Vol. 12, 2023, 132-142

DOI: 10.17512/bozpe.2023.12.15

Construction of optimized energy potential

Budownictwo o zoptymalizowanym potencjale energetycznym

ISSN 2299-8535 e-ISSN 2544-963X

Parameters of a cascade two-stage air-to-water heat pump at low ambient temperature

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Abstract: The results of the numerical calculation of the thermodynamic parameters of a cascade (two-stage) air-to-water heat pump at low atmospheric air temperatures are presented. A Freon R407c is used in the first circuit and a R152a is used in the second circuit. Calculations are made for ambient temperatures from 7°C to –25°C. The effect of underrecovery temperatures in heat exchange equipment is shown. Cycles with R245fa, R142b, R114, R123, R290, R236fa, R600a refrigerants, a mixture of isobutane and isopentane have been investigated. The effect of the complexity of a two-stage scheme on the efficiency of the cycle has been evaluated. The use of more complex schemes (for example, with liquid injection) does not affect the thermodynamic perfection of the heat pump and may be appropriate only to ensure operational reliability. Reducing the underrecovery temperature to 1 K is shown to increase the thermodynamic efficiency of the installation.

Keywords: heat pump, low air temperature, refrigerant, thermodynamic parameters

Please, quote this article as follows:

Redko A., Ujma A., Pavlovskyi S., Redko I., Gvozdetskii O., Chaika Y. & Borodai D., Parameters of a cascade two-stage air-to-water heat pump at low ambient temperature, Construction of Optimized Energy Potential (CoOEP), Vol. 12, 2023, 132-142, DOI: 10.17512/bozpe.2023.12.15

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Introduction

Modern heat pump units (HPU) manufactured by Mitsubishi Electric (Zubadan), Viessmann, Cooper & Hunter, Helio therm and others operate in air-to-water mode at outdoor temperatures up to –30°C. However, their performance decreases at air temperatures below –15°C. Heat pumps operate according to a single-stage scheme. The Zubadan heat pump thermal circuit uses the two-phase refrigerant injection technology to the compressor when the outdoor temperature drops to ensure stable performance. The results of heat performances of the system tested at –20°C ambient temperature are presented in the paper (Fei Qin et al., 2015). The system cycle process with refrigerant injection is analyzed using a lgP-H diagram.

Air-to-water heat pumps are successfully operated in countries with harsh climatic conditions (Lei et al., 2023). Heat pumps are characterized by a high energy efficiency class, thus for 1 kW of power consumed from the network, 4.2-5.1 kW of thermal energy is produced.

In Ukraine, heat pumps are used in a bivalent heating system (a heat pump and a backup boiler) (Matsevity, 2014; Redko et al., 2020; 2021a; 2021b; 2021c). The following paper (Gschwend et al., 2016) presents simulations of two-stage heat pump systems. These simulations are carried out for central and northern locations in the US. The focus of this paper is the applicability of cold climate heat pumps in these climates. Simulation results show the most significant energy saving (up to 90%) can be achieved by improving building insulation and heat recovery of the ventilation system. A second major improvement is the usage of two-stage heat pumps systems, which drastically reduces the need for backup heating systems and leads to seasonal efficiencies (SCOPs) in the area of 3.0, or in other words an energy saving of 20-35%. The use of heat pumps in district heating is one of the most promising technologies for enhancing the efficiency of systems and for meeting the 2030 and 2050 European energy and climate targets (Sayegh et al., 2018). In the paper (Deng et al., 2019), the authors introduce on-going field test results on operational energy performance of 32 heat pumps systems for space heating with various heat sources in cold regions of China.

The works related to the rational design of HPU, as well as the search for their optimal circuit solutions and operating modes are of current interest.

Unlike traditional heat generating equipment, the choice of HPU capacity is mainly determined by its ability to provide the required coolant parameters in accordance with the annual seasonal load schedule (Lisheng et al., 2011; Redko et al., 2020; Wang & Li, 2019; Xing-Qi et al., 2014). It should be noted that the heating load, in principle, is unfavourable for the HPU, because the installed capacity of the HPU, designed for the maximum heating load, has a small number of hours of use. At the same time, the energy consumption in heating HPU at maximum load is 3-4 times higher than that at average load.

It is known that for most regions of Ukraine, single-stage heat pumps can cover the heat load for heating at outdoor temperatures from plus 8°C to minus 5°C. When the outside air temperature drops below minus 5°C, it is necessary to switch to the bivalent-parallel-alternative mode of operation. In this mode, when the temperature of the balancing point (Lisheng et al., 2011; Matsevity, 2014; Redko et al., 2020; Wang & Li, 2019; Xing-Qi et al., 2014) is reached, the configuration of the technological heat supply scheme changes and the heat generator included in it begins to heat the coolant coming after the HPU condenser to the required parameters. The duration of the period of joint operation of the HPU and the heat generator in the climatic conditions of the middle zone is, as a rule, short. It depends on the level of thermal protection of the building envelope, as well as the required coolant temperatures in the supply and return lines of the heating system, determined by the quality control schedule. Thus, for example, at an outdoor air temperature of 12°C, carrying out heating of the heat carrier in the boiler to plus 80ºC, the heat carrier from the return line of the heating system must be supplied to the inlet of the HPU condenser, which during this period has a temperature of the order of plus 60°C. For single-stage heat pumps using a medium-temperature refrigerant as a working medium, the allowable condensation temperature in terms of pressure is also 60°C. Thus, it is no longer possible to condense the refrigerant at high temperatures in the return line. This circumstance greatly narrows the scope of use of the bivalent-parallel-alternative scheme, limiting it only to heating systems with a significant temperature difference in the supply and return lines (Redko et al., 2021a, Redko et al., 2021b, Redko et al., 2021c; Savchenko & Lis, 2020, Shepitchak & Zhelykh, 2020).

The use of high-temperature refrigerants, such as R245fa, R236fa, R142b, R114, R123/R290, in single-stage heat pumps can generally lead to a decrease in the specific heat output of the unit, and, as a result, to an increase in its weight and size characteristics (Matsevity, 2014; Redko et al., 2020; Xing-Qi et al., 2014). If the choice of temperature limits of the cycle turns out to be such that $\pi > 6$, then it is advisable to switch to a scheme with two-stage compression. It should be noted that even if the volumetric productivity of a single-stage compressor is more than 10% higher than the total productivity with two-stage compression, then using a single-stage scheme under operating conditions, despite its relative simplicity, becomes economically impractical, since energy consumption and maintenance costs increase.

The question of the use of two-stage heat pumps in heating systems has long been ignored. Previously, it was believed (Arpagaus et al., 2018; Hosseinnia et al., 2023; Kosmadakis, 2019; Wang & Li, 2019;) that the role of multistage vapor compression thermotransformers in heating technology was not as great as for the purposes of deep cooling in refrigeration technology.

1. Thermodynamic parameters of the cycles of a cascade air-to-water heat pump

Nowadays, there are more than 20 two-stage schemes of vapor-compression refrigeration machines and heat pump installations that are different in terms of equipment composition. Two-stage circuits are formed by including auxiliary elements and additional piping lines in the refrigerant circuit. All the complications of two-stage schemes are aimed at reducing internal losses from irreversibility in the cycle, namely, at reducing the steam temperature at the end of the compression process and increasing the liquid supercooling before throttling.

Preliminary analysis show that for the high-temperature mode of HPU operation, only two-stage schemes with incomplete intermediate cooling by parallel throttling, containing an economizer, are of interest. Schemes with an intermediate vessel, complete intermediate cooling and sequential throttling are unsuitable for HPU. Most of the high-temperature refrigerants being considered (R245fa, R142b, R123, R236fa, R114, R600a, a mixture of isobutane and isopentane) have a rather steep right-hand boundary curve, therefore, with an increase in the condensing temperature, the steam parameters after the second stage compressor for schemes with an intermediate vessel can be in close proximity to the area of wet steam, which is unacceptable.

Using the recommendations of the work, an assessment of the influence of the structural complexity of two-stage schemes on the thermodynamic efficiency of the cycle has been made. It has been established that for the HPU operation range at the condensation temperature $Tc = 80\text{-}95°C$ and the evaporation temperature $Tev = 10\text{-}25°C$, it is advisable to use a scheme with incomplete intermediate cooling, parallel throttling and liquid subcooling.

The analysis of the thermodynamic efficiency of a two-stage HPU scheme with R123 refrigerant, incomplete intermediate cooling, parallel throttling and liquid subcooling shows a significant dependence of the relative exergy losses in the HPU elements on changes in the outside air temperature, which indicates the need to select the optimal modes for regulating the flow of thermal water and refrigerant.

Increasing the condensation temperature of the working substance in HPU up to 90-95°C is not effective enough using, for example, a sequential thermal scheme for switching HPU, two-stage and combined HPU.

Previously, in (Arpagaus et al., 2018; Hosseinnia et al., 2023; Kosmadakis, 2019; Redko et al., 2020; Wang & Li, 2019; Xing-Qi et al., 2014), it was pointed out that it was promising to create two-stage compression units and binary heat pumps with two heat carriers.

The choice of the most effective working substances for specific temperature limits can be carried out by conducting multivariate numerical studies.

Despite the fact that low-temperature heating is widely used abroad, the existing needs and demand for high-temperature heat pumps set the task of their development. Currently, firms in the USA, Japan, and Germany are mastering the industrial production of HPUs that provide heating the coolant in the range of 20-85°C. In 2008-2009 Rotex (Germany) mastered the production of HPU, characterized by the following thermal parameters: thermal power varies from 11 to 16 kW; the conversion factor is 2.88-3.08; evaporation temperature is plus 25°C; condensation temperature is plus 80°C; heat carriers in the binary cycle are R410c and R134a refrigerants.

When developing HPU, various heat carriers are used as working substances, namely organic liquids and their mixtures, freons and their mixtures, ammonia and water-ammonia mixtures, carbon dioxide and others. New generation ozone-safe freons are widely used. When using geothermal energy, the temperature potential of the waste geothermal fluid with a temperature of 15-30°C can be increased by using a heat pump. The choice of the most effective working substances for specific temperature limits can be carried out by conducting multivariate numerical studies.

The purpose of the work is to calculate the thermodynamic parameters of the cycles of a cascade air-to-water heat pump with various freons at various atmospheric air temperatures in order to improve the methodology for increasing their energy efficiency.

2. Method of calculation. Main results

The technological scheme of the cascade two-stage pump is shown in Figure 1.

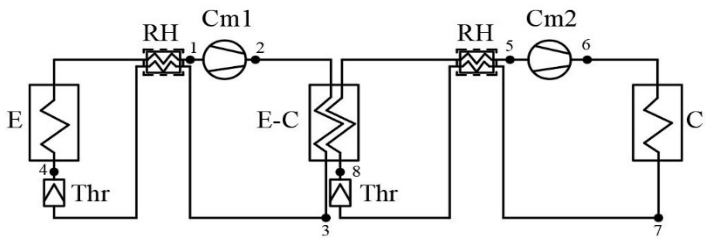

Fig. 1. Thermal scheme of the cascade HPU (*own research*)

The heat pump consists of an evaporator E, a condenser C, an evaporator condenser C-E, two compressors Cm1 and Cm2, regenerative heat exchangers RHE, and throttles. Figure 2 shows the cycles of the lower and upper circuits of the heat pump.

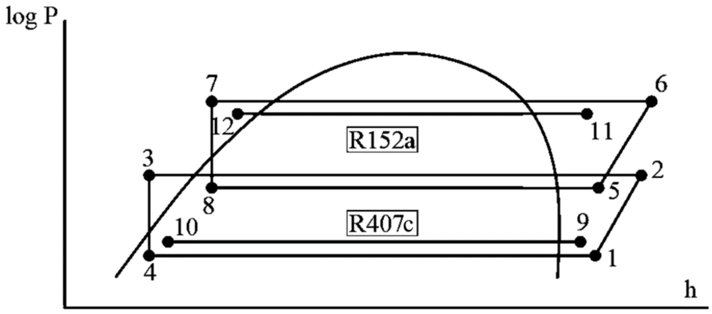

Fig. 2. Cycles of the cascade HPU (*own research*)

1. Mass air flow:

$$m_1 = \left(h_4 - h_1\right)/Cp \cdot \Delta t \tag{1}$$

2. Mass flow of heated water:

$$m_2 = \left(h_7 - h_6\right) / Cp \cdot \Delta t \tag{2}$$

3. The amount of heat transferred from the coolant of the lower cycle (R407c) to the coolant of the upper cycle (R152a):

$$Q_E = m_{\mathrm{R407c}}\left(h_3 - h_2\right) \tag{3}$$

4. Lower cycle conversion factor:

$$\varphi = Q_E / W_1 \tag{4}$$

5. The amount of heat transferred from the coolant of the upper cycle to heated water:

$$Q_E = m_{\mathrm{R152a}}\left(h_7 - h_6\right) \tag{5}$$

6. HPU conversion factor:

$$\varphi = Q_C / \left(W_1 + W_2\right) \tag{6}$$

The physical and chemical properties of freons used in heat pump circuits are as follows:

– Freon R407c (HFc) is an isentropic mixture with a temperature glide. Mixture composition: difluoromethane (R32) – 23%; pentafluoroethane (R125) – 25%; 1.1,1.2 - tetrafluoroethene (R134a) – 52%. Critical temperature is 86.05°C; critical pressure is 4652 kPa; boiling point at atmospheric pressure is 44.0°C (229.15 K). Environmental characteristics are ODP = 0; GWP = 1370.
– Freon R152a (C2H4F2) is difluoroethane. Critical temperature is 113.3°C; critical pressure is 4520 kPa; boiling point is –25°C. ODP = 0; GWP = 140.

The calculation results are given in Tables 1-3.

Case A

- Ambient temperature is 7°C.
- The difference between the temperature of under recuperation and at the pinch point of the evaporator is 5 K.
- The difference between the under-recuperation temperature and at the pinch point in the condenser is 1 K.
- The compressor power of the 1st stage is 0.333 kW.
- Compressor power of the 2nd stage is 1.33 kW.

The values of the parameters at the control points of the heat pump cycles at an air temperature of 7°C are given in Table 1.

Table 1. Parameter values of the heat pump cascade cycle at ambient temperature 7°C (*own research*)

No	*P* [kPa]	*t* [°C]	*x*	*S* [kJ/(kg·K)]	*h* [kJ/kg]	*m* [kg/s]
			air			
9	101.3	7.0	1.0	4.02034	–18.1185	1.0
10	101.3	1.6	1.0	4.00112	–23.4499	1.0
			1st circuit			
1	494.8	2.0	1.0	1.96676	–8877.2988	0.0291
2	500.0	2.5	1.0	1.96707	–7776.9659	0.0291
3	500.0	–3.1	0.173	1.29296	–900.8398	0.0291
4	494.8	–3.4	0.174	1.29313	–9060.8398	0.0291
			2nd circuit			
5	200.0	–2.5	1.0	2.76959	–7250.8825	0.0214
7	774.0	31.0	0.0	1.81063	–7500.3717	00214
8	200.0	31.0	0.197	1.82854	–7500.371	
			water			
11	200.0	30.0	0.0	3.14428	–15833.0356	0.3632
12	200.0	35.0	0.0	3.20440	–15814.6555	0.3632

Case B

- Ambient temperature is –5°C.
- Compressor power of the 1st stage is 0.420 kW.
- Compressor power of the 2nd stage is 1.876 kW.
- The total capacity of the compressors is 2.246 kW.
- Generated amount of heat is 7.48 kW.
- COP conversion factor = 3.3.

The values of the parameters at the control points of the cycles at an air temperature –5°C are given in Table 2.

Case C

- Adiabatic efficiency of compressors is 0.75.
- Ambient temperature is –25°C.
- Inlet water temperature is 35°C.
- Under-recovery and at the pinch-point temperature difference is 5 K. The heat carrier in the 1st circuit is R407c and the one in the second circuit is R152a.
- 1st stage compressor power is 1.228 kW.
- 2nd stage compressor power is 2.041 kW.
- The total capacity of the compressors is 3.269 kW.
- Generated amount of heat is 8.072 kW.
- COP conversion factor = 2.469.

The parameters at the control points of the cycles are given in Table 3.

Table 2. Parameter values of the heat pump cascade cycle at ambient temperature –5°C (*own research*)

No	*P* [kPa]	*t* [°C]	*x*	*S* [kJ/(kg·K)]	*h* [kJ/kg]	*m* [kg/s]
air						
9	101.3	–5.0	1.0	3.97707	–29.9786	1.0
10	101.3	–10.2	1.0	3.9576	–35.1467	1.0
1st circuit						
1	319.7	–10.0	1.0	1.97990	–8884.1335	0.0294
2	500.0	10.9	1.0	1.99259	–8864.8233	0.0294
3	500.0	–3.1	0.173	1.29319	–9060.7778	0.0294
4	319.7	–15.2	0.245	1.30196	–9060.7778	0.0294
2nd circuit						
5	200.0	5.9	1.0	2.79988	–7242.5740	0.0231
6	1080.0	93.0	1.0	2.85644	–7161.4857	0.0231
7	1080.0	40.0	0.0	1.85919	–7485.0003	0.0231
8	200.0	–8.1	0.25	1.88653	–7485.0003	0.0231
water						
11	200.0	35.0	0.0	3.21037	–15812.8354	0.1968
12	200.0	44.4	0.0	3.33200	–15774.7841	0.1968

Table 3. Parameter values of the heat pump cascade cycle at ambient temperature –25°C (*own research*)

No	*P* [kPa]	*t* [°C]	*x*	*S* [kJ/(kg·K)]	*h* [kJ/kg]	*m* [kg/s]
air						
9	101.325	–25	1.0	3.90066	–49.6931	1.0
10	101.325	–25	1.0	3.88081	–54.5712	1.0
1st circuit						
1	137.984	–30.0	1.0	2.00839	–8896.3387	0.0297
2	500.0	28.1	1.0	2.04343	–8854.9445	0.0297
3	500.0	–3.1	0.17	1.29319	–9060.7778	0.0297
4	137.984	–34.9	0.34	1.32530	–9060.7778	0.0297
2nd circuit						
5	200.0	20.6	1.0	2.85172	–7227.7108	0.0237
6	1080.0	108.5	1.0	2.90933	–7141.7090	0.0237
7	1080.0	40.0	0.0	1.85919	–7485.0003	0.0237
8	200.0	–8.1	0.25	1.88653	–7485.0003	0.0237
water						
11	200.0	35.0	0.0	3.21037	–15812.8354	0.2018
12	200.0	45.0	0.0	3.33932	–15772.4580	0.2018

The dependence of the total compressor capacity on the air temperature is shown in Figure 3.

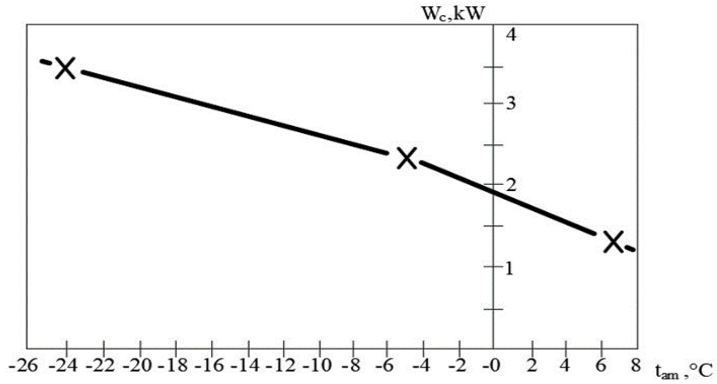

Fig. 3. Total compressor power in the function of ambient temperature (*own research*)

The results of the calculation show that the power of the compressors increases depending on the ambient temperature (from 7°C to –25°C), as shown in Figure 3. The generated amount of heat of the heat pump is from 7.5 kW to 8.0 kW. At the same time, water is heated from 35°C to 45°C in the amount of 0.2 kg/s (12 L/min).

At the same time, 8.13 kW of thermal energy is extracted from the air in the lower circuit at a temperature –25°C and 5.168 kW at a temperature –5°C. The pressure drop during compression in the compressor of the lower circuit is 849.7 kPa, and in the upper circuit is 1280 kPa (pressure drop in the compressor is limited to 1670 kPa). The cascading scheme allows for higher compression.

The results of calculating the thermodynamic parameters of the cascade scheme and the heat engineering characteristics of the air-to-water heat pump are shown in Tables 1-3 at atmospheric air temperatures: –25, –5, 7°C.

The calculation results show that the conversion factor increases as the condensing temperature decreases and the evaporator temperature increases. This also reduces the total capacity of the compressors. The capacity of the condenser increases as the temperature of the condenser (evaporator) increases.

Conclusions

The results of a numerical study confirm the efficiency of a two-stage (cascade) scheme of an air-to-water heat pump in the temperature range –25-45°C. The thermodynamic parameters and characteristics of the heat pump are determined at various air temperatures. It is shown that at an atmospheric air temperature –25°C the heat pump conversion factor is 2.47. The constant generation of heat is shown to be ensured in a cascade heat pump when the temperature of the atmospheric air changes (reduces) to –25°C.

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