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Utilization of Turbo-Expander to Generate Power in Natural Gas Extraction Process

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Abstract

Natural gas is one of the most reliable and general energy sources globally, which their reservoirs always have high pressures. Nowadays, due to the production and supply of a variety of turbine expansions, it is possible to generate power from natural gas extraction. In this research, by considering the operational conditions and exploitation data, the possibility and advantage of using turbo-expander in gas extraction process are explored. Based on the results, 0.26-1.1 MW of power can be generated from each production well using the pressure drop in the turbo-expander with the flow ranges between 0.5-2.0 million STD_m^3/d. The power generation capacity due to natural gas conditions may be associated with the production of gas condensates which has also been studied in this research.

Keywords: Natural Gas, Power, Energy, Turbo-Expander, Condensates

Introduction

The world's limited energy resources have led researchers to optimize energy systems and develop new ways to use more energy resources. Nowadays, much of the world's energy comes from fossil fuels. Considering the depth of extraction and consequent processing costs alongside the limited amount of these fuels, one should always look for new energy sources or optimize the extraction methods to reduce energy loss while using existing energy sources [1,2]. There is always a need to reduce gas pressure in the natural gas industry, which can greatly reduce the energy loss by using turbo-expanders. The turbo-expander technology is currently implemented in natural gas industries to generate mechanical energy [3]. Recently, some details of power generation from high-pressure natural gas and guidelines for designing such installations, taking into account the thermodynamics, gas flow rates and pre-heating requirements, are published by some researchers [4,5].

Most researches have been devoted to design turbo-expanders for various applications [6-8]. In the study of Sun et al., a validated non-equilibrium condensation model is developed to analyze the spontaneous condensation flow in the nozzle, impeller and diffuser passages of cryogenic turbo-expander [9]. The influences of impeller blade profile on spontaneous condensation flow and thermodynamic performance were investigated in the study of Chen et al., while the non-equilibrium spontaneous condensation process in the flow passage is simulated [10].

Turbo-expanders, also called expansion turbines, provide a way to recover the wasted energy in natural gas industries and refineries [11]. Turbo-expanders have a wide range of applications, but here we focus on energy recovery and power generation. Various models of turbo-expander are currently available, ranging from 75 kW to 25 MW, and they can be used for a variety of applications [1].

Any high temperature or high-pressure gas is a potential source for energy recycling. Power generation turbo-expanders can recover the maximum useful energy available in the pressure reduction process, and thus they have high efficiency. The governing principle of the turbo-expander is the conversion of the kinetic energy of gas into useful energy such as electrical energy by turbines [12]. As the gas flows through the high-pressure part into the turbo-expander, it rotates the turbine and provides shaft power as shown in Figure 1. The idealized process for turbo-expander is an isentropic process or constant entropy process in which power is generated by reducing the enthalpy of the passing flow.

An expansion turbine is a centrifugal or axial-flow turbine, through which high-pressure gas is expanded to produce work that is often used to drive a compressor or generator. Therefore, by replacing a control valve (choke valve) or regulator used to reduce gas flow pressure with a turbo-expander, the available energy of the fluid flow that would otherwise be lost can be converted to shaft work for power generation and other uses.

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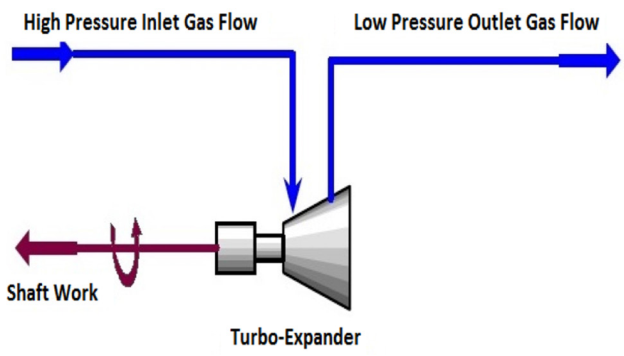

Fig. 1 Turbo-Expander.

It should be noted that the output energy is proportional to the ratio of pressure, temperature and flow rate.

In the gas extraction process, choke valves (or pressure-reducing valves) are generally used to reduce gas pressure. In this study, an effort is made to evaluate the advantage of using turboexpanders instead of choke valves to prevent energy loss in the pressure reduction process. This study shows that significant amounts of energy that was previously wasted can be recovered by which offers a new source of energy.

Materials and Methods

Application of Turbo-expander in Extraction of Natural Gas

Natural gas mainly consists of methane, the shortest and lightest hydrocarbon molecule, as well as various amounts of heavier hydrocarbon gases such as ethane, propane, butane, isobutene, pentane and even higher molecular weight hydrocarbons. Unrefined gas also contains significant amounts of acidic gases such as carbon dioxide or hydrogen sulfide.

Nowadays, various companies have succeeded in developing turbo-expanders with high flow rates and high inlet pressures that can be used in the oil and gas industry. For example, there are examples of turbo-expanders that work by hydrocarbon fluids with an inlet pressure of 206 bar and a temperature range of -195 °C to 260 °C [13].

In this study, to establish a close relationship between the results and the actual conditions of operation, the operation condition of the Dalan zone, one of the operational zones of the South Zagros Oil and Gas Production Company, has been studied. Also, the gas composition of this region was used for the present study which is listed in the Table 1.

Mathematical Modeling

The thermodynamic processes of the gas expansion in the turbo-expander and expansion valve are shown in Figure 2. Both isentropic (state 2s) and non-isentropic (state 2) processes for turbo-expanders are shown in Figure 2. State 2 is the actual state leaving the turbo-expander following an irreversible process due to pressure drop in the non-isentropic process along the flow path. If expansion takes place within the expansion valve, the gas flow expands in a constant enthalpy process (state 3) without producing any work. According to the law of conservation of mass, the fluid components and mass flow rate through the turbo-expander are constant. It can be written as [2].

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} = \dot{m} \tag{1}$$

Table 1 The composition of Natural Gas used in this study.

Gas Component	Molar fraction
Methane	0.8811
Ethane	0.0139
Propane	0.0033
Nitrogen	0.0778
CO_2	0.0162
Benzene	0.0001
Toluene	0.0001
Helium	0.0012
i-Butane	0.0009
n-Butane	0.0013
2-Mpentane	0.0004
2-Mhexane	0.0003
n-Pentane	0.0006
i-Pentane	0.0007
E-Benzene	0.0003
n-Hexane	0.0003
C_7^+	0.0014
H_2O	0.0001

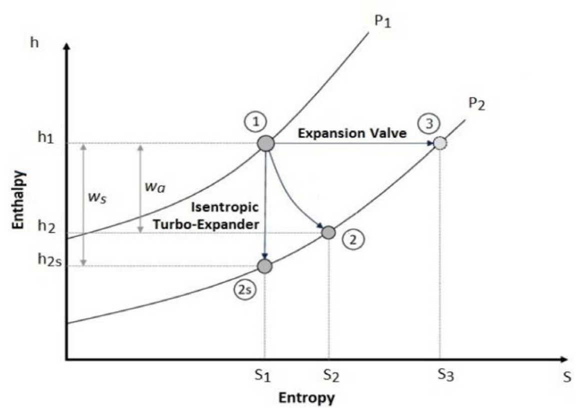

Fig. 2 Enthalpy-entropy diagram for the governing process on turbo-expanders and comparison with expansion valve function.

The first-law or energy balance relation in that case for a general steady-flow system becomes [14]:

$$\dot{Q}_{C.V.} + \sum_{in} \dot{m}\left(h + \frac{V^2}{2} gz\right) = \dot{W}_{C.V.} + \sum_{out} \dot{m}\left(h + \frac{V^2}{2} gz\right) \tag{2}$$

where $\dot{Q}_{C.V.}$ and $\dot{W}_{C.V.}$ are the heat transfer and work done by the working fluid in the control volume, respectively. When there are negligible changes in (1) fluid kinetic and potential energies and (2) also the heat transfer between the control volume and its surroundings (that is, $\Delta ke \cong 0$, $\Delta pe \cong 0$, $\dot{Q}_{C.V.}$), the energy balance equation is reduced further to

$$\dot{W} = \dot{m}(h_{in} - h_{out}) \tag{3}$$

where $\dot{W}$ is the power output of turbo-expander.

Many engineering systems or devices such as pumps, turbines, turbo-expanders, nozzles, and diffusers are essentially adiabatic in their operation, and they perform best when the irreversibilities, such as the friction associated with the process, are minimized. Therefore, an isentropic process can serve as an appropriate model for actual processes. Also, isentropic processes enable us to define efficiencies for processes to compare the actual performance of these devices to the performance under idealized conditions. The term “isentropic” means constant entropy. A process during which the entropy remains constant is called an isentropic process, which is characterized by

$$s_{in}=s_{out} \tag{4}$$

An isentropic process is an idealization of an actual process in the turbo-expander, and it serves as a limiting case for a real process [14].

$$\frac{T_{in}}{T_{out}} = \left(\frac{P_{in}}{P_{out}}\right)^{(k-1)/k} \tag{5}$$

$$\dot{W} = \dot{m}\, C_p (T_{in}-T_{out}) \tag{6}$$

where k is the ratio of the specific heats, and it is defined for the gas mixture as [15]

$$k=k_{mix}=C_{p_{mix}}/C_{v_{mix}}=C_{p_{mix}}/(C_{p_{mix}}-R_{mix}) \tag{7}$$

$$C_{P_{mi.x}} = \sum x_i C_{p_i} \tag{8}$$

where x_i is the mass fraction for each component of the mixture.

It should be noted that the actual process always deviates from the ideal isentropic process, and in practice, the turbo-expander performance is always lower than its isentropic value. Thus, the efficiency of turbo-expander is found as follow:

$$\eta_{T.E} \cdot \frac{h_1 - h_2}{h_1 - h_{2s}} \tag{9}$$

Since the main purpose of this study is to calculate the electrical power output of the system, so the net output power is obtained from:

$$L = \dot{W}\, \eta T.E.\eta_{generator} \tag{10}$$

The pressure drop occurs in the turbo-expander. Also, the pressure ratio will be calculated from the following equation:

$$\text{Pressure Reduction Ratio} = P_{outlet}/P_{Inlet} \tag{11}$$

Results and Discussion

A pure component of a natural gas system exhibits a characteristic phase behavior, as shown in Figure 3. Depending on the component’s pressure and temperature, it may exist as a vapor, a liquid, or some equilibrium combination of vapor and liquid [15].

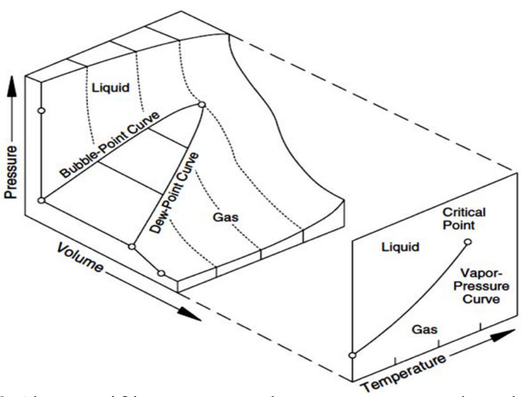

Fig. 3 Characteristic pressure-volume-temperature phase behavior of a pure component [15].

The critical point is the point at which the intensive, or mass-independent, properties of the liquid and vapor phases become identical. This point marks the end of the vapor pressure line and identifies the critical pressure and temperature of the pure component. Therefore, with the help of such graphs found in various petroleum and chemical engineering sources, it is possible to estimate the amount of condensate at any pressure and temperature for natural gas [16,17]. However, nowadays, several softwares such as AspenOne can be used to calculate the amount of condensate under various conditions of temperature and pressure in natural gas as used in this study. This section presents the final results to evaluate the effects of several key parameters on the turbo-expander performance range in actual situations. When efficiency of the turboexpander is assumed to be 100%, the output power is derived. In Figure 4, the amount of isentropic turbo-expander power for various amounts of pressure reduction is shown. In this figure, four batch diagrams can be seen for different gas flow rates. In each batch, four diagrams for each different initial temperature are presented. In Figure 5, one of these batches for the rate of 1500000 STD_m³/d to establish more details is shown separately.

As shown in Figure 4, the power output increased with an increase in gas flow rate through the turbine and an increase in pressure drop rate, which is about 1.1 MW for the rate of 2000000 STD_m³/d and 60 °C initial temperature.

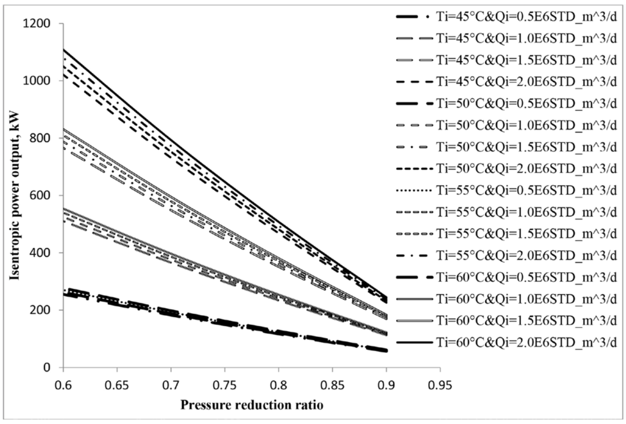

Fig. 4 Isentropic power output of turbo-expander in different flow rates and initial temperatures for gas with initial pressure 150 bar.

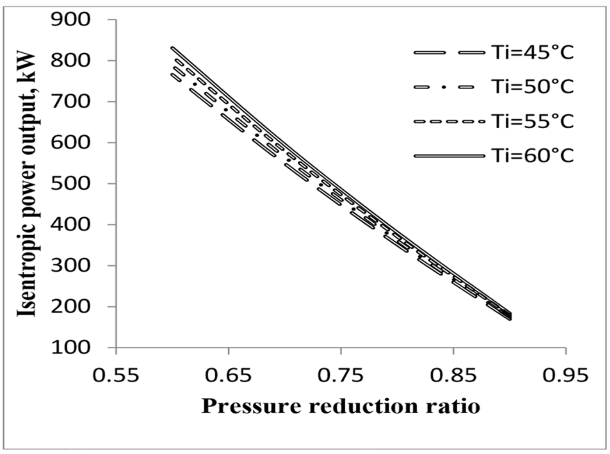

Fig. 5 Variation of isentropic power output as a function of pressure reduction ratio in different initial temperatures for gas with initial pressure 150 bar and flow rate of 1500000STD_m³/d.

Figure 5 shows the effect of the initial temperature on the power output of the expansion turbine, which always increases with an increase in input temperature. This effect is greater for the small values of pressure reduction ratios. The effect of initial temperature has negligible for the lower pressure drop (Higher pressure reduction ratio). The effect of gas flow rate on power generation is shown in Figure 6. As can be seen in this figure, the power output increases with a rise in the working fluid flow rate.

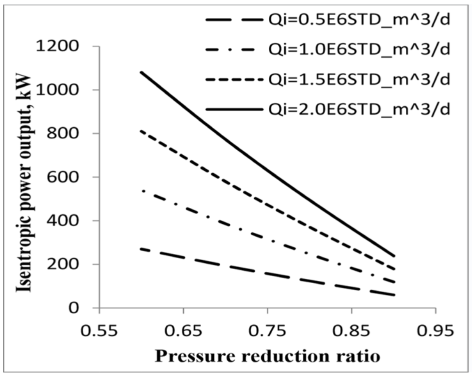

Fig. 6 Variations of power output with pressure reduction ratio at different flow rates for initial pressure of 150 bar and initial temperature of 55°C.

The effect of the pressure reduction ratio on the outlet temperature of the turbo-expander is shown in Figure 7. It is shown that by decreasing the pressure reduction ratio, outlet temperature decreases. The decrease in temperature is associated with an increase in the amount of gas condensates in the turbo-expander outlet, so it is a limiting factor in its application. In Figure 8, the amount of gas condensates formed at the turbo-expander outlet is shown. As can be seen, the amount of generated liquids decreases by increasing the inlet temperature and decreasing the pressure drop. As shown in Figure 8, the condensate formation value is zero for the pressure reduction ratio of 0.9. It will also be zero for the inlet temperature of 60 °C at a pressure reduction ratio of 0.8. Therefore, using a turbo-expander is very suitable if the initial conditions of natural gas match the ranges mentioned above. The low percentage of gas condensates can be neglected, and it will be separated by separators in separation centers. In this study, the lower limit for temperature about 10 °C and liquid phase about 0.5% (volume fraction) is considered. But, according to the liquid and gas velocity ratio in the pipeline, the upper value of the volume fraction of produced gas condensates should be in the range of 0.01 to 0.001 [18].

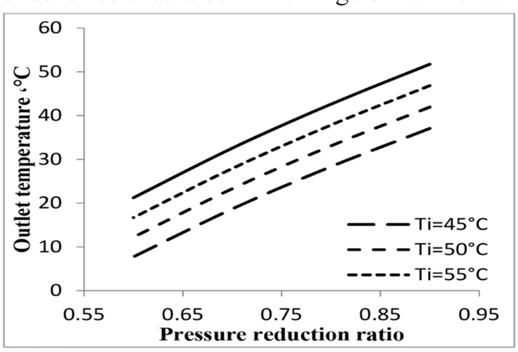

Fig. 7 Outlet temperature of turbo-expander as a function of pressure reduction ratio in different initial temperatures for initial pressure of 150 bar.

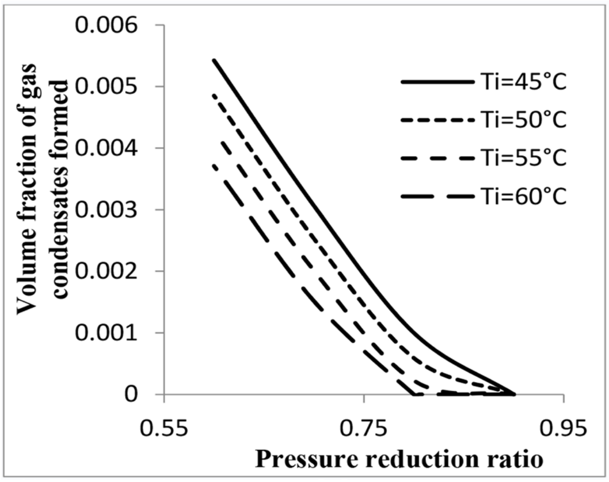

Fig. 8 Volume fraction of gas condensates formed in turbo-expander outlet as a function of pressure reduction ratio for initial pressure of 150 bar.

However, suppose the thermodynamic expansion process of the turbo-expander is carried out in the non-isentropic process. In that case, its efficiency is reduced, and the output power, outlet temperature, and condensate formed will be changed. It should be noted that the actual turbo-expander process differs from the ideal process primarily because of irreversibilities in the expansion turbine and because of pressure drop in the flow passages.

Figure 9 shows the change in the outlet temperature of the turbo-expander, assuming that leaving the isentropic process only affects the temperature. It can be seen in this figure that the actual outlet temperature is higher than the isentropic outlet temperature. A rise in outlet temperature of turbo-expander lowers the amount of gas condensate, as shown in Figure 9.

Figure 10 shows the volume fraction of gas condensate formed in turbo-expander outlet as a function of pressure reduction ratio in different efficiencies. As shown in this figure, in lower pressure reduction ratios, more condensate is formed.

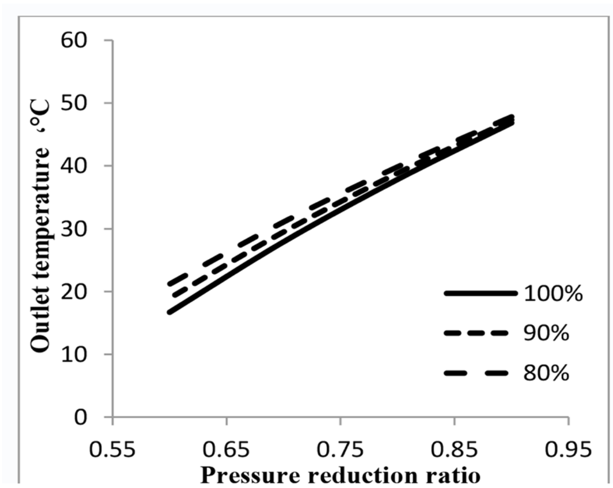

Fig. 9 Turbo-expander outlet temperature variations in different efficiencies for initial pressure of 150 bar and initial temperature of 55 °C.

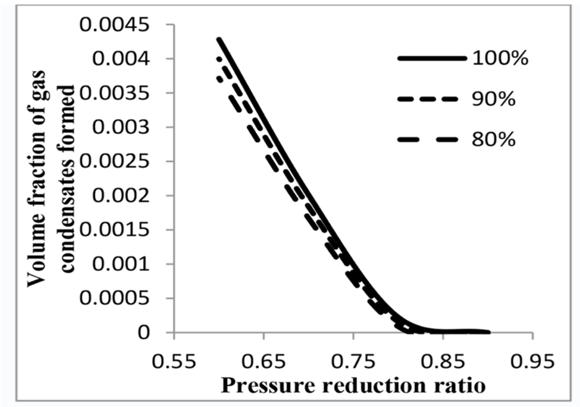

Fig. 10 Volume fraction of gas condensate formed in turbo-expander outlet as function of pressure reduction ratio in different efficiencies for initial pressure of 150 bar and initial temperature of 55 °C.

Conclusions

In this study, the advantage of using a turbo-expander instead of expansion valves in natural gas extraction was investigated. The conditions of gas exploitation of Dalan natural gas reservoir were studied, and corresponding diagrams were extracted to show the power output, temperature drop and amount of condensates formed under different conditions of turbo-expander.

According to the results of this research, from a gas well, depending on the amount of pressure drop in the turbo-expander, for a range of flow rate of 0.5-2.0 STD-m3/d, 0.26-1.1 MW power can be generated. Although turbo-expanders are manufactured nowadays with a wide temperature range and different percentages of formed condensate at the outlet, due to the conditions of the operation and transmission in pipelines, it must be tried to minimize the amount of condensed liquids.

The results indicate that with a decrease in the pressure reduction ratio, the outlet temperature always drops, and the formed condensate increases. Therefore, the use of turbo-expander is not suitable for high-pressure drop and low inlet gas temperatures, which is observed for 45 °C and 0.6 pressure reduction ratios. As shown in the results of this study, for the low-pressure reduction ratio, the condensate formation value is zero. Hence, using a turbo-expander is very suitable for these conditions of natural gas.

It should be noted that turbo-expanders will always have some energy loss, which will reduce their efficiency. In this study, diagrams for 80% and 90% efficiency have been presented, which show that although the reduction in efficiency leads to less power generation, but due to the increase in outlet temperature and the reduction of gas condensate formed, so makes the use of turbo-expander more appropriate for the gas extraction process.

According to this research, it was found out that using turbo-expanders in the gas extraction process would generate power and prevent a lot of energy loss. In addition, considering the gas production capacity in Iran, which amounted to about 800 million standard cubic meters per day in 2019, a significant amount of power can be generated by this method that can reach a value of about 440 MW.

Acknowledgments

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Nomenclatures

$\dot{Q}$:Heat transfer rate (W)
$\dot{W}$:Work rate (W)
$\dot{m}$:Mass flow rate (kg/s)
h: Enthalpy (kJ/kg)
V: Velocity (m/s)
Z: Elevation (m)
P: Pressure (bar)
T: Temperature (K)
L: Power (W)
s: Entropy (kJ/kg K)
x: Mass fraction
C_p: Constant-pressure specific heat(kJ/kg K)
C_v :Constant-volume specific heat (kJ/kg K)
k: The ratio of specific heats (C_p/C_v)

Greek Symbols

η: Efficiency

Subscripts

C. V.: Control volume
T. E.: Turbo-Expander
i: Component
s: Isentropic

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