Thermodynamic modeling of a heat pump unit as part of a cogeneration turbine operating in ventilation mode

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Abstract. Currently, much attention is paid to improving the efficiency of energy generating facilities through the use of secondary energy resources. This article presented a cogeneration turbine in the general energy scheme with a heat pump installation. During the winter period, due to the increased consumption of heat energy, heat-generating turbines, by closing the regulating diaphragm, are switched to full steam extraction mode from the low-pressure flow part to heat the supply water. In this case, the last stages of the turbine are transferred to the ventilation mode with the release of heat due to the vortex flows on the blades of the low pressure part. In order to obtain additional heat energy for network water, the article considers the option of using secondary energy resources with the help of a heat pump unit operating in a steam turbine plant scheme. All necessary calculations of thermodynamic modeling were carried out according to the EES heat pump, the program working as part of a cogeneration turbine, and the results were obtained of increasing the efficiency of using secondary energy resources with various refrigerants.

1. Introduction
Modern cogeneration steam turbines are designed to generate both electrical and thermal energy. However, in winter, in order to maximize the generation of thermal energy, practically all the steam from the flowing part of the low pressure cylinder to the fully closed regulating diaphragm is sent to the mains water heaters, and the turbine stages located behind the low-pressure part are forced to work in the range of ventilation modes. However, due to the reduction of steam consumption to the leakage through the gaps closed by the control diaphragm in the flowing part of low pressure, eddy currents of secondary flows are formed, which lead to the heating of the last stage working vanes to unacceptably high temperature [1]. The relevance of the topic is to increase work efficiency and reduce losses in the cycle of cogenerating turbine plants. Therefore, the development of new ways to improve the efficiency of modern cogeneration turbines without reducing their reliability, maneuverability and available power.

2. Materials and methods
At working blades of the last stage of the CHP turbine T-180 / 210-12.8, operating in the range of ventilation mode, the temperature reaches 240 °C [2]. At the same time, the reduced steam flow through the closed regulating diaphragm is 0.072. To remove heat generated as a result of ventilation losses in the stages of the low-pressure part, in practice, three main methods are usually used: a ventilation pass of steam through specially designed slots closed by a regulating diaphragm; application of a special cooling steam in the steps of the low-pressure part; injection of condensate into the space of the outlet pipe [3, 4].

However, these methods do not make it possible to use the heat released as a result of the ventilation losses, directly for heating the network water. To solve the problem, the authors propose to select the heat of superheated steam at the outlet from the last stage and use the heat pump plant to direct it to the heating of the network water (figure 1) [5, 6].
A compression type of the heat pump comprises a series-connected evaporator 1, a compressor 2, a heater 3 and a valve 4 connected to the inlet of the evaporator 1. The evaporator 1 is structurally located along the periphery of the flow part of the outlet branch pipe 5 of the low-pressure part. This arrangement makes it possible to organize the heat removal from the vortex flow of superheated steam passing in the peripheral zone of the outlet branch pipe 5. The heater 3 is connected to the regular heater 6 of the network water.

![Figure 1. Fragment of the thermal scheme of cogeneration steam turbines plant with a heat pump system.](image)

3. Results
As an example, a heat calculation was performed for the heat pump plant of the T-180 / 210-12.8 steam turbine unit built into the scheme, which operates in the CHP mode [7]. In [8], page 171 of the presented results for measuring the distribution of steam through the height of the working blade of the last stage of this turbine operating in the range of the purge ventilation mode, it is shown that the maximum relative steam flow rate ($\bar{q} = 0.8$) is concentrated on the peripheral part of the blade, which is 20% of its height, when the pressure in the condenser is $P_1=3.75$ kPa, the vapor velocity at the outlet from the peripheral zone of the blade is 158 m/s. In this case, at such a rate, the vapor flows in the annular space of the evaporator 1, which is designed as a two-row, corridor-type heat exchanger located across the direction of steam movement. In order to increase the heat transfer coefficient, the outer surface of the evaporator tubes is made in the form of annular ribs. The R-123, R-113 and R-718 refrigerants, capable of operating over a wide range of temperatures, were chosen as the working fluids in calculating the heat pump plant. Calculation of the coefficient of heat transfer $\alpha_2$ for boiling refrigerant in the cylindrical tubes of the evaporator 1 was carried out according to Lobuntsov's formula [9]:

$$\alpha_2 = b \left[ \frac{\lambda^2}{\sqrt[3]{\rho T}} \right]^{1/3} q^{2/3}$$

where $b = 0.075 \left[ 1 + 10 \left( \frac{\rho' - \rho}{\rho'} \right)^{2/3} \right]$
\( \lambda \) – is the coefficient of thermal conductivity of the refrigerant, \( \nu \) – is the coefficient of kinematic viscosity of the refrigerant, \( T_s \) – is the saturation temperature of the refrigerant, \( \rho \) and \( \rho' \) – are the densities of the liquid and vapor phases of the refrigerant, respectively. Calculation of the heat transfer coefficient \( \alpha_2 \) from the vapor to the outer surface of the evaporator tubes 1 was carried out according to the criterion equation [10]:

\[
Nu = 0.2Re^{0.65}Pr_l^{0.33} \left( \frac{Pr_l}{Pr_{ct}} \right)^{0.25}
\]

(2)

where \( Nu = \frac{a_2d_1}{\lambda_1} \)

\( d_1 \) – is the external diameter of the tube, \( \lambda_1 \) – is the thermal conductivity of the vapor,

\( Re = \frac{wd_1}{\nu_1} \)

\( w \) – the vapor velocity, \( \nu_1 \) – the kinematic viscosity of the vapor, \( Pr_l \) and \( Pr_{ct} \) – the Prandtl number at steam and wall temperatures, respectively.

The heat flux from the superheated steam vortex flow in the low-pressure part can be determined by the formula [11]:

\[
Q = Gc_p(T_{n1} - T_{n2})
\]

(3)

where \( G \) – is the steam flow rate in the low-pressure part, \( c_p \) – the heat capacity of the steam, \( T_{n1} \) and \( T_{n2} \) – are the vapor temperatures respectively at the inlet and outlet of the evaporator 1. On the other hand, this heat flux was directed to evaporate the refrigerant in the evaporator, and it is equal to

\[
Q = K(T_{n1} - T_\phi)F
\]

(4)

where \( T_\phi \) – the refrigerant temperature at the evaporator outlet, \( F \) – the area of the heat exchange surface of the evaporator 1, \( K \) – the heat transfer coefficient of the evaporator.

\[
k = \frac{1}{\frac{F_{TP}}{a_1} + \frac{F_P}{\lambda_{TP}}} \cdot \frac{1}{a_2}
\]

(5)

where \( F_{TP} \) and \( F_P \) – the areas of the outer surface of the tube and the surface of the edges of these tubes respectively, \( \delta_1 \) – the thickness of the tube wall, \( \lambda_{TP} \) – the coefficient of thermal conductivity of the tube.

Calculation of the heat pump was carried out according to the method described in [12].

The conversion coefficient \( COP \) is found from the formula

\[
COP = \frac{Q}{Ncom}
\]

(6)

where \( Ncom \) – the power of the compressor, \( \Delta Q = Q - Ncom \).

4. Discussion

A program was created in this study [13] that simulates the operation of a heat pump unit operating in the heat recovery scheme selected from an outlet nozzle of a heat and power generation turbine operating in ventilation mode.

This program allows to optimize the operation of the heat pump unit for the Rankine cycle with different refrigerants (R123, R113, R718, etc.) [14] and it also allows to consider various designs of the evaporator and condenser for the operation of the heat pump unit at different temperatures and pressures of the refrigerant. In particular, according to the created program, the conversion factors \( COP \) and \( \Delta Q \) were calculated [15][16] at the boiling temperature of the refrigerant from 40 °C to 70 °C. The results of the calculations are shown in figures 2, 3, 4, 5, 6 and 7.
Thus, the heat losses of the heat turbine T-180/210-12.8, operating in the ventilation mode, are 4 MW and can be returned for additional heating of the network water by a heat pump. At the same time, as the temperature of saturated vapor, \( T_s \) increases from 40 °C to 70 °C for R718, the heat pump conversion coefficient \( COP \) increases from 37 to 57.1. But on the other hand, the amount of heat taken from the evaporator decreases from 4470 kW to 3840 kW. A further increase in \( T_s \) can lead to an increase in the overall dimensions of the heater 3 since the return network water enters the heating circuit with a temperature of 75 °C. In this case, the temperature decreases to 5 °C.

![Figure 2. Coefficient of performance for different temperatures of the refrigerants (R123, R113 and R718).](image1.png)

![Figure 3. Relation between coefficient of performance (COP) and different temperatures of the refrigerant R718.](image2.png)
Figure 4. Relation between coefficient of performance (COP) and different temperatures of the refrigerant R123.

Figure 5. Relation between coefficient of performance (COP) and different temperatures of the refrigerant R113.

Figure 6. Relation between coefficient of performance (COP) and different temperatures of the refrigerant R718 and compressor power ($N_{com}$).
5. Conclusions
1. The use of the heat pump plant in the thermal scheme of the turbine T-180/201-12.8, operating in the ventilation mode, allows additionally to obtain commercial heat up to 4 MW.
2. As a working fluid in the heat pump plant, it is more promising to use R-718 (water) refrigerant, which has a significantly higher conversion factor than R-123 and R-113 for the same initial parameters.
3. The location of the evaporator 1 at the periphery of the exhaust pipe will practically not affect the trajectory of the steam flow in it and will not lead to a significant decrease in the efficiency of the steam-turbine unit operating in the condensation mode.
4. As a result of thermodynamic modeling, dependences $\Delta Q = f (\text{COP})$ were obtained for various refrigerants.

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