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Integration of ground and ventilation air energy for heating buildings

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Abstract: The authors have developed an improved version of the vapor compression system for heating and cooling civil buildings on the basis of the soil and ventilation air integrated energy. It is characterized by increased energy efficiency and the possibility of redistribution with automatic regulation of generated heat in subsystems. The results of the analytical study of the system established a multifactorial dependence of the actual conversion factor and led to greater efficiency in the transformation of the extracted heat from the soil and ventilation air. It simulates the multifactorial effect of the initial parameters and operating conditions on the possibility of highly efficient use of integrated energy in cold, warm and transitional periods of the year. The integrated use of soil energy and air flows in heat pump heating systems is distinguished by the possibility of its controlled redistribution with a decrease in the intensity of heat extraction via a ground heat exchanger, as well as the possibility of reducing the depth of the wells and the corresponding costs for the arrangement and operation of probe heat exchangers.

Keywords: heat pump, heat supply, conversion factor

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Introduction

It is known (Matsevite et al., 2007) that universal heat pump technologies based on the energy potential of various types of low-temperature sources (LTS) are becoming more and more widely used for heat and cold supply in buildings in all seasons. Heat pump systems for extracting heat from removed ventilation air, in comparison with the use of other LTS, are distinguished by minor capital and operating costs, and therefore their use is promising for the available temperature and resources (DSTU B V.2.5-44, 2010).

Utilization of heat of removed from the air of ventilated premises is characterized by the relatively constant temperature throughout the year (15-25°C), consequently, its use as an LTS allows one to reduce power, or completely eliminate the need for an additional energy source (AES) in heat pump heating systems (HPHS). Unlike traditional recuperation systems, including those with an intermediate energy carrier and rotating regenerative heat exchangers, heat pump systems can recycle the ventilation emissions heat throughout the year with the required depth of cooling.

Soil is one of the universal heat and power LTS for HPHS with a temperature of (8-12°C) at a depth of up to 150 m from the earth's surface (Lund, 2004), excluding the top layer up to 15 m, which is exposed to periodic temperature effects and solar radiation of varying intensity. In addition, soil is distinguished by the ability to accumulate available heat from the removed ventilation emissions, outdoor air (Vasil'ev et al., 2014; Vasil'ev et al., 2017), solar radiation (Heinrich et al., 2014) and other LTS.

Currently, the most reasonable way of extracting heat from soil mass (Dokumentacija; www.viessmann.ru; www.domteplo.ru) is based on drilling wells with an arrangement of tubular U-shaped loop heat exchangers, mostly made of plastic pipes. Characteristically, this method is used for heat pump systems with the general use of heating separately from the outside air, ground and exhaust air streams for low-temperature water heating, as well as air with reversible operation of the corresponding system (Heinrich et al., 2014), and makes up more than 85% of the total amount, generated for heating of buildings in Western Europe. Practical implementation of the analyzed systems ensures further improvement in the economy of traditionally burned fuel and the quality of the environment.

On the basis of results presented in (Bezrodnij et al., 2013; Bezrodnij et al., 2018; Vasil'ev, 2006; Petrash, 2014; Zabarnij et al., 2012; Patent 108184), one obtains that with the integrated use of several LTS in heat and cold supply systems for buildings, the possibilities for the selection and consumption of heat are expanded. The foregone predetermines the expediency of improving systems for utilizing heat from the removed air, and also when cooling the supply air in the warm season, with justification of alternating modes of selection and accumulation of heat in the soil mass.

Thus, the results of the analysis of known domestic and foreign scientific and technical developments demonstrate the need for further improvement of heating pump systems with the development of new technical solutions and relevant research. One should search for the rational conditions for the simultaneous use of the energy potential of soil and ventilation emissions for the highly efficient heat and cooling supply of civil buildings in the corresponding periods of the year.

1. The purpose and objectives of the study

The purpose of this work is the primary task of improving and developing new technical solutions with a theoretical substantiation of the efficiency saving energy in the process of simultaneous use of the available potential of soil and air flows and with the determination of rational subsystems operating modes for the selection, transformation and consumption of heat by subscriber systems with enhanced capabilities to reduce capital and operating costs compared to traditional HPS.

2. Research system and methodology

The structural and functional arrangement of the improved heat pump heating systems based on the energy of a binary low-temperature source is shown in Figure 1.

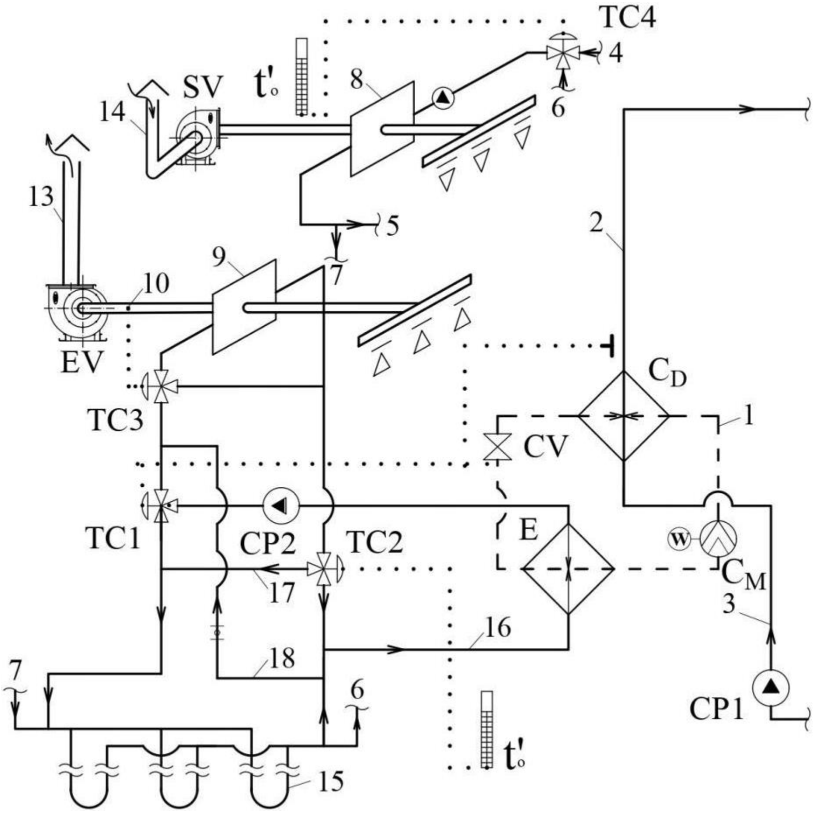

Fig. 1. Schematic diagram of a heat pump heat supply system based on integrated soil and ventilation air energy: 1 – heat pump circuit pipeline; 2, 3 – distribution and collection lines; 4, 5 – pipelines branch; 6, 7 – supply and return pipelines of the ground heat exchanger respectively; 8 – air heater; 9 – surface heat exchanger for exhaust air cooling; 10 – air humidity sensor; 13, 14 – ducts for exhaust air intake and the supply air; 15 – ground heat exchanger; 16 – general pipeline of the LTS heat selection subsystem; 17 – bypass pipeline section; 18 – bypass piping with check valve; E – evaporator; C_M – compressor; C_D – condenser; CV – choke valve; TC1-TC4 – temperature consumption controllers in the subsystem of heat extraction from LTS; CP1, CP2 – circulating pumps; EV, SV – exhaust and supply ventilation units respectively (*own study*)

The system consists of the corresponding circuits of the heat pump energy flows transformation, subscriber heat consumption and heat extraction from LTS.

The heat pump circuit contains a pipeline “1” connected in series with a evaporator E, compressor C_M with an external drive power W, condenser C_D and a choke valve CV, in which circulates the working fluid, for example freon. The local heat supply subsystem contains a closed pipeline circuit for the circulation of the energy carrier with common distribution “2” and collection lines “3” for a low-temperature system of radiant panel heating and hot water supply, as well as branch pipelines “4” and “5” for heating the air in the heater “8” of the supply ventilation system SV.

The circuit for extracting heat from the ground and removed ventilation air contains two parallel-connected circuits, respectively, with a ground heat exchanger “15” and a surface heat exchanger “9” for cooling ventilation emissions.

An aqueous solution of ethylene glycol circulates in the subsystem, which is pre-cooled in the evaporator E of the heat pump to a lower temperature relative to the temperature of the untouched soil mass. After the evaporator, the energy carrier under the action of the circulating pump CP2 moves with its division by a three-way temperature regulator of the flow rate TC1 into two parallel flows for entering the ground heat exchanger “15” and the surface heat exchanger “9” for cooling the exhaust air. After extracting off heat in each of them with different temperatures, the heated mixture of energy carriers through the corresponding pipelines with a total flow through pipeline “16” enters the original cycle for cooling in the evaporator E of the heat pump. The temperature flow controller TC2 with the adjacent bypass pipeline section “17” and the bypass pipeline “18”, with a check valve, convert the circuit to a variant of the sequential movement of the energy carrier through the surface air cooling heat exchanger “9” and a ground heat exchanger “15” with its subsequent entry into the evaporator and bypass pipeline.

Consequently, setting of two three-way temperature regulators PT1 and PT2 with the corresponding switching of the circulating energy carrier provides both parallel and sequential or combined modes of operation of the ground heat exchanger “15” and the surface heat exchanger “9” for cooling the exhaust air.

In the design mode for the cold season, the heated mixture with the total consumption of the energy carrier after the ground heat exchanger “15” and the air cooling heat exchanger “9” enters the evaporator E for its subsequent cooling. In this mode, in the surface heat exchanger “9”, the process of a reasonable depth of cooling of the removed ventilation air is realized during the process of the regulated energy carrier circulation intensity by the temperature controller PT3 under the influence of the pulse signal of the relative humidity sensor “10”, which is installed in the duct after the heat exchanger “9” for cooling the exhaust air. Characteristically, in the condensation mode of air cooling, the efficiency of heat recovery increases to 10%. As a result, not only the energy-efficient operation of heating and ventilation systems and hot water supply is ensured, but also the accumulation of excess heat in the soil mass with an adjustable cooling depth of the removed ventilation air.

When the system is operating in the cold season with a mode that differs from the calculated one, and the outside air temperature is less than its calculated value for the heating period, the position of the three-way temperature controller TC2 with a bypass pipeline section "17" ensures the circulation of an additional part of the energy carrier through the pipeline into the ground heat exchanger "15" with a subsequent flow into the evaporator E and the bypass line "18" for further heating of the liquid in the heat exchanger „9" from cooling the exhaust air. Such a combined mode of operation of the heat extraction circuit is most appropriate for the beginning of the heating period with reduced extraction and subsequent accumulation of heat in the soil mass with its initially low temperature. When the outside air temperature rises during the heating period, using the TC2 temperature regulator, the recirculation part of the passing liquid increases and it enters the ground heat exchanger to accumulate excess heat in the soil mass.

Cooling of the supply air in the warm period of the year occurs in the same heater "8" and the pipeline circuit when it is automatically connected by pipelines "6" and "7" to the ground heat exchanger "15" through of the temperature flow controller TC4 when the outside air temperature reaches typical levels for the beginning of the inter-heating period. At the same time, in the resulting pipeline circuit with a heater "8", which performs the functions of a supply air cooler, the circulating energy carrier in the soil mass is subsequently cooled with a corresponding accumulation of excess heat without energy consumption in the HPS.

Thus, in the analyzed system, a year-round process of heat extraction from the removed ventilation air in heat exchanger "9" is realized with its subsequent vapor compression transformation for subscriber heat consumption subsystems. Moreover, the heat from the cooling of the supply air in the heater "8" is additionally recovered in the warm season by circulating the energy carrier through the ground heat exchanger "15" without energy consumption in the HPS with the corresponding accumulation of excess heat in the soil mass in the automatic process of operational regulation of the improved HPHS.

3. Analytical research

From the presented system (Fig. 1), it follows that for conventionally ideal thermal insulation of pipelines, the temperature of the energy carrier at the inlet to the surface heat exchanger for cooling the exhaust air "9" and the ground heat exchanger "15" can be taken equal to its initial temperature at the outlet after the HPS evaporator. Recommended values are within +5–-5°C, and therefore it is logical to assume, that $t_{mix.out} = t_{cal.in.} = t_{gr.in.}$. For the conditions of parallel operation of the collector or "probe" heat exchanger "15" in the process of extracting soil heat with a flow rate $G_{gr.}$ and with the heat exchanger set to removed ventilation air "9" with a flow rate $G_{cal.}$, total flow rate of the circulating energy carrier through the evaporator G_E can be determined accordingly: $G_E = G_{gr.} + G_{cal.}$. Denoted as the "α" part of the circulation of energy consumption $G_{gr.}$ through a ground heat exchanger "15" relative to total evaporator flow G_E, i.e. $G_{gr.} = \alpha \cdot G_E$, then the rest of the circulated energy carrier passes through the surface heat exchanger "9", in this connection $G_{cal.} = (1 - \alpha) \cdot G_E$.

From the structural and functional mechanism of the system, it follows that the weighted average temperature of the mixture of the heated energy carrier after the ground "15" and surface "9" heat exchangers at the inlet to the HPS evaporator is determined by the ratio of their flows and temperatures. In this case, the heat flow of the energy carrier in the pipeline after the ground heat exchanger "15" at the inlet to the HPS evaporator is determined in the following form:

$$Q_{gr.} = \alpha \cdot G_E \cdot c_{gr.} \cdot t_{gr.,out}, \tag{1}$$

where $c_{gr.}$ – specific heat capacity of the circulating fluid at the outlet from the soil massif, J/(kg·°C).

The temperature of the energy carrier after surface heat exchanger "9" with a flow rate $G_{cal.}$, in which the heat of the removed ventilation air is perceived with a flow rate G_{air}, is determined based on the heat balance of the initial temperature of the liquid at the inlet $t_{cal.in.}$ with its subsequent heating in this heat exchanger. Therefore:

$$t_{cal.out.} = t_{cal.in.} + \frac{G_{air.} c_{air.} \left(t_{air.in.} - t_{air.out.}\right)}{G_{cal.} \cdot c_{wat.}}, \tag{2}$$

where c_{air} and $c_{wat.}$ are the corresponding specific heat capacities of air and circulating liquid, J/(kg·°C).

It is natural that the second term on the right side of equation (2) reflects the subsequent heating of the energy carrier in the surface heat exchanger "9", which is determined on the basis of the corresponding heat balance by the ratio of the cooling heat of the removed ventilation air to the liquid flow rate circulating through it.

Therefore, the heat flux $Q_{cal.}$ of the energy carrier after the heat exchanger "9", which perceives the heat of the removed air, is represented in the following form:

$$Q_{cal.} = (1-\alpha) \cdot G_E \cdot c_{cal.} \left\{ \left[t_{cal.in.} + \frac{G_{air.} c_{air.} \left(t_{air.in.} - t_{in.out.}\right)}{G_{cal.} \cdot c_{wat.}} \right] - t_{mix.out.} \right\}. \tag{3}$$

Consequently, the heat flux taken by the ground heat exchanger "15" and the surface heat exchanger for cooling air "9", which is perceived by the circulating energy carrier in the evaporator after appropriate transformations, is represented in the following form:

$$\begin{gathered} Q_E = G_E \cdot c_{mix.} \times \\ \times \left\{ \alpha \cdot c_{gr.} \cdot t_{gr.out.} + (1-\alpha) \left[t_{cal.in.} + \frac{G_{air.} c_{air.} \left(t_{air.in} - t_{air.out}\right)}{G_{cal.} \cdot c_{wat.}} \right] - t_{mix.out.} \right\}, \end{gathered} \tag{4}$$

where $c_{mix.}$ is the specific heat capacity of the circulating liquid after the evaporator, J/(kg·°C).

From the presented scheme (Fig. 1), it follows that the estimated capacity of the HPS condenser is determined by the total need for the required heat for heating and ventilation processes and hot water supply, and therefore the dependence for its determination takes the form:

$$Q_{cond.} = G_{h.v.} \cdot c_{h.v.} \cdot \left(t'_f - t'_r\right) + G_{h.v.} \cdot c_{h.s.} \cdot \left(t_{h.w.} - t'_{c.w.}\right), \tag{5}$$

where t'_f and t'_r are the variable values of heat carrier temperatures in the feed and return lines during operational regulation of the heating system, °C; $t_{h.w.}$ is the rated temperature of the energy carrier in the hot water supply system, °C; $t'_{c.w.}$ is the variable throughout the year temperature of cold water entering the building, °C; $G_{h.v.}$ and $G_{h.s.}$ is the energy flow rates, respectively, in the heating and ventilation system and the hot water supply system, kg/sec.; $c_{h.v.}$ and $c_{h.s.}$ are specific heat capacities of the corresponding energy carriers, J/(kg·°C), whose values in the analyzed temperature range can be conventionally assumed to be equivalent.

Denoting β as part of the required heat flux $Q_{h.v.}$ for heating and ventilation processes in relation to the total value of the generated heat $Q_{cond.}$ in the condenser $Q_{h.v.} = \beta \cdot Q_{cond.}$.

As a result, for hot water supply it is as follows: $Q_{h.w.} = (1 - \beta) \cdot Q_{cond.}$.

For the recommended two-pipe low-temperature radiant panel heating systems with their characteristic increased thermal-hydraulic stability, the dependence of the temperature difference between hot and cooled energy carriers in the process of operational regulation (Belenkij, 1963) is represented as the following dependence:

$$t'_f - t'_r = \left(t_f - t_r\right) \cdot \left(\frac{t_{in} - t'_{out}}{t_{in} - t_{out}}\right)^{0,5}, \tag{6}$$

where: t_f and t_r are rated values of heat carrier temperatures in the feed and return lines of the heating system, °C; t_{out} and t'_{out} are the rated and current values of outside air temperatures, °C; t_{in} is the average air temperature inside the building, °C.

Based on the results of a field study (Vysockaja, 2015), changes in the temperature of cold water in a flow-through mode at the inlet to residential buildings in the southern region of Ukraine for central water supply systems from the river Dniester, the authors established the dependence of seasonal changes in the average temperature of cold water $t'_{c.w.}$ at the entrance to the building, which at the corresponding outdoor temperature t'_{out} can be determined according to dependency:

$$t'_{c.w.} = \left(9 + \frac{t'_{out}}{5} \cdot \Delta\right), \text{°C}, \tag{7}$$

where Δ is the coefficient of change in the temperature of cold water, which in the ranges of negative and positive outside air temperatures equals 1.0 on 2.0.

Thus, based on equation (5), taking into account dependencies (6) and (7), the capacity of the HPS condenser, determined by the total heat demand for heating and ventilation processes and hot water supply, is represented by the following generalized relationship:

$$Q_{cond.} = G_C \cdot c_{h.v.} \cdot \left[\beta\left(t_f - t_r\right) \cdot \left(\frac{t_{in} - t'_{out}}{t_{in} - t_{out}} \right)^{0,5} + (1-\beta) \cdot \left(t_{h.w.} - \left(9 + \frac{t'_{out}}{5} \cdot \Delta \right) \right) \right]. \quad (8)$$

It is known that the energy efficiency of HPS is determined by the actual conversion coefficient in the form of the ratio of energy flows (Bezrodnij, 2013) according to the dependence:

$$\varphi = \frac{Q_{cond.}}{W_{comp.}}, \quad (9)$$

where $W_{comp.}$ is the thermal equivalent of the external drive power in the operation of the heat pump compressor, which is determined (Gorshkov, 2004; Martynovskij, 1977; Petrash, 2014) according to the conditions of the HPS energy balance in the form:

$$W_{comp.} = Q_{cond.} - Q_E. \quad (10)$$

From correlation (9), the actual conversion factor in the analyzed building heat supply system is determined taking into account the relationship (10) and generalized dependencies for multifactor heat fluxes of the evaporator (4) and condenser (8), based on the integrated energy technological potential of the soil and ventilation air.

From the analysis of the circuit (Fig. 1) it is evident that interest is not only in the influence of possible redistribution of circulating energy sources in the circuits in the selection of heat exchanger “9”, the cooling ventilating air and soil heat exchanger “15”, but also taking into account the structural ratio of subscriber heat consumption for heating and ventilation processes and hot water supply in the annual interval of operational regulation with the identification of rational conditions for the seasonal accumulation of excess heat in the soil mass for its subsequent use in the cold season.

Estimation of the dependence of the energy efficiency of the analyzed system according to equations (4), (8) and (9) on the initial parameters and operating conditions, the following values were used as the initial values: $\bar{G}$ = 1-2 (Dokumentacija; www.viessmann.ru; www.domteplo.ru); α = 0.3-0.7; $t_{gr.,out.}$ = 7°C; $t_{mix.,in.}$ = 2°C; β = 0.7-0.9; (t_f – t_r) = (50 – 40)°C; $t'_{out.} = (-18\text{-}+8)$°C; $t_{h.w.}$ = 50°C. Wherein, the temperature of the energy carrier $t_{cal.,out.}$ at the outlet of the surface heat exchanger “9” in the process of cooling the removed ventilation air, determined by its initial temperature at the inlet $t_{cal.,in.}$ with subsequent heating, depending on the ratio of the heating and heated with subsequent heating, depending on the ratio of the heating and heated medium according to the heat balance, is represented as the second term in equation (2) and, accordingly, appearing in (3) according to the heat balance, is represented as the second term in equation (2) and, accordingly, appearing in (3). Based on preliminary calculations, its values were considered in the range $t_{cal.,out.}$ = = 6-14°C.

In Figure 2 graphical dependence of the conversion coefficient is shown when the outside air temperature changes during the heating period with a characteristic redistribution of the energy carrier between the ground and surface heat exchangers for cooling the exhaust air at characteristic values of the heating and ventilation load (β = 0.8-0.9) in the general structure of subscriber heat consumption.

The graphs show that with a decrease in the ratio of circulating energy carriers between the ground heat exchanger and the heat exchanger for cooling the removed ventilation air, the efficiency of the conversion of energy flows increases significantly. From the presented graph one obtains that it is possible to increase the efficiency of energy conversion in heat supply systems of buildings, which are distinguished by increased values of heating and ventilation flows in the structure of total heat consumption, indirectly reflecting the feasibility of maximum use of the available energy potential of cooling the removed ventilation air.

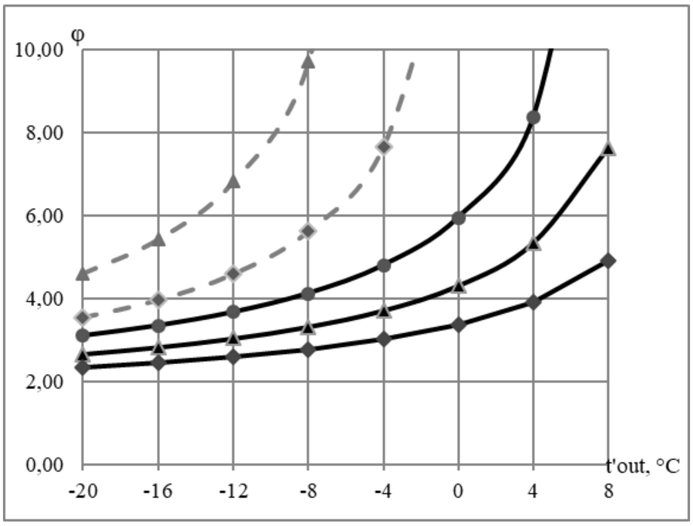

Fig. 2. Dependence of the conversion factor on the outside air temperature with different ratios of energy carriers in the ground and surface heat exchangers with a characteristic component of heat for heating and ventilation processes (*own study*):
——— – β = 0.8; – – – – β = 0.9;
● – α = 0.3; ▲ – α = 0.5; ◆ – α = 0.5

Conclusions

1. Based on the results of an analytical study of the proposed system for heating and cooling buildings, a multifactorial dependence of the actual conversion

coefficient was established to assess the efficiency of extracted heat transformation, allowing to simulate the multifactorial influence of initial parameters and operating conditions on the possibility of highly efficient using integrated energy in cold, warm and transitional periods of the year.

2. Rational operating conditions of an improved system based on integrated soil energy and ventilation air for heat and cold supply of buildings with heat accumulation in the soil mass during warm and transitional periods of the year for its subsequent use with increased efficiency in heating, ventilation and hot water supply processes have been determined.
3. A significant increase in the energy efficiency of the system is ensured with an increase in the outside air temperature, which is primarily due to a corresponding decrease in the temperature of the energy carrier and the consumed energy for heating and ventilation processes under conditions of rational heating of the energy carrier in the heat exchanger for cooling the exhaust air to a temperature of 10°C. The foregone indirectly predetermines the advisability of reducing the depth of expensive wells or the number of probe heat exchangers.

Bibliography

Belenkij, E.A. (1963) *Racional'nye sistemy vodjanogo otoplenija*. Leningrad, Gosstrojizdat.

Bezrodnij, M.K., Puhovij, I.I. & Kutra, D.S. (2013) *Teplovinasosi ta ïh vikoristannja*. Kyiv, NTUU «KPI».

Bezrodnij, M.K., Pritula, N.O. & Oslovs'kij S.O. (2018) *Termodinamichna efektivnist' teplonasosnoï shemi opalennja z vikoristannjam teploti gruntu i stichnih vod*. «KPI»: Energetika ta novi energogenerujuchi tehnologiï. Kyiv, 7-14.

Dokumentacija dlja planirovanija i proektirovanija teplovyhnasosov. Buderus. Source: http://www.adeptamasa.com/doc_proect/Logatherm%20WPS_draft.pdf.

DSTU B V.2.5-44:2010 Inzhenerne obladnannja budinkiv i sporud. Proektuvannja sistem opalennja budivel' z teplovi minasosami [Tekst]. Chinnijvid 2010-09-01. Kyiv, Minregionbud Ukraïni.

Gorshkov, V.G. (2004) *Teplovyenasosy*. *Analiticheskij obzor*. Spravochnik Promyshlennogo Oborudovanija, 2, 47-80.

Heinrich, H., Shofmann, P. & Zotti, A. (2014) *Warme Pumpen*. Wien.

Lund, J. (2004) *Geothermal (ground-source) heat pumps a world overview*. GHC BULLETIN, September, 1-10.

Martynovskij, V.S. (1977) *Cikly, shemy i harakteristiki termotransformatorov*. M., Jenergija.

Matsevite, Y., Macevityj, J.M., Chirkin N.B., Bogdanovich, L.S. &. Klepanda, A.S. (2007) *O racional'nom ispol'zovanii teplonasosnyh tehnologij v jekonomike Ukrainy*. Jenergosberezhenie. Jenergetika. Jenergoaudit, 3, 20-31.

Patent 108184 Ukraïni, MPK (2015.01), F24D 17/02. Sistema teploholodopostachannja na osnovi geliogruntovoï energiï ta utilizovanoï teploti vitjazhnogo ventiljacijnogo povitrja i stichnih vod [Tekst]. V.D. Petrash, O.A. Polomannij, M.V. Vysoc'ka. Odes'ka derzhavna akademija budivnictva ta arhitekturi. opubl. 25.03.2015, Bjul. 6/2015.

Petrash, V.D. (2014) *Teplonasosnye sistemy teplosnabzhenija*. Odessa: «VMV».

Vasil'ev, G.P. (2006) *Teplohladosnabzhenie zdanij i sooruzhenij s ispol'zovaniem nizkopotencial'noj teplovoj jenergii poverhnostnyh sloev Zemli*. M., Granica.

Vasil'ev, G.P., Gornov, V.F. & Kolisova, M.V. (2014) *Issledovanie ocenki jeffektivnosti kombinirovannogo ispol'zovanija tepla grunta i atmosfernogo vozduha v teplonasosnyh sistemah teplohladosnabzhenija*. Jenergobezopasnost' i Jenergosberezhenie, 1, 20-24.

Vasil'ev, G.P., Gornov, V.F., Konstantinov, P.I., Kolisova, M.V. & Korneva, I.A. (2017) *Analiz izmenenija temperatury grunta na osnove mnogoletnih izmerenij*. Inzhenerno-stroitel'nyj Zhurnal, 4(72), 62-72.

Vysockaja, M.V. (2015) *Holodnaja voda kak nizkotemperaturnyj istochnik dlja teplonasosnyh sistem teplohladosnabzhenija zdanij*. Jenergojeffektivnost' v stroitel'stve i arhitekture. Kyiv, KNUBA, 7, 41-46.

www.viessmann.ru/ru/products/Heat_pumps/

www.domteplo.ru/GHP_Vaillant

Zabarnij, G.M., Masljukova, Z.V., Didkivs'ka, G.G. & Primak A.I. (2012) *Ocinka energetichnogo potencialu sistem teplopostachannja na osnovi teplovih nasosiv, sho vikoristovujut' niz'kopotencijnu energiju gruntu, povitrja ta pidzemnih vod pripoverhnevih vodonosnih gorizontiv*. Vidnovljuvana Energetika, 4, 70-79.