Increasing the Efficiency of NPP by Using the Heat Pump for Heat Supply

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ABSTRACT

Nuclear (NPP) and thermal power plants (TPP) even in condensing mode are used for heat supply during winter. The power plant heats up network water using a steam from turbine. The possibility of using a waste heat of NPP for heat supply with a heat pump (HP) is considered.

Analysis the efficiency of the HP to use a low potential heat from cooling water of steam condenser was performed. An algorithm for calculating the thermal pump was developed taking into account the properties of the refrigerant R134, which is ecologically acceptable. Two options for the location of the HP evaporator are considered: directly in the condenser and in the form of a remote location heat exchanger. The dependence of the coefficient of performance (COP) on the pressure in steam condenser is evaluated

The economic effect of the HP using due to the evaluation of the electricity production is analysed. When using HP, the heating plant does not consume a steam for heating network water. Electricity production, taking into account the power of the compressor, may increase. The additional amount of heat supply makes the use of HPs economically justified.

Keywords: *heat supply, heat pump, coefficient of performance, NPP efficiency*

1 INTRODUCTION

NPPs and TPPs, even in condensing mode, are used in winter for heat supply. The district heating plant heats the district heating water with steam extracted from the turbine. Additionally, the possibility of using NPP waste heat for heat supply will be considered. It is very attractive to use a large amount of low-potential heat that is discharged into the environment through a condenser. It is possible to increase the potential of this heat by using a heat pump (HP). The use of HP has a certain list of limitations and a set of standard solutions [1, 2] in particular for power plants [3, 4].

The aim of the work is to analyse the feasibility of using the heat transferred in the condenser for heat supply using the heat pump.

To solve this problem, it is necessary to calculate the thermal cycle of HP for the parameters that can provide NPPs. The data available in the literature are not suitable because they differ significantly from the potential of the low-temperature source used for heat supply. Thus, in [5] the values of the coefficient of performance (COP) for a ground heat exchanger with a temperature of 10 ℃ are given. The COP values vary from 2.5 to 3.5 depending on the temperature of the heating water.

2 THE SELECTION OF FREON

In the following, a heat pump is considered, in the cycle of which the working fluid is freon R134a (CF_3CFH_2). It is less harmful to the environment compared to R12 (CF_2Cl_2), the production and use of which was limited in the 90s [6]. Harmfulness of R12 is due to its molecule contains chlorine atoms, which have high ozone depletion potential (ODP), which leads to the formation of ozone holes. The R134a molecule consists only of hydrogen, fluorine and carbon atoms and does not contain chlorine atoms, so it has no effect on the ozone layer.

It should be noted that according to the second environmental indicator - GWP, which determines the global warming potential, the use of R134a might also be limited [7]. Increased absorption of infrared radiation causes an increase in global surface temperature. The impact of CO_2 is taken as the basic value of the global warming potential. In comparison, the potential for R134a is GWP = 1300 [5], i.e. much more influential than carbon dioxide for the global warming process.

3 DETERMINATION OF FREON PROPERTIES

The heat pump cycle consists of four processes - isobaric heating and cooling, compression and throttling, which occur below the critical point. For freon R134a, the critical state corresponds to the parameters: pressure 4.059 MPa and temperature 101,1 ^{o}C. These processes take place in the saturation region and partially in the state of superheated freon vapour. Thermodynamic parameters of the working fluid must be determined at each point of the cycle and this can be done in different ways - the calculation of thermodynamic properties by formulas, using tables, online calculators and even mobile applications [8-12].

An in-depth review of the set of empirical formulas published in [8] found that they are insufficient for calculating all properties of freon, including enthalpy and entropy, in the saturated and superheated state. Publication [13], which deals with the calculation of oxygen properties by a similar method and provides a listing of the calculation program, only confirms the previous conclusion. Thus, the possibility of calculating the properties of freon R134a by empirical formulas is limited and does not meet the needs of engineering calculation.

In subsequent calculations, thermodynamic parameters in the SI system were used. It should be noted that in international practice, British and other systems are also used [14], which differ not only in units of measurement, but also in the base point of reference. In the SI system, and for saturated liquid at T = 0 °C were chosen as the reference point for determining enthalpies and entropies (H = 0 and S = 0 at -40 °C for ENG units).

The choice of data for the calculation was complicated by the fact that different sources give different dependences for enthalpy and entropy on temperature [15, 16]. In one case, there is a clear error, because in the above data with increasing temperature, the enthalpy of saturated fluid decreases. Qualitative data are given in diagrams [17-19]. However, the calculations on the chart are complicated due to the large scale and the corresponding inaccuracy. Therefore, the method of interpolation of table data was chosen to calculate the properties of freon [20].

Since the table of properties at saturation shows the data in increments of one degree, the formula of linear interpolation was adopted for use. Parameters after compression in the compressor are determined by the properties of superheated freon. The table of superheated freon vapour properties has a step of 5 degrees, but checking the result of linear extrapolation in the area between the saturation parameters and the first data in the table of superheated steam gave an error of 0.03%. Therefore, to obtain the properties of superheated freon, the method of linear interpolation in the middle of the tabular data and their extrapolation in case of need to determine the properties of the parameters beyond the extreme tabular data was adopted.

The following function modules have been developed to calculate the freon properties in Visual Basic Application (VBA) for EXCEL:

at saturation

- saturation temperature and pressure ;
- enthalpy of saturated liquid and vapor, respectively,
- the entropy of saturated liquid and vapor, respectively, ;

for superheated freon vapour

- temperature or ;
- enthalpy ;
- entropy .

A few words about the algorithm for determining the enthalpy by pressure and temperature. First, two values are selected at a given pressure, which determine the properties of superheated freon by the pressure values between which the set pressure value is located. On each selected value, two temperature values are selected, between which there is a set temperature value. At these temperatures, the enthalpy values are selected and interpolation is performed for a given temperature value. According to the two values of enthalpy, interpolation is performed according to the set pressure value. The definition of other properties is similar.

The values of specific enthalpy and entropy of superheated freon vapour can also be determined by the formulas [21]:

(1)

(2)

where is the heat capacity of superheated freon vapour depending on temperature and pressure.

4 METHOD OF CALCULATION OF THE HEAT PUMP SCHEME

The Fig. 1 presents two variants of the thermal scheme with HP and the corresponding cycle in the t,-s-diagram (see Fig. 2). The Fig. 1a shows scheme when freon fed directly to the condenser IV, where it evaporates due to condensation of steam in condenser. Saturated vapour of freon is fed to the compressor I, where it is compressed to a pressure corresponding to the required temperature of mains water, which is sent to the consumer of heat supply. After compression, the freon is fed to the heat exchanger II, where it heats the district heating water (mains water) to the set temperature. At the same time, the vapour of freon condenses. The resulting condensate is throttled in the throttle valve to the required pressure, which corresponds to the evaporation temperature due to low potential heat.

If it is decided to use HP at TPP or NPP, it is not always possible to use a steam turbine condenser as a freon evaporator. In this case, the scheme Fig. 1b. is used. The entire flow of heated circulating water or its part from the turbine condenser IV is fed to the remote evaporator of freon V. Cooled circulating water is fed to the inlet of the circulating pump VI. When the HP is switched off in summer or the circulating water is partially used, it is cooled in cooling tower VII.

The HP cycle in the t, s-diagram contains: the process of cooling and condensation of freon 1-2 in the heat exchanger II; throttle process 2-3 in throttle valve III; the process of evaporation of freon 3-4 in condenser IV or evaporator V and the process of compression 4-5 in compressor I. The processes of the freon condensation and evaporation are isothermal taking into account the reduction of pressure as a result of hydraulic resistance. The compression process is isentropic, taking into account the adiabatic efficiency of the compressor. The throttling process is isenthalpic.

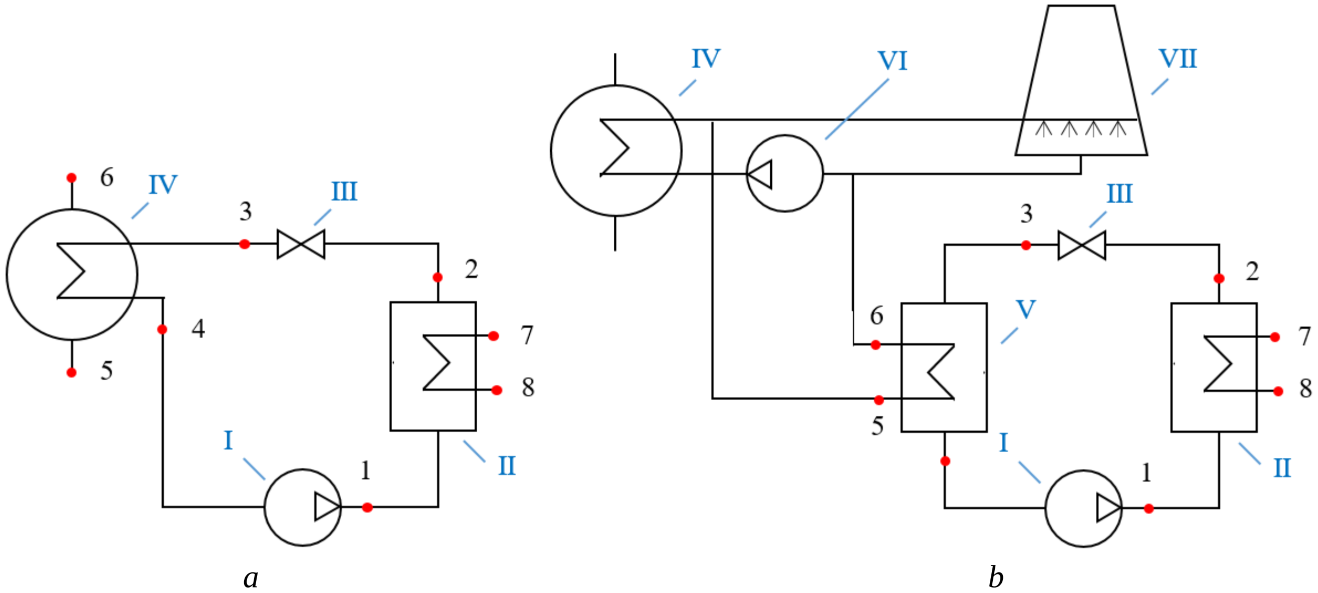

Figure 1: Schematic Diagram of the Heat Pump.

Location of the heat exchange surface of the evaporator:
a - in the condenser of the steam turbine, b - remote evaporator of freon
I - compressor; II - freon condenser; III - throttle device; IV - steam condenser;
V - remote evaporator; VI - circulating pump; VII - cooling tower.

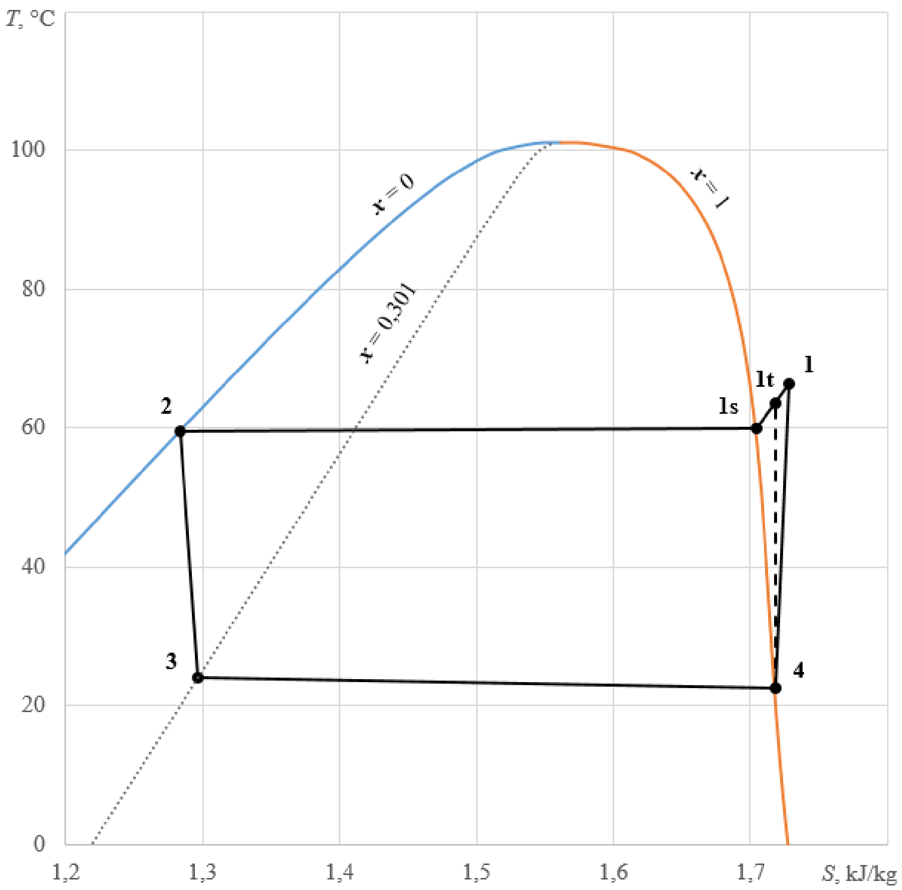

Figure 2: The t-s Diagram of Heat Pump Cycle

Consider the algorithm for calculating the thermal scheme of HP.

Input data:

The calculation is performed for the mass flow of freon . Steam condensing temperature in the turbine condenser (low potential heat sources) at a given pressure in the turbine condenser 4 kPa.

Mains water temperature for heat supply . Adiabatic compressor efficiency . The temperature difference in the condenser and in the evaporator is the same . The pressure loss in the evaporator and in the refrigerant condenser: .

Calculation of HP cycle

Freon temperature in the evaporator

. (3)

Freon condensation temperature in the condenser:

. (4)

Evaporator inlet pressure in point 3 : .
The pressure after the evaporator in point 4:

(5)

Freon pressure in the condenser in point 1: .
The pressure after the condenser in point 2:

(6)

Compressor inlet temperature: .
Enthalpy and entropy at the evaporator outlet:

Condenser outlet temperature .
Enthalpy and entropy of fluid saturation in point 2:

When throttling freon in the throttle valve, the enthalpy does not change, therefore

Determine the quality and entropy of saturated freon in point 3 [16]:

(7)

(8)

During isentropic compression in the compressor, the entropy does not change (Fig. 1) . Determine the temperature at a given pressure in the compressor, which corresponds to this entropy [8]: . Enthalpy of freon at this temperature

The temperature of the low potential heat source . The temperature of district heating water .

The defined parameters of the freon at the nodal points of the cycle are given in Table. 1.

Table 1: Properties of Freon R134a at Points of the Thermal Scheme (Fig. 1)

Points					
1	66,42	16,828	435,18	1,7279	-
1t	63,53	16,828	431,83	1,7180	-
1s	60,02	16,828	427,10	1,7039	1
2	59,49	16,628	287,09	1,2837	0
3	24,00	6,464	287,09	1,2970	0,301
4	22,42	6,164	411,25	1,7180	1

The difference of enthalpy in the compressor at isentropic compression

(9)

The actual difference of enthalpy in the compressor takes into account the efficiency of the compressor:

(10)

Enthalpy at the compressor outlet:

(11)

Power of the condenser, which is the useful power:

(12)

Compressor power

(13)

The coefficient of performance (COP) for HP cycle

(14)

The defining parameters for HP effectiveness are the parameters of the low-potential heat source and the parameters of district heating water. Let's analyse the impact of these parameters on the COP. In fig. Figure 3 shows the dependence of COP on the condensation temperature of freon vapour in the turbine condenser.

From the above results, we can conclude that with increasing pressure in the turbine condenser, the efficiency of the heat pump increases. The dependence is almost linear. Increasing the temperature of mains water reduces the efficiency of HP. At the design value of 5 kPa in the turbine condenser K-1000-5,8/50 [22], which corresponds to a temperature of 32,9 ° C, the efficiency of the heat pump is COP = 6,84. It should be noted that with increasing condensing pressure, the efficiency of the turbine decreases. This means that when using a heat pump, the pressure in the turbine condenser can to have an optimal value.

Data in the Table 2 show the dependence of COP on the mains water temperature.

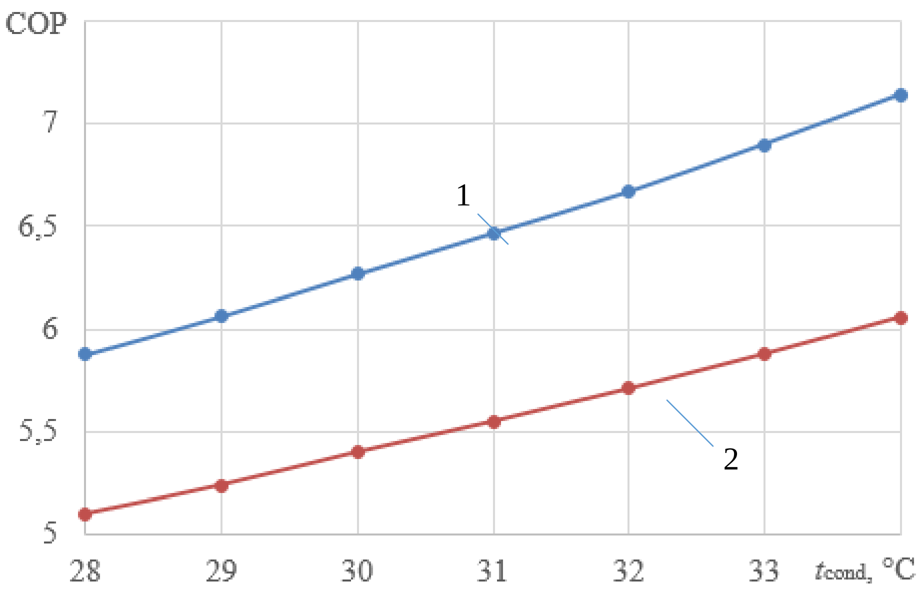

Figure 3: Dependence of COP on the Condensation Temperature of the Freon Vapour in the Condenser of the Turbine.

Mains water temperature , °C: 1 - 55, 2 - 60

Table 2: Dependence of Heat Pump Efficiency on Mains Water Temperature (at the condensation temperature of steam in the turbine condenser 29 ° C)

, °C	50	55	60	65	70
COP	7,132	6,065	5,244	4,590	3,459

The following conclusions can be drawn from the results of the calculation. With increasing temperature of mains water, which is sent to the consumer, the efficiency of HP decreases. As increases by five degrees, the COP decreases by approximately one. It can be stated that even at = 70 °C (this means that the condensation temperature of freon must be at least 75 °C) the efficiency of HP remains relatively high (COP > 3).

These data are obtained if the heat exchange surface of the evaporator is located directly in the condenser. At a remote location, the efficiency will decrease. In this case, the source of low potential heat will be cooling water. At cooling water temperature: Thus, the boiling temperature of freon will be at 16 °C. The results of the calculation relative to the scheme of Figs. 3 are given in Table. 3. The efficiency of HP is equal to COP = 4,854, i.e. will decrease by 29% compared to the option with evaporator placed in the condenser. This means that option with freon evaporator placed in the condenser of the steam turbine is better.

Table 3: Estimated indicators of the HP scheme when using a remote evaporator (Fig. 3b)

5	32,88	29	21	16	30,91	119,05	150,05

If the obtained COP values are more than 3, it is possible to draw a conclusion about the thermodynamic efficiency of using a heat pump for heat supply. Because in this case, the user will practically transfer the heat that was spent on obtaining the electricity used for the compressor work. The efficiency of conversion of thermal energy into electricity at the NPP is = 33,3 %.

Let's analyse the economic result of using a heat pump integrated into the condenser of the turbine. The meaning of the calculation is to estimate the possible profit in the production of additional electricity. When using HP, the district heating plant does not consume steam to heat the mains water and electricity generation will increase.

Determine the amount of additional electric power. The NPP with WWER-1000 unit provides heat supply in the amount of Q = 116.5 MW when operating in the district heating mode. Mains water is heated by extraction from a high-pressure cylinder at a temperature of 231,3 ºC. Calculate what electric power this heat could generate. The electrical efficiency of this cycle will be equal to [22]:

(15)

where

- thermal efficiency of the cycle of using this amount of heat –

- isentropic efficiency of the turbine, ;

- mechanical efficiency of the turbine, ;

- electrical efficiency of the generator, .

Thus, when using HP turbine will produce additional power :

(16)

During operation of HP of the same thermal power at design parameters of steam condensation (5 kPa) and mains water temperature 55 ºC the power of compressors will be

(17)

Thus, during the operation of HP will be additionally released power:

(18)

This will provide additional NPP profit during the heating period days per year and at the electricity tariff is :

(19)

The obtained result is evidence of the feasibility of further economic research, which will take into account all the costs and all the profits from the introduction of HP for heat supply from power plants.

In the future, the pressure in the condenser of the turbine should be optimized also in relation to the exergy efficiency of the power supply complex, which includes a turbine unit and a heat pump. As decreases, the power generation of the turbine unit will increase, but the COP of HP decreases and the compressor power will increase. Thus, it is necessary to check the possible range of change in winter for the presence of maximum exergetic efficiency, which takes into account the combine generation of electricity and heat.

5 CONCLUSIONS

Improving of the efficiency of the power plant (TPP or NPP) by the using of a heat with low potential from the unit condenser in a heat pump was considered. HP used for heat supply system instead of the heat supply unit, in which the mains water is heated by the steam extracted from turbine.

The calculations of the heat pump were done according to the developed technique at the condensation temperature range 28… 34 ºC and at a temperature of mains water 55 °C. Under these conditions the coefficient of performance of the heat pump are changed in range COP= 5,85…7,14. It concerns the HP scheme variant with the evaporator of freon placed directly in the condenser.

If the evaporator of freon is placed separately the coefficient of performance is decreased COP = 4,85 ( ), i.e. it is less by 29 %. When using HP for heat supply, the steam from the turbine for the mains water heating is not extracted and the electrical power of unit increases. If the heat capacity of the heat consumer is equal 116,5 MW the required power of HP compressors is 17,03 MW then additional electrical capacity of the turbine is 35,41 MW. Thus, the use of HP for heat supply using vapour condenser as a low potential energy source makes it possible to increase electricity production of the power unit by 18,38 MW.

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