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DEVELOPMENT OF ENERGY-SAVING TECHNOLOGY FOR MAINTAINING THE FUNCTIONING OF HEAT PUMP POWER SUPPLY

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Розроблено інтегровану систему підтримки функціонування теплонасосного енергопостачання на основі прогнозування зміни температури місцевої води. Зміна витрати пари холодагента, числа обертів електродвигуна компресора відбувається при вимірюванні температури холодагента на виході із конденсатора, тиску випаровування, тиску конденсації та частоти напруги.

Виконано комплексне математичне моделювання теплонасосної системи, що базується на інтегрованій системі підтримки розряду грунту на рівні 10–8 °C. Визначено витрату холодагента, потужність електродвигуна компресора, напругу, частоту напруги, число обертів електродвигуна компресора, коефіцієнт продуктивності теплонасосної системи для встановлених рівнів функціонування. Встановлено параметри конвективного теплообміну в конденсаторі, постійні часу та коефіцієнти математичних моделей динаміки зміни температури місцевої води, витрати пари холодагента, числа обертів електродвигуна компресора.

Здобуто функціональну оцінку зміни температури місцевої води в діапазоні 35–55 °C впродовж опалювального сезону, витрати пари холодагента, числа обертів електродвигуна компресора. Визначення підсумкової функціональної інформації надає можливість приймати наступні випереджуючі рішення: на підтримку зміни тиску випаровування щодо зміни витрати пари холодагента для цифрового управління; на підтримку зміни тиску випаровування щодо зміни витрати пари холодагента та на зміну частоти напруги щодо зміни числа обертів електродвигуна компресора для частотного управління. Тому, запропоновано прогнозування зміни температури місцевої води на основі вимірювання температури холодагента на виході із конденсатора. Саме ця оцінка у співвідношенні з вимірюваним тиском випаровування, входить до складу аналітичних визначень витрати холодагента та числа обертів електродвигуна компресора. Здобуття такої оцінки та вимірювання частоти напруги надає можливість упереджено впливати на узгодження функціонування зовнішнього та внутрішнього контурів теплонасосної системи як при цифровому, так і частотному управлінні

Ключові слова: теплонасосна система, частотне управління, цифрове управління, тиск випаровування, тиск конденсації

1. Introduction

Under conditions for saving natural fuel and decreasing harmful emissions into the atmosphere, heat pump power supply with the use of renewable power sources is gaining further development [1–3]. Thus, for example, the heat pump, using fermented must as a low-potential energy source, is recommended in order to maintain the operation of the biogas plant as a part of a cogeneration power system. Due to additional biogas production, the proposed technology provides an opportunity to increase marketability of a biogas plant and decrease the cost value of production of electricity and heat in the range of 20–30 % [3].

The outside air as a low-potential power source is available in heat pump power supply, but the change of air temperature in a wide range makes it difficult to maintain functioning of heat pump systems [1].

A heating system of the soil–water type requires the construction of special soil heat exchangers for heating the brine – 30 % ethylene glycol solution that is fed to the evaporator of a heat pump [2]. A vertical heat exchanger occupies a smaller area than a horizontal one but requires additional capital investments for drilling wells. It is possible to ensure heat extraction within 30–100 W per one meter of the length of a vertical heat exchanger depending on a soil type at the depth of 40–150 m, where the soil temperature is 8–10 °C. The design of a vertical heat exchanger of the tube-in-tube type is most common. Brine, after transferring heat to a refrigerant in the evaporator of a heat pump, is fed by the circulation pump to the soil heat exchanger on one line, the other line provides a lift of the brine to the evaporator of a heat pump. Improvement of the tools for using the soil heat discharge is based on the development of new structures of vertical heat exchangers for the intensification of the heat exchange process. Thus, for example, in paper [4], the optimal structural parameters, such as length and diameter, were established based on mathematical modeling of a specially developed spiral heat exchanger. In paper [5], it was proposed to use the foundation for location of spiral heat exchangers with the purpose of optimization of a heat exchange surface. Heat pumps of the soil-water type are most common, but their use should not disturb the natural soil relaxation in the period when the heating does not function. Thus, an integrated system of maintaining soil heat discharge within 8–10 °C during the heating season as part of a dynamic system: exchanger-evaporator of the heat pump was proposed for this purpose. Based on prediction of a change in soil temperature, changes in brine flow rate were ensured based on the change of rotation rate of the motor of the circulation pump at measuring the brine temperature at the outlet of the evaporator. The exact initial and final term of soil heat discharge was established [6].

Under condition of the change of parameters of renewable power sources, a relevant task regarding further development of technologies of heat pump power supply is to maintain efficiency of heat pump systems based on coordination of production and power flow rate in terms of energy saving.

It is known that frequency control is based on changing the rotation rate of a compressor due to a change in supply voltage frequency but measures the change in pressure of a refrigerant in the evaporator. Digital regulation of the spiral compressor is based on modification of the period of compression of refrigerant vapor based on a specially developed electromagnetic valve, but also assesses a change in evaporation pressure. Maintaining the productivity of the heat pump system requires ensuring that changes in pressure evaporation in the evaporator should correspond to a change in pressure of refrigerant condensation in the condenser, which is difficult to execute based on measuring a change in refrigerant pressure in the evaporator.

For this purpose, it is necessary to predict a change in the temperature of local water when measuring the refrigerant temperature at the outlet of the condenser. Making advancing decisions on a change in refrigerant flow rate and the number of rotations of the motor of a compressor when measuring evaporation pressure, condensation pressure and voltage frequency will ensure a compliance of a change in evaporation pressure in the evaporator with a change in refrigerant condensation pressure in the condenser.

2. Literature review and problem statement

The known methods of improvement of heat pump systems – economic, exergy, thermal economy – allow determining the optimal operating conditions at the static level, which complicates the coordination of production and energy consumption under operation conditions. Thus, in paper [7], a comparative analysis of the energy and exergy methods in terms of optimization of performance coefficient of the heat pump system was made. The preference was given to the exergy method based on assessment of the influence of refrigerant flow rate on the power of the compressor. In paper [8], the authors performed mathematical modeling of components of the heat pump system regarding further modeling at the level of the heat pump system, but at the static level. Moreover, the dynamic approach to mathematical modeling of the heat pump system was considered in this paper, but only for the purpose of optimization of designing heat pump systems and stabilization of control systems. Paper [9] focuses on the dynamic approach to optimization of heat pump systems. The impact of intensification of heat exchange on the performance of the heat pump system, but with the use of mathematical models with concentrated parameters, was established. Assessment of a change in parameters only in time does not provide the possibility of predicting changes of parameters of the technological process.

Due to complexity of the analysis of heat pump systems at the static level regarding the performance optimization, the authors propose the use of combined systems of heat pump power supply with connection, for example, of photo cells [10]. The influence of a change of solar radiation on maintaining the power of a compressor regarding a decrease in the term of using an external electric network, also at the static level, was described in paper [10]. The technological scheme of heat pump power supply regarding the use of a heat pump heater from a solar engine was proposed in paper [11]. The possibility to raise the suction pressure of a compressor and increase the efficiency of operation of the heat pump system with the use of energy and exergy methods, also at the static level, was determined. In addition to the static models, the models [12] that set the goal of control of heat pump systems with the use of intelligent networks are also known. But the means, proposed in this work, require measurement of process parameters. This approach cannot be used for making advancing decisions to maintain the operation of heat pump systems.

According to the technological scheme, it is possible to separate two circuits of heat pump power supply: external and internal. The external circuit provides extraction of heat from a low-potential power source regarding evaporation of the refrigerant in the evaporator of the heat pump. The internal circuit is made up from the evaporator, the compressor and the condenser. It is the evaporator and the condenser, located in the internal circuit, which are connecting elements of the external and internal circuits. A change in vapor flow rate that is fed to suction in the compressor of the heat pump depends both on a change in parameters of a low-potential power source and changes in flow rate. There are various means to maintain the performance of heat pump systems in terms of the impact on the compressor operation. Thus, the cyclic mode applies the on-off principle, which requires additional electricity consumption to start the compressor. Frequency control is based on changing compressor rotation rate due to a change in supply voltage frequency regarding changes in performance but measures only a change in refrigerant pressure in the evaporator. Digital control of the spiral compressor is based on the modification in the period of time of refrigerant vapor compression based of a specially developed electromagnetic valve in respect to the axial alignment of spirals, but it also assesses a change in evaporation pressure. Maintaining the performance of the heat pump system requires ensuring the compliance of a change in evaporation pressure in the evaporator with a change in refrigerant condensation pressure in the condenser of the heat pump. With the view to energy saving, maintaining of functioning of heat pump power supply should be based on prediction of a change in local water temperature when measuring the temperature the refrigerant at the outlet of the condenser. It is necessary to make advancing decisions regarding a change in refrigerant flow rate and rotation rate of the compressor motor when measuring evaporation pressure, condensation pressure and voltage frequency when it comes to both frequency and digital control. This substantiates the need for research in this direction.

3. The aim and objectives of the study

The aim of present research is to develop the energy-saving technology of maintaining functioning of heat pump power supply as a component of the technological system.

The set goal will be accomplished if the following tasks are solved:

– substantiation of the necessity of analytical estimates of a change in the local water temperature, refrigerant vapor flow rate. the number of rotations of the compressor motor regarding making advancing decisions on a change in performance of the heat pump system;

– measuring the refrigerant temperature at the outlet of the condenser, evaporation pressure, condensation pressure, and voltage frequency;

– development of the structural scheme and integrated mathematical and logical modeling as for obtaining the reference and functional estimation of a change in the local water temperature, refrigerant vapor flow rate, the number of rotations of the compressor motor;

– to offer an integrated system of maintaining functioning of the heat pump power supply at the decision-making level.

4. Materials and methods of the study regarding maintaining the functioning of heat pump power supply

4. 1. Mathematical substantiation of the architecture of the technological system

Based on the methodological and mathematical substantiation of the architecture of technological systems [13, 14], the architecture of the technological system of functioning of heat pump power supply was proposed. Its basis is the integrated dynamic subsystem, which includes the following dynamic systems: the soil heat exchanger – evaporator, the evaporator – compressor, the compressor – condenser. Other units that are the parts of the technological system are the units of charge, discharge and the unit of functional evaluation of efficiency coefficient, which are in the coordinated interaction with the dynamic subsystem (Fig. 1).

Using formula (1), mathematical substantiation of the architecture of the technological system of heat pump power supply functioning was described:

$$TSFHP(\tau)=\left\{\begin{bmatrix}\begin{bmatrix} IDS(\tau)(PIDS(\tau)\left\langle\begin{matrix}x_0(\tau),x_1(\tau),x_2(\tau),\\ f(\tau),K(\tau),y(\tau,z),\\ d(\tau)\end{matrix}\right\rangle, \\ Z(\tau),PIDS(\tau)),R(\tau),(PB_i(\tau)\end{bmatrix}\\ \left\langle x_1(\tau),f_i(\tau),K_i(\tau),y_i(\tau)\right\rangle),\end{bmatrix}\right\} \quad (1)$$

where $TSFHP(\tau)$ is the technological system of functioning of heat pump power supply; $IDS(\tau)$ is the integrated dynamic subsystem (the soil heat exchanger – evaporator; evaporator – compressor; compressor – condenser); $PIDS(\tau)$, $PB(\tau)$ are the properties of the elements of the integrated dynamic subsystem, and of units of the technological subsystem, respectively; τ is the time, s; z is the length coordinate, m; $x(\tau)$ is the impacts; $f(\tau)$ is the diagnosed parameters; $K(\tau)$ is the coefficients of mathematical description; $y(\tau, z)$ is the output parameters; $d(\tau)$ is the dynamic parameters; $Z(\tau)$, $R(\tau)$ are the logical relations in $IDS(\tau)$, $TSFHP(\tau)$, respectively.

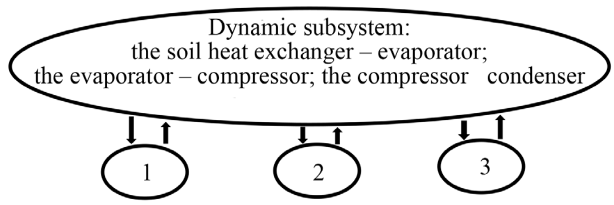

Fig. 1. The architecture of the technological system of functioning of heat pump power supply: 1 — charge unit; 2 — unit of functional estimation of performance coefficient; 3 — discharge unit

Indices: i is the number of elements of the technological system; 0, 1, 2 are the initial stationary mode, external and internal character of impacts.

4. 2. Mathematical substantiation of maintaining the functioning of heat pump power supply

The mathematical description (2) of maintaining the functioning of heat pump power supply, based on mathematical substantiation of the architecture of technological systems, methodology of mathematical description of dynamics of power systems, the method of cause-effect relations graph was proposed [13, 14]:

$$SFHP(\tau)=\left\{\begin{bmatrix}\begin{bmatrix}(IDS(\tau)(PIDS(\tau),MMIDS(\tau,z),\\ CIDS(\tau),LCIDS(\tau)\\ \left\langle\begin{matrix}x_0(\tau),x_1(\tau),x_2(\tau),f(\tau),K(\tau),\\ y(\tau,z),d(\tau),FIIDS(\tau)\end{matrix}\right\rangle,\\ LMDIDS(\tau),MDIDS(\tau),NCF(\tau),\\ SIDS(\tau),LSIDS(\tau)\\ \left\langle f(\tau),K(\tau),y(\tau,z),d(\tau),FIIDS(\tau)\right\rangle\\ PIDS(\tau))),R(\tau),\\ (PB_i(\tau)\left\langle\begin{matrix}x_1(\tau),f_i(\tau),\\ K_i(\tau),y_i(\tau)\end{matrix}\right\rangle)CNCF(\tau)),\end{bmatrix}\end{bmatrix}\right\}, \quad (2)$$

where $SFHP(\tau)$ is the support of the functioning of heat pump power supply;

$IDS(\tau)$ is the integrated dynamic subsystem that includes the dynamic system: the soil heat exchanger – evaporator, the evaporator – compressor, the compressor – condenser;

$PIDS(\tau)$, $PB(\tau)$ are the properties of the elements of the integrated dynamic subsystem, of units of the technological system, respectively;

$MMIDS(\tau, z)$ is the mathematical modeling of soil temperature dynamics, refrigerant vapor flow consumption, the number of rotations of the compressor motor, the local water temperature;

MIIDS(τ) is the boundary permissible change in soil temperature, refrigerant vapor flow rate, the number of rotation of the compressor motor, the local water temperature;

CIDS(τ), *MDIDS*(τ), *SIDS*(τ) are the control of working capacity, decision making, state identification in the dynamic subsystem, respectively;

LCIDS(τ), *LMDIDS*(τ), *LSIDS*(τ) are the logical relations in *CIDS*(τ), *MDIDS*(τ), *SIDS*(τ), respectively;

FIIDS(τ) is the functional summarizing information regarding decision making in the dynamic subsystem;

NCF(τ), *CNCF*(τ) are the new conditions of functioning, confirmation of new conditions of functioning from the units of the technological system;

R(τ) is the logical relations between the dynamic subsystem and the units of discharge, charge, functional estimation of performance coefficient, which are included in the technological system of functioning of heat pump power supply.

x(τ) – impacts;

f(τ) – diagnosed parameters;

K(τ) – coefficients of mathematical description;

y(τ,*z*) – output parameters;

d(τ) – dynamic parameters;

z – length coordinate, m;

τ – time, s. Indices: *i* is the number of elements of *SFHP*(τ); 0, 1, 2 are the initial mode, external, internal character of impacts.

Mathematical descriptions of (1, 2) allow maintaining functioning of the heat pump power supply using the following actions:

– control of working capacity (*CIDS*(τ)) of the dynamic subsystem: the soil heat exchanger – evaporator, evaporator – compressor, compressor – condenser based on mathematical (*MMIDS*(τ,*z*)) and logical (*LCIDS*(τ)) modeling regarding obtaining the reference (*MIIDS*(τ)) estimation of a change in soil temperature, refrigerant flow rate, the number of rotations of the electric motor of the compressor, local water temperature;

– control of working capacity (*CIDS*(τ)) of the dynamic subsystem: the soil heat exchanger – evaporator, evaporator – compressor, compressor – condenser based on mathematical (*MMIDS*(τ,*z*)) and logical (*LCIDS*(τ)) modeling regarding obtaining the functional (*FIIDS* (τ)) estimation of a change in soil temperature, refrigerant flow rate, the number of rotations of the electric motor of the compressor, local water temperature;

– decision making (*MDIDS* (τ)) with the use of functional information (*FIIDS* (τ)), obtained based on logical modeling of obtained (*LMDIDS*(τ));

– decision making on a change of voltage frequency with the use of the functional estimation of a change in soil temperature, refrigerant flow rate, the number of rotations of the electric motor of the compressor, local water temperature (*FIIDS*(τ));

– identification (*SIDS*(τ)) of new conditions of functioning heat pump power supply (*NCF*(τ)) based on logical modeling (*LSIDS* (τ)) and confirmation of new conditions of functioning based on (*R*(τ)) from the units of the technological system.

4. 3. Mathematical modeling of dynamics of change in the local water temperature

According to formulas (1), (2), it was proposed to predict changes in local water temperature when measuring the refrigerant temperature at the outlet of the condenser of the heat pump. Transfer function for the channel: "local water temperature – refrigerant flow rate" was obtained as a result of solving the system of nonlinear differential equations. A change in the local water temperature both in time and along the condenser length coordinate at a change in refrigerant flow rate is represented as follows:

$$W_{t_w - G_{r1}} = \frac{K_r \varepsilon \left(1 - L_r^*\right)}{(T_w S + 1)\beta - 1}\left(1 - e^{-\gamma\xi}\right), \qquad (3)$$

where

$$K_r = \frac{m\left(\theta_0 - \sigma_0\right)}{G_{r0}}; \quad \varepsilon = \frac{\alpha_{r0} h_{r0}}{\alpha_{w0} h_{w0}};$$

$$L_r^* = \frac{1}{L_r + 1}; \quad L_r = \frac{G_r C_r}{\alpha_{r0} h_{r0}}; \quad \varepsilon^* = \varepsilon(1 - L_r^*);$$

$$\gamma = \frac{(T_w S + 1)\beta - 1}{\beta}; \quad \xi = \frac{z}{L_w}; \quad T_w = \frac{g_w C_w}{\alpha_{w0} h_{w0}};$$

$$\beta = T_m S + \varepsilon^* + 1; \quad T_m = \frac{g_m C_m}{\alpha_{w0} h_{w0}},$$

where *a* is the heat transfer coefficient, kW/(m^2·K); *C* is the specific thermal capacity, kJ/(kg·K); *G* is the substance flow rate, kg/s; *g* is the specific weight of substance, kg/m; *h* is the specific surface, m^2/m; t_w, θ, σ are the temperatures of local water, distribution wall, refrigerant, K; *z* is the coordinate of the condenser length, m; T_w, T_m are the time constants, that characterize thermal accumulating capacity of local water, metal, s; *m* is the indicator of dependence of heat transfer coefficient on flow rate; ι is the time, s; *S* is the Laplace transform parameter; *S*=ω*j*; ω is the frequency, 1/s.

Indices: 0 is the original stationary mode; 1 is the inlet to the condenser; w is the internal flow – local water; m is the metal wall; r is the external flow – refrigerant.

Transfer function for the channel: "local water temperature – refrigerant vapor flow rate" was obtained based on the solution of the system of nonlinear differential equations using the Laplace transform. A system of differential equations includes the equation of state as an assessment of a physical model of the dynamic system: compressor – condenser of the heat pump. The system of differential equations also includes the equation of energy of transferring and receiving media – the refrigerant and local water, respectively, and the equation of thermal balance for the wall of the condenser.

The equation of energy of the receiving medium was developed with representation of a change on the local water temperature both in time and along the spatial coordinate, which coincides with the direction of the flow of medium motion. The equation of energy of transferring medium includes coefficient K_r, which assesses a change in the temperature of the refrigerant at the outlet of the condenser of the heat pump that is measured.

4. 4. Mathematical modeling of dynamics of change in the refrigerant vapor flow rate

According to formulas (1), (2), the estimation of a change in refrigerant flow rate as for feed to compressor suction when measuring evaporation pressure at the outlet of the evaporator and condensing pressure at the outlet of the condenser of the heat pump was proposed. For this purpose, the system of differential equations, which includes the equation of refrigerant energy, local water, and the equation of thermal balance for the wall of the condenser was supplemented by equation of refrigerant continuity. The result of solving the system of differential equations using the Laplace transform is the transfer function for the channel: "flow rate of refrigerant – refrigerant pressure", which assesses a change in flow rate of refrigerant at a change of evaporation pressure and condensation pressure:

$$W_{G_r-p_1}=\frac{\chi_p S}{\gamma}\left(1-e^{-\gamma_1\xi}\right), \quad (4)$$

where

$$\chi_p=-f_s\frac{\partial\rho}{\partial p};\ \ \gamma=\frac{(T_w S+1)\beta-1}{L_w\beta};$$

$$T_w=\frac{g_w C_w}{\alpha_{w0}h_{w0}};\ \ L_w=\frac{G_w C_w}{\alpha_{w0}h_{w0}};$$

$$\beta=T_m S+\varepsilon^*+1;\ \ T_m=\frac{g_m C_m}{\alpha_{w0}h_{w0}};$$

$$\varepsilon^*=\varepsilon(1-L_r^*);\ \ \varepsilon=\frac{\alpha_{r0}h_{r0}}{\alpha_{w0}h_{w0}};$$

$$L_r^*=\frac{1}{L_r+1};\ \ L_r=\frac{G_r C_r}{\alpha_{r0}h_{r0}};\ \ \xi=\frac{z}{L_w},$$

where a is the heat transfer coefficient, kW/(m^2·K); p is the refrigerant pressure, MPa; f_s is the intersection for refrigerant passing, m^2; C is the specific thermal capacity, kJ(kg·K); G is the substance flow rate, kg/s; ρ is the refrigerant density, kg/m^3; g is the specific weight of substance, kg/m; h is the specific surface, m^2/m; z is the spatial coordinate of the condenser length, m; T_w, T_m are the time constants that characterize thermal accumulating capacity of local water, metal, s; S is the parameter of Laplace transform; $S=\omega j$; ω is the frequency, 1/s.

Indices: 0 is the original stationary mode; 1 is the inlet to the condenser; w is the internal flow – local water; m is the metal wall; r is the external flow – refrigerant.

4. 5. Mathematical modeling of dynamics of change in the number of rotations of compressor motor

According to formulas (1), (2), estimation of a change in the number of rotations of the compressor motor of the heat pump was proposed. Parameters that are measured are the following: evaporation pressure at the outlet of the evaporator, condensation pressure at the outlet of the condenser and voltage frequency. The transfer function for the channel: "the number of rotations of the motor of the heat pump compressor – voltage frequency" takes the following form:

$$W_{n-f_1}=\frac{K_f\chi_p S}{\gamma}\left(1-e^{-\gamma_1\xi}\right), \quad (5)$$

$$K_f=\frac{60f(1-s)}{p_n};\ \ \chi_p=-f_s\frac{\partial\rho}{\partial p};\ \ \gamma=\frac{(T_w S+1)\beta\text{-}1}{L_w\beta};$$

$$T_w=\frac{g_w C_w}{\alpha_{w0}h_{w0}};\ \ L_w=\frac{G_w C_w}{\alpha_{w0}h_{w0}};$$

$$\beta=T_m S+\varepsilon^*+1;\ \ T_m=\frac{g_m C_m}{\alpha_{w0}h_{w0}};$$

$$\varepsilon^*=\varepsilon(1-L_r^*);\ \ \varepsilon=\frac{\alpha_{r0}h_{r0}}{\alpha_{w0}h_{w0}};$$

$$L_r^*=\frac{1}{L_r+1};\ \ L_r=\frac{G_r C_r}{\alpha_{r0}h_{r0}};$$

$$\gamma_1=\frac{(T_w S+1)\beta-1}{\beta};\ \ \xi=\frac{z}{L_w},$$

where a is the heat transfer coefficient, kW/(m^2·K); p is the refrigerant pressure, MPa; f_s is the intersection for refrigerant passing, m^2; f is the voltage frequency, Hz; p_n is the number of poles pairs; C is the specific thermal capacity, kJ/(kg·K); G is the substance flow rate, kg/s; ρ is the density of refrigerant, kg/m^3; g is the specific weight of substance, kg/m; h is the specific surface, m^2/m; z is the coordinate of the condenser length, m; T_w, T_m are the time constants that characterize thermal accumulating capacity of local water, metal, s; S is the parameter of Laplace transform; $S=\omega j$; ω is the frequency, 1/s.

Indices: 0 is the original stationary mode; 1 is the inlet to the condenser; w is the internal flow – local water; m is the metal wall; r is the external flow – refrigerant.

The true part of transfer function (3) regarding estimation of a change in local water temperature was separated:

$$O_1(\omega)=\frac{(L_1A_1)+(M_1B_1)K_r\varepsilon(1-L_r^*)}{(A_1^2+B_1^2)}. \quad (6)$$

Coefficient K_r includes the temperature of a separating wall θ:

$$\theta=(\alpha_w(\sigma_1+\sigma_2)/2)+A(t_1+t_2)/2)/(\alpha_w+A), \quad (7)$$

where σ_1, σ_2 are the temperatures of refrigerant at the inlet and at the outlet of condenser, K, respectively;

$$A=1/(\delta_m/\lambda_m+1/\alpha_r), \quad (8)$$

where δ is the thickness of the wall of the condenser, m; α is the heat transfer coefficient, kW/(m^2·K); λ is the thermal conductivity of metal of the condenser wall, kW/(m·K); t_1, t_2 are the temperatures of local water at the inlet and outlet of the condenser, K, respectively. Indices: r is the external flow – refrigerant; w is the internal flow – local water.

To use the true part of $O_1(\omega)$, the following coefficients were obtained:

$$A_1=\varepsilon^*-T_wT_m\omega^2;\ \ A_2=\varepsilon^*+1;$$

$$B_1=T_w\varepsilon\omega+T_w\omega+T_m\omega; \quad (9)$$

$$B_2=T_m\omega;\ \ C_1=\frac{A_1A_2+B_1B_2}{A_2^2+B_2^2};\ \ D_1=\frac{A_2B_1-A_1B_2}{A_2^2+B_2^2}; \quad (10)$$

$$L_1=1-e^{-\zeta C_1}\cos(-\xi D_1);$$

$$M_1=-e^{-\zeta C_1}\sin(-\xi D_1). \quad (11)$$

The real part of transfer function (4) concerning evaluation of a change in refrigerant vapor flow rate was separated:

$$O_2(\omega)=\chi_p L_w (C_1 L_1)-(D_1 M_1). \quad (12)$$

To use the real part of $O_2(\omega)$, the following coefficients were obtained:

$$A_1=-T_m\omega^2;\ \ A_2=\varepsilon-T_w T_m\omega^2;\ \ B_1=(\varepsilon+1)\omega;$$

$$B_2=T_w\varepsilon\omega+T_w\omega+T_m\omega+\varepsilon, \quad (13)$$

$$C_1=\frac{A_1A_2+B_1B_2}{A_2^2+B_2^2};\ \ D_1=\frac{A_2B_1-A_1B_2}{A_2^2+B_2^2}; \quad (14)$$

$$L_1=1-e^{-\zeta C_1}\cos(-\xi D_1);\ \ M_1=-e^{-\zeta C_1}\sin(-\xi D_1). \quad (15)$$

The real part of transfer function (5) regarding evaluation of a change in the number of rotations of the compressor motor was separated:

$$O_3(\omega)=K_f\chi_p L_w (C_1 L_1)-(D_1 M_1). \quad (16)$$

To use the real part of $O_3(\omega)$, the following coefficients were obtained:

$$A_1=-T_m\omega^2;\ \ A_2=\varepsilon-T_w T_m\omega^2;\ \ B_1=(\varepsilon+1)\omega;$$

$$B_2=T_w\varepsilon\omega+T_w\omega+T_m\omega+\varepsilon, \quad (17)$$

$$C_1=\frac{A_1A_2+B_1B_2}{A_2^2+B_2^2};\ \ D_1=\frac{A_2B_1-A_1B_2}{A_2^2+B_2^2}; \quad (18)$$

$$L_1=1-e^{-\zeta C_1}\cos(-\xi D_1);$$

$$M_1=-e^{-\zeta C_1}\sin(-\xi D_1). \quad (19)$$

Implementation of transfer functions (3) to (5), obtained based on the use of the operational method for solving the systems of nonlinear differential equations, maintain the parameter of Laplace transform $-S(S=\omega j)$, where ω is the frequency, 1/s. To transfer from frequency region to time region, real parts (6), (12), (16), obtained as a result of mathematical treatment of transfer functions, were separated. These parts are included in integrals (20)–(22), providing the opportunity to obtain dynamic characteristics of the local water temperature, refrigerant flow rate, the number of rotations of the compressor motor, respectively, using the reverse Fourier transform.

$$t_w(\tau,z)=\frac{1}{2\pi}\int_0^\infty O_1(\omega)\sin(\tau\omega/\omega)d\omega; \quad (20)$$

$$G_r(\tau,z)=\frac{1}{2\pi}\int_0^\infty O_2(\omega)\sin(\tau\omega/\omega)d\omega; \quad (21)$$

$$n(\tau)=G_r(\tau,z)K_f(\tau)=$$
$$=\frac{1}{2\pi}\int_0^\infty O_3(\omega)\sin(\tau\omega/\omega)d\omega, \quad (22)$$

where t_w is the temperature of local water, K; G_r is the refrigerant flow rate, kg/s; n is the number of rotations of the electric motor of the compressor, rpm.

5. Results of research into technology of maintaining the functioning of heat pump power supply

According to formulas (1), (2), an integrated mathematical modeling of the heat pump power supply with the use of the developed block diagram was performed (Fig. 2).

According to the proposed block diagram (Fig. 2), Tables 1–3 shows the results of an integrated mathematical modeling of heat pump power supply. Thermal and physical properties of refrigerant R 134a were used. Estimation of the boundary change of the refrigerant temperature at the outlet of the evaporator of the heat pump was established based on maintaining soil heat discharge at the level 10–8 °C [6].

Based on the proposed mathematical substantiation of maintaining functioning of heat pump power supply (1), (2), the block diagrams (Fig. 3, 4) regarding working capacity and maintaining the efficiency of the heat pump system was developed.

The comprehensive integrated system of maintaining functioning of the heat pump power supply was developed (Tables 4, 5). The system is based on prediction of a change in local water temperature regarding a change in refrigerant vapor flow rate, the number of rotations of the electric motor of the compressor. Temperature of the refrigerant at the outlet of the condenser, evaporation pressure, condensation pressure and voltage frequency are measured continuously.

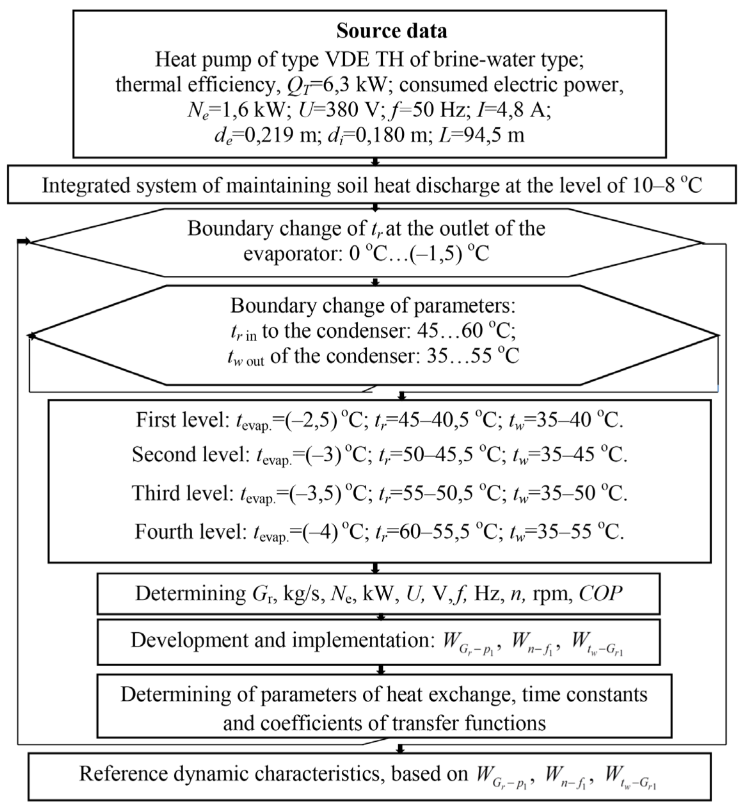

Fig. 2. Block diagram of the integrated mathematic modeling of heat pump power supply: U – voltage, V; f – voltage frequency, Hz; I – current, A; d_e, d_i – external, internal diameter of the soil thermal exchanger, respectively, m; L – length of heat exchanger, m; t_r, t_w, t_{evap} – temperature of refrigerant, local water, refrigerant evaporation, respectively, K; $t_{r\ in}$, $t_{w\ out}$ – temperature of refrigerant at the inlet, local water at the outlet, respectively, K; Gr – refrigerant flow rate, kg/s; n – number of rotation of the electric motor of the compressor, rpm; COP – coefficient of performance of the heat pump system

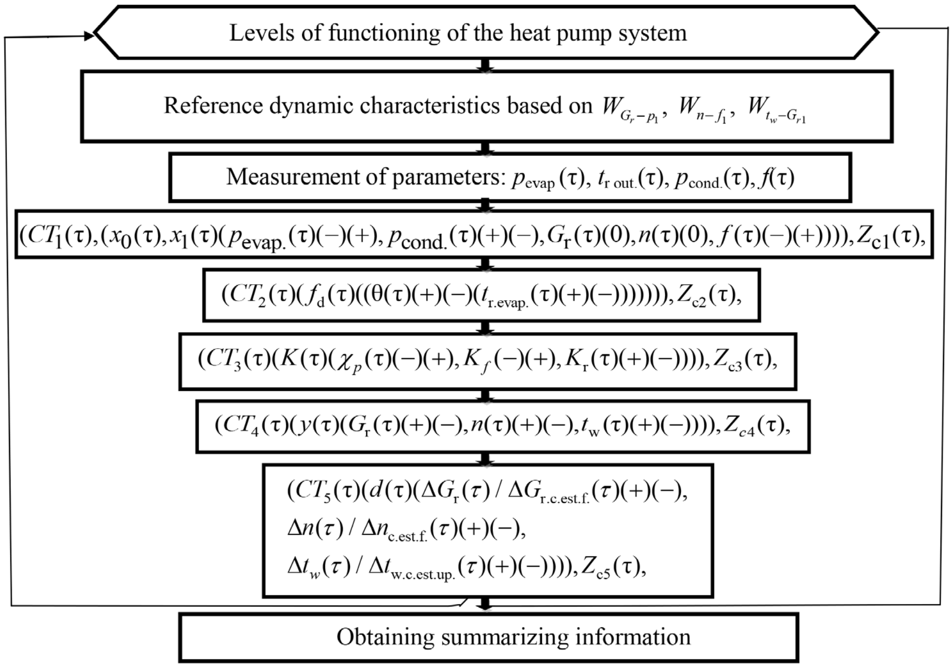

Fig. 3. Block diagram of working capacity control of the heat pump system: $t_{r\ out.}$ t_w, θ are the refrigerant temperatures at the outlet of the condenser, of local water, of separating wall, respectively, K; $p_{evap.}$, $p_{cond.}$ are the evaporation pressure, condensation pressure, respectively, MPa; G_r is the refrigerant flow rate, kg/s; f is the voltage frequency, Hz; n is the number of rotations of the electric motor of the compressor, rpm; CT is the event control; Z is the logic relations; d is the dynamic parameters; x is the impacts; f_d is the diagnosed parameters; y is the original parameters; K is the coefficients of mathematical description; COP is the coefficient of efficiency of the heat pump; ι is the time, Indices: c is the working capacity control; e is the reference value of a parameter, c. est. f. is the constant, estimated value of the parameter of the first level of functioning; c. est. up.is the constant, estimated value of the parameter of the upper level of functioning; 0, 1, 2 is the initial stationary mode, external, internal parameters; 3 is the coefficients of equations of dynamics; 4 is the essential diagnosed parameters; 5 is the dynamic parameters

Table 1

Mode parameters of the heat pump system

Levels of functioning	G_r, kg/s	N_e, kW	U, V	f, Hz	n, rpm.	COP
Level 1	0.0368	1. 23	266. 96	35.13	1053.9	5.13
Level 2	0.0380	1.39	302.17	39.8	1194	4.53
Level 3	0.0391	1.55	337.17	44.36	1330.8	4.06
Level 4	0.0398	1.73	380	50	1500	3.63

Note: G_r is the refrigerant flow rate, kg/s; N_e is the power of electric motor of compressor, kW; U is the voltage, V; f is the voltage frequency, HZ; n is the number of rotations of electric motor of compressor, rpm; COP is the coefficient of efficiency of heat pump system

Table 2

Parameters of heat exchange in the composition of mathematical models of dynamics

Levels of functioning	Parameter		
	α_r, W/(m^2·K)	α_w, W/(m^2·K)	k, W/(m^2·K)
Level 1	1193.16	873.18	496.38
Level 2	1245.17	972.35	536.83
Level 3	1319.06	1129.83	597.21
Level 4	1467.69	1423.96	706.81

Note: α_r is the coefficient of convective heat exchange from the refrigerant to the wall of condenser, W/(m^2·K); α_w is the coefficient of convective heat exchange from the condenser wall to local water, W/(m^2·K); k is the heat transfer coefficient, W/(m^2·K)

Table 3

Values of time constants and coefficients of mathematical models of dynamics

Levels of functioning	T_w, s	T_m, s	L_w, m	ε	L_r, m	L_r^*	ε*	ζ
Level 1	5.36	1.78	23.37	1.5883	2.60	0.28	1.1460	0.72
Level 2	5.02	1.60	20.98	1.5523	2.59	0.28	1.1177	0.67
Level 3	4.14	1.38	18.06	1.3570	2.57	0.28	0.9770	0.62
Level 4	3.28	1.90	14.31	1.1980	2.44	0.28	0.8506	0.60

Local water temperature at the established period of time is determined as follows:

$$t_{wi+1}(\tau)=t_{wi}+\Big(\big(\Delta t_{wi+1}(\tau)/\Delta t_{w.c.est.up.}(\tau)-$$
$$-\Delta t_{wi}(\tau)/\Delta t_{w\ c.est.up.}(\tau)\big)\big(t_{w2}-t_{w1}\big)\Big), \qquad (23)$$

where t_w is the local water temperature, °C; t_{w1}, t_{w2} are the initial and final values of local water temperature, °C, respectively; i is the number of levels of functioning; τ is the time, s. Index: c. est. up. is the constant estimated value of the parameter of the upper level of functioning.

Thus, for example, in the period of time of $94.5{\cdot}10^5$ s (2625 hours) after the beginning of a heating season, which was selected for the city of Kyiv (Ukraine) and is 4,448 hours, absolute value of local water temperature using formula (23) is:

$$40.17\ °C=39.12\ °C+(0.2585-0.2059)(55\ °C-35\ °C).$$

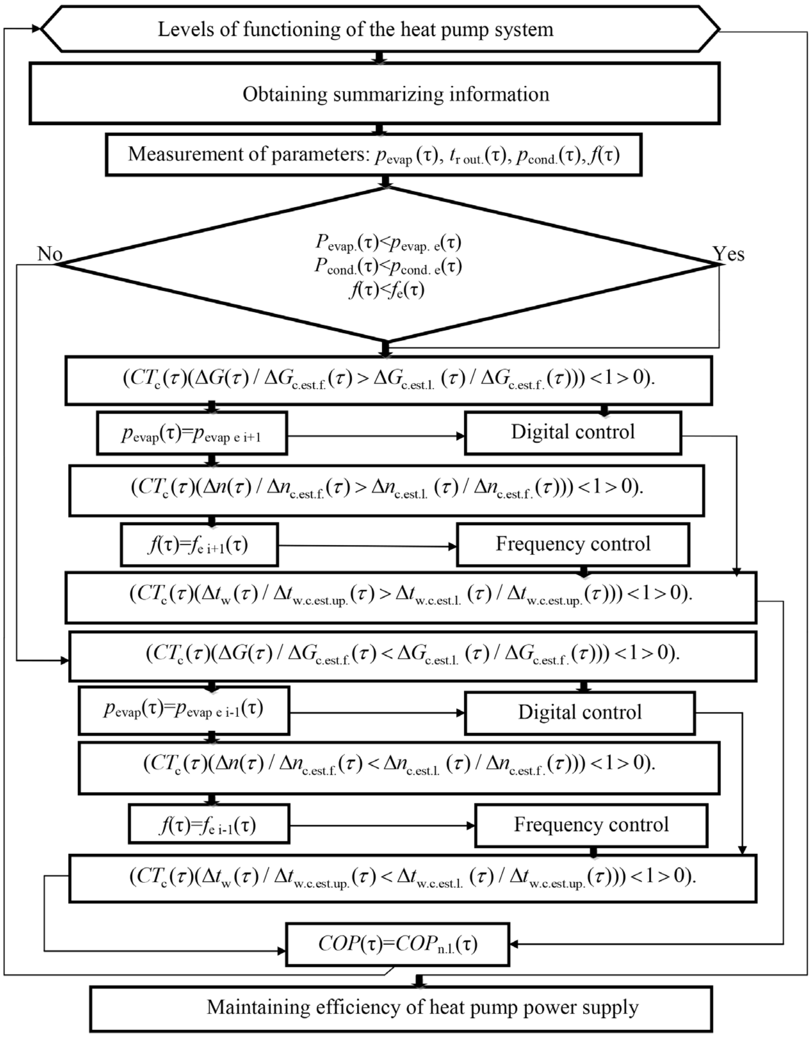

Fig. 4. Block diagram of maintaining efficiency of heat pump power supply: $t_{r\,out}$, t_w are the refrigerant temperatures at the outlet of the condenser, of local water, respectively, K; $p_{evap.}$, $p_{cond.}$ are the evaporation pressure, condensation pressure, respectively, MPa; G_r is the refrigerant flow rate, kg/s; f is the voltage frequency, Hz; n is the number of rotations of the electric motor of the compressor, rpm; COP is the coefficient of efficiency of the heat pump system; ι is the time, Indices: e is the reference value of the parameter; c. est. f. is the constant, estimated value of the parameter of the first level of functioning; c. est. up. is the constant, estimated value of the parameter of the upper level of functioning; l., n. l. are the level, new level of parameters, respectively

Refrigerant flow rate in the established period of time is determined as follows:

$$G_{i+1}(\tau)=G_i+\big(\big(\Delta G_{i+1}(\tau)/\Delta G_{c.est.\,f.}(\tau)-$$
$$-\Delta G_i(\tau)/\Delta G_{c.est.\,f.}(\tau)\big)\big(G_2-G_1\big)\big), \qquad (24)$$

where G is the refrigerant flow rate, kg/s; G_1, G_2 is the initial and final values of refrigerant flow rate, kg/s, respectively (Table 5); i is the number of levels of functioning of heat pump power supply; τ is the time, s. Index: c. est. f. is the constant estimated value of the parameter of the first level of functioning

The number of rotations of the electric motor of the compressor in the established period of time is determined as follows:

$$n_{i+1}(\tau)=n_i+\big(\big(\Delta n_{i+1}(\tau)/\Delta n_{c.est.f..}(\tau)-$$
$$-\Delta n_i(\tau)/\Delta n_{c.est.f}(\tau)\big)\big(n_2-n_1\big)\big), \qquad (25)$$

where n is the number of rotations of the electric motor of the compressor, rpm; n_1, n_2 are the initial and final values of the number of rotations of the electric motor of the compressor, rpm, respectively; i is the number of levels of functioning of the heat pump power supply; τ is the time, s. Index: c. est. f. is the constant estimated value of the parameter of the first level of functioning.

Thus, for example, in the period of time of $94{,}5\cdot10^5$ s (2625 hours) after the beginning of a heating season, which was selected for the city of Kyiv (Ukraine) and is 4448 hours, absolute values of refrigerant flow rate, of number of rotations of the electric motor of the compressor with the use of formulas (24), (25) are:

0.0373 kg/s=0.0369 kg/s+(0.9610–
–0.8291)(0.0398 kg/s–0.0368 kg/s),

1184 rpm=1136.7 rpm+(0.9395–
–0.8334)(1500 rpm–1053.9 rpm).

Graphic dependence of a change in local water temperature within the heating period regarding making decisions on the change in efficiency of the heat pump power supply is shown in Fig. 5.

Thus, for example, in the period of time $94.5\cdot10^5$ s (2625 hours) regarding heating local water from 40.17 °C to 45.36 °C at digital control, it is necessary to make the advancing decision to increase refrigerant flow rate up to the level of 0.0373 kg/s. Frequency control needs making additional decision to increase voltage frequency by 13.2 %, to increase the number of rotations of the electric motor of the compressor up to the level of 1184 rpm.

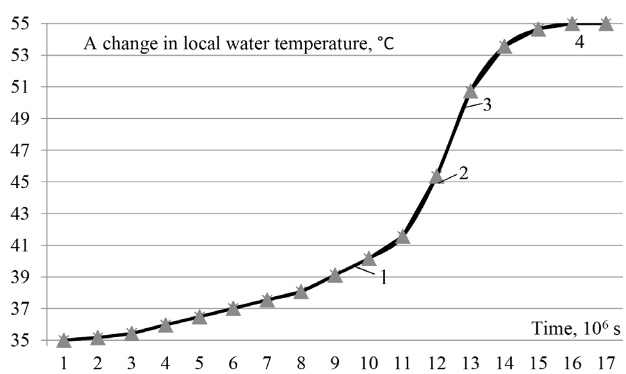

Fig. 5. Maintaining efficiency of heat pump power supply, where 1, 2, 3, 4 are the first, second, third and fourth levels of functioning, respectively

Table 4

Integrated system of maintaining a change in the local water temperature

Time, τ, 10^5 s	A change in parameters	$\Delta t_w(\tau)/\Delta t_{w.\,c.\,est.\,up.}(\tau)$	$t_w(\tau)$, °C
0	$t_{r\,in}$=45 °C; $t_{r\,out.}$=39.55 °C; f=35.13 Hz	0	35
10.5	$t_{r\,in}$=45 °C; $t_{r\,out.}$=39.55 °C; f=34.55 Hz	0.0086	35.17
21	$t_{r\,in}$=45 °C; $t_{r\,out.}$=39.6 °C; f=33.97 Hz	0.0218	35.43
31.5	$t_{r\,in}$=45 °C; $t_{r\,out.}$=39.7 °C; f=33.38 Hz	0.0489	35.97
42	$t_{r\,in}$=45 °C; $t_{r\,out.}$=39.8 °C; f=32.8 Hz	0.0744	36.48
52.5	$t_{r\,in}$=45 °C; $t_{r\,out.}$=39.9 °C; f=32.21 Hz	0.1007	37.01
63	$t_{r\,in}$=45 °C; $t_{r\,out.}$=40 °C; f=31.63 Hz	0.1270	37.54
73.5	$t_{r\,in}$=45 °C; $t_{r\,out.}$=40.1 °C; f=31.05 Hz	0.1533	38.07
84	$t_{r\,in}$=45 °C; $t_{r\,out.}$=40.3 °C; f=30.46 Hz	0.2059	39.12
94.5	Making decision on a change in voltage frequency: f=39.8 Hz; $t_{r\,in}$=45 °C; $t_{r\,out.}$=40.5 °C	0.2585	40.17
105	$t_{r\,in}$=50 °C; $t_{r\,out.}$=44.2 °C; f=37.52 Hz	0.3282	41.56
115.5	Making decision on a change in voltage frequency: f=44.36 Hz; $t_{r\,in}$=50 °C; $t_{r\,out.}$=45 °C	0.5184	45.36
126	$t_{r\,in}$=55 °C; $t_{r\,out.}$=50.1 °C; f=30.46 Hz	0.7865	50.73
136.5	Making decision on a change in voltage frequency: f=50 Hz; $t_{r\,in}$=60 °C; $t_{r\,out.}$=55.1 °C	0.9283	53.57
147	$t_{r\,in}$=60 °C; $t_{r\,out.}$=55.4 °C; f=50 Hz	0.9821	54.65
157.5	$t_{r\,in}$=60 °C; $t_{r\,out.}$=55.5 °C; f=50 Hz	1	55
161.6	$t_{r\,in}$=60 °C; $t_{r\,out.}$=55.5 °C; f=50 Hz	1	55

Note: $t_{r\,in}$, $t_{r\,out}$, t_w are the refrigerant temperature at the inlet of the condenser, at the outlet of the condenser, local water temperature, respectively, °C; f is the voltage frequency, Hz; τ is the time, s. Index:c. est. up. is the constant estimated value of the parameter of the upper level of functioning

Table 5

Integrated system of a change in refrigerant flow rate and the number of rotations of the compressor

Time, τ, 10^5 s	A change in parameters	$\Delta G(\tau)/\Delta G_{c.est.f.}(\tau)$	$\Delta n(\tau)/\Delta n_{c.est.f.}(\tau)$	$G(\tau)$, kg/s	$n(\tau)$, rpm
0	p_{evap}=0.266 MPa; p_{cond}=1.008 MPa; f =35.13 Hz	1	1	0.0368	1053.9
10.5	p_{evap}=0.265 MPa; p_{cond}=1.009 MPa; f=34.55 Hz	0.9979	0.9815	0.0368	1062.1
21	p_{evap}=0.265 MPa; p_{cond}=1.012 MPa; f =33.97 Hz	0.9932	0.9605	0.03681	1071.5
31.5	p_{evap}=0.2645 MPa; p_{cond}=1.014 MPa; f=33.38 Hz	0.9899	0.9505	0.03682	1076
42	p_{evap}=0.264 MPa; p_{cond}=1.016 MPa; f=32.8 Hz	0.9866	0.9341	0.03683	1083.3
52.5	p_{evap}=0.2635 MPa; p_{cond}=1.019 MPa; f=32.21 Hz	0.9820	0.9005	0.03684	1098.3
63	p_{evap}=0.263 MPa; p_{cond}=1.022 MPa; f=31.63 Hz	0.9775	0.8988	0.03685	1099.1
73.5	p_{evap}=0.2625 MPa; p_{cond}=1.028 MPa; f=31.05 Hz	0.9686	0.8567	0.03688	1117.9
84	p_{evap}=0.262 MPa; p_{cond}=1.034 MPa; f=30.46 Hz	0.9610	0.8334	0.0369	1136.7
94.5	Making decision on a change in voltage frequency;f=39.8 Hz; p_{evap}=0.261 MPa; p_{cond}=1.1851 MPa	0.8291	0.9395	0.0373	1184
105	p_{evap}=0.260 MPa; p_{cond}=1.325 MPa; f=37.52 Hz	0.7945	0.7685	0.0376	1260.3
115.5	Making decision on a change in voltage frequency: f=44.36 Hz; p_{evap}=0.258 MPa; p_{cond}=1.3391 MPa	0.7841	0.9297	0.0378	1317.9
126	p_{evap}=0.254 MPa; p_{cond}=1.4298 MPa; f=30.46 Hz	0.7285	0.8005	0.038	1406.9
136.5	Making decision on a change in voltage frequency: f=50 Hz; p_{evap}=0.252 MPa; p_{cond}=1.5206 MPa	0.7262	0	0.038	1500
147	p_{evap}=0.252 MPa; p_{cond}=1.5206 MPa; f=50 Hz	0	0	0.0398	1500
157.5	p_{evap}=0.252 MPa; p_{cond}=1.5206 MPa; f=50 Hz	0	0	0.0398	1500
161.6	p_{evap}=0.252 MPa; p_{cond}=1.5206 MPa; f=50 Hz	0	0	0.0398	1500

Note: Gr is the refrigerant flow rate. kg/s; n is the number of rotations of the electric motor of the compressor, rpm; pevap. is the evaporation pressure, MPa; pcond. is the condensation pressure, MPa; f is the voltage frequency, Hz; τ is the time, s. Index: c. est. f. is the constant estimated value of the parameter of the first level of functioning

Such actions will make it possible to increase vapor compression in the compressor by increasing the vapor temperature at the inlet to the condenser up to the level of 50 °C. Correctness of subsequent local water heating is proved by the use of an integrated system of maintaining the discharge of soil, the temperature of which in this period of time is 9.21 °C [6].

6. Discussion of results of studying the energy-saving technology for maintaining the functioning of heat pump power supply

The integrated system of maintaining the efficiency of the heat pump power supply based on coordination of power production and consumption under power saving conditions.

During control of heat pump power supply, measurement of refrigerant pressure in the heat pump evaporator is usually used. But the use of such measurement is substantiated only under condition of evaluation of a change in refrigerant temperature and under condition when coordination of power production and consumption is not considered. When trying to go beyond these limits in order to enhance the efficiency of the heat pump power supply, the on-off functioning mode is used. When using frequency or digital control, an additional complication of electric circuits is used. There occur objective difficulties, associated with considerable costs of both electrical energy for starting processes and additional capital investments. The way of overcoming these difficulties is presented. It is based on the fact that prediction of a change in local water temperature with measurement of the refrigerant temperature at the outlet of the condenser of the heat pump was proposed. This estimation in the ratio of the measured evaporation pressure is a part of the proposed analytical determining of the refrigerant flow rate as for the digital control and the number of rotations of the electric motor of the compressor for frequency control. This estimation provides an opportunity to influence in advance the coordination of functioning of the external and internal circuits of the heat pump system, i.e. energy production and consumption. The outcome of the conducted research is the results of a comprehensive mathematical and logical modeling of the heat pump power supply using the boundary change in refrigerant temperature at the outlet of the heat pump evaporator, determined on the basis of the integrated system of maintaining soil heat discharge at the level of 10–8 °C. Thus, we established the relationship of a change in refrigerant evaporation temperature in the evaporator as a part of the internal circuit of the heat pump system and an external circuit as for maintaining of a change in soil temperature within the heating period.

Making advancing decisions on a change in vapor flow rate, influencing the action of an electromagnetic valve for digital control, allows ensuring a change of compression of refrigerant suction steam into the compressor and a change in compression of vapor injection into the condenser. Making advancing decisions to change voltage frequency as for a change of the number of rotations of the electric motor of the compressor also allows setting the compliance of compression of vapor suction with a change in injection compression at frequency control. Establishment new vapor parameters regarding the transition to a new level of operation maintains the new coefficient of efficiency of the heat pump system. The results of the study are the continuation of the research in the direction of coordination of production and energy flow rate [3, 6, 13, 14]. The presented results can be used in the development of intelligent systems of functioning of controllers of heat pump systems with both frequency and digital control. The development of this research is the planned testing of the research under the conditions of using heat pump systems of different types and different capacities as a part of the proposed technological system. This is due to the use of different sources of low-potential energy for determining various means of maintaining of a change of parameters during the technological process as, for example, in energy saving technology of biogas production based on the heat pump power supply, the energy source for which is fermented must [3]. When using heat pump systems of various power and improving intelligent systems of controllers functioning, it is necessary to establish the means of control: frequency or digital, in order not to cause unnecessary additional capital investment.

7. Conclusions

1. Prediction of a change in local water temperature at measuring the refrigerant temperature at the outlet of the condenser was proposed. Making advancing decisions on changing the refrigerant flow rate and the number of rotations of the electric motor of the compressor at measuring evaporation pressure, condensation pressure and voltage frequency ensures compliance of a change in evaporation pressure in the evaporator with a change in refrigerant condensation in the condenser.

2. The comprehensive mathematical modeling of the heat pump system, based on an integrated system of maintaining the soil heat discharge at the level of 10–8 °C. The boundary change of the refrigerant temperature at the outlet of the evaporator was established: 0 °C...(–1,5) °C. Mode parameters of the heat pump system, parameters of heat exchange in the condenser, time constants and coefficients of mathematical models of dynamics for the established levels of functioning were determined. The standard dynamic estimations of a change in local water temperature, refrigerant vapor flow rate, the number of rotations of the electric motor of the compressor were obtained. Logical modeling of control of working capacity of the heat pump system, which follows the causation principle, was performed.

The logical unit has the components that evaluate: a change of refrigerant temperature at the outlet of the condenser, evaporation pressure, condensation pressure, voltage frequency that are measured; a change in the temperature of the condenser wall; a change of coefficients of mathematical models of dynamics, $\varkappa_p$, K_f, K_x; a change in local water temperature, refrigerant flow rate, the number of rotations of the electric motor of the compressor; a change of dynamic parameters; the resulting unit of working capacity control for obtaining a functional estimation of a change of local water temperature, refrigerant vapor flow rate, the number of rotations of the electric motor of the compressor.

3. Maintaining the functioning of the heat pump power supply for prediction of a change in local water temperature at continuous measurement of the refrigerant temperature at the outlet of the condenser, evaporation pressure, condensation pressure and voltage frequency was proposed. Maintaining a change in performance of the heat and pump system is based on a comparison of evaporation pressure,

condensation pressure, voltage frequency that are measured with the reference values. Determining of summarizing functional information provides an opportunity to make the following advancing decisions: to maintain a change in evaporation pressure as for a change in refrigerant vapor flow rate for digital control: to maintain a change in evaporation pressure as for a change of refrigerant vapor flow rate and a change of voltage frequency as for a change of the number of rotations of the electric motor of the compressor for frequency control.

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Досліджено вплив установки надроторних елементів у вигляді статорної втулки з повздовжніми прямими пазами над передвключеним колесом на характеристики модельного шнековovідцентрового ступеня. Проведено фізичний експеримент з використанням планування для вирішення проблеми оптимізації геометричних параметрів статорної втулки з повздовжніми прямими пазами в багатофакторній задачі покращення кавітаційно-ерозійних характеристик шнекововідцентрового ступеня за допомогою надроторних елементів. Було визначено спектр збуджуючих частот коливань досліджуваного шнекововідцентрового ступеня від кавітаційних процесів для можливості використання параметра стійкості до кавітаційної ерозії у якості параметра оптимізації. Експериментальним шляхом визначено оптимальні розміри надроторної втулки досліджуваного шнекововідцентрового ступеня: Z=32, b=14, l_1=20, l_2=20. Це дозволило покращити кавітаційно-ерозійні якості шнекововідцентрового ступеня без зміни габаритних розмірів та не погіршити напірні та енергетичні характеристики. Проведено додатковий фізичний експеримент за допомогою альтернативного методу визначення кавітаційно-ерозійних якостей для підтвердження отриманих у дослідженні результатів завдяки застосуванню надроторних елементів у шнекововідцентровому ступені. Використання надроторних елементів у складі шнекововідцентрових ступенів зазвичай обмежувалося лише потребами підвищення кавітаційних якостей ступеня. В рамках дослідження, що описано в даній статті, запропоновано використання цього елементу для боротьби з негативними наслідками кавітаційної ерозії. Була підтверджена можливість такого використання та розроблені науково-методичні рекомендації щодо проектування надроторних елементів у складі шнекововідцентрового ступеня. Впровадження удосконалених перших шнекововідцентрових ступенів з надроторними елементами в існуючі конструкції відцентрових насосів дозволить збільшити наробітку до відмови, що актуально для всіх галузей промисловості, де використовуються відцентрові насоси

Ключові слова: відцентровий насос, шнекововідцентровий ступінь, надроторні елементи, кавітаційно-ерозійні характеристики

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IMPROVEMENT OF CAVITATION EROSION CHARACTERISTICS OF THE CENTRIFUGAL INDUCER STAGE WITH THE INDUCER BUSH

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1. Introduction

Today one of the main problems is a lack of energy resources. Therefore, one of the main tasks of Ukraine and the world is to reduce inefficient energy consumption significantly. It is well known that pumps are used almost in all industries, and according to various estimates consume from 20 % to 25 % of all electricity produced in the world, and in some industries, this value can range to 50 % [1]. One indicator of the pump energy efficiency is the cost of its life cycle, the analysis of