Self-contained gas turbine unite with turbine expander drive of the compressor for the gas distribution station

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Abstract

This paper deals with the gas turbine installation with turbine expander drive of the compressor. It is shown that the offered installation has greater power capacity in comparison with the existent power turbo expanders and this unit can work in the regardless of steam turbine mode.

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1. Introduction

In connection with uninterrupted growth of the energy consumption and limited sources of hydrocarbon fuel, the currency of waste energy utilization grows. Among the waste energy utilization problems, the important problem is excessive gas pressure utilization on the gas-distribution stations and in the first place on the gas-distribution station of the thermal power plant, where the natural gas flow is very big.

At present this problem is solving by application of the turbine expanders used the potential energy of natural gas and directly connected with electric generator.

However the opinion about the reasonability of the usage of such units on the thermal power plants is argued.

It should be taken into consideration that the gas directed to the consumer with lower pressure then in the gas-main pipeline must have near the same temperature, which was at the inlet of the turbo expander. Therefore the gas must be heated behind or after the turbo expander and the quantity of the heat input must be equivalent to the additional generated power.

From this point of view the turbo expander unit is the transformer of the thermal energy into electric one. However if this installation is included in thermal scheme of the thermal power plant and for the gas heating the steam bled form the steam turbine is used the steam turbine power capacity decrease induced by the steam extraction appears less than the power capacity generated by the turbo expander.

This research deals with the methodology of the additional power capacity estimation which generated by the turbo expander and with the dependence of power capacity from the gas expansion ratio. These results are compared with results which can be reaches in case of the

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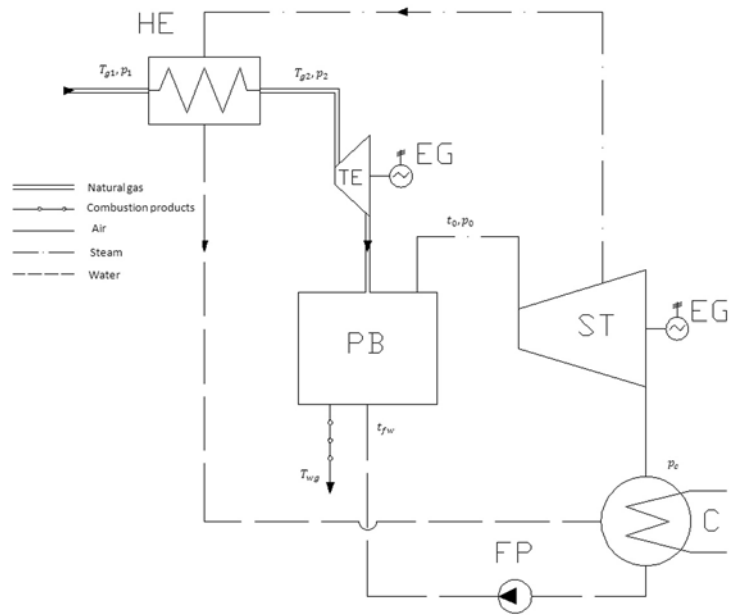*Corresponding author. Tel.: +79 037 887 237, e-mail: R-Andrey2007@yandex.ru.

Fig. 1. Thermal diagram of the turbo expander installation: HE – heat exchanger for gas heating; TE – turbo expander; EG – electric generator; PB – power boiler; ST – steam turbine; C – condenser; FP – feed pump

application of gas turbine unit with the turbo expander drive of the compressor on the thermal power plant.

Naturally such gas turbine units which aren't connected with the objects which consume the gas behind turbo expander can operate at the various gas expansion ratios with the high electric efficiency level.

2. The estimation of the efficiency of the turbo expanders application in the thermal power plants thermal schemes

At present the simplest scheme of the natural gas potential energy utilization is used on the thermal power plants. This scheme is presented in Fig. 1.

In this scheme the high-pressure gas goes to the power boiler through the turbo expander directly connected with electric generator. In the expand process the gas pressure and temperature decrease. So far as the gas entered to the boiler must have definite temperature and pressure the heat exchanger is installed in front of turbo expander where the gas is heated by the bled steam of turbo installation. The steam condensate generated in the heat exchanger goes back to the cycle and enter to the boiler.

The steam extraction from the steam turbo installation cycle leads to decreasing of the steam turbine power. The effectiveness of the turbo expander application will be determined by the difference value between the turbo expander power capacity and decrease of steam turbine power capacity.

The useful electric power of the turbo expander is determined by the following expression:

$$N_{te} = B \cdot c_{p\_ch4} \cdot T_{g2} \cdot (1 - \delta^{-m_{ch4}}) \cdot \eta_{te} \cdot \eta_m \cdot \eta_{eg}, \tag{1}$$

where B – the natural gas flow through turbo expander; c_{pm} – the average gas heat capacity in the expansion process; T_{g2} – the gas temperature in front of turbo expander; δ – the gas expansion ration in the turbo expander; η_{te} – the turbo expander internal efficiency factor; η_m – mechanical efficiency of the turbo expander; η_{eg} – efficiency of the electric generator; m_{ch4} – is defined by the adiabatic index of the natural gas by the expression:

$$m_{ch4} = \frac{k-1}{k}.$$

It is supposed that the required natural gas temperature in front the boiler equals to the natural gas temperature in front of the heat exchanger $t_{g2} = t_{g1}$. Then the gas temperature in front of turbo expander defines by the following expression:

$$T_{g2} = \frac{T_{g1}}{1 - (1 - \delta^{-m_{ch4}}) \cdot \eta_{te}}. \tag{2}$$

The substitution of the expression (2) in to (1) takes us the expression for the turbo expander power capacity:

$$N_{te} = B \cdot c_{p\_ch4} \cdot T_{g1} \cdot F(\delta) \cdot \eta_m \cdot \eta_{eg}, \tag{3}$$

where $F(\delta)$ is the gas expansion ratio function in the turboexpander:

$$F(\delta) = \frac{(1 - \delta^{-m_{ch4}})}{1 - (1 - \delta^{-m_{ch4}})}. \tag{4}$$

Let's find the steam turbine power capacity decrease resulted from the steam extraction for the gas heating. Let thinking that the steam turbine internal efficiency doesn't change because of little steam extraction. In this case the steam turbine power capacity decrease may be estimated by the following expression:

$$\Delta N_{st} = G_{extr} \cdot (h_{extr} - h_c), \tag{5}$$

where G_{extr} – the flow of extracted steam, h_{extr} – the extracted steam enthalpy, h_c – the steam enthalpy at the end of expansion process.

After the proper transformations the expression (5) may be presented it the following view:

$$\Delta N_{st} = c_{p\_ch4} \cdot B \cdot T_{g1} \cdot \eta_{te} \cdot F(\delta) \cdot Y(p_n, t_n, \eta_{oi}, \delta), \tag{6}$$

where $Y(p_n, t_n, \eta_{oi}, \delta) = \frac{h_{extr} - h_c}{h_{extr} - h_s}$.

In the issue the additional power capacity which may be obtained in case of application of the turbo expander scheme presented on the figure 1 is estimated by the following expression:

$$N_1 = N_{te} - \Delta N_{st} = B \cdot c_{p\_ch4} \cdot T_{g1} \cdot F(\delta) \cdot \eta_{te} \cdot \eta_m \cdot \eta_{eg} \cdot \left(1 - \frac{Y(p_n, t_n, \eta_{oi}, \delta)}{\eta_{eg} \cdot \eta_m}\right).$$

For $m_{ch4} = 0.218$ and $\eta_{te} = 84\%$, the function $F(\delta)$ will be as it is presented in Fig. 2 (curve 1).

It is required to know the expansion process in the steam turbine for the function

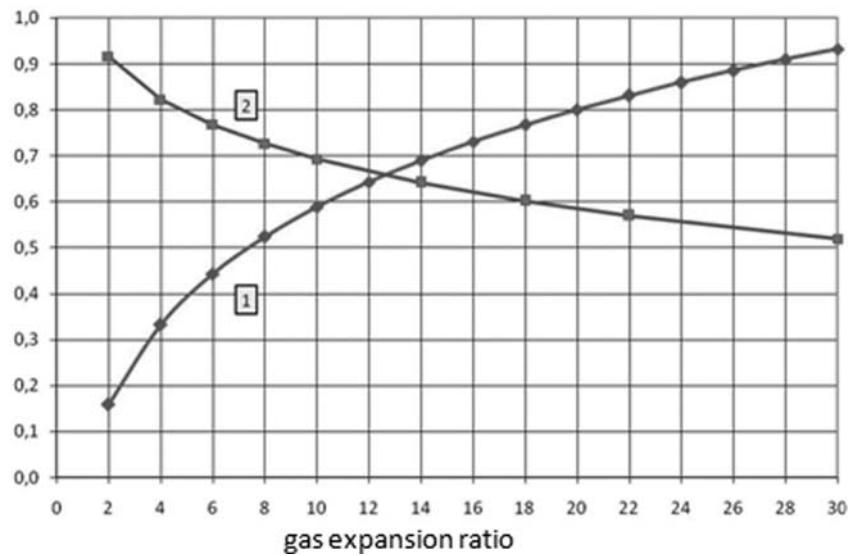$$1 - \frac{Y(p_n, t_n, \eta_{oi}, \delta)}{\eta_{eg} \cdot \eta_m}$$

Fig. 2. $F(\delta)$ and $1 - \frac{Y(p_n, t_n, \eta_{oi}, \delta)}{\eta_{eg} \cdot \eta_m}$ as a function of the gas expansion ratio in the turbo expander δ

estimation. For example, let us consider this process for the steam turbine K-300-240 of LMZ. Let's take the subcooling δt in the heat exchanger equals to 7 K and the natural gas heat capacity c_{p_ch4} equals to 2.39 kJ/kg.

Then the function $1 - \frac{Y(p_n, t_n, \eta_{oi}, \delta)}{\eta_{eg} \cdot \eta_m}$ will take on a form as it is presented in Fig. 2 (curve 1).

Deficient for the calculation values are the following: the mechanical and electric efficiency – $\eta_{eg} = 99\,\%$, $\eta_m = 99\,\%$, the gas temperature in the gas-main pipeline $T_{g1} = T_{g3} = 288.15$ K.

The results of calculations of additional power capacity N_1 which generated at the expense of turbo expander application for the selected steam turbine are presented in Fig. 3 and in Table 1.

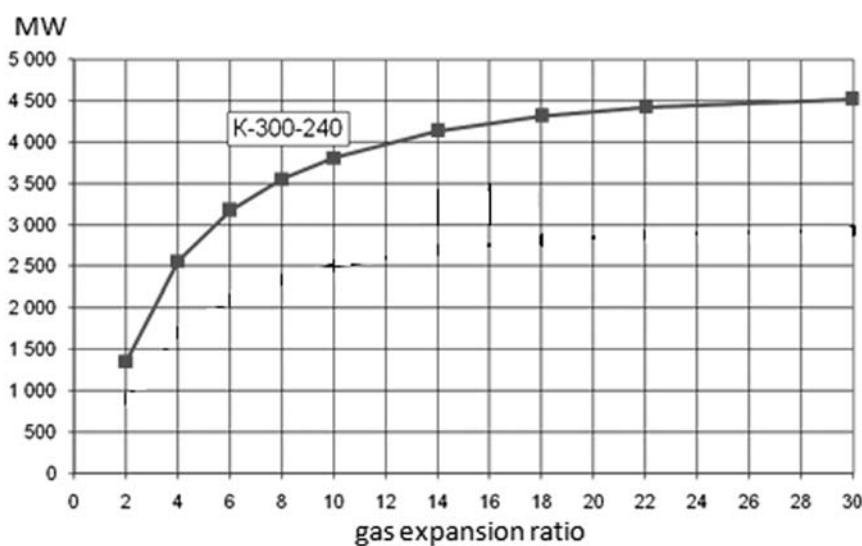
Fig. 3. The addition power capacity of the power generating unit induced by the turbo expander application as a function of the gas expansion ratio δ

Table 1. The results of calculations

No.		parameter	value
1	G_2	Gas flow, kg/s	16.45
2	δ	Gas expansion ratio (from gas-main pipeline to the boiler)	8
3	t_{g1}	The gas temperature in the gas-main pipeline, °C	15
4	t_{g2}	The gas temperature in front of turbo expander, °C	141.9
5	p_{extr}	The pressure in the steam extraction manifold, MPa	0.53
6	t_s	Saturation temperature at the steam extraction pressure, °C	154
7	G_{extr}	Steam flow for the gas heater, kg/s	2.074
8	N_{te}	Turbo expander power capacity, kW	4 893.5
9	ΔN	The steam turbine power capacity decrease because of the steam extraction for the gas heating, kW	1 391.6
10	N_{usf}	The useful power capacity of the unit, kW	3 501.8
11	η_{te}	The turbo expander efficiency, N_{te}/Q	0.98
12	η_{usf}	The efficiency of addition power generation for the whole power generation unit, N_{usf}/Q	0.7

The results of the calculations have shown we can obtain the additional power capacity from 1 400 to 4 500 kW subject to the gas expansion ratio in the turbo expander for the steam turbine K-300-240. So far as the gas expansion ration in the turbo expander is determined by the gas pressure in the gas-main pipeline and by the gas pressure in front of the boiler burners then for the conventional power boilers the value of the gas expansion ratio δ changes in the short range from 6 to 8. Then in according to the curve in Fig. 3 the additional power capacity changes from 3 200 kW to 3 500 kW.

The power capacity of the turbo expander unit subject to the gas expansion ratio lies in the range from 1 500 kW to 8 800 kW. The steam turbine power decrease does not exceed 4 000 kW.

The data in Table 1 show that for the gas expansion ratio in the turbo expander δ equal to 8 in case of the application of the turbo expander in the gas system of the power generation unit K-300-240 LMZ the power capacity of the turbo expander N_{te} is equal to 4 893kW and additional power capacity N_{usf} reaches 3 500 kW.

3. Recovery gas turbine unit with turbo expander drive of the compressor for gas-distribution stations

The executed estimations of the turbo expanders application in the power plants thermal schemes are shown that we can obtain the power capacity increase of power generation unit

K-300-240 by 1 %. This increment may be considerably increase if we will use self-contained gas turbine installation with the turbo expander drive of the compressor on the gas-distribution stations. The scheme of such installation is considered in the works [3, 4] and presented in Fig. 4.
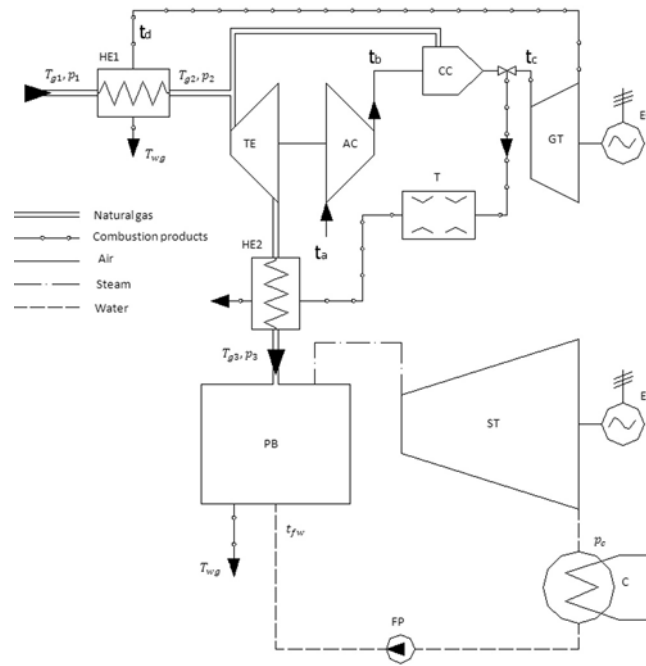

Fig. 4. Thermal scheme of the gas turbine unit with turbo expander drive of the compressor in the power boiler gas feed system: HE – heat exchanger for the gas heating; HE2 – heat exchanger for the gas heating the varying duty; TE – turbo expander; AC – air compressor; GT – gas turbine; EG – electric generator; PB – power boiler; ST – steam turbine; C – condenser; FP – feed pump, T– throttle

In this scheme the high pressure gas goes through the heat exchanger HE1 where it is temperature grows up to $100\,^\circ$C. Such temperature level ensures required temperature in front of the power boiler. Behind the heat exchanger part of the gas flow goes to the gas turbine combustion chamber and the other part enter to the turbo expander. After the expansion in the turbo expander gas enter to the power boiler as in the conventional scheme. The turbo expander drives the air compressor which supplies the combustion chamber with compressed air. The fuel combustion is realized in the combustion chamber. The combustion products expands in the gas turbine which generate the useful electric power and the waste gases of the gas turbine is utilized in the heat exchanger HE1 where the natural gas is heated up to required temperature.

The whole turbo expander power capacity is used for the compressor rotation:

$$N_c = N_{te}.$$

Using this condition and taking the air compression ratio value, we can estimate the air flow through the compressor by the following expression:

$$G_{air} = \frac{N_c}{c_{p\_air} \cdot T_a \cdot \frac{(\varepsilon^{m_{air}} - 1)}{\eta_c}}, \qquad (7)$$

where ε – air compression ratio, c_{p_air} – average air heat capacity in the compression process, T_a – air temperature in front of the compressor, η_c – compressor internal efficiency.

The air temperature behind the compressor is estimated by the following expression:

$$T_b = T_a \cdot \left(1 + \frac{(\varepsilon^{m_{air}} - 1)}{\eta_c} \right).$$

The combustion products temperature must be so that the this products may heat the natural gas in the heat exchange HE1 up to the required temperature.

Let us estimate the required combustion product temperature in front of the heat exchanger from the heat balance equation:

$$T_d = T_{wg} + \frac{B \cdot c_{p\_ch4} \cdot (T_{g2} - T_{g1})}{c_{pg} \cdot G_{air}}.$$

The gas temperature in front of the gas turbine is determined in the following way:

$$T_c = \frac{T_d}{1 - (1 - (\lambda \cdot \varepsilon)^{-m_g}) \cdot \eta_{gt}},$$

where λ – hydrodynamic loss coefficient in the gas turbine and in the heat exchanger; η_{gt} – internal efficiency of the gas turbine.

In the issue the useful power of the gas turbine may be estimated by the following expression:

$$N_2 = c_{pg} \cdot c_{p\_ch4} \cdot T_{g1} \cdot B \cdot F(\delta) \cdot X(\varepsilon) \cdot \eta_{te} \cdot \eta_{gt} \cdot \eta_c \cdot \eta_m \cdot \eta_m \cdot \eta_{eg}. \tag{8}$$

$X(\varepsilon)$ – the function of the air compression ratio in the compressor:

$$X(\varepsilon) = \frac{1}{c_{p\_air} \cdot \eta_m} \cdot \left[\frac{c_{pg} \cdot T_{wg} \cdot \eta_m \cdot \eta_c}{c_{p\_air} \cdot T_a \cdot (\varepsilon^{m_{air}} - 1)} + 1 \right] \cdot \left[\frac{1 - (\lambda \cdot \varepsilon)^{-m_g}}{1 - (1 - (\lambda \cdot \varepsilon)^{-m_g}) \cdot \eta_{gt}} \right].$$

Let us calculate the absolute efficiency η_{GTU} of the concerned installation, which defines as a ration of the useful power N_2 to the combustion chamber heat power Q_{CC}:

$$\eta_{GTU} = \frac{N_2}{Q_{CC}}.$$

In its' turn Q_{CC}:

$$Q_{CC} = \frac{c_{pg} \cdot G_{air} \cdot (T_c - T_b)}{\eta_{cc}}.$$

Then

$$\eta_{GTU} = \frac{c_{pg} \cdot c_{p\_ch4} \cdot T_{g1} \cdot B \cdot F(\delta) \cdot X(\varepsilon) \cdot \eta_{te} \cdot \eta_{gt} \cdot \eta_c \cdot \eta_m \cdot \eta_m \cdot \eta_{eg} \cdot \eta_{cc}}{c_{pg} \cdot c_{p\_ch4} \cdot T_{g1} \cdot B \cdot F(\delta) \cdot Z(\varepsilon) \cdot \eta_{te} \cdot \eta_m \cdot \eta} = \tag{9}$$

$$= \frac{X(\varepsilon)}{Z(\varepsilon)} \cdot \eta_{gt} \cdot \eta_m \cdot \eta_{eg} \cdot \eta_{cc},$$

where

$$Z(\varepsilon) = \frac{1}{T_a \cdot c_{p\_air} \cdot (\varepsilon^{m_{air}} - 1)} \cdot \left[\frac{T_{wg} + T_a \cdot \frac{c_{p\_air}}{c_{pg}} \cdot \frac{(\varepsilon^{m_{air}} - 1)}{\eta_m \cdot \eta_c}}{1 - (1 - (\lambda \cdot \varepsilon)^{-m_g}) \cdot \eta_{gt}} - T_a \cdot \left(1 + \frac{\varepsilon^{m_{air}} - 1}{\eta_c} \right) \right].$$

The derived expression (9) shows that the gas turbine installation with outer turbo expander drive of the compressor efficiency at the fixed waste gases temperature and specified hydro-dynamic loss coefficient λ depends on the air compression ratio in the compressor only. This dependency is shown in Fig. 5 and all characteristic quantities for the concerned self-contained gas turbine installation with turbo expander drive of the compressor operating jointly with turbo installation K-300-240 LMZ are presented in Table 2.
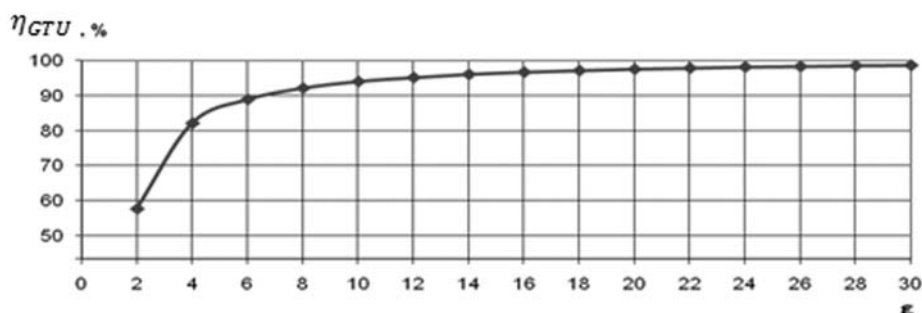

Fig. 5. The gas turbine unit with turbo expander driver of the compressor as a function of the air compression ratio

The thermal scheme calculations of the conventional turbo expander unit connected directly with electric generator and the calculations of the alternative self-contained gas turbine unit with the turbo expander drive of the compressor shows that the passage to more difficult installation utilized the compressed gas power allows us to considerably increase the generating power at the same natural gas flow. So the application of such installation on the thermal power plant gas-distribution station supplying the steam boiler of the power-generating unit K-300-240 the useful power capacity increases from 3.5 MW up to 6.8 MW.

The efficiency of the addition power generation in case of the conventional turbo expander unit is 70 % as the usage of the extracted steam for the gas heating leads to power decrease of the main power-generation unit but in case of the gas turbine installation with turbo expander drive of the compressor application the addition power generation efficiency reaches 98 %. The efficiency is high because the both units utilize the energy of the compressed natural gas at the equivalent temperatures in front of and behind the turbo expander ensuring in such way the transformation of the thermal energy delivered to the heat exchanger into electric one.

If the air compression ratio ε is more than 4, the efficiency of such installation considerably exceed the modern power-generating units of the thermal power plants. Thereafter the specific fuel consumption on the concerned self-contained gas turbine installations with turbo expander drive of the compressor appears considerably lower than the specific fuel consumption on the power-generating units.

4. Conclusions

1. The application of the self-contained gas turbine unit with turbo expander drive of the compressor on the gas-distribution stations allows to increase the additional power generated on the base of compressed natural gas energy in 1.6–2 times in comparison with the conventional turbo expanders.

Table 2. Parameters of self-contained gas turbine installation

No.		parameter	value
1	δ	Gas expansion ratio	8
2	t_{g1}	Gas temperature in the gas-main pipeline, °C	15
3	t_{g2}	Gas temperature in front of the turbo expander, °C	141.9
4	N_{te}	Power capacity of the turbo expander and air compressor, kW	4 893.5
5	ε	Air compression ratio in the air compressor	14
6	t_b	Air temperature in front of the combustion chamber, °C	401.5
7	G_{air}	Air flow at the compressor inlet, kg/s	12.73
8	t_c	The combustion product temperature in front of the gas turbine, °C	874.4
9	t_d	The combustion product temperature behind the gas turbine, °C	402.6
10	T_{wg}	Waste gases temperature behind the heat exchanger, °C	60
11	B	Air flow entered to the combustion chamber, kg/s	0.155
12	N_{gc}	Power consumption for the gas compression in front of the combustion chamber, kW	55.7
13	N_{gt}	Gas turbine power capacity, kW	6 730.4
14	Q_{CC}	Heat power of the combustion chamber, kW	6 871.3
15	η_{te}	Turbo expander efficiency, N_{gt}/Q_{CC}	98

2. So far as the gas temperature in front of and behind the turbo expander must be about the same that for this conditions ensuring all installations which utilize the energy of compressed natural gas must operate with heat supply equivalent to the turbo expander power. From this point of view all considered installations almost completely transforms the delivered heat energy into electric one with very high efficiency.

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