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Study of Preheating Natural Gas in Gas Pressure Reduction Station by the Flue Gas of Indirect Water Bath Heater

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Abstract- In most gas pressure reduction stations, natural gas pressure is reduced through the throttle valve. In regulators due to the Joule-Thompson effect, pressure drop causes temperature reduction. To refuse obstruction of pipelines, indirect water bath heaters warm flow natural gas before passing through regulator. In this paper Mahshahr CGS elected as a case study and thermal efficiency of heater installed in this station computed briefly. Results illustrate that heat losses from stack is the main factor in low efficient heater. To use the thermal energy of flue gases, a shell and tube heat exchanger is proposed to preheat inlet natural gas to heater. The specification of heater is calculated by considering some assumptions at first. Then the effect of using heat exchanger on fuel consumption in atmospheric burner of heater is studied. Results shows that using a heat exchanger can reduce total thermal needed energy from hater 1.3MJ in a year. This means that more than 141000 m^3 fuel is saved and it causes a significant economic saving and environmental pollution decreasing. Payback period of this proposal is 1.2 years and net present value with interest rate of 14% is positive after 2 years. Therefore by using a simple heat exchanger the efficiency of indirect water bath heater improves 11%.

Keywords- *Indirect water bath heater; Heat exchanger; Heat recovery from product combustion;Preheating inlet gas .*

I. INTRODUCTION

Natural gas is extracted and transported in high pressure, but consumers need it in lower pressure and this is the necessity of constructing gas pressure reduction stations. Figure 1 has depicted the natural gas transportation system. AGA-8 equation of state describes the relation of inlet pressure to regulator and the content of temperature drop precisely. This process is adiabatic, steady state and Irreversible. Indirect water bath heaters installed before the regulators to increase natural gas temperature and prevent hydrated formation. Since the indirect heaters are the most consumers with low efficiency in gas pressure reduction station, optimization of these devices is one of the main goals in countries with high level of natural gas reserves like Iran.

In this regard Khalili et al. [1] collected some data such as natural gas flow rate, inlet and outlet temperature of natural gas, fuel consumption of heater and temperature of inlet and outlet combustion product through the stack. Then he concluded that heater efficiency in Shahrekord station is not more than 47%. Riahi et al. [2] optimized the combustion efficiency in indirect water bath heaters of ardabil city gate stations. Results showed that regulation of burner and using barometric damper can decrease heat losses and increase efficiency. In case of feasibility studying of accompanying uncontrolled linear heater with solar system Farzaneh-Gord et al. [3] proposal could reach to saving of 100kWh energy and 27000 USD. None of the past studies use thermal energy of flue gases to preheat inlet natural gas. This paper presents a new solution that could be applied in all gas pressure reduction station.

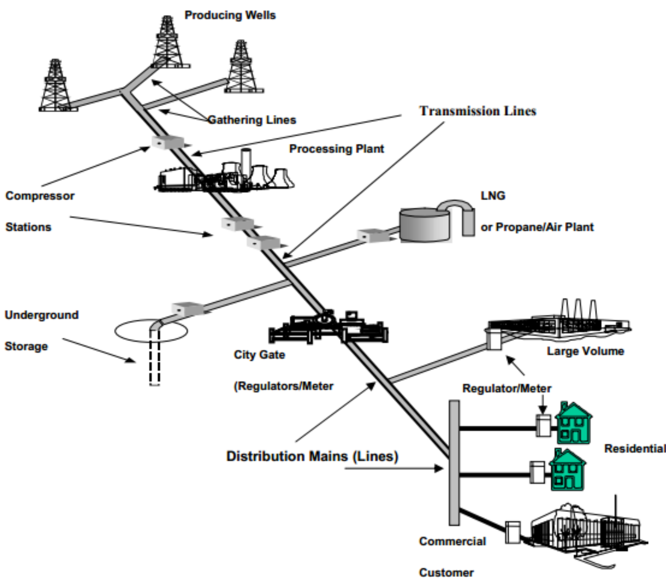

Figure 1. Natural gas transportation system[4]

II. THE PERFORMANCE OF HEATER

Figure2 shows the schematic of indirect heater operation. Natural gas should not be exposed to direct flame. Thermal energy generated in burner is transferred to natural gas stream by a medium (water) indirectly. This is the reason that this type of heaters called indirect heaters.

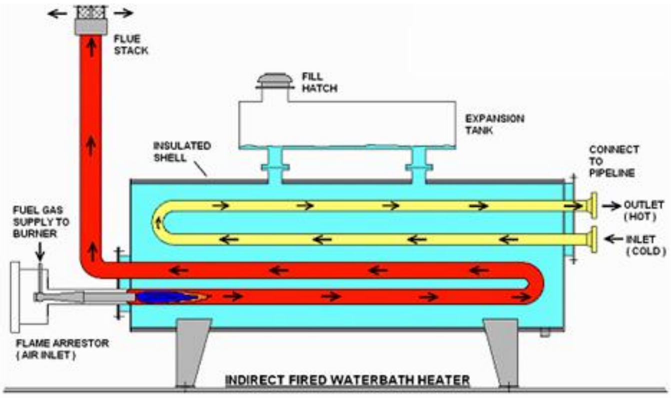

Figure 2. Schematic diagram of heater performance [5]

III. THE EFFICIENCY OF HEATER BEFORE USING HEAT EXCHANGER

Considering the water bath heater as a control volume, input energy is the thermal energy generated by burning fuel in burner and wasted energy is the heat loss from stack and surface of heater. Therefore thermal efficiency of heater is calculated as blow [2]:

Heater efficiency = [(Input energy – wasted energy) / input energy] × 100 (1)

Input energy represented by $\dot{Q}_{fuel}$ and the equations due to calculate effective parameters in equation (1) is depicted in table 1.

IV. USING SHELL AND TUBE HEAT EXCHANGER

The main target of all researches on indirect heaters is to decrease fuel consumption in these devices. In this regard the proposed design of using heat exchanger to recover thermal energy of flue gas has shown in Figure3. By starting of burner flu gas will pass through the stack. Regarding to figure1 stack should be removed and flu gas should come to heat exchanger. Natural gas passes through the exchanger before going to heater on the other side. In heat exchanger natural gas with high pressure assumed to be in tube and flue gas in shell. After heat transferring from hot combustion product (400˚C) to cold natural gas, flue gas discharged to ambient with lower temperature and preheated natural gas enters to indirect water bath heater.

Heater installed in mahshahr city gate station operates in seven months. Results shows in five other months of the year that heater is off the temperature of natural gas is more than 30˚C. Therefore heater applied to raise the temperature of natural gas to 30˚C [6]. Using heat exchanger causes natural gas enters heater with higher temperature. Since the heat transfer rate and fuel consumption of heater depends on ΔT between inlet and outlet temperature of natural gas, by increasing the inlet temperature, ΔT, heat transfer rate and fuel consumption will decline.

TABLE 1. EQUATIONS USED TO CALCULATE HEATER EFFICIENCY [7] ,[8]

$$\dot{Q}_{fuel} = \dot{Q}_{losses} + \dot{Q}_{NG} + \dot{Q}_{stored} = \dot{Q}_{stack} + \dot{Q}_{surf_h} + \dot{Q}_{NG} + \dot{Q}_{stored} \tag{2}$$

$$\dot{Q}_{NG} = \dot{m}_{NG}\,(\mathrm{h}_{out} - \mathrm{h}_{in}) = \dot{m}_{NG} \int_{\mathrm{T}_{NG-1}}^{\mathrm{T}_{NG-2}} C_{P\,NG}\,\mathrm{dT} \tag{3}$$

$$C_{\mathrm{p}_{mix}} = \sum \mathrm{X}_{\mathrm{i}} \times C_{\mathrm{p}_{\mathrm{i}}} \tag{4}$$

$$T_{NG-1} = T_{soil} = 0.0084 T_{am}^{2} + 0.03182 T_{am} + 11.403 \tag{5}$$

$$\dot{Q}_{stack}\left(kw\right) = \dot{m}_{product}\,(h_{out_{stack}} - h_{in_{stack}}) = \dot{m}_{product}\,\Big(\int_{\mathrm{T}_{m,o}}^{\mathrm{T}_{m,i}} \mathrm{C}_{p_{product}}\,\mathrm{dT} + \int_{\mathrm{T}_{am}}^{\mathrm{T}_{m,o}} \mathrm{C}_{p_{product}}\,\mathrm{dT}\Big) = \dot{m}_{product} \int_{\mathrm{T}_{am}}^{\mathrm{T}_{m,i}} \mathrm{C}_{p_{product}}\,\mathrm{dT} \tag{6}$$

$$\dot{Q}_{surf_h} = \frac{T_w - T_{am}}{\dfrac{\ln(\frac{r_{o,steel}}{r_{i,steel}})}{2\pi K_{steel} L_{steel}} + \dfrac{\ln(\frac{r_{o,wool}}{r_{i,wool}})}{2\pi K_{wool} L_{wool}} + \dfrac{\ln(\frac{r_{o,al}}{r_{i,al}})}{2\pi K_{al} L_{al}} + \dfrac{1}{h_{air}.A_{heater}}} \tag{7}$$

$$\frac{\mathrm{T_w - T_{NG-2}}}{\mathrm{T_w - T_{NG-1}}} = \mathrm{e^{Y}}, \mathrm{Y} = \frac{-\pi \mathrm{D_{oc} L_c U_c}}{\dot{\mathrm{m}}_{\mathrm{NG}} \mathrm{C}_{\mathrm{p_{NG}}}} \tag{8}$$

$$\mathrm{T_w = (T_{NG-2} - T_{NG-1} e^{Y}) / (1 - e^{Y})} \tag{9}$$

$$\dot{Q}_{stored}\,(kW) = \int_{i}^{i+1} \mathrm{m_w C_{p_w}}\,\mathrm{dT} = \frac{\mathrm{m_w C_{p_w} (T_{w(i+1)} - T_{w(i)})}}{3600} \tag{10}$$

$$F_{fuel} = \frac{\dot{Q}_{fuel}}{\eta_b LHV} \tag{11}$$

$$\dot{m}_{fuel} + \dot{m}_{air} = \dot{m}_{product} \tag{12}$$

$$\dot{Q}_{surf} + \dot{Q}_{NG} + \dot{Q}_{stored} + \dot{m}_{Product} \int_{\mathrm{T}_{am}}^{\mathrm{T}_{m,i}} \mathrm{C}_{p_{product}}\,\mathrm{dT} = F_{fuel}.\eta_b.LHV \tag{13}$$

$$\dot{m}_f = \frac{F_{fuel}.\rho_{methane}}{3600} \tag{14}$$

$$\dot{n}_{fuel} = \frac{\dot{m}_f}{M_{methane}} \tag{15}$$

$$\dot{m}_{air} = \dot{n}_{air} \times M_{air} \tag{16}$$

$\dot{Q}_{fuel}$: Heating duty of heater provided by burning natural gas as fuel
$\dot{Q}_{NG}$: Required energy to heat NG
$\dot{Q}_{stack}$: Energy losses from stack
$\dot{Q}_{surf_h}$: Energy losses from the surface
$\dot{Q}_{stored}$: Energy to enhance the temperature of heat transfer environment
$\dot{m}_{NG}$: Mass flow rate of NG
h_{in} : Enthalpy of input natural gas
h_{out} : Enthalpy of output natural gas
X_i : Mass fraction of components
C_{p_i} : specific heat
F_{fuel}: volumetric flow rate
LHV: Lower heating value
M: Molecular weight
$\dot{m}_{product}$: Mass flow rate of combustion products
$T_{m,i}$: Mean temperature at the inlet of the stack
$T_{m,o}$: Mean temperature at the outlet of the stack
T_w: water temperature inside the heater
T_{am}: Temperature of ambient
K: Thermal conductivity
A: Heat transfer surface
h: Convective Heat transfer coefficient
D_{oc}: External diameter of coil
L_c: Length of coil
U_c: Total heat transfer coefficient in indirect heaters
m_w: Mass of water in heater
η_b : burner combustion efficiency
$\dot{n}_{fuel}$:Fuel molar flow rate

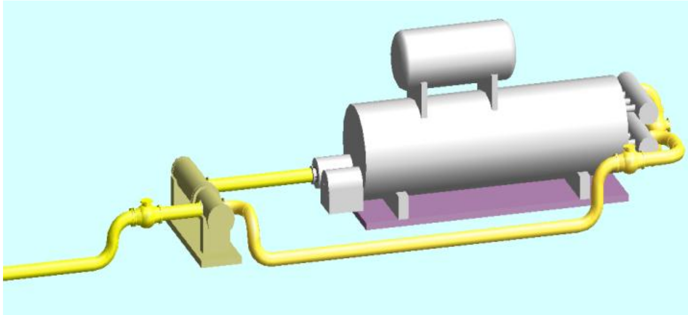

Figure 3. Schematic diagram for proposed system [9]

Shell and tube heat exchanger consists of a bundle of tubes held in a cylindrical shape by plates at either end called tube sheets. The tube bundle placed inside a cylindrical shell. The size of the exchanger is defined as the total outside surface area of the tube bundle .To obtain the specification of needed heat exchanger some effective parameters on heat exchanger operating should be assumed. Then the exact content of them will be calculated with trial and error. The first hour of the first day of the year is chosen as the basis. It is one of the coldest days in the year. Considering $\dot{m}_{NG}$, $\dot{m}_{product}$, $T_{\text{NG}-1}$ (as inlet cold fluid), $T_{\text{product}-1}$ (as inlet hot fluid), $C_{\text{p}_{\text{NG}}}$ and $C_{\text{p}_{\text{product}}}$, to increase $T_{\text{NG}-2}$ until 22˚C , $T_{\text{product}-2}$ is computed 220˚C. In this regard hypothesis is as blow table:

TABLE 2. ASSUMPTION TO PREDICT HEAT EXCHANGER SPECIFICATION

Symbol	Definition	Content
d_{it}	Tube internal diameter (m)	0.097
d_{ot}	Tube outside diameter (m)	0.103
R_{ft}	Fouling resistance (m^2.K/W)	0.000176
K	Tube thermal conductivity (W/m.K)	60
PR	Pitch ratio	1/25
WS	Wall shear	25%
CL	Tube count calculation constant	1
CTP	Tube layout constant	0.93
h_o	Shell Convective Heat transfer coefficient (W/m^2.K)	80
h_i	Tube Convective Heat transfer coefficient (W/m^2.K)	450
L	Tube length (m)	3
F	Correction factor	0.9

Above measures on initial parameters led to the following conclusions have been illustrated in table3.

TABLE 3. OBTAINED RESULTS FOR HEAT EXCHANGER SPECIFICATION

Symbol	Definition	Content
U	Total heat transfer coefficient (W/ m^2.K)	66.2
A	Area (m^2)	20
D_s	Shell Diameter (m)	0.7
N_T	Number of tunes	22

If there is no change in fluid phases and specific heat assumed to be constant, heat transfer rate in a heat exchanger will be as below [10]:

$$Q = \left(\dot{m}c_p\right)_h (T_{h1} - T_{h2}) \tag{17}$$

$$Q = \left(\dot{m}c_p\right)_c (T_{c2} - T_{c1}) \tag{18}$$

Temperature difference between hot and cold fluids $(\Delta T = T_h - T_c)$ changes with displacement in heat exchanger. Therefore in heat transfer analysis an average content for temperature difference between hot and cold fluid is defined as mean temperature difference [10]:

$$Q_{hex} = UA\Delta T_m \tag{19}$$

Heat exchangers surfaces are always exposed to fouling. Fouling layer causes an increase in thermal resistance between two fluids. This effect considered with another thermal resistance called fouling factor R_f inverse of total heat transfer coefficient represents R_0 as total thermal resistance. R_0 is sum of all thermal resistances which are connected in series as follows [10]:

$$R_t = R_h + R_{h,f} + R_w + R_{c,f} + R_c \tag{20}$$

R_h= Thermal resistance of hot side = $\dfrac{1}{(\eta_o hA)_h}$

R_c= Thermal resistance of cold side= $\dfrac{1}{(\eta_o hA)_c}$

$R_{h,f}$= fouling resistance at hot side

$R_{c,f}$= fouling resistance at cold side

R_w= Wall thermal resistance= $\dfrac{\ln(\frac{d_0}{d_i})}{2\pi k_w L N_l}$

In which N_l is the number of tube and calculated as blow:

$$N_t = \frac{A}{\pi . d_o . L} \tag{21}$$

Therefore total heat transfer coefficient obtained from equation (22):

$$U_0 = \frac{1}{\frac{r_o}{r_i}\frac{1}{h_i} + \frac{r_o}{r_i}R_{fi} + \frac{r_o \ln(\frac{r_o}{r_i})}{K_w} + R_{fo}\frac{1}{h_o}} \tag{22}$$

V. RESULT AND DISCUSSION

The result of proposed plan for heat recovery of thermal energy of flue gas is represented in this part.

Refer to Fig 4, average T_{NG-1} will increase about 2.3˚C among the year by applying the heat exchanger. This leads to decline in man content of energy dedicated to raise natural gas temperature as Fig 5.

$\dot{Q}_{NG}$ will decrease 25% after using heat exchanger to apply thermal energy of flue gas.

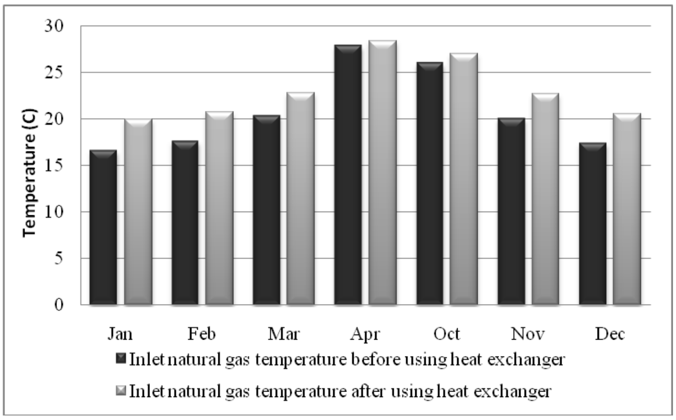

Figure 4. Mean inlet natural gas temprature

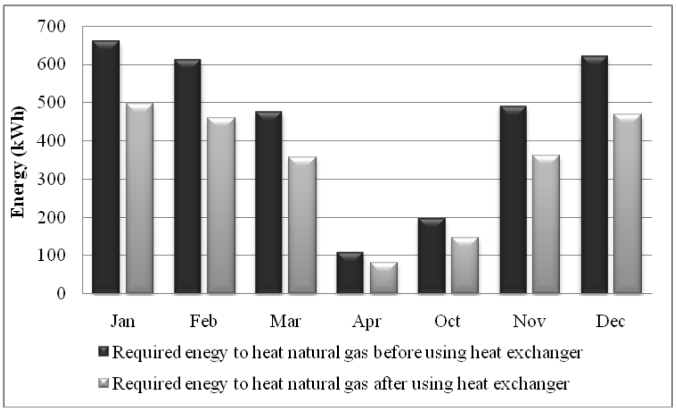

Figure 5. Required energy for heating natural gas

Fig 6 shows Mass flow rate of combustion products will be reduced about 946kg in a year and the consequence of environmental pollution due to discharged flue gas will decline.

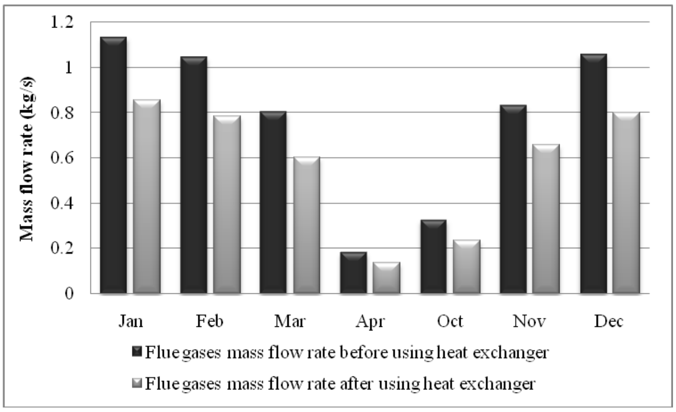

Figure 6. Mass flow rate of combustion products

Fig7 illustrates that heat loss due to exited combustion products will be reduced 24% by apply it in heat exchanger.

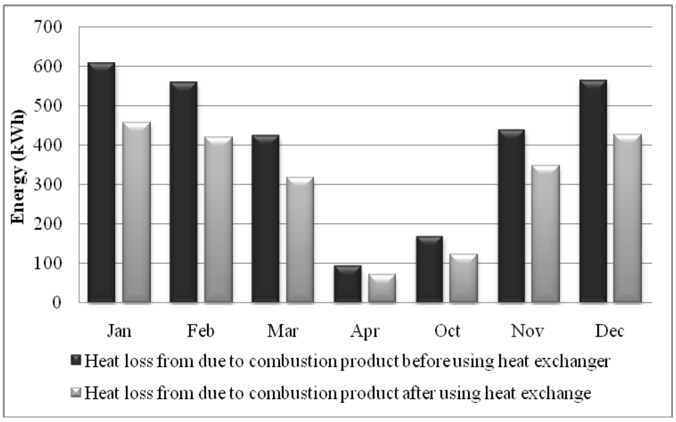

Figure 7. Heat loss due to combustion product

New content of heater efficiency is calculated by equation (23). Therefore average efficiency of heater increase 11% as showed in Fig8.

$$\eta_{heater} = \frac{\dot{Q}_{NG+}\dot{Q}_{stored} + \dot{Q}_{Hex}}{\dot{Q}_{fuel}} \times 100 \qquad (23)$$

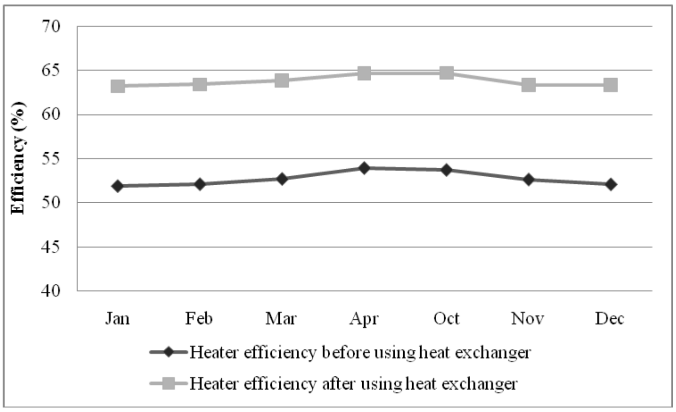

Figure 8. Mean heater efficiency

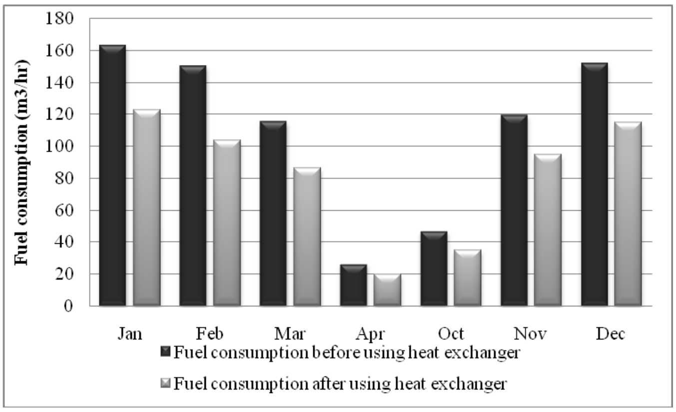

Figure 9. Fuel consumption

In this case new $\dot{Q}_{fuel}$ which has direct relation with fuel consumption represented as below:

$$\dot{Q}_{fuel} = \dot{Q}_{NG} + \dot{Q}_{stack} + \dot{Q}_{surf} + \dot{Q}_{stored} + \dot{Q}_{Hex} \quad (24)$$

Proposed plan saving has been depicted in Fig10.

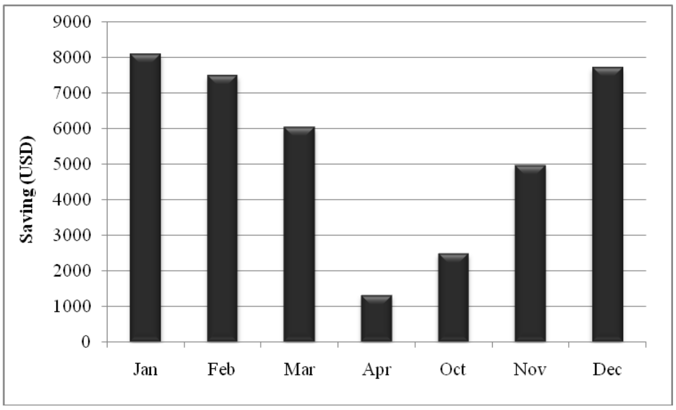

Figure 10. Saved money by proposed system

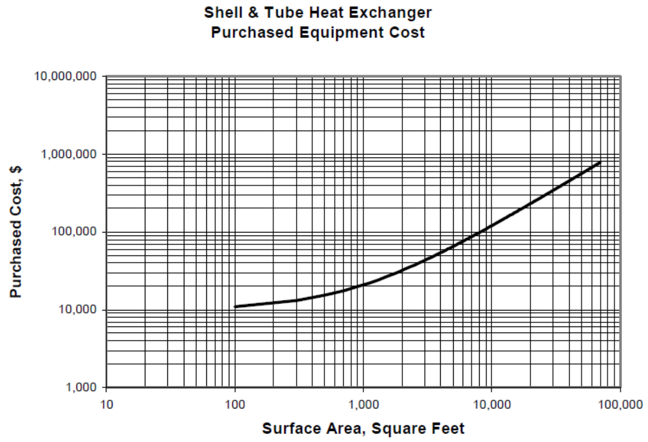

Figure 11. Diagram of capital cost for heat exchanger [11]

Capital cost to purchasing shell and tube heat exchanger could be obtained from Fig11. Cost analysis of proposed system has been presented in table 7. Regarding to total capital cost and annual benefit, payback period is calculated as 1.2 years.

TABLE 4. COST ANALYSIS FOR PROPOSED SYSTEM

The capital cost for purchasing heat exchanger	39300 US$
Total capital cost for the system installation	3000 US$
Annual O&M costs	5200 US$
Cost of natural gas per cubic meter	0.28 US$
Annual fuel saving	39700 US$

Considering equation (25), net present value (NPV) for the proposed system with 14% interest rate is shown in Fig12. NPV will be positive after 2 years and this could show that proposed system is economical [12].

$$NPV = \frac{R_t}{(1+i)^t} \quad (25)$$

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 as zero if NPV supposed to be positive after only one year and it clears the strength of proposed plan

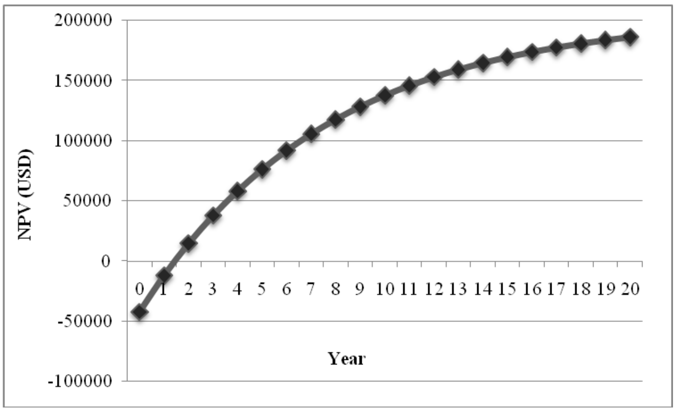

Figure 12. NPV for proposed system

VI. CONCLUSION

One of the main ways of high pressure natural gas transportation is passing through pipelines. Gas pressure reduction station has built to reduce gas pressure. Process of passing through regulator lead to forming hydrated and pipe obstruction. Indirect water bath heater is a common device to heat natural gas before regulation. Most of thermal energy generated in burner of these heaters wasted by flue gas exit. In this paper a solution represented to recover the energy of combustion product by means of adding a shell and tube heat exchanger and transfer it to inlet natural gas. In heat exchanger natural gas with high pressure assumed to be in tube and flue gas in shell. Result showed this plan can increase heater efficiency 11% and decrease fuel consumption 25%. Economic analysis depicted the strength of proposed system. Finally it can be claimed that indirect heaters are high consumption devices in natural gas transportation industry and it is necessary to revise their mechanism. Combustion product exit is the main factor for being low efficiency of these heaters. Therefore studying on ways of using thermal energy of flue gases or making lower the outlet temperature of them could be the best solution.

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NOMENCLATURE

η	Efficiency (%)
ρ	Density (kg/m^3)
N_T	Total number of tubes
f	Fouling
h	Hot fluid
c	Cold fluid
T_{am}	Ambient temperature (K)
T_w	Water temperature in heater (K)
T_{st}	Water temperature in storage tank (K)
$T_{NG\text{-}1}$	Natural gas temperature at entrance (K)
$T_{NG\text{-}2}$	Natural gas temperature at exit (K)
A	Effective surface area (m^2)
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)
d	Diameter
U	Total heat transfer coefficient