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Aspen Plus Simulation of Power Generation Using Turboexpanders in Natural Gas Pressure Reduction Stations

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ABSTRACT

This paper investigates the economic feasibility of installing a turboexpander in parallel with the throttling valve of a city gate station for the purpose of distributed electricity generation through exergy recovery from pressurized natural gas. The preheating requirements for preventing hydrate formation due to pressure reduction are provided by the combustion of a small fraction of the outlet natural gas stream. As a case study, the simulation of Tehran No.2 City Gate Station demonstrates an exergy loss of more than 36.5 million kWh per year for the present throttling valves. Thermo-economic analyses gives the optimum operating conditions for electricity generation through a turboexpander. Optimization of preheating temperature leads to an exergy recovery of >60%, cost to generate electricity of <$0.04/kWh, and discounted payback period of around ~4 years. Simulation results can be used for designing an automatic preheating temperature control system to optimize exergy recovery via turboexpander under the variable operating conditions of a pressure reduction station.

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1. Introduction

Currently, natural gas (NG) has become the obvious alternative for crude oil because of its ease of operation, less risky transport and lower environmental footprint. These factors lead to increased NG consumption and transport to remote areas. Economic transport of NG requires the reduction of its specific volume through either pressurization or liquefaction. Theoretically, it is possible to recover the energy consumed for volume reduction as the gas reaches its destination. However, the current throttling valves used to reduce NG pressure at the pressure reduction stations within the conventional NG transport system destroy this latent energy via the irreversible Joule-Thompson effect in an isenthalpic process. On the other hand, the isentropic expansion produces the maximum work that is greater than the real expansion due to friction losses. By replacing a throttling valve with a turboexpander in a NG pressure reduction station, the energy content of pressurized gas is partially harnessed and utilized for moving the truboexpander wheel. The mechanical work generated by the expansion turbine can then be used for moving a coupled generator shaft and thus generating electric power.

Recently, harnessing the potential energy of NG distribution system by means of turboexpanders has captured worldwide attention [1-10]. Researchers have developed models to estimate the amount of exergy recovery and the economics of turboexpander installation in NG pressure reduction stations [6, 8]. Rahman (2010) studied power generation from pressure reduction in NG network [11]. Taheri-Seresht et al. (2010) estimated 96% energy recovery and a return on investment of 2 years by installing turboexpanders in Tehran City Gate Station (CGS No. 2) [7]. Howard (2009) proposed a hybrid turboexpander and fuel cell system for NG preheating and power recovery at NG pressure reduction stations with minimum environmental pollutions [12]. Kostowski (2010) developed a thermo-economic model based on exergy analysis to perform a feasibility study on energy recovery within the conventional NG transport system [6]. Farzaneh-Gord and Sadi (2008) analyzed different scenarios of using turboexpander in an Iran's city gate station for simultaneous refrigeration and electricity generation [13]. Kostowski and Uson (2013) estimated a remarkable performance ratio (generated power to burnt fuel) of 52.6% for a novel system consisting of an internal combustion engine and an organic Rankine cycle in a NG expansion plant [14]. He and Ju (2013) performed an exergy analysis to design and optimize a liquefaction process using NG pipeline pressure energy [15].

When NG is subjected to temperature and pressure variations, the water and hydrocarbon molecules are prone to form solid hydrate molecules. The formation of these solids are worrying because they can cause potential limitation in flow and abrasion of equipment. The allowable water content of NG is usually determined by the process equipment. This is of particular importance to high-speed machinery like turboexpanders. To avoid hydrate formation at the outlet of the turboexpander, the inlet stream must be preheated. The current practice in a pressure reduction station is to increase the gas temperature by burning a small fraction of NG in a preheater upstream of the expansion process so that the final outlet gas temperature at low pressure is kept outside the hydrate formation region. As the preheater is the most important energy consumer of a pressure reduction station, controlling its performance is of technical and economic significance. Khalili *et al.*(2011) calculated less than 47% thermal efficiency for indirect water bath heaters that are typically installed in Iran's NG pressure reduction stations [16]. In order to reduce the NG consumption, Azizi et al. (2014) suggested using the flue gas of an indirect water bath heater to partially preheat NG and found an 11% improvement of thermal efficiency [17]. Farzaneh-Gord *et al.* (2011) performed a feasibility study on the application of solar heaters together with uncontrolled linear heaters in NG pressure drop stations [18]. Ashouri et al. (2014) found that using a controller to adjust the preheating temperature results in 43% savings on preheater energy consumption and less than one year return on investment [19]. Pozivil (2004) used Hysys to find that for a given pressure ratio, the temperature difference caused by a turboexpander is much higher than that of a throttling valve [2].

Another significant challenge of NG pressure reduction stations is controlling the temperature under changing pressure and also the flow rate of gas during the hours of a day and days of a year. As the pressure reduction stations are inherently subject to seasonal fluctuations of inlet NG pressure and flow rate, the preheating temperature should be adjusted accordingly. Regardless of inlet temperature, pressure and flow rate of inlet NG, the final outlet pressure must be kept constant while its final temperature must not be less than 5 °C to prevent hydrate formation.

The aim of this paper is to design an optimum expansion process that utilizes expansion turbines to generate electricity from recoverable energy of NG that would otherwise be wasted by the throttling valves of existing pressure reduction station. This study differs from the previous ones by taking into account the effect of fluctuations in the operating conditions of a pressure reduction station causing off-design turboexpander isentropic efficiency as well as optimizing preheating temperature while avoiding hydrate formation. An implication of this research is the optimal design of a turboexpander coupled with an automatic preheating temperature control system which leads to maximum electricity generation under variable operating conditions.

2. Model Development

2.1. Exergy Analysis

According to the second law of thermodynamics, the generated entropy in any real process is equivalent to the loss of exergy. Exergy, by definition, is the maximum useful work that is derived from a specific stream of energy and/or material with respect to its surrounding environment. For exergy analysis of the expansion turbine system, the exergy fluxes crossing the system boundary may comprise the following components:

1. Chemical exergy, bound to the content of combustible components as well as to the composition other than that of the environment. The chemical exergy of NG was calculated after Szargut (2009) for high-methane gas [20]:

$$ex_{Ch}^{NG} = 1.04 \times LHV \tag{1}$$

The calculation of the chemical exergy of flue gases takes into account only the change of the components concentration compared to the environment:

$$ex_{Ch}^{FG} = RT_0 \sum_{i=1} x_i . \ln\left(x_i / x_{i,0}\right) \tag{2}$$

where xi,0 denotes the composition of given species in the environment.

$$ex_{Ph} = h - h_0 - T_0\left(s - s_0\right) \tag{3}$$

2. Physical exergy, resulting from pressure or temperature different from ambient conditions, defined as:

$$ex_{Ph} = h - h_0 - T_0\left(s - s_0\right) \tag{3}$$

where the subscript 0 indicates the state of the fluid under ambient conditions and the enthalpy (h) and entropy (s) are functions of its temperature, pressure and compounds composition.

3. The exergy of work and heat exchanged with the surroundings. While the exergy of work is equal to the work done, the exergy of heat exchanged with a heat source or sink depends on its temperature:

$$ex_{Heat} = q\frac{T - T_0}{T} \tag{4}$$

For systems other than a heat engine, the second-law efficiency is defined as the ratio of recovered exergy to supplied exergy. Since exergy does not obey the conservation law, a quasi-balance of exergy has to be closed taking into account the internal exergy loss (δex):

$$\sum ex^{in} = \Delta ex_{system} + \sum ex^{out} + \delta ex \tag{5}$$

Note that the heat losses to the environment have a zero exergy content. The chemical exergy of the main gas flux is not included as it is not destroyed in the system. Furthermore, the exergy of flue gases is not included as the recovered exergy because they are released to the environment.

The net electric power is estimated as:

$$W_{Gen} = \eta_{GB}.\eta_{Gen}.\eta_T.F_{NG}.\left(h_2^{NG} - h_3^{NG}\right) \tag{6}$$

where η_{GB} is the mechanical efficiency of the gearbox that connects the turboexpander to a generator and η_{Gen} is the electrical efficiency of the generator that are assumed constant for simplicity.

2.2. Economic Analysis

The aim of economic analysis is to study the effect of design parameters on the feasibility of turboexpander installation in a pressure reduction station. In order to take into account the time value of money, the Net Present Value (NPV) and discounted payback period are used [21]. The differential NPV method analyzes the changes of the cash flow elements due to turboexpander installation compared to the existing pressure reduction station. Assuming discrete year-end cash flows and using discrete compounding, the Net Present Value (NPV) is defined as:

$$NPV = -TCI + \sum_{j=1}^{n} \frac{\Delta\left(CF_j\right)}{\left(1+m_{ar}\right)^j} \tag{7}$$

where n is the operation time in years, mar is the minimum acceotable rate of return, TCI is the total capital investment and $\Delta(CF_j)$ is the change in annual cash flow due to turboexpander installation.

The total capital investment consists of fixed capital investment (FCI) and working capital investment (WC) listed in detail as percentage of fixed capital investment in Table A1. See Appendix B for further information on the estimation of purchased installed equipment cost (Eq).

In order to estimate NPV, the change in annual cash flow due to turboexpander installation is obtained. The annual Cash Flow including depreciation for jth year is:

$$CF_j = N_{Pj} + d_j \tag{8}$$

where d_j is the annual depreciation calculated from the straight-line method by dividing Total Capital Investment by operation time:

$$d_j = 0.85 \times TCI/n \tag{9}$$

and N_{Pj} is Net Profit after Taxes calculated as:

$$N_{Pj} = \left(ES_j - TOC_j\right) \times \left(1-\varphi\right) \tag{10}$$

where ES_j is the income from electricity sales, TOC_j is the Total Operating Costs for jth year and φ is the fractional income tax rate.

The Total Operating Costs for jth year is calculated as:

$$TOC_j = DOC_j + FC_j + POV_j + GE_j \tag{11}$$

where DOC_j is Direct Operating Costs, FC_j Fixed Charges, POV_j Plant Overhead Costs and GE_j General Expenses. Table A2 of Appendix A summarizes a detailed list of all the operating costs with their typical ranges [21] and selected values [22] for a pressure reduction station. The costs of raw material and utilities include NG and electricity. While the price of NG is expected to vary with time, constant NG price throughout the entire operation time and equal electricity sale and purchase prices are assumed in our economic analysis. According to Iran Ministry of Power (https://www.tasnimnews.com) and National Iran Gas Company (http://www.iraniangas.ir), the average electricity and NG prices in year 2016 are \$0.05/kWh and \$0.04/Nm^3 (~\$1.22/MMBtu), respectively. Furthermore, we assume that the pressure reduction station is operating for 90% days of a year, 35% of a worker time with an average labour cost of 25.58 \$/h is spent for the operation of a turboexpander [21]. Note that different costs are updated by Chemical Engineering Plant Cost Index (CEPCI) as reported in Appendix B.

Therefore, the change in annual cash flow due to turboexpander installation is:

$$\Delta\left(CF_j\right)=\Delta\left(N_{Pj}\right)+\Delta\left(d_j\right)=\left(ES_j-\Delta\left(TOC_j\right)\right)\times\left(1-\varphi\right)+\Delta\left(d_j\right) \quad (12)$$

where $\Delta(TOC_j)$ and $\Delta(d_j)$ are the change in annual Total Operating Costs and Depreciation due to turboexpander installation. Obviously, the income from electricity sales is zero for the current pressure reduction station that operates with throttling valves.

Assuming Total Capital Investment is provided by internal resources (i.e. no bank loans), straight-line depreciation and constant change in annual cash flow, we obtain:

$$NPV=-TCI+\Delta\left(CF_j\right).\sum_{j=1}^{n}\frac{1}{\left(1+m_{ar}\right)^j} \quad (13)$$

The ratio of Net Present Value to Total Capital Investment shows the relative feasibility of an investment:

$$NPVR=NPV/TCI \quad (14)$$

where $NPVR>0$ shows the investment is economically justified.

Another important economic indicator is the discounted payback period calculated by putting $r=mar$ and solving for the time necessary to achieve $NPV=0$, i.e. n. Thus:

$$NPV=0\rightarrow TCI=\Delta\left(CF_j\right).\sum_{j=1}^{n}\frac{1}{\left(1+m_{ar}\right)^j} \quad (15)$$

3. Results and Discussion

Tehran No. 2 City Gate Station (CGS No. 2) is selected for a case study because of its continuous NG flow throughout the year, as well as its sufficient space for installation of new equipment and its closure to industrial and residential areas with high demand for electricity consumption. Because of the real and non-polar nature of NG components reported in Table 1, the thermodynamic calculations are based on Peng-Robinson equation of state [23]. Furthermore, the NG physical properties and hydrate formation curve estimated by Aspen Hysys V.10 are given in Table 2 and Figure 1, respectively.

Table 1. Chemical composition of NG at CGS No. 2

Species	N_2	CH_4	CO_2	C_2H_6	C3H8	iC_4H_{10}	nC_4H_{10}	iC_5H_{12}	nC_5H_{12}	C_6H_{14}	Sum
Mole %	3.70	1.10	89.80	3.70	0.98	0.22	0.29	0.10	0.07	0.04	100.0

Table 2. Physical properties of NG at CGS No. 2.

Molecular Weight	Specific gravity	Density (15 °C & 1 atm)	Mass LHV
17.925 g/mol	0.62	0.760 kg/m^3	45431.84 kJ/kg

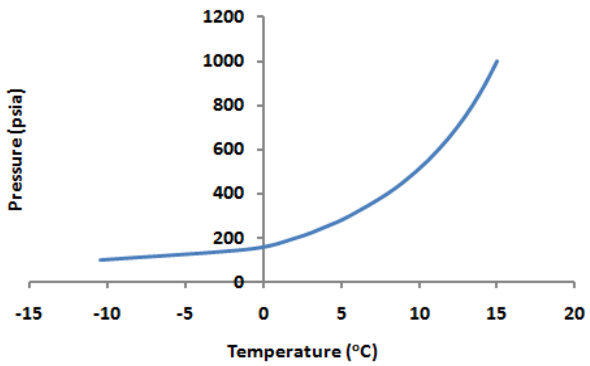

Figure 1. Pressure-temperature curve for hydrate formation of NG.

The average monthly operating conditions for Tehran No.2 City Gate Station are given in Table 3. While the NG outlet pressure and outlet temperature do not change appreciably, the inlet pressure and volumetric flow rate are subject to significant variations due to seasonal changes. The monthly inlet gas temperature can be estimated on the basis of inlet gas pressure and hydrate formation curves (see Figure 1). Given the monthly inlet and outlet gas pressures reported in Table 3, the inlet and outlet gas temperatures are required to be higher than 15°C and 5°C to avoid hydrate formation. Because of the CO_2 content of NG, hydrate can form at higher temperatures. For a conservative design, inlet and outlet gas temperatures of 25°C and 10°C are used. Table 4 reports the physical exergy loss by existing throttling valves for each month of the year. The total exergy loss of more than 36’500’000 kWh per year in only one of Iran’s city gate stations reveals the significant potential for exergy recovery through turboexpander installation. Table 5 includes the model input parameters used for the energy, exergy and economic analyses done by Aspen Hysys V.10 simulations.

Table 3. Monthly operating conditions of NG at CGS No. 2.

The months of year (in Iranian calendar)	Inlet pressure (MPa)	Outlet pressure (MPa)	Outlet temperature (°C)	Volumetric flow rate (Nm^3/h)
The first month of spring (1)	4.4	1.7	7.7	176880
The second month of spring (2)	4.6	1.7	10.7	104105
The third month of spring (3)	4.9	1.7	8.0	170514
The first month of summer (4)	5.3	1.7	8.7	154239
The second month of summer (5)	5.3	1.7	8.8	132929
The third month of summer (6)	4.9	1.7	10.2	184663
The first month of autumn (7)	4.2	1.7	10.4	178689
The second month of autumn (8)	3.9	1.8	10.4	262381
The third month of autumn (9)	2.4	1.8	10.8	346470
The first month of winter (10)	2.3	1.7	10.9	427071
The second month of winter (11)	3.2	1.8	10.3	398845
The third month of winter (12)	3.0	1.8	10.4	355395
Average	4.07	1.76	9.8	241015

Table 4. Physical exergy loss in Tehran No.2 City Gate Station.

The months of year (in Iranian calendar)	Inlet exergy (kJ/kmol)	Outlet exergy (kJ/kmol)	Exergy loss (kJ/kmol)	Exergy loss (kWh)
The first month of spring (1)	9114.16	6947.24	2166.92	3133454.27
The second month of spring (2)	9202.13	6940.72	2261.40	1924649.31
The third month of spring (3)	9379.39	6946.53	2432.85	3391390.12
The first month of summer (4)	9522.44	6944.93	2577.52	3250107.79
The second month of summer (5)	9522.44	6944.70	2577.74	2801309.10
The third month of summer (6)	9379.39	6941.72	2437.67	3680072.77
The first month of autumn (7)	9025.82	6941.31	2084.50	3045100.97
The second month of autumn (8)	8824.82	7025.31	1799.51	3860004.45
The third month of autumn (9)	7720.05	7123.15	596.90	1690718.71
The first month of winter (10)	7631.52	7005.93	625.59	2184190.32
The second month of winter (11)	8402.34	7132.90	1269.44	4139217.76
The third month of winter (12)	8276.89	7097.46	1179.42	3426753.49
Sum				36'526'969

Table 5. Model input parameters for simulations.

Inlet design pressure	6.89 MPa	Ambient temperature	25°C
Outlet design pressure	1.72 MPa	Flue gas temperature	120°C
Inlet design temperature	25°C	Excess air	15%
Outlet design temperature	10°C	Min acceptable rate of return	6%
Heater outlet design temperature	40°C	Fractional income tax rate	20%
Design volumetric flow rate	600,000 Nm^3/h	Project operation time	15 yr
Min-max turbine efficiency	10% - 90%	Electricity price	$ 0.05/kWh
Heat exchanger efficiency	90%	NG price	$ 1.22/MMBtu
Electric heater efficiency	100%	Operating time	0.9 × 365 days

Figure 2 illustrates the process flow diagram of CGS No. 2. As seen, the current pressure reduction station consists of three parallel identical lines each of which equipped with an electric heater and a throttling valve. The economic analysis of the current city gate station provides a basis of comparison with the proposed model. As no electricity is being generated in the current station, its revenue from electricity sale is zero. On the other hand, the current city gate station purchases electricty from an outside supplier to provide the gas preheating requirements.

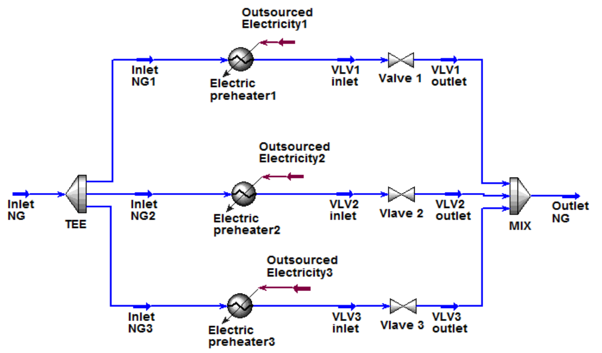

Figure 2. Process flow diagram of CGS No.2 in Aspen Hysys V.10.

Figure 3 shows the process flow diagram for the proposed model, i.e. turboexpander installation in parallel with the existing throttling valve. For simplicity, only one of the three lines of Tehran No. 2 City Gate Station is shown. In our model, instead of the current electric heater, a combination of a boiler and a gas-gas heat exchanger is used to preheat the NG. A fraction of the outlet gas stream is recyled and combusted with air in a boiler and the hot flue gas is used to exchange heat with cold NG inlet stream. A conversion reactor in Aspen Hysys V.10 is used to simulate the complete combustion of NG in the boiler. The logical operators in Figure 3 include.

- SET-1 sets the molar flow rate of inlet air assuming complete combustion of NG with 15 mole % excess air.
- SET-2 sets 10% heat loss from the boiler thermal power assuming a boiler efficiency of 90%.
- ADJ-1 adjusts the fraction of burned NG until the flue gas temperature is converged to a set point, here 120°C.
- ADJ-2 adjusts the fraction of inlet gas that bypasses through the throttling valve until the outlet gas mixture temperature is converged to a set point. Normally, to prevent hydrate formation and optimize energy consumption, the outlet temperature of pressure reduction stations is set to 10°C which is also a suitable temperature for NG combustion efficiency.

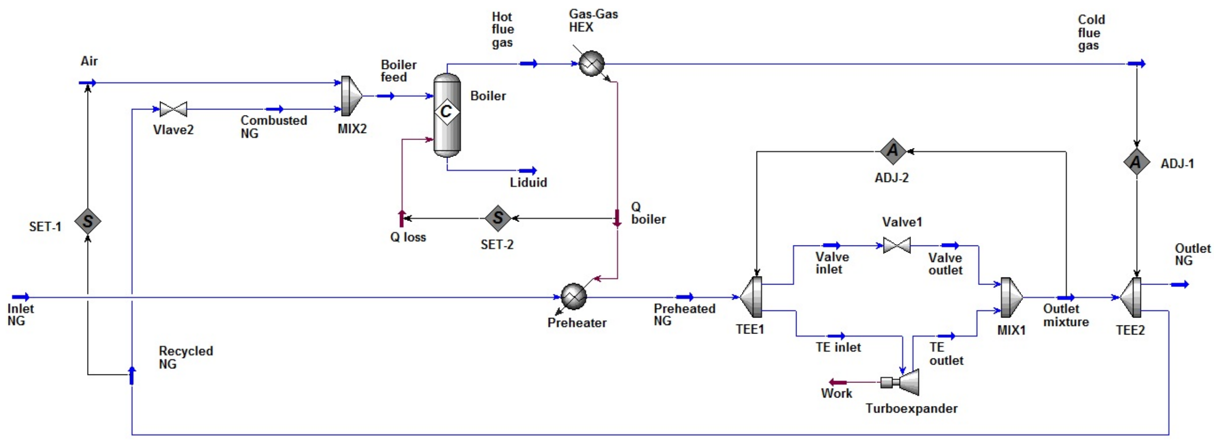

Figure 3. Simulation of turboexpander installation in Tehran No. 2 City Gate Station with Aspen Hysys V.10.

Sensitivity analysis are carried out on design specifications, i.e. preheating temperature and turbine isentropic efficiency. As seen in Figure 4, the fraction of NG that is combusted in the boiler increases with increasing preheating temperature, but is independent of turboexpander efficiency. As seen in Figure 3, the inlet gas flow is divided into two parallel streams. One stream passes through the turboexpander, while the other bypasses through the throttling valve. The fraction of inlet NG flow passing through the turboexpander is adjusted so that the outlet gas mixture temperature is ≥10°C. For a given pressure drop, the temperature drop caused by the turboexpander is much larger than that of the throttling valve. Given the outlet gas mixture temperature ≥10°C, the fraction of NG stream that can pass through the turboexpander is increased by increasing the preheating temperature and/or decreasing the turboexpander isentropic efficiency as seen in Figure 5. In other words, for every turboexpander isentropic efficiency, there is a preheating temperature beyond which the entire gas flow can pass through the turboexpander.

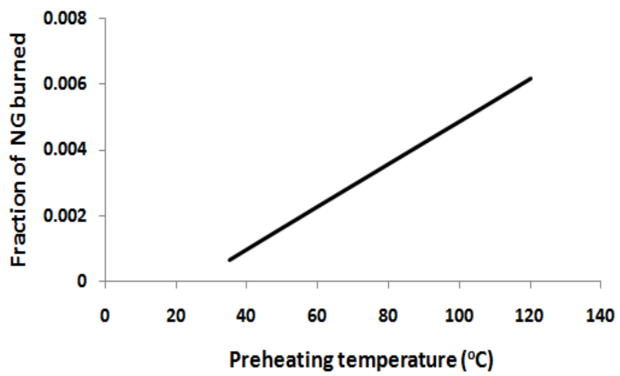

Figure 4. Fraction of burned NG as a function of preheating temperature.

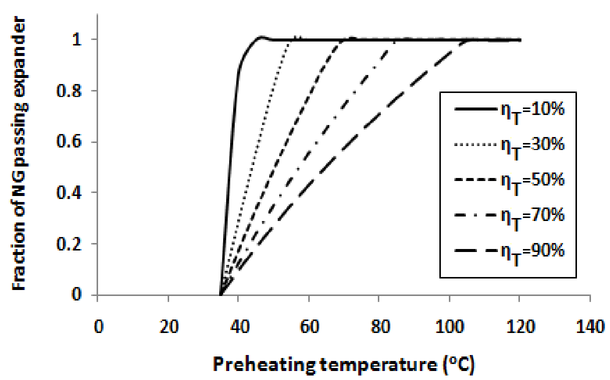

Figure 5. Fraction of total NG through turboexpander as a function of preheating temperature and turboexpander efficiency.

The results of energy and exergy analyses are shown in Figures 6 to 8. As seen in Figure 6, the specific work generated by the turboexpander is increased by increasing its isentropic efficiency and/or increasing preheating temperature. Furthermore, for each turboexpander isentropic efficiency, there is a minimum preheating temperature below which hydrate formation occurs in the turboexpander outlet stream, i.e. the turboexpander outlet gas temperature must be less than its hydrate formation temperature. Therefore, at higher turboexpander isentropic efficiencies, greater preheating temperatures are required to avoid hydrate formation.

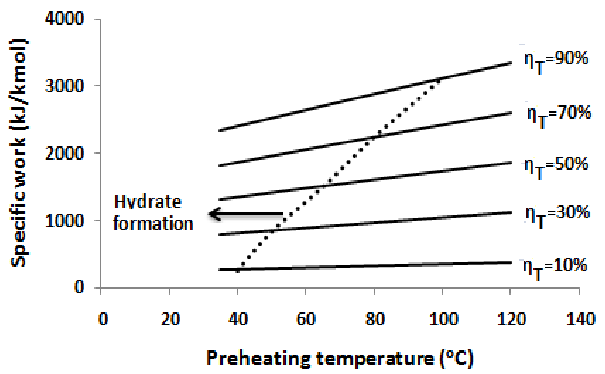

Figure 6. Specific work generated by turboexpander as a function of design specifications. Preheating temperatures below the dotted curve lead to hydrate formation in turboexpander outlet stream.

Figure 7 illustrates the generated work to burned NG ratio as a function of design specifications. Obviously, the work-to-fuel ratio increases by increasing the turboexpander isentropic efficiency. Furthermore, for each turboexpander isentropic efficiency, there is a preheating temperature where the work-to-fuel ratio is maximum. The optimum preheating temperature occurs exactly when the preheating temperature is high enough to allow the passage of the entire NG stream through the turboexpander. Below this temperature, a fraction of NG stream has to bypass through the throttling valve to maintain the outlet gas mixture temperature at 10°C. Preheating at temperatures greater than the optimum temperature leads to overheating of turboexpander inlet gas stream and therefore outlet gas mixture temperatures that are unnecessarily greater than 10°C. Note that by operating at preheating temperatures equal to or higher than the optimum preheating temperature, hydrate formation is avoided in the turboexpander outlet stream.

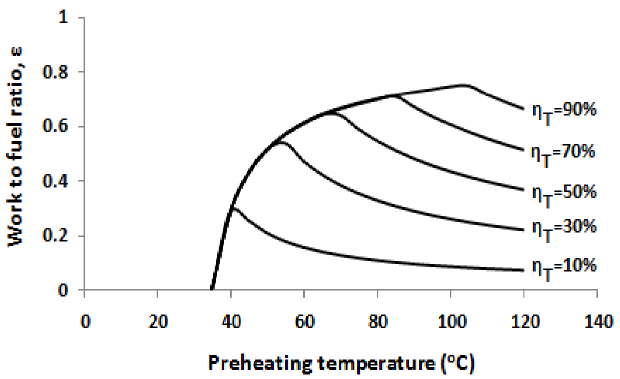

Figure 7. Generated work to burned NG ratio as a function of design specifications.

The effect of design specifications on second-law efficiency of the system is shown in Figure 8. Clearly, greater turboexpander isentropic efficiencies lead to increased exergy recovery from NG inlet stream and thus increased second-law efficiency. As with the work-to-fuel ratio, for each turboexpander isentropic efficiency, there is a preheating temperature where the second-law efficiency is maximum. This optimum preheating temperature occurs just as the entire NG flow can pass through the turboexpander in order to main the outlet gas mixture temperature at 10°C. Below this temperature, a fraction of NG has to bypass through the throttling valve resulting in less exergy recovery by the turboexpander. Above the optimum preheating temperature, the second-law efficiency drops rapidly due to exergy loss associated with NG preheating that is provided by heat exchange with hot flue gases. This shows that an alternative energy source with lower temperature could increase the second-law efficiency. That is why water is used as an indirect heating medium in most city gate stations.

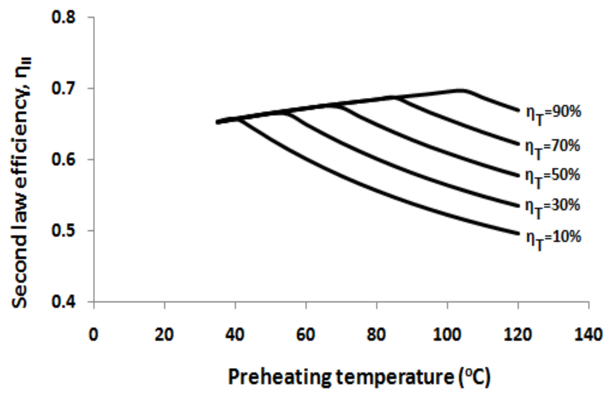

Figure 8. Second-law efficiency as a function of design parameters.

The optimum preheating temperature for each turboexpander isentropic efficiency is illustrated in Figure 9. As the turboexpander isentropic efficiency can be expressed as a function of inlet NG pressure and mass flow rate [4], the preheating temperature can be optimized based on these operating conditions.

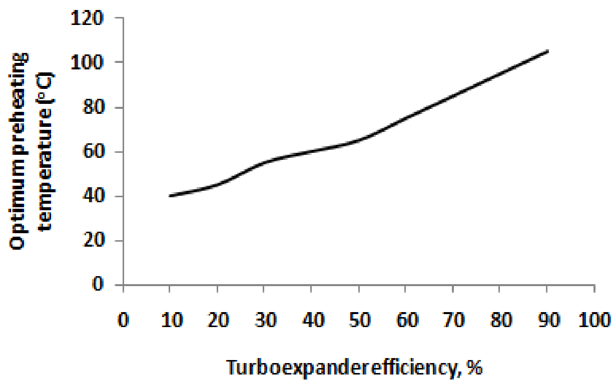

Figure 9. Optimum preheating temperature as a function of turboexpander isentropic efficiency.

The results of economic analyses based on constant operating conditions throughout the year are shown in Figures 10 to 12. As seen in Figure 10, while NPVR increases with increasing turboexpander isentropic efficiency, it sharply decreases with increasing preheating temperature due to the increased NG consumption. Figure 11 shows the number of years required for return on investment in turboexpander installation project as a function of design parameters.

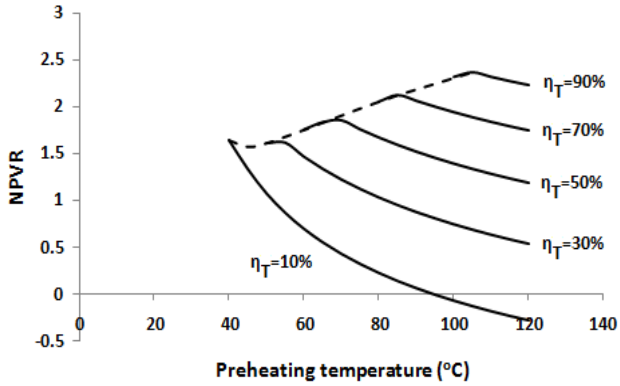

Figure 10. Net Present Value Ratio as a function of design parameters.

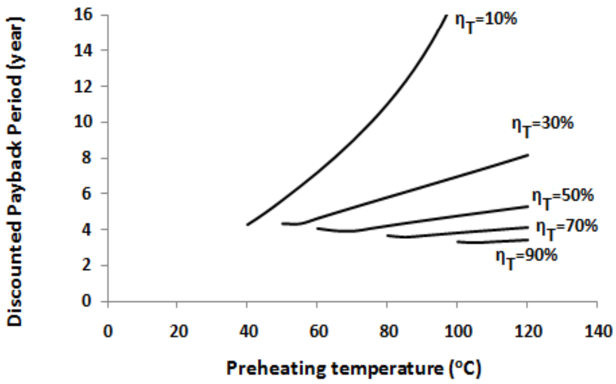

Figure 11. Discounted payback period as a function of design parameters.

Figure 12 shows the cost to generate electricity (i.e. total operating costs ($/yr) divided by annual electricity generation rate (kWh/yr) as a function of design specifications. As expected, the cost to generate electricity increases with increasing preheating temperature and decreasing turboexpander isentropic efficiency. Furthermore, it seems that for turboexpander isentropic efficiencies ≥30%, the cost of generated electricity is always less than the electricity price reported in Table 5.

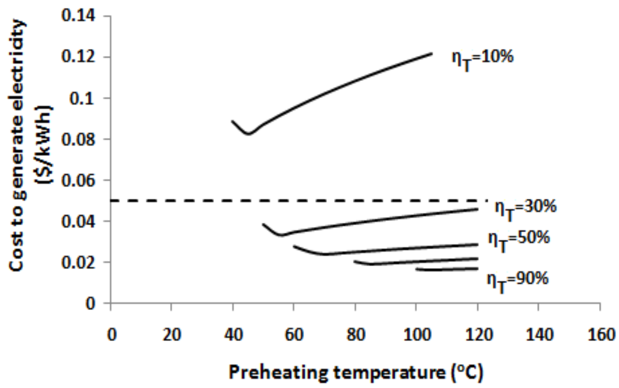

Figure 12. Cost to generate electricity as a function of design specifications. Dashed line is the electricity price.

Note that the large variations in NG inlet pressure and mass flow rate forces a turboexpander to operate away from its design point and, therefore, with a lower efficiency. It is clear that the change in turboexpander efficiency affects the optimum preheating temperature, the rate of power generation, NG consumption and thus project economics. Therefore, it is highly recommended to also optimize the turboexpander design specifications so that it could operate efficiently throughout the year. In particular, using a radial inflow turbine in a pressure reduction station is highly recommended to widely control the flow through the expander.

4. Conclusions

Installing a turboexpander in parallel with the current throttling valves of Tehran No.2 City Gate Station (CGS No.2) and burning a fraction of NG to avoid hydrate formation at the outlet stream proved to be economically attractive because of the lower prices of NG compared to electricity prices that is currently used in the electric preheaters of CGS No.2. Assuming constant yearly operating conditions, a discounted payback period of around 4 years and Net Present Value Ratio of >1 is estimated based on Iranian NG and electricity prices in 2016. Energy, exergy and economic analyses reveal great potential for exergy recovery in

CGS No.2 which suffers from an outstanding total exergy loss of 36'500'000 kWh per year. In general, this potential is highly dependent on available pressure drop and NG flow rate of a city gate station that is subject to daily and monthly variations. Operating at optimum preheating temperature is particularly important to set the basis for automatic temperature control system especially under variable operating conditions. To improve the economics of the process, the turboexpander design specifications should also be optimized according to the variable operating conditions of a city gate station.

Acknowledgment

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Nomenclature

CEPCI	Chemical Engineering plant cost index, -
CFj	Annual cash Flow for j^{th} year, $/yr
dj	Annual depreciation for j^{th} year, $ /yr
DOCj	Annual direct operating costs for j^{th} year, $/yr
ESj	Annual income from electricity sales for j^{th} year, $/yr
Eq	Purchased installed equipment cost, $
ex	Exergy, kJ/kmol
FCj	Annual fixed charges for j^{th} year, $/yr
FCI	Fixed capital investment, $
F_{NG}	Natural gas molar flow rate, kmol/s
GEj	Annual general expenses for j^{th} year, $/yr
h	Enthalpy, kJ/kmol
LHV	Lower heating value, kJ/kmol
mar	Minimum acceptable rate of return, -
n	Operation time, yr
NPj	Annual net profit after taxes for j^{th} year, $/yr
NPV	Net Present Value, $
P	Electric/thermal power, kW
POVj	Annual plant overhead costs for j^{th} year, $/yr
q	Heat, kJ/kmol
R	Universal ideal gas constant, 8.314 J/mol.K
s	Entropy, kJ/kmol.K
T	Temperature, K
TCI	Total Capital Investment, $
TOCj	Annual total operating costs for j^{th} year, $/yr
W_{Gen}	Generated power, kW
WC	Working capital investment, $
xi	Molar composition of species i, -

Greek

ε	Work-to-fuel ratio, -
η	Efficiency, -
φ	Fractional income tax rate, -

Subscripts

Ch	Chemical
FG	Flue gas
GB	Gear box
Gen	Generator
i	Species number, -
j	Year number, -
NG	Natural gas
Ph	Physical
T	Turbine

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Appendix A

Table A1. Estimation of total capital investment for pressure reduction station.

		Typical ranges of FCI, % [21]	Selected % of FCI	Normalized % of FCI
Direct Costs	Purchased delivered equipment	15-40	30	25.00
	Purchased equipment installation	6-14	10	8.33
	Instrumentations and controls	2-12	9	7.50
	Piping	4-17	13	10.83
	Electrical systems	2-10	7	5.83
	Buildings	2-18	4	3.33
	Yard improvements	2-5	3	2.50
	Service facilities	8-30	10	8.33
	Land	1-2	0	0.00
Indirect Costs	Engineering and supervision	4-20	12	10.00
	Construction expenses	4-17	6	5.00
	Legal expenses	1-3	2	1.67
	Contractor's fee	2-6	3	2.50
	Contingency	5-15	10	8.33
	HSE modification	1	1	0.83
Fixed Capital Investment = Direct Costs + Indirect Costs			120	100.00
Working Capital = 15/85*(Fixed Capital Investment)				
Total Capital Investment = Fixed Capital Investment + Working Capital				

Table A2. Estimation of total operating costs for pressure reduction station.

<table>
<tr><th colspan="2"></th><th>Typical ranges [21]</th><th>Selected values [22]</th></tr>
<tr><td rowspan="8">Direct Operating Costs
(=66% TOC)</td><td>Raw materials</td><td>10-80% TOC</td><td>Annualized NG cost</td></tr>
<tr><td>Operating labour (OL)</td><td>10-20% TOC</td><td>35% of a worker</td></tr>
<tr><td>Direct supervisory and clerical (Sup)</td><td>10-20% OL</td><td>20% OL</td></tr>
<tr><td>Utilities (electricity, cooling water, etc.)</td><td>10-20% TOC</td><td>Annualized electricity cost</td></tr>
<tr><td>Maintenance and repairs (M&R)</td><td>2-10% FCI</td><td>4% annualized FCI</td></tr>
<tr><td>Operating supplies</td><td>10-20% M&R</td><td>15% M&R</td></tr>
<tr><td>Laboratory charges</td><td>10-20% OL</td><td>15% OL</td></tr>
<tr><td>Patents and royalties</td><td>0-6% TOC</td><td>0</td></tr>
<tr><td rowspan="5">Fixed Charge
(=10-20% TOC)</td><td>Depreciation</td><td>Linear</td><td>(0.85*TCI) / n</td></tr>
<tr><td>Local taxes (property)</td><td>1-4% FCI</td><td rowspan="4">3% annualized TCI</td></tr>
<tr><td>Insurance</td><td>0.4-1% FCI</td></tr>
<tr><td>Rent</td><td>0</td></tr>
<tr><td>Financing (interest)</td><td>Own capital investment</td></tr>
<tr><td>Plant Overhead Costs
(=5% TOC)</td><td>Safety, protection, restaurant, etc</td><td>5-15% TOC</td><td>60% (M&R+ OL+ Sup)</td></tr>
<tr><td colspan="4">Operating Costs = Direct Operating Costs + Fixed Charges + Plant Overhead Costs
(=70-85% TOC)</td></tr>
<tr><td rowspan="3">General Expenses
(=15-25 % TOC)</td><td>Administrative costs</td><td>2-5% TOC
or
15-25% OL</td><td rowspan="3">20% OL</td></tr>
<tr><td>Distribution & marketing</td><td>2-20% TOC</td></tr>
<tr><td>Research & development</td><td>5% TOC</td></tr>
<tr><td colspan="4">Total Capital Investment = Fixed Capital Investment + Working Capital</td></tr>
</table>

Appendix B

The purchased installed cost of the current electric heaters is estimated from [21] that is updated using Chemical Engineering Plant Cost Index :

$$Eq = 306.6 \times \left[P(kW) \right]^{0.852} \times \frac{CEPCI\ 2016}{CEPCI\ 2002} \quad \text{(B1)}$$

For turboexpander and boiler two methods are used:

1. Purchased turboexpander prices as a function of its delivered power within the range of 10 to 10,000 kW [21]:

$$Purchased\ Price\ of\ Turbine\ in\ 2016, USD = 2573 \times \left[P(kW) \right]^{0.617} \times \frac{CEPCI\ 2016}{CEPCI\ 2002} \quad \text{(B2)}$$

•

According to Table A.1, the purchased equipment installation cost is almost one-third of its purchased price. Therefore, the purchased price of equipment is 75% of its purchased installed price.

2. 2) The power scale method estimates the installation costs of turboexpander and boiler based on prices of reference devices with known electric/thermal powers [6]:

•
• Purchased installed price of reference turboexpander:

$$\text{Gascontrol: } Eq_{ref} = 45958.8\,\$, P_{ref} = 15\ kW \text{ Electric power} \quad \text{(B3)}$$

Purchased installed price of reference boiler:

$$\text{De Dietrich: } Eq_{ref} = 86216.8\,\$, P_{ref} = 280\ kW \text{ Thermal power} \quad \text{(B4)}$$

$$Eq = Eq_{ref} \left(\frac{P}{P_{ref}} \right)^{a} \times \frac{CEPCI\ 2016}{CEPCI\ 2009} \quad \text{(B5)}$$

where a=0.6 for turboexpander and a=0.73 for boiler [6].

Note that in the above economic analysis, the arithmetic mean of purchased installed price of turboexpander is used.

According to "www.chemengonline.com", Chemical Engineering Plant Cost Index for the years required in this research are listed in Table B1.

Table B1. Chemical Engineering Plant Cost Index

Year	2001	2002	2009	2016
CEPCI	394.3	395.6	521.9	541.7

شبیه سازی تولید توان الکتریکی با استفاده از توربین های انبساطی در ایستگاه های تقلیل فشار گاز طبیعی با استفاده از نرم افزار Aspen Plus

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چکیـــــــده

این مقاله امکان سنجی اقتصادی تولید برق پراکنده از بازیافت اکسرژی جریان گاز طبیعی تحت فشار از طریق نصب توربین انبساطی به موازات شیر فشار شکن در ایستگاه تقلیل فشار شهری را مورد مطالعه قرار می دهد. احتیاجات پیش گرمایشی مورد نیاز برای جلوگیری از تشکیل هیدرات بر اثر افت فشار توسط احتراق کسر کوچکی از جریان گاز طبیعی خروجی تامین می گردد. به عنوان مطالعه موردی، شبیه سازی ایستگاه دروازه شهری شماره ۲ شهر تهران بیانگر اتلاف اکسرژی بیش از ۳۶/۵ میلیون کیلووات ساعت در سال توسط شیرهای فشار شکن موجود است. آنالیزهای ترمودینامیکی و اقتصادی، شرایط عملیاتی بهینه برای تولید برق توسط توربین انبساطی را بدست می دهند. بهینه سازی دمای پیش گرمایش منجر به بیش از ۰۶ درصد بازیافت اکسرژی، هزینه برق تولیدی کمتر از ۴۰،۰ دلار بر کیلووات ساعت و زمان بازگشت سرمایه حدود ۴ سال می گردد. از نتایج شبیه سازی ها می توان برای طراحی سیستم کنترل خودکار دمای پیش گرمایش به منظور بهینه سازی بازیافت اکسرژی توسط توربین انبساطی تحت شرایط عملیاتی متغیر ایستگاه تقلیل فشار گاز استفاده نمود.

واژگان کلیدی: گاز طبیعی، ایستگاه تقلیل فشار، توربین انبساطی، شبیه سازی فرایند.