Thermodynamic analysis and optimization of low-boiling fluid parameters in a turboexpander

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Abstract. The influence of the initial parameters of a low-boiling working fluid on the thermodynamic efficiency for two turbo-expander cycles (with a heat exchanger at the outlet of the turbo-expander and without a heat exchanger) is considered. For each of the studied cycles, the dependences of the exergy efficiency on the temperature of the low-boiling working fluid before the turboexpander at a constant pressure and the dependence of the exergetic efficiency on the pressure of the low-boiling working fluid before the turbo-expander at a constant temperature were obtained. The dependences of exergy losses on the elements of the studied cycles on the parameters of a low-boiling working fluid are constructed and their analysis is carried out. For the considered schemes, the dependences of the exergy efficiency on pressure are constructed at various temperatures of the low-boiling working fluid in front of the turboexpander. An analysis of the results showed that at any temperature of a low-boiling working fluid, it is possible to determine the pressure at which the exergy efficiency of the investigated circuit will be maximum. Graphic dependencies are obtained that are characterized, from a thermodynamic point of view, by the optimal parameters of a low-boiling working fluid. Comparison of these dependences revealed that, over the entire range of studied temperatures (from 100 °C to 300 °C), a cycle with a heat exchanger at the outlet of the turboexpander has a large exergy efficiency. These graphical dependencies make it possible to determine the optimal parameters of the working fluid in the turboexpander cycle, as well as to predict the change in the exergy efficiency of the installation with changing parameters of the working fluid.

1. Introduction

Trigeneration complexes are an effective means of satisfying consumers with electricity, heat and cold, which are necessary in various industries, in the service sector, and in housing and communal services [1,2]. The energy source for such plants are: solar energy, biofuel, geothermal energy, thermal waste from enterprises, etc. Examples of schemes for such installations are discussed in [3-5]. One of the main cycles of such complexes is a turbo-expander cycle, in which low-boiling working fluids (LBWF) (with a lower boiling point than that of water) are used as a working fluid.

Large studies in the field of increasing the efficiency of cycles have been carried out for steam turbines, where water vapor is used as a working fluid. The main way to increase the efficiency of the steam cycle is to increase the parameters of the steam in front of the turbine [6].

It is known that the T-s water diagram has a saturation line similar to "wet" low boiling fluids. However, some low-boiling fluids have a "dry" or "isentropic" saturated steam line in the T-s diagram [7].

Preliminary studies have shown that it is possible to increase the efficiency of turbo-expander cycles by increasing the parameters of a low-boiling working fluid (both temperature and pressure) before the turbo-expander, but it is not clear to what extent it is advisable to increase these parameters and which parameters are considered optimal.

The aim of this work is to study the influence of the parameters of a low-boiling working fluid in front of a turboexpander on the thermodynamic efficiency of a turboexpander cycle and determining the thermodynamically optimal working fluid parameters.

2. Thermodynamic analysis technique

Evaluation of the effectiveness of the studied cycles is possible using exergy analysis, the result of which is the determination of exergy efficiency [8-10], relative units:

$$\eta_e = \frac{\sum E_E}{\sum E_C} = 1 - \frac{\sum D}{\sum E_C}$$

where $\sum E_E$ – exergy flows, the sum or difference of which determines the resulting effect; $\sum E_C$ – exergy flows whose sum or difference determines costs; $\sum D$ – cycle exergy loss.

Determine the exergy loss in the cycle is necessary to determine the exergy efficiency. These losses can be represented as the sum of the exergy losses for the elements of the studied cycle, which depend on the temperature and pressure of the low-boiling working fluid in front of the turboexpander, kJ/kg:

$$\sum D(P_3,T_3) = D_{RB}(P_3,T_3) + D_{TE}(P_3,T_3) + D_{HE}(P_3,T_3) + D_{CD}(P_3) + D_P(P_3)$$

where D_{RB} – loss of exergy in the recovery boiler; D_{TE} – loss of exergy in a turboexpander; D_{HE} – exergy loss in the heat exchanger; D_{CD} – loss of exergy in the condenser; D_P – loss of exergy in the pump.

Loss of exergy in the recovery boiler 1 (рис. 1, 2), kJ/kg:
– for circuit with heat exchanger (рис. 2):

$$D_{RB}(P_3,T_3) = E_q + E_{2'}(P_3,T_3) - E_3(P_3,T_3)$$

– for circuit without heat exchanger (рис. 1):

$$D_{RB}(P_3,T_3) = E_q + E_2(P_3) - E_3(P_3,T_3)$$

where E_q – exergy of flue gases supplied to the recovery boiler; $E_{2'}(P_3,T_3)$ – exergy of a low-boiling working fluid at the inlet to the recovery boiler (after heating in a heat exchanger); $E_2(P_3)$ – exergy of a low-boiling working fluid at the inlet to the recovery boiler (without heating in a heat exchanger); $E_3(P_3,T_3)$ – exergy of a low-boiling working fluid at the outlet of the recovery boiler.

Exergy of heat supplied to the recovery boiler, kJ/kg:

$$E_q = Q_1 \times \eta_t^{RCC} = Q_1 \times (1 - \frac{T_0}{T_{hfg}})$$

where Q_1 – amount of heat supplied to the recovery boiler with flue gas; η_t^{RCC} – thermal efficiency of a reversible Carnot cycle; T_0 – ambient temperature; T_{hfg} – temperature of a hot source of thermal energy, in this case, hot flue gas entering the recovery boiler.

Exergy losses in a Turbo Expander 2 (рис. 1, 2), kJ/kg:

$$D_{TE}(P_3,T_3) = E_3(P_3,T_3) - E_4(P_3,T_3) - L_{aTE}(P_3,T_3) \times \eta_{mechTE} \times \eta_{gen}$$

where $L_{aTE}(P_3,T_3)$ – actual work done in a turboexpander; $E_4(P_3,T_3)$ – exergy of LBVF at the exit of the turboexpander; η_{mechTE} – mechanical efficiency of a turboexpander; η_{gen} – generator efficiency.

Actual work done in an expander, kJ:

$$L_{aTE}(P_3,T_3)=(h_3(P_3,T_3)-h_4(P_3,T_3))\times G_{LBVF}$$

where G_{LBVF} – consumption of low-boiling working fluid in the cycle.

Exergy loss in the heat exchanger 6 (рис. 2), kJ/kg:

$$D_{HE}(P_3,T_3)=(E_4(P_3,T_3)-E_5(P_3))-(E_{2'}(P_3,T_3)-E_2(P_3))$$

where $E_5(P_3)$ – LBVF vapor exergy after cooling in a heat exchanger.

Loss of exergy in the condenser 4 (рис. 1, 2), kJ/kg:

$$D_{CD}(P_3)=E_5(P_3)-E_1$$

where E_1 – exergy of LBVF at the output of the condenser.

Loss of exergy in the pump 5 (рис. 1, 2), kJ/kg:

$$D_P(P_3)=L_{aP}(P_3)-(E_2(P_3)-E_1)$$

where $L_{дH}$ – actual work performed by the pump.

Actual work performed by the pump, kJ:

$$L_{aP}(P_3)=(h_2(P_3)-h_1)\times G_{LBVF}$$

3. Investigated schemes and analysis of the results

Consider the influence of the temperature of the working fluid in front of the turboexpander on the efficiency of the turbine expander cycle without a heat exchanger (Fig. 1). As a low-boiling working fluid, R236EA freon was selected, which has a "dry" saturation line characteristic, a zero ozone layer destruction potential and a global warming potential of 1370.

The following elements are presented on the diagram (Fig. 1,2): 1 – recovery boiler; 2 – turboexpander; 3 – generator; 4 – condenser; 5 – pump; 6 – heat exchanger.
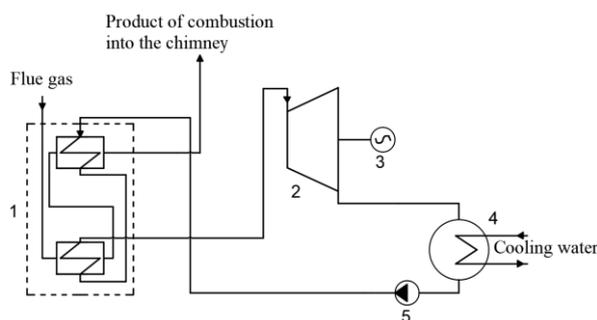

Figure 1. Scheme of a turboexpander cycle without a heat exchanger.
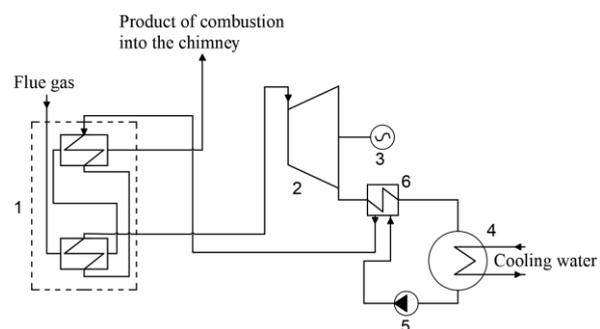
Figure 2. Scheme of a turbo-expander cycle with a heat exchanger at the outlet of the turboexpander.

The cycle of the investigated circuit for various temperatures of a low-boiling working fluid is presented in Figure 3. Cycles (Fig. 3,4) consist of the following processes: 1–2 – pressure increase of a low-boiling working fluid in the pump 5 (fig. 1,2); 2–3 heating and steam generation in the recovery boiler 1 (Fig. 1); 3 – 3a, 3 – 3b, 3 – 3c, 3 – 3d, 3 – 3e – overheating in the boiler superheater 1 (Fig. 1); 3 – 4, 3a – 4a, 3b – 4b, 3c – 4c, 3d – 4d, 3e – 4e – expansion of the low boiling fluid in a

turboexpander 2 (Fig. 1,2); 4 – 1, 4a – 1, 4b – 1, 4c – 1, 4d – 1, 4e – 1 – cooling and condensation of the vapor of the working fluid in the condenser 4 (Fig. 1,2).
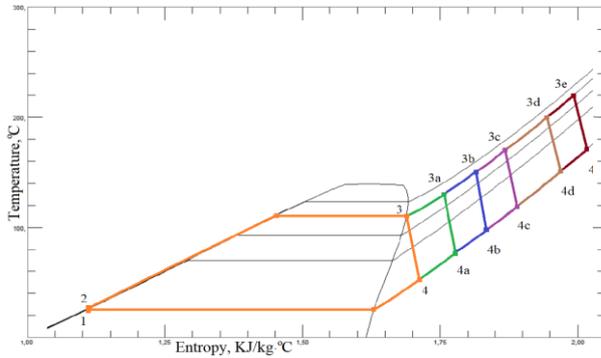

Figure 3. The cycle of a turboexpander unit at various temperatures of a low-boiling working fluid (without a heat exchanger).
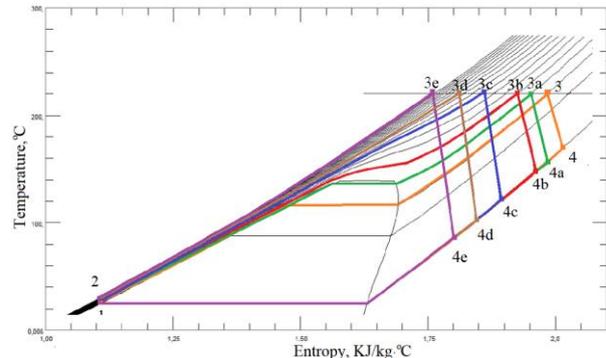
Figure 4. The cycle of the turboexpander unit at various pressures of the low-boiling working fluid (without heat exchanger).

The calculation results are presented in Figure 5 (a) in the form of the dependence of the exergy efficiency on the temperatures of the low-boiling working fluid in front of the turboexpander. From this dependence, it can be noted that an increase in temperature leads to a slight increase in exergy efficiency but only to a certain point, after which it begins to decrease. Having studied the dependence of the exergy losses on the elements of the studied cycle on the temperature of the low-boiling working fluid in front of the turboexpander (Fig. 3 (a)), it can be noted that the exergy efficiency is mainly influenced by the exergy losses in the recovery boiler (decrease with increasing temperature) and the exergy losses in the condenser (increase with increasing temperature). The steeper characteristic of the exergy losses in the condenser as compared with the exergy losses in the recovery boiler leads to the fact that the curve of the dependence of the exergy efficiency on temperature ceases to grow at a certain moment, changes its slope and begins to decrease.
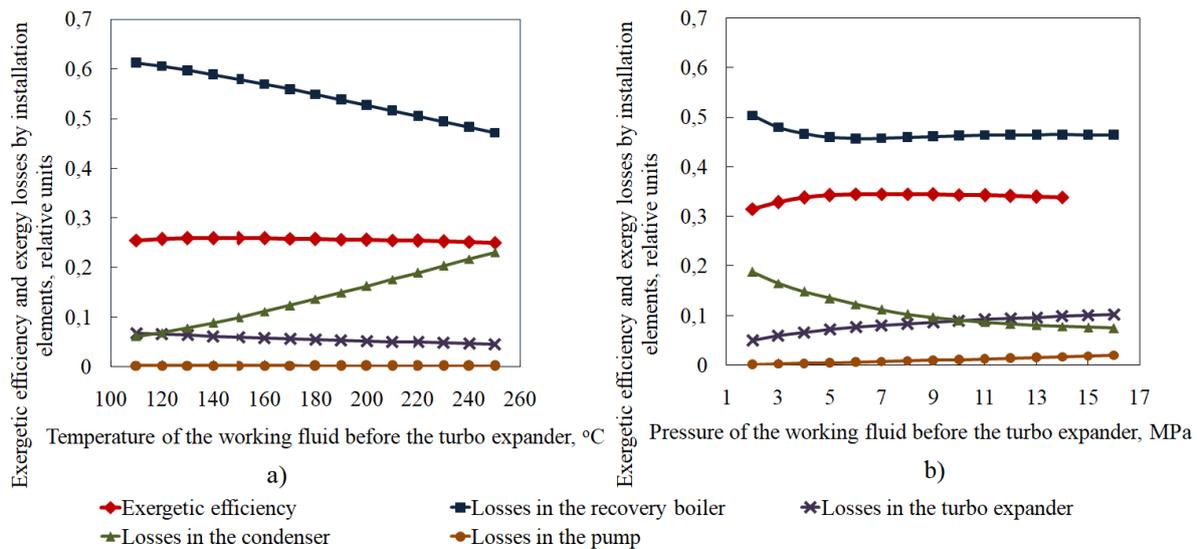

Figure 5. Dependence of exergy efficiency and exergy losses on the elements of the studied cycle on the temperature of the low-boiling working fluid in front of the turboexpander **(a)** and from the pressure of a low-boiling working fluid in front of the turboexpander **(b)** (turboexpander cycle without heat exchanger).

Let us consider the effect of the pressure of a low-boiling working fluid on the efficiency of the turbo-expander cycle (Fig. 1) when using the same working fluid (R236EA). The cycle of the investigated circuit at various pressures of the low-boiling working fluid is presented in Figure 4.

The research results are presented in Figure 5 (b) in the form of the dependence of the exergy efficiency on the pressure of the low-boiling working fluid in front of the turboexpander. From this dependence, it can be noted that an increase in pressure, as well as an increase in temperature, leads to an increase in the efficiency of the test cycle only to a certain point, after which a further increase in pressure becomes impractical. Having studied the dependence of the exergy losses on the elements of the studied cycle on the pressure of the low-boiling working fluid in front of the turboexpander (Fig. 5 (b)), it can be noted that with increasing pressure the exergy losses in the condenser decrease, the exergy losses in the turboexpander increase, and the exergy losses in the recovery boiler first decrease, then they change their direction and begin to grow slightly.

In some studies, to cool an overheated low-boiling working fluid that has left the turbine expander, it is proposed to use a heat exchanger [11], in which the low-boiling working fluid vapor is cooled to the boiling point at a given pressure, heating the condensate of the working fluid obtained in the unit's condenser.

Similar studies were carried out for a circuit with a heat exchanger at the outlet of the turboexpander (Fig. 2). Freon R236EA was adopted as a working fluid.

The cycle of the studied circuit for different temperatures of LBVF is shown in Figure 6. The cycles (Fig. 6,7) consist of the following processes: 1–2 – pressure increase of a low-boiling working fluid in the pump 5; 2 – 2', 2 – 2'a, 2 – 2'b, 2 – 2'c, 2 – 2'd – heating the working fluid in a heat exchanger 4; 2' – 3, 2'a – 3, 2'b – 3, 2'c – 3, 2'd – 3 – heating and steam generation in the recovery boiler 1; 3 – 3a, 3 – 3b, 3 – 3c, 3 – 3d – overheating of the working fluid in the superheater of the recovery boiler 1; 2'–3, 2'a – 3a, 2'b – 3b, 2'c – 3c, 2'd – 3d, 2'e – 3e – heating, vaporization and overheating in the recovery boiler 1; 3 – 4, 3a – 4a, 3b – 4b, 3c – 4c, 3d – 4d – expansion of the low boiling fluid in a turboexpander 2; 4 – 5, 4a – 5, 4b – 5, 4c – 5, 4d – 5 – refrigerant vapor cooling in a heat exchanger 6; 5 – 1 – cooling and condensing the refrigerant vapor in the condenser 4.
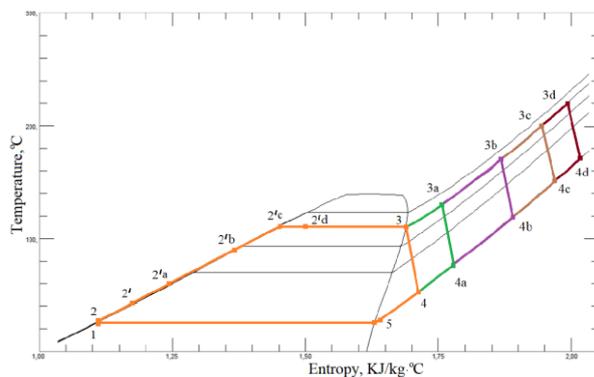

Figure 6. The cycle of a turboexpander unit at various temperatures of a low-boiling working fluid (with a heat exchanger).
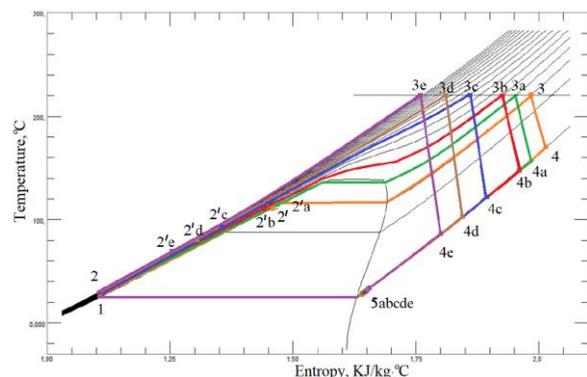
Figure 7. The cycle of the turboexpander unit at various pressures of the low-boiling working fluid (with heat exchanger).
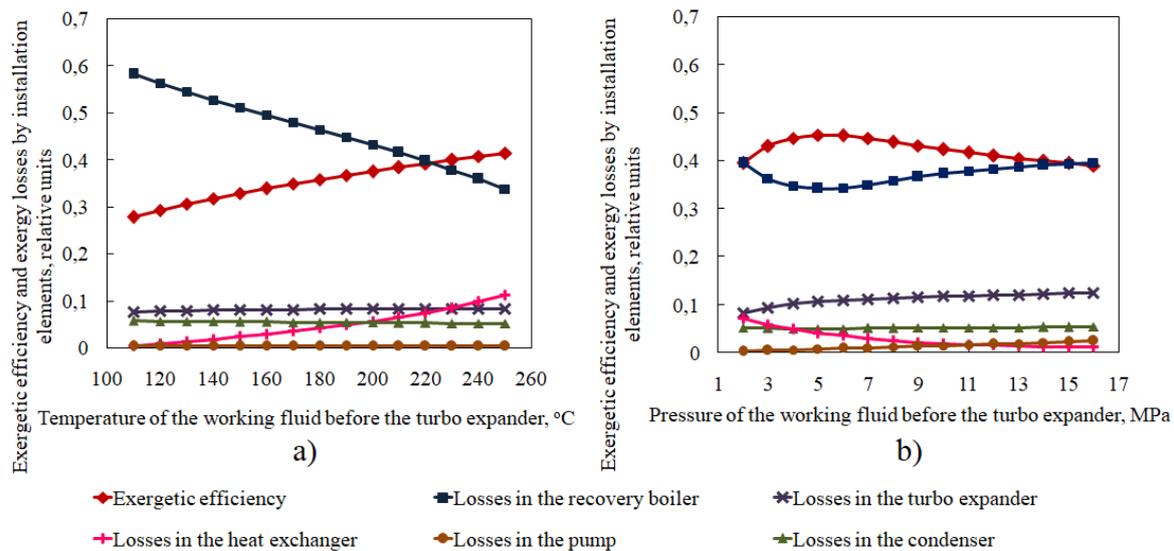
Figure 8. Dependence of exergy efficiency and exergy losses on the elements of the studied cycle on the temperature of the low-boiling working fluid in front of the turboexpander **(a)** and from the pressure of a low-boiling working fluid in front of the turboexpander **(b)** (turboexpander cycle with heat exchanger).

The calculation results are presented in Figure 8 (a) in the form of the dependence of the exergy efficiency on the temperatures of the low-boiling working fluid in front of the turboexpander. From this dependence it can be noted that, unlike the circuit without a heat exchanger, any increase in temperature leads to an increase in exergy efficiency. At the same temperature of the working fluid in front of the turboexpander (Fig. 1,2), the values of exergy efficiency are higher in the scheme with a heat exchanger. An analysis of the dependence of exergy losses on the elements of the studied cycle on the temperature of the low-boiling working fluid before the turbine expander (Fig. 8 (a)) showed that the exergy efficiency is mainly influenced by exergy losses in the heat exchanger (increase with increasing temperature) and exergy losses in the recovery boiler (with increasing temperature are decreasing).

Let us consider the effect of pressure of a low-boiling working fluid on the exergy efficiency of a turbo-expander with a heat exchanger (Fig. 2). As a low-boiling working fluid, as in previous cases, Freon R236EA was selected. The cycle of the investigated circuit is shown in Figure 7.

The research results are presented in Figure 8 (b) in the form of the dependence of the exergy efficiency on the pressure of the low-boiling working fluid in front of the turboexpander. From this dependence, it can be noted that with increasing pressure, the exergy efficiency increases only to a certain pressure, after which it changes its direction and begins to decrease. The reason for this result is a change in the shape of the exergy loss curve in the waste heat boiler (Fig. 8 (b)), which first decreases with increasing pressure, and then change directions and begin to grow.

For the studied circuits (Fig. 1,2), the dependences of the exergy efficiency on pressure are constructed at various temperatures of the low-boiling working fluid in front of the turbo expander. For each dependence of exergy efficiency, the point with the maximum value is determined. This point shows the maximum exergy efficiency that can be obtained for a given temperature of a low-boiling working fluid in front of a turboexpander. Combining these "maximum" points, we obtain a curve characterizing the optimal ratio of temperature and pressure in front of the turboexpander to obtain maximum exergy efficiency (Fig. 9 (a),(b)). Thus, by constructing a line of maximum efficiency for a particular circuit, it is possible to quickly and with a sufficient degree of accuracy determine the optimal pressure of a low-boiling working fluid in front of a turbo-expander for a given temperature, as well as predict the exergy efficiency of the installation when changing the parameters of the working fluid.
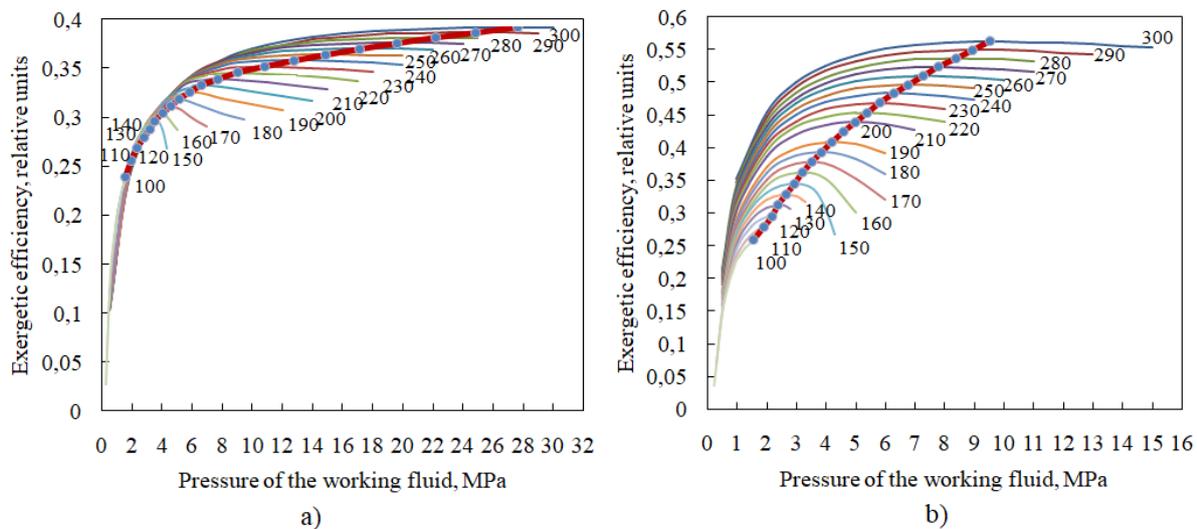
Figure 9. Dependence of exergy efficiency on pressure at various temperatures of a low-boiling working fluid for a circuit without a heat exchanger **(a)** and for a circuit with a heat exchanger **(b)**.

Comparing the lines of maximum efficiency for two circuits with and without a heat exchanger shown in Figure 9, it was found that over the entire range of studied temperatures (from 100 °C to 300 °C), a cycle with a heat exchanger at the exit of the turbine expander has a large exergy efficiency. You can also notice that at the same temperature of the working fluid, the value of the optimal pressure for a circuit without a heat exchanger is many times greater than for a circuit with a heat exchanger.

4. Conclusion

An increase in the temperature of the working fluid in front of the turboexpander without changing the pressure, as well as an increase in pressure without changing the temperature in the turbine expander without a heat exchanger, do not always lead to an increase in the exergy efficiency of the cycle. Exergy analysis showed that changing the parameters of the working fluid in front of the turboexpander primarily affects the loss of exergy in the recovery boiler and the loss of exergy in the condenser.

In turn, increasing the temperature of the working fluid without changing the pressure in the turboexpander cycle with a heat exchanger leads to an increase in exergy efficiency over the entire range of temperatures studied (from 100 °C to 300 °C), however, increasing pressure without changing the temperature does not always lead to a positive effect. Changing the parameters of the working fluid in front of the turboexpander has a major effect on the loss of exergy in the recovery boiler and the loss of exergy in the heat exchanger.

It was revealed that at any temperature of a low-boiling working fluid, it is possible to determine the pressure at which the exergy efficiency of the circuits under study will be maximum. The lines of maximum exergetic efficiency were constructed for the working fluid under study, which make it possible to determine the optimal pressure of the low-boiling working fluid in front of the turbo-expander for a given temperature, as well as to predict the exergetic efficiency of the installation when the parameters of the working fluid are changed. Under the same initial conditions, a cycle with a heat exchanger at the outlet of the turboexpander has a large exergy efficiency.

5. References
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