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Exergy Recovery in Gas Pressure Compression Stations (GPCSs)

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Abstract: The exergy analysis is a proper method for performance evaluation of industrial systems. A generic and detailed analysis of the GPCSs on the second gas pipeline of Iran is made by the means of exergy. The two main improvement measures of fuel pre-heating and steam injection technologies are presented for the current conventional stations. Steady state equations regarding the second law of thermodynamics and the chemical and physical exergy analysis are presented as well. The results indicate that the improved cycle is a more energy saving one, with an overall efficiency and net output power. The exegetic efficiency of every gas turbine of the improved station is increased by 31% in average and their exergy destruction is decreased by 84%. The amount of total exergy saving for the case study would be 552 MW. A higher overall efficiency can be achieved by an increase in both the turbine inlet temperature (TIT) and steam mass flow (SMF).

Keywords: Gas Compression Stations, Steam Injection, Exergy Analysis, Exergy Destruction.

1. Introduction

The compression stations raise gas pressure to compensate the pressure drops resulting from frictional gas flow in the pipeline. This compression is usually generated in centrifugal or reciprocating compressors driven by gas turbines or electric motors. Simple and regenerative gas turbine cycles are the candidates for the distributed systems because of their simple construction (Chaczykowski et al., 2011; Safarian et al., 2013; Nishida et al., 2005).

Gas turbines in simple-cycle mode are applied in utilities for limited peak power generation, industrial facilities use gas turbine units for on-site power generation. To increase total efficiency of the units, the gas turbines are usually applied together with processes of heat production, like hot water and steam production. The performance of industrial gas turbines has improved due to considerable investments in research and development in conversion efficiency of fuel to electricity, plant capacity, availability and reliability (Poullikkas, 2005).

Gas turbines are vastly applied together with steam cycle, either to generate only electricity, or to cogenerate both electrical power and heat for industrial processes and district heating. Gas-fired combustion turbines and combined-cycle plants are expected to capture over 47% of the international and 80% of the U.S. new-generation market in the next decade (Najjar, 2005). However this method may not be feasible in a small scale power plant(s) because of extra capital investments needed for a high-pressure steam generator, a steam turbine, a condenser and special water treatment facilities (Korobitsyn, 2002).

Najjar et al., (1992) studied on industrial, design, applications and economics of cogeneration. They suggested utility of the waste energy in the form of cogeneration in preventing further degradation of the environment and achieving the objective of

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greater economic competitiveness. Najjar and Akyurt (1993) utilized the waste heat from exhaust gases of gas turbine as the fuel in cogeneration technology; they proved that the combined cycles boost the output power and efficiency in a considerable manner in comparison with steam power plants and reduced the capital cost by converting an existing steam plant to combined cycle.

Nishida et al., (2005) introduced two types of regenerative steam injection gas turbine (RSTIG) and compared their obtained results with that of the simple, regenerative, water injection and STIG cycles. They revealed that the thermal efficiencies of the RSTIG systems are higher than that of all conventional cycles and the specific power is greater than the regenerative cycle.

Ghazikhani et al., (2011) run exergy analysis on two new high-performance cycles for gas turbine. They introduced evaporating gas turbine with air bottoming cycle (EGT-ABC), and steam injection gas turbine with air bottoming cycle (STIG-ABC). They revealed that at the same up-cycle pressure ratio and turbine inlet temperature (TIT), a higher overall efficiency can be achieved through the EGT-ABC cycle.

Safarian and Bararzadeh, (2013) assessed the performance of water and steam injection in the gas turbine incorporated to ABC based on an exergy analysis. They found that the steam injection cycles have minimum exergy loss and maximum output power at the same bottoming cycle pressure ratio and turbine inlet temperature. This phenomenon is due to more heat recovery at the regenerator in the SI-ABC cycles, lower exhaust temperature and more inlet mass to bottoming turbine.

Since the temperatures of the exhaust gases from simple and regenerative cycles are high, steam can be generated in a heat-recovery steam generator (HRSG).The generated steam can be utilized in several thermal processes or as the working fluid for a gas or steam turbine. In STIG systems, the steam is returned to the gas turbine system and utilized as the working fluid together with air, and this leads to more electric generation efficiency in simple and Regenerative cycles, Nishida et al., (2005).

Steam injection is a technique able to increase a plant's ability to generate extra power without burning extra fuel, while requiring moderate capital investment Koivu, (2007). A generic and detailed analysis of the GPCSs on the second pipeline of Iran is presented by the means of exergy analysis and thermodynamic laws in this article. The considered cycles are in form of current conventional station (simple) and improved station with fuel pre-heating and steam injection systems. All the concentration is on the released exhaust gasses from the gas turbine of the gas combustion system. The pre-heating process is accomplished by heating the inlet fuel before it is introduced to the combustion chamber of the system by the means of heat exchanger, and taking advantage of the high temperature of the hot exhaust gases from the gas turbine. Cooling the exhaust gases in a heat recovery steam system in the presence of fresh water leads to steam production which in turn can be injected at the combustion chamber level to save a portion of the fuel consumed in combustion, hence, a decrease in operational cost. Both the improvements would contribute in achieving higher system overall efficiency and lower exergy destruction rates as well as enhancing the combustion process.

2. Process Description

2.1. Assumptions

The overall efficiency and exergy loss of GPCS are estimated based on the operating conditions which are provided by Iran hydrocarbon balance, (2009) and National Iranian Gas Company, (2011). The main assumptions adopted in this analysis are reported in Table1.

Table 1. Conditions and Assumption Used in Calculation

Thermodynamics Assumptions	
Isentropic compressor efficiency	80%
Isentropic turbine efficiency	80%
HRSG pressure drop (water and steam)	6%
HRSG pressure drop (flue gases)	2%
Heat exchanger pressure drop ($\Delta P_{H.E.Gas}$)	2%
Combustion chamber pressure drop	2%
Combustion efficiency	98%
Air compressor pressure ratio (r_c)	5
Gas turbine pressure ratio (r_t)	6
(Fuel temperature) T_{fuel} °C	25
(Air temperature) T_{air} °C	25
Air- fuel ratio (m^3/m^3)	11.42

2.2. Process Approach

This study consists of the calculating the three cycles of: simple GPCS, GP-GPCS and ST-GP-GPCS. The simple GPCS is presented in Fig. (1).

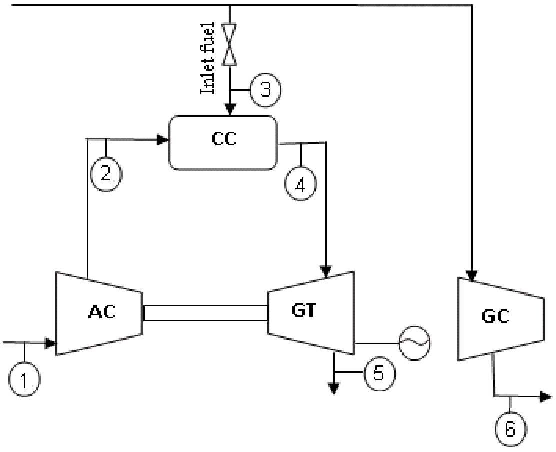

Figure 1. Schematic Diagram of GPCS

Due to high pressure of inlet natural gas, the reduction valves must be applied, (state 1), next, the compressed air (state 2) combusts with medium pressure fuel (state 3) and consequently fired-gases exit from combustion chamber and enter the turbine (state 4), where output power is obtained by considering the combustion chamber pressure drop, efficiency and the fuel mass flow rate

$$H_4 - H_2 = \eta_{CC}\, \dot{m}_{fuel}\, Q_{LHV} \tag{3}$$

$$\eta_{GT} = \frac{h_4 - h_5}{h_4 - h_{5s}} \tag{4}$$

$$W_{net} = H_4 + H_1 - H_5 - H_2 \tag{5}$$

The operating conditions and calculation results like the output power, net output power and overall efficiency for GPCSs are tabulated in Table 2.

The ST-GP-GPCS system, heat exchanger (for pre-heating the inlet natural gas), HRSG and a mixer (for production and injection of steam) are the main differences between simple GPCS and ST-GP-GPCS, Fig. (2). In the first improved step the TIT is increased to 1200 °C by contacting high temperature exhaust gases (state 8) and inlet

Table 2. Net Output Work and Overall Efficiency, GPCS

Station name	Inlet fuel flow rate (m^3/s)	TIT (°C)	P_{in} (MPa)	P_{out} (MPa)	W_t (MW)	W_{net} (MW)	η (Efficienty)
Patave2	4.03	1000	4.90	7.27	29.3	11.96	9
Dorahan2	4.02	1000	5.31	7.27	29.26	11.92	8.98
Dehagh2	3.93	1000	5.61	7.27	28.6	16.12	12.43
Esfahan	4.01	1000	3.92	7.20	29.2	15.12	11.42
Ghom2	3.78	1000	4.99	7.20	27.5	12.96	10.39
Noorabad	3.96	1000	5.17	7.44	28.8	16.25	12.44
Farashband	3.95	1000	4.80	8.38	28.75	14.74	11.31
Ghazvin2	3.78	1000	5.25	7.24	27.5	13.84	11.1

Table 3. The Main Results of GP- GPCS and ST-GP-GPCS Calculations

	Improved 1 [a]				Improved 2 [b]				
Station	TIT (°C)	W_t (MW)	W_{net} (MW)	η (%)	TIT (°C)	SR (kg/kgFuel)	W_t (MW)	W_{net} (MW)	η (%)
Patave2	1200	31	13.65	10.28	1200	7.5	34.5	34.5	12.92
Dorahan2	1200	30.8	13.45	10.14	1200	7.5	34.1	16.76	12.63
Dehagh2	1200	30.2	17.7	13.66	1200	7.5	39.92	20.44	15.76
Esfahan	1200	30.6	16.5	12.48	1200	7.5	33.5	19.42	14.67
Ghom2	1200	29	14.45	11.60	1200	7.5	31.2	16.66	13.36
Noorabad	1200	30.3	17.75	13.59	1200	7.5	33	20.45	15.66
Farashband	1200	29	14.45	11.60	1200	7.5	31.2	16.66	13.36
Ghazvin2	1200	29	15.35	12.30	1200	7.5	31.2	17.54	14.07

a GP-GPCS
b SI-GP-GPCS

fuel (state 3) leading to overall efficiency of every GPCS with an 11% on average increase. In the second improved step steam is produced by exchanging heat between the hot exhaust gases (state 9) and the fresh water (state 5) in the HRSG.

Here, the steam (state 6) is then mixed with air and injected into the combustion chamber or turbine inlet. Steam injection into turbine inlet improves the SFC by half and boosts power by 1.5 fold in comparison with the combustor, though it costs less in terms of compressor operability.

Finally the fired-gases and steam expand in the turbine and thereafter the cycle is repeated again. In this case the overall efficiency of GPCSs is increased by 31% on average, since the amount of steam per unit flow of fuel is 7.5 (kg/kg-fuel), Eq. (6). The main results of the calculations on GP-GPCS and ST-GP-GPCS are tabulated in Table 3.

$$\eta_{Mean} = \frac{\sum_{i=1}^{n}\left(\frac{\eta_{developed} - \eta_{simple}}{\eta_{simple}}\right)}{n} \qquad (6)$$

The total net output power for this case study would be 1.11 times the simple GPCS obtained through gas pre-heating (TIT 1200 °C) and by developing conventional stations with gas pre-heating and steam injection (TIT and SR are 1200°C and 7.5 kg/kg-fuel), respectively. The total net output power would be 1.3 times the simple GPCS.

2.3. Effect of Injected Steam Flow

Description steam injection is applied in industrial engines to reduce NOx emissions and boost power. SFC is improved as steam is increased through the heat generated from exhaust gasses in a HRSG. The chilling effect of steam in the combustor is much less and more uniform than that of liquid water; hence, there is little or no increase in CO emission or combustion noise.

To increase the output power and improve operating efficiency of the basic Brayton cycle, high quality steam should be applied. In this article the amount of steam ratio for every GPCS varies between 0 to 25 (kg/kg-fuel) and the turbine inlet temperature is 1200 °C, these variations cause specific net output work change from 0.2 to 6 (kW/kg/hr), Fig. (3).

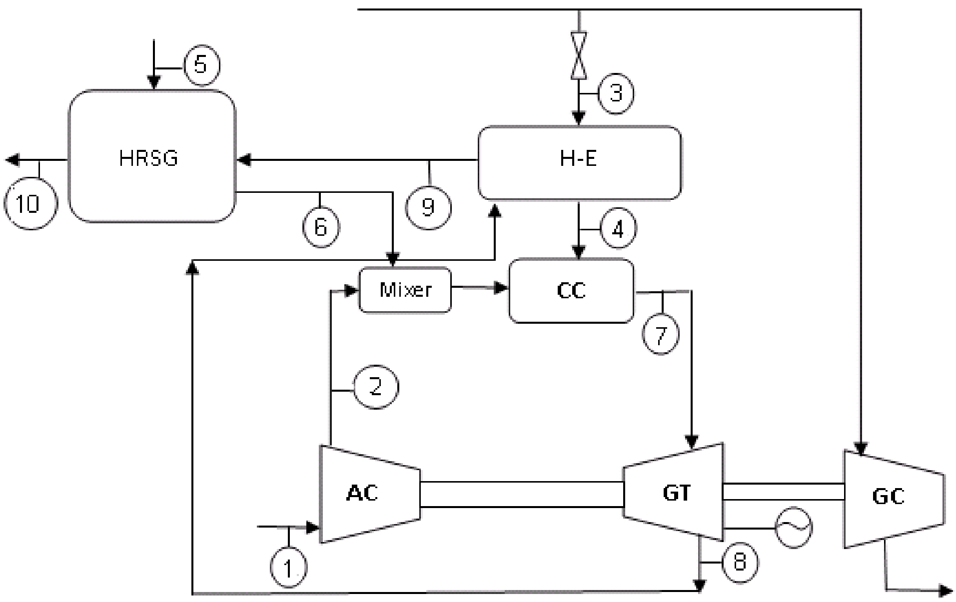

Figure 2. Schematic Diagram of ST-GP-GPCS

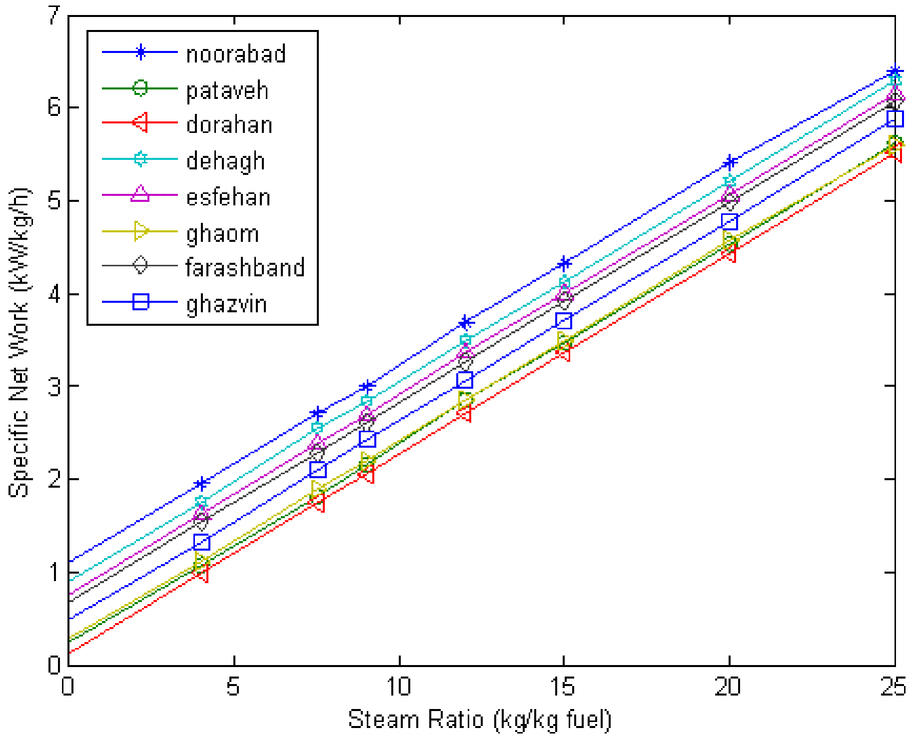

Figure 3. Specific Net Output Work Proximity for all Stations

3. Exergy Evaluation of GPCS

The exergy of a system is the maximum obtainable work as the system comes to equilibrium with the surroundings. The first law of thermodynamics makes only an energy balance of a system or a control volume; it does not make any distinction between different forms of energy, particularly work and heat, or heat at different temperatures (internal energy). It is the secon thermodynamic law, verified in engineering viewpoint; all forms of energy are not of the same quality. Energies of both the systems may be equal quantitatively but not qualitatively. Exergy is a measure of energy quality and exegetic efficiency (or second law) is a measure of a perfect thermal system. While energy of a system in a process remains constant, a portion of its exergy may be dropped and another is always destroyed (Nag, 1995; Kotas, 1985). Exergy flow rate of a system is composed of kinetic, potential, physical and chemical:

$$E = E_k^{\bullet} + E_p^{\bullet} + E_{ph}^{\bullet} + E_{ch}^{\bullet} \quad (7)$$

And the specific exergy rate is:

$$e = e_k + e_p + e_{ph} + e_{ch} \quad (8)$$

where,

$$e = \frac{E^{\bullet}}{m^{\bullet}} \quad (9)$$

In this study the kinetic (e_k) and potential (e_p) exergy are assumed negligible. Physical exergy is the obtainable work by taking the substance through reversible processes from its initial state temperature T and pressure P, to the state determined by temperature T_0 and pressure P_0 of the environment and be calculated through the following equation introduced by, Ameri and Enadi, (2012):

$$e_{ph} = h - h_0 - T_0(s - s_0) \quad (10)$$

where, *h* is the specific enthalpy and *s* the specific entropy .

The physical exergy can be split into a thermal and a pressure component, named mechanical component. By adopting the perfect gas laws and assuming a constant specific isobaric heat capacity follows (Safarian & Aramoun, 2015):

$$e_{ph} = h - h_0 - T_0[S - S_0 - RLn(\frac{P}{P_0})] \quad (11)$$

is yield.

Chemical exergy is equal to the maximum amount of obtainable work when the substance under consideration is brought from the environmental state, defined by the parameters T_0 and P_0, to the reference state by processes involving heat transfer and exchange of substances only through the environment. The value of chemical exergy for the flue gas can be calculated through the following equation introduced by Safarian and Aramoun, (2015):

$$e_{ch} = \sum_i (xe_{ch})_i - RT_0 \sum_i x_i Lnx_i \quad (12)$$

where, $e_{ch,i}$ is the standard exergy of component gas *i*, and *xi* is the volume percent of the component gas *i*, and R, is the gas constant. Standard enthalpy of devaluation and exergy of some substances are tabulated in Table 4 (Dincer & Rosen, 2007).

Table 4. Standard Enthalpy of Devaluation and Exergy of some Substance

substance	ΔH_0 (kJ/kmol)	e_{ch} (kJ/kmol)
CH_4	802320	836510
CO_2	0	19.87
H_2O	0	9.5
N_2	0	0.72
O_2	0	3.97

By determining the physical and chemical exergy, the energy balance and exergy efficiency for each component is evaluated. The energetic or second law efficiency is defined by:

$$\eta_{ex} = \frac{\text{Exergy rate of Product}}{\text{Exergy rate of Fuel}} \quad (13)$$

The exergy losses can be calculated from exergy balance equation for steady-state processes through the following equation introduced by Boonnasa and Namprakai, (2004) and Cornelisson, (1997):

$$E^{\bullet}_{loss} = \sum_j (E^{\bullet}_j)_{in} - \sum_i (E^{\bullet}_i)_{out} - W \quad (14)$$

Calculation results of exergy flow rate for the mentioned simple GPCS and ST-GP-GPCS are tabulated in Tables 5 and 6, and the other results of evaluation, exergy efficiency and exergy destruction of gas turbines are tabulated in Table 7. The results are calculated in constant steam ratio of (kg/kg-fuel), compressor pressure ratio and turbine pressure ratio of 7.5, 5 and 6, respectively.

Table 5. Exergy Flow Rate for Simple GPCS (MW)

	Simple Cycle				
	E_1	E_2	E_3	E_4	E_5
Patave2	0	8.72	139.5	138	77.9
Dorahan2	0	8.52	139.3	155	84
Dehagh2	0	8.51	136.2	168	96.9
Esfahan	0	8.51	139	155	84
Ghom2	0	8.19	130.9	218	61
Noorabad	0	8.57	137	157	86.4
Farashband	0	8.56	136.9	160	82
Ghazvin2	0	8.19	130.9	180	79.8

Since, HRSG and gas-pre heater are applied here, the exegetic efficiency of every gas turbine is increased by 31% in average and their exergy destruction is decreased by 84%. Finally, due to more heat recovery and lower exhaust temperature in the ST-GP-GPCS in comparison with simple GPCS, the total exergy saving for the second gas pipeline is 552 MW. In addition, output power is increased as the inlet flow rate rises.

Table 6. Exergy Flow Rate for ST-GP-GPCS (MW)

	Improved Cycle									
Station	E_1	E_2	E_3	E_4	E_5	E_6	E_7	E_8	E_9	E_{10}
Patave2	0	8.72	139.49	141.52	0.17	28.19	491.90	388.13	441.31	9.67
Dorahan2	0	8.52	139.32	141.35	0.19	30.50	525.75	404.78	466.82	12.53
Dehagh2	0	8.51	136.17	138.15	0.21	33.59	510.26	394.45	535.53	11.42
Esfahan	0	8.51	139.03	141.05	0.19	30.43	524.64	403.93	465.84	12.50
Ghom2	0	8.19	130.94	219.01	0.26	28.73	791.00	601.14	497.16	16.24
Noorabad	0	8.57	137.11	139.11	0.20	30.02	537.61	412.42	457.64	12.45
Farashband	0	8.56	136.89	138.88	0.21	29.78	512.93	394.91	467.03	12.44
Ghazvin2	0	8.19	130.94	132.85	0.26	28.93	745.92	593.07	475.78	12.69

Table 7. Evaluation of Exergy Efficiency (%) and Exergy Destruction (MW) of Gas Turbine

			Simple	Improved
Station	η_{exergy}	ELoss	η_{exergy}	ELoss
Patave2	8.57	77.9	12.50	9.67
Dorahan2	8.56	84	12.03	12.53
Dehagh2	11.84	96.9	15.01	11.42
Esfahan	10.88	84	13.97	12.50
Ghom2	9.90	61	12.72	16.24
Noorabad	11.85	86.4	14.92	12.45
Farashband	10.77	82	13.87	12.44
Ghazvin2	10.57	79.8	13.40	12.69

4. Conclusions

The effect of steam injection and gas pre-heating in the GPCSs on the second gas pipeline of Iran is assessed. The total net output power of the case study is 1.11 times the simple GPCS when gas pre-heating (TIT: 1200 °C) is applied, and the overall efficiency of every GPCS is increased by 11% in average.

By developing conventional stations with gas pre-heating and steam injection (TIT: 1200°C and SR: 7.5 kg/kg-fuel), the total net output power would be 1.3 times the simple GPCS, and the overall efficiency of every GPCS would enhance by 31% in average.

Based on the exergy analysis on the ST-GP-GPCSs, the exegetic efficiency of every gas turbine is increased by 31% in average and their exergy destruction is decreased by 84%. The amount of total exergy saving for the case study would be 552 MW.

Nomenclature

AC	air compressor
CC	combustion chamber
e	specific exergy (kJ/kg)
E	exergy (kJ)
GC	gas compressor
GPCS	gas pressure compression station
GP-GPCS	gas preheating-gas pressure compressing station
h	specific enthalpy (kJ)
H-E	heat exchanger
HRSG	heat recovery steam system
m	mass flow rate (kg/s)
P	Pressure (kPa)
P_0	ambient pressure (kPa)
rc	pressure ratio of compressor
rt	pressure ratio of turbine
R	specific gas constant (kJ/kgK)
s	specific entropy (kJ/kgK)
SR	steam ratio(kg/kg-fuel)
SI-GP-GPCS	steam injection-gas preheating-gas pressure compressing station
SFC	specific fuel consumption
T_0	ambient temperature (K)
TIT	turbine inlet temperature (°C)
x	volume percent of the component gas
W	work interaction (kJ)
η	efficiency

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