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EFFICIENCY OF OPERATION OF A HEAT-PUMP SYSTEM IN TRANSITIONAL MODES

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Abstract:
Heat-nosed technologies are widely available all over the world. In some countries, up to 70% of the need for thermal energy is provided by this technology. Heat pumps in heating and hot water systems operate on a specific cycle. The work cycle in-volves starting a heat pump, heating the coolant and stopping. The number and duration of cycles depend on the need for thermal energy. When starting and getting to work mode, the heat pump works in an unsteady mode. This article is dedicated to the study of this regime.

Keywords: Heat Pump; Hot Water Supply; Heating.

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1. Introduction

Motivation

Unsteady nature of processes in most main units of heat pumps (HP) in transitional operating modes is the main obstacle when defining the real energy efficiency of heat-pump systems (HPS).Experimental studies of such modes were taken on a stand simulating real operational conditions of HPS when heat supplying for objects of low-rise construction.

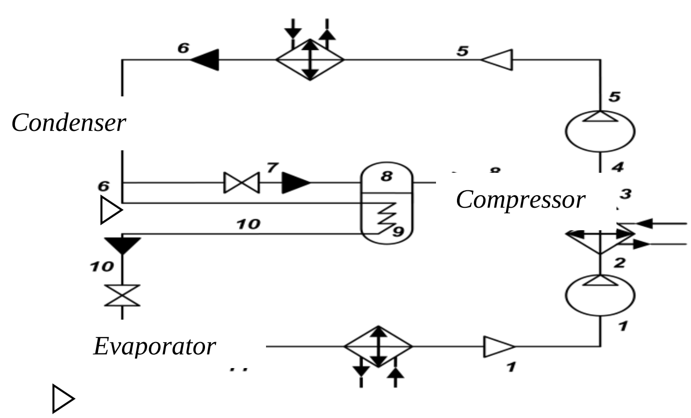

Figure 1: Gashydraulicchartofvapor compression HP.
- liquid; - vapor; - liquid-vapor mixture;
1- 11 – placement of temperature sensors on fre online.

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Continuous phase-by-phase measurement of the electric power consumed by the compressor was carried out using the analytical instrument KAEP-02 (КАЭП-02). The gas-hydraulic circuit of HP of Vitocal-350 brand based on a scroll compressor with intermediate throttling with location of temperature sensors is shown in Figure 1. A non-azeatropic mixture of R-407 freons (R-134a - 52%, R32-23%, R-125-25%) was used as a working material.

2. Method

Two main operating modes of the HP were investigated: 1) from the moment the HP was first turned on to reach the operating mode; 2) maintaining the required temperature of the heat carrier of the heat supply system. The dynamics of temperature changes (Fig. 2) gives a visual representation of the complex nature of the processes in the main HP apparatuses.

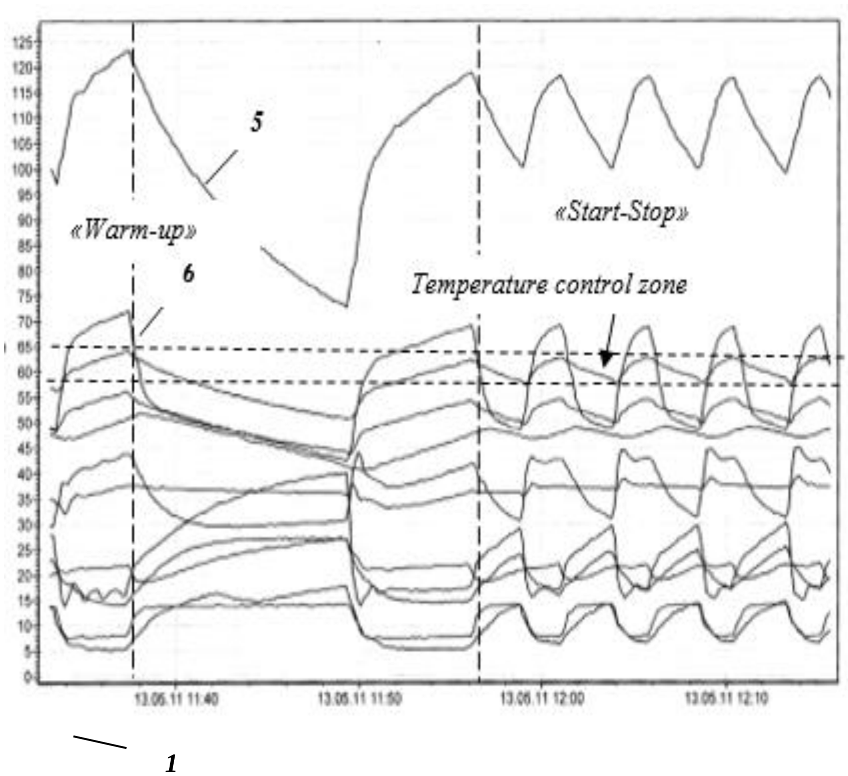

Figure 2: Indications of temperature sensors during operation (For location of the indicated temperature sensors, see Fig. 1).

Entering the operating mode. Changes in the loads of the main devices (Fig. 3) from the moment they are turned on until they enter the operating mode ("Warm-up") are complex. In this case, the values of the conversion coefficient (Fig. 4) first reach their maximum, and then monotonously decrease.

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Q,N,KBт

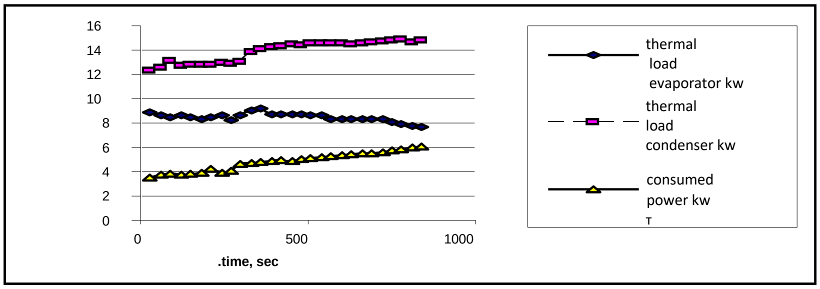

Figure 3: Current heat loads of devices and power consumption

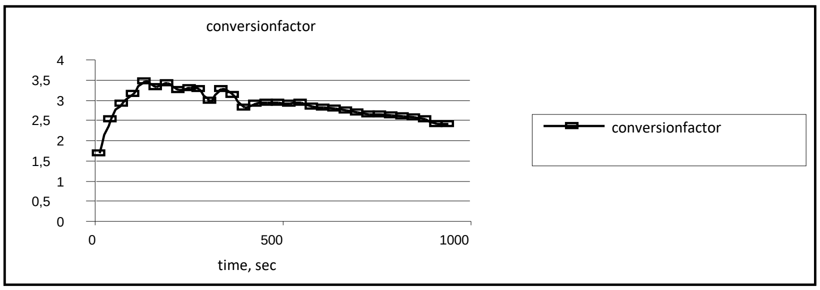

Figure 4: Current conversion rate

Average conversion rate:

$$\overline{\varphi} = \int_0^{\tau} Q d\tau \Big/ \int_0^{\tau} N d\tau \approx \overline{Q} / \overline{N} = \sum_{i=1}^{n=32} Q_i \Big/ \sum_{i=1}^{n=32} N_i = \sum_{i=1}^{n=32} \varphi_i \Big/ 32 = 2{,}95$$

Where: $\overline{Q}$, $\overline{N}$, n - accordingly, the total amount of heat generated, energy expended, the number of intermediate 32 equal time periods for averaging the result for the entire period τ.

“Start-stop” operating mode. This mode is characterized by episodic shutdown and turning on of the compressor to maintain the heat carrier temperature of the heat supply system in a predetermined narrow control range from $t_{\max}$ to $t_{\min}$. The values of the parameters within one cycle are presented in the table.

Table 1: Discrete values of parameters of cycle and heat carriers of heat supply systems and low-grade heat source.

	N, kw	**ΔT evaporation, °C**	**ΔT condensation °C**	**Q evaporator, kw**	**Q condenser, kw**	**φ**
0	*6,10*	*6,3*	*21,2*	*8,06*	*13,31*	*2,2*
1	0	5,7	20,3	7,29	12,75	-
2	0	0,6	5,4	0,77	3,39	-

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3	0	0,1	2	0,13	1,26	-
4	0	0	1,4	0	0,88	-
5	0	0	1,3	0	0,82	-
6	5,41	0,8	3,2	1,02	2,01	0,37
7	5,72	5,2	13,8	6,65	8,67	1,52
8	5,93	6,4	19,8	8,19	12,43	2,10
9	5,99	6,2	20,9	7,93	13,13	2,19
10	6,10	6,3	21,2	8,06	13,31	2,20
00	*0*	*5,7*	*20,3*	*7,29*	*12,75*	-

The energy efficiency of the main operating mode as a whole, taking into account the almost identical repeatability of such mini cycles (see Fig. 2) for one such cycle, was determined by us as follows: $\overline{\varphi} = \int_0^{\tau_1+\tau_2} Q_K d\tau \Big/ \int_0^{\tau_2} N d\tau \approx \overline{Q}/\overline{N} = \{c_P W\rho(t_{\max} - t_{\min})\tau_1 + c_P W\rho(t_{\max} - t_{\min})\tau_2\}/\overline{N} = 2{,}36$

Where: c_p, W, ρ - accordingly, specific heat, volumetric flow rate and density of the heat carrier of the heat supply system; and, accordingly, the duration of half-cycles of heating and cooling the coolant; - energy costs for the compressor electric drive when the temperature of the coolant increases from $t_{\min}$ to $t_{\max}$.

3. Conclusions

Thus, the calculation of $\overline{\varphi}$ requires the correct consideration of the non-stationary heat and mass transfer processes in units of HP, the specifics of the operating modes of HPS, and the development of new methods for summarizing data.

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