ISME

A. Arabkoohsar*
Ph.D student

M. Farzaneh-Gord†
professor

R.N.N. Koury‡
professor

Proposing a Novel Configuration for CGSs Aimed at Reducing Energy Consumption and Exergy Destruction

A CGS is of the most important parts in natural gas transmission pipelines in which the high inlet natural gas (NG) pressure is reduced to much lower value by employing a throttling valve. This pressure drop causes hydrate forming, preventing stable NG flow through the pipeline. To prevent hydrate forming, NG is usually preheated by heaters which burns remarkable amount of NG. In this work, a novel configuration for CGSs is proposed in which solar heat is utilized to decrease fuel consumption and a turbo expander is employed to utilize the available exergy in the NG stream to produce power. Finally, an economic analysis is done on the proposed configuration based on NPV method in order to calculate optimum cost of capital and the configuration efficiency.

Keywords: Natural gas pressure drop station, Turbo-expander, Solar heater system, Net present value

1 Introduction

NG which is extracted and refined in refineries is mostly transmitted to consuming points by transmission pipelines. Considering the long distance which NG has to pass from the refinery to the consuming points, it must overcome friction losses along the path. Consequently, NG is pumped into the pipeline at much higher pressure than consumption values and secondly numerous attenuator and booster pressure stations worked out along this way in order to pressure adaption. CGSs which are usually located close to the cities or other consumption points such as big factories are mainly the most important kinds of these stations. Through a CGS, NG pressure has to be dropped from nearly 1000psi to almost 250psi. There are also two further pressure reduction steps. The first one is happened at TBS stations in which NG pressure is decreased from 250psi to 60psi. The last step of pressure reduction is carried out by a small regulator at consumption points which converts NG pressure from 60psi to 1/4psi [1].

*Corresponding author, Ph.D student, Department of Mechanical Engineering, Minoodasht Branch, Islamic Azad University, Minoodasht, Iran, mani.koohsar@yahoo.com
†Professor, Department of Mechanical Engineering, Shahrood University of Technology, Shahrood, Iran, mahmood.farzaneh@yahoo.co.uk
‡Professor, Department of Mechanical Engineering, Federal University of Minas Gerais (UFMG), BH, Brazil

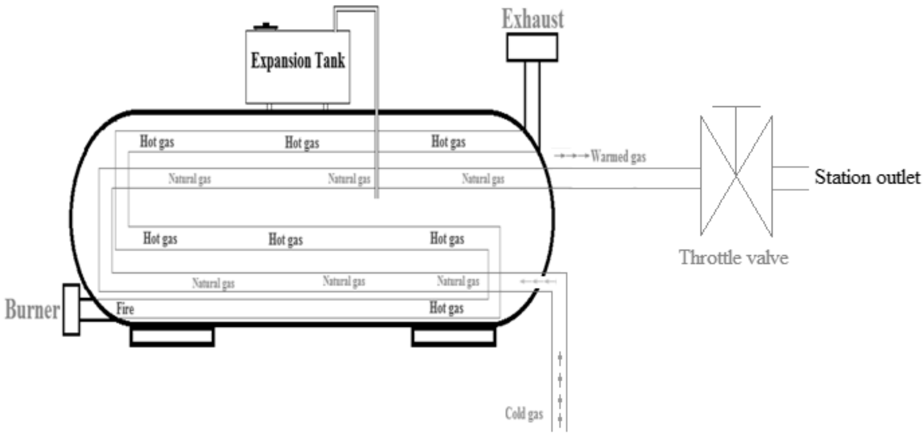

Figure 1 Schematic diagram of a natural gas pressure reduction station

At CGS, due to both considerable mass flow rate and pressure of the passing NG, its exergy undoubtedly has a significant value and this is exactly the thing could make possible various optimizations. On the other hand, pressure drop in NG leads to temperature drop due to NG positive Joule-Thompson coefficient. It should be noted that, there is a limitation for the minimum allowable temperature of each NG named hydration point. Hydration point is a temperature in which suspended water droplets in NG start to freeze and then the transmission pipeline may be blocked. Hydration point is depended on NG compositions. In order to prevent gas hydrate forming, it should be preheated before the pressure reduction in CGSs [2]. Linear heaters are mostly used for doing this task in CGSs. The heater burns a remarkable portion of the flowing NG as its main fuel. Figure (1) illustrates a detailed schematic diagram of a CGS.

In this study, two suggestions are simultaneously presented for optimizing NG pressure reduction station patterns. The first one is about adding a solar heater set aimed to decrease heating duty of the active line heater and subsequently decrease fuel consumption in the station. In fact, a lot of studies have been done about employing renewable and sustainable energies for either domestic or industrial applications all over the world. Three items could be pointed out as the main reasons of this notable attendance of the world to renewable energies. Firstly, huge reserves of fossil fuels which have been stacked under the Earth's layers after millions of years are being discharged due to the indiscriminate use of human. Secondly, because of many difficulties on the way of exploitation and refinement of fossil fuels, these fuels are mainly too expensive. Adverse effects of combustion of these fuels on the environmental so could be mentioned as the third reason. Therefore, renewable energies are gradually considered as the main substitute of fossil fuels. Among all kind of sustainable energies, solar energy is the most important one for energy consumers due to being free and endless. Although, there are many different layouts and designs for providing and employing solar heat, in all of them the most important part of the system is solar collector. The device receives and absorbs solar radiation and transfers it to an operating fluid which usually is water. Solar energy usage for industrial applications is still not widespread enough especially for those applications which need low temperature heat [3].

Norton has presented a thorough research about the most common solar energy utilization for providing required heat for various applications [4].The research consists of a comprehensive background about agricultural and industrial solar energy applications and some practical examples as well. Spate et al. have introduced a solar heater system taking advantage of non-concentrator collectors suitable for either big bakeries or kitchens in developing countries [5]. Benz et al. introduced two different solar systems for operating in a dairy factory in Germany, applied assessments demonstrates that both introduced systems could be extensible for domestic as well as heating applications [6]. Through another paper, he did a thorough study on non-concentrator collectors' applications for using in food production industry in Germany [7]. In another research, Li and Young have analyzed Hong Kong's potential for providing solar heat [8]. Low temperature solar heat producer systems may not be appropriate enough for most big industries due to enough access to surplus heat in these places but for other places which this surplus heat is not available utilization of solar energy in order to generate required heat seems a very sensible idea [9]. As explained before, the most important advantage of these places is that low temperature and low pressure heat is needed, therefore non-concentrator collectors which are simpler and so cheaper could be used.

The second suggestion in this work along with applying solar heating system, is replacing the throttling valve by a turbo expander. This makes the possibility of capturing the high available exergy in the flowing NG. A lot of studies have been done in order to evaluate the exergy amount of NG pipelines. Bisio introduced a special system consists of a mechanical air compressor in order to utilize this exergy [10]. Turbo expanders are also equipments which are used for generating electricity from this exergy. The device usually is used for providing NG flow in very low temperatures in ethane exploitation and liquid NG refineries. Since the turbo expander net work is gotten from expanding high pressure NG, so the expansion procedure could be considered as an isentropic process through a reasonable approximation. There are many various turbo expander models which operate in range of 75-130kW and isentropic efficiency 84-86% [11].Greeff et al. have done a thorough study on feasibility of using turbo expanders in exothermic chemical mixing procedures which leaded to considerable energy providence in the processes [12]. Hinderink et al. introduced a method to calculate the existing exergy in multi-component liquids and two-phase flows [13]. This method divides a stream exergy into three different terms and exergy variations due commixture are separately calculated from their chemical and physical exergy. Pozivil studied feasibility of employing turbo expanders in NG pressure reduction points using HYSIS software and assessed effects of these turbines isentropic efficiency on temperature and pressure drop of the NG as well as electricity generation [14]. Farzaneh-Gord and Magrebi analyzed exergy destruction in Iran NG pressure drop stations and found out that the total of 4200 MW electricity could be generated in these stations[15]. Farzaneh-Gord et al. studied the methods of utilizing the existing pressure exergy in the CGS of Bandar Abbas refinery [16]. They also did a comprehensive research on enhancing the output energy of CGSs applying turbo expanders and presented some useful suggestions [17].

2 The Proposed System

In order to assess the proposed system effectiveness, Birjand station with capacity of 60000 m^3/hr has been chosen as a typical station. Birjand is the capital of South Khorasan province with latitude 32^o and longitude 59^o. Pay attention to Birjand's latitude, it obviously has a high potential about receiving solar energy among other cities of Iran.

Figure (2) illustrates a schematic diagram of the proposed system for implementation in Birjand station. Considering the figure, in the first step of the modification a solar heater system has been attached to the line heater for reducing fuel consumption of the heater. Through the next step, the throttling valve is substituted by a work producer set consisting of a turbo expander set and an electricity generator with their peripherals.

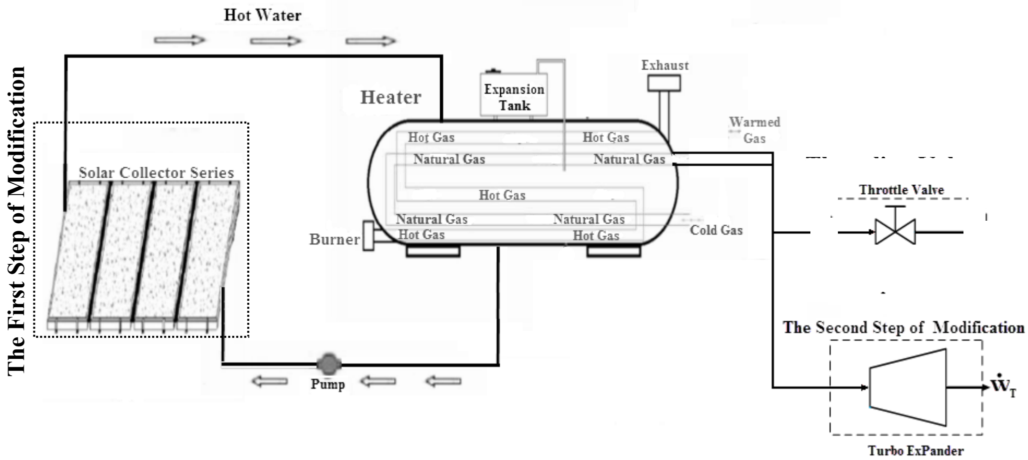

Figure 2 The proposed system schematic diagram

3 Formulation for CGS Initial Plan

Considering the above presented information about CGSs, the optimum temperature which NG has to reach before the throttling valve could be found as below:

$$T_{NG-2} = T_{hyd} + 5 + \Delta T_{tv} \tag{1}$$

Where, $T_{hyd}, \Delta T_{tv}$ refer to hydrate forming temperature and temperature drop in throttling valve for NG respectively. ΔT_{tv} is hourly recorded at the station place for all days of the year and the existing reports about Birjand station shows that this value is always a number between 12 to 15 ^{o}C. It also should be noted that the value 5 in the above equation has been considered as a confidence factor. The energy amount which NG has to gain to reach this temperature can be calculated as follow:

$$\dot{Q}_{NG} = \dot{m}_{NG} \cdot c_{p-NG}(T_{NG-2} - T_{NG-1}) \tag{2}$$

In which, $\dot{m}_{NG}, c_{p-NG}$ are mass flow rate and heating capacity of the NG respectively. T_{NG-1} is also inlet gas temperature to the station and could be gained by the below equation [18]:

$$T_{NG-1} = 0.0084T_o^2 + 0.318T_o + 11.40 \tag{3}$$

Regarding the fact that the CGS inlet gas is exited from 1.5m depth of soil, the above equation computes the soil temperature at the depth of 1.5 m as a quadratic function of the ambient temperature. On the other hand, the correlation between inlet and outlet NG into/from the heater should be written as follow [19]:

$$\frac{T_w - T_{NG-2}}{T_w - T_{NG-1}} = e^Y, \quad Y = \frac{-\pi\, D_{oc}\, L_c\, U_c}{\dot{m}_{NG} \cdot c_{p-NG}} \tag{4}$$

In the above equation, T_w, D_{oc}, L_c are the heater water temperature, outer diameter of the heater coils and the coils length, respectively. Also, U_c is overall heat transfer coefficient between the heater coils and water which could be found using the following equation:

$$\frac{1}{U_c} = \frac{D_{oc}}{D_{ic}h_{ic}} + R''_{ic}\frac{D_{oc}}{D_{ic}} + \frac{\frac{D_{oc}}{2}\ln\frac{D_{oc}}{D_{ic}}}{K_c} + R''_{oc} + \frac{1}{h_{oc}} \tag{5}$$

Where, $K_c, D_{oc}, D_{ic}, R''_{oc}, R''_{ic}, h_{oc}, h_{ic}$ are conductivity heat transfer coefficient, outer and inner diameter, outer and inner thermal resistance of sediments and outer and inner convection heat transfer coefficient of the heater coils, respectively. However, the other study shows that the line heaters overall heat transfer coefficient is 586 W/m^2.K [20].Assuming hourly variation of the heater water temperature (3600s), the heater heating duty to provide required heat for warming up the NG can be calculated by:

$$\dot{Q}_h = \dot{Q}_{NG} + m_w \cdot c_{pw} \cdot (T_w^{i+1} - T_w^i)/3600 \tag{6}$$

The fuel mass flow rate for providing this amount of energy could be computed by equation below:

$$\dot{m}_f = \frac{\dot{Q}_{NG} + m_w \cdot c_{pw} \cdot (T_w^{i+1} - T_w^i)/3600}{LHV \cdot \eta_h} \tag{7}$$

In two above equations, LHV, m_w, c_{pw} are lower heating value of the fuel, the heater water mass and water heat capacity respectively. It's also noteworthy that the line heaters thermal efficiency (η_h) is usually is number between 35-45% and the value of 40% through this research has been applied in all steps of calculations as it has been found as an appropriate value in the previous study of authors [20].

Physical exergy of the NG stream which is destroyed by the throttling valve could be earned as follow:

$$Ex = \dot{m}\left((h - h_o) - T_o(s - s_o)\right) \tag{8}$$

Where, h_o, s_o, h and s are the NG enthalpy and entropy in the ambient situations and in specific situations respectively. T_o and $\dot{m}$ also refer to ambient temperature and NG mass flow rate respectively. It also should be mentioned that EES engineering software has been employed for computing the above passive parameters.

4 Formulation for the Proposed CGS Plan

There are two important parameters through the formulation of CGS initial plan which are fuel consumption and exergy destruction rates at the station, while for the proposed system, the available solar energy and the net producible work by the turbo expander in addition to the two aforementioned parameters both must be calculated.

4.1 Calculation of Available Solar Energy

Regarding this fact that no high temperature heat is needed in CGSs, flat plate solar collectors are employed through this research. Useful energy rate which could be gained by operating fluid from a flat pale collector could be computed as follow [21-25]:

$$\dot{Q}_{solar} = A_c F_R \left\{S - U_l (T_{fi} - T_o)\right\} \tag{9}$$

Where U_l is overall loss coefficient of the collector and it is the summation of 3 loss coefficients as: U_t loss coefficient from top, U_b loss coefficient from back and U_e loss coefficient from edge of the collector [21].

$$U_l = U_t + U_b + U_e \tag{10}$$

U_t could be calculated as below equation [21-25]:

$$U_t = \left\{ \frac{N}{\frac{C}{T_{pm}}(\frac{T_{pm}-T_o}{N+f})^e} + \frac{1}{h_w} \right\}^{-1} + \frac{\sigma(T_{pm}+T_o)(T_{pm}^2+T_o^2)}{(\varepsilon_p + 0.00591.N.h_w)^{-1} + \frac{2N+f-1+0.133\varepsilon_p}{\varepsilon_g} - N} \tag{11}$$

Where ε_p, ε_g, T_{pm}, T_{am} and N are the plate and the cover emittance, the absorber plate average and ambient temperature and the number of glass layers. The other parameters could be calculated as below:

$$f = (1 + 0.089 \cdot h_w - 0.1166 \cdot h_w \cdot \varepsilon_p)(1 + 0.07866 \cdot N) \tag{12}$$

$$C = 520 \cdot (1 + 0.000051 \cdot \beta^2) \tag{13}$$

$$e = 0.43(1 - \frac{100}{T_{pm}}) \tag{14}$$

h_w is wind coefficient from upside of the collector which could be calculated as bellow:

$$h_w = 9.4 \times V_m^{1/2} \tag{15}$$

The losses coefficient through the back of collector could be calculated using bellow equation:

$$U_b \cong \frac{K \cdot A}{d} \tag{16}$$

Where k is thermal conduction factor, A is the collector back area, and d is the collector insulator thickness. For loss coefficient through the edge of the collector the following equations could be employed:

$$Ue \cong \frac{(U \cdot A)_e}{A_c} \tag{17}$$

$$(U \cdot A)_e = \frac{P \times d_c \times K_e}{d_e} \tag{18}$$

A_c, P, d_c, d_e and K_e are the absorber plate area, the collector perimeter, the collector thickness and the edges insulator thickness and thermal conductivity factor from the edges. The value of T_{pm} could be calculated from the following equation [22]:

$$T_{pm} = T_{fi} + \frac{Q_{solar}/A_c}{F_R.U_l}.(1 - F_R) \tag{19}$$

Where

$$F_R = \frac{\dot{m} \cdot C_p}{A_c \cdot U_l}\left[1 - \exp(\frac{-A_c.U_l.F'}{\dot{m} \cdot C_p})\right] \tag{20}$$

$$F' = \frac{1}{WU_l\left[\frac{1}{\left(\frac{1}{U_l(W-D_{ot})\varphi} + \frac{\delta_a}{KaD_{ot}}\right)^{-1} + U_l D_{ot}}\right] + \frac{1}{\pi D_{it} h_{fi}}} \tag{21}$$

In the last two equations F_R and F' are removal factor and collector efficiency factor, respectively. Also $\dot{m}$, δ_a, D_{ot}, D_{it}, K_a and h_{fi} are the operating fluid mass flow rate through the collector, the absorber plate thickness, the inner and outer diameters of the collector tubes, the thermal conductivity factor of the absorber plate and the thermal convection factor between the operating fluid and the collector tubes, respectively.

The functional dependence of the collector efficiency on the meteorological and system operation values can be represented by the equation below [21]:

$$\eta_i = 0.78 - 1.4\frac{(T_{pm} - T_o)}{I_T} - 0.09\frac{(T_{pm} - T_o)^2}{I_T} \tag{22}$$

Which I_T is radiated solar flux on slopped collector. The collector efficiency has been calculated by the manufacturer based on En-12975-2 Standard [26].

It should be noted that numerical solution methods must be employed for solving equation 9 and Newton-Raphson method in Matlab programming language have been used by the authors. It's also noticeable that the employed flat plate solar collectors in this work are industrial available collectors in Iran's market. Table 1 presents the details of the applied collectors [26].

Table 1 Properties of employed solar collectors

Characteristic	Information
Collector Length	200 Cm
Collector Wide	95 Cm
Collector Thickness	9.5 Cm
Cover Matter	Glass
Cover Thickness	4 mm
Absorber Plate Thickness	0.5 mm
Tubes Inner Diameter	10 mm
Tubes outer Diameter	12 mm
Tubes space	150 mm
Plate Area	1.51 m^2
Plate Matter	Copper

4.2 The Heater Hearting Duty Calculation in Presence of Solar System

While the solar heater system works as an auxiliary system aimed to decrease the fuel consumption, for calculating the heater heating duty, equation 6 should be rewritten as:

$$\dot{Q}_h = \dot{Q}_{NG} + \left(m_w \cdot c_{pw} \cdot (T_w^{i+1} - T_w^i)/3600\right) - n \cdot \dot{Q}_u \tag{23}$$

Where, the parameter n refers to number of collector modules in the proposed system. The value of n should be selected based on a thermo-economical analysis which is described in the next sections. Considering the above equation, the fuel mass flow rate of the heater could be found as:

$$\dot{m}_f = \frac{\dot{Q}_{NG} + \left(m_w \cdot c_{pw} \cdot (T_w^{i+1} - T_w^i)/3600\right) - n \cdot \dot{Q}_u}{LHV \cdot \eta_h} \tag{24}$$

4.3 The Producible Net Work by Turby Expander

Prior studies clearly show that NG pressure reduction process could be utilized to generate electricity and cooling employing a suitable system [10-17]. Since the proposed system is equipped with an electricity generation pack including a turbo expander accompanying with an electricity generator machine, so the far more temperature heat is needed rather than the initial plan of CGS. The authors' prior study demonstrates that temperature 85^oC is needed at the turbo expander entrance in order to prevent gas hydrate forming after the expansion process through the turbine regarding 5^oC confidence factor [27].The maximum obtainable work from the turbine could be calculated by:

$$\dot{W}_{rev} = Ex_i - Ex_o \tag{25}$$

In which, Ex is the NG stream exergy and could be found using equation 8. Knowing thermodynamics second law efficiency of the turbo expander ε, actual work amount which could be done by the turbine can be calculated by:

$$\dot{W}_T = \varepsilon \cdot \dot{W}_{rev} \tag{26}$$

It's noteworthy that the employed value of the exergetic efficiency in this study was 75% [17]. Knowing electricity generator machine efficiency η_g, the obtainable electrical energy could be gained as:

$$P = \eta_g \cdot \dot{W}_T \tag{27}$$

Electricity generator machine efficiency has been chosen 90% through this research [17].

5 The Exergy Destruction Rate in the Conventional/Improved Configuration

Exergy is an important definition of the second law of thermodynamics which can determine the maximum obtainable net work in a material or energy stream. The rate of exergy destruction through a control volume could be found as follow [28, 29]:

$$\dot{E}_d = \sum_k (1 - T_o / T_k)\dot{Q}_k - \dot{W} + \sum_i \dot{m}(e_{f-in} - e_{f-out}) \tag{28}$$

Where, $\dot{E}_d, \dot{W}, \dot{Q}_k, T_o$ are the rate of exergy destruction, the amount of work done, the amount of heat transfer from the control volume and ambient temperature, respectively. e_f also refers to the total amount of physical, chemical, potential and kinetic exergy of the control volume. It's noteworthy that the kinetic and potential terms of exergy are negligible in this work.

Physical exergy could be obtained as below:

$$e_f^{\,ph} = (h - h_o) - T_o(s - s_o) \tag{29}$$

Where h and s are enthalpy and entropy in the target stream temperature and h_o and s_o refer to enthalpy and entropy of the stream in the ambient temperature, respectively. Chemical exergy also depends on the target stream nature which is thoroughly discussed in the next subsections.

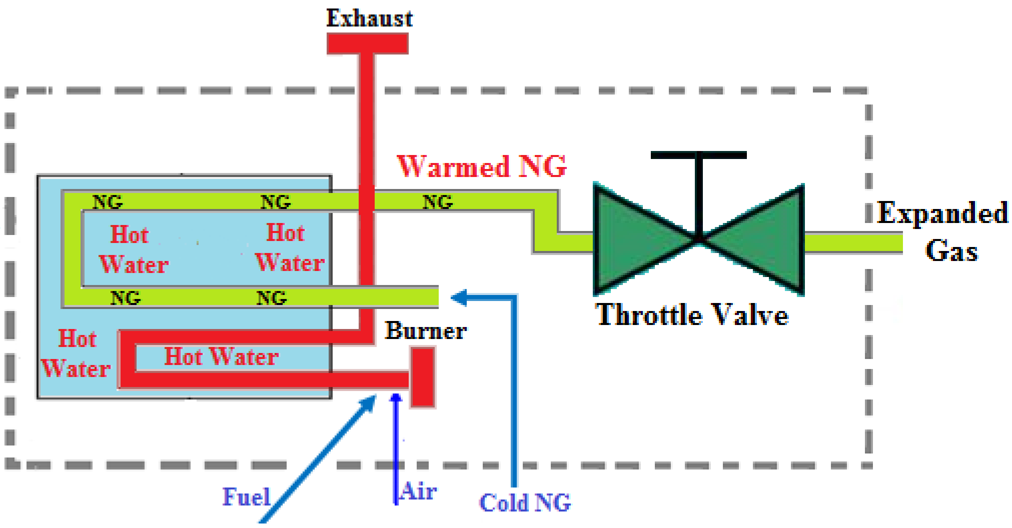

Figure 3 The conventional configuration control volume details

In this section, a comprehensive exergy analysis is done on the both conventional and improved configuration of CGS in order to assess the proficiency of proposed system from the aspect of exergy destruction.

5.1 Exergy Destruction Rate in the Conventional Configuration

Figure (3) demonstrates the control volume detail related to the conventional configuration. The exergy balance correlation for the above control volume in steady state could be written as follow:

$$\Delta\dot{E}_{f-t} - \Delta\dot{E}_{NG} = \dot{E}_d \tag{30}$$

Where $\dot{E}_d$, $\Delta\dot{E}_{NG}$ and $\Delta\dot{E}_{f-t}$ are the exergy destruction rate, the exergy balance for the passing NG through the pipeline and the exergy balance for the fire-tubes of the heater, respectively. The NG exergy balance should be written as below:

$$\Delta\dot{E}_{NG} = \dot{m}(e_{f-in} - e_{f-out}) = \dot{m}_{NG}\left[(h_{in} - h_{out}) - T_o(s_{in} - s_{out})\right] \tag{31}$$

Where h_{in}, h_{out}, s_{in} and s_{out} are the inlet and outlet enthalpies and entropies of the NG to/from the control volume, respectively. Similarly, the exergy balance for the fire-tubes could be obtained as:

$$\Delta\dot{E}_{f-t} = \dot{E}_{fuel} + \dot{E}_{air} - \dot{E}_{exh} \tag{32}$$

In the above correlation, fuel, air and exh refer to the burned fuel, the inlet air to the fire-tubes which is required for combustion procedure and the outlet combustion production from the heater exhaust. The three items in the right side of the above equation are hereunder discussed.

The principal correlation for calculation of the fuel exergy is:

$$\dot{E}_{fuel} = \dot{n}\left\{(\bar{h} - \bar{h}_o) - T_o(\bar{s} - \bar{s}_o) + \bar{e}^{ch}\right\} \tag{33}$$

In the above equation, $\bar{e}^{ch}$, $(\bar{s} - \bar{s}_o)$ and $(\bar{h} - \bar{h}_o)$ are the chemical exergy of each component in the consuming fuel, the molar entropy difference and the molar enthalpy difference comparing the reference condition. As the fuel is entered the fire-tube at reference condition, the aforementioned items should be zero. In this equation, $\dot{n}$ refers to the molar flow rate of the fuel which can be determined as:

$$\dot{n} = \frac{\dot{m}}{M} \quad \& \quad \dot{m} = \frac{\dot{Q}_h}{\eta_{th} \times LHV_f} \tag{34}$$

In which $\dot{m}, M, LHV, \eta_{th}$ are the fuel mass flow rate, molecular weight, lower heating value and the heater thermal efficiency, respectively. The proportion of the target fuel compositions is [30]:

$$0.98\, CH_4 + 0.0075\, C_2H_6 + 0.0075\, N_2 + 0.005\, CO_2 \tag{35}$$

Regarding the above proportions, it could be noted that with a negligible deviation the main fuel could be assumed as pure methane.

As the inlet air is entered fire-tube in the reference condition, the middle item in the right side of Eq.32 is zero.

For calculation of the exergy of outlet combustion production for the heater exhaust, the following equation could be used:

$$\dot{E}_{exh} = \sum_i \dot{n}_i \left\{ (y_i\, \bar{e}_i^{ch} + \bar{R} T_O\, y_i \ln y_i) - T_O(\bar{s}_i - \bar{s}_O) + (\bar{h}_i - \bar{h}_O) \right\} \tag{36}$$

Where, $y_i, \dot{n}_i$ are the molar ratio and the number of moles for each component in the combustion production, respectively. For calculation of these two items, the following combustion reaction should be completed. It should be noted that the combustion reaction is completed based on the prior study of the author which shows the combustion procedure in a linear heater is usually done with approximately 300% theoretical air.

$$CH_4 + 6(O_2 + 3.76N_2) \rightarrow (CO_2) + 2(H_2O)_{(g)} + 22.56(N_2) + 4(O_2) \tag{37}$$

It should also be noted that in Eq.36, $\bar{s}_i, \bar{h}_i$ are molar entropy and enthalpy of each of the combustion production components in the exhaust outlet temperature. For computing the exhaust outlet temperature the thermodynamics first law should be written for the control volume.

$$\frac{\dot{Q}_{NG}}{\dot{n}_f} + \bar{h}^o{}_{CH_4} = (\bar{h}^o{}_{CO_2} + \Delta\bar{h}_{CO_2}) + 2(\bar{h}^o{}_{H_2O(g)} + \Delta\bar{h}_{H_2O(g)}) + 22.56(\Delta\bar{h}_{N_2}) + 4(\Delta\bar{h}_{O_2}) \tag{38}$$

Extracting the required values from the thermodynamics tables as well as using the known parameters in the above equation, the exhaust outlet temperature and subsequently the exhaust outlet exergy could be calculated. Knowing the exergy value change through the fire-tubes and NG pipeline, the rate of destroyed exergy in the control volume will be calculable. Exergy efficiency or the target control volume could be defined as:

$$\eta_{II} = \frac{The\ Effective\ Exergy}{The\ Supplied\ Exergy}, \quad where \quad Supplied\ Exergy = Fuel\ Exergy \tag{39}$$

5.2 Exergy Destruction Rate for the Improved Configuration

As it will be demonstrated in the results section, the CGS with both solar energy and turbo expander is much more impressive that a CGS with only solar energy, therefore, the exergy destruction assessment is only done for the CSG equipped to both solar heating system and turbo expander. The corresponding control volume for this configuration is shown in figure (4).

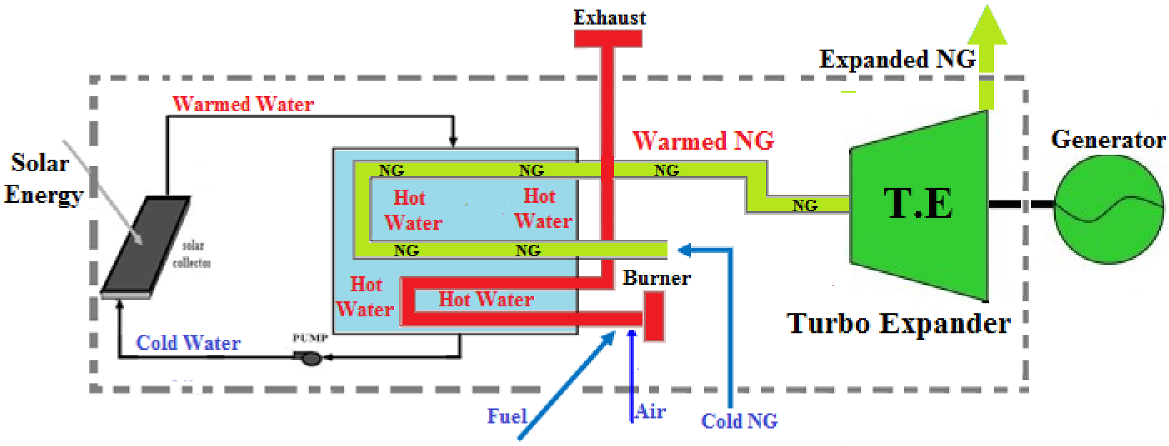

Figure 4 The improved configuration control volume

In the steady state, the exergy balance for the above control volume should be written as:

$$\dot{E}_d = \Delta\dot{E}_{f-t} + \dot{E}_{solar} - \Delta\dot{E}_{NG} - \dot{W} \quad (40)$$

Where $\dot{E}_{solar}$ is solar exergy which is entered through the control volume employing flat plate solar collectors. The available solar exergy for the above control volume could be calculated as below [29]:

$$E_{solar} = \eta_o I_T A_p \{1 - (T_o/T_s)\} \quad (41)$$

In which η_o is optical efficiency and could be calculated as follow:

$$\eta_o = S/I_T \quad (42)$$

T_s is also the effective temperature of the sun and equal to 4350 K. For calculation of the fire-tube and the NG exergies the same correlations as the conventional configuration can be used. The same correlation as Eq.39 could be used for calculation of the exergy efficiency, though the effective and supplied exergy parameters are different than it used to be. The effective exergy is the sum of the NG exergy and net producible work rate and the supplied exergy is the summation of solar and fuel exergies. It is worth mentioning that, the turbo expander doesn't affect the outlet NG conditions because the outlet conditions of the NG should be stable and constant in anyway.

6 Results and Discussion

The key parameter for calculating the temperature of the CGS inlet NG, the absorbable solar energy by the solar collectors and the NG stream exergy is definitely the ambient temperature. Figure (5) illustrates Birjand's daily average ambient temperature as well as the CGS inlet NG temperature in (2011). As expected the warm months of the year begin from late May up to early October and ambient temperature varies between 16 and about 40°C, whereas through other months it changes from -10 to 20°C.

Figure (6) presents the NG pressure and the mass flow rate of Birjand station. Obviously from the figure, in contrast with cold months of the year in which the NG pressure and mass flow rate are remarkably high, both the pressure and the mass flow rate decrease due to consumption drop during the hot months of the year.

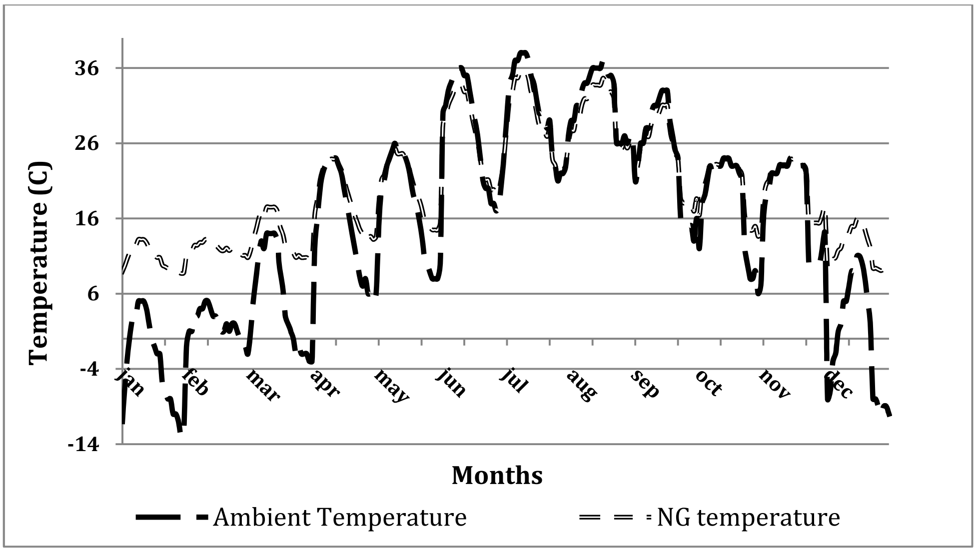

Figure 5 Birjand ambient and CGS inlet gas temperatures in 2011

One of the most important parameters in this research is the available solar energy amount which could be utilized by solar collectors and calculated by the related equations. It's noticeable that, the slop angle of the flat plate collectors is really effective on the absorbable amount of solar energy, therefore, the optimum slop angle must be found. For this purpose, it's suggested to add a number in the range of 10-12 to the latitude of the target place in Iran [24]. The authors have found 45^{o} as the best slop angle based on a trial and error manner for Birjand city. Figure (7) presents the absorbable solar energy by 1m^2 area collector as well as the obtainable energy amount by the operating fluid.

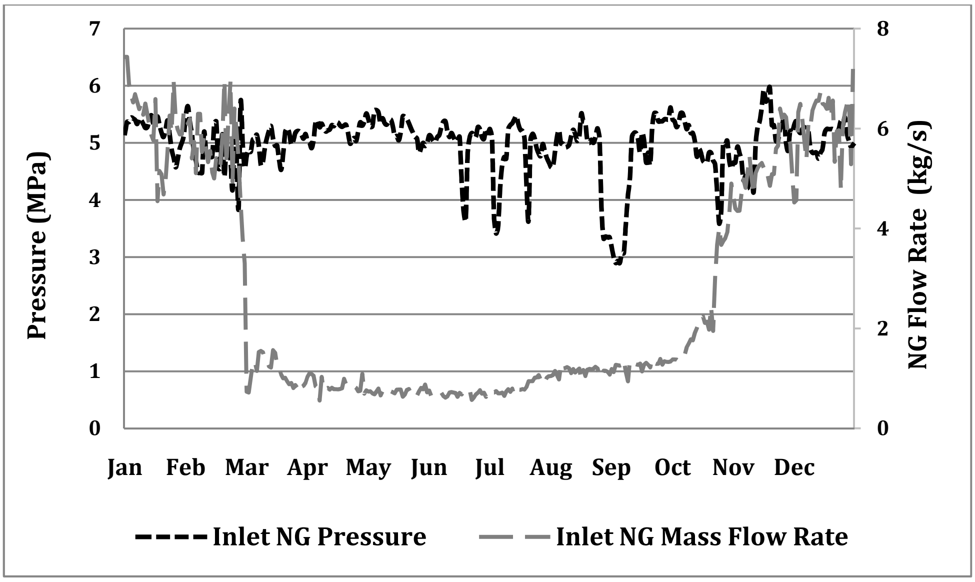

Figure 6 Natural gas pressure and mass flow rate in 2011

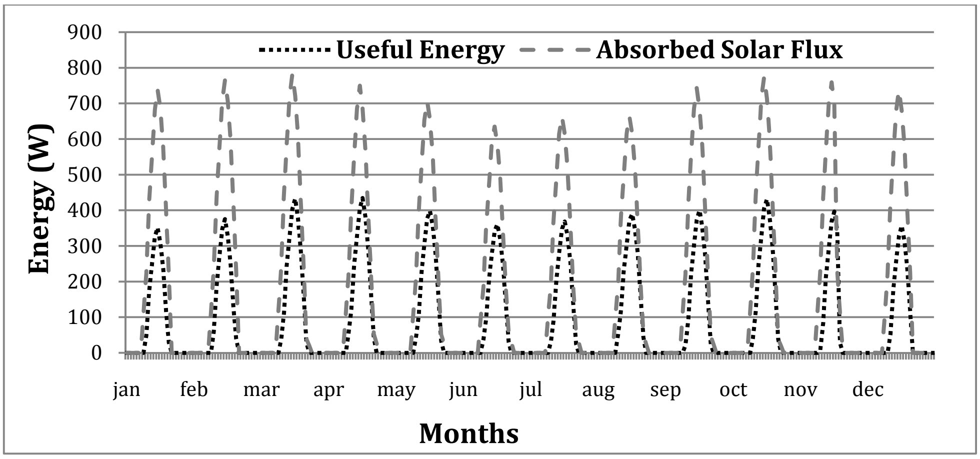

Figure 7 Monthly and hourly average useful energy versus absorbed solar flux

In general the heater heating duty could be discussed from 4 aspects. Firstly, the heating duty in the CGS initial plan; secondly, if the only target is to reduce fuel consumption and the project doesn't aim to take advantage of the available exergy in the station; Thirdly, in contrast with the prior assumption the only target is to take advantage of the existing availability in the station without any plan for reducing the fuel consumption and the last aspect refers to the new proposed plan which contains both the solar system and the electricity generation part. What's clear is that when the turbo expander is added to the system a marked leap in heating demand will occur because when the system is equipped to a turbo expander the required temperature is 85^{o}C while in the system without turbo expander; the required temperature is only about 20 to 25^{o}C. Figure (8) presents a comparison between the heating duty of the heater in the four different aforementioned cases.

Figure (9) shows a comparison on the monthly average (hourly too) fuel saving in two with/without turbo expander systems in (2011) aimed to show the solar heater system effectiveness. Obviously from the graph, the solar system in the system with turbo expander is dramatically more effective than the system without turbo expander due to the much more number of the employed collectors. The question may arise here is that "why are the employed collector numbers different in the two systems?"

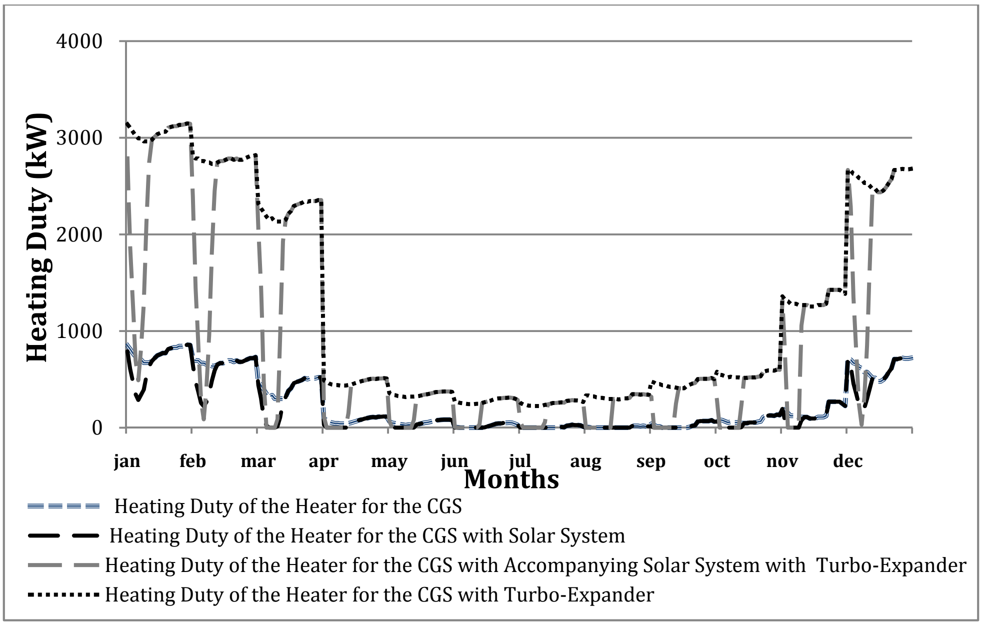

Figure 8 Comparison between monthly and hourly average heating duty of the heater for four different cases

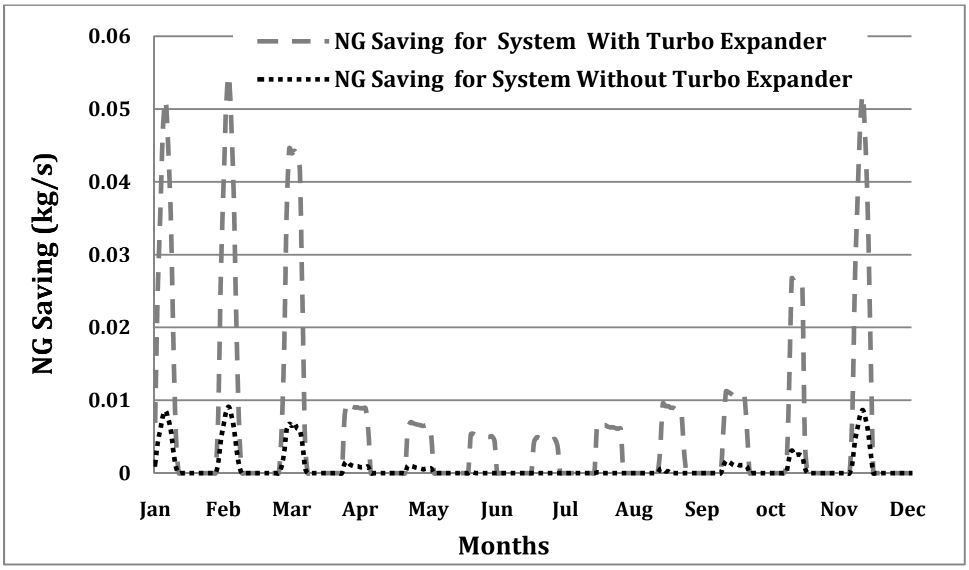

Figure 9 Fuel saving in with/without turbo expander systems

The inquiry can be answered by an economical analysis which is mostly applied to estimate the cost of capital for each industrial project. Based on this economical principle, both annual fuel cost for the proposed system and the cost of capital curves should be drown in the same graph, the conflux point reveals the optimum cost of capital which is the purchase and the installation price of the solar system and the electricity generation pack [24]. Figure (10) reveals the optimum capital cost for the both with/without electricity generator systems based on the aforementioned principle. Obviously from the figure, for the proposed system equipped to the electricity generator pack, the optimum capital cost is 251251 USD for 1500 collectors, whereas for the system without any turbo expander the optimum collector numbers are 250 which lead to 52000 USD. It's noticeable that for the system with turbo expander, the electricity generator package price has not been considered in the above graph because the graph must only include the price of equipments which cause to reduce fuel consumption.

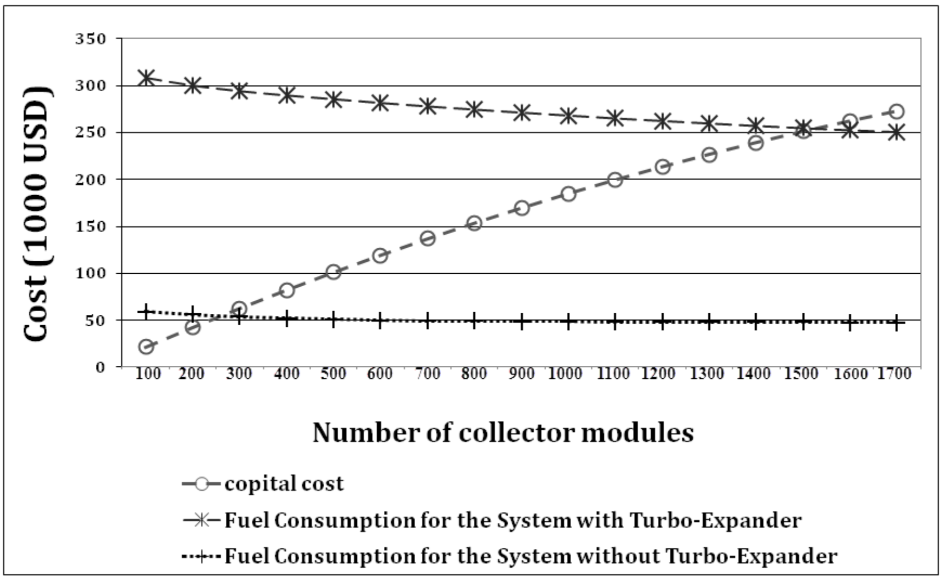

Figure 10 Thermo-economical survey for finding the optimum capital cost

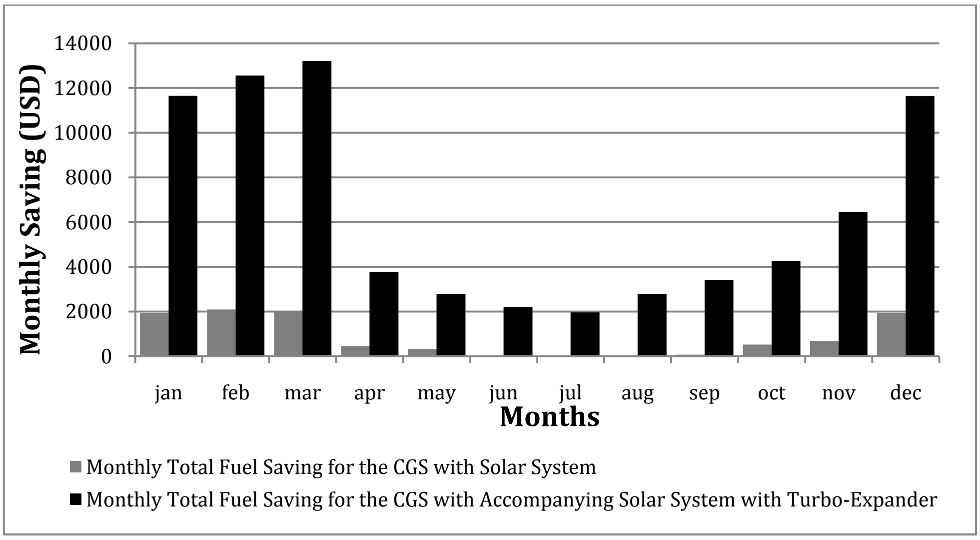

Figure 11 Total monthly fuel saving for both with/without electricity generator system in 2011

Figure (11) represents a comparison on the total monthly fuel saving between the systems with/without electricity generator package. As expected, providences during cold months are much more than hot months of the year. Figure (12) shows the monthly average net producible work by the turbo expander. As it's clear from the graph, the maximum value belongs to both January and February by approximately 1000 kW. The decreasing trend of the producible net work begins with ambient temperature growth and a sudden collapse is observed in April by almost 200 kW. After that, a roughly steady trend is seen till October and then the increasing trend of the curve begins, so that the producible net work reaches to 800 kW in December. It should be noted that the capital cost of the system equipped to the electricity generation package includes the package cost of 1651031 USD in addition to the solar system cost. Considering the turbo expander net work, the total annual producible electricity will be 3177252 kWh and the electricity price is 0.215 USD/kWh in Iran. Gathering the annual fuel saving by the solar system (62207 USD) and the generated electricity (683109 USD) the total benefit of the proposed system for (2011) can be determined. On the other hand for the system without the electricity generator pack, the annual fuel saving is almost 8500 USD.

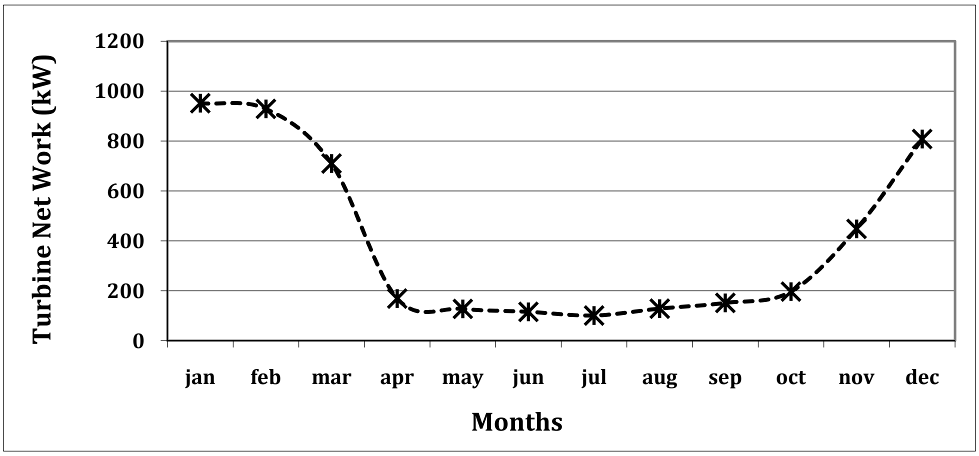

Figure 12 The monthly average producible net work by the turbo expander

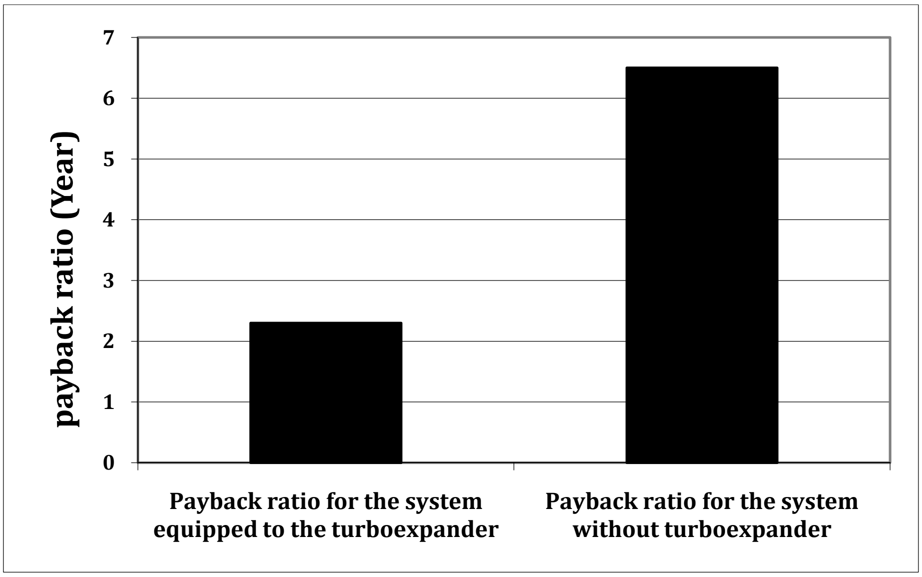

Figure 13 Simple payback ration comparison

Regarding the presented economical information above, the payback ratio could be calculated for each system. There are some authentic methods to survey proposal project impacts and both simple payback ration and NPV method have been employed for this research. Figure (13) compares the simple payback ratio for the system with/without turbo expander dividing the total capital cost into the annual benefit of each system. Obviously from the figure, the simple payback ratio for the system equipped to the turbo expander pack is about 2.3 years, while this parameter for the other system is roughly 6.5 years. The main problem with this kind of economical survey is that some vital and effective parameters such as inflation rate, depreciation and subsidiary costs are not considered through the analysis. That's why this project is analyzed by NPV method as a comprehensive method of economical assessment, too. In this method NPV which refers to investment return period is calculated as:

$$NPV = \sum_{t=1}^{N} \frac{R_t}{(1+i)^t} \tag{43}$$

Where i, t represent interest/inflation rate and the time of cash flow respectively, while R_t is the net cash flow i.e. cash inflow cash outflow [31]. The equation above could specify that after how many years the capital cost could be returned by the proposed system. Figure (14) reveals NPV values for both systems.

Confirming the results of simple payback ratio analysis, the investment return period for the configuration with the work production unit is much better than the other system. The NPV for the configurations with/without the work production unit are 3.5 and 11.5 years, respectively. The next two figures are presented to show the results of exergy analysis and make a comparison between the two conventional and improved configurations of the CGS from the aspect of exergy efficiency. Figure (15) shows the rate of solar exergy entered the system. Although the solar exergy is totally freely added to the system, it should be considered in the computations. Making a comparison between the above figure and figure (7), it could be realized that the solar exergy trend is just the same as the trend of absorbed solar flux curve. That's surely because the solar exergy amount is a functional of the absorbed solar flux based on Eq. 42.

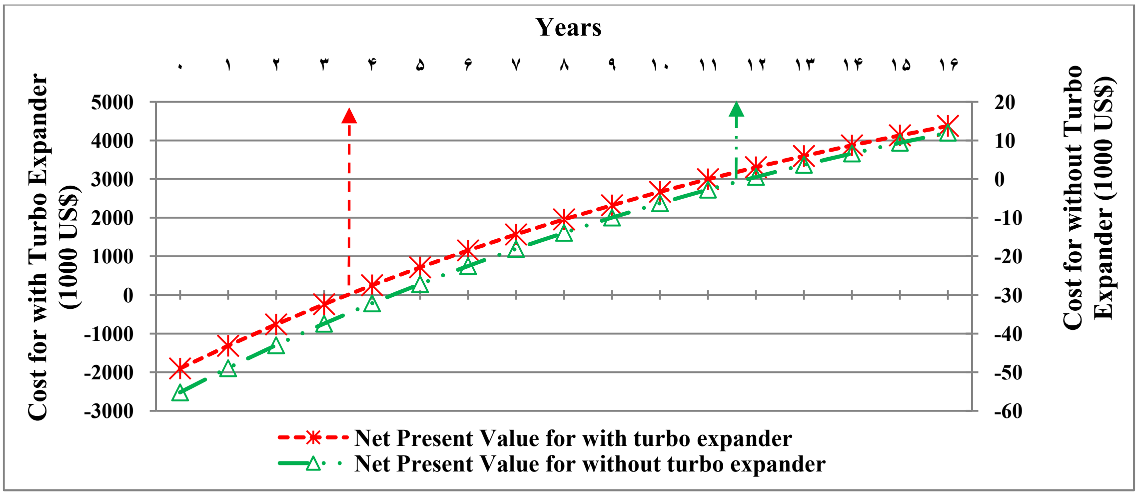

Figure 14 The NPV analysis results

Figure (16) presents the results of comparison between the exergy efficiency in the improved/conventional configuration. According to the definition of the exergy efficiency, it is undoubtedly too low in the conventional configuration of the CGS because of the high rate of the exergy destruction in the heater (specially the high rate of the outlet exergy from the exhaust). In contrast with the conventional configuration, the proposed system will lead to much higher exergy efficiency because: firstly, the heating duty of the heater has been reduced considerably (consequently, the exhaust waste exergy amount has been decreased remarkably) and secondly, the added turbo expander to the system, utilizes a significant portion of the available exergy to produce power.

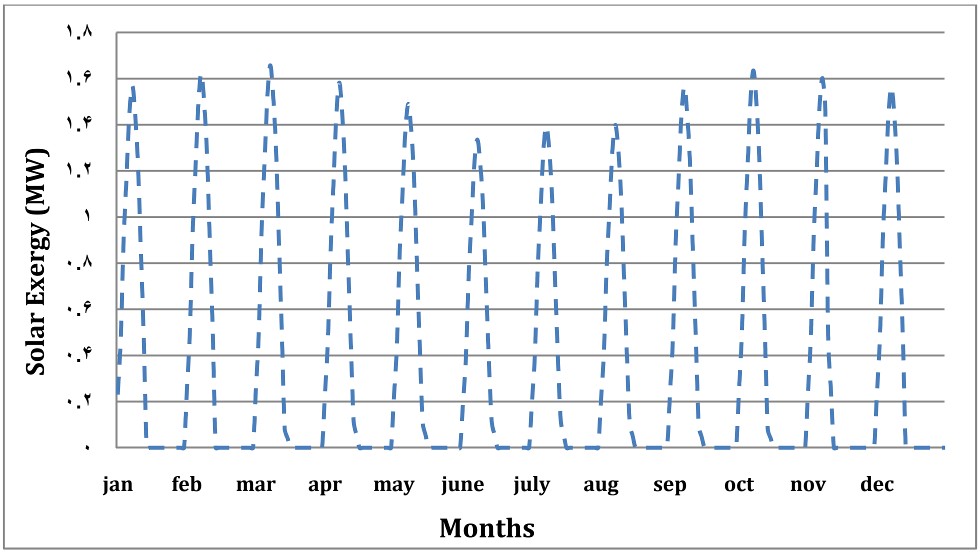

Figure 15 The solar exergy amount added to the system

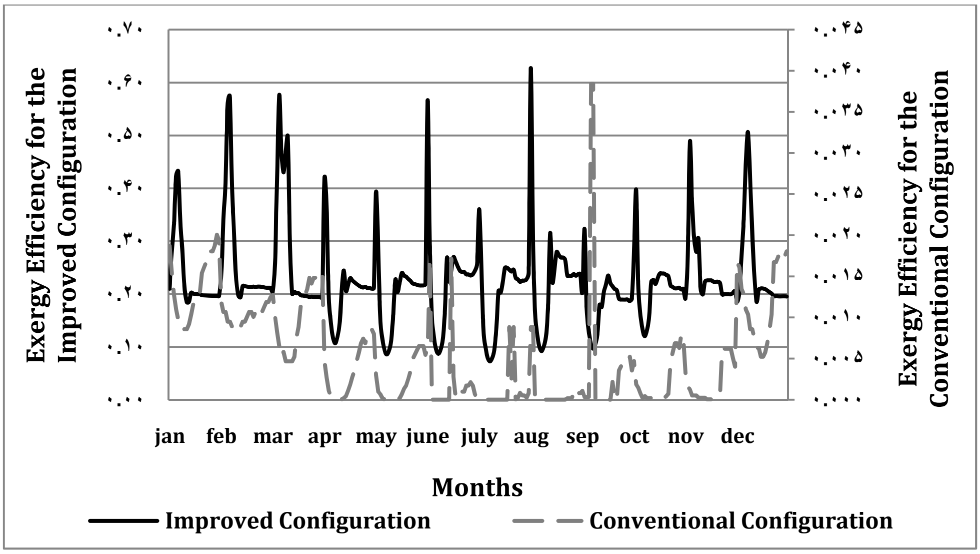

Figure 16 The exergy efficiency comparison

7 Conclusion

NG pressure reduction stations are high potential places for implementing appropriate improvement projects. In this work, two steps of optimizations have been proposed in CGS stations. The first step is adding a solar heater system to the station in order to reduce fuel consumption of the heater and the other is substituting the throttling valve by a turbo expander and an electricity generator for utilizing available exergy in NG stream. Regarding the fact that, Iran is one the biggest NG producers of the world, the appropriate geographical situation of Iran which makes it a high potential country in the field of available solar energy and also pay attention to the results of this research, the implementation of the proposed system for all Iran's CGSs is strongly recommended.

References

[1] Edalata, M., and Mansoori, G. A., "Buried Gas Transmission Pipelines: Temperature Profile Prediction through the Corresponding States Principle", Energy Sources, Part A: Recovery, Utilization, and Environmental Effects, Vol. 10, Issue. 4, pp. 247-252, (1988).

[2] Fundamentals of Natural Gas Pressure Drop Stations Design, Iranian National Gas Company, (2005).

[3] Kalogirou, A.S., "Solar Thermal Collectors and Applications", Progress in Energy and Combustion Science, Vol. 30, 231–295, (2004).

[4] Norton, B., "Solar Process Heat: Distillation, Drying, Agricultural and Industrial uses", Proceedings of ISES Solar World Congress, Jerusalem, Israel on CD-ROM, Jerusalem, Israel, (1999).

[5] Spate, F., Hafner, B., and Schwarzer, K., "A System for Solar Process Heat for Decentralised Applications in Developing Countries", Proceedings of ISES Solar World Congress on CD-ROM, Jerusalem, Israel, (1999).

[6] Benz, N., Gut, M., and Rub, W., "Solar Process Heat in Breweries and Dairies", Proceedings of EuroSun 98, Portoroz, Slovenia on CD-ROM, (1998).

[7] Benz, N., Gut, M., and Beikircher, T.," Solar Process Heat with Nonconcentrating Collectors for Food Industry", Proceeding of ISES Solar World Congress on CD ROM, Jerusalem, Israel, (1999).

[8] Li, H., and Yang, H., "Potential Application of Solar Thermal Systems for Hot Water Production in Hong Kong. Applied Energy, Vol. 86, pp. 175–180, (2009),

[9] Baldini, A., Manfrida1, G., and Tempesti, D., "Model of a Solar Collector/Storage System for Industrial Thermal Applications", Int. J. of Thermodynamics Vol. 12, No. 2, pp. 83-88, (2009).

[10] Bisio, G., "Thermodynamic Anaiysis of the use of Pressure Exergy of Natural-Gas", Energy, Vol. 20, No. 2, pp. 161-167, (1995).

[11] Dresser-Rand: "FCC Power Recovery Expanders", Available at: http://www.dresserrand.com/turbo/eq/expand.asp

[12] Greeff, I. L., Visser, J. A., Ptasinski, K. J., and Janssen, F. J. J. G., "Using Turbine Expanders to Recover Exothermic Reaction Heat Flow Sheet Development for Typical Chemical Processes", Energy, Vol. 29, pp. 2045–2060, (2004).

[13] Hinderink, A. P., Kerkhof, F. P. J. M., Lie, A. B. K., de Swaan, J., Arons and van der Kooi, H. J., " Exergy Analysis with a Flow Sheeting Simulator, Part 1: Theory; Calculating Exergies of Material Streams", Chem. Eng. Sci., Vol. 51, No. 20, pp. 4693–4700, (1996).

[14] Pozivil, J., "Use of Expansion Turbine in Natural Gas Pressure Reduction Stations", J. Acta Montanistica Slovaca, Vol. 9, pp. 258-260, (2004).

[15] Farzaneh-Gord, M., and Magrebi J.," Exergy of Natural Gas Flow in Iran's Natural Gas Fields", International Journal of Exerg, Vol. 6, No. 1, pp. 131-142, (2009).

[16] Farzaneh-Gord, M., Manzari, M., Magrebi, M., Hashemi, S., Eftekhari, H., and Farokhi A.,. "Using Pressure Exergy of Natural Gas in Bandar-Abbas Refinery Gas Pressure Drop Station, the 2nd ICMSAO, pp. 24-27, (2007), March. Abu Dhabi, UAE.

[17] Farzaneh-Gord, M., Hashemi, S., and Sadi, M., "Energy Destruction in Iran's Natural Gas Pipe Line Network", Energy Exploration and Exploitation, Vol. 25, pp. 393-406, (2007).

[18] Najafi-mod, M.H., Alizadeh, A., Mohamadian, A., and Mousavi J., "Investigation of Relationship between Air and Soil Temperature at Different Depths and Estimation of the Freezing Depth (Case study: Khorasan Razavi), Journal of Water and Soil of Ferdowsi University of Mashhad, Vol. 22, No. 2, pp. 456-466, (2008).

[19] Incropera, F.P., and DeWitt, D.P., "*Fundamentals of Heat and Mass Transfer*", 5th Ed. New York, John Wiley, (2002).

[20] Arabkoohsar, A., Farzaneh-Gord, M., Deymi-Dashtebayaz, M., Machado, L., and Koury, R. N.N., "Energy and Exergy Analysis of Natural Gas Pressure Reduction Points Equipped with Solar Heat and Controllable Heaters", Renewable Energy, Vol. 72, pp. 258-270, (2014). December

[21] Zekai S., "*Solar Energy Fundamentals and Modelling Techniques*", 3rd Ed., London, Springer 180, (2008).

[22] Duffie, J.A., and Beckman, W.A., "*Solar Engineering of Thermal Processes*", Second Ed, New York, Willy, (1991).

[23] Yik, F.W.H., Chung, T.M., and Chan, K.T., "A Method to Estimate Direct and Diffuse Radiation in Hong Kong and its Accuracy", Transactions of the Hong Kong Institution of Engineers, Vol. 2, pp. 23–29, (1995).

[24] Farzaneh-Gord, M., Arabkoohsar, A., Rezaei, M., Deymi-Dashtebayaz, M., and Rahbari, H. R., "Feasibility of Employing Solar Energy in Natural Gas Pressure Drop Stations", Journal of the Energy Institute, Vol. 84, No. 3, pp.165, (2011)

[25] Farzaneh-Gord, M., Arabkoohsar A., and Deymi-Dashtebayaz, M., "Feasibility of Accompanying Uncontrolled Linear Heater with Solar System in Natural Gas Pressure Drop Stations", Energy, Vol. 41, pp. 1-9, (2012).

[26] http://solar.polar.ir/.

[27] Farzaneh-Gord, M., and Sadi, M., "Enhancing Energy Output in Iran's Natural Gas Pressure Drop Stations by Cogeneration", Journal of the Energy Institute, Vol. 81, No. 4, pp. 191-196, (2011).

[28] Moran, M. J., and Shapiro, H. N., "*Fundamentals of Engineering Thermodynamics*", Wiley & Sons, (2015).

[29] Bejan, A., "*Advanced Engineering Thermodynamic*", Second Edition, Wiley & Sons, pp. 133-137 & 462-465, New York, (1988).

[30] Farzaneh-Gord, M., and Rahbari, H. R., "Developing Novel Correlations for Calculating Natural Gas Thermodynamic Properties", Chemical and Process Engineering, Vol. 32 No. 4, pp. 435-452, (2011). DOI: 10.2478/v10176-011-0035-1

[31] Farzaneh-Gord, M., Arabkoohsar, A., Deymi Dashtebayaz, M., and Khoshnevis, A. B., "New Method of Solar Energy Application in Greenhouses in Order to Decrease Fuel Consumption", International Journal of Agricultural and Biological Engineering, Vol. 6, No. 4, pp. 1-11, (2013).

Nomenclature

A_c	Collector surface area	(m^2)
CGS	City Gate Stations, natural gas pressure reduction point	
c_{pw}	Specific heat capacity of the water in the tank	(W/kg-K)
$c_{p\text{-}NG}$	Natural gas specific heat capacity in constant pressure	(W/kg-K)
D_{ic}	Internal diameter of the coil	(m or Cm)
D_{oc}	External diameter of the coil	(m or Cm)
Ex	Exergy	(kJ)
$\bar{e}^{ch}$	Chemical exergy	(kJ)
$\dot{E}_d$	Exergy destruction rate	(kW)
$\dot{E}_{exh}$	The outlet exergy from the heater exhaust	(kW)
$\dot{E}_{solar}$	Solar exergy	(kW)
$\Delta\dot{E}_{f-t}$	Fire tube exergy balance	(kW)
$\Delta\dot{E}_{NG}$	NG exergy balance	(kW)
F_R	Removal factor of the collector	
h	Specific enthalpy in a specific temperature	(kJ/kg)
h_{ic}	Internal heat transfer convection coefficient of the Coil	(W/m^2-K)
h_o	Specific enthalpy in ambient temperature	(kJ/kg)
h_{oc}	External heat transfer convection coefficient of the Coil	(W/m^2-K)
i	Interest rate	
I_b	Beam component of the radiated solar flux	(W/m^2)
I_d	Diffuse component of the radiated solar flux	(W/m^2)
I_g	Reflected component of the radiated solar flux	(W/m^2)
I_T	Radiated solar flux to slopped flat plat collector	(W/m^2)
K_c	Effective thermal conductivity of the Coil	(W/m-K)
L_c	Coil length	(m or cm)
LHV	Lowering heating value of fuel (Here natural gas)	(kJ/kg)
M	Molecular weight	(kg/kmol)
$\dot{m}_f$	Mass flow rate of the fuel consumed by the heater	(kg/s)
m_w	Mass of existent water in the tank	(kg)
$\dot{m}$	Mass flow rate	(kg/s)
$\dot{m}_{NG}$	Mass flow rate of the natural gas	(kg/s)
NPV	Net present value	
P	Producible power	(W)
$\dot{Q}_{gh}$	Heat transfer rate into the natural gas main stream	(kW)
$\dot{Q}_h$	Heat transfer rate produced by the heater	(kW)
$\dot{Q}_{solar}$	The heat transfer rate produced by the solar system	(kW)
R''_{ic}	Inner thermal resistance of sediments	(m^2.K/W)
R''_{oc}	Outer thermal resistance of sediments	(m^2.K/W)
S	Absorbable solar flux	(W/m^2)
s	Specific entropy in a specific temperature	(kJ/kg)
s_o	Specific entropy in the ambient temperature	(kJ/kg)
t	Year counter	
T_{hyd}	Temperature that hydrating occurs on it	(^oC-K)
$T_{NG\text{-}1}$	Natural gas temperature before heater	(^oC-K)
$T_{NG\text{-}2}$	Natural gas temperature after heater	(^oC-K)

T_w	Temperature of water in the tank	(^oC or K)
T_{fo}	Outlet water temperature from the collector	(^oC-K)
T_o	Dead state temperature	(^oC-K)
T_{pm}	The average temperature of the collector	(^oC-K)
ΔT_{tv}	Temperature drop in natural gas through throttle valve	(^oC-K)
U_b	Loss coefficient from back of collector	(W/m^2-K)
U_e	Loss coefficient from edges of collector	(W/m^2-K)
U_c	Total heat transfer coefficient	(W/m^2-K)
U_l	Total loss coefficient from collector	(W/m^2-K)
U_t	Loss coefficient from up of collector	(W/m^2-K)
$\dot{W}_{rev}$	Rate of maximum obtainable work	(W)
$\dot{W}_T$	Rate of maximum producible work by the turbine	(W)
y_i	Molar ratio	

Greek symbols

η_h	Heater thermal efficiency
η_g	Generator efficiency
η_{II}	The exergetic efficiency
ε	The second law efficiency
ε_p	Plate emittance
ε_g	Glass emittance
τ	Glass transmission factor
α	Absorber plate absorption factor
β	Slop angle
ρ	Solar radiation reflection factor
$(\tau\alpha)$	Average transmission-absorption factor of the collector

چکیده

ایستگاه سی جی اس یکی از مهمترین بخش های صنعت انتقال گاز طبیعی می باشد که در آن فشار گاز طبیعی ورودی به ایستگاه بوسیله یک شیر اختناق تا سطوح بسیار پایینتری کاهش می یابد. این کاهش فشار می تواند موجب ایجاد هیدراته در جریان گاز طبیعی گردیده و باعث انسداد خط لوله انتقال گاز طبیعی گردد. برای جلوگیری از تولید هیدراته، گاز طبیعی معمولاً با استفاده از هیترهایی که مقدار زیادی گاز طبیعی می سوزانند گرم می شود. در این مقاله، یک طرح جدید برای ایستگاه سی جی اس ارائه می گردد که در آن استفاده از انرژی خورشیدی برای کاهش مصرف سوخت هیترها به همراه استفاده از یک توربین انبساطی برای بهره گیری از اگزرژی موجود در جریان گاز طبیعی برای تولید توان پیشنهاد می گردد. در پایان، یک آنالیز اقتصادی مبسوط به روش ان پی وی بر روی طرح پیشنهادی برای توجیه آن از نظر اقتصادی نیز انجام می پذیرد.