tag{23}$$

6   Calculation analysis of the ventilation and air conditioning heat pump system

The calculation analysis of the HPS parameters is performed for a typical production premise with a given technological mode. The production workshop of the Roshen confectionery factory in the Kyiv region was chosen as a prototype [17]. To ensure comfortable working conditions in the

workshop premise, the following indoor air parameters were set [14]: room temperature $t_2 = 18°C$; relative humidity in the room $\varphi_2 = 50\%$; supply air overheating for the ventilation system $\Delta t_{\text{vent}} = 3°C$. Moisture contents of air at the inlet and outlet of the premise were determined under given conditions, i.e. at points 1 and 2 of the workflow diagram in Fig. 2: $d_1 = 5.3$ g/kg$_{\text{d.a.}}$ and $d_2 = 6.5$ g/kg$_{\text{d.a.}}$.

The calculation analysis using relations (6), (9), (11), (16), (17), (19), and (23) by the method of successive approximations makes it possible to assess effects of changes in temperature and relative humidity of the environment, the recovery efficiency, characteristics of an object of ventilation and air conditioning both on the system's parameters (air temperature at the outlet of the HP evaporator and condenser (Fig. 3), the ratio of fresh supply air (Fig. 4), the HP refrigeration coefficient (Fig. 5)) and the thermodynamic efficiency of the whole system (Fig. 6).

Following values are taken for the recovery efficiency $\eta_r = 0$, 0.4, 0.6, 0.8. The case when $\eta_r = 0$ corresponds to the HPS which was analysed in [7], i.e. ventilation and air conditioning without prior recovery of the cold energy from the exhaust air.

When choosing the value of K, the following assumption is taken into account. According to the method given in [7] for the selected production premise, it was calculated that $K = 0.23$ (the ratio of heat flow to the premise to energy required for ventilation). The obtained value is true because for industrial and public buildings energy input requirements for ventilation can be several times higher than the corresponding heat inflow [16]. Considering this the following values are taken: $K = 0.1, 0.3, 0.5$.

Figure 3a–3c shows graphical relationships between air flow temperatures at the outlet of the HP condenser, t_c, and evaporator, t_{ev}, the environment temperature and relative humidity (a), the value of K (b) and the recovery efficiency (c) (t_c, $t_{\text{ev}} = f(t_0, \varphi_0, K, \eta_r)$). Temperatures t_c are limited by a critical value at which normal operation of the HP condenser is still possible. It is seen that application conditions of this system deteriorate with increasing environment relative humidity. The application range of this system is from $t_0 = 27°C$ ($\varphi_0 = 80\%$) to $t_0 = 34°C$ ($\varphi_0 = 40\%$), which is 1–2°C wider than in the case without recuperation [7]. Temperatures t_{ev} at $t_0 > 20°C$ almost do not depend on the relative humidity φ_0 and the temperature t_0 and on average are 6°C. The use of recuperation has the greatest positive effect on the temperature after the HP evaporator, t_{ev}, while temperature limits after the HP condenser t_c are almost unchanged. Thus, as in [7], this system is suitable for applica-
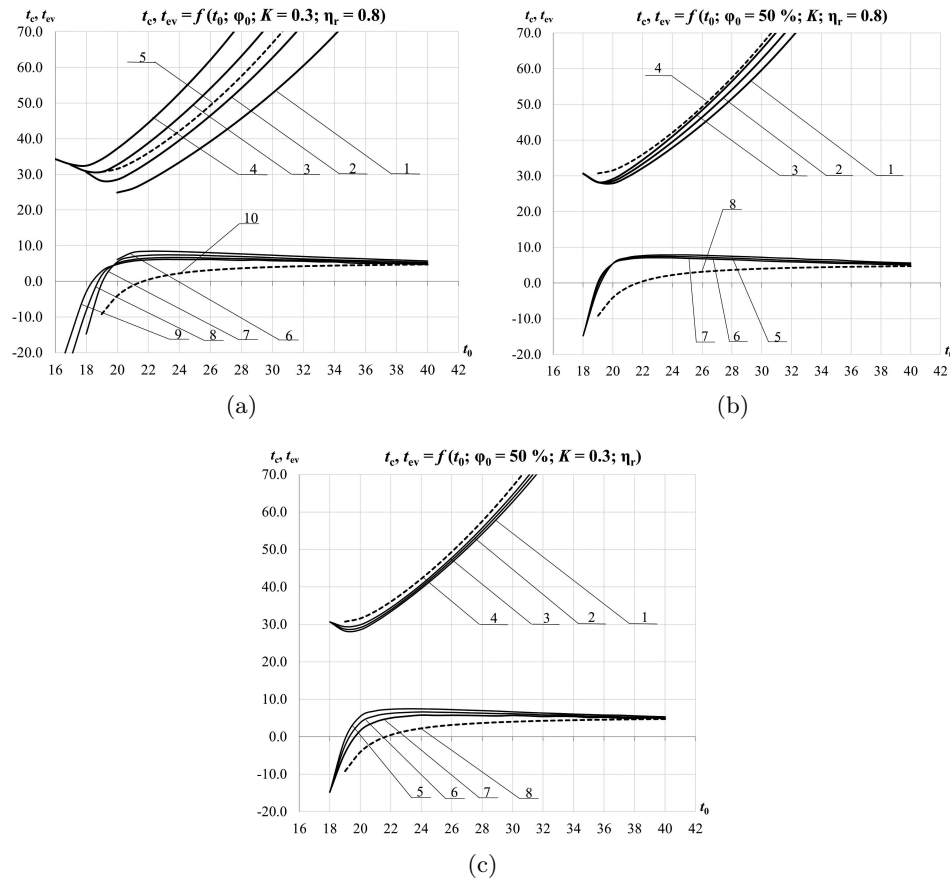
Figure 3: Relationships between air flow temperatures at the outlet of the HP condenser and evaporator, the environment temperature and relative humidity (a), the value of K (b) and the recovery efficiency (c): a) $K = 0.3$ and $\eta_r = 0.8$: 1–4 (t_c) – $\varphi_0 = 40\%, 50\%, 60\%$, and 70%; 5 ($t_c$) – $\varphi_0 = 50\%$ and $\eta_r = 0$; 6–9 (t_{ev}) – $\varphi_0 = 40-70\%$; 10 (t_{ev}) – $\varphi_0 = 50\%$ and $\eta_r = 0$; b) $\varphi_0 = 50\%$ and $\eta_r = 0.8$: 1–3 (t_c) – $K = 0.1, 0.3, 0.5$; 4 $t(_c)$ – $K = 0.3$ and $\eta_r = 0$; 5–7 (t_{ev}) – $K = 0.1, 0.3, 0.5$; 8 (t_{ev}) – $K = 0.3$ and $\eta_r = 0$; c) $\varphi_0 = 50\%$ and $K = 0.3$: 1–4 (t_c) – $K - \eta_r = 0.8, 0.6, 0.4, 0.0$; 5–8 ($t_{ev}$) – $\eta_r = 0.8, 0.6, 0.4, 0.0$.

tion in countries with a temperate continental climate with low relative humidity.

An important characteristic of this system is the ratio of fresh air at the entrance to the system K_0 which is then supplied to the HP evaporator. The corresponding calculated data for the variable ratio of fresh air obtained by Eq. (16) are shown in Fig. 4.
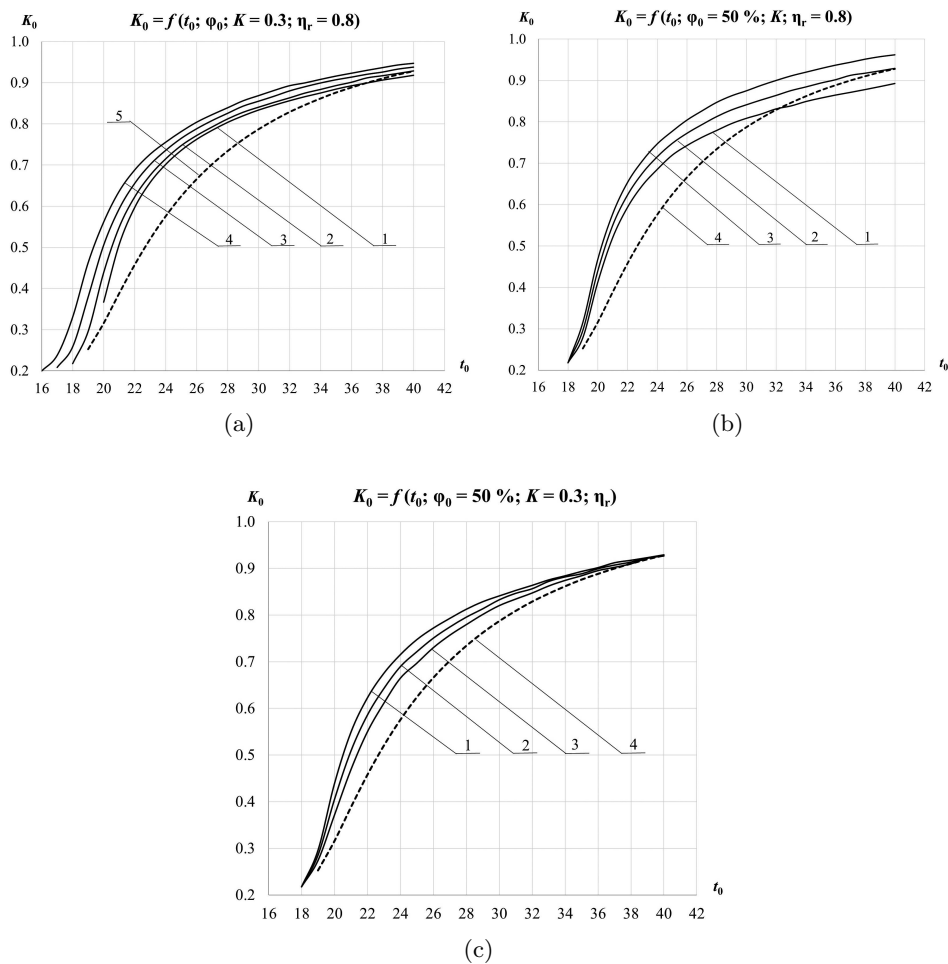
Figure 4: Relationships between the variable ratio of fresh supply air, the environment temperature and relative humidity (a), the value of K (b) and the recovery efficiency (c): a) $K = 0.3$ and $\eta_r = 0.8$: 1–4 – $\varphi_0 = 40\text{--}70\%$; 5 – $\varphi_0 = 50\%$ and $\eta_r = 0$; b) $\varphi_0 = 50\%$ and $\eta_r = 0.8$: 1–3 – $K = 0.1, 0.3, 0.5$; 4 – $K = 0.3$ and $\eta_r = 0$; c) $\varphi_0 = 50\%$ and $K = 0.3$: 1–4 – $\eta_r = 0.8, 0.6, 0.4, 0.0$.

Plots show that the ratio of fresh supply air to maintain the set comfortable parameters of indoor air should vary depending on both the temperature t_0 and the relative humidity φ_0 of the environment. At a constant temperature t_0 the demand for fresh air increases with increasing relative humidity φ_0, recovery efficiency η_r and the coefficient K, i.e. the heat flow to the premise. As φ_0 increases, the enthalpy h_0 increases as well. This, along with the
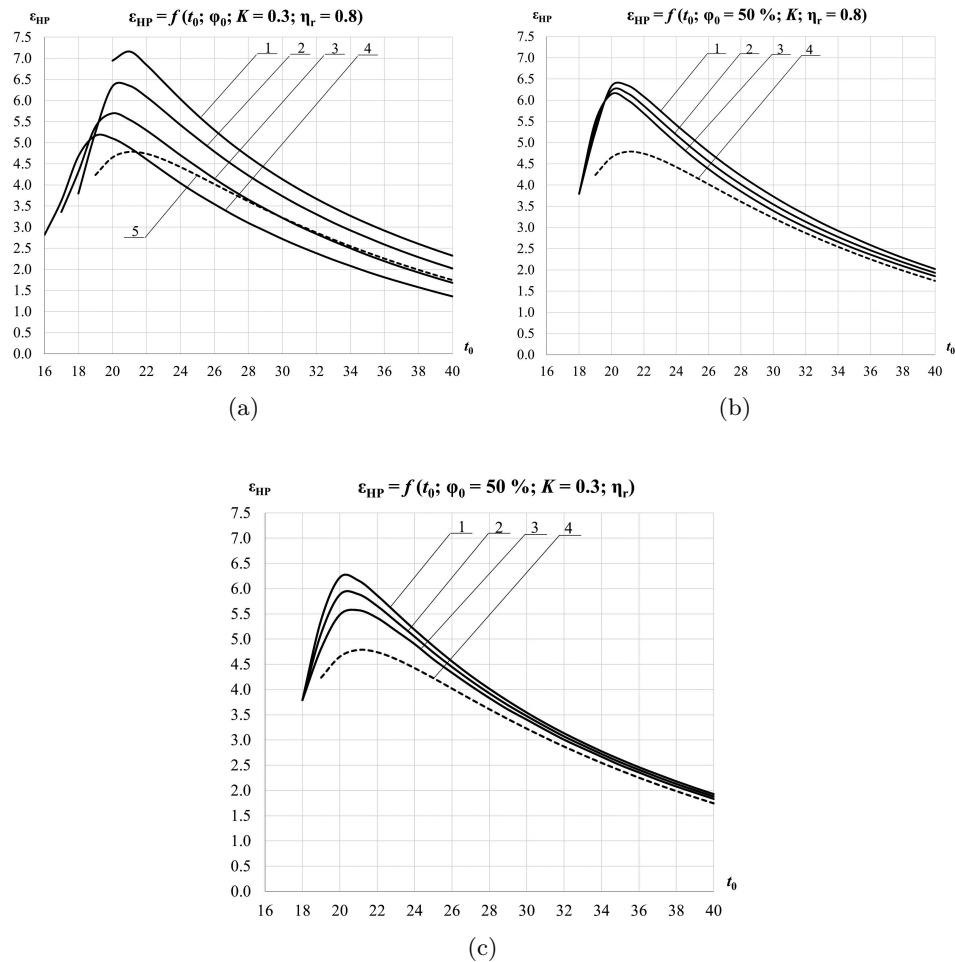
Figure 5: Relationships between the HP refrigeration coefficient, the environment temperature and relative humidity (a), the value of K (b) and the recovery efficiency (c): a) $K = 0.3$ and $\eta_r = 0.8$: 1–4 – $\varphi_0 = 40$–70%; 5 – $\varphi_0 = 50\%$ and $\eta_r = 0$; b) $\varphi_0 = 50\%$ and $\eta_r = 0.8$: 1–3 – $K = 0.1, 0.3, 0.5$; 4 – $K = 0.3$ and $\eta_r = 0$; c) $\varphi_0 = 50\%$ and $K = 0.3$: 1–4 – $\eta_r = 0.8, 0.6, 0.4, 0.0$.

higher utilisation of the exhaust air cold energy in the recuperator, leads to an increase in the temperature t_c of the exhaust air after the HP condenser. As a result, the mixing chamber receives a larger share of fresh supply air at higher relative humidity φ_0. Therefore, the design of the ventilation and air conditioning HPS should be carried out with a variable supply of fresh air, because otherwise the system will not provide a given technological mode
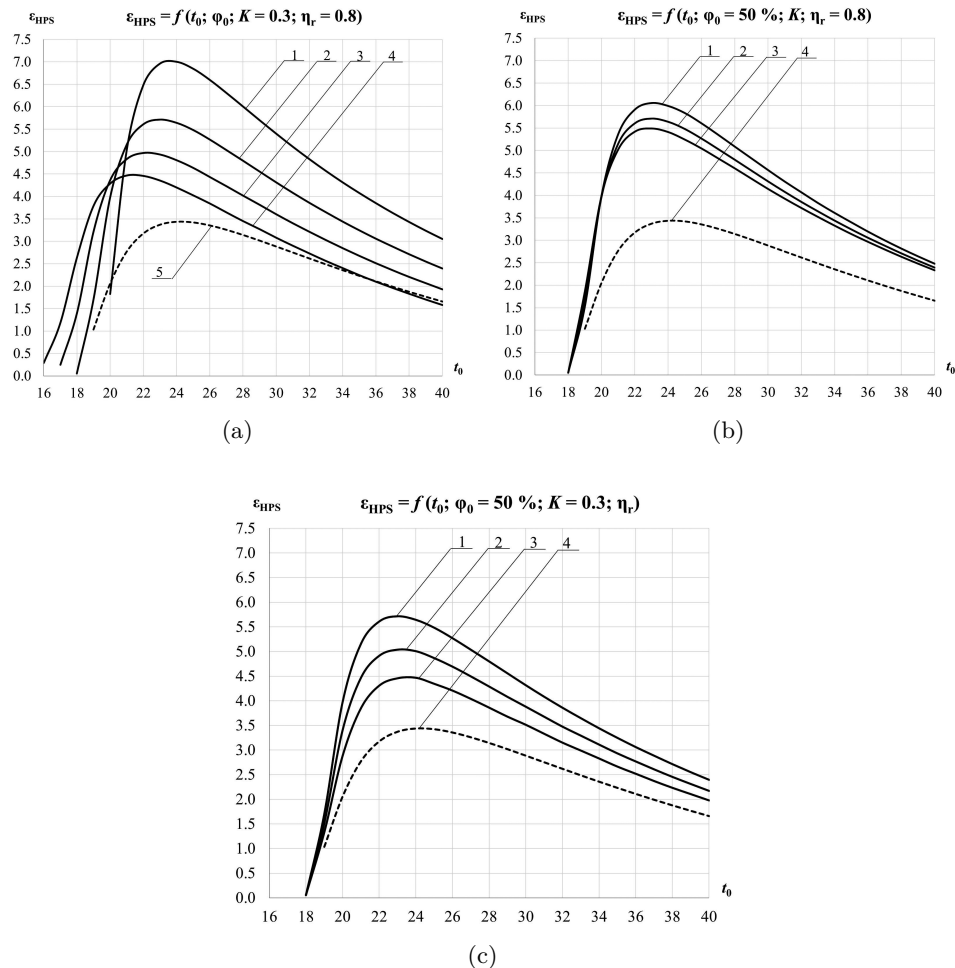
Figure 6: Relationships between the HPS refrigeration coefficient, the environment temperature and relative humidity (a), the value of K (b) and the recovery efficiency (c): a) $K = 0.3$ and $\eta_r = 0.8$: 1–4 – $\varphi_0 = 40-70\%$; 5 – $\varphi_0 = 50\%$ and $\eta_r = 0$; b) $\varphi_0 = 50\%$ and $\eta_r = 0.8$: 1–3 – $K = 0.1, 0.3, 0.5$; 4 – $K = 0.3$ and $\eta_r = 0$; c) $\varphi_0 = 50\%$ and $K = 0.3$: 1–4 – $\eta_r = 0.8, 0.6, 0.4, 0.0$.

inside the premise with changes in environmental parameters. In general, the use of recuperation increases the ratio of fresh supply air in the total air flow.

Based on the calculation analysis of relations (17) and (23) graphical relationships between HP and HPS refrigeration efficiencies, temperature and relative humidity of the environment, recovery coefficient and various val-

ues of K were obtained ($\varepsilon_{\mathrm{HP}}$, $\varepsilon_{\mathrm{HPS}} = f(t_0,\ \varphi_0,\ K,\ \eta_{\mathrm{r}})$) (Figs. 5 and 6). The HP refrigeration coefficient is characterized by higher values in the range of moderate temperatures $t_0 = 20-30°\mathrm{C}$ comparing to the system without recuperation. However, with a further increase in t_0 and φ_0 a significant decrease in the HP efficiency is visible (Fig. 5).

Figure 6 shows that the energy efficiency of this HPS reaches highest values also in the area of relatively low environment temperatures and largely depends on the relative humidity and the recovery efficiency. Values of the HPS refrigeration efficiency do not depend much on the premise characteristics, i.e. the value of K, in the entire temperature range of t_0.

In general, it can be noted that the preliminary recovery of the exhaust air cold energy to precool fresh supply air significantly increases the energy efficiency of the entire heat pump ventilation and air conditioning system to maintain comfortable and technological conditions in the production premise.

7   Conclusions

1. The research of this ventilation and air conditioning HPS showed the advantage of the cold energy recovery, compared to the system studied in [7], to provide a higher HPS refrigeration coefficient in the operating range of environment temperatures at considered values of the environment relative humidity. Theoretically, the maximum efficiency of the system (COP of the system) is about 4 (at critical air temperatures after the HP condenser at air parameters $t_0 = 32-34°\mathrm{C}$ and $\varphi_0 = 40-50\%$).

2. The studied system of ventilation and air conditioning inside premises, where maintenance of technological modes (temperature and humidity) during the warm season is required, can work effectively to some moderate values of environment parameters which further increase leads to critical temperatures of the exhaust air after the HP condenser.

3. The HPS has the highest energy efficiency in the area of relatively low environment temperatures and relative humidity. This suggests that such HPS is suitable for application in countries with temperate continental climates, which are characterized by low relative humidity.

Results of the study will be used for further scientific work on the improvement of existing heating, ventilation and air conditioning heat pump systems.

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