IJE TRANSACTIONS C: Aspects Vol. 27, No. 12, (December 2014) 1815-1822

International Journal of Engineering

Journal Homepage: www.ije.ir

Operation Analysis of Rotary Tools of Compressor Station Using Exergy Approach

M. Mohamadi-Baghmolaei[a], J. Sargolzaei*[a], F. Tabkhi[b]

[a] Department of Chemical Engineering, Ferdowsi University of Mashhad, Mashhad, Iran
[b] Department of Chemical Engineering, Persian Gulf University, Bushehr, Iran

PAPER INFO

Paper history:
Received 09July 2013
Received in revised form 15May 2014
Accepted 14August 2014

Keywords:
Exergy Analysis
Compressor Station
Turbine
Fuel Consumption
Efficiency

ABSTRACT

In this study, operation of compressor station has been investigated by exergy approach. Exergy analysis is a thermodynamic method which shows the irreversibility of a system quantitatively. Gas compressors are used to compensate the pressure drop along the gas pipeline significantly. The compression process causes temperature rise of gas; in this regard gas cooler is applied to reduce the temperature. The compressors are run at variable speeds; fuel consumed in combustion chamber is changed accordingly. The case study is focused on transmission of sour gas through IGAT5 pipeline, located in south of Iran.Exergy destruction, exergetic efficiency, fuel consumption of station and effect of each elements of station on total destruction is measured for speed range of 5200 to 7000 rpm of compressor. It is showed that about 90 precent of destruction is possessed by turbine. Fuel consumption of station depending on compressor speed and increasing trend is seen to start from 0.924 kg/s at 5200 rpm and end in 1.2 kg/s at 7000 rpm.

doi: 10.5829/idosi.ije.2014.27.12c.03

1. INTRODUCTION

Investment and expenses in gas transmission networks are so high and notable, hence a small progress in optimizing the operation of these networks concludes saving a large amount of money. The main issue in gas industries is the long distance between the gas reservoir and gas markets, which causes pressure drop along the pipelines. Compressor stations have been established to overcome pressure decline by providing sufficient energy.

Each compressor station consists of three dominant parts: compressor which is installed to boost the gas, turbine that provides driving force for compressor and cooler to reduce the outlet gas temperature of compressor. Occasionally these units are capable to adjust themselves by a range of throughput received and required compression ratio. That's why in some cases compressors are installed in three patterns: parallel, series or combination.

*Corresponding Author's Email: sargolzaei@um.ac.ir (J. Sargolzaei)

Turbine power usually is provided by fossil fuels or electric energy. In this paper the required energy for boosting the gas, is obtained via portion of natural gas through the pipeline. It should be mentioned that the large volume of the gas flows through the pipeline during the transmission process, so each attempt to quantify the fuel consumption in a station can be used for any fuel minimization and optimization [1]. In this regard, choosing an acceptable and reliable analyzing method that convinces all aspects of the problem is extremely important. Thus a thermodynamic study, exergy analysis, is used to analyze the operation of compressor station main tools. This analysis gives an accurate measurement for any optimization problem concluding minimization of fuel consumption. This method is a good way to find value of irreversibility of a system and understand which part of the system possesses the maximum and minimum irreversibility. Exergy of a system with respect to a reservoir is the maximum work done by the system during a transformation which brings it into equilibrium with surrounding. In the other words, the useful work potential of a system at the specified state is called exergy. Exergy analysis is a useful method in every branch of science considering the work done by Jafarmdar [2] that is the numerical investigation on combustion of air and hydrogen using exergy analysis or improvement of engines operation [3].

There are different methods and techniques to optimize and model the gas networks [4]. As an example Tabkhi et al. [5] have presented a mathematical model for minimizing the total cost of high-pressure natural gas network. Their optimization strategy provides the main design parameters of the pipelines such as pressure, flow rate and fuel consumption. So many methods have been used for optimization and minimization of energy in compressor stations such as work which was done by Borraz-Sánchez and Ríos-Mercado [6]; they used tabu search for their optimization problem. Also exergy analysis is applied for estimating the operation of system; in this regard an exergy based analysis on gas transmission systems was carried out by Evanko [7] to estimate the quality of energy utilization. The second-law efficiency of an arbitrary selected structure of gas transmission system was calculated under full load operation of compressor station and the gas was assumed to be ideal. Chaczykowski et al. [8] have continued using this method to detect the highest exergetic efficiency and the lowest exergetic destruction of overall compression when the electric coolers wereloaded partly or fully by real gas consideration. Zanchini and Terlizzese [9] made an important effort to find the expression for molar exergy and flow exergy of chemical fuels. Exergy analysis and operation improvement of the turbo-compressor package and its air cooler is the objective of this study. Providing useful information to improve gas transmission is an important duty.

2. THERMODYNAMIC AND MATHEMSTICAL MODEL

In this part, a comprehensive model which includes turbine and compressor under steady state condition in a specific boosting station is taken into account. Fuel consumption of turbine is determined by getting help of characteristic curves of turbine and compressor which are unique for each of them. Finally exergy analysis of system can be possible with this thermodynamic model.

2. 1. Compressibility Factor and Other Properties

Traditionally compressibility factor of natural gas as a real gas is calculated by equation of states, otherwise it can be estimated by empirical equations [10]. Here this factor is expressed in the form of the following EOS based on corresponding states theory:

$$\boldsymbol{Z = 1 + \left(0.257 - 0.533\left(\frac{T_c}{T}\right)\right)\frac{p}{p_c}} \quad (1)$$

The pseudo-critical properties of natural gas, T_c and, P_c can be calculated using the simple mixing rules using critical properties of natural gas components.

A method which is described below, utilizes an adjustment factor depending on the amount of carbon dioxide and hydrogen sulfide existing in the sour gas [11]:

$$\varepsilon = 66.67(A^{0.9} - A^{1.6}) + 8.33(B^{0.5} - B^4) \quad (2)$$

where A is the sum of the mole fractions of CO_2 and H_2S, while B is the mole fraction of H_2S.

The pseudo-critical temperature is modified to get the adjusted pseudo-critical temperature T'_{pc} from Equation (3):

$$T'_{pc} = T_{pc} - \varepsilon \quad (3)$$

Similarly, the pseudo-critical pressure is adjusted as follows:

$$P'_{pc} = \frac{p_{pc} \cdot T'_{pc}}{T_{pc} + B(1-B)\varepsilon} \quad (4)$$

Ideal heat capacity can be obtained from Perry's handbook [12] for each component of natural gas.
The real heat capacity, C_p, is estimated by Equation (5) as a function of ideal heat capacity, temperature, and pressure, and gas specific volume, v:

$$C_p = C_p^0 - T\int_0^p \left(\frac{\partial^2 v}{\partial^2 T}\right)_p dp \quad (5)$$

Entropy and enthalpy changes are computed by integration from initial state to final state:

$$s(T_2, p_2) - s(T_1, p_1) = \int_{T_1}^{T_2} \frac{C_p^0}{T} dT - \int_{p_1}^{p_2} \frac{R}{p}\left[Z + T\left(\frac{\partial Z}{\partial T}\right)_p\right] dp \quad (6)$$

$$h(T_2, p_2) - h(T_1, p_1) = \int_{T_1}^{T_2} C_p^0 dT - \int_{p_1}^{p_2} \frac{RT^2}{p}\left(\frac{\partial Z}{\partial T}\right)_p dp \quad (7)$$

where C_p^0, R, and Z are ideal heat capacity, universal gas constant, and compressibility factor, respectively.

2. 2. Compressor Calculations and Characteristic Curves

Centrifugal compressor can be characterized by means of its delivered flow rate and its pressure ratio, the ratio between suction pressure and discharge pressure.

The compressor performance can be obtained from a detailed mathematical model based on mechanical conservation laws [13] but due to complexity of the models and amount of initial data required, characteristic curves are generally constructed by vendors.

The compressors within the station are modeled using compressor map-based polynomial equation. Normalized head, h_p/ω^2, and normalized flow rate, Q_s/ω, are used to describe the performance map of compressors where ω is the rotational speed of compressor [14]. The normalized head is obtained by curve fitting for compressor and is shown in Equation (8):

$$\frac{h_p}{\omega^2} = a_1 + a_2\frac{Q_s}{\omega} + a_3(\frac{Q_s}{\omega})^2 \quad (8)$$

Also second degree polynomial is suggested to calculate the polytropic efficiency:

$$\eta_p = a_4 + a_5\frac{Q_s}{\omega} + a_6(\frac{Q_s}{\omega})^2 \quad (9)$$

Turbines can be characterized like compressors and their performance can be obtained from their characteristic curves and usually provided by manufacturer.

Polytropic head on the characteristic curve is a function of compression ratio where M is the molecular weight of gas mixture and n is polytropic exponent:

$$\mathbf{h_p} = \frac{\mathbf{Z_sRT_s}}{\mathbf{M}}\frac{\mathbf{n}}{\mathbf{n-1}}[\left(\frac{\mathbf{p_d}}{\mathbf{p_s}}\right)^{\frac{\mathbf{n-1}}{\mathbf{n}}} - \mathbf{1}] \quad (10)$$

In the aforementioned equation, compressibility factor and temperature are in the suction state. Isentropic exponent is the function of compressibility factor and real heat specific capacity by considering Uilhoorn's study [15].

$$\boldsymbol{k} = \left[\mathbf{1} - \frac{\boldsymbol{R}}{\boldsymbol{C_p}}(\boldsymbol{Z} + \boldsymbol{T}\left(\frac{\partial \boldsymbol{Z}}{\partial \boldsymbol{T}}\right)_{\boldsymbol{p}})\right]^{-\mathbf{1}} \quad (11)$$

Polytropic exponent is commonly related to isentropic exponent [10]:
By consideringpolytropic efficiency, the outlet temperature of compressed gas can be found [8]:

$$T_d = \frac{T_s}{\eta_p}((\frac{Z_s}{Z_d})(\frac{p_d}{p_s})^{\left(\frac{K-1}{K}\right)} - 1) + T_s \quad (12)$$

where s and d in the aforementioned equation are suction and discharge state, respectively.

2. 3. Gas Cooler To reduce the compressed gas temperature, coolers are installed after compressor. The output temperature of gas cooler can be computed based on effectiveness-number of transfer unit ($\boldsymbol{\varepsilon}$-NTU) method [16]. The effectiveness factor,, of heat exchanger is calculated as follows, where$\dot{Q}$ and h, are the heat transfer rate and enthalpy:

$$\dot{Q} = \dot{Q}_{max} = \dot{m}_c(h_1 - h_2) \quad (13)$$

The product of heat capacity, C_p, and mass flow rate, $\dot{m}$, for the so called minimum florid is minimum, in order to reach the maximum temperature difference [13].

$$\dot{Q}_{max} = C_{min}(T_{gas,in} - T_{air,in}) \quad (14)$$

Both cool and hot fluid can be the minimum fluid, this would be possible when mass flow rate multiple to specific heat capacity is the minimum one.

$$C_{min} = min({}_{air}C_{p,air}, {}_{c}C_p) \quad (15)$$

Effectiveness is a function of geometry of exchanger and flow direction. Gas cooler are ordinary cross-flow exchanger and air jet is allowed to mix. The relation for cross-flow where one fluid is allowed to mix is:

$$= \frac{(1 - e^{(-C(1-e^{(-NTU)}))})}{C} \quad (16)$$

Equation (19) is proper when C_{max} is allowed to mix, if C_{min} permitted to combine, the equation will alter to Equation (17):

$$= 1 - e^{(-C^{-1}(1-e^{(-CNTU)}))} \quad (17)$$

The number of transfer unit and heat capacity ratio is expressed, respectively as below:

$$NTU = \frac{U_{ac}A_{ac}}{C_{min}} \quad (18)$$

$$C = \frac{C_{min}}{C_{max}} \quad (19)$$

where A_{ac} is the air-side total heat transfer surface including both fin area and bare area of tube surfaces, and U_{ac} is the overall heat transfer coefficient.

Here C_{max}, is the maximum value of mass flow rate multiple to heat capacity. It can be computed as below:

$$C_{max} = max({}_{air}C_{p,air}, {}_{c}C_p) \quad (20)$$

Fans efficiency is supposed to be fixed, in this regard for finding air volume, Q, needed for cooling propose, relation between pressure drops, P, and fan speed, N, and its required electric power with its speed, so called fan's laws are applied[1].

2. 4. Exergy Balance Exergy, also called availability or work potential, is the maximum useful work that can be obtained from a system at a given state in a given environment. In other words, the most work can be acquired from a system is called exergy. Note that exergy has the physical dimension of energy.

Exergy analysis is based upon the second law of thermodynamics, which stipulates that all macroscopic processes are irreversible. Every such irreversible process entails a non-recoverable loss of exergy, expressed as the product of the ambient temperature and the entropy generated [17]. Some of the components of entropy generation can be negative, but the sum is always positive.

Exergy can be transferred to or from a system by heat, work, and mass transfer. Then the exergy balance for any system undergoing any process can be expressed more explicitly as follows, where $\dot{x}$ is exergy flow rate in Watt [18]:

[1]Gas Processors Suppliers Association. Engineering Data Book, Section 23: Physical PropertiesTulsa, Oklahoma , (1998)

$$\dot{X}_{heat} - \dot{X}_{work} + \dot{X}_{mass,in} - \dot{X}_{mass,out} - \Delta\dot{X}_{destroyed} = \left(\dot{X}_2 - \dot{X}_1\right)_{cv} \quad (21)$$

Turbo-compressor is considered as an open system which mass is allowed to enter the boundary of system and exit from it. Under steady stateconditions, the termon the right side of equation will be crossed off and Equation (24) will alter to Equation (22):

$$\dot{X}_{heat} - \dot{X}_{work} + \dot{X}_{mass,in} - \dot{X}_{mass,out} = \Delta\dot{X}_{destroyed} \quad (22)$$

$\Delta\dot{X}_{destroyed}$is the exergy destruction in the system. By substituting the rate of heat exergy and mechanical work into Equations (22), it can be calculated as Equation (23), where$\dot{W}$ and T_0 are mechanical work and temperature at ambient air condition, respectively:

$$\Delta\dot{X}_{destroyed} = \sum(1 - \frac{T_0}{T})\,\dot{Q} - \dot{W} + \dot{X}_{mass,in} - \dot{X}_{mass,out} \quad (23)$$

Index 0 denotes the dead state. In exergetics when a system is in thermodynamic equilibrium with its environment, it is denoted by a subscript zero. At this point no more work can be done.

The flow exergy of a gas stream is related to its basic thermodynamic properties by introducing specific exergy, ψ, which is a function of enthalpy, *h*, entropy, *s*, flow velocity, *w*, height, *z*, and chemical exergy, ψ^{CH}:

$$\frac{\dot{X}}{\dot{m}} = \psi = [(h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz + \psi^{CH}] \quad (24)$$

Generally the thermodynamic inefficiency caused by friction and heat transfer, shows itself in exergy destruction, therefore the exergy destruction of turbine and compressor have been investigated in this study.

Exergy destruction of turbine is written as below:

$$\Delta\dot{X}_{gt} = 10^3\dot{m}_f\psi_f - P_{gc} \quad (25)$$

where $P_{gt}$is power of gas turbine and $\dot{m}_f$ is the mass flow rate that is consumed by turbine in kg/s.The chemical exergy of fuel is presented by ψ_f. For engineering purposes Szargut suggested a simple approximation where chemical exergy is equal to LHV of fuel multiplied to a constant [19].

The chemical exergy in the literature is at the standard condition. Zanchini and Terlizzese [9] have investigated the evaluation of flow exergy of pure chemicals for non-standard conditions. Their study has shown that chemical flow exergy is a decreasing function of temperature and relative moisture. The variation of chemical exergy by temperature and pressure has been investigated by Chaczykowski et al. [8].Fuel exergy in Equation (24) can be written using below relationship:

$$M(\psi_f - \psi^{CH}(T_0,p_0)) = (h - h_0) - T_0(s - s_0) \quad (26)$$

Exergy destruction of gas compressor can be presented using mechanical efficiency and isentropic efficiency which is substituted by an expression; this expression is written as a function of isentropic exponent, *k*, polytropic exponent, *n*, and P_{gc} that is power of gas compressor [12]:

$$\Delta\dot{X}_{gc} = P_{gc}\left(1 - \eta_m \frac{(\frac{p_d}{p_s})^{\frac{k-1}{k}} - 1}{(\frac{p_d}{p_s})^{\frac{n-1}{n}} - 1}\right) \quad (27)$$

Exergy destruction of cooler can be computed by considering the consuming work which is needed to overcome hydraulic resistance, exergy destruction associated with heat and fan electric power, P_{ac}:

$$\Delta\dot{X}_{ac} = -\left(1 - \frac{T_0}{T_{air,out}}\right)\dot{Q} + P_{ac} + \left[(h_{in} - h_{out}) - T_0(s_{in} - s_{out}) + \frac{{w_{in}}^2 - {w_{out}}^2}{2}\right] \quad (28)$$

Employing exergetic efficiency is a good way to analyse the operation of thermodynamic systems. Exergetic efficiency is expressed as a ratio of the exergy that leaves the system to the total exergy thatenters it. Accordingly, exergetic efficiency will become as below:

$$\eta_{tc} = 1 - \frac{\Delta\dot{X}_{destryed}}{\dot{X}_{in}} \quad (29)$$

3. CASE STUDY

Exergy analysis of turbo-compressor and its cooler within a compressor station has been carried out in this study. This station which is installed in south of Iran on IGAT5 pipeline, has been mounted to boost sour gas; one of the longest sour gas pipeline in the world. This station consists of four compressors which are all placed in parallel mode. Any of these compressors can be in service optionally, in accordance with receiving pressure, volumetric flow and delivery pressure. However, at least one compressor should be in stand-by mode.

The sour gas boosting in compressor station, consist of fourteen components. Mole fractions and physical propertiesof gas components are presentedin literature [20]. Modified pseudo-critical pressure and temperature of natural gas considered in compressibility factor calculationsareshownin Table 1, where pressure and temperature are in psi and R ,respectively.

It is assumed that delivery capacity at the station is $6.0 \times 10^4 m^3/h$ at 38°C and 67.81 bar. The maximum driver efficiency is 0.35 based on turbine characteristic curves and mechanical deriver is supposed to be 0.98. As it is mentioned before, for calculating the fuel consumption in turbine the polytropic efficiency, polytropic head and driver efficiency are needed. Hence they are extracted from manufacturer data [21], Coefficients of Equations (8), (9) and (13) are set in Table 2. $a_7$and a_8 have obtained at 38°C.

Design parameters and manufacturer data for gas cooler are as follow, heat transfer area of cooler is 35296 m^2, volumetric air flow needed for cooling propose is 141.1m^3/s, when air temperature is 38°C. The power of single fan with maximum speed of 235 rpm is 25.9 kW for each 12 fans, where the overall heat transfer coefficient is 21.78 W/(m^2.K) and cooler pressure drop is supposed to be 0.00144 bar based on documentary data.

4. RESULTS AND DISCUSSIONS

In this study, the changes in exergy destruction of turbo-compressor, cooler and fuel consumption under variable compressor speeds are presented. According to the value of driver efficiency, it is seen that turbine exergy destruction has a large magnitude in comparison with that of compressor and cooler. It is considered that compressor boosts the gas at variable speeds: 5200 to 7000 rpm. Under aforementioned operating conditions, the gas density at suction side is 54.67 kg/m^3 and 915 kg/s, sour gas is entering into the station and divided equally into three lines of parallel compressor. By increasing the compressor speed, it is expected that fuel consumption starts a rising mode to provide energy needed to transmit the gas. Although this amount of fuel use is unfavorable but for reaching the desired head of energy sometimesit is inevitable.

Some important data of compressor station for five states were calculated and shown in Table 3. The driven data in this table clearly shows the relation between fuel consumption of gas turbine, output power of turbine and also the fuel exergy. As the speed rises, more fuel is needed for providing the required energy and also output power increases consequently. This increase of output power can be recognized by rate of fuel LHV. This table also shows the maximum potential work (rate of fuel exergy) that can be extracted from fuel entering to the combustion chamber of turbine, but only part of this great potential work of fuel can be converted to real work because of driver efficiency. Therefore the large magnitude of exergy destruction will appear. Power of air cooler is supposed to be fixed for each compressor speeds. As the compression ratio of station grows up, the gas turbine power and rate of fuel needed to burn in combustion chamber is increased slightly. Compression ratio changes versus speed from 1.13 to 1.43 by a nearly linear mode.The trend of fuel consumption versus speed is shown in Figure 1. As it was mentioned the more the fuel enters the combustion chamber the more exergy destruction of turbine is expected therefore it is acceptable to see equal trend of turbine exergy destruction and fuel consumption as a reason of their dependency. Exergy destruction of turbine, compressor, cooler and total destruction is depicted in Figure 2 to realize the effect of each one on total destruction.

The total destruction of station is influenced by turbine destruction more than compressor and cooler due to magnitude of turbine's exergy destruction. It can be seen that the turbine destruction line and total destruction are so close to each other and have the same mode.

TABLE 1. Effect of adjustment factor

Critical pressure	Critical temperature	Suction side compressibility factor
676.32	360.18	0.8753
Modified critical pressure	**Modified critical temperature**	**Modified suction compressibility factor**
665.51	354.46	0.8813

TABLE 2. Coefficients of compressor polytropic head and efficiencyalong with driver efficiency

Coefficient	a1	a2	a3	a4
Value	4.65E-4	2.074	-1.85E3	-0.921
Coefficient	**a5**	**a6**	**a7**	**a8**
Value	4483	-3E+6	1.32E-8	0.0579

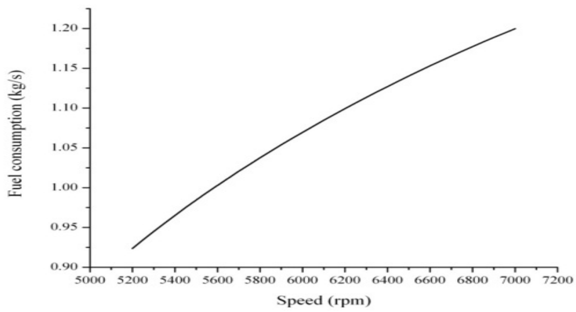

Figure 1. Fuel consumption versus compressor speed

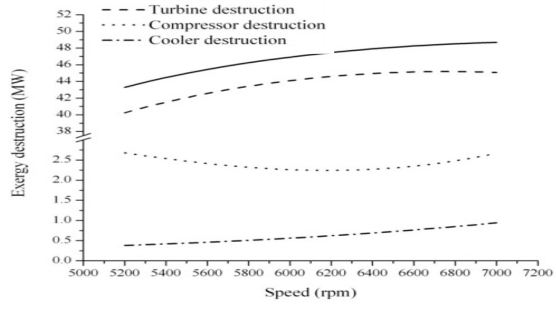

Figure 2. The change in exergy destruction of turbine, compressor and cooler versus various speeds

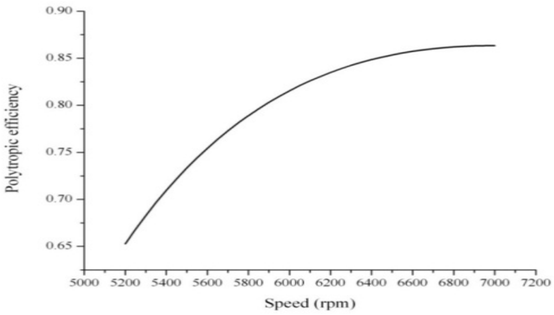

Figure 3. Polytropic efficiency versus compressor speed

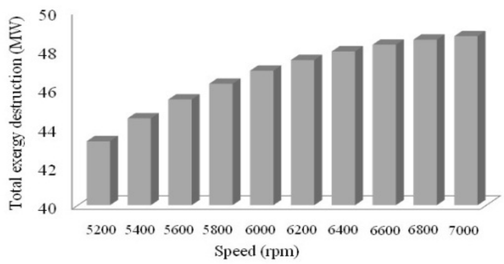

Figure 4. Total exergy destructionof station versus compressor speed

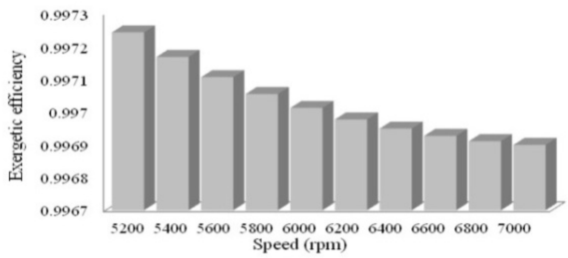

Figure 5. Exergetic efficiency versus compressor speed

TABLE 3. Important data of station in five states

Parameter	Speed				
	5200	**5650**	**6100**	**6550**	**7000**
Fuel consumption (kg/s)	0.924	1.012	1.048	1.147	1.199
GT power (MW)	7.33	9.31	11.49	13.94	16.71
Rate of fuel LHV (MW)	47.14	51.64	55.36	58.53	61.24
Rate of fuel exergy (MW)	47.56	52.10	55.85	59.05	61.78
AC electric power (MW)	0.31	0.31	0.31	0.31	0.31

Exergy destruction of compressor that is illustrated in Figure 2 is started with the falling and rising trend during the operation and does not have the uniform mode. This concavity occurred due to tradeoff between polytropic efficiency and head of compressor. Both polytropic efficiency and compressor head affect the compressor destruction and both increase by raising the speed but not by the same magnitude. This tradeoff can be recognized better by considering Equation (37). At right side of this equation, the mechanical driver is multiplied to an expression which is actually isentropic efficiency that rises by the same trend of polytropic efficiency that is shown in Figure 3.So when the polytropic efficiency increases subsequently isentropic efficiency increases too. Mechanical driver is assumed to be fixed and is multiple to the rising term that is actually isentropic efficiency. But after a specific speed between 6000 and 6500, increase in power of compressor (compressor head) affected the compressor destruction more than increase of isentropic efficiency that its rising rate is become slighter than old one; consequently compressor destruction become more than later state and the rising trend appeared. Polytropic efficiency severely depends on operating conditions especially the volumetric flow of compressed gas. Based on compressor characteristic curve, the polytropic efficiency when the inlet volumetric flow is less grows much more compared to the state that more volume entering, for the same compressor speed raises. The outlet temperature is raised and affected by compressor speed; consequently the air cooler destruction isinfluenced and obeys this raising trend.

The growth of total exergy destruction instation causes the reduction of exergetic efficiency which is a scale to show how environmentally-friendly the operation is. Total exergy destruction of station is shown in Figure 4, also variation of exergetic efficiency is plotted in Figure 5. Obviously, the exergetic efficiency is affected from total exergy destruction station and its falling trend observed here is due to the destruction rise.The change in exergetic efficiency of turbo-compressor is started from 0.997 and ended to 0.9968 depending on operating conditions.

TABLE 4. Exergy analysis of each element of station in five states

Parameter	Speed				
	5200	**5650**	**6100**	**6550**	**7000**
Turbine destruction (MW)	40.23	42.79	44.36	45.10	45.07
Comp destruction (MW)	2.68	2.39	2.25	2.32	2.67
Cooler destruction (MW)	0.44	0.53	0.65	0.81	1.00
Total destruction (MW)	43.34	45.70	47.26	48.23	48.74
Exergetic efficiency (%)	99.72	99.71	99.70	99.69	99.68

Another important parameter in exergy analysis is driver efficiency. Driver efficiency is changing in a rising mode during the increasing speed. Based on Equation (16), there is an approximately liner relation between output power of turbine and driver efficiency for an exact temperature. These changes in driver efficiency and polytropic efficiency are not a general rule for every turbo-compressor and differ vis-à-vis the characteristic curves of turbine and compressor. As seen in Table 4, exergy analysis of compressor station for each turbo-compressor and its cooler is calculated. As can be seen from this table more than 90 precent of total exergy destruction is possessed by turbine for each state in which the second place is allocated by compressor destruction. The falling and rising mode of compressor exergy destruction is appeared here, where the exergy destruction of compressor at speed of 6100 rpm is less than destruction at speed of 5650 and more than 6550 rpm.

5. CONCLUSIONS

An exergy based analysis has been performed to investigate the operation of gas turbine and compressor within a boosting station installed on a trunk pipeline.By this method, it is possible to find out the parameters involved in fuel consumption and to quantify some economic aspects of energy. This thermodynamic study gives a point of view to estimate the magnitude of irreversibility in station. The presented model helps for operating the station at different conditions such as discharge pressure. The results show that biggest irreversibility belongs to gas turbine as a result of driver efficiency which has a small magnitude. In addition, by increasing the compressor speed at the declared condition, the irreversibility and fuel consumption come up due to receiving the required discharged pressure, consequently exergy destruction of cooler also changes at the same trend.

6. REFERENCES

1. Wu, S., Rios-Mercado, R.Z., Boyd, E.A. and Scott, L.R., "Model relaxations for the fuel cost minimization of steady-state gas pipeline networks", ***Mathematical and Computer Modelling***, Vol. 31, No. 2, (2000), 197-220.
2. Jafarmadar, S., "The numerical eexergy analysis of H2/AIR combustion with detailed chemical kinetic simulation model",***International Journal of Engineering Transection C***, Vol. 25, (2012), 239-248.
3. Rath, M., Acharya, S., Patnnaik, P. and Roy, S., "Exergy and energy analysis of diesel engine using karanja methyl ester under varying compression ratio", ***International Journal of Engineering-Transactions B: Applications***, Vol. 27, No. 8, (2014), 1259-1268.
4. Taherinejad, M., Hosseinalipour, S. and Madoliat, R., "Steady flow analysis and modeling of the gas distribution network using the electrical analogy", ***International Journal of Engineering***, Vol. 27, No. 8, (2014), 1269-1276.
5. Tabkhi, F., Pibouleau, L., Azzaro-Pantel, C. and Domenech, S., "Total cost minimization of a high-pressure natural gas network", ***Journal of Energy Resources Technology***, Vol. 131, No. 4, (2009),1-12.
6. Borraz-Sanchez, C. and Rios-Mercado, R.Z., "Improving the operation of pipeline systems on cyclic structures by tabu search", ***Computers & Chemical Engineering***, Vol. 33, No. 1, (2009), 58-64.
7. Evenko, V., "Thermodynamic analysis of hydrocarbon gas transportation system", ***Chemical and Petroleum Engineering***, Vol. 33, No. 4, (1997), 390-395.
8. Chaczykowski, M., Osiadacz, A. and Uilhoorn, F., "Exergy-based analysis of gas transmission system with application to yamal-europe pipeline", ***Applied Energy***, Vol. 88, No. 6, (2011), 2219-2230.
9. Zanchini, E. and Terlizzese, T., "Molar exergy and flow exergy of pure chemical fuels", ***Energy***, Vol. 34, No. 9, (2009), 1246-1259.
10. Menon, S. and Menon, P., "Gas pipeline hydraulics, Trafford Publishing, (2013) 35-41.
11. Ludwig, E.E., "Applied process design for chemical and petrochemical plants: Gulf Professional Publishing, Vol. 2, (1997) 102-115
12. Perry Robert, H., Green Don, W. and Maloney James, O., "Perry's chemical engineers' handbook", ***Mc Graw-Hills New York***, (1997) 56-64.
13. Galindo, J., Arnau, F., Tiseira, A. and Piqueras, P., "Solution of the turbocompressor boundary condition for one-dimensional gas-dynamic codes", ***Mathematical and Computer Modelling***, Vol. 52, No. 7, (2010), 1288-1297.
14. Odom, F.M. and Muster, G.L., "Tutorial on modeling of gas turbine driven centrifugal compressors", in PSIG Annual Meeting, Pipeline Simulation Interest Group.,(2009), 21-30.
15. Uilhoorn, F., "Dynamic behaviour of non-isothermal compressible natural gases mixed with hydrogen in pipelines", ***International Journal of Hydrogen Energy***, Vol. 34, No. 16, (2009), 6722-6729.
16. Holman, J., Heat transfer, *9th*., McGraw-Hill.(2002), 112-120
17. Szargut, J., "Exergy analysis", ***The Magazine of Polish Academy of Sciences***, Vol. 3, No. 7, (2005), 31-33.
18. Cengel, Y.A., Boles, M.A. and Kanoglu, M., "Thermodynamics: An engineering approach, McGraw-Hill New York, Vol. 5, (2011), 41-50.
19. Szargut, J., "Exergy method: Technical and ecological applications, WIT press, Vol. 18, (2005), 23-28
20. Winnick, J., "Chemical engineering thermodynamics, John Wiley & Sons, (1997), 45-62
21. Mohamadi-Baghmolaei, M., Tabkhi, F. and Sargolzaei, J., "Exergetic approach to investigate the arrangement of compressors of a pipeline boosting station", ***Energy Technology***, Vol. 2, No. 8, (2014), 732-741.

Operation Analysis of Rotary Tools of Compressor Station Using Exergy Approach

M.Mohamadi-Baghmolaei[a],J. Sargolzaei[a], F. Tabkhi[b]

[a] *Department of Chemical Engineering, Ferdowsi university of Mashhad , Mashhad, Iran*
[b]*Department of Chemical Engineering, Persian Gulf University, Bushehr, Iran*

PAPER INFO

Paper history:
Received 09July 2013
Received in revised form 15 May 2014
Accepted 14 August 2014

Keywords:
Exergy Analysis
Compressor Station
Turbine
Fuel Consumption
Efficiency

چکیده

در این تحقیق با استفاده از تحلیل اگزرژی به بررسی عملکرد ایستگاه تقویت فشارپرداخته شده است. تحلیل اگزرژی روشی ترمودینامیکی است که با استفاده از آن بازگشت ناپذیری های سیستم به صورت کمی محاسبه می شود. کمپرسورهای مورد استفاده در ایستگاه جهت غلبه بر افت فشار به وجود آمده در امتداد خط لوله به کار گرفته شده اند. عمل تراکم منجر به افزایش دمای گاز متراکم شده می شود در نتیجه استفاده از کولرهای هوا جهت خنک سازی آن اجتناب ناپذیر می باشد. کمپرسورها با سرعتهای متغیر قادر به تراکم گاز می باشند، مقدار سوخت مصرف شده در محفظه ی احتراق نیز متعاقباً تغییر می کند. برای مطالعه موردی خط لوله ی انتقال گاز ترش IGAT5، در جنوب ایران انتخاب شده است. برای دامنه سرعت rpm ۵۲۰۰ تا ۷۰۰۰ تخریب اگزرجی، بازده اگزرجتیک، مصرف سوخت و تأثیر هر یک از اجزاء ایستگاه بر روی تخریب اگزرجی کلی اندازه‌گیری شده است. نتایج نشان دهنده آن است که بیش از ۹۰ درصد از تخریب اگزرجی کل مربوط به توربین می‌باشد. همچنین مصرف سوخت ایستگاه با سرعت کمپرسورتغییرمی‌کند و از kg/s ۰/۹۲٤ در سرعت ۵۲۰۰rpm آغاز می‌شود و تا kg/s ۱/۲ در سرعت rpm ۷۰۰۰ پایان می‌یابد.

***doi**: 10.5829/idosi.ije.2014.27.12c.03*