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ANALYSIS OF THE EFFICIENCY OF ECO-FRIENDLY REFRIGERANTS OF A NEW GENERATION IN THE OPERATION OF HIGH TEMPERATURE HEAT PUMPS

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ABSTRACT

The article deals with high-temperature heat pump operation using refrigerant R-134a for heating DH system water. High temperature heat pumps of large capacity with two-stage centrifugal compressors can provide hot water heating up to 85 - 90 ° C and have a coefficient of heat transformation μ = 2.3 – 2.6 running by wastewaters depending on the temperature of refrigerant in the evaporator. The results of the study allow determining the most effective thermal circuits with high temperature heat pumps for district heating systems in various climatic regions. It was found that the efficiency of heat pump as a heat source depends on the type of refrigerant used as working fluid. The application of new refrigerant R-1234yf of a fourth-generation gives high efficiency to air conditioning systems. It ensures minimal negative impact on the environment because it has zero ozone-depleting potential and minimal potential of global warming. However, the preference of R-1234yf is not optimal for high temperature heat pumps due to low values of the coefficient of heat transformation. Therefore, it is necessary to develop new refrigerants that do not affect global warming and the ozone layer.

Key words: heat pump; refrigerant; ozone-depleting potential; global warming potential; coefficient of heat transformation.

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1. INTRODUCTION

Nowadays, one of the factors hindering widespread fields of heat pump systems applications is the lack of suitable working fluids, which, on the one hand,would satisfy the environmental requirements (Kyoto and Montreal agreements) and, on the other hand, would have high thermodynamic properties.

Working fluids for vapor compression heat pumps can be presented in the form of following substances (or mixtures of substances) having a number of following basic properties [1]:

- low normal (at atmospheric pressure) boiling and vaporizing temperature $t_{ИН}$, favorably contributing to the evaporation process when feeding low-grade heat (for a given ambient temperature) at pressure $P_И$ (slightly exceeding the atmospheric pressure to prevent from suckingthe air into the system of working fluid);

- low condensation pressure P_K at the required reheat temperature, which contributes to simplification of the compressor construction determined by compression ratio $\frac{P_K}{P_И}$;

- high heat of vaporization within the temperature range, providing high values of heat capacity and coefficient of heat transformation of heat pumps;

- non-toxicity, non-combustibility, explosionsafety;

- high chemical stability, inertness of chemical composition towards materials of heat pumpconstruction and lubricating oils.

2. Calculation of high temperature heat pump running by wastewaters

In the heating sector in Russia and Kazakhstan, centralized systems dominate. An alternative to themmay be high-temperature heat pumps with environmentally friendly refrigerant of the 3^{-rd} and 4^{-th} generation. As example, one can consider a heat pump with two-stage centrifugal compressor and an intermediate vessel with heat power of 17 MW [2]. Figure 1 presents a diagram of this heat pump with temperature in the evaporator – 3,5°C and the capacitor- 90°C. Refrigerant R-134a (1,1,1,2-tetrafluoroethanCH_2F-CF_3),with temperature of 101,08 °C and pressure of 40,603 bar at critical point is usedas working fluid in the heat pump. R-134a is refrigerant of the third generation; ithas no effect on the ozone layer (ODP=0) but it has influence on the greenhouse effect upon global warming potential GWP = 1340 compared to CO_2

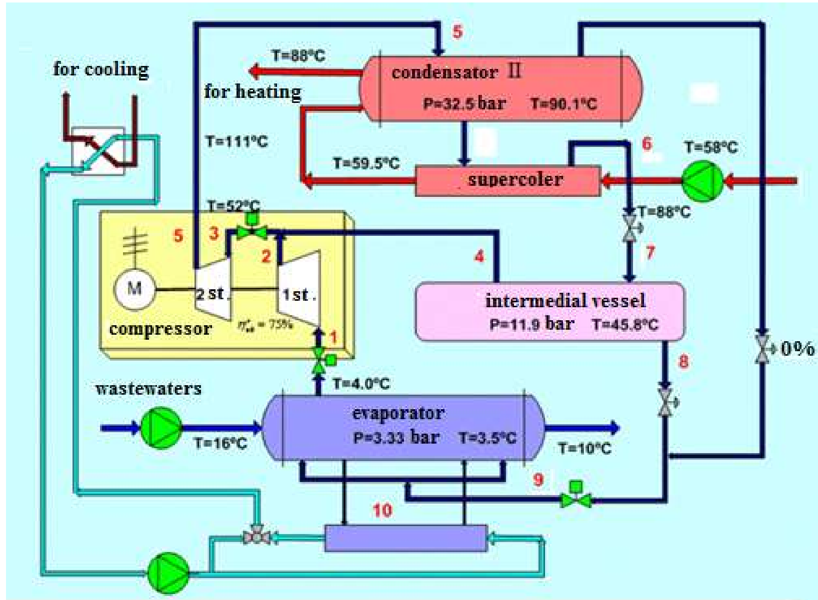

Fig.1. - Heat diagram of high temperature heat pump

1-10 – operating points of a thermodynamic process for converting heat in heat pumps; M –clutch of centrifugal compressor

The behavior of thermodynamic parameters (pressure P, temperature T, molar volume V, heat capacity c_p,, enthalpy H and entropy S) of the tested refrigerant at all points of operating process is described by Peng–Robinson's equation of state for real gas, which is one of the most accurate in the area close to the critical point [3]:

$$P = \frac{RT}{V-b} - \frac{a\alpha}{V(V+b)+b(V-b)}, (1)$$

where$P$ – pressure (MPa) T – temperature (K),V - molar volume (м^3/kmol), R =0,0083144 MPa·m^3/kmol·K [kJ/(molK)], a, b, α - coefficients determined on the basis of thermodynamic properties of a refrigerant at the critical point [3]. The coefficient of compressibility Z is included into the equation of state for real gas in the following form

$$Z = \frac{PV}{RT}. 2)$$

The valuechanges of the enthalpy $H_{P,T}$ [J/mol] and those of entropy $S_{P,T}$ [J/(mol•K)] for pressure P and temperature T during the sequential transition from one point to another in operating processare determined basedonthe thermodynamic calculations, the algorithm of which is presented in [3]. The error in determining thermodynamic parameters in accordance with mentioned algorithms compared with tabular data [4] did not exceed 2%.

When calculating a two-stage heat pump with intermediate vessel and two-stage throttling the flow rate of refrigerant in the circuits of low-$G_L$and high-pressure $G_H$is determined from energy balance for the intermediate vessel in adiabatic conditions:

$$G_H(h_4 - h_8) = G_L(h_4 - h_9), (3)$$

whereh_4, h_8 and h_9 - specific enthalpy [kJ/kg] at operating process points 4, 8 and 9 (see Fig.1).

Hence, the ratio of the use of refrigerant in circuit of high pressure G_H to that of low pressure G_L is:

$$\frac{G_H}{G_L} = \frac{(h_4 - h_9)}{(h_4 - h_8)} = \frac{1+\delta}{1} = \frac{422{,}7 - 262{,}5}{422{,}5 - 340{,}7} = 1{,}954 \text{ kJ/kg}, (4)$$

whereδ –quantity of steam in the intermediate vessel in relation to the steam of 1-ststage of the compressor, so, (1+ δ) –steam quantity in the 2-nd stage of the compressor. So, we get δ = 0.954.

The enthalpy of steam at point 3 going to the 2nd stage of high-pressure compressor is determined by the equation of mixing steam going out of the intermediate vessel (point 4) and of stage 1 of low pressure compressor (point 2) [5]:

$$h_3 = \frac{h_2 + \delta h_4}{1+\delta} = \frac{439{,}8 + 0{,}954 \cdot 422{,}7}{1{,}954} = 431{,}4 \text{ kJ/kg}. \qquad (5)$$

While determining specific enthalpies h_2 and h_5 the isentropic efficiency was taken equal to 0.75 in the calculations of compression processes in compressor stages [5].

A heat pump with a condenser and cooler can generate heat capacity Q_{cond} = 17000 kW and heat the water from$t_{гв1}$ = 58 ° C to $t_{гв2}$ = 88 ºC. The refrigerant flow rate in the upper circuit is equal to:

$$G_H = \frac{Q_{конд}}{(h_5 - h_7)} = \frac{17000}{(462{,}5 - 340{,}7)} = 139{,}6 \frac{кг}{с} = 502{,}5 \frac{т}{ч}. \qquad (6)$$

The refrigerant flow rate in the lower circuit is equal to:

$$G_L = \frac{G_H}{1+\delta} = \frac{139{,}6}{1{,}954} = 71{,}44 \frac{кг}{с} = 257{,}2 \frac{t}{h}. \qquad (7)$$

The consumption of mechanical energy to drive the 1stand 2nd stage of the compressor as a whole is:

$$N_{к1} = G_L(h_2 - h_1) = 71{,}44(439{,}8 - 403{,}5) = 2593 \text{kW}; \qquad (8)$$

$$N_{\kappa 2} = G_H(h_5 - h_3) = 139{,}6(462{,}5 - 431{,}4) = 4341\,\text{kW}; \qquad (9)$$

The electromechanical efficiency of motor at the compressor shaft is equal to $\eta_{\text{em}} = 0.98$.

The consumption of electrical energy to drive the compressor is:

$$N_э = \frac{N_{э\Sigma}}{\eta_{эм}} = \frac{6934}{0{,}98} = 7075\,\text{kW}. \qquad (10)$$

The coefficient of heat transformation is defined by the formula:

$$\mu = \frac{Q_{конд}}{N_э} = \frac{17000}{7075} = 2{,}4\,. \qquad (10)$$

Supplying cooled water to the evaporator at summer season instead of wastewater, a heat pump can produce some cold for air conditioning systems in addition to the heat. Thus,the cooling capacity will be equal to:

$$Q_{исп} = G_L(h_1 - h_{10}) = 71{,}44(403{,}5 - 262{,}5) = 10073\,\text{kW}. \qquad (11)$$

3. Analysis of efficiency of two-stage heat pumps for heating DH system water

The diagrams of thermal processes (pressure P - enthalpy H) and (temperature T – entropy S) are constructed based on the results of calculations and represented in Fig. 2 and 3.

As seen from heating water in district heating system from 50 ° C to 88 ° C by the most simple and diagrams of thermal processes, two-stage circuit withan intermediate vessel allows to provide high temperature reliable way [2,6]. The intermediate vessel acts as a separator of phase at intermediate pressure after getting a liquid-vapor mixture (point 8 in Fig.1 and diagrams in Fig.2 and 3) and superheated steam (point 2); it is the most easy way to create a two-stage system (without the risk of ingress of liquid into the second stage of the compressor from point 3). An additional increase in efficiency is given by supercooling the refrigerant in the supercooler (process 6-7), because the heat load transmitted to the consumer increases in the condenser (process 5-6) without increasing the flow rate of the refrigerant [7].

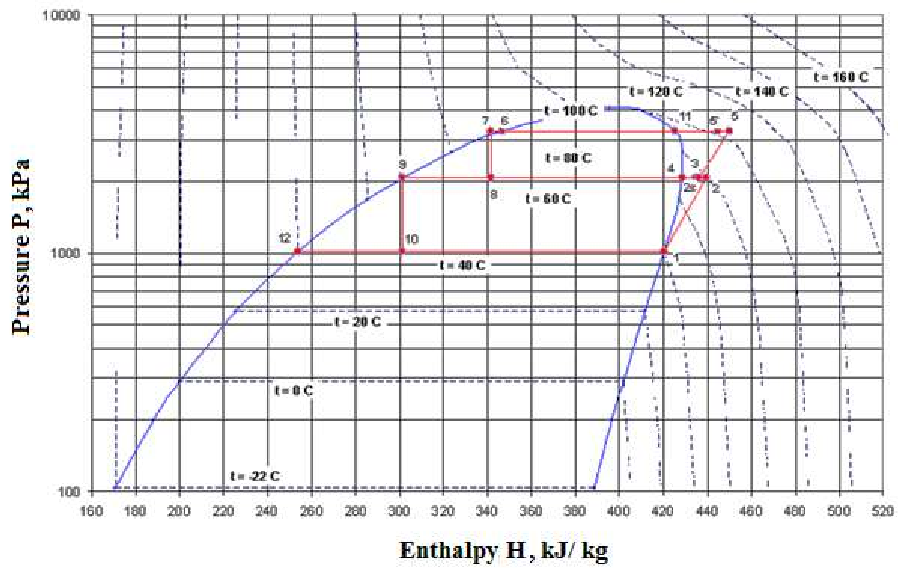

Fig.2. *P-H* - diagram of thermal processes in the operation of two-stage heat pumpbased on refrigerant R-134a running by wastewaters

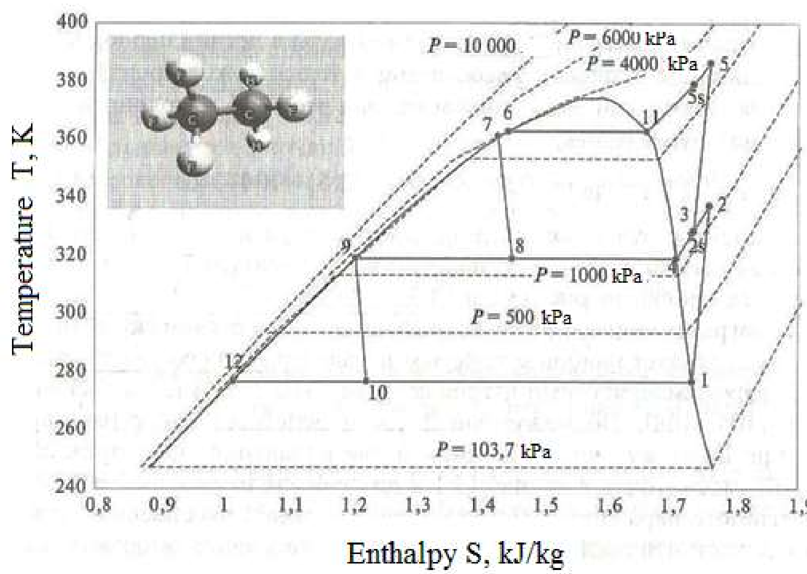

Fig. 3.*T-S*-diagram of thermal processes in the operation of two-stage heat pumpbased on refrigerant R-134a running by wastewaters

The coefficient of heat transformation μ = 2.4 turned out not to be large. However, one should take into consideration that the accounting variant of heat pump with a greater range of difference between the refrigerant temperature in the evaporator and that in the condenser wasespecially chosen

$$\Delta t = t_{cond} - t_{evap} = 90.1 - 3.5 = 86.6\ ° C, (12)$$

which is not available for most other types of heat pumps.

The complexity of heat load regulation for heat consumers during the day and season places high demands on the operation control systems of heat pumps. Modern heat pumps allow adjusting the heat load in the range of 10-100% due to the change in the position of control apparatus before the compressor stages.

High temperature heat pumps can heat water delivered to consumers located in the immediate vicinity of Central heat station (CHS), peak boiler, etc. In this case, the heat pump uses the heat of reverse water that returns to CHS from district heating system.

The calculationsof the above heat pump operating on the heat of reverse water were carried out. It was assumed that the refrigerant temperature in the condenser is equal to 90.1 ºC, and in the evaporator 40 ºC. Some water coming out of thereturn pipes of the heating system flows to the evaporator where it is cooled from 58 to 46 ° C and then returns to the CHS. The water from the return pipe of the internal circuit of heating system of local consumers is directed to the heat pump condenser and heated from 58 to 88 ° C to return to the consumers.

When thermal capacity of the condenser Q_{cond} = 17000 kW the consumption of electric energy to drive the compressor will be $N_э$ = 4050 kW, and the coefficient of heat transformation μ = 4.20.

4. Development of refrigerants for heat pumps not affecting global warming and the ozone layer

R-134a is the refrigerant of the third generation that does not affect the ozone layer (ODP = 0), but influences the greenhouse effect having global warming potential GWP = 1340 compared to CO_2.

Currently, refrigerants of the 4th generation with GWP < 150 are being developed. Refrigerant R-1234yf (CF_3-CF=CH_2) [8] is the isomer of fluoride propylene (3,3,3,2-tetrafluoropropylene) with GWP = 4 and ODP = 0, the main purpose of which is the use in automobile an air conditioning system.

According to developed EU Directive [9] since 2011, all new cars must use refrigerants with low impact on global warming. From 2017, all cars must have air conditioners and refrigerants.

It can be assumed that after the refrigerants of the 4-th generation for car air conditioning systemsone can proceed to eco-friendly refrigerants for stationary air conditioning systems, heat pumps and refrigeration machines.

R-1234yf does not provide all the needs of systems based on the principle of heat transformation. For comparison, its critical temperature 96ºC is 5 degrees lower than R-134a, and therefore, the preference of R-1234yf may not be optimalin the conditions of high temperature heat pumps.

It is necessary to look for new refrigerants and to prove that they have GWP < 150, and also they are non-toxic and non-explosive.

We need to find new refrigerants of the 4-th generation, predict and confirm their thermodynamic properties, develop technology of their production, evaluate their energy and environmental efficiency, as replacersfor the refrigerants of the 3-rd generation.

Thus, the calculations show that high temperature heat pumps of large capacity with two-stage centrifugal compressors can provide hotwater heating up to 85-90 ° C. These pumps can have a coefficient of heat transformation μ=2,3÷2,6 running by wastewaters, as well as marine and river water; μ=4,0÷5,0 running by return water depending on the temperature of the refrigerant in the evaporator.

5. Insights

1. The calculations show that high temperature heat pumps of large capacity with two-stage centrifugal compressors can provide hot water heating up to 85 - 90 ° C.These pumps can have a coefficient of heat transformation μ=2,3÷2,6 running by wastewaters, as well as marine and river water and μ=4,0÷5,0 running by return water depending on the temperature of the refrigerant in the evaporator.

2. The obtained results of energy characteristics of high temperature heat pumps allow determining the most effective thermal schemes of utilizing heat pump units in district heating systems in different climatic regions.

3. It is necessary to develop new types of refrigerants for high temperature heat pumps that do not affect global warming and the ozone layer.

6. CONCLUSION

The relevance of application of heat pumps of large capacity for district heating systems on a real example of the use of graywater, as a source of low-grade energy to operate heat pumps has been proved. The analysis of efficiency of two-stage heat pumps operation for heating water in DH systems has been carried out.

The need for new refrigerants of the 4-th generation not affecting global warming and the ozone layer has been considered.

Well-predicted and confirmed thermodynamic properties of these refrigerants will ensure to develop new technologies of their production, evaluate their energy and environmental efficiency, as replacers for the refrigerants of the 3-rd generation.

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