Thermodynamic analysis of the cycle efficiency of the utilization units for gas transmission systems

I S Antanenkova\textsuperscript{1,2} and A A Vetrenko\textsuperscript{1,3}

\textsuperscript{1}National Research University «Moscow Power Engineering Institute», Russia
\textsuperscript{2}antanenkovais@mail.ru, \textsuperscript{3}alvetrenko@mail.ru

Abstract. A schematic solution is proposed for organizing an auxiliary energy module that converts the thermal energy of the exhaust gases of the GTU, which serves as a drive for the gas compressor unit in a thermal power circuit on a low-boiling working fluid, into electrical energy (based on the technology of the organic Rankine cycle). The results of thermodynamic analysis of cycle efficiency on different working fluids (Rc318, R744, R245fa, R601, R290) are presented. The influence of the operating parameters of the cycle on the energy efficiency of the unit is studied.

Gas compression at compressor stations (CS) is the most energy-intense technological process in gas transportation systems \cite{1}, therefore, the solution of the problem of reducing energy costs for gas pumping, first of all, should be aimed at improving the efficiency of the "heart" of the CS - gas compressor unit (GCU). The number of installed GPA in Russia exceeds 4000 PCs., 85\% of which are driven by gas turbine units (GTU) \cite{2}. Given the large fleet of GCU in Russia, the heat loss with the exhaust gases of the gas turbine drive is huge (according to various estimates, up to 80 GW), which makes the task economically feasible.

The analysis of the technological parameters of such plants allows judging on applicability of the technology of the organic Rankine cycle (ORC) to improve energy efficiency.

One of the most common GPA in Russia is GPA-16, which can be driven by various GTU (table. 1).

\begin{table}
\centering
\begin{tabular}{|c|c|c|c|}
\hline
GCU & GTU & Efficiency coefficient, (%) & The flow rate of exhaust gases, (kg/s) & Exhaust gas temperature, (°C) \\
\hline
GCU-16 “Volga” & NK-16ST & 29.0 & 103.1 & 378 \\
GCU-16 “Volga” & NK-38ST & 38.0 & 102.0 & 445 \\
GCU-16 “Volga” & NK-16-18ST & 31.0 & 106.0 & 427 \\
GCU-16 “Ladoga” & T16 & 36.0 & 54.6 & 492 \\
GCU-16 P “Ufa” & AL-31ST & 36.5 & 64.5 & 490 \\
\hline
\end{tabular}
\caption{The main technical characteristics of gas turbines for GCU-16.}
\end{table}
As an example to analyze the effectiveness of the proposed solution, GTU series NK-16ST was employed. The choice of this unit was due to its wide prevalence in the largest gas transportation system of Russia Gazprom PAO [3].

The proposed thermal scheme of the utilization circuit of the ORC module (figure 1) contains a turbine with an electric generator, in which the working substance (freon) is expanded, being preheated in a heat exchanger-utilizer of exhaust gases of GTU (freon steam generator). The exhaust steam of freon in the turbine still has a relatively high potential (temperature), for the efficient use of which a recuperator is provided in the scheme. Downstream the heat exchanger, the freon goes to the condenser where it is cooled and condenses due to the transfer of heat to atmospheric air. After the condenser, the freon in the liquid state is pumped sequentially into the recuperator and the steam generator. The supercritical thermodynamic cycle realized by such a unit is presented in figure 2.

\textbf{Figure 1.} Principle heat scheme of the ORC-module for utilizing the heat of exhaust gases of the GTU.

\textbf{Figure 2.} Supercritical cycle of the ORC module on a low-boiling working fluid (taking into account irreversible losses in the elements of the unit).

ORC implementation is possible on various working fluids (WF). To analyze the energy efficiency of the utilization plant on the exhaust of the gas turbine drive GCU, the following WF were selected: carbon dioxide (R744); octafluorocyclobutane (Rc318); 1.1.1.3.3-pentafluoropropane (R245fa), pentane (R601) and propane (R290). The thermodynamic characteristics of these substances are shown in table 2.

\textbf{Table 2.} Fluid Information

| Characteristic             | Rc318 | R744 | R245fa | R601 | R290 |
|---------------------------|-------|------|--------|------|------|
| Chemical formula          | C4F8  | CO2  | C3H3F5 | C5H12| C3H8 |
| Molar mass, (kg/kmol)     | 200.03| 44.01| 134.05 | 72.15| 44.10|
| Critical temperature, (K) | 388.38| 304.13| 427.16 | 469.70| 369.89|
Critical pressure, MPa | 2.78 | 7.38 | 3.65 | 3.37 | 4.25  
Boiling point $t_s$, (K) (by $p = 0.1013 \text{ MPa}$) | 267.17 | 220.09 | 288.28 | 309.21 | 231.03  
Heat of evaporation at $t_s$, (kJ/kg) | 116.75 | 344.77 | 196.05 | 357.56 | 425.60  
Triple point temperature, (K) | 233.35 | 216.59 | 171.05 | 143.47 | 85.53  
Ozone-depleting potential (ODP) | 0 | 0 | 0 | 0 | 0  
Global warming potential (GWP) | 9100 | 1 | 930 | 11 | 3  

The following values were taken as initial data for the calculation of the thermodynamic cycle of the ORC-module:
- temperature of the exhaust gases of GTU before FSG: 380 °C;
- temperature of the exhaust gases of GTU after FSG: 150 °C;
- freon pressure before the turbine: from 6 to 20 MPa;
- condenser temperature: 25 °C;
- internal relative efficiency of the turbine $\eta_T^i$: 75%;
- internal relative efficiency of the pump $\eta_P^i$: 50%;
- undercooling of freon in the recuperator: 10 °C;
- underheating of freon in the FSG: 20 °C;
- minimum temperature difference at the condenser: 10 °C.

For determining thermophysical properties of the considered WF references, proper experimental data obtained in MPEI and the base of thermophysical properties of substances NIST REFPROP 8.0 were used.

The internal efficiency of the ORC-module was determined by the following dependence:

$$\eta_i = \frac{(h_1 - h_2) - (h_5 - h_4')}{h_1 - h_6},$$

where $h_1$, $h_2$, $h_4'$, $h_5$, $h_6$ is the WF enthalpy at characteristic points of the cycle (figure 1).

The results of calculation of efficiency coefficient of the ORC-module at different pressures at the inlet to the freon turbine for the substances are shown in figure 3.

Figure 3. Dependence of the ORC-module efficiency on freon pressure at inlet to the turbine.
Dashed lines in figure 3 indicate the calculation results taking into account additional underheating of the WF in the presence of an intermediate circuit between the exhaust gases of the GTU and the combustible WF of the utilization plant (the traditional scheme of the ORC with a thermal oil circuit).

According to the criterion of the maximum possible efficiency of the cycle, the optimal values of freon pressure at the turbine inlet for most of the substances under consideration have been determined (24.2% at 12 MPa at R245fa, 22.4% at 12 MPa at R601, 18.2% at 16 MPa at R290, 17.1% at 12 MPa at R318). When operating the cycle with R744, the predicted optimum is obtained at too high pressure (above 30 MPa), which will, ceteris paribus, increase capital costs of the unit and requirements for its reliability and operational safety.

Maximum efficiency of the unit is received when using the R245fa as a WF. Thus, it is advisable to further study the effect of the operating parameters of the cycle on the energy efficiency of the ORC-module on R245fa at a freon pressure of 12 MPa at the turbine inlet.

In addition to the initial pressure, the thermodynamic efficiency of the GTU cycle with the ORC-module is significantly affected by the temperature of the atmospheric air. First, atmospheric air cools the condenser of the utilization power plant, and secondly, air is used as a working substance in the cycle of GTU. For figure 4 the nature of the dependence of the temperature and exhaust gas flow rate, as well as the power of the considered GTU series NK-16ST when the outside air temperature changes is presented (the presented dependences are obtained on the basis of the passport data of the unit).

![Figure 4](image)

**Figure 4.** The parameters of the gas turbine unit at different temperatures of atmospheric air:
1 - Relative turbine exhaust gas temperature \( T/T_{\text{max}} \) (\( T_{\text{max}}=401.2 \degree \text{C} \));
2 - Relative exhaust flow rate \( G/G_{\text{max}} \) (\( G_{\text{max}}=115.1 \text{ kg/s} \));
3 - Relative Power of GTU \( N/N_{\text{max}} \) (\( N_{\text{max}}=19200 \text{ kW} \)).
Figure 5. Efficiency coefficient $\eta_t$ (1) and the ratio of power $N_{ORC}$ to power GTU $N_{ORC}/N_{GTU}$ (2) at various temperatures of atmospheric air.

In figure 5 results of calculation of efficiency coefficient (1) and power $N_{ORC}$ (2) of the ORC-module on R245fa related to the corresponding power of GTU $N_{GTU}$ (figure 4) at temperature change of atmospheric air are presented.

The output power of the ORC module, kW, was determined as follows:

$$N_{ORC} = N_{FT} - N_p,$$

where $N_{FT} = m_{WF} \cdot (h_1 - h_2)$ is the freon turbine power, kW; $N_p = m_{WF} \cdot (h_3 - h_4)$ is the pump power, kW; $m_{WF}$ is the the flow rate of the WF determined from the heat balance equation of the FSG, kg/s.

Contrary to expectations, the cycle efficiency of the ORC-module increases from 23 to 27.5% with the increase of atmospheric air temperature from -50 to -6 °C. The increase in the efficiency of the unit is due to the compensation of heat losses in the condenser by increasing the degree of regeneration in the ORC (the share of heat transferred in the heat exchanger inside the cycle) with an increase in the temperature of the WF at the outlet from the freon turbine. A further increase in the temperature of the atmospheric air (from -6 to 40 °C) leads to the fact that the degree of regeneration in the ORC reaches its maximum at a constant temperature of the gases at the outlet of the freon steam generator (150 °C). Consequently, a further increase in the exhaust gas temperature of the GTU leads to an increase in the heat supplied to the steam generator and a decrease in the available heat transfer of the freon turbine with an increase in the pressure in the condenser, that is the efficiency of the ORC-module decreases up to 15.6%.

The dependence of the power of the ORC-module $N_{ORC}$ in the considered interval of temperatures of atmospheric air is similar to $\eta_t$. In the temperature range from -50 to -6 °C $N_{ORC}$ on R245fa is increasing from 2.1 to 5.4 MW; and from -6 to 25 °C it is reduced from 5.4 to 4.9 MW. With an increase in air temperature from 25 to 40 °C, the ratio of power $N_{ORC}/N_{GTU}$ practically does not change.

The results obtained during the study allow drawing the following conclusions:

1. The ORC technology is applicable to increase the energy efficiency of modern gas turbine-driven GCU.
2. The thermodynamic efficiency of the ORC and the optimal parameters of the plant differ significantly when using different working substances.
3. The power and energy efficiency of the ORC-module on the exhaust of GTU on GCU significantly depends on the temperature of the atmospheric air.
4. For the parameters of the considered GTU series NK-16ST, the most effective mode of operation of the ORC-module when operating on R245fa with an optimal freon pressure at the inlet to the turbine is 12 MPa; at an atmospheric air temperature of -6 °C, the efficiency of such a unit is 27.5%, and the electric power is 5.4 MW.

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