Energy, exergy, exergoenvironmental, and exergoeconomic (4E) analyses of a gas boosting station

Mohammad Salimi Delshad\textsuperscript{1} | Ali Momenimovahed\textsuperscript{1} \textcopyright | Mohammad Sh. Mazidi\textsuperscript{2} | Mehdi A. Ehyaei\textsuperscript{3} | Marc A. Rosen\textsuperscript{4}

\textsuperscript{1}Department of Mechanical Engineering, Imam Khomeini International University, Qazvin, Iran
\textsuperscript{2}Optimization and Development of Energy Technologies Division, Research Institute of Petroleum Industry (RIPI), Tehran, Iran
\textsuperscript{3}Department of Mechanical Engineering, Pardis Branch, Islamic Azad University, Pardis New City, Iran
\textsuperscript{4}Faculty of Engineering and Applied Science, University of Ontario Institute of Technology, Oshawa, ON, Canada

Abstract
Energy, exergy, exergoenvironmental, and exergoeconomic analyses of a natural gas boosting station are presented using a real-gas model and actual operational data. The effect of various performance parameters on the thermodynamic efficiencies, specific fuel consumption (SFC), gas-phase emissions, and cost rates are assessed. The results show that, for the actual operational conditions of the gas boosting station, the exergy efficiencies are 76.1\% and 73.9\% in the hot and cold seasons, respectively. Moreover, the energy analysis at partial load reveals that the SFC varies from 0.285 kg/kWh to 0.302 kg/kWh, respectively, in maximum and minimum ambient temperatures. The exergoeconomic analysis along with the exergoenvironmental analysis shows that the total cost rate of gas boosting stations in hot and cold ambient conditions is 7390 US$/h and 8070 US$/h, respectively, with more than 60\% related to the environmental impact. In this system, the highest exergoeconomic factor is attributed to the centrifugal gas compressor at 40.9\%-44.2\% and the lowest to the air cooler at 0.030\%-0.036\%, depending on the ambient temperature, which specifies the balance between capital cost and the cost of exergy destruction. The cost rate of the exergy destruction is more pronounced in the combustion chamber, and the overall cost rate of the exergy destruction can be improved significantly by increasing turbine inlet temperature which needs additional investment cost for the system.

\textbf{KEYWORDS}
efficiency, environmental impact, exergy, fuel consumption, gas boosting station, gas turbine

1 | INTRODUCTION

Natural gas is a popular energy source in comparison with other fossil fuels because it emits lower greenhouse gases.\textsuperscript{1,2} It has many commercial and residential consumers around the world, based on a complex and vast distribution network. More than 30\% of the cost of natural gas is attributed to the transmission and distribution of refined gas to the users, largely due to the long distance between gas reservoir and gas markets.\textsuperscript{3} Gas boosting stations are the heart of the gas transmission networks, which are employed to increase the pressure of...
the gas to drive it through the pipeline systems. Gas compression is normally accomplished with reciprocating or centrifugal compressors driven by electro-motors or gas turbines. Centrifugal compressors are widely used for the systems with relatively high mass flow rates due to their simplicity and reliability compared to screw or reciprocating compressors. Due to compression, the gas temperature increases significantly which causes a rise in its specific volume and velocity, and consequently, the pressure drops along the pipeline. Therefore, gas coolers are utilized in the discharge section of the compressors to offset the impacts of gas temperature rise.\textsuperscript{4}

In a natural gas boosting station, the gas turbine consumes 3\%-5\% of the transmitted natural gas to drive the gas compressor.\textsuperscript{2} Many efforts have been made in the last decades to decrease the fuel cost by reducing the cost of compression. Chebouba et al.\textsuperscript{5} used a metaheuristic optimization algorithm to minimize the power consumption of a compressor station by adjusting the nodal pressure in a linear gas transmission system including several compressor stations. To decrease the energy consumption in a boosting station, Fasihizadeh et al.\textsuperscript{6} found the optimum nodal pressure values in the national Iranian gas network. This was done by setting the proper pipeline capacity and optimum inlet pressure at each port using adequate number of compressors. They showed that the operation hours of the turbo-compressors can also be reduced by optimizing the transmission process which in turn causes a postpone in the overhaul time. In order to find the optimum performance of a natural gas network, Alinia Kashani and Molaei\textsuperscript{7} presented a multiobjective optimization model considering two objective functions: maximizing the gas delivery flow and line pack and minimizing the operating cost defined as the sum of the fuel consumption and CO\textsubscript{2} emission costs.

In addition to energy analysis, exergy analysis is a powerful method for performance assessment, design, and optimization of thermal systems. Exergy analysis can provide an understanding of the quality and degradation of energy in a thermodynamic process. It highlights the sources, types, and magnitudes of irreversibilities or inefficiencies in thermal systems. Such information can be useful for improving or optimizing the efficiency, environmental impact, and cost-effectiveness. It can also be used in benchmarking studies.\textsuperscript{8,9}

Evenko\textsuperscript{10} performed an exergy analysis to assess the efficiency of a gas transmission system under full load operating condition. Chaczykowski et al.\textsuperscript{4} determined the exergy destruction of an existing gas transmission system including compressor station and pipelines in order to compare the performance of the gas transmission system at part and full loads for several cooler operating points. Baghmolaei et al.\textsuperscript{11} assessed the exergy destruction of eight configurations of multistage turbo-compressors with intercoolers, to find the optimum arrangement of compressors in order to minimize the fuel consumption in a sour gas pipeline boosting station. They reported 2.93 kg/s, 132 MW, and 99.4\% for the fuel consumption rate, exergy destruction rate, and exergy efficiency, respectively, for the optimum configuration. Kostowaski et al.\textsuperscript{12} performed energy and exergy analyses to compare three different scenarios based on the installation of a heat recovery system to improve the thermodynamic performance of a gas compressor station with various types of compressor units. They showed that the energy and exergy efficiencies of their case study plant at the operational point are significantly lower than the corresponding values in the nominal state. Although much research has been carried out to enhance the performance of gas transmission networks, it is still necessary to evaluate the performance parameters of gas boosting stations and identify the contribution of different system components into the losses which can then be used to further improve the performance of the gas boosting stations in order to reduce the cost of transmission and distribution of natural gas.

In addition to the cost of fuel consumption, the environmental cost of gas boosting stations should also be taken into account. It has been shown that the environmental cost of power generation systems can be several times higher than the cost of their fuel consumptions.\textsuperscript{8} Energy, exergy, environmental, and economic analyses have been conducted to improve the performance of different types of power generation systems; however, these evaluations have not been performed for a gas boosting station yet.

Mohtaram et al.\textsuperscript{13} carried out energy and exergy analyses of an ammonia water combined cycle to find the optimum pressure ratio. Their results show that, with increasing compressor pressure ratio, the exergy destructions of the gas turbine, high-pressure compressors, and intercooler increase while the exergy destruction of the recuperator decreases and the power generation of the gas turbine cycle increases. Furthermore, the lowest fuel consumption belongs to the cycle with the compressor pressure ratio of 7.5. Ameri et al.\textsuperscript{14} performed energy, exergy, and exergoeconomic analyses for a steam power plant at various loads and ambient temperatures. They found that the condenser and boiler are responsible for the main energy losses and exergy destructions, respectively. They also showed that the energy loss rate from the condenser and the boiler to the environment is 306.9 MW and 67.6 MW, respectively. Singh et al.\textsuperscript{15} performed a thermoeconomic analysis and optimization of a power cycle including Rankine, Brayton, and Kalina cycles.
that is, a triple power cycle. They showed that the total cost rate defined as a thermodynamic objective function decreases with decreasing ambient temperature and the minimum amount of total cost rate occurs for a gas topping cycle pressure ratio of 14. Ehyaei et al.\textsuperscript{16} carried out exergy, economic, and environmental (3E) assessments to obtain the output power, energy and exergy efficiencies, and environmental and electrical costs of a gas turbine power plant using an inlet air cooling system. They found that the exergy efficiency of the power generation plant increases, and the electricity production cost decreases by the implementation of an absorption chiller air inlet cooling system. Yazdi et al.\textsuperscript{17} studied the effects of air cooling on gas turbine performance conducting energy, exergy, economic, and environmental analyses. They found that the exergy efficiency was improved by up to 2.5% and 1.5% for cities with humid and dry climates, respectively, when an absorption chiller was implemented in the cycle. They also showed that using the inlet air cooling system can significantly reduce NO\textsubscript{x} emission as well as electricity generation costs.

This work presents a comprehensive model (energy, exergy, exergoeconomic, exergoenvironmental) for integration of gas cycle with gas boosting station. Although several models existed in literature in this regard, a complete analysis with real data has not existed. Also, no previous researches have been done with this complete analysis. The energy and exergy efficiencies of the boosting station are determined by comparing the driver-specific fuel consumption (SFC) in hot and cold seasons. Moreover, the contributions of the elements in the gas boosting station to the destroyed exergy are quantified. Furthermore, the total cost rate of the gas boosting station is obtained with exergoeconomic analysis and the impact on the surrounding environment is quantified with exergoenvironmental analysis.

Additional innovations and original contributions of this paper are as follows:

- Developing an accurate model to predict the performance of the gas boosting station and achieving good agreement between model and test results.
- Performing 4E analyses on the gas boosting station using a real-gas model with a compressibility factor and performance parameters for real operational conditions.
- Obtaining the pollutant emissions and determining the contribution of the environmental impact costs to the total cost rate of gas boosting stations.
- Providing applicable suggestions to increase efficiencies, improve the deviation from the reversible process, and reduce the exergy destruction cost of a gas boosting station.

2 | MODELING AND ANALYSIS

2.1 | Process description and assumptions

A typical gas boosting station design is illustrated in Figure 1. It consists of a dual-spool (two shaft) gas turbine (GT), a centrifugal gas compressor (GC), and an air cooler (AC). In a dual-spool gas turbine, the power turbine and the gas generator are installed on separate shafts, thus permitting the power turbine to rotate independently of the gas generator since the power turbine is supported on its shaft bearings independently. This configuration is used for variable speed applications such as driving compressors, typically in the range between 25% and 100% of the full speed. The gas turbine runs on natural gas with a lower heating value (LHV) of around 50 MJ/kg. The fuel flow rate and exhaust temperature are 1.85 kg/s and 510\degree C, respectively, based on the ISO standard conditions. The compressor and turbine isentropic efficiencies and the compressor pressure ratio are also assumed to be 83%, 91%, and 10.9, respectively.

A single-stage centrifugal compressor with a polytropic efficiency of 82.6% is used to boost the natural gas and an induced type air cooler with 8 fans is implemented to reduce the natural gas discharge temperature. The assumptions used to model the system include\textsuperscript{4,11}:

1. The gas turbine, gas compressor, and air cooler work at part-load, off-design condition.
2. Enthalpy and entropy are a function of both temperature and pressure.
3. The thermodynamic properties of natural gas are calculated using the real-gas model instead of ideal gas model.

2.2 | Energy and exergy analyses

The system power generation, fuel consumption, and energy efficiency can be estimated performing the energy balances for the gas boosting station and its components (see Figure 1).

2.2.1 | Air compressor

Air at ambient pressure, $P_1$, and temperature, $T_1$, is drawn into the compressor. The compressor outlet temperature ($T_2$) is dependent on the pressure ratio ($r_p$) and the compressor isentropic efficiency ($\eta_{ac}$) as well as the specific heat ratio ($k$). The power used by the air compressor as
well as the isentropic efficiency of the air compressor can be expressed, respectively, as:\(^1^8\)

\[ W_{ac,act} = m_{air} (h_{2a} - h_1) = m_a \left( \int_{T_1}^{T_{2a}} C_{p,air} (T) \, dT \right) \]  

(1)

\[ \eta_{ac} = \frac{W_{ac,isen}}{W_{ac,act}} = \frac{h_{2a} - h_1}{h_{2a} - h_1} \]  

(2)

Entropy change across the air compressor can be calculated from:\(^1^6\)

\[ s_{2s} - s_1 = (s_{gen})_{ac} - R \ln \frac{P_2}{P_1} = \int_{T_1}^{T_{2a}} \frac{C_{p,air} (T)}{T} \, dT - R \ln \frac{P_2}{P_1} \]  

(3)

In an isentropic process, \( s_{2a} - s_1 = 0 \) and Equation (3) permit an iterative solution for the air compressor’s isentropic discharge temperature \( T_{2a} \).

### 2.2.2 Combustion chamber

Natural gas is burnt with air in the combustion chamber at constant pressure resulting the temperature to rise from \( T_2 \) to \( T_3 \). An energy rate balance for the combustion chamber can be expressed as:\(^1^9\)

\[ \dot{m}_{air} C_{p,air} T_2 + \dot{m}_f LHV + \dot{m}_f C_{pf} T_f = \dot{m}_{exh} C_{p,exh} T_3 \]  

(4)

where \( \dot{m}, T, \) and \( C_p \) are, respectively, mass flow rate, temperature, and specific heat at constant pressure.

Considering the pressure drop inside the combustion chamber, the pressure of the exhaust gas can be expressed as:\(^2^0\)

\[ P_3 = P_2 \left( 1 - \Delta P_{CC} \right) \]  

(5)

The composition of the natural gas is shown in Table 1. The chemical composition is determined with gas
chromatography based on the ASTM D1945 standard. The mole fraction of compounds under 0.01% are reported as trace and neglected in the combustion reaction.

The combustion reaction can be determined from the atomic balance and consideration of the excess air as follows:13,21

\[ y_{H_2O} = \frac{1.9452}{4.76a + 1.0156} \]  
\[ y_{CO_2} = \frac{0.9992}{4.76a + 1.0156} \]  
\[ y_{O_2} = \frac{a - 1.9608}{4.76a + 1.0156} \]  
\[ y_{N_2} = \frac{3.76a + 0.032}{4.76a + 1.0156} \]  
\[ AFR = \frac{m_a}{m_f} \]  
\[ a = \frac{AFR \times \frac{M_{fuel}}{M_{air}}}{4.76} \]

Therefore, the total mass of products is \( 4.76a + 1.0156 \). The mole fraction of the product species as well as the air-to-fuel ratio can also be calculated from:

\[ \begin{align*}
\Delta s_3 - \Delta s_5 &= (s_{gen})_g - R\ln \frac{P_3}{P_5} = \int_{T_{sa}}^{T_3} C_p(T) dT - R\ln \frac{P_3}{P_5} = 0 \tag{12}
\end{align*} \]

where \( \frac{P_3}{P_5} \) is the turbine expansion pressure ratio. The gas turbine output power can also be calculated from:18

\[ \dot{W}_{gt} = \dot{m}_{ech} (h_3 - h_{5a}) = \dot{m}_{ech} \int_{T_{sa}}^{T_3} C_p(T) dT \]  
\[ \eta_{gt} = \frac{\dot{W}_{gt, isen}}{\dot{W}_{gt, act}} = \frac{h_{5a} - h_3}{h_{5a} - h_3} \]  
\[ W_{net} = \dot{W}_{gt} - W_{ac} \]  

Finally, the specific fuel consumption (SFC) for the gas turbine can be expressed as follows:22

\[ SFC = \frac{3600}{\dot{m}_{air} \left( \frac{\dot{W}_{ac}}{m_{air}} - \frac{\dot{W}_{ac}}{m_{air}} \right)} AFR \]  

where \( \eta_{mech} \) is gas turbine mechanical efficiency and AFR is the air-to-fuel ratio.

### 2.2.3 Gas turbine

In an isentropic process, the entropy change in the gas turbine is obtained from:18

\[ \Delta s_3 - \Delta s_5 = (s_{gen})_g - R\ln \frac{P_3}{P_5} = \int_{T_{sa}}^{T_3} C_p(T) dT - R\ln \frac{P_3}{P_5} = 0 \]  

The isentropic turbine efficiency \( (\eta_{gt}) \) can be written as:

\[ \eta_{gt} = \frac{\dot{W}_{gt, isen}}{\dot{W}_{gt, act}} = \frac{h_{5a} - h_3}{h_{5a} - h_3} \]  

and the gas turbine net output power as:

\[ W_{net} = \dot{W}_{gt} - W_{ac} \]  

### 2.2.4 Centrifugal gas compressor

The performance of the centrifugal compressor can be obtained from mathematical models using conservation principles or a neural network approach.23,24 However, experimental characteristic curves are normally provided by the manufacturers because of the complexity of the models.4

The isentropic efficiency of the gas compressor can be calculated from:

\[ \eta_{gc} = \frac{h (P_7 \cdot T_7) - h (P_6 \cdot T_6)}{h_7 - h_6} \]  

where \( \eta_{mech} \) is gas turbine mechanical efficiency and AFR is the air-to-fuel ratio.
\[
\dot{W}_{\text{g,c}} = \frac{1}{\eta_{\text{mech}}} \dot{m}_g \left[ h \left( P_7 \cdot T_{2c} \right) - h \left( P_6 \cdot T_6 \right) \right] \tag{18}
\]

In a turbo-compressor system, \(\dot{W}_{\text{g,c}}\) is equal to the power generated by the gas turbine.

The entropy and enthalpy changes across the centrifugal compressor can be calculated from: \(^{26}\)

\[
s \left( T_7 \cdot P_7 \right) - s \left( T_6 \cdot P_6 \right) = \int_{T_6}^{T_7} \frac{C_p}{T} dT - \int_{P_6}^{P_7} \frac{R}{P} \left[ Z + T \left( \frac{\partial Z}{\partial T} \right)_p \right] dP \tag{19}
\]

\(T_{2c}\) can be obtained iteratively from the above equation assuming that \(s_7 - s_6 = 0\).

\[
h \left( T_7 \cdot P_7 \right) - h \left( T_6 \cdot P_6 \right) = \int_{T_6}^{T_7} C_p dT - \int_{P_6}^{P_7} \frac{R T^2}{P} \left( \frac{\partial Z}{\partial T} \right)_p dP \tag{20}
\]

Here, \(R\), \(C_p\), and \(Z\) are, respectively, specific gas constant, ideal heat capacity, and compressibility factor.

The gas compressor actual discharge temperature \((T_7)\) can be calculated iteratively after determining the actual compressor head using the compressor isentropic efficiency formula (Equation 17).

The compressibility factor of the natural gas can be calculated from the equation of states or estimated from empirical equations. In the present study, the equation suggested by the AGA (American Gas Association) is used to calculate the compressibility factor: \(^{25}\)

\[
Z = 1 + \left[ 0.257 - 0.533 \left( \frac{T}{T_c} \right) \right] \left( \frac{P_{\text{avg}}}{P_c} \right) \tag{21}
\]

\[P_{\text{avg}} = \frac{2}{3} \left( P_6 + P_7 - \frac{P_6 P_7}{P_6 + P_7} \right) \tag{22}\]

The pseudo-critical temperature \((T_c)\) and pseudo-critical pressure \((P_c)\) can be calculated using the critical temperature and pressure of the components of natural gas (see Table 1, \(^{25}\)).

The heat capacity of each component of natural gas can be estimated from: \(^{26}\)

\[
C_{p,i}^0 = C_1 + C_2 \left[ \frac{C_3}{T} \sinh \left( \frac{C_3}{T} \right) \right] + C_4 \left[ \frac{C_3}{T} \cosh \left( \frac{C_3}{T} \right) \right] \tag{23}
\]

where \(C_1\), \(C_2\), \(C_3\), \(C_4\), and \(C_5\) are constant. These constant values can be found from \(^{26}\) for different compounds of natural gas. The ideal gas heat capacity for the natural gas can then be written as: \(^{26}\)

\[
C_p^0 = \sum C_{p,i}^0 y_i \tag{24}\]

Finally, the real heat capacity which is a function of temperature, pressure, ideal heat capacity, and specific volume \((v)\) can be calculated from: \(^{26}\)

\[
C_p = C_{p,i} - \int_0^P \left( \frac{\partial^2 v}{\partial T^2} \right)_p dP \tag{25}\]

### 2.2.5 Air cooler

Air coolers are implemented as after coolers at the discharge section of the compressor to reduce the gas temperature before the introduction to the pipeline. In this work, horizontal cross-flow air coolers are used with the cooling air passes from 8 parallel bundles of finned tubes.

For the exergy analysis of the air cooler, the gas temperature at the gas compression station outlet and the amount of heat exchanged are required. Two common methods can be used to determine the gas temperature: the logarithmic mean temperature difference (LMTD) and the effectiveness number of the transfer unit \((\varepsilon\text{-NTU})\). Here, the \(\varepsilon\text{-NTU}\) method is used due to the unknown gas inlet and outlet temperatures. The energy rate balance for the air cooler at steady-state conditions is: \(^{27}\)

\[
\dot{Q} = \varepsilon \dot{Q}_{\text{max}} = \dot{m}_g \left( h_7 - h_8 \right) \tag{26}\]

where \(\dot{Q}, \varepsilon, \dot{Q}_{\text{max}}\) are the actual heat rate exchanged in the air cooler, effectiveness factor, and maximum heat transfer rate, respectively. For a given inlet and outlet temperatures for the cold and hot fluids, \(\dot{Q}_{\text{max}}\), is calculated from:

\[
\dot{Q}_{\text{max}} = \left( \dot{m} C_p \right)_{\text{min}} \left( T_{g,\text{in}} - T_{a,\text{in}} \right) \tag{27}\]

where \(\left( \dot{m} C_p \right)_{\text{min}}\) is the minimum heat capacity rate for the fluid \((W/K)\) which can be selected from both hot or cold fluids when the product of the mass flow rate and the heat capacity is minimum:

\[
c_{\text{min}} = \left( \dot{m} C_p \right)_{\text{min}} = \min \left( \dot{m}_g C_{p,g}, \dot{m}_a C_{p,a} \right) \tag{28}\]

\[
c_{\text{max}} = \left( \dot{m} C_p \right)_{\text{max}} = \max \left( \dot{m}_g C_{p,g}, \dot{m}_a C_{p,a} \right) \tag{29}\]

The heat capacity ratio is expressed as \(c = \frac{c_{\text{min}}}{c_{\text{max}}}\).

The effectiveness factor depends on the heat exchanger flow arrangement and geometry. Cross-flow heat exchangers are used as the air coolers, and the external flow is permitted to mix. The following equations are used for
a cross-flow air cooler with one flow mixed and one flow unmixed:

For \( C_{\text{max}} \) mixed and \( C_{\text{min}} \) unmixed:

\[
\varepsilon = 1 - \exp \left[ -c \left( 1 - e^{-N} \right) \right]
\]

For \( C_{\text{min}} \) mixed and \( C_{\text{max}} \) unmixed:

\[
\varepsilon = 1 - \exp \left\{ - \left( \frac{1}{c} \right) \left[ 1 - \exp (-Nc) \right] \right\}
\]

where \( N \) is the number of transfer units (NTU), expressed as:

\[
N = \frac{U_{\text{AC}} A_{\text{AC}}}{(m C_p)_\text{min}}
\]

where \( U_{\text{AC}} \) is the overall heat transfer coefficient of the heat exchanger and \( A_{\text{AC}} \) is the air-side heat transfer area, which are extracted from the vendor datasheet. In the off-design operation, the heat exchanged in the air cooler is different than the design point reported in the cooler specification. For a fixed diameter of fan blade, the fan speed varies which causes the fan motor power to change. Here, the required electric power for part-load operation and the air flow rate for use in Equation (27) are obtained from the following equation:

\[
\dot{m}_a = \frac{C_{p-g}}{C_{p-a}} \dot{m}_g
\]

\[
\frac{n_{\text{AC-part-load}}}{n_{\text{AC-design}}} = \frac{m_{\text{a-part-load}}}{m_{\text{a-design}}} \frac{T_{\text{amb}}}{T_{\text{a-design}}}
\]

\[
\frac{W_{\text{AC-part-load}}}{W_{\text{AC-design}}} = \left( \frac{n_{\text{AC-part-load}}}{n_{\text{AC-design}}} \right)^3 \frac{T_{\text{a-design}}}{T_{\text{amb}}}
\]

where \( n \) is the fan rotational speed.

Considering the gas boosting station as a control volume, its energy efficiency can be expressed as:

\[
\eta_{\text{energy}} = \frac{W_{\text{gt}} - W_{\text{ac}} - W_{\text{ge}} + Q_{\text{AC}}}{\dot{m}_j \text{LHV} + W_{\text{elec}}}
\]

Energy analysis cannot indicate where and how irreversibilities occur during the process. Exergy analysis can be performed to evaluate the irreversibilities and to specify how the system performance can be improved. Exergy can be separated into four parts: physical, chemical, potential, and kinetic. In this research, potential and kinetic exergy are taken to be small and thus negligible because of minor changes in elevation and speed. In general, the exergy rate balance for any system is given by:

\[
\dot{E}_{\text{heat}} + \dot{in} \sum (\dot{m} e)_i = \dot{out} \sum (\dot{m} e)_c + \dot{E}_{\text{Work}} + \dot{E}_D
\]

where \( \dot{E}_D \) is the exergy destruction rate and other terms are defined as follows:

\[
\dot{E}_{\text{heat}} = \left(1 - \frac{T_0}{T_1}\right) Q_l
\]

\[
e = e_{ph} + e_{ch}
\]

\[
\dot{E}_{\text{Work}} = W
\]

\[
e_{ph} = (h - h_0) - T_0 (s - s_0)
\]

where the subscript 0 denotes the reference environment condition.

For a gas mixture, the specific chemical exergy can be calculated from:

\[
e^{\text{ch}} = \frac{1}{M_{\text{fuel}}} \left[ \sum_{i=1}^{n} y_i \overline{e}_{i}^{\text{ch}} + R_u T_0 \sum_{i=1}^{n} (y_i \text{ln} y_i) + \overline{g}_E \right]
\]

where \( \overline{e}_{i}^{\text{ch}} \) is the molar chemical exergy for constituent \( i \) of the mixture with the values reported in, \( \overline{g}_E \) is the Gibbs free energy on molar basis. When the pressure of the gas mixture is relatively low, \( \overline{g}_E \) is negligible.

In the current study, the exergy of each flow, as well as the exergy changes, is calculated for different components of the gas boosting station.

The equations for the exergy efficiency and exergy destruction rate are shown in Table 2 for different components. In general, the main mechanisms of entropy generation are heat transfer and friction. The exergetic inefficiencies of a gas boosting station are mainly caused by exergy destruction through heat transfer to the surrounding with large temperature differences. The maximum irreversibility occurs in the combustion chamber resulting from the combustion reaction as well as heat transfer of hot stream with high-temperature difference. Dunbar et al. showed that entropy production in the combustion chamber is due to fuel oxidation/diffusion, mixing, and internal heat transfer. Similarly, the air cooler exergy destruction is caused by the temperature difference between cold and hot streams.

The overall exergy efficiency of gas boosting station can be written as the flow exergy of natural gas exiting the boosting station divided by the sum of physical and chemical exergy flows into the gas boosting station:
which is considered to be constant at 0.02 s.43

In order to study the effect of ambient conditions on these emissions, the quantities of CO and NOx gases generated by the combustor are calculated as a function of adiabatic flame temperature of the primary zone of the combustor40,41 in order to study the effect of ambient conditions on these emissions:

\[
T_{pz} = A \sigma^a \exp \left( \beta (\alpha + \lambda)^2 \right) \alpha^x \theta^y \psi^z
\]

where \( \sigma \) is the dimensionless pressure \( \frac{P}{P_{ref}} \) (\( P \) is combustion pressure, \( P_2 \)), \( \theta \) is the dimensionless temperature \( \frac{T}{T_{ref}} \) (\( T \) is the combustor inlet temperature, \( T_2 \)), and \( \psi \) is the \( \frac{C}{Z} \) atomic ratio. \( \sigma = \varphi - 0.7 \) for \( \varphi \geq 1 \) and \( \sigma = \varphi \) for \( \varphi \leq 1 \) (where \( \varphi \) is the equivalence ratio). Furthermore, \( x^* \), \( y^* \), and \( z^* \) are quadratic functions of \( \sigma \) which can be written as:41

\[
x^* = a_1 + b_1 \sigma + c_1 \sigma^2
\]

\[
y^* = a_2 + b_2 \sigma + c_2 \sigma^2
\]

\[
z^* = a_3 + b_3 \sigma + c_3 \sigma^2
\]

where \( A, \alpha, \beta, \lambda, a_i, b_i, \) and \( c_i \) are constant. These constant values are provided in reference.42

The NOx and CO pollutant emissions (in grams per kilogram of fuel) can be calculated from:41

\[
m_{NOx} = \frac{0.15E16 \cdot 10^{0.5} \exp \left( - \frac{71100}{T_{pz}} \right)}{p_2^{0.5} \left( \frac{\Delta P}{P_2} \right)^{0.5}}
\]

\[
m_{CO} = \frac{0.179E9 \exp \left( \frac{2800}{T_{pz}} \right)}{p_2^{2.8} \left( \frac{\Delta P}{P_2} \right)^{0.5}}
\]

where \( \frac{\Delta P}{P_2} \) is the dimensionless pressure drop across the combustor and \( r \) is the residence time in the combustion zone which is considered to be constant at 0.02 s.43

The exergoenvironmental analysis is a combination of environmental and exergy analyses which shows the impact of exergy destruction on the environment. The exergoenvironmental factor which is directly proportional to the exergy destruction rate can be calculated from:44,45

\[
f_{el} = \frac{\dot{E}_{D-tot}}{\sum \dot{E}_{in}}
\]

The effectiveness factor for environmental damage is defined as:44,45

\[
\theta_{el} = \frac{f_{el}}{C_{el}}
\]

where \( C_{el} \) is the exergoenvironmental impact coefficient and can be calculated from:44,45

\[
C_{el} = \frac{1}{\eta_{ex}}
\]

Here, \( C_{el} \) denotes the environmental damages caused by a thermal system. Enhancing the exergoenvironmental impact has a positive influence on the environment and is obtained from:44,45

\[
\theta_{el} = \frac{1}{\theta_{el}}
\]

Finally, the exergy stability factor is defined as:39,44,45

\[
f_{es} = \frac{\dot{E}_{D-tot}}{\dot{E}_{D-tot} + \dot{E}_{out-tot} + 1}
\]

2.4 Exergoeconomic analysis

By combining exergy analysis with economic principles, exergoeconomic analysis can help the designers to improve the system operation cost-effectively.46 Exergoeconomic analysis provides useful insights into the cost rate of products, and their ties to component costs, which can indicate the economic behavior of the thermal systems. For each system component, the cost rate balance is written as follows:20,47

\[
\sum C_{out, k} + \dot{C}_{w, k} = \sum C_{in, k} + \dot{C}_{q, k} + \dot{Z}_k
\]

where \( C \) is the stream cost rate (US$/h) and \( Z_k \) is the capital cost of the component \( k \), while \( \dot{C}_{w, k} \) and \( \dot{C}_{q, k} \), respectively, are the cost rates associated with work and heat interactions for the component \( k \). The cost rate can be defined for stream \( j \) as:20,47
where $E_j$ is the exergy for the stream $j$ and $c_j$ is the specific cost. Therefore,

$$\sum (c_j E_j)_k + c_{w,k} W_k = \sum (c_i E_i)_k + c_{q,k} E_{q,k} + Z_k$$  \hfill (57)

The fuel and product streams for different components of the gas boosting station are defined in Table 3.

The component capital cost rate $Z_k$ includes the cost of equipment purchase and maintenance and can be obtained from.\cite{20,36}
### TABLE 4 System component cost equations

| Component                  | \( \dot{E}_F (MW) \) | \( \dot{E}_P (MW) \) |
|----------------------------|------------------------|------------------------|
| Air compressor             | \( \dot{W}_{ac} \)     | \( \dot{E}_2 - \dot{E}_1 \) |
| Combustion chamber         | \( \dot{E}_2 + \dot{E}_f \) | \( \dot{E}_3 \) |
| Gas turbine                | \( \dot{E}_3 - \dot{E}_s \) | \( \dot{W}_{gg} \) |
| Gas compressor             | \( \dot{W}_{gc} \)     | \( \dot{E}_7 - \dot{E}_6 \) |
| Air cooler                 | \( \dot{W}_{AC} \)     | \( \dot{E}_8 \) |

### TABLE 5 Exergoeconomic equations of gas boosting station components

| Component                  | Cost function (US$)                                                                 |
|----------------------------|--------------------------------------------------------------------------------------|
| Air compressor             | \( \frac{44.71\text{US$}}{0.95 - \dot{E}_{out}} \ln \left( \frac{\dot{E}_2}{\dot{E}_1} \right) \) |
| Combustion chamber         | \( \frac{28.98\text{US$}}{0.995} \left[1 + \exp \left(0.015 \left(T_3 - 1540\right)\right)\right] \) |
| Gas turbine                | \( \frac{301.45\text{US$}}{2.39 - \dot{m}_f} \ln \left( \frac{\dot{E}_3}{\dot{E}_5} \right) \left[1 + \exp \left(0.025\left(T_3 - 1570\right)\right)\right] \) |
| Centrifugal gas compressor | \( 7900 \ (\text{hp})^{0.82} \)                                                    |
| Air cooler                 | \( 30 \ (\text{Area})^{0.4} \)                                                   |

### TABLE 6 Specific costs of fuel, electricity, and pollution damage

| Parameter | Unit | Cost  | Source(s) |
|-----------|------|-------|------------|
| \( c_f \) | US$/MJ | 0.004 | 52         |
| \( c_{elec} \) | US$/kWh | 0.220 | 39         |
| \( C_{CO} \) | US$/kg | 0.02086 | 41        |
| \( C_{NOx} \) | US$/kg | 6.853 | 41         |

### TABLE 7 Comparison of the gas turbine model and datasheet real data

| Parameter | Unit | Simulation | Datasheet | Deviation (%) |
|-----------|------|------------|-----------|---------------|
| \( T_1 \) | K    | 321.1      | 321.1     | 0             |
| \( P_1 \) | mbar | 990        | 990       | 0             |
| \( r_p \) | -    | 10.9       | 10.9      | 0             |
| \( m_f \) | kg/h | 6613       | 6613      | 0             |
| \( m_s \) | kg/s | 114.41     | 114.41    | 0             |
| \( T_5 \) | K    | 844.4      | 807.2     | 4.6           |
| \( W_{net} \) | kW  | 24 058     | 24 420    | 1.48          |
$Z_k$ is the cost of component $k$ (in US$). The investment and installation cost of each piece of equipment can be calculated from the equations shown in Table 4. $\varphi$ is the maintenance factor (1.06 in the current study), $t_w$ is the annual number of working hours for the system, and CRF is the capital recovery factor which depends on the system lifetime and interest rate and can be calculated from:

$$\text{CRF} = \frac{i(1+i)^L}{(1+i)^L - 1}$$  \hspace{1cm} (60)

where $i$ is the interest rate and $L$ is the system lifetime. For the component $k$, the exergy destruction cost rate in (US$/s)$ can be expressed as:

$$\dot{C}_{D,k} = c_{F,k} \dot{E}_{D,k}$$  \hspace{1cm} (61)

where $c_{F,k}$ is the unit fuel cost in $k$th input line of the system (US$/MJ)$.

To find the exergy cost for each component, the cost balance equations are solved. In some components, the number of inlet and outlet flows is more than one for each stream. In these cases, auxiliary exergoeconomic equations shown in Table 5 along with the cost balance equations provide a matrix that results in the balance cost in each flow. To evaluate all cost parameters, linear equations can be used as follows:

$$[E_k] \times [C_k] = [Z_k]$$  \hspace{1cm} (62)
The exergoeconomic factor is defined as follows:\textsuperscript{20,35}
\[ f_k = \frac{\dot{Z}_k}{C_{D,k} + Z_k} \] (63)

A higher exergoeconomic factor is yielded as a result of lower exergy destruction rates. For component \( k \), a relatively low value of \( f_k \) reveals inefficiencies, and a high value of \( f_k \) implicates that capital investment is high.\textsuperscript{39}

The total cost rate of the gas boosting station can be calculated from:\textsuperscript{20}
\[ C_{\text{total}} = C_f + C_{\text{elec}} + \sum Z_k + \sum C_{D,k} + C_{\text{env}} \] (64)

where the cost rates can be determined for fuel, electricity, and environmental impact as follows:
\[ C_f = c_f m_f LHV \] (65)
\[ C_{\text{elec}} = c_{\text{elec}} W_{AC} \] (66)
\[ C_{\text{env}} = C_{CO} m_{CO} + C_{NOx} m_{NOx} \] (67)

The values for \( c_f, c_{\text{elec}}, C_{CO}, C_{NOx} \) are reported in Table 6.

3 | RESULTS AND DISCUSSION

To determine the system performance parameters and data required for exergy, environmental, and economic analyses of the gas boosting station, a simulation program is developed. In the present study, the performance of the natural gas compressor station is evaluated for various real-world operating conditions reported in the distributed control system. The mathematical modeling and 4E analyses calculations are performed with the help of computer codes written in Fortran software. Energy and exergy balances and efficiencies are formulated for each component, including the air compressor, combustion chamber, gas turbine, centrifugal compressor, and air cooler, as well as whole system. The results from the energy analysis are used to do the exergy, exergoenvironmental, and exergoeconomic analyses.

3.1 | Validation of the model

To validate the model, several parameters as shown in Tables 7 and 8 are compared with performance data of the site test datasheet which are provided by the system supplier (see ref.\textsuperscript{53,54}). According to the statistics of the boosting station, the system operates always based on nominal load. Therefore, the site test report does not contain any data from other load conditions. The comparison results are reported in Tables 7 and 8 separately for the gas turbine and gas compressor based on actual performance conditions. The test location and environmental conditions of the test site are as follows:

- Industrialized area
- Petrochemical industry
- Marine side conditions
- The high concentration of particles

As shown in Tables 7 and 8, the deviations between simulation and experimental results are less than 7% which can be generally considered as good agreement.

3.2 | Results of energy and exergy analyses

According to the gas turbine performance curve, the fuel consumption, airflow, and exhaust temperature vary with the air compressor inlet temperature. In cold and hot seasons, the minimum and maximum recorded ambient temperatures are 5°C and 48°C, respectively, and the compressor inlet pressure is 0.992 bar.

For the evaluated case in the present study, the natural gas is produced in an ethane recovery plant (C2 plant) where the extracted natural gas after filtration and drying enters the ethane recovery plant in order to separate ethane and other heavier hydrocarbons from that. The remaining sale gas then enters the gas boosting station to increase its pressure for supplying through the gas distribution network. The temperature and pressure of inlet sale gas to the boosting stations are almost constant during the year due to its upstream process. Therefore, the centrifugal compressor usually works at the same load condition. So, the

![Figure 2](image-url) Energy efficiencies of the driver and driven equipment in hot and cold seasons (for \( W_{\text{net}} = 25.6 \text{ MW} \)
results are representative of a fixed 25.6 MW power transmitted from gas turbine to the centrifugal compressor. Further characteristics of the gas turbine, gas compressor, and air cooler are presented in Tables 9-11.

According to the system schematic (Figure 1) and energy equations, the performance of the centrifugal compressor and air cooler is directly linked to the power generation of the gas turbine. When a gas turbine for mechanical driver application is running, adequate air flow through the compressor is controlled by adjusting the position of inlet guide vanes (IGVs) and turbine nozzles and also the rotational speed (rpm) of the high-pressure rotor.

In this study, to obtain the exergy destructions and exergy efficiencies as well as the results of exergoenvironmental and exergoeconomic analyses, several iterative solutions have been conducted to find the AFR for 25.6 MW power generation of the gas turbine with the same performance parameters in cold and hot seasons.

Figure 2 provides the energy efficiency of the gas turbine system as well as the isentropic efficiency of the gas compressor and change of compressibility factor in the suction and discharge sections for the hot and cold seasons. The isentropic efficiency of the gas compressor is slightly higher in summer. Moreover, the gas turbine

FIGURE 3 Exergy destruction rates of gas boosting station and its components (for TIT = 1300 K)

FIGURE 4 Effect of air compressor pressure ratio and ambient condition on total exergy destruction rate of overall system (for TIT = 1300 K)

FIGURE 5 Effect of air compressor pressure ratio and ambient condition on gas turbine exergy efficiency (for TIT = 1300 K)

FIGURE 6 Effect of air compressor pressure ratio and ambient condition on specific fuel consumption (for TIT = 1300 K)
energy efficiency is 1.4% higher in the hot seasons. This is consistent with the results reported in 55 and confirms that heating the inlet air temperature causes an increase in turbine efficiency at part-load conditions.

Figure 3 illustrates the exergy destruction rates of gas boosting station components in hot and cold seasons. As can be seen, the total exergy destruction in winter is 2.9% higher in comparison with summer. As expected, the most significant part of the overall exergy destruction is linked to the combustion chamber, at 64.6% and 60.2% in cold and hot seasons, respectively. This is due to the chemical reaction in the combustion process and heat transfer inside the combustion chamber which makes the combustor a source of significant exergy destruction. The combustor exergy efficiencies are 64.5% in maximum and 63.2% in minimum ambient temperatures.

The inefficiencies in the combustion chamber can be lowered by preheating the combustion air, enhancing the combustion process, and reducing AFR.39,56 This high exergy destruction rate of the combustor leads to interest in direct energy conversion devices such as fuel cells, in which chemical energy is directly converted to electrical energy without the intermediate production of heat.57
Although the compressor exergy destruction rate is lower in lower ambient temperature by up to 17%, the reduction in combustion chamber inlet temperature causes an inefficient combustion process.

It can be seen from Figures 4 and 5 that raising the air compressor pressure ratio lowers the irreversibilities of the gas turbine and the overall system. The effect of ambient conditions on the performance of the gas boosting stations is assessed considering the influence of pressure ratio on the overall exergy destruction rate for maximum and minimum recorded ambient temperatures. For a turbine inlet temperature (TIT) of 1300 K and fixed power, Figure 4 shows that the total exergy destruction rate decreases as the air compressor pressure ratio rises. Furthermore, in summer the gas boosting station destroys less exergy in comparison with winter.

Similarly, the impact of pressure ratio and ambient conditions on the exergy efficiency of the natural gas-fired gas turbine is illustrated in Figure 5. As pressure ratio rises, the exergy efficiency of the gas turbine increases, lowering the fuel consumption of the gas turbine cycle. This leads to reducing the significant amount of fuel chemical exergy which is supplied to the combustor. In the case of gas boosting stations where exhaust gases are directly released to the atmosphere, employing a recuperator can be useful to enhance the performance of the air compressor at low ambient temperatures and to recover waste heat to preheat air entering the combustion chamber.

It can be seen from Figures 5 and 6 that the rise in the efficiencies of the system due to an increase in the air compressor pressure ratio improves the fuel consumption. Figure 6 also shows that, for a 60% boost in the pressure ratio from 10.9 to 17.44, the amount of specific fuel consumption is improved by 20.5% in winter and 19.3% in summer.

The energy and exergy efficiencies as well as specific fuel consumption of the gas turbines in normal operation of gas boosting stations in both seasons based on Equation (16) are compared, and the results are shown in Figure 7. The energy and exergy efficiencies of the boosting station are, respectively, 4.4% and 3.0% higher in summer. In contrast, the value of SFC is 6% higher in winter.

### 3.3 Exergoenvironmental analysis

Figures 8 and 9 illustrates the variations in the emission rates of NOₓ and CO with pressure ratio. Although SFC declines as pressure ratio increases, the combustion chamber primary zone temperature dramatically increases due to the rise in compressor outlet temperature and pressure. The rate of increase in NOₓ emission is more sensitive to pressure ratio in the hot seasons as seen in Figure 8. However, the CO emission is reduced significantly by increasing the pressure ratio and ambient temperature (see Figure 9).

Figure 10 shows the exergy stability and exergoenvironment factors as well as the environmental damage effectiveness factor of the system in hot and cold seasons. According to Equation (50), the greatest exergoenvironmental factor is attributed to the high exergy destruction rate. The exergoenvironmental factor is slightly higher in summer because of the high input exergy of the system. Moreover, in the cold season, the gas boosting station supplies more desired exergy output. The lower exergoenvironmental factor results in a more sustainable system. Also, the lower environmental damage effectiveness
factors are more favorable. This factor is 0.393 and 0.401 in hot and cold seasons, respectively. This index is in direct relationship with both exergy destruction and exergy efficiency values. Therefore, during the normal operation of the system, a 3% rise in exergy efficiency in summer results in a 2% reduction in environmental damage effectiveness factor in comparison with winter. Based on Equation (54), low exergy stability factor values are preferred because they involve less exergy destruction. It can be observed from Figure 10 that the value of this factor in winter is slightly higher because of the higher exergy destruction.

The output power of the gas turbine and its efficiency is strongly related to the TIT. The air-to-fuel ratio decreases when TIT increases causing the total exergy destruction rate of the system to drop. Therefore, the exergoenvironmental index is improved effectively by increasing TIT. As can be seen from Figure 11, the exergoenvironmental factor can be improved by more than 18% in hot season when TIT increases from 1200 to 1500 K. Moreover, exergy stability and environmental damage effectiveness factors have the greatest decrease in hot season, about 9.6% and 13.1%, respectively.

### 3.4 Exergoeconomic analysis

The exergoeconomic results of each stream of the gas boosting station are presented in Table 12.

The exergoeconomic parameters for each component of the proposed system for the normal operating conditions in both hot and cold seasons are listed in Table 13.

As shown in Table 13, the combustion chamber exhibits the greatest value of $\dot{C}_D + \dot{Z}_k$ and a low exergoeconomic factor in both seasons. As mentioned before, the low value of $f_k$ in the combustor reveals inefficiencies which can be enhanced by increasing the capital investment through increased gas turbine inlet temperature (considering limitations in the value of TIT due to metallurgical conditions). In the case of centrifugal compressors, the exergoeconomic factor is high in both seasons due to the large capital investment cost. This costly investment is mainly attributed to the extensive power requirement, which needs to be improved. Turbomachinery sections of the turbine significantly contribute to exergy destruction and result in substantial exergy destruction cost rates of 195.8 US$/h and 175.0 US$/h for the air compressor and 147.8 US$/h and 136.6 US$/h for the gas turbine in the hot and cold seasons, respectively. In the air cooler, considerable differences between values of $\dot{Z}_k$ and $\dot{C}_D$ reveal a small exergoeconomic factor. The fuel costs in the air cooler are 12.54 US$/GJ and 13.08 US$/GJ in the hot and cold seasons, respectively, which are the highest fuel costs among all components. This causes a high exergy destruction cost rate. In this case, an increase in heat transfer area causes $\dot{Z}_k$ to rise and results in a higher $f_k$, which implies better performance of coolers.

Figure 12 demonstrates the effect of TIT and ambient condition on the exergy destruction cost rate of the gas boosting station. As can be seen from the figure, increasing TIT from 1200 K to 1500 K improves the cost rate of the exergy destruction by more than 15% in hot and cold seasons for the whole system. At a fixed air compressor pressure ratio, a higher TIT results in a higher exhaust temperature which can be used in a heat recovery system. Different strategies have been proposed in 12 in order to recover the energy and exergy in a gas compressor station.

According to Equation (64), the total cost rate of the gas boosting station is 7390.9 US$/h in summer and 8074.2 US$/h in winter (9.2% higher in winter).

A breakdown of total cost rates is illustrated in Figures 13 and 14. It can be seen that more than 60% of the total costs are related to the environmental impact costs in the hottest and coldest environmental conditions. The contribution of capital investment cost is 2.2% higher in the hot season and the contribution of exergy destruction cost rate decreases by 0.77% in winter. The slight reduction in electricity cost in winter implies a decrease in fan load due to lower ambient temperatures.

| Component          | Hot season | Cold season |
|--------------------|------------|-------------|
| $c_f$ (US$/GJ)     | $\dot{Z}_k$ (US$/h) | $\dot{C}_D$ (US$/h) | $f$ (%) | $c_f$ (US$/GJ) | $\dot{Z}_k$ (US$/h) | $\dot{C}_D$ (US$/h) | $f$ (%) |
| Air compressor     | 7.3        | 30.5        | 195.8     | 13.5    | 7.94        | 28.5        | 175.0     | 14.03   |
| Combustion chamber | 5.64       | 6.24        | 667.9     | 0.92    | 5.59        | 5.84        | 730.0     | 0.79    |
| Gas turbine        | 5.94       | 56.1        | 147.8     | 27.5    | 6.16        | 54.3        | 136.6     | 28.46   |
| Centrifugal compressor | 7.3      | 124.2       | 157       | 44.16   | 7.94        | 124.5       | 179.5     | 40.94   |
| Air cooler         | 12.54      | 0.023       | 65.0      | 0.03    | 13.08       | 0.023       | 63.9      | 0.036   |
Detailed energy, exergy, exergoenvironmental, and exergoeconomic analyses are presented for a natural gas boosting station using a real-gas model and actual operational data. The impacts of several operating parameters on efficiencies, exergy destruction, specific fuel consumption, pollutants emission, and cost rates are investigated. To validate the simulation, selected turbine and centrifugal gas compressor parameters are compared against real performance data and good agreement was observed. The main conclusions from the present study are as follows:

- Energy and exergy efficiencies for the natural gas boosting station are, respectively, 9.5% and 76.1% for the hot season and 9.1% and 73.9% for the cold season. Furthermore, higher efficiencies in the hot season reduce fuel consumption by 6% relative to the cold season.
- The main exergy destruction is attributed to the combustion chamber where 64.6% and 60.2% of total exergy destruction of the gas boosting station occurs in the hot and cold seasons, respectively. However, less than 3% of the total exergy destruction rate occurs in the air cooler regardless of the ambient temperature.
- Raising the air compressor pressure ratio decreases the total exergy destruction rate, specific fuel consumption, and CO emission. However, this increases the NOx emission.
- For the gas boosting station, the exergoenvironmental factors are 0.300 and 0.296, the effectiveness factors are 0.393 and 0.401, and the exergy stability factors are 0.281 and 0.284, for operation in hot and cold seasons, respectively.
- For the gas boosting station, the highest exergoeconomic factors are attributed to the centrifugal gas compressor, at 44.2% in the hot season and 40.9% in the cold season, mainly due to the large capital investment cost. In contrast, the lowest exergoeconomic factor belongs to the air cooler, at 0.030% and 0.036% in maximum and minimum ambient temperatures, respectively.
- The cost rate of exergy destruction decreases by more than 15% when turbine inlet temperature increases by 25% from 1200 K, and exergoenvironmental index is improved for all components of the gas boosting station.
- The total cost rates of the gas boosting station are about 7390 US$/h in summer and 8070 US$/h in winter, of which the major contributions are due to environmental impact.

**FIGURE 12** Effect of TIT and ambient condition on cost rate of exergy destruction

**FIGURE 13** Breakdown (in %) of total cost rate for the overall system in the hot season

**FIGURE 14** Breakdown (in %) of total cost rate for the overall system in the cold season

**NOMENCLATURE**

\[ A \] area (m²)
\[ a_1, a_2, a_3 \] constant parameters
\[ b_1, b_2, b_3 \] constant parameters
\( C \)  
\( \text{cost (US$)} \)

\( C \)  
\( \text{heat capacity ratio} \)

\( c_1, c_2, c_3 \)  
\( \text{constant parameters} \)

\( C_{ei} \)  
\( \text{exergoenvironmental impact coefficient} \)

\( C_p \)  
\( \text{specific heat at constant pressure (kJ/kg K)} \)

\( \text{CRF} \)  
\( \text{capital recovery factor} \)

\( e \)  
\( \text{specific exergy (kJ/kg)} \)

\( e_{ch} \)  
\( \text{specific chemical exergy (kJ/kg)} \)

\( E \)  
\( \text{exergy (kJ)} \)

\( E_{D} \)  
\( \text{exergy destruction rate (kW)} \)

\( E_{f} \)  
\( \text{exergy rate of fuel (kW)} \)

\( E_{p} \)  
\( \text{exergy rate of product (kW)} \)

\( f_{ei} \)  
\( \text{exergoenvironmental factor} \)

\( f_{es} \)  
\( \text{exergy stability factor} \)

\( f_k \)  
\( \text{exergoeconomic factor} \)

\( gE \)  
\( \text{Gibbs free energy on molar basis (kJ/kmol)} \)

\( H \)  
\( \text{specific enthalpy (kJ/kg)} \)

\( H_p \)  
\( \text{power (hp)} \)

\( I \)  
\( \text{interest rate} \)

\( L \)  
\( \text{lifetime of system (years)} \)

\( LHV \)  
\( \text{lower heating value (kJ/kg)} \)

\( m \)  
\( \text{mass flow rate (kg/s)} \)

\( n \)  
\( \text{fan rotational speed (rpm)} \)

\( N \)  
\( \text{number of transfer units} \)

\( P \)  
\( \text{pressure (kPa)} \)

\( Q \)  
\( \text{heat transfer rate (kW)} \)

\( r_p \)  
\( \text{pressure ratio} \)

\( R \)  
\( \text{specific gas constant (kJ/kg K)} \)

\( R_a \)  
\( \text{universal gas constant (kJ/kmol K)} \)

\( s \)  
\( \text{specific entropy (kJ/kg K)} \)

\( t_w \)  
\( \text{annual duration of working (hours)} \)

\( T \)  
\( \text{temperature (K)} \)

\( T_{pz} \)  
\( \text{flame adiabatic temperature in combustor primary zone (K)} \)

\( U \)  
\( \text{overall heat transfer coefficient (W/m²K)} \)

\( V \)  
\( \text{gas specific volume (m³/kg)} \)

\( V \)  
\( \text{velocity (m/s)} \)

\( W \)  
\( \text{work rate (kW)} \)

\( x^*, y^*, z^* \)  
\( \text{quadratic functions} \)

\( Y \)  
\( \text{mole fraction} \)

\( Z \)  
\( \text{compressibility factor} \)

\( 
\( \dot{Z} \)  
\( \text{capital cost rate (US$/h)} \)

\( \psi \)  
\( \text{H/C atomic ratio} \)

\( \epsilon \)  
\( \text{effectiveness factor} \)

\( \pi \)  
\( \text{dimensionless pressure} \)

\( \tau \)  
\( \text{residence time in combustion zone (s)} \)

SUBSCRIPTS

0  
\( \text{reference state condition} \)

1, 2, ..., 8  
\( \text{points in Figure 1} \)

A  
\( \text{air} \)

Ac  
\( \text{air compressor} \)

AC  
\( \text{air cooler} \)

act  
\( \text{actual} \)

avg  
\( \text{average} \)

Cc  
\( \text{combustion chamber} \)

E  
\( \text{output} \)

elec  
\( \text{electricity} \)

env  
\( \text{environment} \)

ex  
\( \text{exergy} \)

exh  
\( \text{exhaust} \)

F  
\( \text{fuel} \)

F  
\( \text{fuel stream} \)

G  
\( \text{gas} \)

gc  
\( \text{centrifugal gas compressor} \)

gen  
\( \text{generation} \)

gt  
\( \text{gas turbine} \)

I  
\( \text{input, component number} \)

isen  
\( \text{isentropic} \)

J  
\( \text{stream number} \)

K  
\( \text{component number} \)

mech  
\( \text{mechanical} \)

P  
\( \text{product} \)

Q  
\( \text{heat} \)

S  
\( \text{isentropic} \)

tot  
\( \text{total} \)

SUPERSCRIPTS

–  
\( \text{unit-mole basis} \)

·  
\( \text{rate} \)

ORCID

Ali Momenimovahed  
https://orcid.org/0000-0002-1639-3860

REFERENCES

1. Mikolajková M, Haikarainen C, Saxén H, Pettersson F. Optimization of a natural gas distribution network with potential future extensions. Energy. 2017;125:848-859.

2. Mahmoudimehr J, Sanaye S. Minimization of fuel consumption of natural gas compressor stations with similar and dissimilar turbo-compressor units. J Energy Eng. 2014;140(1):04013001.

3. Hamedi M, Zanjirani Farahani R, Husseini MM, Esmaelizadeh GR. A distribution planning model for natural gas supply chain: a case study. Energy Pol. 2009;37(3):799-812.
42. Gülder Ö. Flame temperature estimation of conventional and future jet fuels. J Eng Gas Turbines Power-transactions ASME. 1986;108:376-380.
43. Rizk N, Mongia HC. Semianalytical correlations for NOx, CO, and UHC emissions. ASME J Eng Gas Turbines Power. 1993;115(3):612-619.
44. Midilli A, Dincer I. Development of some exergetic parameters for PEM fuel cells for measuring environmental impact and sustainability. Int J Hydrogen Energy [Internet]. 2009;34(9):3858-3872. https://doi.org/10.1016/j.ijhydene.2009.02.066
45. Ratlamwala TAH, Dincer I, Reddy BV. Exergetic and environmental impact assessment of an integrated system for utilization of excess power from thermal power plant. In: Dicer I, Colpan C, Kadioglu F (eds), Causes, Impacts and Solutions to Global Warming. New York, NY, USA: Springer. 2013;(pp. 803-824). https://doi.org/10.1007/978-1-4614-7588-0_42
46. Tsatsaronis G. Definitions and nomenclature in exergy analysis and exergoeconomics. Energy. 2007;32(4):249-253.
47. Sayyaadi H, Sabzaligol T. Exergoeconomic optimization of a 1000 MW light water reactor power generation system. Int J Energy Res. 2009;33(4):378-395.
48. Miar Naeimi M, Eftekhari Yazdi M, Reza SG. Energy, exergy, exergoeconomic and exergoenvironmental analysis and optimization of a solar hybrid CCHP system. Energy Sources, Part A Recover Util Environ Eff [Internet]. 2019;41:1-21. https://doi.org/10.1080/15567036.2019.1702122
49. Sahoo PK. Exergoeconomic analysis and optimization of a cogeneration system using evolutionary programming. Appl Therm Eng. 2008;28(13):1580-1588.
50. Roosen P, Uhlenbruck S, Lucas K. Pareto optimization of a combined cycle power system as a decision support tool for trading off investment vs. operating costs. Int J Therm Sci. 2003;42(6):553-560.
51. Couper JR, Penney WR, Fair JR, Walas SM. Chemical Process Equipment: Selection and Design. Houston, Texas, USA: Gulf Professional Publishing; 2005.
52. Shamoushaki M, Ehyaei MA. Exergy, economic, and environmental (3E) analysis of a gas turbine power plant and optimization by MOPSO algorithm. Therm Sci. 2018;22(6 Part A):2641-2651.
53. Linde AG. Datasheet gas turbine driver for sale gas compressor. Report No.: M-DE 1273; 2001.
54. Linde AG. Datasheet sales gas compressor. Report No.: M-DE 1272.001; 2001.
55. Liu ZT, Ren XD, Yan ZY, et al. Effect of inlet air heating on gas turbine efficiency under partial load. Energies. 2019;12(17):3327.
56. Almutairi A, Pilidis P, Al-Mutawa N. Exergetic and environmental analysis of 100 MW intercooled gas turbine engine. Green Energy Technol. 2018;863-882.
57. Yuan D. Energy and Exergy Evaluations on a Microturbine System. Canada: University of Toronto; 2007.

How to cite this article: Salimi Delshad M, Momenimovahed A, Mazidi MS, Ehyae MA, Rosen MA. Energy, exergy, exergoenvironmental, and exergoeconomic (4E) analyses of a gas boosting station. Energy Sci Eng. 2021;9:2044-2063. https://doi.org/10.1002/ese3.966