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Thermodynamic evaluation of gas compression station from the point of energy and exergy view with an approach to reduce energy consumptions and emissions: A case study

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Abstract – The objective of the present study is to evaluate the gas compression station installed in Marand city from the point of energy and exergy view with an approach to reduce energy consumptions and emissions. The use of exhaust gases thermal energy is investigated to produce steam and electrical power. This free extra electrical power can be used in electrical facilities of the gas turbine unit and the gas compression station can be independent from foreign electrical energy sources. Meanwhile replacement of existing gas turbine starting system and the effects of inlet air temperature, steam injection to the combustion chamber and the steam generator pinch point are studied on fuel consumption, net power production, nitrogen oxides emissions and irreversibilities in various loads. The results revealed that in the proposed gas compression station the produced electrical power from the generated steam in the minimum load and inlet air temperature of 288 K is more than the overall electrical energy consumptions. Steam injection with amount of 0.5 kg.s^{-1} to the combustion chamber increased the cycle efficiency from 34% to 37.5% and decreased the emission from 3.1 ppm to 0.45 ppm in full load condition.

Key words: Gas compression station / energy / exergy / emission

1 Introduction

In today's world due to industrial developments and population growth the demand for energy consumption is increasing and efforts to secure energy resources and deliver it to the consumers are of the main challenges. Currently, natural gas is one of the main energy providing sources that has significant contribution in energy supply which is needed for main industries and thermal energy consumption in domestic applications and the role of natural gas in the industrial economy and society is revealed [1].

Delivering in time and much-needed source of energy to the end consumer has paramount importance. Today, natural gas is transported through pipelines. Along the gas pipeline due to frictions and crossing altitudes the gas pressure inside decreases. This pressure drop prevents the gas to move forward along pipeline. To compensate for the pressure drop, gas compression stations along the gas pipelines are used. In gas compression station using centrifugal compressors the pressure of gas through pipeline increases. The driving force of these compressors is provided from the gas turbines which are coupled to them. The most important energy consumptions in gas compression stations are fuel consumption in gas turbines, electrical power consumption of the turbine equipments, electrical power consumption in air coolers to cool down the high-pressure compressed natural gas and the lubrication oil, natural gas consumption in expansion starter turbine and power consumption in the electric heater to increase the temperature of the turbine fuel.

Some of the major problems of the existing system are mentioned below:

(1) In the studied gas compression station the thermal efficiency of the gas turbines is approximately 30% and this means that about 70% of the fuel energy is excreted to the environment without any use as an exhaust heat losses.
(2) There from in gas turbines, natural gas is usually used as the fuel, so the cost of this fuel supply is high.
(3) The existing expansion starter turbine discharges large amount of natural gas into the atmosphere. Due to the high costs to explore, refine and deliver the

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Nomenclature

P	Pressure (bar)	pp	Pinch point (K)
T	Temperature (K)	h	Working fluid enthalpy (kj.kg^{-1})
$\dot{W}$	Power (kW)	LHV	Low heat value (kj.kg^{-1})
$\dot{m}$	Mass flow (kg.s^{-1})		
C_P	Specific heat at constant pressure	**Subscripts and abbreviations**	
S	Entropy (kj.kg^{-1}.K^{-1})	i	Isentropic
I	Irreversibility (kW)	LPT	Low pressure turbine
k_{f}	Forward reaction constant	HPT	High pressure turbine
k_{r}	Reverse reaction constant	comp	Compressor
γ	Specific heat ratio	GT	Gas turbine
$\dot{E}x_{\mathrm{D}}$	Destruction exergy (kW)	st	Steam turbine
$\dot{E}_x$	Working fluid exergy (kW)	HRSG	Heat recovery steam generator

natural gas to end users, this amount of natural gas discharging into environment causes huge losses in financial assets and energy resources and also environmental pollutions.

(4) The use of natural gas in industrial gas turbines due to fewer emissions than other fossil fuels is considered. Among the pollutants produced by combustion of natural gas in gas turbines, nitrogen oxides compounds are the most toxic and most destructive contaminants to the environment that are as the result of a chemical reaction between nitrogen and oxygen at high temperatures.

(5) Dependence of the existing system to electrical power consumption in air coolers, electrical heater and blower fans is one of the major weaknesses of the gas compression stations in a way that in the event of power failure, the unit will stop working.

One of the most effective ways to prevent heat losses in gas turbines is to use heat recovery steam generator system. In this system by the thermal energy of the turbine exhaust gases, steam is generated and is used in steam turbine to produce electrical power. This can be an excellent source of electrical energy supply in required gas turbine equipments and make the gas compression unit independent of electrical energy from an external source. In addition, the amount of steam produced can be used to increase turbine efficiency by steam injection into the combustion chamber. Some of the activities in the field of heat recovery steam generator in gas turbines are as follows.

Sue and Chuang [2] evaluated a combined cycle of gas turbine from the perspective of second law of thermodynamics and calculated the thermodynamic second law efficiency with good accuracy. He observed that the second law efficiency of the combined cycle for the part loads became less than full load and the pinch point enhancement reduced the efficiency of the cycle. Also cooling the entering air to the gas turbine and preheating the fuel increased its net power.

Yang et al. [3] investigated thermodynamically the influences of inlet air cooling using an absorption chiller system in a combined cycle of gas turbine over the efficiency and net power and estimated the resulting savings in power generation in a non-cooling cycle and in the proposed new cycle.

Lee et al. [4] investigated the effects of simultaneous injection of water and steam into the combustion chamber of a gas turbine in a combined cycle of gas turbine. The most important results of this study can be noted to increase in the efficiency and net power in a combined cycle with steam injection to the turbine combustor.

Basrawi et al. [5] investigated the effects of environmental temperature on the performance of combined cycle of gas turbine in areas with cold climates and achieved the following results:

(1) Electrical efficiency decreased with increasing ambient temperature.
(2) With increasing temperature the amount of heat recovered from the exhaust gases increased.
(3) The fraction of heat recovery to produced power, increased with increasing ambient temperature.

Carazas et al. [6] studied the capability of heat recovery system efficiency in a combined cycle power plant.

Sayyaadi and Mehrabipour [7] investigated the second law efficiency of a gas turbine cycle by using a heat exchanger in the turbine exhaust and reheating the intake air. Then they calculated the time of investment return after installing heat exchanger in three different ambient temperatures. Increasing the ambient temperature decreased the second law efficiency of thermodynamics and the time of investment return increased.

Livshits and Kribus [8] proposed a plan that uses solar energy and thermal energy of the exhaust gases, to produce steam. Then steam was injected into the combustion chamber. In this study it was shown that increasing the mass fraction of water vapour in the entering air to the combustion chamber increases the power output of the turbine. The maximum contribution of solar energy to provide the heat needed for steam generation is 50% and a maximum efficiency in this system is 37%. Ganjeh Kaviri et al. [9] studied the combined cycle power plant with HRSG dual pressure and firing system. The effects of HRSG inlet gas temperature on the steam cycle efficiency and CO_2 emission were investigated in this research. The results revealed that increasing HRSG inlet gas temperature led to increase in the thermal efficiency and exergy efficiency of the cycle until 650 °C and after that decreased. Also from the exergy analysis of HRSG, it was cleared that the HP-EV and 2st HP-SH have the most exergy destruction respectively.

Feng et al. [10] discussed about the dual pressure HRSG with three different layouts of Taihu Boiler with specified values of inlet temperature, composition of flue gas, mass flow rate and water/steam parameters as temperature, pressure, steam mass flow rate and heat efficiency of different heat exchangers layout of HRSG. Analysis was based on the laws of thermodynamics and energy balance equations for the heat exchangers. The results of the steam mass flow rate, heat efficiency were obtained for three heat exchangers layout of HRSGs and compared with each other. It was found that the optimization of heat exchangers layout of HRSGs has a great significance for waste heat recovery and energy conservation.

Although there are some papers about heat recovery systems in gas turbines in literature, but they have not investigated the use of this system in a gas compression station and there is not any specific research about the effects of inlet air temperature, steam injection and pinch point on the amount of total energy savings, plant efficiency, NO_x pollution and irreversibilities in real operating conditions of these plants. Also there is not specific research about the possibility of replacing the existing gas turbine starting system with other alternatives.

The aim of this work is to evaluate the gas compression station installed in Marand city from the point of energy and exergy view with an approach to reduce energy consumptions and emissions. In this station there are five gas turbines in parallel formation which supply the driving forces of the gas compressors. These gas turbines are the major consumers of energy in this plant. Due to the low efficiency of the gas turbines, large amounts of energy are wasted through exhaust gases. Furthermore, the starting system of the existing gas turbine is a kind of expansion turbine, with the natural gas working fluid which ends large amounts of natural gas into the atmosphere during turbine start up and adds the inefficiency of the existing system.

In this research the use of exhaust gases thermal energy is investigated to produce steam and electrical power. This free extra electrical power can be used in electrical facilities of the gas turbine unit and the gas compression station can be independent from foreign electrical energy sources. In addition the effects of inlet air temperature, steam injection to the combustion chamber and the pinch point are studied on the combined cycle performance parameters including fuel consumption, net power production, nitrogen oxides emission and irreversibilities for various gas turbine loads. Moreover, the possibility of replacing the existing gas turbine starter system is considered. For this purpose a comprehensive thermodynamic model for the total suggestive gas compression station with all equipments including gas turbine, gas compressor, air coolers and heat recovery steam generator has been developed. This model is capable of providing all the predictions of the gas compressor power consumption, gas turbine power production, gas turbine exhaust temperature, nitrogen oxides emissions, electric power production out of the exhaust gases thermal energy and the amount of heat transfer in all the existing equipments in different working conditions of the gas compressor and governing climate conditions. In addition, it is capable of investigating exergy destructions and second law efficiencies of various components of the system equipments and the factors affecting them.

2 Description of the existing gas compression station

The studied gas compression station has been formed out in four main parts consisting of turbines fuel temperature and pressure control unit, gas turbine and gas compressor, filters and air coolers. Figure 1 schematically shows the configuration of the existing station. Natural gas is received via two different points out of gas transport pipeline and enters in to the plant. At first, the fuel gas is being conditioned including filtration, pressure regulation and temperature adjustment. This gas is used in other components of the station such as expansion starter turbine, buildings and emergency power generator. In the next phase natural gas passes through the filter and then enters into the gas compressor. The compressed high pressure natural gas cools down in air coolers and finally is delivered to the pipeline.

3 Proposed gas compression station

As shown in Figure 1 in the gas compression station, hot exhaust gases of the gas turbine, with the temperature of about 480 °C in station's nominal load are sent into the atmosphere. On the other hand an expansion turbine is used to start up the gas turbine with natural gas working fluid at the temperature and pressure of the fuel gas. High pressure natural gas drives the expansion turbine and then it is vented to the atmosphere. At the moment the gas turbine is being self sustained, the starter turbine stops. In the existing starting system large amount of natural gas is lost every time the turbine is being started and this makes it inefficient and adds pollutions to the environmental as well. In this regard, the most important methods to improve of existing gas compression station and increase the efficiency of the gas turbine cycle are put forward.

1 – In the case of studied gas compression station, it is necessary to note that the applied expansion starter turbine is designed primarily on the bases of steam working fluid, but in the present system without regarding to the issue, the working fluid has been changed to natural gas. This is probably due to the abundant accessibility of natural gas in these plants. The main problems of this system are the loss of large amount of gas during the turbine start up and the

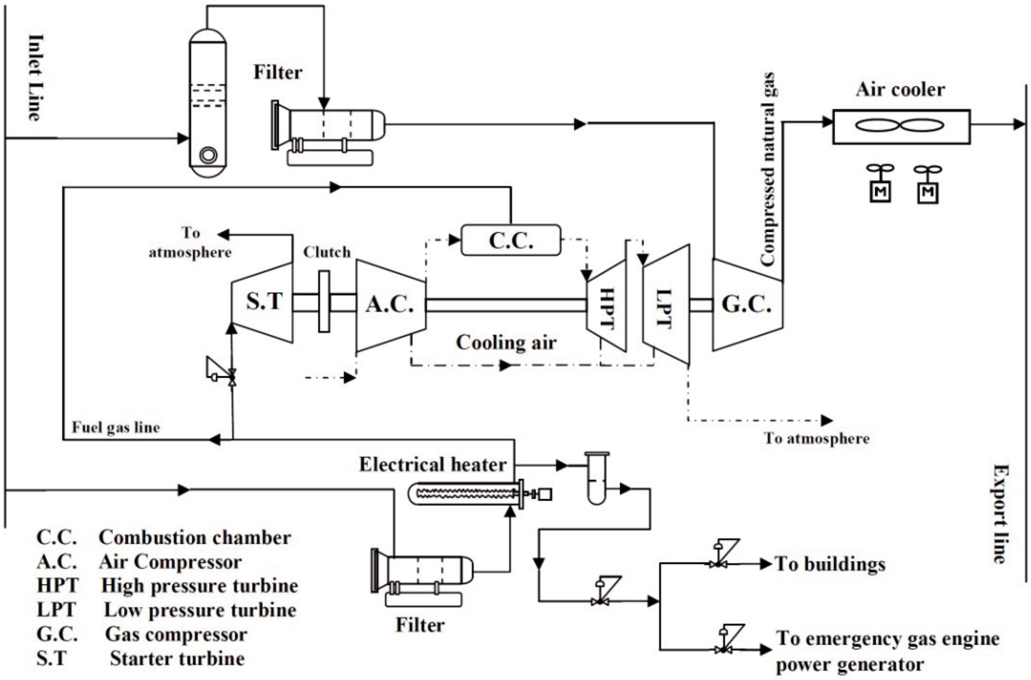

Fig. 1. General scheme of the studied gas compression station.

probability of gas leakage inside the turbine cabin and eventual explosions. In this work it is proposed to use an electrical starter to crank the air compressor of gas turbine. The electrical starter provides smooth acceleration from rest, up to a speed at which the fuel flows through the engine and provides sufficient power for the turbine to be self sustained.

2 – The operation of studied gas compression station is highly related to the external electrical energy sources. Table 1 has summarized the most important electrical energy consumptions in the present plant:

When the gas turbine is running in the full load much more energy is needed than it runs in the minimum load. It is obvious from the table, in the case of external electrical power failure the gas compression unit will stop running immediately. In the proposed system much more than this amount of energy can be produced from the exhaust gas thermal energy by employing a heat recovery steam generator. In this plan after the gas turbine reaches to the steady state operating condition the steam will be generated in the HRSG. This generated steam will produce power in the existing expansion steam turbine. This extra free power can be used in an electrical power generator to produce electrical energy. Thus the gas turbine unit will be independent from external electrical power sources. This generated electrical power is sufficient for the overall energy consumptions of the gas turbine auxiliary equipments.

3 – A mass fraction of steam produced in the heat recovery steam generator can be injected into the combustion chamber of the gas turbine in order to increase turbine net power and reduce emissions of nitrogen oxides. It is also quite practical and executable because steam injection pipelines into the combustion chamber have already been predicted thus there is no need for a change in the structure of a gas turbine combustor.

4 – On the other hand the discharged hot gases emerging from the heat recovery steam generator can be used to increase the temperature of turbine fuel gas hence the electrical power consumption in the electric heater can be removed. Application of these improvements is only possible when the gas turbine is running and according to the special conditions of the plant, its functioning can almost be considered uninterrupted. Therefore, the proposed system will have a good performance all year.

In Figure 2 the proposed gas compression station and the equipments configurations are shown.

4 Gas compression station modelling

To simulate the proposed gas compression station the integrated thermodynamic model has been developed. In the following, the first and second law of thermodynamic modelling of the system components are presented.

4.1 The first law of thermodynamics

Figure 3 shows the overview of existing gas turbine open cycle with T-S diagram. As indicated in the figure, the processes of compression in air compressor and expansions in both high pressure and low pressure turbines

Table 1. Energy consumptions for the existing gas turbine auxiliary equipments.

Equipment type	Unit	Energy consumption in full load	Energy consumption in minimum load
Gas coolers	kW	37 × 4	37 × 1
Ventilation fans	kW	26 × 2	11.1 × 2
Oil coolers	kW	8.8 × 2	2.2 × 2
Oil vapour separator	kW	5.5	5.5
Electrical heater	kW	27	12.7
Total	kW	250	81.8

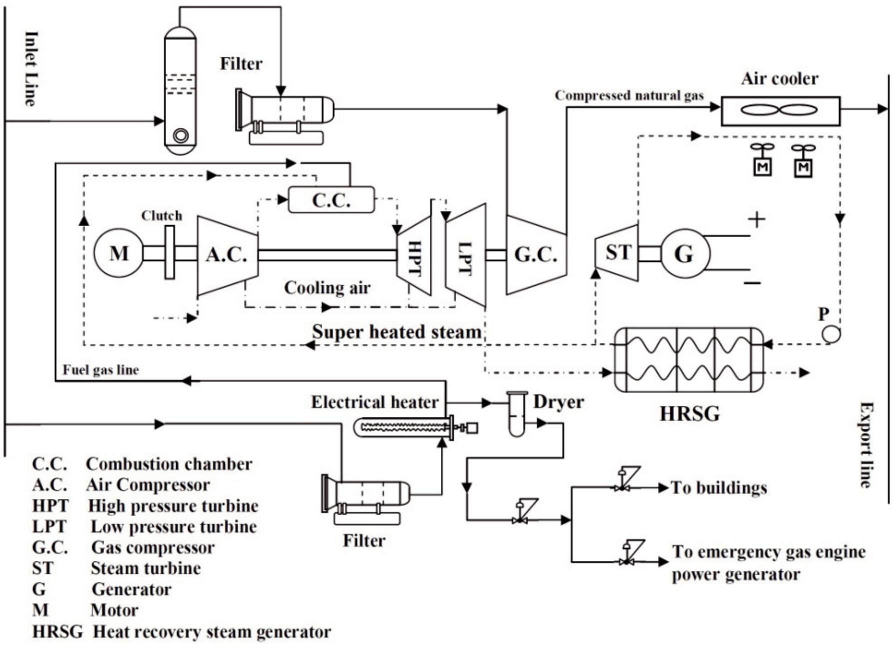

Fig. 2. The proposed system for the studied gas compression station.

Table 2. Studied gas turbine performance parameters.

T_1 (K)	288	P_2 (bar)	12.5
T_2 (K)	652	P_5 (bar)	1.01
T_3 (K)	1310	η_{mech}	33.3
T_4 (K)	994	NO_x emission (ppm)	15.5
T_5 (K)	775	Turbine net power (MW)	11.5
P_1 (bar)	1.013	Gas compressor max power (MW)	7.5

have deviancies from isentropic processes due to the irreversibilities.

Table 2 shows some performance parameters of the studied gas turbine taken out of the model results.

4.1.1 Air compressor

The air compressor of the studied gas turbine is an 11 stage axial compressor with the isentropic efficiency of about 80 percents. The driving force of the air compressor is supplied from the high pressure turbine. Maximum pressure ratio of the air compressor is 15.5:1 and the compression process is irreversible and adiabatic. The model equations for the air compressor are as follows [11]:

$$\dot{W}_{\text{comp,s}} = \int_{T_1}^{T_{2s}} \dot{m}_{\text{air}} C_{p,\text{a}}(T)\,\mathrm{d}T \tag{1}$$

$$\dot{W}_{\text{comp,a}} = \int_{T_1}^{T_{2a}} \dot{m}_{\text{air}} C_{p,a}(T)\,\mathrm{d}T \tag{2}$$

$$\eta_{\text{comp}} = \frac{\dot{W}_{\text{comp},s}}{\dot{W}_{\text{comp},a}} \tag{3}$$

$$\frac{P_2}{P_1} = \left[\frac{T_{2s}}{T_1}\right]^{\left[\frac{\gamma_1}{\gamma_1 - 1}\right]} \tag{4}$$

$$\dot{m}_1 = \dot{m}_{2a} \tag{5}$$

Since the air compressor discharge temperature is between (300–800) K, the heat capacity of air is obtained from relevant Equation (6) [12]:

$$C_{p,a} = 1.04841 + \left(\frac{3.8371}{10^4}\right) T + \left(\frac{9.4537}{10^7}\right) T^2 + \left(\frac{5.49031}{10^{10}}\right) \times T^3 + \left(\frac{7.9298}{10^{14}}\right) T^4 \tag{6}$$

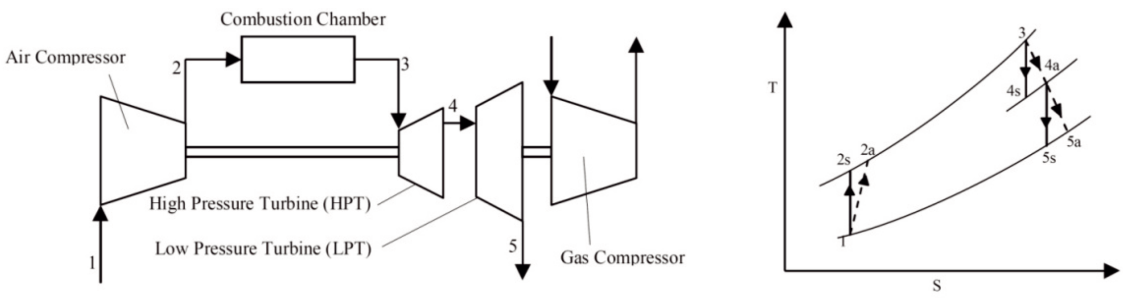

Fig. 3. Gas turbine open cycle scheme and T-S diagram.

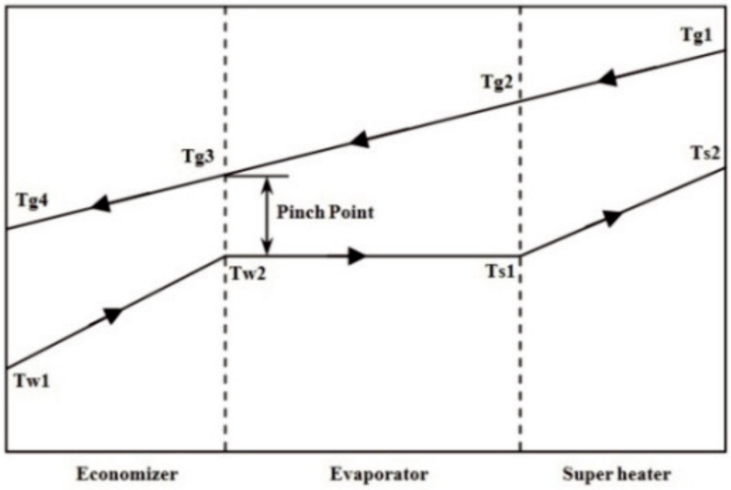

Fig. 4. Hot gas and steam temperature profiles in HRSG.

4.1.2 Combustion chamber

In the combustion chamber of the gas turbine, natural gas fuel with the lower heating value of about 50 (Mj.kg^{-1}) enters in to the air and the combustion process begins and the fuel energy releases. The combustion in the gas turbine is an adiabatic process with the pressure drop of about 0.2 percent of air compressor discharge pressure [13]. Inlet air to the combustor in unimproved and basic gas turbine cycle is a non-steam injected mixture. On the other hand, emissions of the nitrogen oxides are produced in the combustion chamber via the reactions between nitrogen and oxygen. The combustion equations have been carried out based in Equations (7)–(9):

$$\dot{m}_{\text{fuel}}(LHV)_{\text{fuel}} = (\dot{m}_{\text{g}})C_{p,\text{g}}(T_3)T_3 - \dot{m}_{\text{air}}C_{p,a}(T_{2,a})T_{2,a} \tag{7}$$

$$\dot{m}_{\text{g}} = \dot{m}_{\text{air}} + \dot{m}_{\text{fuel}} \tag{8}$$

According to the hot gases temperature range in the combustion chamber which is between (800–2200) K, so the heat capacity is obtained out of Equation (9) [12]:

$$C_{p,\text{g}} = 0.991615 + \left(\frac{6.99703}{10^5}\right)T + \left(\frac{2.7129}{10^7}\right) \times T^2 - \left(\frac{1.22442}{10^{10}}\right)T^3 \tag{9}$$

4.1.3 High and low pressure turbines

Hot gases of the combustion chamber initially enter into the high pressure turbine and then pass through the low pressure one. Both turbines isentropic efficiencies are about 82 percents. The driving forces of the air compressor and the gas compressor are provided via high pressure and low pressure gas turbines respectively. Both turbines' expansion processes are considered irreversible and adiabatic. The equations of the gas turbines modeling are as follows [14]:

$$\dot{W}_{\text{HPT},a} = \int_{T_3}^{T_{4a}} (\dot{m}_g)C_{p,\text{g}}(T)\text{d}T \tag{10}$$

$$\dot{W}_{\text{LPT},a} = \int_{T_{4a}}^{T_{5a}} (\dot{m}_g)C_{p,\text{g}}(T)\text{d}T \tag{11}$$

$$\eta_{\text{turb}} = \frac{\dot{W}_{\text{turb},a}}{\dot{W}_{\text{turb},s}} \tag{12}$$

$$\frac{p_{4a}}{p_3} = \left(\frac{T_{4s}}{T_3}\right)^{\left(\frac{\gamma_3}{\gamma_3 - 1}\right)} \tag{13}$$

$$\frac{p_{5a}}{p_4} = \left(\frac{T_{5s}}{T_4}\right)^{\left(\frac{\gamma_4}{\gamma_4 - 1}\right)} \tag{14}$$

$$p_3 = p_2 - 0.02p_2 \tag{15}$$

4.1.4 Heat recovery steam generator (HRSG)

Figure 4 shows the hot gases and produced steam temperature profiles in the heat recovery steam generator. The HRSG system consists of three sections including economizer, evaporator and super heater. In this system hot gases and steam, flow in opposite directions. Thus, hot gases cross over the super heater, evaporator and the economizer respectively.

The temperature and pressure ranges of steam generator are set on the bases of the expansion steam turbine performance parameters. According to the amount of steam production in the steam generator, a mass fraction of the steam is injected into the combustion chamber in the proper pressure. The combustion chamber pressure varies according to the operating condition of the gas turbine. So the steam pressure is adjusted in to the appropriate pressure. Since the steam pressure is available then the saturation temperature of the desired pressure is achievable. According to the pinch point, the hot gas temperature over the evaporator outlet is obtained. The difference between the economizer inlet and the evaporator outlet temperature is also considered negligible [15]. Employing the energy balance equation between hot gases and steam in the heat recovery steam generator, from evaporator outlet until the super heater exit, the mass flow rate of the produced steam is obtained. In order to calculate the hot gases temperature over the super heater inlet, Equation (18) is used. In the first step the heat capacity of point $g2$ is calculated by the temperature of $g1$. In the second step the new $g2$ temperature is obtained according to the first step's $g2$ temperature. By repeating this corrector-predictor operation, hot gas temperature over the super heater inlet will converge to the true answer. To obtain the temperature of the hot gases over the economizer inlet the same calculating method is used via Equation (19) [15].

$$T_{3g} = T_{w2} + PP \tag{16}$$

$$\dot{m}_s = \frac{\dot{m}_g \left(C_{p,g1}T_{g1} - C_{p,g3}T_{g3}\right)(1-hl)}{(h_{s2} - h_{w2})} \tag{17}$$

$$T_{g2} = \frac{C_{p,g1}T_{g1}}{C_{p,g2}} - \frac{\dot{m}_s \left(h_{2s} - h_{w2}\right)}{\dot{m}_g C_{p,g2}\left(1-hl\right)} \tag{18}$$

$$T_{g4} = \frac{C_{p,g3}T_{g3}}{C_{p,g4}} - \frac{\dot{m}_s \left(h_{w2} - h_{w1}\right)}{\dot{m}_g C_{p,g4}\left(1-hl\right)} \tag{19}$$

4.1.5 Steam turbine

The applied steam turbine is the same starting expansion turbine mounted on the air compressor of the gas turbine. When the gas turbine is self sustained the existing automatic clutch separates the air compressor and the starter turbine and then the starter turbine stops. The existing startup turbine system is essentially a small-scale steam turbine. In the present study it is proposed to couple the starter turbine with the generator to produce electrical power after the gas turbine reaches to the steady state condition. Steam turbine specifications are presented in Table 3.

Table 3. Steam turbine specifications [16].

Parameter	Unit	Value
Maximum power generation	kW	746
Maximum inlet pressure	bar	46
Maximum inlet temperature	K	713
Maximum outlet pressure	bar	11
Maximum speed	rpm	4300
Inlet diameter	mm	100
Exhaust diameter	mm	200

The power production in the steam turbine and the water pump power consumption is calculated using the following equations:

$$\dot{W}_{\rm st} = \dot{m}_{\rm st}(h_{s2} - h_{ex}) \tag{20}$$

$$\dot{W}_{\rm pump} = \dot{m}_{\rm st}(h_{\rm pump,in} - h_{w1}) \tag{21}$$

The gas turbine and combined cycle efficiencies are obtained according to the relations (22) and (23) [17]:

$$\eta_{GT} = \frac{\dot{W}_{{\rm LPT},a}}{\dot{m}_{\rm fuel}{\rm LHV}} \tag{22}$$

$$\eta_{\text{combined cycle}} = \frac{\dot{W}_{{\rm LPT},a} + \dot{W}_{\rm ST} - \dot{W}_{\rm PUMP}}{\dot{m}_{\rm fuel}(LHV)_{\rm fuel}} \tag{23}$$

4.2 Second law of thermodynamics

Exergy analysis of thermodynamic systems helps to improve the ways of energy use in them. Accordingly exergy studies in many systems especially in combined cycle power generation are expanding rapidly. In general exergy consists of four main components of chemical, physical, kinetic and potential exergy. Since the little altitude and speed changes, potential and kinetic exergies are neglected [17]. The physical exergy is defined based on the maximum obtainable work of the system in contact with the environment. The chemical exergy is the chemical potential between the existing and the equilibrium compounds. Using the second law of thermodynamics the exergy balance equation is written as relations (24)–(27):

$$\dot{E}x_Q + \sum_i \dot{m}_{\rm in} ex_{\rm in} = \sum_e \dot{m}_{\rm out} ex_{\rm out} + \dot{E}x_W + \dot{E}x_D \tag{24}$$

$$\dot{E}x_Q = \left(1 - \frac{T_0}{T_{\rm i}}\right)\dot{Q}_{\rm i} \tag{25}$$

$$\dot{E}x_W = \dot{W} \tag{26}$$

$${\rm ex}_{\rm ph} = (h - h_0) - T_0\,(s - s_0) \tag{27}$$

In the present work the exergy of heat transfer is negligible because all processes are considered to be adiabatic. In the following the exergy balance and the second law efficiency equations are written for air compressor, combustion chamber, gas turbine and heat recovery steam generator respectively [18].

Air compressors exergy:

$$\dot{m}_{\text{air}} ex_1 + \dot{W}_{\text{comp},a} = \dot{m}_{\text{air}} ex_{2a} + I_{\text{comp}} \tag{28}$$

$$\eta_{\text{II,comp}} = 1 - \frac{I_{\text{comp}}}{\dot{W}_{\text{comp},a}} \tag{29}$$

Combustor exergy:

$$\dot{m}_{\text{air}} x_{2a} + \dot{m}_{\text{fuel}} ex_{\text{fuel}} = \dot{m}_g ex_3 + I_{\text{comb}} \tag{30}$$

$$ex_{\text{fuel}} = ex_{\text{ph},f} + ex_{\text{ch},f} \tag{31}$$

$$ex_{\text{ch},f} = \zeta(\text{LHV})_{\text{fuel}} \tag{32}$$

For the fuels with C_xH_y compounds [19]:

$$\zeta = 1.22 + 0.0169\frac{y}{x} - \frac{0.0698}{x} \tag{33}$$

Gas turbine exergy:

$$\dot{m}_g ex_3 = \dot{m}_g ex_{5a} + \dot{W}_{\text{LPT},a} + I_{\text{turb}} \tag{34}$$

$$\eta_{\text{II,turb}} = \frac{\dot{W}_{\text{turb}}}{\dot{W}_{\text{turb}} + I_{\text{turb}}} \tag{35}$$

Steam generator exergy:

$$\begin{aligned} I_{\text{HRSG}} &= \dot{m}_{\text{gas}}[(h_{5a} - T_0 s_{5a}) - (h_{g4} - T_0 s_{g4})] \\ &\quad + \dot{m}_{\text{st}}[(h_{w1} - T_0 s_{w1}) - (h_{s2} - T_0 s_{s2})] \end{aligned} \tag{36}$$

$$\eta_{\text{II,HRSG}} = \frac{\dot{m}_{\text{st}}[(h_{w1} - T_0 s_{w1}) - (h_{s2} - T_0 s_{s2})]}{\dot{m}_{\text{gas}}[(h_{5a} - T_0 s_{5a}) - (h_{g4} - T_0 s_{g4})]} \tag{37}$$

4.3 Nitrogen oxides emission in combustion chamber

In the present work the mechanism of thermal NO_x is considered as the production process of nitrogen oxides in the combustion chamber of the gas turbine. Nitrogen oxide production in thermal NO_x mechanism is activated in high temperatures by chemical reactions called developed Zeldovich mechanism [20].

$$O + N_2 \underset{k_{r1}}{\overset{k_{f1}}{\longleftrightarrow}} N + NO \tag{38}$$

$$N + O_2 \underset{k_{r2}}{\overset{k_{f2}}{\longleftrightarrow}} O + NO \tag{39}$$

$$N + OH \underset{k3}{\overset{k_{f3}}{\longleftrightarrow}} H + NO \tag{40}$$

In order to model the NO_x emissions, the combustion chamber temperature is calculated by the first law of thermodynamic and energy balance equations. The amounts of oxygen and nitrogen molecules in this set of equations are obtained from the turbine inlet air mass flow rate. O and OH radical concentrations are calculated from empirical relations (41) and (42) [21,22].

$$[O] = 3.97 \times 10^5 \times T_3^{-0.5} [O_2]^{0.5} \exp\left(\frac{-31\,090}{T_3}\right) \tag{41}$$

$$\begin{aligned} [OH] &= 2.129 \times 10^2 \times T_3^{-0.57} [h_2O]^{0.5} [O]^{0.5} \\ &\quad \times \exp\left(\frac{-4595}{T_3}\right) \end{aligned} \tag{42}$$

Then the numerical solution method is applied to solve the non-linear equation of nitrogen oxide concentration:

$$[NO] = 2k_{f1}[O][N_2] \left[\frac{1 - k_{r1}k_{f2}\frac{[NO]^2}{k_{f1}[N_2]k_{f2}[[O]_2]}}{1 + k_{r1}\frac{[NO]}{k_{f2}[O]_2 + k_{f3}[Oh]}} \right] \tag{43}$$

It is clear from the equations that, this mechanism is highly dependent on the temperature of the combustion products and the higher amount of nitrogen oxides will produce by increasing in the combustion temperature [23].

5 Results and discussion

The results of this study are summarized in two main parts. In the first section the results of the first law of thermodynamics including model validation, inlet air temperature and steam injection influences on the combined cycle performance parameters are discussed. In the second part the effects of the above parameters are investigated on the irreversibility of gas turbine and heat recovery steam generator in various gas turbine loads.

5.1 First law of thermodynamic results

5.1.1 The model validation

After preparing the model and setting up the related equations for the basic gas compression station, the model results are compared with the gas turbine performance maps. Figure 5 compares the calculated turbine power versus fuel consumptions for different ambient temperatures. As outlined in the figures there is a good agreement between model results and the results obtained from gas turbine performance curves.

Figure 6 indicates the power production of the steam turbine versus gas turbine loads in different inlet air temperatures (T_1). By increasing the inlet air temperature and the gas turbine load the steam power production increases due to the increase in exhaust gases temperature. In the studied gas compression station the gas turbine load never falls below 50%. As it is shown in this figure the minimum steam turbine power production is 186 kW which is obtained in the lowest inlet air temperature of 250 K. This amount of power production is much more than the auxiliary equipments of the gas turbine electrical power consumptions in minimum load (Tab. 1). It means that the proposed system is reliable for the whole running conditions of the gas turbine and is able to make independent the gas compression unit from external electrical sources. In the followings the effects of three important parameters including inlet air temperature, steam injection and pinch point are studied on the gas turbine exhaust temperature, NO_x pollutions, steam turbine power production, combined cycle efficiency, fuel consumption and GT and HRSG irreversibilities and second law efficiencies.

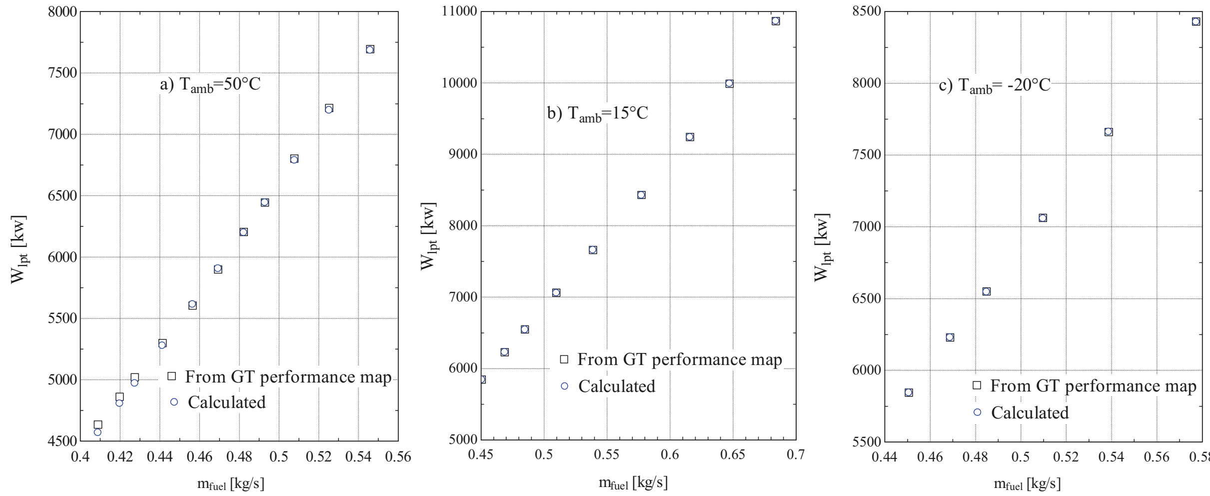

Fig. 5. Model results and GT performance map net power versus fuel consumption in different ambient temperatures.

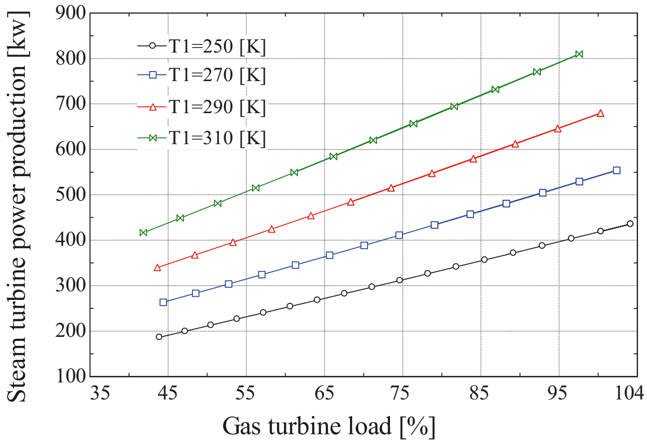

Fig. 6. Steam turbine power production versus GT loads in various inlet air temperatures.

5.1.2 Effects of inlet air temperature

Figure 7 indicates the effects of gas turbine inlet air temperature on the turbine exhaust temperature, steam turbine power generation and emissions of nitrogen oxides for turbine maximum design load and for the gas compressor maximum absorbed power respectively.

By increasing the inlet air temperature the air density and the mass flow rate of the gas turbine decrease. Since the power produced in the gas turbine is dependent on the total mass flow rate, thus reducing the mass flow will reduce power production. To compensate the turbine net power production due to inlet air temperature rise, more fuel consumption is required. Increasing the fuel consumption in the fixed turbine load increases the temperature of the combustion products. Increasing exhaust gas temperature, rises the super heated steam temperature in the heat recovery steam generator, accordingly the power production in the steam turbine enhances. On the other hand, with increasing the temperature of the combustion products in accordance to the Zeldovich mechanism the amounts of nitrogen oxide emissions increase. It is important to note that the maximum permissible temperature in the gas turbine exhaust gases is 800 K. Exhaust temperature control leads to control the combustion products temperature with the amount of fuel consumption in order to protect the turbine blades from burning. Thus, by comparing Figures 7a–7c it is observed that the turbine control system in the maximum design load of gas turbine will limit the fuel consumption in the inlet air temperature of 298 K. In this case the maximum emission in the turbine is 15 ppm and the efficiency of the combined cycle for the considered temperature range is 33% in full load and 29% in part load. It is clear that the produced power in steam turbine in the combined cycle has compensated the efficiency reduction of gas turbine cycle due to inlet air temperature rise.

5.1.3 Effects of steam injection into the combustion chamber

As it was noted in Section 3, the steam injection into the combustion chamber of the gas turbine is one of the other developing methods in gas compression station. Due to the availability of related pipelines of steam injection to the gas turbine combustor, performing this task without overall changes in the gas turbine structure is easily possible. Figure 8 shows the effects of steam injection into the gas turbine combustion chamber, on fuel consumptions, exhaust gas temperatures, steam turbine power production, emissions of nitrogen oxides and the combined cycle efficiency in the inlet air temperature of 288 K and gas turbine various loads.

As Figure 8a shows, by increasing the amount of steam injection into the combustion chamber in constant load the amount of turbine fuel consumption is reduced. Increasing the flow rate of the steam injection into the combustion chamber, increases the gas turbine net power. So to stabilize the gas turbine net power production, the amount of fuel consumption should be decreased

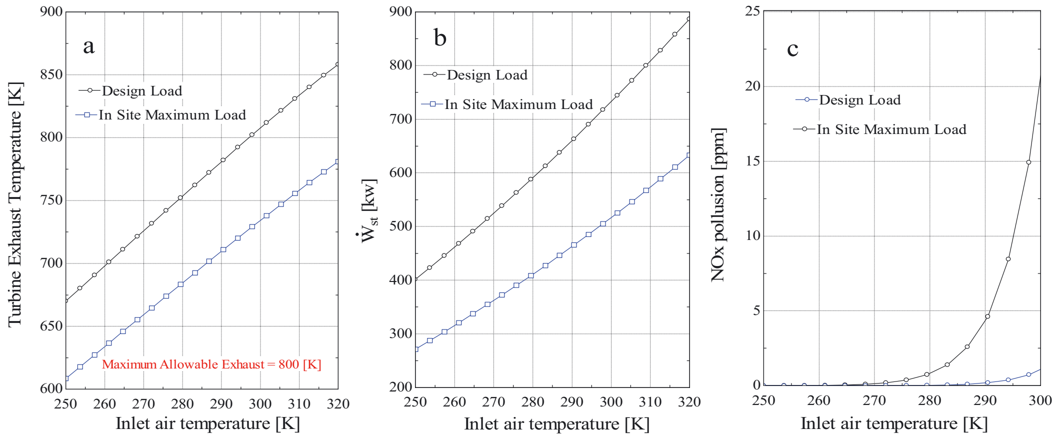

Fig. 7. Effects of inlet air temperature on (a) turbine exhaust temperature (b) steam turbine power (c) NO_x pollution.

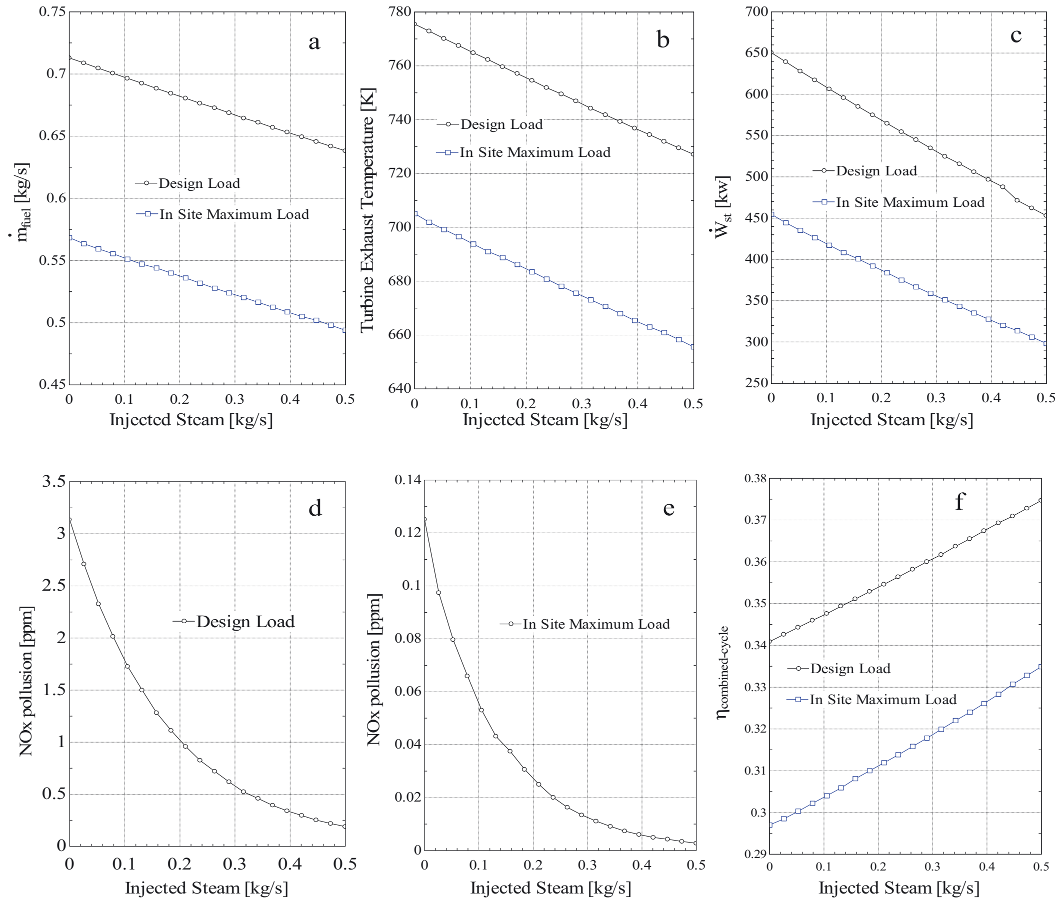

Fig. 8. Effects of steam injection to the gas turbine combustion chamber on (a) fuel consumptions (b) exhaust gas temperatures (c) steam turbine power production (d) NO_x for design load (e) NO_x for site maximum load (f) combined cycle efficiency.

(Fig. 8b). Furthermore the turbine exhaust temperature is decreased due to reduction in fuel consumption and consequently the super heated steam temperature is decreased and power production in steam turbine is lessened (Fig. 8c). On the other hand increasing steam injection to the combustion chamber reduces the mass flow rate of steam turbine. Despite the reduction in steam turbine power production it is still capable of making the gas turbine unit independent from external electrical power source because the total power consumption of the unit is less than the steam turbine produced power. Moreover, the reduction in the combustion products temperature leads to the reduction of nitrogen oxides pollution in the Zeldovich mechanism greatly and this is due to the high sensitivity of NO_x production mechanism to the temperature of the combustion products (Figs. 8d and 8e).

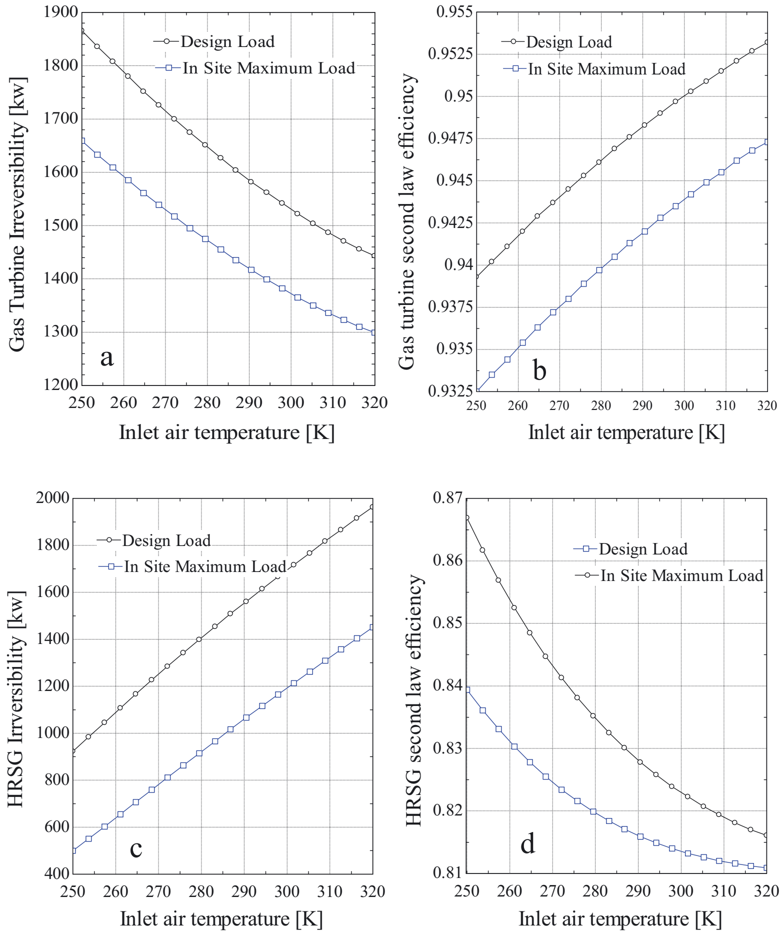

Fig. 9. Effects of inlet air temperature on (a) gas turbine irreversibility (b) gas turbine second law efficiency (c) HRSG irreversibility (d) HRSG second law efficiency.

Figure 8f reveals the effects of steam injection in combined cycle efficiency in various loads. It is observed that increasing the steam injection into the combustion chamber increases combined cycle efficiency according to Equation (23). What can be deduced from this figure is that the effect of fuel consumption reduction in the gas turbine is dominant to steam turbine power production decrease so the efficiency of the combined cycle is increased.

5.2 Second law of thermodynamics results

5.2.1 Effects of inlet air temperature on irreversibilities

Figure 9 shows the irreversibility and second law efficiency of the gas turbine and the steam generator versus inlet air temperature in full and partial loads. In both cases according to Equation (34) the irreversibility of gas turbine is reduced with increasing the inlet air temperature (Fig. 9a). Irreversibility decline means the potency of power production in the system increases and consequently the system efficiency reduces (Fig. 9b). In the heat recovery steam generator the irreversibility increases with increasing the inlet air temperature (Fig. 9c). When the inlet air temperature of the gas turbine increases the temperature of combustion products and exhaust gases increase. By increasing the temperature of the exhaust gases the mass flow rate of steam produced in the steam generator increases and thus according to Equation (36) the irreversibility of steam generator increases. Increasing the irreversibility of the heat recovery steam generator lessens the second law efficiency according to Equation (37) (Fig. 9d).

5.2.2 Effects of steam injection on irreversibilities

In Figure 10 the effects of steam injection into the turbine combustor have been investigated. In this study the inlet air temperature according to the previous section is considered 288 K. Steam injection into the combustion chamber increases the irreversibility and decreases the second law efficiency of the gas turbine. This is completely vice versa in HRSG, so that the steam injection decreased the irreversibility and increased the second law efficiency in HRSG system. In the gas turbine by increasing the steam injection in to the combustor, the entering and outgoing exergies of the turbine decrease due to the

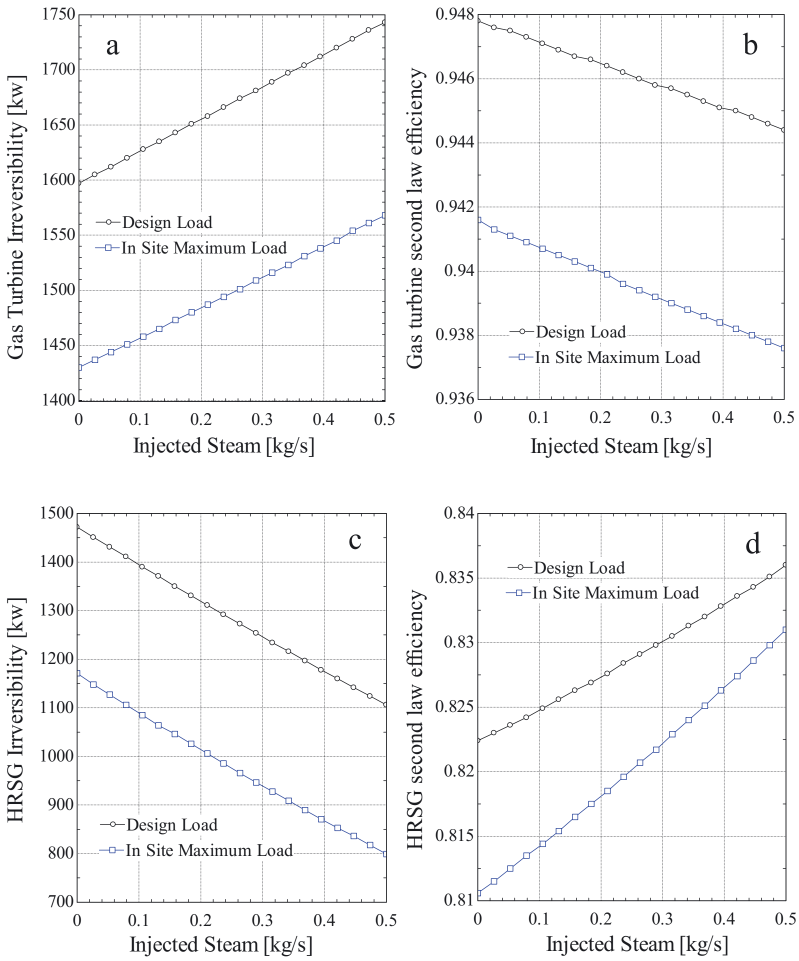

Fig. 10. Effects of steam injection to the combustion chamber on (a) gas turbine irreversibility (b) gas turbine second law efficiency (c) HRSG irreversibility (d) HRSG second law efficiency.

temperature reduction. But the difference between these two quantities which equals to the irreversibility of gas turbine increases according to Equation (34) in the gas turbine's constant load (Fig. 10a). Since the system irreversibility increases and the power production capability decreases, the second law efficiency reduces (Fig. 10b). In contrast to the gas turbine, in steam generator with increasing the steam injection, irreversibility is decreased (Fig. 10c). Steam injection to the combustion chamber decreases the combustion product's and turbine exhaust gases temperature and this causes a reduction in steam generated in HRSG, so the steam turbine mass flow rate decreases. According to Equation (36) the reduction of hot gases temperature and steam mass flow rate in the steam generator are the main factors for the irreversibility decline. The second law efficiency the same as previous part increases by increasing the system performance (Fig. 10d).

5.2.3 Effects of the pinch point on the irreversibilities

The effects of the pinch point on the irreversibility of the heat recovery steam generator have been studied. Figure 11 illustrates the irreversibility of steam generator versus the pinch point in the gas turbine constant load. Increasing HRSG pinch point leads to increase the irreversibility and after a certain point it starts to decrease. Figure 11a shows that without steam injection when the turbine inlet air temperature decreased from 298 K to 288 K in addition to irreversibility decline, the pinch point at which the irreversibility has its maximum value decreased from 28 °C to 24.5 °C. Figure 11b shows the effects of steam injection to the combustion chamber on the steam generator irreversibility in the gas turbine constant inlet air temperature. With increasing in the steam injection into the combustion chamber, the irreversibility increases inconspicuously and the pinch point related to the maximum irreversibility reduced from 37.5 °C to 29 °C. Decreasing the pinch point is due to the intensification of the energy absorption in the steam generator. Thus, if the mass flow rate of steam generator stays constant, decreasing the pinch point will cause an increase in the super heated steam temperature. This action increases the power production of the steam turbine and reduces the steam generator outgoing hot gases temperature. Reducing hot gases temperature declines the exergy

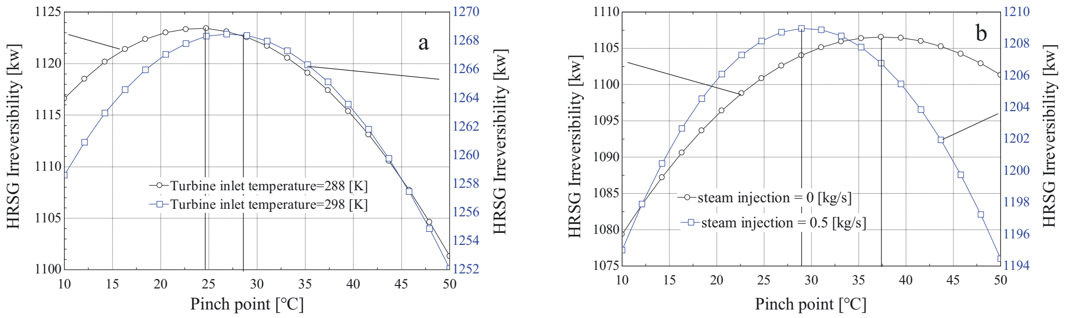

Fig. 11. HRSG irreversibility versus pinch point in gas turbine constant load for (a) various inlet air temperature (b) various steam injection.

destruction as well as the system irreversibility, and it increases the second law efficiency.

6 Conclusion

In the present work a comprehensive thermodynamic model was developed for the gas compression station. The main features of this model are to calculate the gas compressor power consumption, the gas turbine net power production, combustion chamber temperature, the turbine exhaust temperature, heat recovery steam generator analysis and nitrogen oxides emissions in the gas turbine combined cycle. Moreover the gas turbine and the HRSG irreversibilities in actual operating conditions and under different loads were studied. Comparison of model initial results with the gas turbine performance curves indicated a good agreement between them and confirmed the good performance of the model.

In the following the energy and exergy analysis of combined cycle were obtained and the effects of inlet air temperature, steam injection to the combustion chamber and the pinch point were investigated. Simultaneously controlling nitrogen oxides emissions in this method was offered.

– The most important results of inlet air temperature on the gas turbine with heat recovery steam generator are as follows:

1 – The results indicate that in the maximum design load of the gas turbine due to the exhaust temperature limitations from the turbine controlling system, the turbine operation at inlet air temperatures above 298 K would not be possible. Because in this temperature the maximum allowable exhaust temperature is reached and increasing the inlet air temperature from 298 K to higher degrees may damage the turbine blades. But in the maximum load of the gas compressor which is about 65 percents of the gas turbine design load, there will be no limitations in the studied temperature range. Increasing the inlet air temperature increased the exhaust gas temperature. So the amount of steam production in the steam generator and steam turbine power production increased.

2 – Increasing the inlet air temperature led to increase in the temperature of the combustion products and due to the Zeldovich mechanism sensitivity to the temperature, the amount of produced nitrogen oxides emissions in gas turbine increased.

3 – By increasing the inlet air temperature, the amount of irreversibility of gas turbine in constant load decreased and the second law efficiency due to the irreversibility decline increased.

4 – By increasing the inlet air temperature, the mass flow rate of steam in the steam generator increased and this led to increase in steam generator irreversibility. Increasing the steam generator irreversibility decreased the second law efficiency of thermodynamics.

– The most important results of steam injection into the gas turbine combustion chamber are defined as follows:

5 – By increasing the amount of steam injection into the combustion chamber the mass flow rate and the net power production in the gas turbine increased. Thus, to stabilize the gas turbine net power production, the fuel consumption reduced.

6 – Increasing steam injection into the combustion chamber at turbine constant load led to decrease in combustion products temperature and NO_x emissions due to decrease in fuel consumptions.

7 – Increasing the injection rate of steam into the gas turbine combustor decreased the mass flow rate of steam turbine and its power production reduced, but according to the maximum electrical power consumption in each turbo compressor unit, this minimum amount of steam turbine power production was enough to make independent each unit from external energy sources.

8 – Steam injection to the combustion chamber reduced the combustion product's temperature and due to the Zeldovich mechanism sensitivity to the temperature, the amount of nitrogen oxide emissions in a gas turbine decreased.

9 – Despite the reducing of power produced in the steam turbine through steam injection to the combustion chamber, the gas turbine combined cycle efficiency increased. This is due to the reduction in gas turbine fuel consumption in constant load.

– The most important results related to the pinch point are given as follows:

10 – By increasing the pinch point, the amount of steam generator exergy destruction raised and then reduced.

11 – Reducing the inlet air temperature without steam injection reduced the irreversibility of steam generator.
12 – Steam injection and gas turbine inlet air temperature reduction, caused the irreversibility of the steam generator to reach its maximum value in lower pinch points.

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