DETERMINATION OF HEAT PUMPS EFFICIENCY

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Introduction. For optimization of regimes and structure of a heat supply heliosystem with the ground accumulator of heat and heat pump (HP) it is necessary to have a method of a power consumed estimation by a HP [1,2,3]. Results of optimization essentially depend on a way of processes modelling in a power supply system within the limits of the chosen method of calculation. Usually are applied two methods: so-called thermodynamic Karno based on a return cycle, and experimental, with use of the results of real installations tests. The first way is not the full proved, besides it has qualitative character. The second, having high reliability, has a number of lacks, as its parametrical limitation of represented data and a way of their representation Ғ usually graphic that is complicated for their application in analytical researches and optimization. Therefore is a necessity to develop the mathematical model well adapted for a wide spectrum of analytical applications.

Analysis of publications. Optimizations of a heat supply systems requests the dependence of the consumed electric power for HP drive from the temperature of the heat- source (ground accumulator or heliocollectors) [1, 2]. These elements are characterized by essential non-stationary regimes of work, and, hence, variable influence on HP. The analytical definition of the electric power from thermal loading of the evaporator and coefficient of performance is as follows

$$N_{\text{к}} = Q_0 / (COP - 1),$$

where Q_0 — thermal loading of the evaporator; COP — HP coefficient of performance.

Due to thermodynamic method coefficient of performance is defined as follows [1, 2, 3, 4]

$$COP = COP_{\text{K}} \nu,$$

where COP_{K} — HP coefficient of performance in a return cycle of Carnot, $COP_{\text{K}} = \left(1 - \dfrac{T_{\text{e}}}{T_{\text{c}}}\right)^{-1}$; ν — degree of thermodynamic perfection HP, under recommendations [4] accept $\nu = 0{,}6$; T_{e} - boiling temperature in the evaporator; T_{c} - condensation temperature in the condenser.

The coefficient of performance in a return cycle of Carnot includes also other factors [1, 2, 3]

$$COP_{\text{K}} = \left(1 - \frac{T_{\text{e}}}{T_{\text{c}}}\right)^{-1} = \left(1 - \frac{273 + t_{\text{e}}^{\text{o}} - \Delta t_{\text{e}}}{273 + t_{\text{c}}^{\text{o}} - \Delta t_{\text{c}}}\right)^{-1},$$

where t_e^o — outlet temperature of the flow form heat source in evaporator; t_c^o— inlet temperature of a heating system flow in the condenser; Δt_e— difference of temperatures of the flow form heat source and coolant at outlet of evaporator; Δt_c— difference of temperatures of a coolant and the heating system flow at outlet of condenser.

According recommendations both Δt_e and Δt_c are accepted to be equal 5 degrees and independent from regimes of HP work [1, 4]. Thus, the model of a thermodynamic method assumes direct dependence of internal parameters HP from external. However, considering presence of complicated processes in HP such as boiling, an overheat of steam, etc., it demands a special substantiation.

Optimizations of system operating modes usually is carried out with the fixed constructive structure of system. In this case it is necessary to find out, when the change of internal regimes parameters of system while optimization requests the change of its structure.

Goal. To develop the method of adequate describing functional depending between internal and external factors of HP operation.

Description of method. It is considered HP which a structure consists of following elements: evaporator, compressor, condenser and a throttle. For investigation of functional tie between defining parameters of HP the mathematical model of the processes in the considered device has be developed. The model is based on definition the areas of zones of boiling and an overheat in the evaporator, and also zones of cooling and condensation in the condenser with use of dimensionless specific thermal loadings of heat exchangers [5]. The system of the equations of considered model includes enthalpy balances of HP cycle and the equations of heat transfer in the evaporator and the condenser. The stable work of heat exchange elements HP within the limits of model is provided with the compressor which productivity and difference of pressure should correspond to an outlet steam of the working agent formed as a result of boiling. As the working agent has been chosen cool agent R–12.

In Fig. 1 are given the data of comparison analysis for change of the electric power consumed by a drive of compressor HP as a dependence on a heat source temperature at modelling of processes of heat exchange (1,2) and a thermodynamic method (3). It is easy to see that for thermodynamic model (a curve 3) power decreases, but for developed model (curves 1 and 2) grows. Thus character of curves 1 and 2 are proved by experimental data with the assumption that a difference of temperatures between heat flows in the evaporator is constant [6].

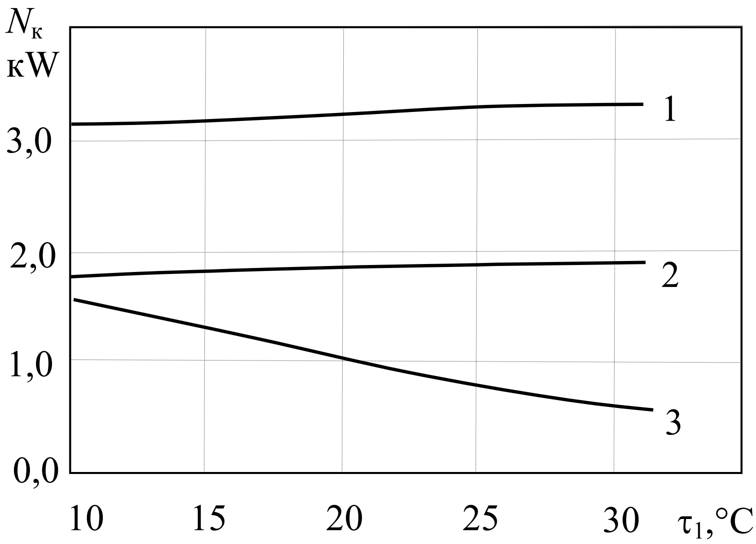

Fig. 1. Change of consumed electrical power for a drive of compressor HP depending on temperature of a heat source for modelling of processes of heat exchange (1,2) and thermodynamic method (3). Nominal pressure in the evaporator for curves, MPa: 1 – 0,2; 2 – 0,33; 3 – 0,33

Change of pressure in the evaporator, and, hence, the change of temperature of a boiling coolant, does not change character of the received dependence (curves 1 and 2). However pressure decrease in the evaporator leads to increase consumed of the electric power at the same temperatures of a heat source. Source temperature τ_1 has a small influences at boiling temperature t_0 (and pressure) in the evaporator.

For example the increase τ_1 on 20 K results increases t_0 only on 1,5...2 K. It is necessary to notice that change of pressure in the evaporator without connection with above noted factor, is result of a variation of others characteristics of HP such as: volume productivity of the compressor, areas of heat exchanger (evaporator and condenser) etc.

The temperature of evaporation only a little depends on external conditions so with increase the temperature of a source the difference of temperature in the evaporator Δt_e also grows, that is in contradiction with model in [1].

The differ of received data from thermodynamic method is a result of existing the various heat exchange zones in a evaporator. Not only evaporation zone, but also the zone of an overheat of steam under certain conditions can be generated. Both zones are quite differ by intensity of heat exchange: intensity in the first it is considerable bigger. In Fig. 2 is given dependence of heat exchange area in evaporator HP from the temperature of heat source. Corresponding: the ratio of overheat areas $F_{\text{oh}} / F_{\text{o}}$ (1); the ratio of an evaporation areas $F_{\text{oe}} / F_{\text{o}}$ (2).

As it follows from curves in Fig. 2 grow of temperature leads to essential redistribution of zones sizes: the boiling zone decreases, and an overheat increases and its area can exceed a boiling zone considerably. The similar phenomenon takes place in the condenser where an area of heat exchange is redistributed between cooling and condensation zones. Hence, to search for optimum operating mode HP as function only of boiling and condensation temperatures, as appears from model [1] is not correct.

In Fig. 3 the dependence of coefficient of performance from temperature of a heat source is shown. The data is received for two variants of initial conditions of calculation HP which can be named by nominal. Under these conditions coolant evaporation is up to a condition of dry sated steam and a heat exchange area represents only one zone F evaporation. All other modes received, for example, with a change of heat source temperature, concern to so-called not nominal. In these cases a warm supply Q_0 can exceed nominal or to be less than it.

Curves are built at different nominal temperature of a source: τ_1=10 °C for a curve 1 and τ_1=20°C for a curve 2, all other initial parameters are the same. In a case when a regime assumes increase τ_1 comparable.

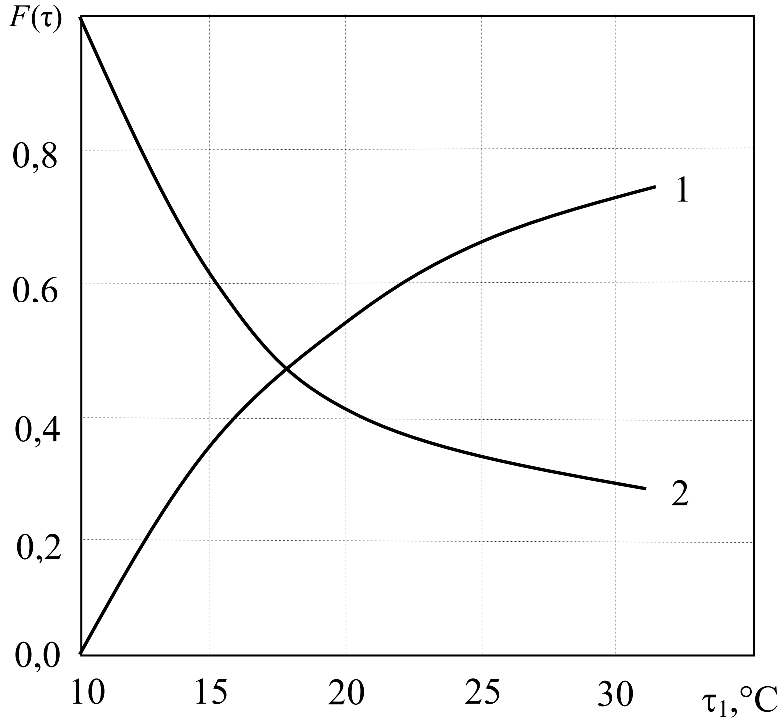

Fig. 2. Dependence of heat exchange area in evaporator HP from the temperature of a heat source. $F(\tau)$ corresponding: the ratio of overheat areas F_{oh} / F_o (1); the the ratio of an evaporation areas F_{oe} / F_o (2)

with nominal, there is a grow of COP, and in the evaporator two zones are formed. Decrease τ_1 from nominal causes reduces the COP. Thus in the evaporator is formed one zone – evaporations one, but a steam is damp. That is caused by a bigger size of heat exchange surface which is a main parameter of calculation for a nominal mode. For growing of a source temperature there is a pressure change, both bottom, and top. Accordingly, the temperature in the evaporator and the condenser change. However the parameter $\frac{T_e}{T_c}$, as has shown in the analysis, depends on temperature of a heat source only a little.

Its changing in considered limits of the initial data variation are differ not more than on 1…2 %.

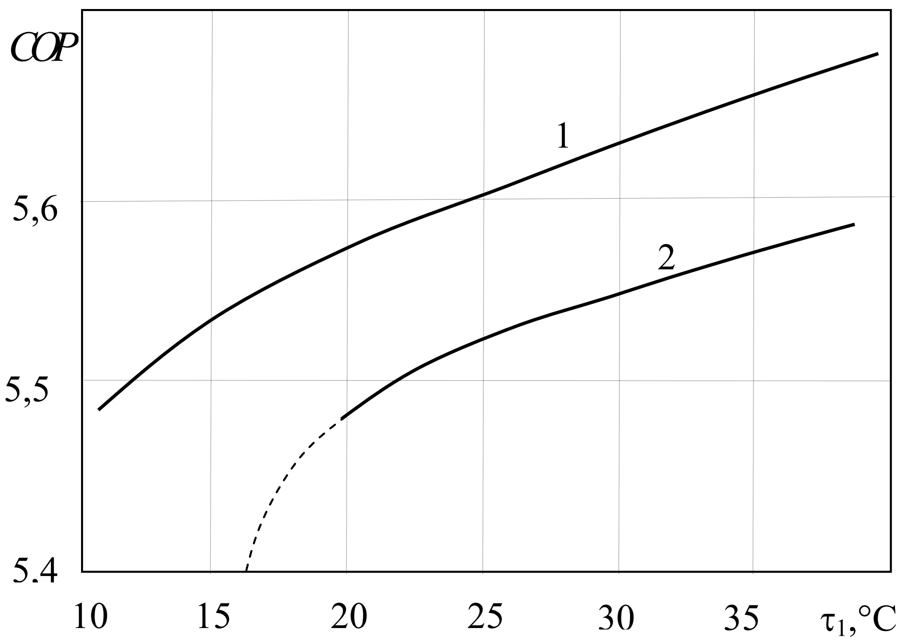

Fig. 3. The function of coefficient of performance from temperature of a heat source for two variants of this temperature as the nominal: 1 – 10 °C; 2 – 20 °C.

For co-ordinates $COP - \frac{\tau_1}{\tau_1^n}$, where τ_1^n temperature of a heat source for nominal conditions, it is possible to receive combination of curves for the different τ_1^n where $\tau_1 > \tau_1^n$. Changing of nominal pressure in the evaporator, and, hence, an evaporation temperatures, leads to displacement of curves along co-ordinate COP. COP will increase with grow of the ratio of nominal pressure in the evaporator and pressure in condenser (and, accordingly increase $\frac{T_e}{T_c}$).

Results. Results of investigations can be given as

$$COP = 0,01 \cdot \exp\left(7,3501\frac{T_e}{T_c}\right) + \left(0,5219\frac{T_e}{T_c} - 0,3919\right)\frac{\tau_1}{\tau_1^n},$$

This formula allows to make an estimation of overall efficiency HP as at a design (a variation τ_1^n and $\frac{T_e}{T_c}$) , as well as in case of regime parametre τ_1 changing for given strucure HP. The differ of developed method from experimental data of various type HP tests consists in possibility of its use for developing the rational operating modes and schemes of heliosystem.

Conclusions. The mathematical model of the heat pump on the basis of consideration thermodynamic and heat mass exchange processes in its elements is developed. The investigations allowed to develop the well-founded analytical method supplementing integrated models of a heat supply systems of any configuration with use of HP. It is shown that application of a thermodynamic model for the alternative analysis of operating modes HP is not correct. The formula connecting HP coefficient of performance with key parameters $\frac{T_e}{T_c}, \frac{\tau_1}{\tau_1^n}$ is found and described. This formula can be used also for optimization of regimes and schemes of HP.

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