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Effect of Various Parameters on Indirect Fired Water Bath Heaters' Efficiency to Reduce Energy Losses

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Abstract- Indirect fired water bath heaters are found in a wide range of applications in gas industry. Natural gas is transferred in high pressure in order to reduce the pipeline's sizing. As consumers need gas in lower pressure, it should be reduced in gas pressure reduction stations. The main application of indirect heaters is to heat high pressure gas prior to pressure reduction. This prevents hydrate formation that can occur because of the temperature drop due to the Joule-Thompson effect. In recent years, because of energy crisis and environmental problems due to fossil fuels, the necessity of optimizing these high consumption devices is more evident than past. In this paper one city gate station located in Mahshahr city has been considered as a case study. Heat losses and efficiency of the heater installed in this station has been calculated. Heater efficiency was obtained about 52% that is not acceptable. Therefore a solar system has been proposed to provide part of heat demand. Proposed system includes an array of solar collectors with a storage tank. The system is capable to save more than 39000m^3 of NG in a year. Afterwards, the processes that lead to heat natural gas in indirect fired water bath heaters have been modeled. Therefore, using heat loss from stack for preheating the necessary air for combustion has been investigated. At the end, a cycle to generate power from the thermal energy of flue gases has been modeled. The results show that heating the combustion air will decrease fuel consumption and increase heater efficiency. Moreover, with considering the proposed system as a power plant, 168MWh power can be generated in a year that causes considerable income for the station.

Keywords- *Indirect fired water bath heater; Gas pressure reduction station; Reduce energy losses.*

I. INTRODUCTION

In most of gas pressure reduction stations, gas pressure is reduced by a throttle valve. The thermodynamic process that is occurred in regulators of the station is passing high pressure gas through an orifice that make it expanded. Energy equation for this process is [1]:

$$h_i + \frac{V_i^2}{2} = h_e + \frac{V_e^2}{2} \tag{1}$$

Where h_i, h_e, V_i and V_e are the enthalpy and velocity of natural gas at the inlet and outlet of regulator. Gas pressure reduction could be assumed as a constant enthalpy process. Joule-Thompson coefficient (μ_{JT}) for constant enthalpy process is [1]:

$$\mu_{JT} = \left(\frac{\partial T}{\partial P}\right)_h \tag{2}$$

In which, T is temperature and P is pressure. μ_{JT} for this process is positive. Therefore pressure drop causes temperature drop. To prevent hydrate formation, natural gas should be preheated before pressure reduction. The temperature of outlet gas after pressure reduction should be about 15˚C [2]. Therefore it is necessary to use heaters to raise the temperature of natural gas in city gate stations (CGS). Indirect fired water bath heaters are the most common heaters in gas stations. These heaters are designed based on API12K standard [3]. Since millions of cubic meters fuels are consumed in these heaters around the world, the performance of these devices should be monitored more precisely. In this regard khalili [4] calculated the heat losses and efficiency of the heater at Shahrekord station. Riahi [5] has presented some solutions to optimize the combustion efficiency of the heaters installed at Ardabil station. Farzaneh-Gord [2] has carried out some considerable efforts about using solar system at Sari station. This paper has tried to have a comprehensive attitude on indirect fired water bath heaters and the results and solutions that presented about the heater as a case study can be applied for other cases. Mean ambient temperature in one hour intervals is the main data for this research. At first one day of a month has been chosen as an average day that the climate and ambient temperature of these days are very similar to the related month with acceptable accuracy. The dates of these days in the months that heater operates, are shown in table1 [6]:

TABLE 1. DATE OF AVERAGE DAYS

January	February	March	April	October	November	December
17	16	16	15	15	14	10

II. THE MECHANISM OF HEATER

In common heaters as shown in Fig1, the flame and combustion products in burner enter to fire tube and increase its temperature. Fire tube transfers heat to water that has surrounded it. Hot water causes an increment in natural gas temperature that is in process coil. At the end, flue gases exit from the heater through the stack. It is considered about 70 BTU/h heating duty for heater for one cubic meter of station capacity.

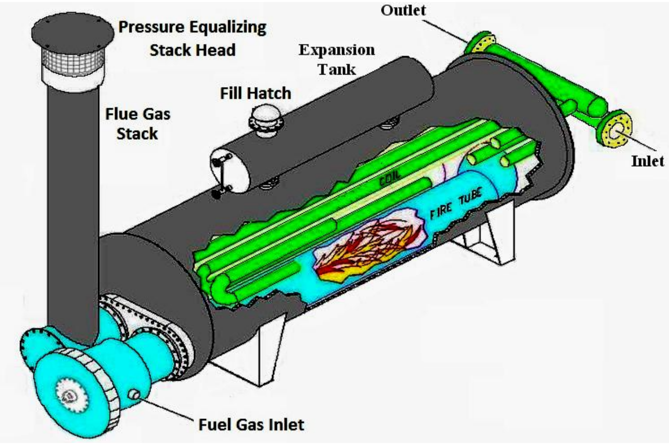

Figure 1. Schematic diagram of heater mechanism [7]

III. ENERGY BALANCE FOR INDIRECT WATER BATH HEATERS

The process of the combustion in indirect heaters is assumed to be a steady state process at a constant pressure and all gases are ideal. Natural gas is a mixture of some hydrocarbons like methane, ethane, propane, and butane. The result of gas analysis in Mahshahr station is shown in table 2 [8].

TABLE 2. VOLUMETRIC ANALYSIS OF NATURAL GAS (MAHSHAHR STATION-IRAN)

Constituent	Percent by volume
Methane (CH_4)	83.23
Ethane (C_2H_6)	10.97
Propane (C_3H_8)	3.41
Iso-Butane (C_4H_{10})	0.37
N-Butane (C_4H_{10})	0.87
Iso-pentane (C_5H_{12})	0.14
N-pentane (C_5H_{12})	0.11
Carbon dioxide (CO_2)	0.36
Nitrogen (N_2)	0.54
Hydrogen sulfide (H_2S)	4.2 ppm vol
Sodium & potassium (Na + K)	0.5 ppm vol
Calcium (Ca)	2 ppm vol
Other	2 ppm vol

Energy balance for indirect heater as a control volume is:

$$\dot{Q}_{fuel} = \dot{Q}_{losses} + \dot{Q}_{NG} + \dot{Q}_{stored} = \dot{Q}_{stack} + \dot{Q}_{surf_h} + \dot{Q}_{NG} + \dot{Q}_{stored} \quad (3)$$

At equation (3) $\dot{Q}_{fuel}$ is the heating duty of heater provided by burning natural gas as fuel. $\dot{Q}_{losses}$ is the wasting energy, including the stack and wall wasting energies ($\dot{Q}_{stack}$, $\dot{Q}_{surf_h}$). $\dot{Q}_{NG}$ is the required energy to heat NG and $\dot{Q}_{stored}$ is the energy to enhance the temperature of heat transfer environment from the minimum to maximum working temperature during a cycle [5].

A. *Required Energy for heating of natural gas flow*

Natural gas in process coils can be considered as a volume control and kinetic and potential energy of that can be supposed to be constant, so the heat transferred to natural gas is calculated according to equation 4:

$$\dot{Q}_{NG} = \dot{m}_{NG}(\mathrm{h}_{out} - \mathrm{h}_{in}) = \dot{m}_{NG}\int_{\mathrm{T_{NG-1}}}^{\mathrm{T_{NG-2}}} C_{P\,NG}\,\mathrm{dT} \quad (4)$$

Where $\dot{m}_{NG}$ is the mass flow rate, h_{out} and h_{in} are the enthalpy of output and input natural gas. By thermo-dynamical calculations the exact value of the constant-pressure specific heat of natural gas is determined as a function of temperature [1]:

$$C_{\mathrm{p_{mix}}} = \sum \mathrm{X_i} \times C_{\mathrm{p_i}} \quad (5)$$

Where parameter $\mathrm{X_i}$ is the mass fraction of components and $C_{\mathrm{p_i}}$ is its specific heat of them. The temperature of gas in pipelines is the same as the soil surrounded them. Najafi-mod et al.[9] proposed an empirical correlation to determine soil temperature from ambient temperature as follows:

$$T_{NG-1} = T_{soil} = 0.0084T_{am}^2 + 0.03182T_{am} + 11.403 \quad (6)$$

Mentioned heater is off in five months of year (May, June, July, August, September). By considering ambient temperature in these months and Putting them in to equation 6, it is clear that NG with entrance temperature about 30°C does not need preheating. Therefore in this paper T_{NG-2} has been assumed 30°C.

B. *Stack Losses*

Methane is supposed as the combustion fuel for simplicity. So the general equation of combustion reaction of this hydrocarbon fuel with air is written as:

$$\begin{aligned} &C_aH_b + A(a+b/4)\times(O_2 + 3.76N_2 + 4.76\times1.61\omega H_2O) \rightarrow \\ &aCO_2 + [b/2 + 4.76\times1.61\times A\times(a+b/4)\ \omega]H_2O + \\ &(A-1)\times(a+b/4)O + 3.76A(a+b/4)N_2 \end{aligned} \quad (7)$$

Where parameter A represents equivalence ratio and shows the percentage of excess air. The parameter ω is the amount of humidity ratio of the ambient air (combustion air). Excess air wastes energy by carrying heat up the stack. Regarding the 50% excess air and the humidity ratio of 46% (based on site condition), Calculating stack loss for heater was done:

$$\begin{aligned} &\dot{Q}_{stack}(kw) = \dot{m}_{product}(h_{out_{stack}} - h_{in_{stack}}) = \\ &\dot{m}_{product}\left(\int_{\mathrm{T_{m,o}}}^{\mathrm{T_{m,i}}} \mathrm{C}_{p_{product}}\,\mathrm{dT} + \int_{\mathrm{T_{am}}}^{\mathrm{T_{m,o}}} \mathrm{C}_{p_{product}}\,\mathrm{dT}\right) = \dot{m}_{product}\int_{\mathrm{T_{am}}}^{\mathrm{T_{m,i}}} \mathrm{C}_{p_{product}}\,\mathrm{dT} \end{aligned} \quad (8)$$

Where $\dot{m}_{product}$ refers to mass flow rate of combustion products and $T_{m,i}$, $T_{m,o}$ stand for the mean temperature at the inlet and outlet of the stack. In the process of combustion the hydrogen content of fuel is converted to H_2O, which normally leaves the stack as water vapor, carrying with it the heat required to convert it from liquid to vapor. However it can be ignored in comparison with the heat loss of flue gases.

C. Heat losses from the walls

To calculate the wasting energy from the walls of heater, temperature of inner wall is considered as the water temperature. The framework of heater is formed from steel, glass wool and aluminum. By using equation 9 and having the value of convection coefficient, the wasting from the surfaces is calculated [10]:

$$\dot{Q}_{surf_h} = \frac{T_w - T_{am}}{\dfrac{\ln(\frac{r_{o,steel}}{r_{i,steel}})}{2\pi K_{steel} L_{steel}} + \dfrac{\ln(\frac{r_{o,wool}}{r_{i,wool}})}{2\pi K_{wool} L_{wool}} + \dfrac{\ln(\frac{r_{o,al}}{r_{i,al}})}{2\pi K_{al} L_{al}} + \dfrac{1}{h_{air}.A_{\ heater}}} \qquad (9)$$

Where T_w and T_{am} are the water temperature inside the heater and the temperature of environment (K) respectively; L is length of each layer (m); K is heat transfer coefficient (W/m.K); h is heat transfer coefficient of air ($W/m^2.K$) and A is the heat transfer surface (m^2). Based on the climate condition, the average wind velocity at the station is about 6m/s [11] and the convection heat transfer coefficient (external flow) was calculated 23W/m²K [4]. The immersed coil in water bath could be considered as a pipe in constant temperature environment [2] as below [10]:

$$\frac{T_w - T_{NG-2}}{T_w - T_{NG-1}} = e^Y, Y = \frac{-\pi D_{oc} L_c U_c}{\dot{m}_{NG} C_{p_{NG}}} \qquad (10)$$

Where, D_{oc}, L_c, U_c are external diameter, length and overall heat transfer coefficient of the coil respectively. The previous studies suggest that U_c in indirect heaters is about $568W/m^2K$ [12]. T_w from the above equation determined as follows:

$$T_w = (T_{NG-2} - T_{NG-1} e^Y)/(1 - e^Y) \qquad (11)$$

D. Required energy to heat water

Assuming an one hour time period (i.e. 3600s) and constant C_{p_w} , $\dot{Q}_{stored}$ is estimated as:

$$\dot{Q}_{stored}(kW) = \int_i^{i+1} m_w C_{p_w} dT = \frac{m_w C_{p_w} (T_{w(i+1)} - T_{w(i)})}{3600} \qquad (12)$$

Where m_w and C_{p_w} are the mass and specific heat capacity of water as heat transfer environment respectively.

E. Heater efficiency

Heater was considered as a control volume and all the input and output energies of the system borders determined. So thermal efficiency is obtained using equation (13) :

heater efficiency = [(Input energy – wasted energy) / input energy] × 100 (13)

Input energy is the heating duty of the heater is provided by burning natural gas as fuel ($\dot{Q}_{fuel}$) and wasted energies are heat losses from the stack and wall. Considering thermal efficiency of the heater, one could calculate the fuel volume flow rate F_{fuel} , as below:

$$F_{fuel} = \frac{\dot{Q}_{fuel}}{\eta_b LHV} \qquad (14)$$

In which, *LHV*, is the lower heating value of the fuel (here Natural gas) and η_b is the burner combustion efficiency (80%).

IV. USING SOLAR ENERGY

In this part the main goal is to decline fuel consumption in indirect fired water bath heaters in CGS thanks to solar energy. Since these heaters have attracted a great deal of confidence in natural gas heating field, the most desirable proposed system to grow efficiency is the one that makes minimum change in their structure. The temperature of water in heater varies between 30˚C to 55˚C in different months, so the solar collector module could be interconnected in parallel flat plate array. Moreover this system has been equipped with storage tank as shown in fig2. Common heaters do not have automatic control system and heating duty should be changed with manual setting. Therefore in proposed plan, it is assumed that burner operates with constant heating duty. When the temperature of outlet gas is more than 30˚C, the solenoid valve in burner will be closed.

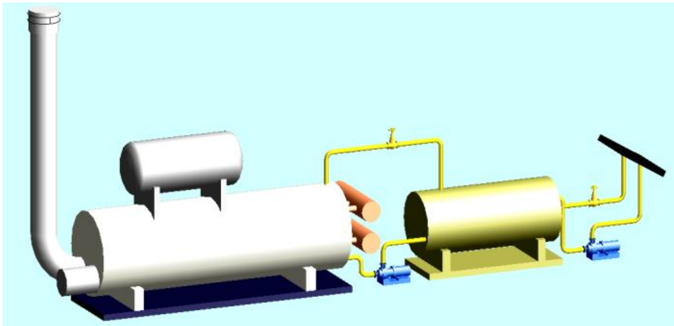

Figure 2. Schematic diagram for proposed solar system

A. Energy balance for proposed solar system

Considering fig 2, one could identify two thermodynamic systems as the storage tank and the heater. For simplicity of equations, the temperature of water in storage tank has been assumed to be constant. For storage tank the energy balance is:

$$m_{st}.C_{p_w} \frac{dT_{st}}{dT} = \dot{Q}_{solar} - \dot{Q}_{load} - \dot{Q}_{surf_{st}} \qquad (15)$$

Where m_{st} and T_{st} are the mass and temperature of water in storage tank respectively, $\dot{Q}_{solar}$ represents the useful solar energy gained by collectors, $\dot{Q}_{surf_{st}}$ is the wasting energy from the wall of storage tank that could be calculated by equation 9 and $\dot{Q}_{load}$ stands for the rate of energy transfers from the storage tank to the heater as follows:

$$\dot{Q}_{load} = \dot{m}_w C_{p_w} (T_{st} - T_w) \tag{16}$$

In which $\dot{m}_w$ is the mass flow rate between storage tank and heater. Since the length of storage tank to obtain $\dot{Q}_{surf_{st}}$ is unknown, it could be follower of mass or volume of water in storage tank (equation17) that will be optimized.

$$\rho_w = \frac{m_{st}}{V_{st}} = \frac{m_{st}}{A_{st} L_{st}} \ , \quad L_{st} = \frac{m_{st}}{\rho_w . A_{st}} \tag{17}$$

Where ρ, V, A, L and m are density, volume, area, length and mass of water in storage tank. For second control volume (heater) energy balance is:

$$m_w C_{p_w} \frac{dT_w}{dt} = \dot{Q}_{fuel} + \dot{Q}_{load} - \dot{Q}_{NG} - \dot{Q}_{stack} - \dot{Q}_{surf_h} \tag{18}$$

B. *Available solar energy*

Solar energy calculation is carried out by empirical equations. In this paper, equation represented by Duffie [13] has been used to calculate solar energy in Mahshahr. These correlations are shown as table 3:

TABLE 3. EQUATIONS USED TO CALCULATE SOLAR ENERGY

$I_T = I_b R_b + I_d \left(\frac{1+\cos\beta}{2}\right) + I \rho_g (\frac{1-\cos\beta}{2})$	(19)
$R_b = \frac{\cos\theta}{\cos\theta_z}, \cos\theta = \sin\delta \sin\varphi \cos\beta - \sin\delta \cos\varphi \sin\beta \cos\gamma +$ $\cos\delta \cos\varphi \cos\beta \cos\omega + \cos\delta \sin\varphi \sin\beta \cos\gamma \cos\omega + \cos\delta \sin\beta \sin\gamma \sin\omega$	(20), (21)
$\cos\theta_z = \cos\varphi \cos\delta \cos\omega + \sin\varphi \sin\delta$	(22)
$\omega = (Solar\ Time - 12) \times 15^{\circ}$	(23)
$Solar\ Time = standard\ time + 4(L_{st} - L_{loc}) + E$	(24)
$E = 229.2(0.000075 + 0.001868\cos B -$ $0.032077\sin B - 0.014615\cos 2B - 0.04089\sin 2B)$	(25)
$B = (n-1)\frac{360}{365}$, $\delta = 23.45\sin(360\frac{284+n}{365})$	(26), (27)
$I = I_0\left[a_0 + a_1 e^{(\frac{-k}{\cos\theta_z})}\right]$, $\begin{cases} a_0 = 0.2538 - 0.0063(6-A)^2 \\ a_1 = 0.7678 + 0.0010(6.5-A)^2 \\ k = 0.249 + 0.081(2.5-A)^2 \end{cases}$	(28), (29)
$I_0 = \frac{12\times3600}{\pi} G_{sc}\left(1 + 0.033\cos\frac{360n}{365}\right)\times$ $\left[\cos\varphi \sin\delta(\sin\omega_2 - \sin\omega_1) + \frac{\pi(\omega_2-\omega_1)}{180}\sin\varphi \sin\delta\right]$	(30)
$\frac{I_d}{I} = \begin{cases} 1.0 - 0.249k_T\ for 0 \le k_T \le 0.35 \\ 1.557 - 1.84k_T\ for 0.35 < k_T < 0.75, k_T = \frac{I}{I_0} \\ 0.177\ for k_T > 0.75 \end{cases}$	(31)

I_T : Total radiation	L_{loc} : Longitude of location
I_b : Direct Normal Radiation	E : Equation of time
R_b : Ratio of beam radiation	B : constant
I_d : Diffuse radiation	I_0 : Extraterrestrial radiation
I : Reflected radiation	a_0 :constant
ρ_g : Reflection coefficient	a_1 :constant
β: Slope	k : constant
θ : Angle of incidence	A :altitude (km)
θ_z: Zenith angle	G_{sc} : Solar constant
φ: Latitude	k_T : Hourly clearness index
ω: Hour angle	n : Number of day in year
δ: Surface azimuth angle	γ : surface azimuth angle
L_{st} : Standard meridian	

Geographic information and environmental conditions in order to use in above equations and calculate the amount of solar energy at mentioned location are shown in table 4. Furthermore, heater specification has been represented in table5.

TABLE 4. GEOGRAPHIC CONDITION AND COLLECTOR INSTALLATION STATUS

Parameter	Value
φ	30˚
Lst	49˚
A	3m
β	30˚
γ	0˚
Gsc	1367 w/m2
ρg	0.6

TABLE 5. HEATER SPECIFICATION IN MAHSHHR CGS

Diameter of fire tube	30in
Water capacity	34m3
Number of coils	4
Length of a coil	8
Shell length	8.35m
Shell diameter	2.4m
Heater capacity (NG)	125000 m3/h

The useful energy gained by a flat plate collector could be calculated as:

$$Q_{sol} = \eta_{coll} I_T A_{coll} \tag{32}$$

Where A_{coll} is the area of all collectors (The area of chosen collector is $1.9m^2$).The collector efficiency obtains from equation 33 [2]:

$$\eta_{coll} = 0.78 - 1.4\frac{(T_{st} - T_{am})}{I_T} - 0.09\frac{(T_{st} - T_{am})^2}{I_T} \tag{33}$$

By integration of equation 15 in terms of time, the new water temperature in storage tank is calculated as follows:

$$T_{st,new} = T_{st,old} + \frac{\Delta t}{(mC_p)_{st}}[\dot{Q}_{solar} - \dot{Q}_{load} - \dot{Q}_{surf_{st}}] \tag{34}$$

In proposed system, it is assumed that whole water in storage tank is circulated between heater and storage tank for one time in a day. It means:

$$m_{st} = \dot{m}_w \times 3600 \times 24 \tag{35}$$

V. THE EFFECT OF PREHEATING AIR COMBUSTION AND HEAT RECOVERY

The temperature of flame is one of the important factors to raise heat transfer from flame to fire tube, water and finally natural gas in coils. One of the methods for using the energy of hot exhaust flue gases is to use them to preheating the fuel or oxidant in order to increase the adiabatic temperature of flame.

However, preheating oxidant (air combustion) due to much mass flow rate in comparison to fuel has more influence on combustion efficiency. The processes that lead to heat natural gas in indirect fired water bath heaters have been modeled to investigate this issue (Fig 3). In this model energy balance has been observed as equation 3.

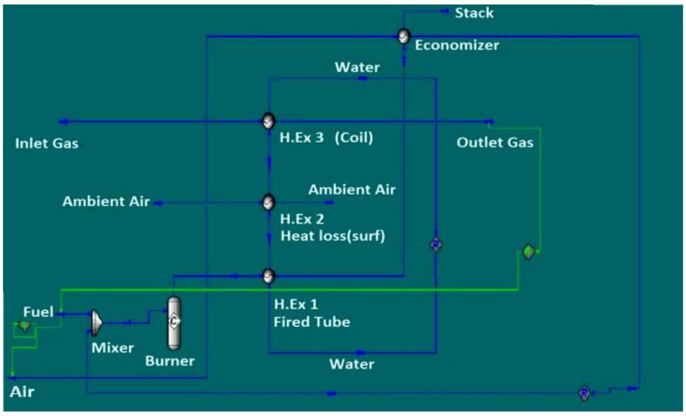

Figure 3. Schematic model of heating the natural gas in heaters

As a result, combustion air that is the same as ambient temperature can be heated by hot flue gases thanks to a heat exchanger. Recycle burners could be employed instead of common atmospheric burners in heaters. These burners have been designed in such a way that guide flue gases to a double pipe heat exchanger by a channel. In this heat exchanger, combustion air flows in inner pipe and flue gases are in outer pipe. Hot exhaust gases transfer part of their heating duty to entered air for combustion and then go out from stack (Fig4).

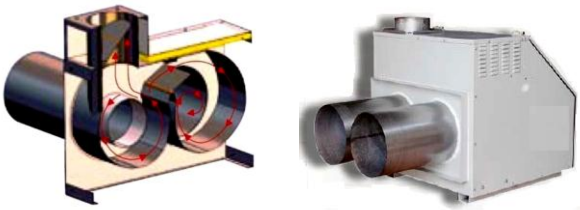

Figure 4. Proposed burner with capability of recycling [14]

Another way to use heat losses from the stack is to apply them in steam turbine in order to generate power. If flue gases from the stack of indirect heaters could be considered as the same as flue gases from gas turbine, by using a heat exchanger and transferring energy from flue gases to water to produce steam, considerable power could be generated from the generator that connected to steam turbine. Steam converts to water by a condenser after passing through steam turbine and the cycle will be closed. Schematic diagram of above description is shown in Fig 5.

In this part the above cycle has been modeled in Fig 6. Composition, temperature and mass flow rate of exhaust gases from hypothesis gas turbine are the same as exhaust gases from heater in different months.

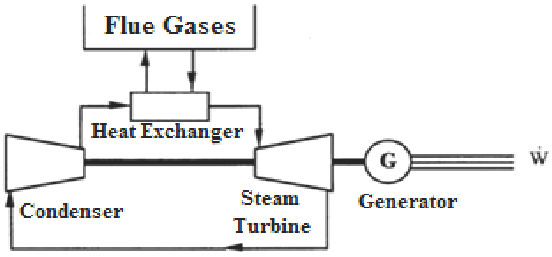

Figure 5. Schematic diagram of generating power from flue gases [15]

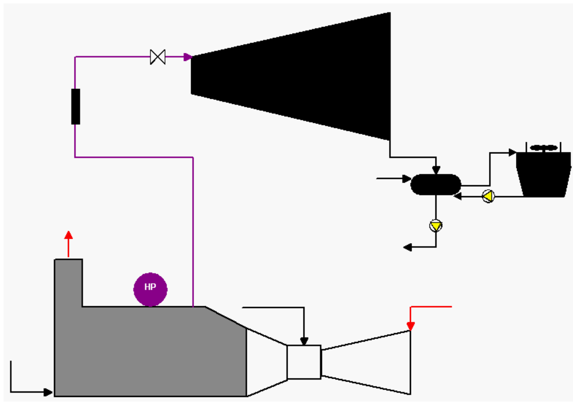

Figure 6. Schematic model for generating power

VI. RESULT AND DISCUSSION

The values of $\dot{Q}_{NG}$, $\dot{Q}_{stack}$, $\dot{Q}_{surf_h}$ and $\dot{Q}_{stored}$ has been represented in table 6. Contents of energy used to heat NG and heat losses of flue gases is much more than energy used to heat water and heat losses from the walls.

TABLE 6. HEATING ENERGY IN DIFFERENT PARTS *(KWH)*

	$\dot{Q}_{NG}$	$\dot{Q}_{stack}$	$\dot{Q}_{surf_h}$	$\dot{Q}_{stored}$
Jan	660.17	607.21	5.91	18.33
Feb	611.35	557.63	5.40	19.96
Mar	474.62	423.21	4.02	18.60
Apr	108.85	93.89	0.71	14.38
Oct	196.51	168.99	1.45	22.94
Nov	490.36	438.01	4.17	17.32
Dec	621.65	563.40	5.48	13.25

Mean efficiency of heater during the months that heater operates has been illustrated in Fig 7. As it is clear, heater efficiency is directly proportional to ambient temperature. However heater efficiency does not exceed 54% and it means approximately half of heating energy provided by burning natural gas in burner will be wasted.

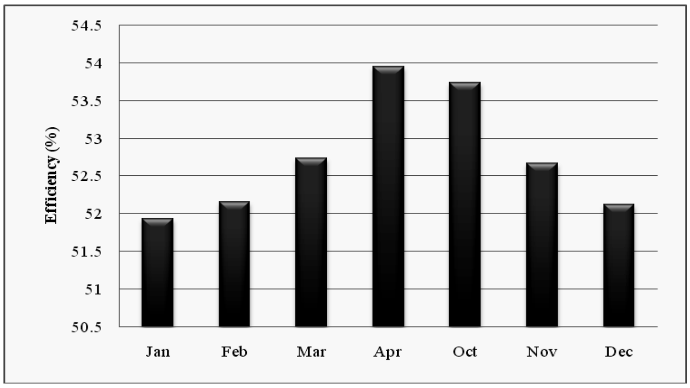

Figure 7. Mean efficiency of heater

Total radiated solar flux and average hourly on slopped collector in 7 months could be seen in Fig 8, 9 respectively. As it is expected solar energy in warmer months is more available like April. Since in warmer months, the temperature of natural gas is relatively high due to ambient temperature, required energy to heat NG is relatively less and solar energy does not have expected effect on fuel saving.

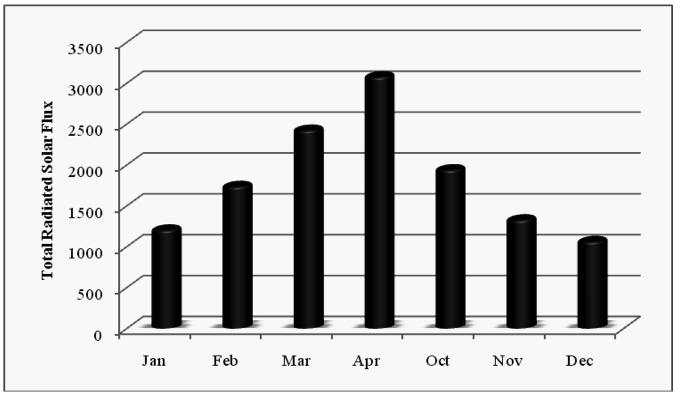

Figure 8. Total radiated solar flux in Mahshahr

In addition, solar energy at noon is more than other hours of day just the same as the ambient temperature.

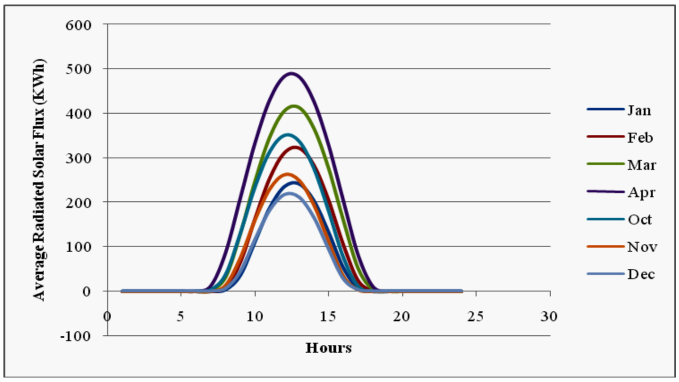

Figure 9. Average hourly radiated solar flux

The rate of energy transferred from the storage tank to heater, $\dot{Q}_{load}$, in one hour intervals is displayed in Fig 10.

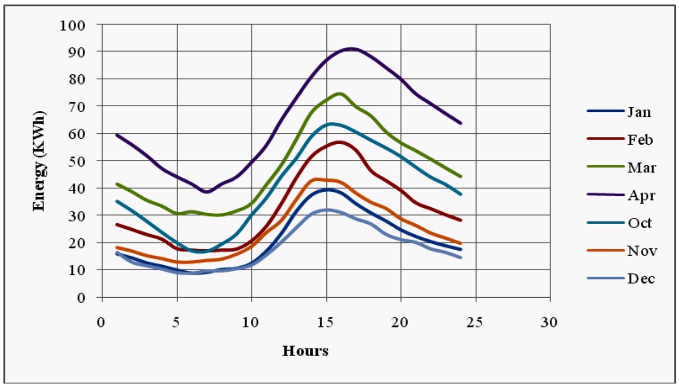

Figure 10. Average rate of transferred energy

In early hours of morning, energy transferred between heater and storage tank is lowest. The capacity of storage tank in order to gain maximum saving in fuel consumption was obtained about 90 liter per one collector. Average hourly temperature of water in storage tank is depicted in Fig 11.

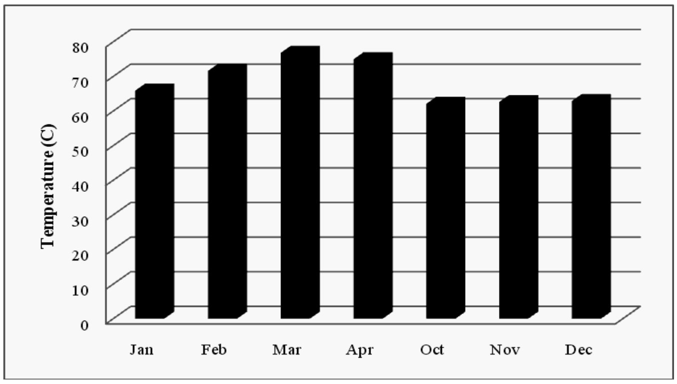

Figure 11. Average hourly temperature of water in storage tank

Fig 12 shows a comparison heating duty of heater when solar system is and is not utilized. Averagely 60kwh required energy will decrease in one hour intervals when CGS equipped with solar system.

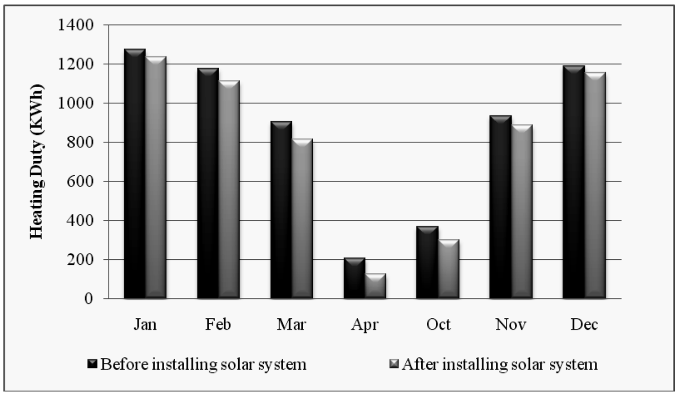

Figure 12. Heating duty comparison before and after installing solar system

Considering Fig 13, one could select 370 as an optimum value for the number of collector modules.

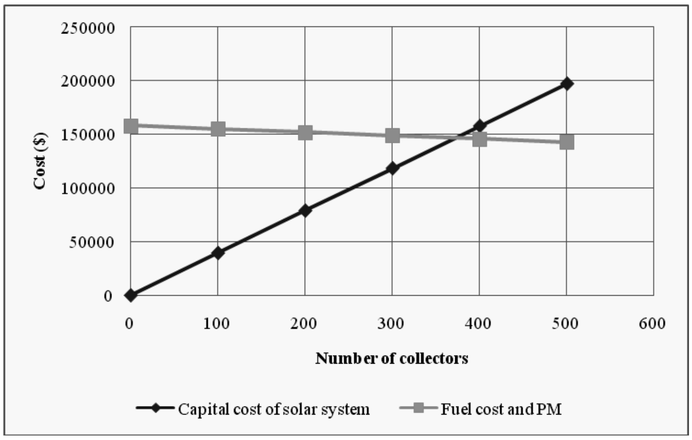

Figure 13. Optimum value for the number of collectors

Consequently, monthly fuel saving in Mahshahr CGS with solar system is shown in Fig 14.

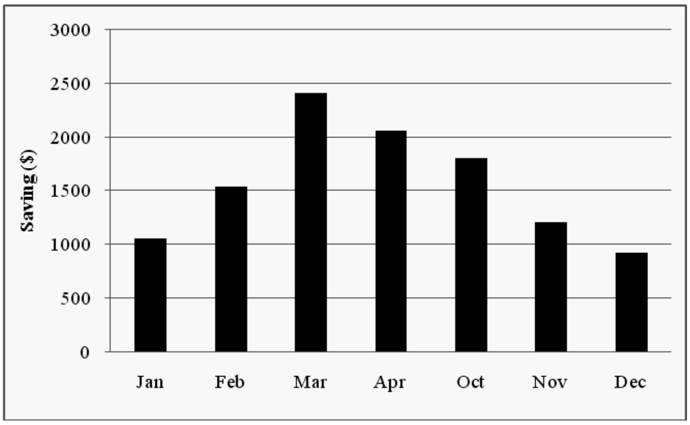

Figure 14. monthly fuel saving thanks to solar system

The cost analysis of the proposed solar system has been presented in table 7. The total capital cost is 146150 USD and the cost effectiveness analysis shows that annual benefit is about 10947 USD. Therefore, the payback period is calculated as 18 years.

TABLE 7. COST ANALYSIS FOR SOLAR SYSTEM [2]

The capital cost for 370 collectors	66600 US$
The capital cost for the storage tank	50000 US$
Total capital cost for the system installation	29550 US$
Annual O&M costs	2900 US$
Cost of natural gas per cubic meter	0.28 US$
Annual fuel saving	10947 US$

Considering equation 36, net present value (NPV) for the proposed solar system with 7% interest rate is shown in Fig 15. NPV will not be positive after 30 years and this could show that proposed system is not economical.

$$NPV = \frac{R_t}{(1+i)^t} \tag{36}$$

Where t is the time of the cash flow, i is the discount rate and R_t is the net cash flow. Internal rate of return (IRR) for this project is determined 3.6%. It is clear that interest rate especially in Iran is much more than 3.6%.

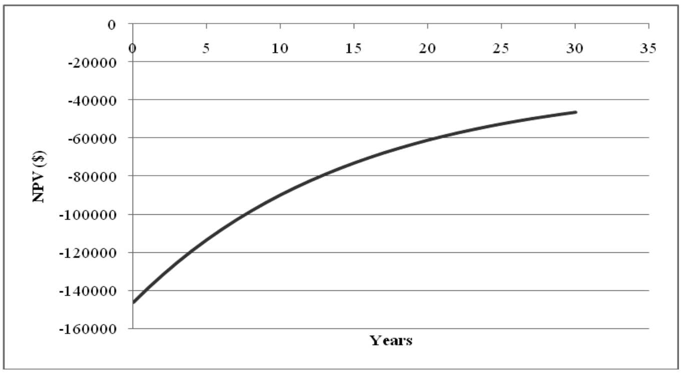

Figure 15. NPV for proposed solar system

Fig 16 illustrates heat losses from the stack in comparison with heating duty of heater that is provided by burning natural gas as fuel.

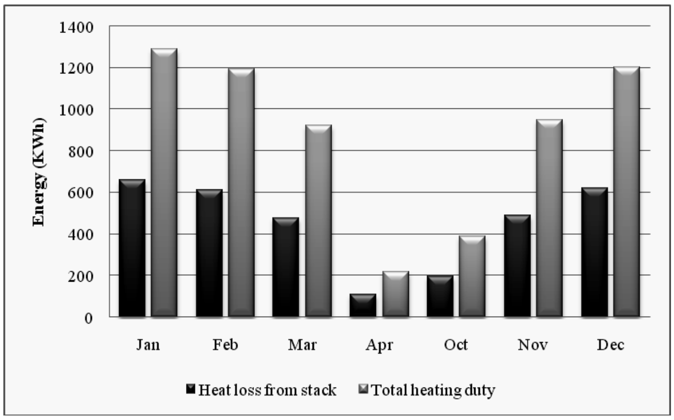

Figure 16. Heat losses from the stack in comparison with heating duty of heater

$\dot{Q}_{stack}$ is about 46% and $\dot{Q}_{surf_h}$ is 1% of $\dot{Q}_{fuel}$. The changes in fuel consumption due to preheating the air combustion are shown in Fig 17.

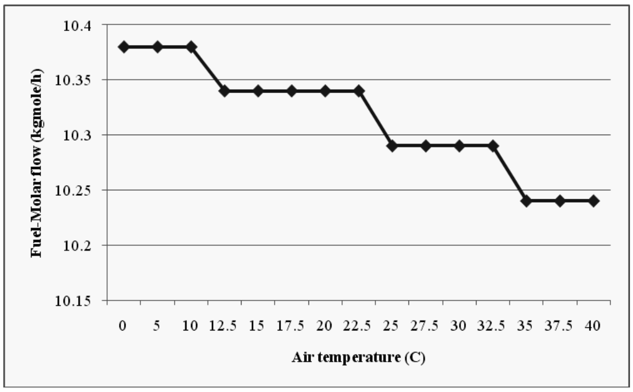

Figure 17. Changes in fuel consumption due to preheating air combustion

In a particular case study, by increasing the temperature of air combustion about 40˚C, fuel consumption is decreased about 0.15*Kgmole/h* while T_{NG-2} has been fixed.

Mean power energy that could be reached from the heating energy of flue gases is displayed in table 8 for the months that heater operates. Considering the proposed system as a 50*kW* power plant, capital and variable costs of the main parts is shown in table 9.

TABLE 8. MEAN POWER ENERGY REACHED FROM THE HEATING ENERGY OF FLUE GASES

Month	Mean ambient temperature (˚C)	$\dot{m}_{stack}$ (t/h)	Mean generated power (kW)
Jan	12.2	4.08	51.1
Feb	14	3.76	46.65
Mar	18.8	2.89	35
Apr	29.2	0.65	0
Oct	26.8	1.17	13.85
Nov	18.3	2.99	36.29
Dec	13.8	3.8	47.19

TABLE 9. CAPITAL AND VARIABLE COSTS OF PROPOSED POWER PLANT

	Steam turbine	Generator	Heat exchanger	Cooling system
Capital cost	630 ($/kW)	230 ($/kW)	80 ($/kW)	45 ($/kW)
Annual O&M costs	3.75 ($/kW)	2.3 ($/kW)	1.6 ($/kW)	1.35 ($/kW)
Lifetime	20 years	20 years	30 years	30 years

VII. CONCLUSION

In this paper as a case study namely Mahshahr CGS was considered. The efficiency of heater installed in this station calculated about 52%. Heat losses from the stack of heater identified as the main factor for this low efficiency. At first, a solar system has been proposed to provide part of heat demand. The results show that the system can reduce the heating duty of heater about 652GJ and 39000m^3 of fuel can be saved in a year. Payback period for this plan is about 18 years while NPV will not be positive even after 30 years. The main reason for this non profitability is the climate condition of CGS considered as case study. Mahshahr has been placed in a warm zone of Iran and the heater is off for 5 months. Therefore the proposed system has no benefit in these 5 months. One of the methods for using the energy of flue gases, is to apply them for preheating the air combustion. Increasing the temperature of air combustion causes growing heater efficiency and reducing fuel consumption. Moreover, energy of hot exhaust gases could generate considerable amount of power. In this case study proposed system generates 168MWh power in a year. This power can be used in other parts of station or sold to government about 0.1 USD/MWh. The system can be considered as a power plant that payback period is calculated about 3 years.

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NOMENCLATURE

T_{am}	Ambient temperature (K)
T_w	Water temperature in heater (K)
T_{st}	Water temperature in storage tank (K)
T_{NG-1}	Natural gas temperature at entrance (K)
T_{NG-2}	Natural gas temperature at exit (K)
A	Equivalence ratio
Q_{NG}	Energy for heating NG flow (kWh)
Q_{fuel}	Heating duty of heater (kWh)
Q_{stack}	Heat losses from stack (kWh)
Q_{stored}	Energy for heating water (kWh)
Q_{surf}	Heat loss from the walls (kWh)
Q_{sol}	Solar energy absorbed by collectors (kWh)
Q_{load}	Transferred energy between heater and tank (kWh)
Cp	constant pressure heat capacity (kJ/kg.K)
$\dot{m}$	Mass flow rate (kg/h)

Greek symbols

η	Efficiency (%)
ω	Humidity ratio (%)
μ_{JT}	Joule-Thompson coefficient
ρ	Density (kg/m^3)