ENERGY RECOVERY IN PRESSURE REDUCTION STATIONS OF NATURAL GAS USING TURBOEXPANDERS

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Natural gas transport, and consumption take place at different pressure levels. The shift from the pressure of the transport system to that corresponding to the distribution is carried out in dedicated reduction stations. This article analyzes the solution of exploiting the energy potential of reducing the gas pressure from 20 bar to 6 bar by expanding it in a turboexpander. The main disadvantage in this case is the considerable decrease in temperature at the exit of the turboexpander. Considering the necessity of gases heating prior to their entry into the turboexpander, was analyzed two solutions namely: - The natural gas preheating solution by burning a gas share in a furnace and expanding it in a turboexpander; - Gas intermediate preheating solution and fractional expansion.

Keywords: natural gas, power generation, pressure reduction, turbo expander

1. Introduction

Natural gas transport, distribution and consumption take place at different pressure levels. The shift from the pressure of the transport system to that corresponding to the distribution is carried out in dedicated reduction stations, called Control and Measuring Stations (CMS), Fig. 1.

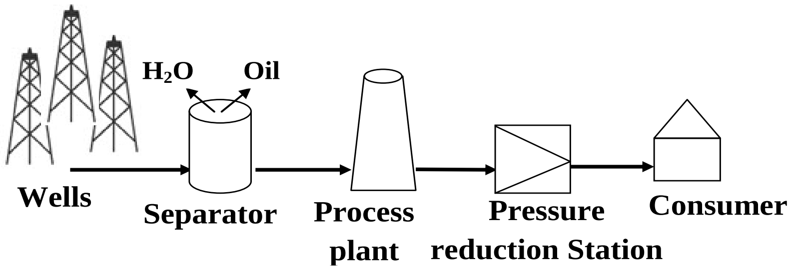

Fig. 1. Scheme of natural gas supply chain

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Natural gases are transported through high-pressure main pipelines between 10 and 20 bar so that the transport should be more efficient, then taken over by CMS, reducing pressure to 2.5 - 5 bar and distributed to industrial or home consumers.

The solution currently used in CMS is to expand the natural gas from transport pressure to usage one. From the energy point of view, expansion leads to an important loss of energy.

A modern solution is to use a turboexpander in order to reduce the natural gas pressure.This solution allows to obtain a mechanical work based on this expansion.

Worldwide, there are numerous applications regarding the exploitation of the energy coming from reducing the pressure of the gases in the transport and distribution networks using turboexpanders in countries such as Canada, England, Italy, the Czech Republic and Iran.

2. Existing limitations regarding the expansion of the natural gas in a turboexpander

The main drawback of the natural gas adiabatic expansion process is the considerable decrease in temperature at the exit of the turboexpander reaching negative temperatures. This is explained by the production of mechanical work during the adiabatic expansion due to the diminution of the gases internal energy.

Another limitation refers to the content of water vapors found in natural gases mixtures, which are considered impurities. Their presence involves operational problems through the formation of hydrates, corrosion, high pressure drops and consequently a turbulent flow causing the flow efficiency to decrease [1, 2]. Still another negative aspect is that the presence of water vapors in natural gases reduces the calorific value.

The solution that is required to overcome these limitations that lower the gas temperature when exiting the turboexpander is to preheat the gas before the expansion process.

The lost energy, when natural gases expand, is used by the turboexpander to trigger an electric generator, as can be seen in Fig. 2.

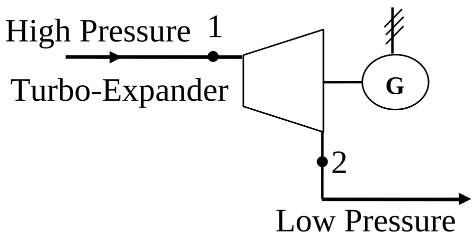

Fig. 2. Schematic concept for Turbo-Expanders

The main drawback in this case is the considerable decrease in temperature when coming out from the turboexpander.

To highlight this phenomenon, was considered the average temperature of the natural gas t_1 at the entrance in the turboexpander to be between 5 °C and 31 °C, was showed the results in Fig. 3.

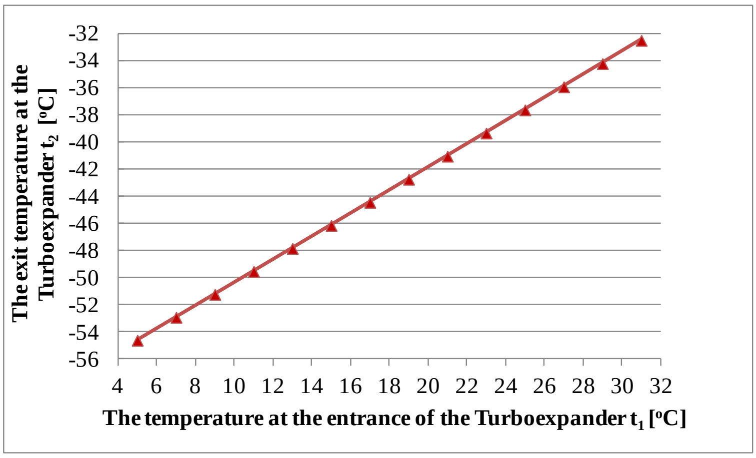

Fig. 3. Gas temperature at the exit of the Turboexpander, depending the entrance temperature, for a pressure reducing from 20 bar to 6 bar

The adiabatic expansion process leads to a considerable decrease of the temperature at the exit of the turboexpander to $t_2 = -55\,°C$, in the case of pressure reduction from 20 bar to 6 bar (Fig. 3). The power produced in the adiabatic expansion takes place based on the decrease of the gases internal energy, obtaining a negative temperature of $t_2 = -32\,°C$ for a gases temperature at the entrance in the turboexpander of 31 °C. This being unacceptable from a technical point of view, that is why we need various gas pre - heating solutions before entering the turboexpander. An example in this regard was analyzed in paper [3], where the gas flow was reduced from a pressure of (49 - 55) bar to 18 bar and in order to avoid lowering the temperature of the natural gas below the temperature of the dew point, it was preheated to a temperature of 83 °C.

Considering the necessity of gases heating prior to their entrance in the turboexpander, was analyzed two solutions namely:

- The natural gas preheating solution by burning a gas share in a furnace and expanding it in a turboexpander.
- The natural gas preheating solution by means of a furnace and expanding it in a two-body turboexpander.

Between the gas inlet pressure and the exit pressure there is a temperature drop of about 4.5 – 6 °C/MPa known as the Joule - Thompson effect, which depends on its state and composition, as Poživil shows in a study on the natural gas transportation network in the Czech Republic, while in the case of using a turboexpander the temperature can also decrease by 15 - 20 °C/MPa [4].

In order to avoid high cooling, the gas was preheated before expanding it in the turboexpander so that the exit temperature was maintained at 3 °C, which is the temperature above the dew point of the condensable hydrocarbons.

Worldwide, there are numerous applications in terms of energy exploitation resulting from the reduction of gases pressure in transport and distribution networks using turboexpanders. In Canada, in 2008, a 2.2 MW turboexpander and combustion cells plant was put into operation. The use of combustion cells using natural gas along with the turboexpander in the natural gas pressure reduction system led to an increase in its efficiency by 10% for a flow of 12000 m^3/h of gas [5].

In London, in 2009, a natural gas pressure reducer with a turboexpander and a biofuel generator for gases preheating was put into operation. The preheating temperature of the gases is very important because it is an additional energy consumption, as evidenced in the work of Zabihi and Taghizadeh [6], by installing a high-performance temperature controller in the gas pressure regulating station. The results obtained by means of the Hysys simulation software were compared with those measured in the gas pressure reduction stations, leading to a 65% reduction of the fuel used to preheat gas. This reduction was carried out by imposing a temperature of 15 °C of the CH_4 at the exit of the second regulation stage, thus avoiding the additional energy consumption for preheating.

The variation in gas flow influences the efficiency of the turboexpander, this being evidenced in several papers [7]. In his paper, Zehtabian analyzes the gas flow variation in one year at a gas-pressure reduction station equipped with a turboexpander in Takestan, Iran, with a flow of 20000 m^3/h. The energy produced in this case is of 1104737 kWh. The study shows that the efficiency of the turboexpander drops due to the gas flow variation to 67.79 % compared to the nominal efficiency of 80% [8]. In another study conducted in Italy [9] it was pointed out that a reduction in the natural gas pressure from (1.13 - 5.1) MPa to (0.15 - 0.6) MPa and a variable gas flow between 5000 - 30000 N m^3/h leads to a power output of the turboexpander between 300 - 1400 kW.

The article analyzes several variants involving the expansion of natural gas in a turboexpander, eliminating the expansion stage and at the same time obtaining the electric energy as can be seen in Fig. 4.

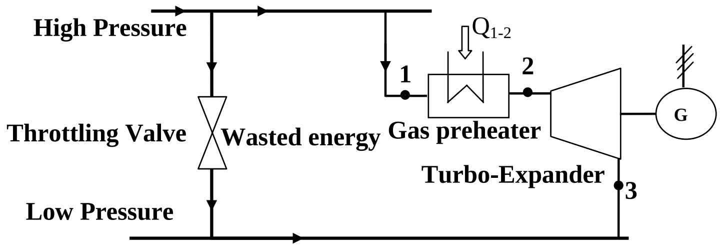

Fig. 4. Schematic concept for use Turbo-Expanders in natural gas networks

3. The solution of pre-heating the natural gas before it is expanded in a turboexpander

Was considered a scheme of preheating the natural gas by burning a share of the gas in a furnace and expanding it in a turboexpander.

Is require that the temperature of the gases at the exit of the turboexpander remains above the dew point of condensable hydrocarbons and the water dew point of about 3 °C in order to avoid the formation of hydrates. In Fig. 5 presents the use of a furnace to preheat the gas before entering the turboexpander.

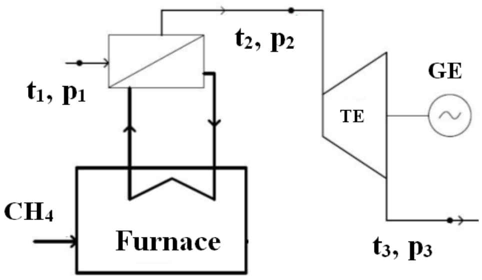

Fig. 5. Natural gas preheating by burning a quota/ share of the natural gas in a furnace

The natural gas expansion process in the turboexpander is considered to be irreversibly adiabatic, with isentropic efficiency. Using the notations in Fig. 5 and a specialized software, REFPROP it was determined the CH_4 thermodynamic properties and then was calculated the power resulting from the process [10].

$$\eta_{TE} = \frac{h_2 - h_{3r}}{h_2 - h_{3t}}, \tag{1}$$

$$h_{3r} = h_2 - \eta_{TE}\left(h_2 - h_{3t}\right) \tag{2}$$

Where: η_{TE} - the isentropic efficiency of the turboexpander

h_2 - the gas enthalpy at the outlet/exit of the first preheating stage

h_{3t} - the theoretical enthalpy after the first stage of expansion

h_{3r} - the real enthalpy after the first stage of expansion

In this case, was varied the gases temperature at the inlet of the turboexpander from 30 °C to 90 °C, by calculating the gases exit temperature t_3, the thermal load of the furnace $\dot{Q}$, the power produced P_{T_1}, the gross electrical power at the generator terminals *GP* and the fuel consumption required for the preheating of the gas natural *BF*.

Where:

$$\dot{Q} = D_{massCH_4} \cdot (h_2 - h_1) \tag{3}$$

$$P_{T_1} = D_{massCH_4} \cdot (h_2 - h_3) \tag{4}$$

$$GP = P_{T_1} \cdot \eta_g \cdot \eta_m \tag{5}$$

$$BF = \frac{D_{massCH_4} \cdot (h_2 - h_1)}{H_i \cdot \eta_F} \tag{6}$$

Here, was also calculated a quality indicator – the Efficiency ratio I, which defines the ratio between the mechanical work produced by the turboexpander through the gas expansion and the energy needed for the preheating of the natural gas in the furnace.

$$I = \frac{P_{T1}}{\dot{Q}} \tag{7}$$

According to the National Meteorology Administration, in Romania the average annual atmospheric temperature is of about 10 °C, which is why was considered in the analysis that this should also be the temperature of the gas. The input data that were used in this study are presented in Table 1.

Table 1

Input parameters are shown in Table 1.

	Notation	Value	UM
Temperature	t_1	10	°C
Mass Flow of CH_4	D_{CH_4}	0.60	kg/s
Isentropic Efficiency of the turboexpander	η_{TE}	0.8	-
Generator Efficiency	η_g	0.93	-
Mechanic Efficiency	η_m	0.92	-
Furnace Efficiency	η_F	0.9	-
Inferior calorific value of natural gas	H_i	50000	kJ/kg

In this case, was analyzed the natural gas pressure reduction from 20 bar to 6 bar. Fig. 6 shows that the preheating temperature of the gas t_2 must be of at least 80 °C to ensure a positive gas temperature of the 10 °C at the exit of the turboexpander. In this case a power of 78 kW was obtained at the generator. Here, as well, the value of the quality indicator I drops with the increase of the preheating temperature of the gas, this having a value of 0.90 for the temperature mentioned at the exit of the turboexpander.

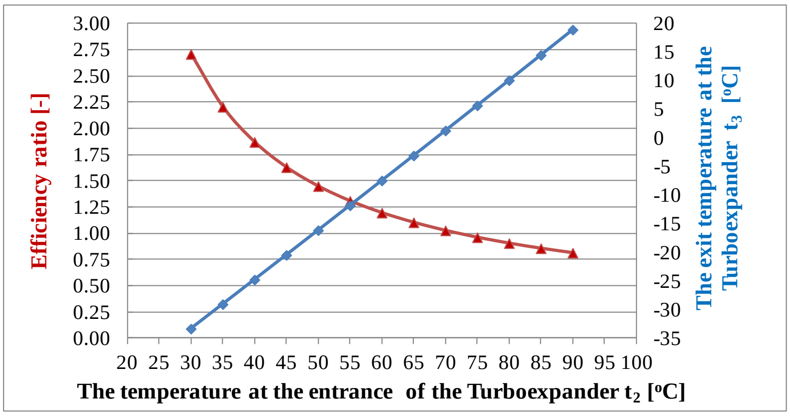

Fig. 6. Reducing gas pressure from 20 bar to 6 bar

During the gas expansion process in the turboexpander, regardless of the pressure levels at which one operates, one intends to obtain a quality indicator, meaning an Efficiency ratio I as high as possible, but which should meet the limitation imposed by the positive temperature at the exit of the turboexpander.

Fig. 7 shows the variation of the power and of the thermal load with the increase of the turboexpander inlet temperature. The intersection point between the power produced by the expansion of the gas in the expander and the thermal load consumed by the furnace is at a temperature of 75 °C.

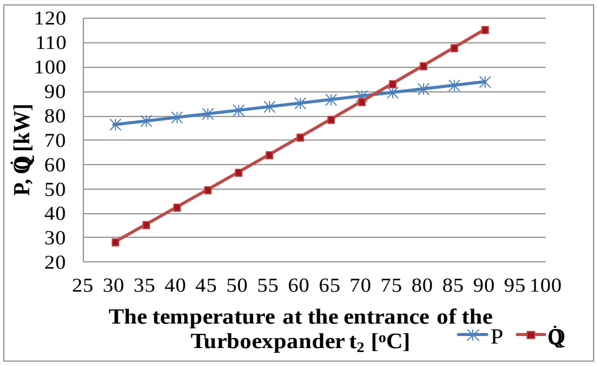

Fig. 7. Variation of the power and of the thermal load of the furnace in the case of gas pressure reduction from 20 bar to 6 bar

To meet the limitation of the gas exit temperature, this must be heated up to 80 °C leading to a higher thermal load than the power produced.

4. Intermediate pre-heating solution of the natural gas and fractional expansion

Was considered a turboexpander with fractional expansion and intermediate heating. The scheme also benefits from an initial preheating when entering the first stage.

The two-stage gas intermediate preheating solution and fractional expansion is shown in the equivalent scheme in Fig. 8, where it is observed the preheating circuits of natural gas before expanding in each of the two turboexpanders.

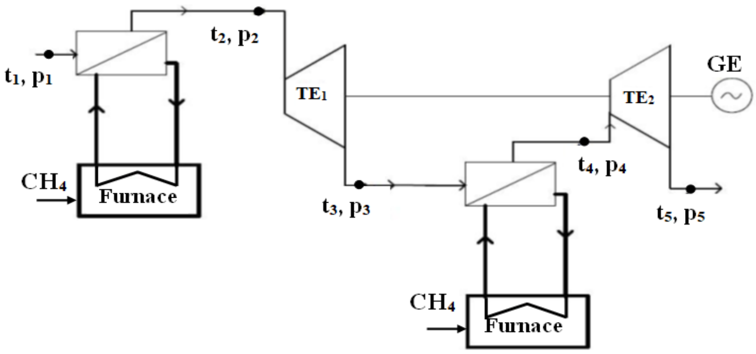

Fig. 8. Preheating natural gas in two stages with fractional expansion.

In this case, the efficiency ratio is defined as the ratio between the mechanical work produced by the two turboexpanders and the thermal load consumed by the two gas preheating circuits.

$$I = \frac{P_{T1} + P_{T2}}{\dot{Q}_1 + \dot{Q}_2} \tag{8}$$

In this case was analyzed two aspects, namely: a first aspect related to the identification of the optimal intermediate operation pressure $p_3 = p_4$ and the second aspect related to the gas pressure reduction analysis from 20 bar to 6 bar, using different schemes of the gas preheating furnace;

a) *The identification of the optimal intermediate pressure p_3, in the case of the natural gas fractional expansion.*

To determine the intermediate pressure p_3, was calculated the power produced by the two turboexpanders, by varying the gas preheating temperature at the exit of the two furnace stages $t_2 = t_4$ from 30 °C to 60 °C.

The reduction of gas pressure was carried out from 20 bar to 6 bar, in Fig. 9 and the temperature at the exit of the two preheating stages $t_2 = t_4$ was varied between 30 °C and 60 °C. It is noticeable that the maximum value of the Power $P_1 + P_2$ is obtained for an optimal intermediate pressure of 11 bar. Also, in this graph shows the optimal intermediate pressure is small influenced by the preheating temperature of the gas.

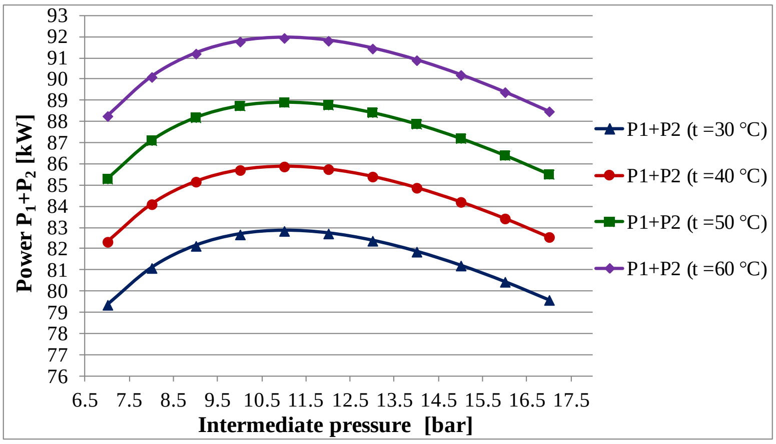

Fig.9. Variation of the intermediate pressure depending on the Power for different temperatures of gas preheating

Fig. 10 shows, depending on the optimal intermediate pressure determined, the increase of the thermal load in the furnace, with the gas pre-heating temperature.

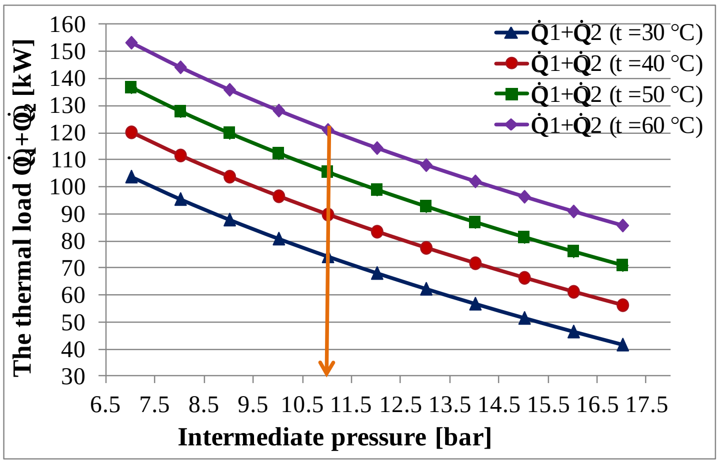

Fig.10. Variation of the intermediate pressure depending on the thermal load of the furnace for different temperatures of gas preheating

Fig. 11 shows that the quality indicator has a slight decrease with the increase in the temperature for the previously set optimal pressure of 11 bar.

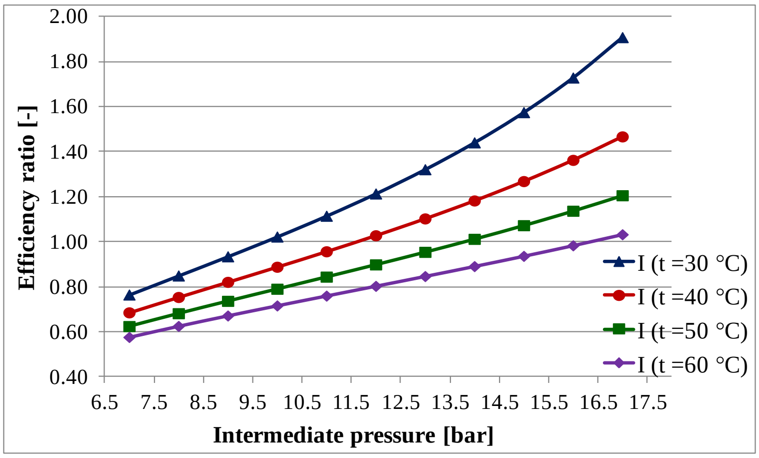

Fig.11. Variation of the intermediate pressure depending on the efficiency ratio, for different temperatures of gas preheating

Following the analysis, for the two levels of the gas pressure reduction and for different preheating temperatures, it resulted that the power produced in the two stages reaches a maximum value at an optimal intermediate pressure. This optimal intermediate pressure results as the geometric mean between the inlet pressure in the first stage and the exit one from the second stage.

$$p_3 = \sqrt{p_1 \cdot p_5} \tag{9}$$

b) Parameter analysis to reduce the gas pressure from 20 bar to 6 bar in the case of the fractionated expansion, using a furnace.

In the case of the gas fractional expansion from 20 bar to 6 bar, was maintained the optimal intermediate pressure $p_3 = 11$ and was varied the gas preheating temperature $t_2 = t_4$ between 20 °C and 80 °C before expanding it in the two stages of the turboexpander. Fig. 12 shows that the temperature up to which the gas has to be preheated $t_2 = t_4$ before entering the turboexpander is of about 45 °C, so that the temperature at the exit of the turboexpander is not too low but reaches $t_3 = 10.17°\text{C}$ and $t_5 = 10.91°\text{C}$ respecting the limitations imposed by the occurrence of the hydrates. Fig. 13 shows the variation of the power, the thermal load and the quality indicator, Efficiency ratio I. In this case a power of 84 kW was obtained at the generator, for temperature $t_2 = t_4$ about 45 °C.

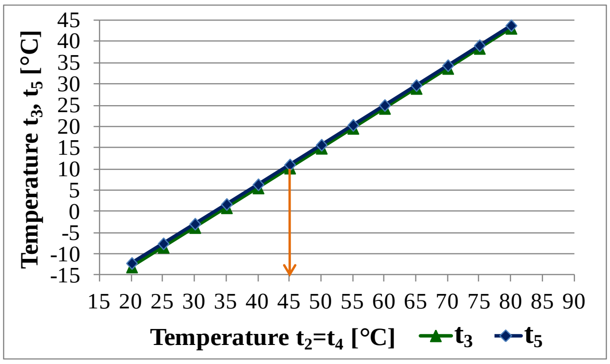

Fig.12. Variation of the temperature for gas preheating in the furnace.

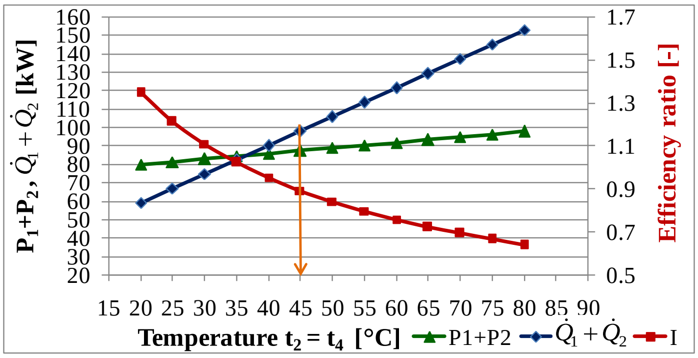

Fig.13. P, $\dot{Q}$, I variation depending on the gases preheating temperature.

5. Conclusions

The calculations indicate that reducing the natural gas pressure from 20 bar to 6 bar by expanding it in a turboexpandor leads to negative gas temperatures at the turboexpandor outlet, which is not technologically acceptable.

The solution of preheating the natural gas up to 80 °C by burning a share of the gas in a furnace and expanding it in a turboexpandor led to a gas temperature of 10 °C at the turboexpandor outlet, thus removing the temperature limitation issue. In this case a power of 78 kW was obtained at the generator.

As a result of the calculations was observed that in the case of the fractional expansion with intermediate heating, the power obtained in both stages reaches a maximum value for an optimal intermediate pressure of 11 bar. The optimal pressure is determined as the geometric mean between the inlet pressure in the first stage and the outlet pressure from the second stage, having little influence from the preheating temperature of the gas.

In the case of fractional expansion, it was observed that little preheating of the CH_4 was required before expansion. A temperature of 45 °C was sufficient for achieving a positive gas temperature at the turboexpandor outlet. This scheme can be applied if geothermal energy sources are used for preheating, by utilizing heat pumps, which can provide an increase of 50 K up to 130 K in the temperature between source and user in the case of industrial high temperature heat pumps [11].

The power obtained in the case of fractional expansion is insignificant compared to other technologies in which this process is applied, such as fractional expansion in a gas turbine, considering that the turboexpander plays a role similar to our case [12, 13]. In the case of CH_4, this fact can be explained by the less divergent character of the isobaric curves in the temperature - entropy diagram.

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