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Research Article

Combined heat pump heating and ventilation system using heat of soil, sewage water and ventilation emissions

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ABSTRACT

Currently, one of the main energy consumers in many countries is the communal sector. In this case, the bulk of the energy is supplied by fossil sources. In this regard, studies in the field of alternative energy sources that can help solve not only the problem of energy saving, but also the problem of environmental pollution and prevent the approaching environmental collapse deserve special attention. One of the most promising sources of alternative energy in the field of communal utilities is heat pump that use the low-temperature heat of renewable sources and the heat of the upper layers of the soil. Soil, as an energy source, has an almost constant and sufficiently high temperature level, which determines the sufficient efficiency of using its heat. However, the widespread occurrence of this type of heat pump is hindered by the high cost of the ground heat exchanger and its installation. In this regard, the development of new heat supply schemes having higher energy efficiency is relevant. To solve this problem, a combined heating and ventilation scheme based on ground heat pump was developed. The feature of this scheme is the use of additional low-temperature energy sources, such as the heat of ventilation emissions and wastewater of a heat supply facility. On the basis of the scheme the thermodynamic model in which the equation of the basic balance of separate elements and the scheme as a whole found the defining parameters characterizing work of system is developed. An algorithm for determining parameters at the nodal points of the chain is proposed, on the basis of which a numerical analysis of the circuit is performed. The results of the analysis are presented in the form of graphical dependencies. Features of operation of the scheme depending on the set initial parameters and ambient temperature are shown. Conclusions are made on the possible reduction of capital and operating costs for heating and ventilation due to the proposed solution.

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INTRODUCTION

Currently, energy consumption in the construction sector is about 40% of total global energy consumption [1]. This energy is mainly supplied through fossil fuels (coal, gas, oil), which cause about 36% of greenhouse gas emissions in the world [2]. According to a study by the International Energy Agency (IEA), energy consumption in the public sector is 41% of total energy consumption in Europe [3]. Therefore, at the moment, the issue of energy efficiency and resource conservation is particularly acute.

To solve these problems, it is necessary to reduce energy consumption in the communal sector. This can be achieved through energy-saving solutions in construction (minimization of heat loss, recovery) and through the introduction of alternative highly efficient energy sources, one of which is a heat pump [4].

Heat pump technologies deserve special attention in the global energy market. The conversion coefficient of such devices ranges from 3-8 units, depending on the heat source of the lower circuit. The most effective from this point of view are ground-based heat pumps that use energy accumulated in the soil to meet the heat supply needs of the building. Their main advantage is an almost constant temperature of a low-grade heat source, which ensures a stable system operation. The conversion coefficient of soil heat pumps, compared to air pumps, is high and stable year-round, therefore, it allows to achieve significant energy savings during long-term operation, but requires significant capital investments during construction (soil heat exchanger, well drilling, installation works). In this regard, studies are gaining relevance with the aim of increasing the energy and overall efficiency of heat supply systems using ground heat pump units (HPU).

Recently, a lot of research has been done on the development and optimization of soil HPU systems. Cho and Choi [5], made a quantitative assessment of the impact of design parameters on the operation of the installation. It was determined that reducing the length of the ground heat exchanger (GHE) is a determining factor in saving the total capital cost of installing the system. Choi et al. [6] performed a numerical simulation of a vertical type GHE under intermittent conditions in moist soil. It is concluded that it is very important to consider soil moisture when designing and evaluating the GHE. Lee et al. [7,8] carried out a mathematical analysis of the heat transfer efficiency in the soil contour and compared the existing models of the structures of the vertical GHE, as a result of which several of the most effective ones were identified. A lot of research work was carried out to optimize the design of the soil contour system of HPU [9-14].

From the above review of the literature, we can conclude that the development in this direction was mainly associated with the design of new structural solutions, which can increase the efficiency of the system, however, such solutions have a negligible effect on the scheme as a whole. In our opinion, work related to the development and study of combined heat supply systems using various types of heat pumps (including ground pumps), heat recovery units, and also additional low-potential energy sources can be no less effective. For example, in works [15, 16, 17], fundamentally new heat supply schemes were developed using heat pump technologies and several low-temperature heat sources, the results of a thermodynamic analysis of the developed schemes are presented, and optimal conditions for the operation of the ground heat exchanger are studied. In these studies, it was proposed to use a combination of lower heat sources, such as soil, conditionally clean wastewater, ventilation emissions. As a result of such a combination, not only the operational characteristics of the heat supply systems improved, but also the capital costs decreased (installation of the system, well drilling, ground heat exchanger dimensions).

This paper is a logical continuation of previous work. A feature of the proposed combined scheme with a ground HPU, as an object of study, is the use of additional heat sources both in the upper circuit due to the installation of a heat exchanger and utilization of the heat of ventilation emissions for preheating the supply air, and in the lower circuit due to the use of heat conditionally clean wastewater for additional heating of the coolant after the ground heat exchanger.

DESCRIPTION OF THE DESIGNED SYSTEM

The developed heating and ventilation scheme based on a ground-based heat pump with the additional use of the heat of conditionally clean wastewater and ventilation emissions from a house is shown in Figure 1.

The main feature of this solution is the combination of soil VT with additional low potential energy sources, such as heat of ventilation emissions Q_v and wastewater Q_{SW}. According to the scheme, on one hand, a wastewater heat exchanger (H_{SW}) is installed in the lower (ground) circuit of the heat pump, due to which the glycol solution, which took the heat Q_{GHE} from the ground, is heated from the temperature t_{GHE}^{ex} to t_1, thereby utilizing the waste heat of the sewage. On the other hand, a heat exchanger-recuperator (R) is installed in the original circuit, with which fresh air in the mechanical ventilation system is heated from temperature t_0 to t_H, due to the heat of the exhaust air, which is cooled from t_{in} to t_{or}. Further, the supply air stream passes through the heater (H), where it heats up to the set temperature t_{in} in the room.

In this scheme, due to the ground heat pump, the heat consumption is covered both for ventilation Q_v and for heating Q_h. Using an additional low-potential heat source, it becomes possible to increase the efficiency of the heat supply scheme based on ground heat pumps due to the waste heat of sewage and exhaust air. Due to this solution, it is

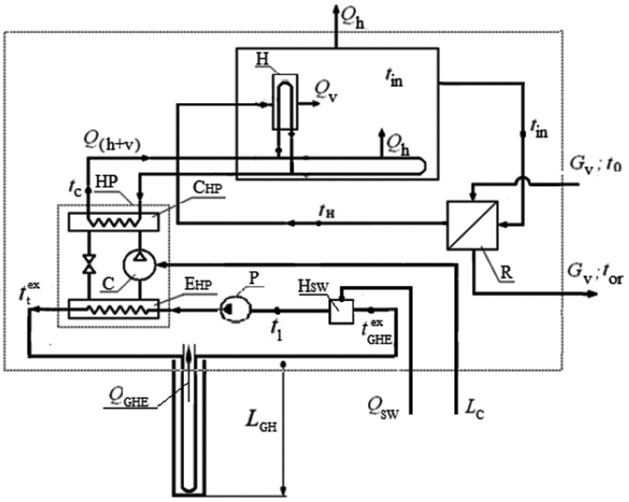

Figure 1. Schematic diagram of the combined heating and ventilation system based on ground heat pump with the additional using of heat of ventilation emissions and wastewater: HP–heat pump; C_{HP} – condenser; E_{HP} – evaporator; C – compressor; H_{SW} – heat exchanger for recover heat of sewage water; H–heater of fresh air; P – pump; R – recuperator.

possible to reduce not only the operational costs of heat supply (electricity), but also capital - by reducing the size of the ground heat exchanger.

THERMODYNAMIC ANALYSIS OF THE DEVELOPED SCHEME

The thermodynamic analysis of the developed scheme begins with the determination of the main quantities at the nodal points of the system and the assessment of the influence of additional energy sources on the parameters of the scheme. Using the heat of wastewater, it is necessary to provide for their separation into conditionally clean (shower, bath, washbasins, sinks for washing dishes) and cold (toilet) [18]. This separation is necessary for the intensification of processes in heat exchangers-utilizers, because the temperature of conditionally clean wastewater can reach an average of 32°C, while for cold this temperature does not exceed 10°C. The power of such a heat source was estimated in [15] and its level was determined by the ratio of heat for hot water supply Q_{HWS} to heat for heating Q_h. It was assumed that

$$Q_{SW} = Q_{HWS}, \tag{1}$$

then Q_{SW} can be estimated as

$$Q_{SW} = K^e Q_h^e, \tag{2}$$

where K^e – coefficient characterizing the quantitative ratio of heat on the hot water supply to the heat on heating at the calculated outdoor temperature, and Q_h^e – heat flow for heating under design conditions, respectively, kW.

If the right side of equation (2) is multiplied and divided by the heat consumption under current conditions Q_h, then a similar expression can be obtained that is valid at any ambient temperature

$$Q_{SW} = KQ_h, \tag{3}$$

where the coefficient K, in turn, is defined as

$$K = K^e \frac{t_{in} - t_0^e}{t_{in} - t_0}, \tag{4}$$

t_{in} – the required temperature level in a heated room, °C; t_0– temperature of atmospheric air, °C; t_0^e – design outside air temperature, °C.

Next, we determine the temperature of the glycol solution t_t^{ex} at the outlet of the HP evaporator from the balance equation of the soil heat exchanger, at a given and fixed temperature t_{GHE}^{ex} at the outlet of the GHE

$$t^{ex} = t_{GHE}^{ex} - \frac{4q_G L_{GH}}{w\pi d_{in}^2 \rho_t c_p}. \tag{5}$$

The specific total external energy consumption for the heating system can be determined similarly [19]

$$l_{h+v} = \frac{(L_c + L_P)}{Q_C} = \frac{L_c + L_P}{Q_h + Q_v}, \tag{6}$$

where L_c, L_P – external energy consumption for driving the compressor of HP and the lower circuit pump, respectively, kW; Q_C – heat flow diverted from the condenser of HP, kW.

To assess the efficiency of a real heat pump, we use the coefficient of performance defined as

$$\varphi = \varphi^e \eta_{HP}, \tag{7}$$

where – η_{HP} loss coefficient in HP, which is accepted at the level η_{HP}= 0,6 [18].

The COP of the ideal Carnot cycle can be written in this form [18]

$$\varphi^e = \left[1 - \frac{T_E}{T_C}\right]^{-1} = \left[1 - \frac{273 + t^{ex} - \Delta t_E}{273 + t_C + \Delta t_C}\right]^{-1}, \tag{8}$$

where T_E – absolute temperature of refrigerant evaporation in the evaporator of HP, K; T_C – absolute condensation temperature of the refrigerant in the condenser of HP, K; t^{ex} – coolant temperature at the outlet of the evaporator of HP, °C; t_C – water temperature at the outlet of the condenser of HP, °C;

Δt_E – temperature difference between the coolant and refrigerant flows at the outlet of the heat pump evaporator,

°C; Δt_C – temperature difference between the flows of the refrigerant and the heating coolant in the heating system at the outlet of the condenser of HP, °C. According to the recommendations in [18], it can be accepted that for a heat transfer fluid in a heat exchanger evaporator – $\Delta t_E = 5$°C, for water at the condenser of HP – $\Delta t_C = 5$°C.

Coolant temperature t_C, supplied to the heating system, determined by the ratio of [18]

$$t_c = t_{\text{in}} + \left(t^e - t_{\text{in}}\right)\left[\frac{\left(t_{\text{in}} - t_0\right)}{\left(t_{\text{in}} - t_0^e\right)}\right]^{\frac{1}{(1+n)}}, \tag{9}$$

where t_{in} – indoor air temperature is 20 °C; t_0 – outlet air temperature, °C; t^e – design temperature of the heating medium in the heating system at design temperature of ambient air t_0^e; $n = 0$ for low temperature heating systems. The estimated coolant temperature in the heating system is accepted $t_0^e = 45$°C.

External energy consumption the pump drive in equation (6) is defined as

$$L_P = \frac{V_t\left(\Delta p_{\text{sw+e}} + \Delta p_{\text{GHE}}\right)}{\eta_p \eta_{\text{dr}}}, \tag{10}$$

where $\Delta p_{\text{sw+e}}$, Δp_{GHE} – hydraulic pressure losses in the evaporator circuit, wastewater heat exchanger and in the ground heat exchanger, respectively, kPa; η_p i η_{dr} – efficiency of the pump and drive unit, respectively. We assume that pump efficiency is $\eta_p = 0{,}8$, and for drive unit $\eta_{dr} = 0{,}95$ [18].

The pressure loss in the vertical GHE is determined by the Darcy-Weisbach equation

$$\Delta p_{\text{GHE}} = \lambda \frac{\rho_t w^2}{2} \frac{L_{\text{GHE}}}{d_{\text{in}}}, \tag{11}$$

where λ – hydraulic friction coefficient, $L_{\text{GHE}} = 2L_{\text{GH}}$ – GHE pipe length, m.

The coefficient of hydraulic friction is determined depending on the flow regime in the circuit as:

- with laminar flow in smooth pipes (Re < 2300) - Poiseuille law

$$\lambda = 64/Re, \tag{12}$$

with turbulent flow (Re > 2300) - according to the Blasius equation

$$\lambda = \frac{0{,}3164}{Re^{0{,}25}}, \tag{13}$$

where $Re = \frac{wd_{\text{in}}}{\nu_t}$, and ν_t – kinematic viscosity of the coolant, m²/s.

The next step in the analysis is to evaluate the parameters of ventilation emissions. To do this, it is necessary to characterize the efficiency of the recuperator by the value of the recovery coefficient η_r [17]

$$\eta_r = \frac{t_{\text{in}} - t_{\text{or}}}{t_{\text{in}} - t_0}. \tag{14}$$

Based on equation (14), we determine the temperature of the ventilation air $t_н$ и t_{or} [18]

$$t_H = t_0 + \left(t_{\text{in}} - t_0\right)\eta_r. \tag{15}$$

$$t_{\text{or}} = t_{\text{in}} - \left(t_{\text{in}} - t_0\right)\eta_r. \tag{16}$$

To determine the quantitative characteristics of ventilation emissions, we introduce the coefficient m, defined as the ratio of the heat spent on ventilation of the room to the heat for heating

$$m = \frac{Q_v}{Q_h}. \tag{17}$$

Then the amount of heat spent on heating and ventilation will be equal

$$Q_v + Q_h = \frac{Q_v}{m} + Q_v = Q_v\left(\frac{1+m}{m}\right). \tag{18}$$

Further, after determining the main quantities at the nodal points and evaluating additional sources of heat, we proceed to assess the efficiency of the circuit. For this, we write the balance equation of the heat supply

$$G_v C_v t_0 + L_C + Q_{\text{GHE}} + L_P + Q_{\text{SW}} = Q_v + G_v C_v t_{\text{or}}. \tag{19}$$

From the balance equation of the lower circuit of HP we have

$$L_P = Q_E - Q_{\text{GHE}} - Q_{\text{SW}} = L_P\left(\varphi - 1\right) - Q_{\text{GHE}} - Q_{\text{SW}}. \tag{20}$$

Substituting equation (20) into (19), we have

$$L_c + L_c\left(\varphi - 1\right) + G_v C_v t_0 = Q_h + G_v C_v t_{\text{or}}, \tag{21}$$

where, after simple mathematical transformations, we can get an equation to determine the work of compressor

$$L_c = \frac{1}{\varphi} G_v C_v \left(t_{\text{in}} - t_0\right)\left(\frac{1+m}{m} - \eta_r\right). \tag{22}$$

Further, taking into account expressions (22), (10) and (18), equations (6) can be written as

$$l_{\text{h+v}} = \frac{\frac{1}{\varphi} G_v C_v \left(t_{\text{in}} - t_0\right)\left(\frac{1+m}{m} - \eta_r\right) + \frac{G_t C_t}{\eta_p \eta_{\text{dr}}} \frac{\left(\Delta p_{\text{sw+e}} + \Delta p_{\text{GHE}}\right)}{\rho_t C_t}}{G_v C_v \left(t_{\text{in}} - t_0\right)\left(\frac{1+m}{m}\right)}, \quad (23)$$

or after simplifications in the form

$$l_{\text{h+v}} = \frac{1}{\varphi}\left(1 - \eta_r \frac{m}{m+1}\right) + \frac{G_t C_t}{G_v C_v}\left(\frac{m}{m+1}\right) \frac{1}{\left(t_{\text{in}} - t_0\right)} \frac{\left(\Delta p_{\text{sw+e}} + \Delta p_{\text{GHE}}\right)}{\eta_p \eta_{\text{dr}} \rho_t C_t}. \quad (24)$$

It is seen that in equation (24) a complex arises $\frac{G_t C_t}{G_v C_v}$, which is the ratio of water equivalents of the coolants of the system and requiring additional definition. To do this, we write equation (19) as follows

$$\frac{G_t C_t \left(t_t^{\text{in}} - t_t^{\text{ex}}\right)}{\varphi - 1} + \frac{G_t C_t}{\eta_p \eta_{\text{dr}}} \frac{\left(\Delta p_{\text{sw+e}} + \Delta p_{\text{GHE}}\right)}{\rho_m C_m} + G_t C_t \left(t_{\text{GHE}}^{\text{ex}} - t_{\text{GHE}}^{\text{in}}\right) + K G_v C_v \left(t_{\text{in}} - t_0\right)\frac{1}{m} = G_v C_v \left(t_{\text{in}} - t_0\right)\frac{1}{m} + G_v C_v \left(t_{\text{or}} - t_0\right) \quad (25)$$

From equation (25) we obtain the dependence for determining the complex $\frac{G_t C_t}{G_v C_v}$

$$\frac{G_t C_t}{G_v C_v} = \frac{\left(t_{\text{in}} - t_0\right)\frac{1}{m}\left(1 - K\frac{\varphi}{\varphi - 1}\right) + \left(t_{\text{or}} - t_0\right)}{\frac{4 q_G L_{\text{GH}}}{w \pi d_{\text{in}}^2 \rho_t c_p} \frac{\varphi}{(\varphi - 1)} + \frac{\left(\Delta p_{\text{sw+e}} + \Delta p_{\text{GHE}}\right)}{\rho_t C_t \eta_p \eta_{\text{dr}}}}. \quad (26)$$

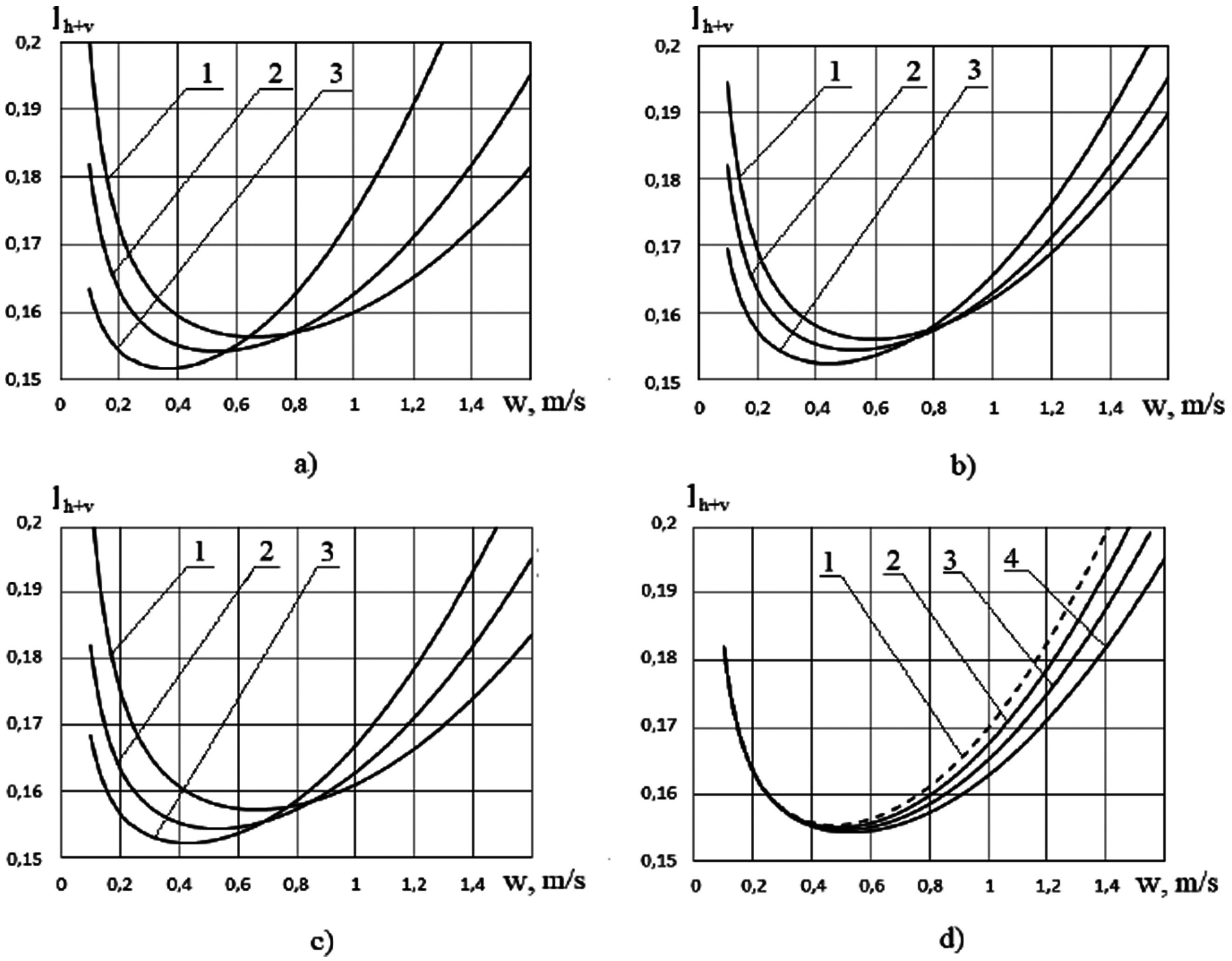

Figure 2. The dependence of the specific consumption of external energy for the heating and ventilation system from the coolant velocity in lower circuit: (a) 1 – 3 – q_G = 75; 50; 25 W/m accordingly; (b) 1 – 3 – L_{GH} = 100; 75; 50 m accordingly; (c) 1 – 3 – d_{in} = 0,025; 0,032; 0,04 m accordingly; (d) 1 – 4 – K^e = 0; 0,1; 0,2; 0,3 accordingly.

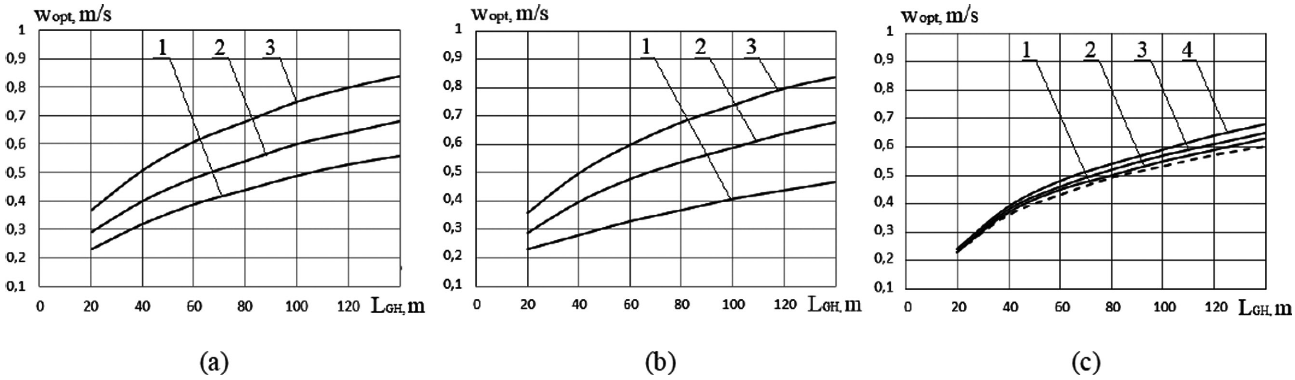

Figure 3. The dependence of the optimal coolant velocity in lower circuit from the depth of well: (a) 1 – 3 – d_{in} = 0,04; 0,032; 0,025 m accordingly; (b) 1 – 3 – q_G = 25; 50; 75 W/m accordingly; (c) 1 – 4 – K^e = 0; 0,1; 0,2; 0,3 accordingly.

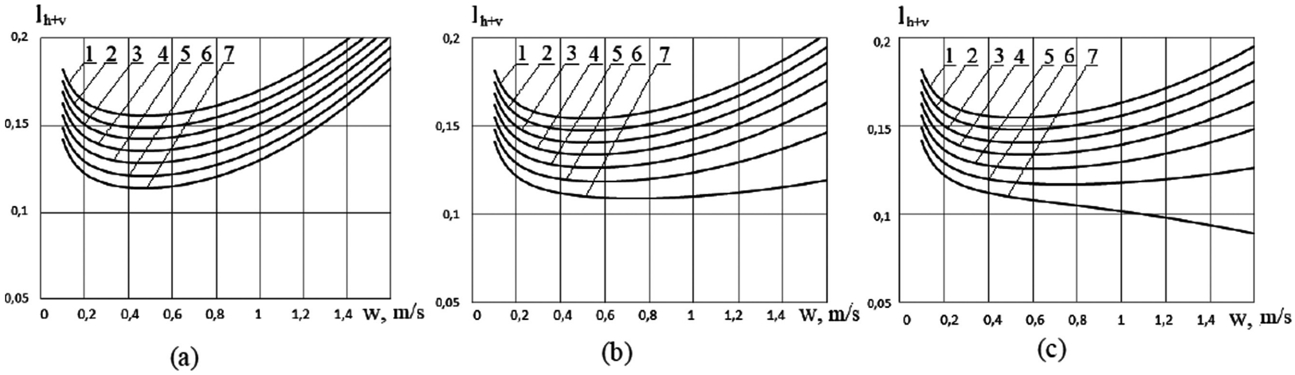

Figure 4. The dependence of the total consumption of external energy for heating and ventilation system from the coolant velocity in lower circuit: (a) K^e=0, (b) K^e =0,2, (c) K^e =0,3; 1 – 7 – t_0= -20…10 °C.

Then, taking into account (26), the specific consumption of external energy for heating and ventilation will be determined as follows

$$l_{\text{h+v}} = \frac{1}{\varphi}\left(1-\eta_r \frac{m}{m+1}\right) + \left(\frac{\left(1-K\frac{\varphi}{\varphi-1}\right)+m\frac{\left(t_{\text{or}}-t_0\right)}{\left(t_{\text{in}}-t_0\right)}}{\frac{4q_G L_{\text{GH}}}{w\pi d_{\text{in}}^2 \rho_t C_{@}} \frac{\varphi}{(\varphi-1)} + \frac{\left(\Delta p_{\text{sw+e}}+\Delta p_{\text{GHE}}\right)}{\rho_t C_t \eta_p \eta_{\text{dr}}}} \right) \frac{1}{1+m} \frac{\left(\Delta p_{\text{sw+e}}+\Delta p_{\text{GHE}}\right)}{\eta_p \eta_{\text{dr}} \rho_t C_t}. \tag{27}$$

Also, an interesting characteristic for evaluating the effectiveness of the system is the specific load on the GHE, which is defined as the ratio Q_{GHE} to Q_{h}+ Q_{v}. With the help of studies of this value, it becomes possible to evaluate the effect of replacing the soil heat exchanger with additional heat sources, which are proposed for combination. This effect, in turn, is directly related to capital costs for the construction of the ground circuit of HP (well drilling, installation of the circuit) and operating costs for the pump.

To determine this quantity, we use the equation

$$\frac{Q_{\text{GHE}}}{Q_{\text{h+v}}} = \frac{G_t C_t \left(t_{\text{GHE}}^{\text{ex}} - t_{\text{GHE}}^{\text{in}}\right)}{G_v C_v \left(t_{\text{in}} - t_0\right)\frac{1}{m}}. \tag{28}$$

In view of equation (26) for $\frac{G_t C_t}{G_v C_v}$, as well as equations (5) for the temperature difference at the outlet and entrance of the GHE, we finally have a dependence for determining the specific load of the GHE

$$\frac{Q_{\text{GHE}}}{Q_{\text{h+v}}} = \left(\frac{\left(1-K\frac{\varphi}{\varphi-1}\right)+m\frac{\left(t_{\text{or}}-t_0\right)}{\left(t_{\text{in}}-t_0\right)}}{\frac{4q_G L_{\text{GH}}}{w\pi d_{\text{in}}^2 \rho_t C_p} \frac{\varphi}{(\varphi-1)} + \frac{\left(\Delta p_{\text{sw+e}}+\Delta p_{\text{GHE}}\right)}{\rho_t C_t \eta_p \eta_{\text{dr}}}} \right) \frac{4q_G L_{\text{GH}}}{w\pi d_{\text{in}}^2 \rho_t C_p} \frac{1}{(1+m)}. \tag{29}$$

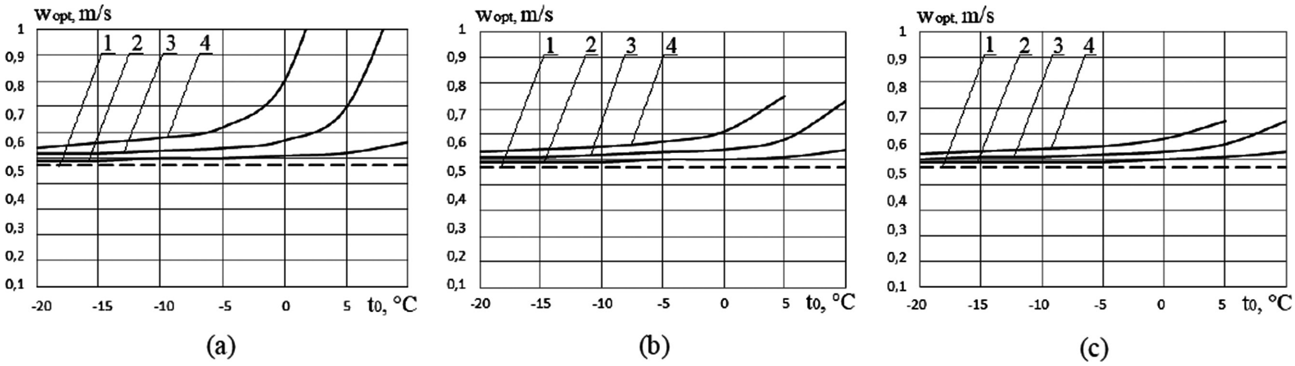

Figure 5. The dependence of the optimal coolant velocity in lower circuit from the temperature of atmospheric air: (a) m=0, (b) m=1, (c) m=2; 1 – 4 – K^e =0…0,3.

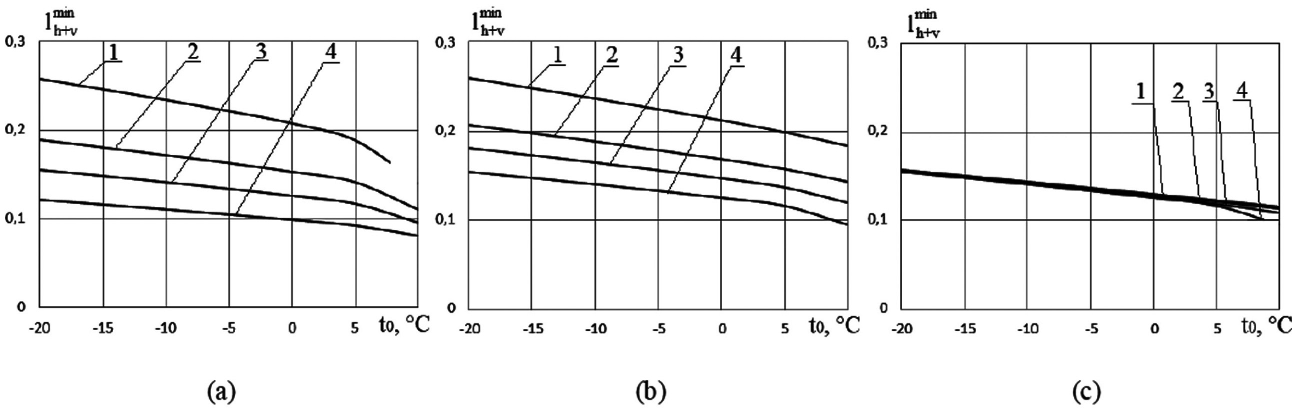

Figure 6. The dependence of the minimal unit consumption of external energy for system from the temperature of atmospheric air: (a) 1 – 4 – m =0…2; (b) 1 – 4 – η_r=0…0,8; (c) 1 – 4 – K^e =0…0,3.

CALCULATION CIRCUIT ANALYSIS

A numerical method was used to determine the main quantities characterizing the efficiency of the heat pump heating and ventilation system.

Variable parameters that are decisive in the calculations are taken at the level of real for the target objects according to preliminary studies [15,17]:

- the proportion of the waste water heat in the total heat consumption for heating under design environmental conditions K^e = 0…0,3;
- the ratio of the amount of heat spent on heating the ventilation air to the amount of heat on heating m = 0…2,0;
- efficiency of recuperative air heater according to recommendations [20] η_r = 0,4…0,8.

The first stage of evaluating the effectiveness of the proposed scheme is the study of the value of the specific consumption of external energy for the heating and ventilation system [21]. The dependence of this quantity on the velocity of the coolant of the lower circuit for the variable parameters q_G, L_{GH}, d_{in} and K^e is shown in Figure 2.

After analyzing the graphical dependences obtained as a result of the calculation, we can conclude that the influence of the use of wastewater of the building as an additional heat source on the total specific energy consumption is rather insignificant compared to the initial scheme (about 2% reduction). It is clearly seen that, as in previous studies [15], there is an optimal coolant velocity at which the specific energy consumption is minimal. A further task of the analysis is to study the dependence of the optimal speed on the determining quantities of the circuit (Figure 3).

It can be seen from the graphs that the change in the values of the pipeline diameter and the specific average annual heat capacity of the ground heat exchanger has a significant effect on the optimum speed, while the change in the share of wastewater heat does not significantly affect (displacement optimum speed in the direction of growth by 10%).

The results of calculating the optimal speed make it possible to pre-evaluate the operating conditions of the soil heat exchanger and provide design solutions that will meet the optimal conditions for using the heat of the soil under various operating conditions of the circuit.

To predict the change in the energy load of the heat supply system under different climatic conditions [22], the dependences of the specific consumption of external energy on the ambient temperature were obtained (Fig. 4).

It can be seen from the graphs that the minimum specific energy consumption clearly decreases with an increase in the ambient temperature and with an increase in the relative power of the wastewater heat exchanger (K^e) shifts towards an increase in the outside air temperature and the speed of the lower circuit coolant.

Quite interesting is the fact that at K^e = 0.3 (Figure 4, c) it is clearly seen that the last curve (t_0=10°C) falls out of the general regularity of the curves and the phenomenon of a minimum of specific energy consumption disappears as such. This is due to the fact that when the temperature of the outside atmospheric air rises, a moment comes when the heat demand for heating and ventilation is completely covered by the heat supplied to the lower circuit from the waste water heat exchanger, and the ground heat exchanger starts to work in reverse mode, sharply reducing the heat evaporator load and, accordingly, energy consumption for the HP compressor. Moreover, Figure 5 illustrates the nature of the change in the optimal velocity of the lower coolant from the outside temperature at various values of the relative power of the wastewater heat exchanger (coefficient K^e) and various relative heat consumption for ventilation (coefficient m). It is seen that the deviation of the optimal speed from a constant value increases with an increase in the coefficient K^e, which leads to a decrease in the heat load of the GHE and decreases with an increase in the coefficient m, that is, with an increase in the heat load of the upper circuit of the HP.

From the dependencies shown in Figure 5, it is also seen that there is a certain value of the ambient temperature, at which the optimum of speed begins to shift sharply towards growth. This phenomenon is observed at positive values of the outside air temperature. However, with an increase in the heat fraction for ventilation, the curves smooth out and the system becomes more stable. This phenomenon indicates the efficiency of using the proposed solution to the combination of several low-potential heat sources from an operational point of view [23].

Another important effect of using the combined circuit, from the point of view of operational features, is to reduce the minimum consumption of external energy for the heat supply circuit, which take place at optimal temperatures of the coolant in the lower circuit of the heat pump. The dependences of this quantity on the ambient temperature for variable determining parameters are shown in Figure 6.

It can be seen that the minimum specific consumption of external energy for the generation of a unit of heat significantly decrease with an increase in the heat fraction for ventilation and the recovery coefficient η_r of the recuperator, however, they practically do not depend on the coefficient K^e, which characterizes the relative power of the wastewater heat exchanger. This is due to the fact that the coefficients m and η_r characterize the degree of utilization of heat in the upper circuit and lead to a decrease in the heat consumption for heating and ventilation, and hence to a decrease in energy expenditures for heating, and an increase in the coefficient K^e only leads to the replacement of heat and does not affect the energy consumption of the compressor of HP.

An important characteristic of the proposed scheme is also the influence of the individual elements of the combined scheme on the relative power of the soil heat exchanger, which is directly related to its size [24], and therefore, to the corresponding capital costs for its construction. The dependence of this quantity on the ambient temperature is shown in Figure 7.

As can be seen from Figure 7, a - the effect of the additional installation of a wastewater heat exchanger on the specific load of ground heat exchanger is significant. So,

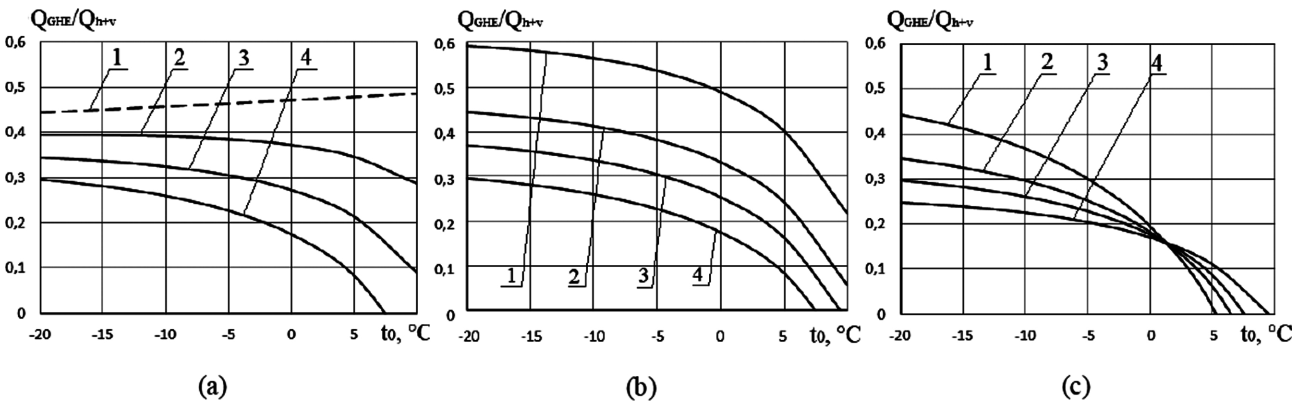

Figure 7. The dependence of the specific load of ground heat exchanger from the temperature of the air: a) 1 – 4 – K^e = 0…0,3; b) 1 – 4 – η_r= 0..0,8 c) 1 – 4 – m = 0…2.

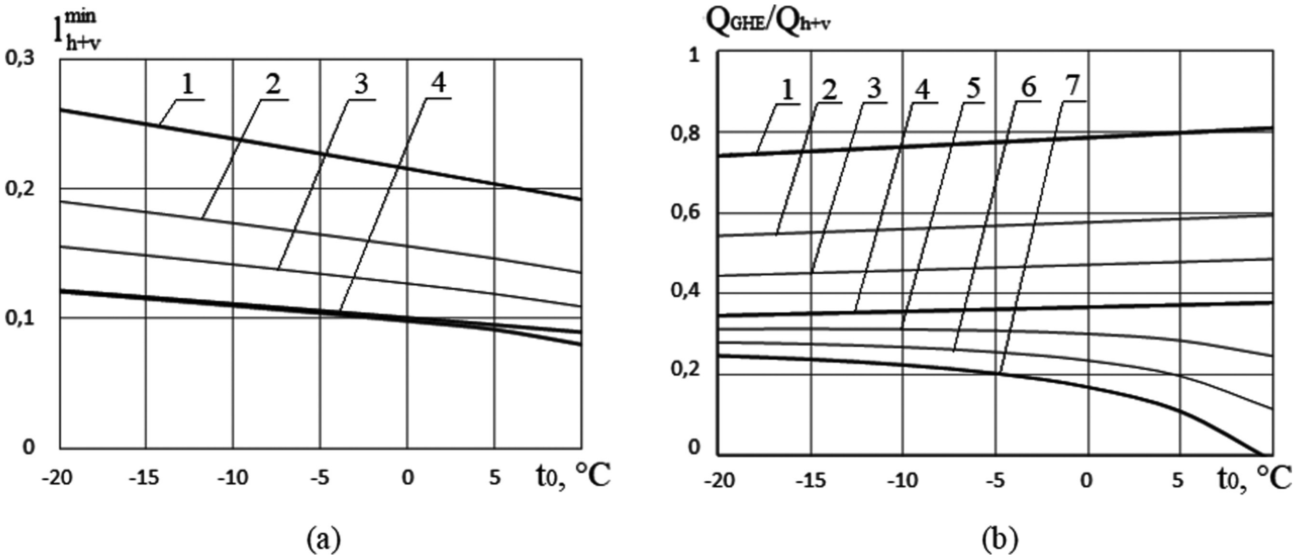

Figure 8. The dependence of the minimal specific energy consumption and the specific load of the ground heat exchanger from the temperature of atmospheric air: a) 1 – m = 0…2; 1 – 4 – K^e = 0 (η_r = 0; 0,4; 0,6; 0,8); 4 – K^e = 0…0,3; b) 1 – m = 0…2; 1 – 4 – K^e = 0 (η_r = 0; 0,4; 0,6; 0,8); 5 – 7 – K^e=0…0,3.

with an increase in the ambient temperature, in the absence of a wastewater heat exchanger, the load on the ground heat exchanger steadily increases, and in the presence of it, on the contrary, it decreases. It should also be noted that the effect of using such a solution is obvious: with an increase in the K^e coefficient from 0 to 0.3, the load on the GHE decreases by 35% at the design temperature t0, and with an increase in this temperature, the effect increases to 80%.

It should also be noted that the efficiency of the recuperation of ventilation emissions has a significant effect on the performance of the GHE: the installation of a highly efficient recuperator can reduce the load on the GHE and, as a result, capital costs by 2 times at the design temperature of the outside air.

It should be noted that no less significant effect is achieved by increasing the heat proportion on ventilation air heating. From the graphical dependencies, which are shown in Figure 7, c it may be seen that at a value of m = 2 (the proportion of the ventilation load of modern shopping and entertainment centers), the required capacity of the ground heat exchanger is almost halved compared to the scheme without ventilation (m = 0, residential private houses). As can be seen from Figure 7, c starting from a certain value of the ambient temperature (under these conditions, about 2.3°C), the noted pattern becomes inverse, that is, an increase in m leads to an increase in the specific load of the ground heat exchanger. This is due to the fact that with an increase in the heat consumption for ventilation (and, therefore, the total heat consumption), the need to use a ground heat exchanger is maintained up to higher outdoor temperatures.

A comparison of the influence of individual elements of the combined scheme on the indicators of its effectiveness is presented in Figure 8. Moreover, as can be seen from Figure 8, a, the specific external energy consumption for HP during operation without a recuperator does not depend on additional heat consumption for ventilation. When working with a recuperator, the specific energy consumption decreases with an increase in both the ventilation flow and the efficiency of the recuperator η_r. With the additional use of a wastewater heat exchanger, the specific consumption of external energy on the heat pump remains almost unchanged.

When analyzing the effect of circuit elements on the specific power of the ground heat exchanger (Figure 8, b), it can be noted that the characteristics of the ventilation flow and the efficiency of the recuperator have a similar effect as on the specific energy consumption of the heat pump, however, the additional use of the wastewater heat exchanger leads to reduce the required capacity of the GHE.

Analyzing the results shown in Figure 8, it should be noted that the use of additional low-temperature energy sources, such as the heat of ventilation emissions and wastewater of the building in the proposed scheme of heat pump heating and ventilation during the billing period, reduces the specific energy consumption of HP (as a result, operating costs) about 2 times, and the specific load on the GHE (as a result, the capital cost of the construction [23]) reduces by more than 3 times.

The thermodynamic analysis presented in this article shows the fundamental possibilities of increasing the efficiency of a combined heat pump system based on a ground heat pump by using additional heat sources in the lower and upper circuits of the system and can be used as a basis for further economic calculations and design of such a heat supply system for specific parameters and application conditions.

CONCLUSION

The use of additional sources of heat in the upper (ventilation emissions) and lower (conditionally clean waste water) circuit of the combined heating and ventilation system with ground heat pump is generally characterized by a significant positive effect.

The utilization of the ventilation emissions heat in the upper circuit using a recuperator for preheating the supply air leads both to a significant saving of external energy on the HP drive and to a decrease in the required capacity of the ground heat exchanger, which results in a decrease in its size and capital costs for its construction. At the same time, both the amount of energy saving on the HP drive and the decrease in the ground heat exchanger sizes increase with an increase in both the relative heat flow to ventilation and the recovery coefficient, which characterizes the efficiency of the recuperator.

The use of a wastewater heat exchanger practically does not affect the optimal value of the coolant speed in the lower circuit of the HP and the total specific consumption of external energy for the drive of the HP compressor and the circulation pump. At the same time, in the design mode and in the cold season, the use of wastewater heat in the lower circuit leads to additional replacement of the heat capacity of the ground heat exchanger, which entails a reduction in its size and corresponding capital costs.

NOMENCLATURE

C_p	Specific heat, kJ / kg °C
d	Diameter, m
G	Coolant mass flow, kg/s
K	Coefficient of hot water supply heat
L	Work, W
l	Specific total energy consumption
m	Coefficient of ventilation emissions heat
n	Temperature level coefficient
Δp	Hydraulic pressure losses, Pa
Q	Heat flow, W
q	Specific heat, W/m^2
Re	Reynolds complex
t	Temperature, °C
V	Coolant volume flow, m^3/sec.
w	Coolant speed, m/s

Greek symbol

η	Efficiency factor
λ	Hydraulic friction coefficient
ν	Kinematic viscosity, m^2/s
ρ	Density, kg/m^3
ϕ	Coefficient of performance

Subscripts

0	Ambient parameters
C	Condenser
c	Compressor
dr	Driven unit
E	Evaporator
GHE	Ground heat exchanger
h	Heating
HP	Heat pump
HWS	Hot water supply
in	Inlet
or	Out of recuperator
P	Pump
r	Recuperator
SW	Sewage water
t	water parameters

Superscripts

e	Design parameters
ex	Outlet parameters

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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