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Features of modeling a highly efficient multistage vapor compression heat pump unit

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Abstract. The increase in the cost of fuel and energy resource and the deterioration of the environment from the combustion of traditional fossil fuel, have led to a great interest in energy-saving technology by using secondary energy resources in the thermal energy of industrial, housing and communal services using heat pump units in Russia and abroad. This paper analyzes the well-known two-stage heat pump units, and reveals their advantages in comparison with single-stage. The modeling of a highly efficient multistage vapor compression heat pump unit is proposed. Moreover, a method for calculating a multistage heat pump unit with a high coefficient of performance is presented. In addition, an example of calculating the thermodynamic cycle of a four-stage heat pump unit is presented. The influence of the number of stages on the increase in coefficient of performance in relation to a single-stage heat pump unit, the effect of the temperature difference between the temperature of the high-potential heat source and the temperature of the low-potential heat source on the coefficient of performance were analyzed. In addition, the influence of the initial value of the temperature of the high-potential heat source before heating during the course in the heat pump unit on the value of coefficient of performance for a different number of stages is analyzed under the condition of a constant difference between the heating temperature of the high-potential heat source at the outlet of the heat pump unit and the temperature of the low-potential heat source.

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Особенности моделирования высокоэффективной многоступенчатой парокомпрессионной теплонасосной установки

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Аннотация. В последнее время вследствие повышения стоимости топливно-энергетических ресурсов и ухудшения экологии от сжигания традиционного органического топлива в России и за рубежом проявился большой интерес к энергосберегающей технологии путем использования вторичных энергоресурсов в тепловой энергии промышленного и жилищно-коммунального хозяйства при помощи теплонасосной установки. В работе анализируются известные двухступенчатые теплонасосные установки, раскрываются их преимущества по сравнению с одноступенчатыми. Предложено моделирование высокоэффективной многоступенчатой парокомпрессионной теплонасосной установки, представлена методика расчета многоступенчатой теплонасосной установки с высоким коэффициентом трансформации теплоты. Приведен пример расчета термодинамического цикла четырехступенчатой теплонасосной установки. Проанализировано влияние количества ступеней на прирост коэффициента преобразования по отношению к одноступенчатой теплонасосной установке, а также влияние разности температур между температурой источника высокопотенциальной теплоты и температурой источника низкопотенциальной теплоты на величину коэффициента трансформации теплоты. Изучено влияние начального значения температуры источника высокопотенциальной теплоты перед нагревом на входе теплонасосной установки на величину коэффициента трансформации теплоты для различного количества ступеней при условии постоянной разности температуры нагрева источника высокопотенциальной теплоты на выходе из теплонасосной установки и температурой источника низкопотенциальной теплоты.

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Introduction

In recent decades in our country and, especially abroad, there has been a great interest in the use of secondary energy resources in thermal energy of industrial, housing and communal services. This is mainly caused by a sharp increase in the cost of fuel and energy resources, in addition, the decrease in their reserves and the environmental consequences of burning traditional fossil fuel. One of the solutions to these problems today is the use of energy-saving technologies based on the use of heat pump units (HPUs). Heat pumps utilize low-potential heat source (LHS) from industrial, domestic and natural sources, and generate high-potential heat, while consuming 1.2–1.3 times less primary energy than with direct fuel combustion[1] [1–4]. It is known that a significant part of heat supply (communal and

[1] ANSI/AMCA 210. *Laboratory methods of testing fans for rating*. Arlington; 1985; ASHRAE Guideline, 2. *Engineering analysis of experimental data*. Atlanta; 1986; ASHRAE Standard 116. *Methods of testing for seasonal efficiency of unitary air-conditioner and heat pumps.* Atlanta; 1993.

industrial) in developed countries is carried out using heat pumps [5; 6].

Heat pumps works based on a reversed thermodynamic cycle on a low-boiling point refrigerant.

The theoretical foundations of an HPU are formed as a result of the study of thermodynamic cycles and processes, the development of a methodology for the selection of structures and calculation of the main elements of heat pumps. The most widespread HPUs at the moment are vapor compression HPUs, in which a vapor-liquid cycle is implemented, where the working fluid (refrigerant) is in the form of liquid, wet steam, superheated steam at pressures and temperatures below critical values.

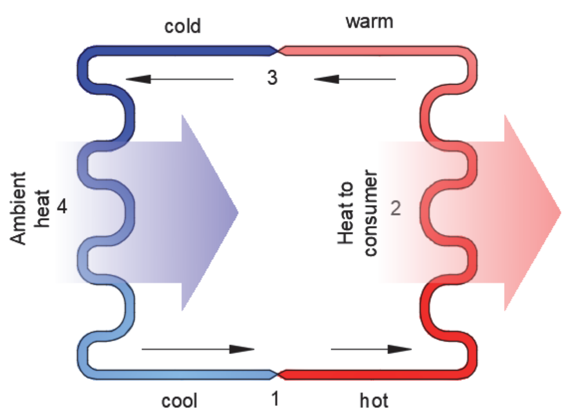

Figure 1. The main elements of HPU

The second problem is to ensure the maximum efficiency of the HPU at variable temperatures of condensation and boiling of the refrigerant, which vary depending on the LHS and the required temperature of the water heated in the condenser. This issue has not been resolved yet. However, an acceptable result can be achieved by fragmentation of the HPU and using circuit solutions, in which hot water is heated in condensers and the LHS is cooled in evaporators when the HPU is switched on in series with a lower compression ratio and counter-flow of heated and cooled water. This method is used by the company "Energy" for relatively powerful HPU (more than 1 Gcal/hour) and leads to an increase of the conversion factor by 1.5–1.8.

Figure 1 schematically shows the main elements of the basic one-stage HPU. Compressor 1 pumps the refrigerant taking into account hydraulic losses to the saturated vapor pressure of the refrigerant in condenser 2. Due to polytropic compression, the temperature of the superheated refrigerant vapor at the inlet to the condenser becomes higher than the temperature of the saturated vapor of the refrigerant in the condenser. Wherein the refrigerant delivers the HHS to the consumer due to the cooling of the superheated vapor, the phase transition from vapor to liquid and the supercooling of the latter. Reducer 3 is required for throttling the refrigerant. Wherein the refrigerant changes from the liquid phase into the vapor phase. The process of the phase transition takes place in the evaporator 4, where heat from the LHS is transferred to the refrigerant.

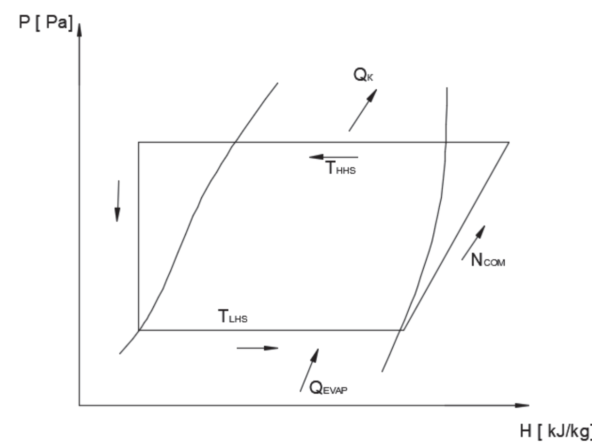

Figure 2. *PH* diagram of the thermodynamic ideal cycle of HPU

Figure 2 shows the *PH* diagram of the thermodynamic cycle of a single-stage HPU. One of the main criteria for the efficiency of the HPU operation is the coefficient of performance (COP):

$$\text{COP} = \frac{Q_k}{N_{\text{com}}},$$

where Q_k is the power of heat flow transferred to the consumer, kW; N_{com} is the consumed power of the HPU compressor, kW.

The COP value is largely dependent on the temperature difference between the LHS and the HHS. The temperatures of the sources are often considered constant, and the COP is compared with the maximum theoretically possible value (according to the Carnot cycle). However, at the inlet and outlet of the HPU, these temperatures are very different (the temperatures of the HHS inlet and outlet appear, as well as for the LHS), therefore, in this case, when calculating the COP, it is usually noted what temperature difference must be taken into account between the HHS and the LHS. For example, the supply of hot water to a residential building (the temperature of the HHS outlet) was carried out using a single-stage HPU, which absorbed heat from the groundwater (the temperature of the LHS inlet). When the temperature difference between HHS and LHS is equal to 10, 30 and 50 °C, the COP is 7, 3.5 and 2.9 respectively, which indicates a decrease in the efficiency of the HPU by 2 or more times [7. P. 61]. Currently, scroll-type compressors with an intermediate supply of vapor-liquid injection of refrigerant into the cavity of the scroll channel are being widely used in HPUs, where, in fact, a single-stage scroll compressor turns into a two-stage.[2] Figure 3 shows a schematic diagram of the HPU with a refrigerant injection circuit, while Figure 4 shows the *PH* diagram of the thermodynamic cycle of this HPU in heating mode.

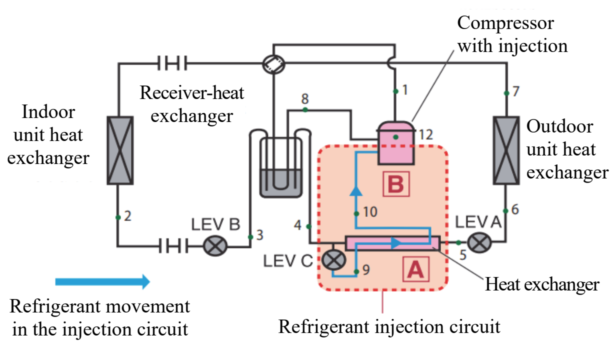

Figure 3. Schematic diagram of an HPU with a refrigerant injection circuit[3]

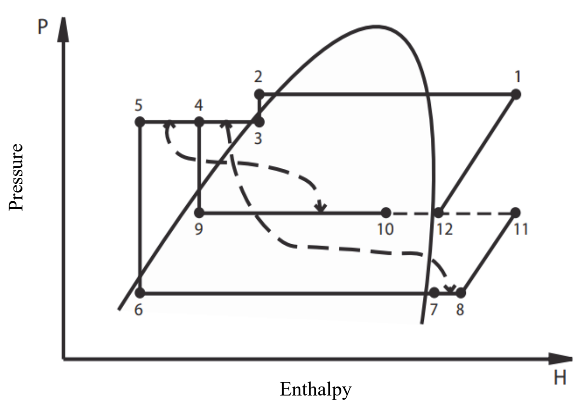

Figure 4. *PH* diagram of the HPU thermodynamic cycle in heating mode

Such an HPU scheme allows to reduce the temperature of the refrigerant during the compression process and thereby reduce the power consumption of the compressor. Therefore, at an ambient air temperature of 7 °C, the HPU heats the air indoors from 20 to 45 °C with a COP coefficient of 5. The disadvantage of the considered HPU scheme is mainly the pumping of the entire refrigerant flow through two compressor stages [8–10].

1. Objectives

The aim of the study is to improve the efficiency of a multistage vapor compression HPU at a high value of temperature difference between HHS and LHS, in addition, to simulate a multistage steam-compression HPU, develop a mathematical model and a calculation method for a multistage HPU, which provides a high COP with a relatively high value of temperature difference between HHS and LHS.

2. Computational studies

2.1. Subject of study

The authors of this work within the framework of the 5–100 Program in the Department of Mechanical and Instrumentation Engineering at the Academy of Engineering in RUDN University developed an experimental model of a two-stage heat pump unit, in which the principle of sequential stepwise heating of the consumer's working medium was tested and experimentally confirmed with simultaneous withdrawal of the refrigerant from each stage. The schematic diagram, in particular, of a three-stage heat pump installation is shown in Figure 5, and the *PH* diagram of the thermodynamic cycle of this HPU in heating mode is shown in Figure 6.

This technical solution is protected by a utility model patent and an invention patent.[4] As it can be seen from the *PH* diagram of the HPU thermodynamic cycle, the total refrigerant flow passes through the compressor of the first stage ($G_1 + G_2 + G_3$), in the condenser of the first stage, the refrigerant with the flow rate G_1 rejects the high-potential heat source HHS to the consumer due to the cooling of the superheated steam, phase transition from vapor to liquid and overcooling.

[2] Heat pump with inverter PUHZ-SHW ZUBADAN INVERTER series. (In Russ.) Available from: https://www.mitsubishi-aircon.ru/product/products/description/puhz-shw_2018.pdf (accessed: 11.05.2021).

[3] Ibid.

[4] Antipov YA, Shatalov IK, Sobennikov EV. *Multistage heat pump unit.* Utility model patent 140197 RF, IPC F25B30/00. No. 2013117874/06. Declared 18.04.2013. Publication 10.05.2014. (In Russ.); Antipov YA, Shatalov IK, Silin AV, Shkarin KV, Sobennikov EV. *Multistage heat pump unit.* Patent for invention 2705696 RF, IPC F25B30/00, F25B1/10. No. 2017102563. Declared 26.01.2017. Publication 25.01.2018. (In Russ.)

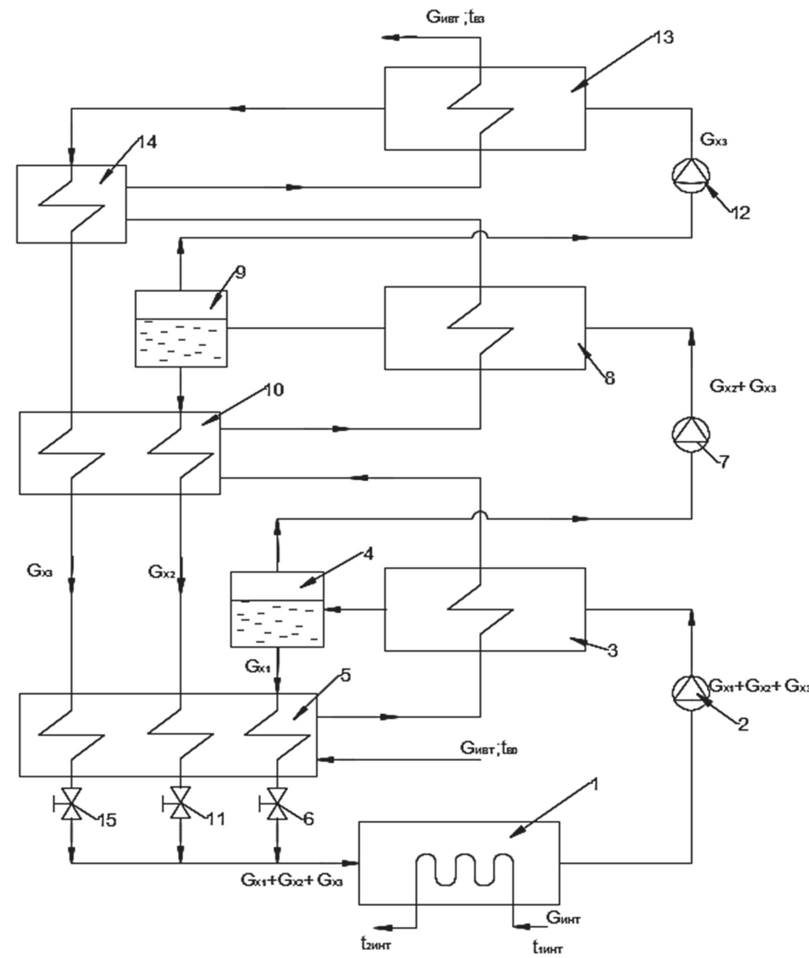

Figure 5. Schematic diagram of a three-stage HPU:
1 – evaporator; *2, 7, 12* – compressor of the first, second and third stages; *3, 8, 13* – condensator of the first, second and third stages; *4, 9* – separators of the refrigerant fractions; *5, 10, 14* – refrigerant subcoolers; *6, 11, 15* – reducer

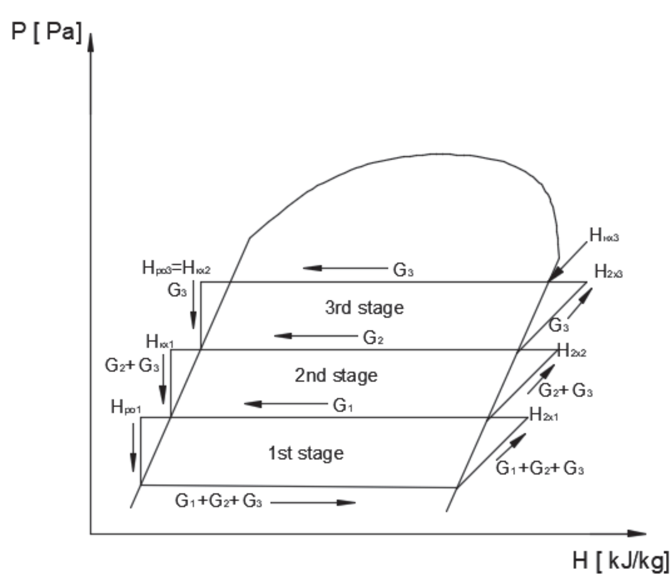

Figure 6. *PH* diagram of the thermodynamic cycle of a three-stage HPU

The remaining total refrigerant flow (G_2 + G_3) is cooled down to the saturated vapor temperature of the first stage condenser and is pumped out by the second stage compressor. Further, in the second stage condenser, the refrigerant with the flow rate G_2, by analogy with the first stage, reject the HHS to the consumer. The remaining refrigerant with a flow rate G_3 through the third compressor enters the condenser of the third stage and, by analogy with the previous stages, through the subcoolers and the corresponding reducer enters the evaporator.

2.2. Calculation method of a multistage compression HPU

The calculation method of a multistage vapor compression HPU is based on the distribution of the fraction of refrigerant flow between all stages, this is initially set by the following initial data:

1. Thermal performance of the multistage HPU – Q, MW.
2. Type of refrigerant.
3. Temperature of the low-potential heat source LHS:
   – *refrigerant temperature at evaporator inlet* t_{H1}, °C;
   – *refrigerant temperature at evaporator outlet* t_{H2}, °C.
4. Temperature of the high-potential heat source HHS:
   – *refrigerant temperature at condenser outlet of the last stage of the HPU* t_{he}, °C;
   – *refrigerant temperature at condenser inlet of the first stage of the HPU* $t_{H1,1}$, °C.
5. The efficiency of the HPU compressors is selected within the range 0.6–0.85.

Then, we determine the value of the temperature rise of the HHS after each stage of the HPU:

$$\Delta t_{st} = \frac{t_{Hz} - t_{H1,1}}{z},$$

where t_{vz} – condenser outlet temperature of the last stage of the HPU; $t_{H1,1}$ – subcooler HHS inlet temperature of the first stage of the HPU; z – number of HPU stages.

According to the *PH* diagram, we determine for each stage the temperature of saturated vapors starting from the condenser of the penultimate stage to the condenser of the first stage according to the formula

$$t_{H[z-i]} = t_{Hz} - \Delta t_{st},$$

where i – the number of the HPU intermediate stage (the stages are counted from the low-pressure stage); z – number of the last stage of the HPU.

6. The saturated vapor temperature in the last stage condenser is calculated by the formula

$$t_{Hz} = t_{hz} + \Delta t_{cz},$$

where Δt_{cz} – underheating of HHS in the last stage condenser; Δt_{cz} = 2–8 °C.

7. A thermodynamic cycle is developed, starting from the last stage of a multistage HPU in the *PH* diagram (Figure 6):

– the isotherms t_{Hz} = const, and t_{z-1} = const are drawn in the area of the *PH* diagram away from the saturation line $x = 0$ to the saturation line $x = 1$;

– compressor inlet temperature of the refrigerant at the last stage is calculated by the formula

$$t_{az} = t_{Hz} + \Delta t_{Pz},$$

where Δt_{Pz} = 1–4 °C – overheating of the refrigerant behind the saturation line;

– taking into account the hydraulic losses of the refrigerant at the inlet and outlet of the compressor of the last stage, the pressure at the compressor inlet is calculated by the formula

$$p_{1(z-1)} = p_{H(z-1)} - \Delta p_{1(z-1)},$$

and

$$p_{2z} = p_{Hz} + \Delta p_{2z},$$

where p_{Hz} and $p_{H(z-1)}$ – saturated vapors pressure of the refrigerant, respectively, of the condensers of the last and penultimate stages of the HPU; Δp_{2z} and $\Delta p_{1(z-1)}$ – pressure loss of refrigerant vapor respectively, at the outlet and inlet to the compressor, which in the first approximation can be taken equal to 1.5% of the pressure at the outlet and inlet to the compressor respectively.

To develop the process of adiabatic work of refrigerant compression from the point of start of compression from the point 1(z–1) an adiabat is drawn to the intersection with the isobar p_{2z} at point $2z_{ad}$, and on the *PH* diagram, we determine the enthalpy of the refrigerant $H_{1(z-1)}$ and $H_{2z_{ad}}$;

– taking into account the efficiency of the compressor, we calculate the work expended in compression of one kg refrigerant of the last stage:

$$L_z = \frac{1}{\eta_i}\left(H_{2z_{ad}} - H_{1(z-1)}\right); \quad (1)$$

– then we calculate the enthalpy of the refrigerant at the compressor outlet of the last stages:

$$H_{2z} = H_{1(z-1)} + L_z.$$

In the *PH* diagram, at the intersection points t_{Hz}, $t_{H(z-1)}$ with the saturation line $x = 0$, we find the saturation enthalpy of the refrigerant H_{cz} and $H_{c(z-1)}$ respectively;

– then we determine the enthalpy of the refrigerant at subcooler outlet of the *i*-th stage of the HPU $H_{p0[z-(n-i)]}$. In the case of heat transfer in the subcooler from the refrigerant to the HHS in the *i*-th stage, it is possible to assume $H_{poi} = H_{cR_{[i-1]}}$.

8. The heat balance equation of the last stage of the HPU as follows:

$$G_{\mathrm{HHS}} C_{P_{\mathrm{HHS}}} \left(t_{Hz} - t_{H(z-1)}\right) = \\ = G_{xz}\left(H_{2z} - H_{H(z-1)}\right)\eta_{\mathrm{eff}},$$

where G_{HHS} and G_{xz} – flowrate in seconds of the HHS and refrigerant in the last stage of the HPU respectively; $\eta_{\mathrm{eff}} = 0.99$ – condenser thermal efficiency; $C_{P_{\mathrm{HHS}}}$ – average mass heat capacity of the HHS in the temperature range from $t_{h(z-1)}$ to t_{Hz}.

Solving the heat balance equation, we calculate the fraction of the refrigerant flowrate α_z of the last stage of the HPU for heating one kilogram of HHS:

$$\alpha_{xz} = \frac{G_{xz}}{G_{\mathrm{HHS}}} = \frac{C_{P_{\mathrm{HHS}}}\left(t_{Hz} - t_{H(z-1)}\right)}{\left(H_{2z} - H_{H(z-1)}\right)\eta_{\mathrm{eff}}}.$$

9. Further, by analogy with the heat balance of the last stage, we compose the equation of the heat balance of the intermediate *i*-th stage of the HPU:

$$G_{\mathrm{HHS}} C_{p_{\mathrm{HHS}}}\left(t_{\mathrm{HHS}_{[i]}} - t_{\mathrm{HHS}_{[i-1]}}\right) = \\ = G_{x[i]}\left(H_{2x_{[i]}} - H_{cx_{[i-1]}}\right)\eta_{\mathrm{eff}} + \\ +\left(G_{xz} + G_{x(z-1)} +{*}{*}{*} + G_{x[i+1]}\right) \times \\ \times \left(H_{2x_{[i]}} - H_{Hx_{[i]}}\right)\eta_{\mathrm{eff}} + \\ +\left(G_{xz} + G_{x(z-1)} +{*}{*}{*} + G_{x[i+1]}\right)\left(H_{cx_{[i]}} - H_{cx_{[i-1]}}\right)\eta_{\mathrm{eff}}.$$

Finally, we obtain:

$$\alpha_{x[i]} = \frac{C_{p_{\mathrm{HHS}}}\left(t_{\mathrm{HHS}_{[i]}} - t_{\mathrm{HHS}_{[i-1]}}\right)}{\left(H_{2x_{[i]}} - H_{kx_{[i-1]}}\right)\eta_{\mathrm{eff}}} - \\ - \frac{\left(\alpha_{xz} + \alpha_{x(z-1)} +{*}{*}{*} + \alpha_{x[z-(i+1)]}\right)\left(H_{2x_{[i]}} - H_{Hx_{[i]}}\right)}{\left(H_{2x_{[z-(n-i)]}} - H_{cx_{[z-(n-i+1)]}}\right)\eta_{\mathrm{eff}}} - \\ - \frac{\left(\alpha_{xz} + \alpha_{x(z-1)} +{*}{*}{*} + \alpha_{x[i+1]}\right)\left(H_{cx_{[i]}} - H_{cx_{[i-1]}}\right)}{\left(H_{2x_{[i]}} - H_{cx_{[i-1]}}\right)\eta_{\mathrm{eff}}}, \qquad (2)$$

where $\alpha_{x[i]} = G_{x[i]}/G_{\mathrm{HHS}}$ – fraction of the refrigerant flowrate of the *i*-th stage of then HPU to the flowrate of one kilogram of HHS.

10. Knowing the fraction of the refrigerant consumption of each stage, it is possible to determine the compressor power of each HPU stage according to the formula:

$$N_{[i]} = G_{\mathrm{HHS}} L_{[i]}\left(\alpha_{xz} + \alpha_{x(z-1)} +{*}{*}{*} + \alpha_{x[i]}\right). \qquad (3)$$

11. The total power of the multistage HPU:

$$N_{\Sigma} = \sum_{i=1}^{z} N_{[i]}. \qquad (4)$$

12. The COP is calculated as follows:

$$\mathrm{COP} = \frac{G_{\mathrm{HHS}} C_{P_{\mathrm{HHS}}}\left(t_{Hz} - t_{H1,1}\right)}{N_{\Sigma}}. \qquad (5)$$

2.3. Results

As an example, we considered a four stage HPU using the following initial parameters: refrigerant – R-600; HHS – water; $t_{H1,1} = 8$ °C – water temperature at the inlet subcooler of the 1st stage of HPU; $t_{Hz} = 88$ °C – water temperature at the outlet of the condenser of the 4th stage of the HPU; $G_{\mathrm{HHS}} = 1$ kg/s – consumption of the heated HHS (water) in a four-stage HPU.

It is assumed that: $\Delta t_{Cz} = 2$ °C – underheating of the HHS in the condenser in each stage of the HPU; $\Delta t_{pz} = 1$ °C – overheating of the refrigerant at the compressor inlet of each HPU stage; $\eta_i = 0{,}85$ – internal efficiency of each compressor in the HPU; $\eta_{\mathrm{eff}} = 0.99$ – thermal efficiency of the condenser at each stage of the HPU.

1. The amount of heating the HHS in each stage of the HPU was calculated:

$$\Delta t_{ist} = \frac{t_{Hz} - t_{H1,1}}{n} = \frac{88-8}{4} = 20\ ^\circ\mathrm{C}.$$

Then the temperature of saturated vapors in the fourth stage condenser is calculated: $t_{H4} = 88 + 2 = 90$ °C; third stage $t_{H3} = 70$ °C; second stage $t_{H2} = 50$ °C; and first stage $t_{H1} = 30$ °C.

2. The thermodynamic cycle of the fourth stage of a multistage HPU in the *PH* diagram is developed.

3. The results of the enthalpy values at the points of the four-stage HPU are presented in Table 1.

Starting from the fourth stage, we calculate the fraction of the refrigerant flowrate in seconds of each HPU stage, to the flowrate of one kilogram of HHS according to the formula (2), the work of compression of 1 kg of refrigerant of the compressor of each stage is calculated by the formula (1), the compressor power of each stage is calculated by the formula (3), the values of which are presented in Table 2.

Table 1

Refrigerant enthalpies at characteristic points of the *PH* diagram of the thermodynamic cycle of a four-stage HPU, kJ/kg

Stage number	H_1	H_2	H_{xH}	H_{xc}	H_{po}
4	682	708.4	707.8	426.5	373.6
3	654	683.45	680.7	373.6	321.6
2	637	658.16	652.8	321.6	271.8
1	597	631.7	625.3	271.8	223.4

Table 2

The calculation results of the four-stage HPU

Stage number	α_{xi}	L_i, kJ/kg	N_i, kW
1	0.1049	32.06	22.685
2	0.1545	27.79	16.75
3	0.1954	24.27	10.88
4	0.2528	21.25	5.37

Note: α_*xi* – refrigerant consumption fraction; *Li* is the work expended to compress 1 kg refrigerant; *Ni* – compressor power of each HPU stage.

4. Using formulas (4) and (5), we calculate the total power of the HPU and COP_4: N_{HPU4} = 55.68 kW; COP_4 = 6.02. For reference, the power of the compressor and COP_1 of a single-stage HPU, other things being equal, are equal to N_{HPU1} = 90.35 kW; COP_1 = 4.71. As can be seen from the above calculations, water heating from 10 to 90 °C with a flowrate of 1 kg/s (the thermal power of water heating is 425.5 kW), replacing the one-stage HPU with a four-stage one, it is possible to reduce the total compressor power by 34.67 kW.

Therefore, replacing a one-stage HPU with a four-stage one will lead to a relative increase ΔCOP = 23%.

According to the above methodology, an additional calculation of two-, three- and five-stage HPU was carried out, the result of which is shown in Table 3.

Table 3 shows that the main relative increase in ΔCOP falls on the two-stage HPU. Further, this increase at subsequent stages decreases sharply and becomes less than 5%.

Table 3

Change in the COP depending on the number of HPU stages

Number of HPU stages	COP	ΔCOP relative to 1-stage HPU, %	ΔCOP relative to the previous stage of HPU, %
1	4.71	0	0
2	5.54	15	15
3	5.84	19.35	5.13
4	6.02	21.76	2.99
5	6.13	23.16	1.79

The influence of the water temperature at the inlet to the first stage subcooler on the efficiency of the multistage HPU operation at a constant temperature t_{Hz} = 90 °C is considered; and the temperature at the inlet to the evaporator t_{H1} = 10 °C.

Table 4

Change in the COP depending on the water temperature at the HPU inlet

Number of HPU stages	COP HPU at different values of $t_{H1,1}$					
	10 °C	20 °C	30 °C	40 °C	50 °C	60 °C
1	4.71	4.53	4.27	4.05	3.80	3.55
2	5.54	5.068	4.74	4.344	4.3335	3.68
3	5.48	5.269	4.854	4.278	4.15	3.797
4	6.02	5.845	4.95	4.562	4.252	–

The calculation results given in Table 4 clearly show a gradual decrease in the COP of HPU with a different number of stages with an increase in the temperature difference ($t_{H1,1}$–t_{H1}).

Conclusion

The outlined method for calculating the thermodynamic cycle makes it possible to simulate a multistage vapor compression heat pump installation, in which, with an increase in the number of stages, the COP increases to 20–23% and the main increase in ΔCOP, equal to 15%, falls on the two-stage HPU.

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