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Research Paper

Performance assessment of vortex tube and vertical ground heat exchanger in reducing fuel consumption of conventional pressure drop stations

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HIGHLIGHTS

- Utilization of geothermal energy and vortex tube in city gate station is investigated.
- The thermal and economic performance of the system was evaluated up to 25 years.
- The proposed system can reduce energy consumption up to 88%.
- The discounted payback period of the offered system is always less than 4.5 years.

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ABSTRACT

In natural gas pressure drop stations, before pressure dropping specially in cold seasons, the natural gas is preheated to prevent gas hydrate formation. Indirect water bath heaters employed for preheating. The heaters have low thermal efficiency and consume a large amount of natural gas for preheating. Due to the abundance of natural gas pressure reduction stations, reducing energy consumption is essential in this sector of the gas industry. In this study to reduce the heater energy consumption an innovative system based on using vortex tube and vertical ground heat exchanger is proposed. A vortex tube is used instead of throttle valve to reduce natural gas pressure. Unlike the throttle valve, vortex tube divides the incoming stream into two cold and hot streams. Cold stream enters the shell and tube heat exchanger and receives geothermal heat. Then the warmed cold stream mix with hot steam outgoing from the vortex tube, and goes towards the heater at low pressure. The proposed system can reduce energy consumption up to 88%, and the discounted payback period is always less than 4.5 years.

1. Introduction

Iran NG (natural gas) consumption is increasing by an accelerating rate. The consumption not only exceeds the expected consumption level, but also is several times higher than the world average consumption level. From 2003 to 2013, the world and Iran NG consumption has been increased by 28.9% and 90%, respectively. Currently, Iran is the third largest NG consuming country with the total amount of 162.2 billion cubic meters [1]. Transporting this considerable amount of NG to end consumers is an energy and cost consuming process. NIGC (National Iranian Gas Company) is in charge of delivering NG to consumers. In recent years, concerns over NG consumption at different internal sectors have been increased and NIGC has begun a program to reduce NG consumption, especially in NG pressure drop stations, which is known as CGS (City Gate Station).

CGS typically is used for NG metering and pressure reducing. The CGS receives high pressure NG (5–7) MPa from transmission pipeline and delivers it to distribution system with the pressure level of (1.5–2) MPa. Thus, keeping NG distribution network pressure at a desirable level is a main objective of the CGS. In Iran's and most of the world's CGSs, the pressure reduction process is generally accomplished by the throttle valve. In this process due to the positive Joule Thomson coefficient of the NG, as the pressure reduces, the NG temperature drops down. The temperature drop in the throttle valve can be about 4.5–6 °C per 1 MPa of pressure drop. In addition, there is a minimum allowable temperature called hydrate-formation temperature, which is a function of thermodynamic condition and composition of NG. The undesirable phenomenon of gas hydrate formation is because of water and liquid

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Nomenclature

$C_{p,NG}$	natural gas thermal capacity
$\dot{Q}$	heat transfer rate (kW)
$\dot{m}$	mass flow rate (kg/s)
m_w	the line heater water mass (kg)
B	borehole distance (m)
BHE	borehole heat exchanger
d	diameter (m)
D	ineffective borehole length (m)
k	thermal conductivity
LHV	lowering heating value
N	number of boreholes
NPV	net present value
PP	payback period
R	thermal resistance
r	number of years
$r_{i,j}$	radial distance between borehole i and j
r_p	pressure ratio
T_{in}	fluid temperature at the ground heat exchanger inlet (°C)
T_{out}	fluid temperature at the ground heat exchanger outlet (°C)
VT	vortex tube
H	active borehole length (m)
q	heat flux (W/m)

Greek letters

γ	specific heat ratio of natural gas
η_h	thermal efficiency of line heater
α	soil thermal diffusivity
μ_c	cold mass fraction
η_v	cooling efficiency of vortex tube

Subscripts

b	borehole
bw	borehole wall
cold	cold stream
cond	conduction
conv	convection
f	fuel
gr	ground
h and *H*	heater
hot	hot stream
hyd	hydrate
i	inner
in	inlet
NG	natural gas
o	outer
OC	external oh heater
tv	throttling valve
w	water

hydrocarbon presence in transmission pipeline. Some undesirable effects of hydrates in transmission system are fouling of the heaters, internal erosion and corrosion of pipelines and even blocking of [2]. To prevent this phenomenon, the NG must be warmed up to a specific level. Generally, line heaters are employed in the CGSs to preheat the NG stream. The heater has a low thermal efficiency and consumes a huge amount of NG as a fuel. Therefore, it is important to look for energy saving schemes in the CGS. Energy studies about the CGS could be categorized into two areas: most studies propose using expanders (generally turbo-expander) for recovering energy of high pressure NG [3–11] and some limited studies paid attention to reduce energy consumption in the CGS. Kargaran et al. [12] proposed the possibility of using VT (vortex tube) instead of throttling valves in NG pressure drop stations. They have studied VT performance with low pressure NG stream experimentally. In using the hot stream of VT, Universal Vortex Incorporation introduced a new device called Vortex Pilot Gas Heater (VPGH) by implementing VT technology. The core of the VPGH is a VT which converts internal energy of the high-pressure gas flow into highly intensive heat flux and prevents pilot gas freezing at NG transmission and distribution pressure regulation stations. Farzaneh-Gord et al. [13] offered and studied a solar system for providing part of heat demand in the Akand CGS. The heater was assumed to be uncontrolled, it means there is no control equipment to adjust heater burner according to energy demand of NG stream which must be preheated and it is adjusted daily or weekly by an operator (human). Economic analysis shown the system net benefit would be come back after 11 years. Their another study [14] revealed that utilizing solar heat with controllable line heater at the Akand CGS gives annual benefit of 27,011 USD with capital cost equal to 144,000 USD. The simple and discounted payback period (DPP) were also ascertained to be 5.5 and 8 years respectively. Lately, in another work, Farzaneh-Gord et al. [15] proposed and studied using of VGHXs (vertical ground heat exchangers) at Gonbad Kavoos CGS to lower fuel consumption. In the proposed system, preheating process of NG takes place in two steps, firstly, NG enters the shell and tube heat exchanger and receives geothermal heat, after then, NG pass through the line heater and its temperature reaches to a desirable level to avoid hydrate formation at the exit of throttle valve. Comprehensive thermo-economic analysis showed that a system comprising 8 boreholes with 150 m depth and 0.15 m diameter each is the most efficient configuration for Gonbad Kavoos CGS. The DPP and IRR (internal rate of return, the higher a project's IRR, the more desirable it is to undertake the project) of the system was computed to be about 5 years and 15.5% respectively. In comparison with utilizing solar systems, the offered system showed good economic performance. Table 1 informs deficiencies of the previous studies and highlights the significance of the current work.

Geothermal energy, is introduced as a sustainable solution for heating and cooling in a wide range of applications from small residences to large commercial buildings. For this purpose, Heat pumps are connected to the ground heat exchangers, which are installed either vertically or horizontally, and heat or cool building space. They provide better performance than other conventional heating and cooling devices such as air source heat pumps [16–20]. Ground coupled Heat pump uses relatively constant temperature of earth for heating and cooling purposes [21,22]. Ground temperature is approximately constant below the 15 m depth [23]. Utilization of underground heat without utilizing heat pump for providing process heat in industrial applications is not common, especially for low temperature cases. It may be due to available waste heat at low temperature in most big industries. However, for industrial sectors in which no waste heat is available, the use of geothermal energy for heat supply has the potential for substantial economization of primary energy resources. Presence of low temperature NG stream in CGS, which must be heated to a desirable temperature level creates a good opportunity for using earth heat. In this case, employing a VT instead of throttle valve not only creates more opportunity for the NG stream to be heated via geothermal energy, but also inherently has a great potential in decreasing the CGS energy consumption. VT is a simple mechanical

Table 1
Deficiencies of the previous proposed systems and significance of the current work.

Authors	Proposed system	Description
Farzaneh-Gord et al. [8]	Employing a solar heating system without storage tank at CGS	• Operate only a few hours of a day • Roughly sensitive to weather conditions • No peak heating energy reduction • High water in heater temperature • Low contribution of the proposed system in fuel saving • Relatively high payback period • Low profitable • Need to constantly clean of solar flat collectors
Farzaneh-Gord et al. [13]	Employing a solar heating system with storage tank at CGS	• Operate only a few hours of a day • Low contribution of the proposed system in fuel saving • High water in heater temperature • Relatively high payback period • Low profitable • Need to constantly clean of solar flat collectors
Kargaran et al. [12]	Proposal of utilizing vortex tube instead of throttle valve at CGS	• The authors only proposed utilizing of VT instead of throttling valve. They only studied NG behaviors in a VT
Farzaneh-Gord et al. [15]	Employing VGHX at CGS	• Low contribution of the proposed system in fuel saving • High water in heater temperature • Relatively high payback period • Low profitable
Ghezelbash et al. (current study)	Employing VGHX and VT	• Operate entire day • Low sensitive to weather conditions • Peak heating energy reduction • High contribution of the proposed system in fuel saving • Low water in heater temperature • Low payback period • Highly profitable

device, which separates a high-pressure stream into cold and hot streams. It was discovered by Ranque and developed by Hilsch. Consequently, it is also called RHVT (Ranque–Hilsch Vortex Tube) [24,25].

In this study, the possibility of reducing NG consumption in the CGS is investigated by employing a VT and VGHX. The proposed system is in series configuration with the heater. The NG passes through the VT and is divided into two separate hot and cold flows. A cold stream is heated in shell and tube heat exchanger and then mixed with the hot stream outgoing from the hot section of the VT. Then, it flows towards the heater. It is noteworthy that existing low pressure and temperature NG at the cold side of VT, causes more heat be absorbed from the ground.

2. Natural gas pressure drop stations

High-pressure NG which is transmitted to cities must be decompressed to the distribution network pressure level of 1.7 MPa. Schematic of the NG pressure drop station is shown in Fig. 1. Before pressure reduction in throttle valve, NG must be preheated to keep the gas temperature above the hydrate formation temperature. This process ensures that no liquid or solid phase exists at the transmission pipeline. The standard temperature of preheated gas is in the range of 30–55 °C [13]. The exact value depends on the inlet pressure. The type of preheater used in CGS is the indirect water bath heater, which also known as line heater. Schematic of line heater is shown in Fig. 1. Heat is produced by burning NG in fire tube,

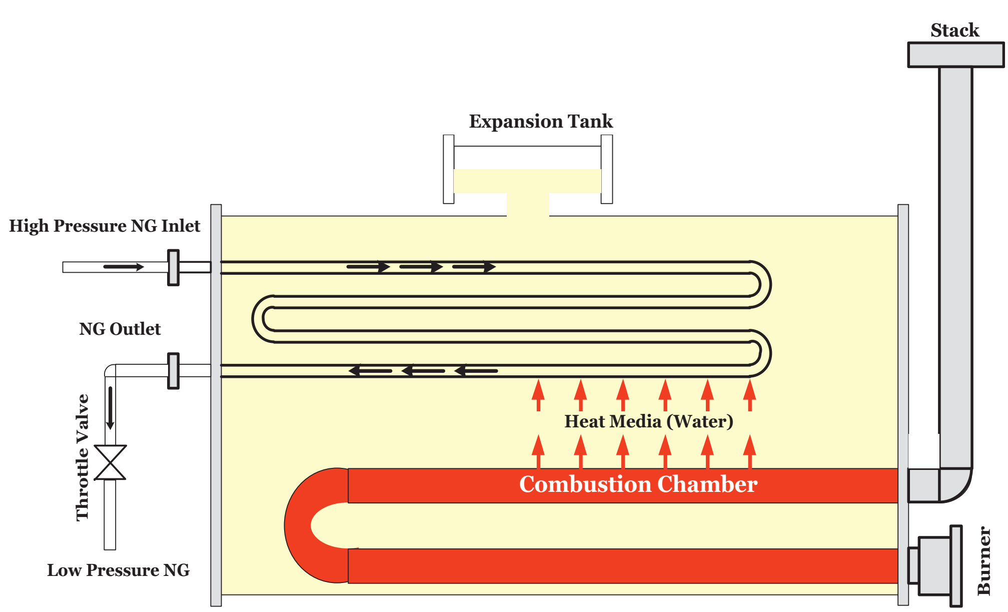

Fig. 1. Schematic of a NG pressure drop station.

and then it is transferred to heat transfer medium (water). Finally, heat is delivered to NG passing through the line heater. Current heaters are not equipped with automatic control systems and they are manually adjusted to meet the heating demands. As a result, fuel consumption of heaters stays constant for a certain time, even when heating demand is low. These types of heaters can easily be equipped with a temperature sensitive equipment to control the amount of fuel consumption. In this paper, it is supposed line heater is equipped with automatic line heater.

For NG, the temperature of hydrate formation depends on its constituents and pressure. By knowing the composition of NG, the temperature of hydrate formation could be computed by using thermodynamic models. And by knowing the outlet NG temperature, the required temperature at the heater outlet could be calculated by using below equation:

$$T_{NG-2} = T_{NG-3} + \Delta T_{tv} \tag{1}$$

In which, T_{NG-2} and T_{NG-3} are the NG temperature at the inlet and outlet of the throttle valve respectively. ΔT_{tv} is temperature drop due to pressure drop in throttle valve.

NG flows through pipes buried in the depth of 1–1.5 m. Consequently, gas temperature in the pipes is equal to the temperature of surrounding soil, due to the long-lived vicinity with soil. Soil temperature is also a function of ambient temperature. In warm seasons, the gas temperature at the station inlet is high enough and most of the warm days don't require being preheated. Thus, in these seasons, the heaters are turned off. Meanwhile, in the cold seasons and depending on geographic location of station, gas temperature at the entrance is low (not lower than 10 °C) due to low ambient temperature. In this study, the inlet gas temperature is assumed to be 10 °C throughout the year. Because of the availability of NG within the station, it is used as fuel. By knowing station inlet and outlet gas pressure and temperature, rate of heat absorbed by the gas passing through the heater, $\dot{Q}_{NG}$, is then could be calculated as:

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

$\dot{m}_{NG}$ is the NG mass flow rate passing through the CGS and h stands for enthalpy. With regard to the thermal efficiency of heaters, η_h, mass flow rate of consuming fuel is calculated as:

$$\dot{m}_f = \frac{\dot{Q}_H}{\eta_h \cdot \mathrm{LHV}} \tag{3}$$

In which, LHV, is the lower heating value of the fuel, $\dot{m}_f$ is fuel mass flow rate, $\dot{Q}_H$ is heating duty of heater and η_h is the thermal efficiency of the heater, which is in the range of 0.35–0.5 [13]. This low efficiency is due to the heat loss of the heater chimney. In this study the thermal efficiency of the heater is considered to be 0.4.

NG is passed through the coils which are immersed at constant temperature water bath. Formulation suggested for tubes enclosed in a constant temperature environment is presented in Eq. (4) [26]:

$$\frac{T_w - T_{NG-2}}{T_w - T_{NG-1}} = \exp(y), \quad y = \frac{-\pi D_{OC} L_C U_C}{\dot{m}_{NG} C_{p,NG}} \tag{4}$$

By rewriting Eq. (4), water temperature of the heater is found as:

$$T_w = \frac{T_{NG-2} - T_{NG-1} \cdot \exp(y)}{1 - \exp(y)} \tag{5}$$

3. Description of the proposed system

In this study an innovative method is proposed to reduce fuel consumption in the CGS. This system is based on utilizing VT as a core of the system and the VGHX, to enhance the amount of fuel consumption reduction.

VT is an inexpensive, safe and reliable device, which is able to supply heating and cooling. Unlike the throttle valve, VT separates the high-pressure gas stream into two hot and cold streams at lower pressure. Inlet gas flows into VT and passes through one or more nozzles. Nozzles create strong vortex flow. In addition to spiral gas flow in VT, gas moves along the tube towards the hot outlet. A part of flow exits with higher temperature from the hot end. A conical valve does not let the whole of the flow exits out. Part of the flow returns back through the main flow and exits from the other end with lower temperature. The cold stream passes through shell and tube heat exchanger and receives geothermal heat. Then warmed cold stream is combined with hot stream NG outgoing from the VT and the temperature of the stream rises at the line heater inlet. Therefore, compared to conventional CGS, the NG stream at the line heater inlet (proposed system) is at the low pressure and high temperature than conventional systems which cause to line heater consume low energy to heat the NG.

The proposed geothermal heat exchanger is a simple shell and tube heat exchanger with water as the working fluid. The hot water enters some long vertical tubes prepared in some deep boreholes. The cold water outgoing from the heat exchanger is taken down of the boreholes by the tubes and during this trip absorbs deep ground heat. The warmed water returns to the heat exchanger and gives back this heat to the cold natural gas stream and raise the NG temperature. Pay attention to the depth of boreholes and other characteristics of the considered geothermal system in the proposed system, the desired geothermal heater system with its boreholes and the other belongings is able to increase the temperature only up to undisturbed ground temperature. And even this temperature would rarely be achieved during the whole year as the water inlet temperature into the boreholes has low values.

4. Mathematical modeling

4.1. Vortex tube

To study the performance of VT, a few parameters should be defined. Cold mass fraction μ_c, is defined as the ratio of the mass flow rate of the cold stream to the mass flow rate of the inlet stream ($\dot{m}_{NG}$). The first law of thermodynamics for a VT can be written as follows:

$$(h_0)_{in} = \mu_c (h_0)_{cold} + (1 - \mu_c)(h_0)_{hot} \tag{6}$$

In which h_0 is stagnation enthalpy. Regardless of the potential and kinetic energy of outgoing flows of VT, energy balance for an ideal gas can be written as follows:

$$C_{p,NG} T_{in} = \mu_c C_{p,NG} T_{cold} + (1 - \mu_c) C_{p,NG} T_{hot} \tag{7}$$

where T_{in}, T_{cold} and T_{hot} are the gas temperature at the inlet, cold and hot outlets respectively. When Eq. (7) is arranged according to μ_c, one can estimate the cold mass fraction by knowing the temperature. The cold mass fraction could be obtained based on the hot and cold temperature differences [24]:

$$\mu_c = \frac{\Delta T_{hot}}{\Delta T_{hot} + \Delta T_{cold}} \tag{8}$$

The cold temperature difference is defined as the temperature difference of NG between the inlet and cold outlet of VT. Similarly, the hot temperature difference is defined as the temperature difference of NG between inlet and hot outlet. These two parameters defined as $\Delta T_{hot} = T_{cold} - T_{in}$ and $\Delta T_{hot} = T_{hot} - T_{in}$. To calculate the cooling efficiency of VT, the principle of adiabatic expansion of an ideal gas is used. This can be written as follows [24]:

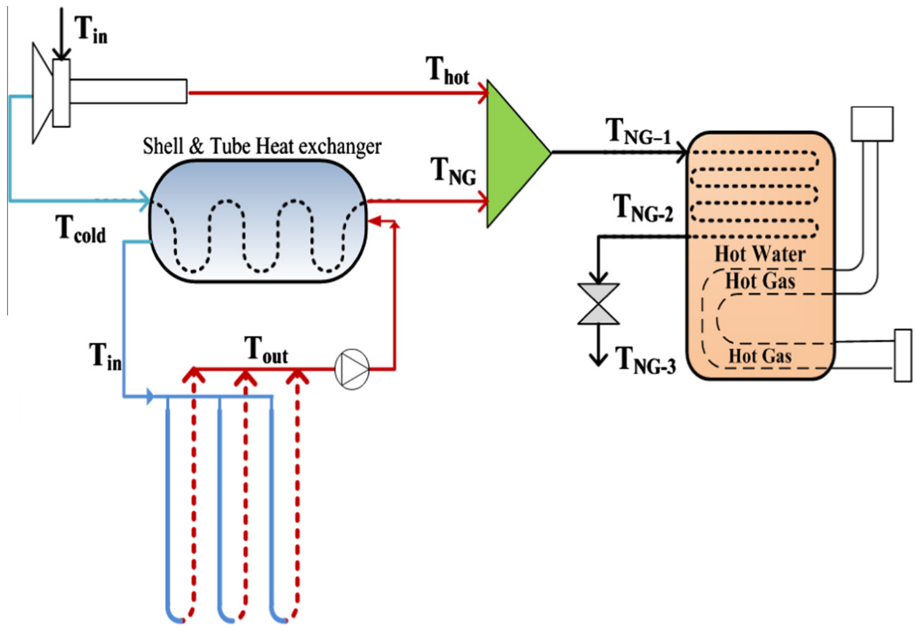

Fig. 2. The CGS equipped with vortex tube and geothermal heat exchanger.

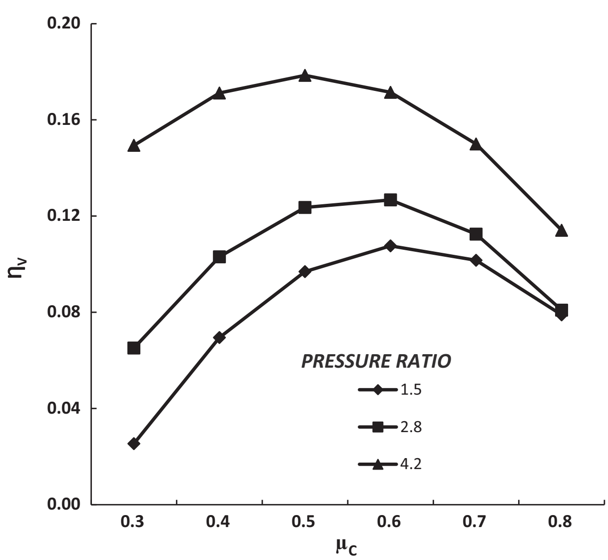

Fig. 3. Isentropic efficiency of vortex tube.

$$\eta_v = \frac{T_{in} - T_{cold}}{T_{in}\left(1 - \left(\frac{1}{r_p}\right)^{\frac{\gamma-1}{\gamma}}\right)} \quad (9)$$

where r_p is the pressure ratio of inlet to outlet stream and γ is the specific heat ratio. For different pressure ratios from 1.5 to 4.2, Farzaneh-Gord et al. carried out experiments [24]. Cooling efficiency of VT in this experiment is shown in Fig. 3. Based on their experiments, the maximum possible errors in the case of temperature and pressure measurement are 4% and 0.4%. For $T_C = -8$ °C, $T_h = 70$ °C and $T_{in} = 16$ °C, uncertainty in the calculation of cold mass fraction was calculated to be 0.5%. In similar way, uncertainty in the calculation of efficiency was calculated and found to be 6%. Table 2 also presents the characteristics and uncertainties of the measurement instruments.

By utilizing the experimental data in Fig. 3, cooling efficiency of VT could be obtained based on the cold mass fraction and pressure ratio as follows:

Table 2
Characteristics and uncertainties of the measurement instruments [24].

Instrument	Range	Uncertainty
Temperature sensors	−200 °C to 850 °C	0.5 °C
Pressure gauge	0–10 bar	1% full scale = 10 kPa
Flow meter	0.16–25 m^3/h	2%

$$\begin{aligned}\eta_v = {} & \frac{(r_p-2)(r_p-1.5)}{3.87}(-0.7204\mu_c^2 + 0.7218\mu_c - 0.1229) \\ & + \frac{(r_p-4.2)(r_p-1.5)}{3.87}(-0.8691\mu_c^2 + 0.9875\mu_c - 0.1529) \\ & + \frac{(r_p-2.8)(r_p-4.2)}{3.87}(-0.8352\mu_c^2 + 1.0258\mu_c - 0.2072)\end{aligned} \quad (10)$$

Now all equations have been specified for thermal analysis of VT and by the following steps, hot and cold temperatures are calculated:

A. First, inlet temperature and pressure are considered.
B. Then, by knowing the desired NG pressure at the exit, pressure ratio is determined.
C. By using experimental efficiency correlation, VT efficiency is computed for different cold mass fractions.
D. Finally, cold and hot NG temperature are given by the following equations:

$$T_{cold} = T_{in} - \eta_v \cdot T_{in} \cdot \left(1 - \left(\frac{1}{r_p}\right)^{\frac{\gamma-1}{\gamma}}\right) \quad (11)$$

$$T_{hot} = \frac{(T_{in} - \mu_c T_{cold})}{(1 - \mu_c)} \quad (12)$$

4.2. Vertical ground heat exchanger

In this study, heat transfer modeling inside and outside of boreholes is separately modeled with the borehole wall acting as the interface. The model for the borehole interior uses steady state heat transfer [27], whereas the outside model must take care of thermal phenomenon from the borehole wall to the surrounding soil and the other boreholes. Ground heat exchanger systems are generally involves one or multiple vertical boreholes. The boreholes have a diameter about 10–15 cm and 100–200 m long [28]. Much researchers studied heat transfer inside and outside of borehole heat exchangers. The methods are based on analytical [29–33], numerical [34–36] and hybrid models [37–39]. Precise solution is the main advantage of the numerical attempts. On the other hand the disadvantages with these methods are low flexibility in programming and time consuming computations which have been led to limited utilization in simulation programs. In contrast, accuracy of analytical solutions is not as high as the numerical models, but offers much flexibility in programming and low computational time. The both advantageous are sufficient to prefer the analytical models to numerical models. Analytical models are based on ILS (Infinite Line Source) [31], CS (Cylindrical Source) [27] and FLS (Finite Line Source) [30].

The ILS and CS were generally used to model heat transfer around the boreholes. Both methods offer a one-dimensional solution to heat transfer from a heat source and neglect axial heat transfer which cause to overestimation of borehole wall temperature for times greater than 3 years [40]. On the other hand, FLS solution can predict 2D heat transfer from a FLS positioned in a semi-infinite medium and subjected to a constant heat transfer rate. Eskilson [38] proposed the analytical solution of the FLS. Zeng et al. [30] presented a new methodology to estimate the temperature in a borefield using the FLS. Contrary to Eskilson work which the temperature was calculated at mid-height of the source, they used the integral mean temperature over the borehole height and it showed better results. Lamarche and Beauchamp [41] simplified the double integral into a single integral in the FLS solution which led to reducing considerably the time required to calculate the integral mean temperature over the borehole height. Claesson and Javed [42] obtained a correlation to predict integral mean temperature of geothermal bore field where boreholes are buried at a distance D from the ground surface. In contrast to Zeng et al. [30] and Lamarche and Beauchamp [41], the FLS was defined by point heat source integral which was initially integrated in space. Then after, the obtained solution is given in the form of an integral in the time domain which provides us with a simpler formulation of the FLS method.

4.2.1. Borehole wall temperature at variable heat flux

By using Claesson and Javed [42] solution and temporal superposition principle for time varying heat flux the average borehole wall temperature, T_{bw}, for the entire set of N boreholes is obtained as:

$$T_{bw} = T_{gr} + \left[\sum_{j=1}^{n_t} \frac{q_j - q_{j-1}}{4\pi k_{gr}} \cdot \int_{\frac{1}{\sqrt{4\alpha(t-t_{j-1})}}}^{\infty} I_e \cdot \frac{I_{ls}(Hs, Ds)}{Hs^2} \cdot ds \right] \tag{13}$$

where j counts the hourly time steps as all the calculation processes are mainly done on a basis of hourly periods. Therefore, q_j represents the absorbed q in time step j and q_{j-1} does that for time step $j-1$. H and D stand for active and inactive borehole length. T_{gr}, k_{gr} and α also refer to the ground temperature, thermal conductivity coefficient and diffusion coefficient of the borehole, respectively.

$$I_e(s) = \frac{1}{N} \sum_{i=1}^{N} \sum_{j=1}^{N} e^{-r_{i,j}^2 s^2} \tag{14}$$

Here $r_{i,j}$ denotes the radial distance between borehole i and j ($i \neq j$). The contribution of the own heat source of the borehole i is obtained for the radial distance r_b.

$$I_{ls}(h,d) = 2ierf(h) + 2ierf(h+2d) - ierf(2h+2d) - ierf(2d) \tag{15}$$

$$h = Hs, \quad d = Ds \tag{16}$$

4.2.2. Fluid temperature at the VGHX exit

Outgoing water temperature from VGHX could be computed by borehole thermal resistance concept:

$$T_{out} = T_{bw} - q \cdot R_b + \frac{q \cdot H}{2\dot{m}_b C_b} \tag{17}$$

R_b is the thermal resistance of the borehole interior. $\dot{m}_b$ and C_b are water mass flow rate in each borehole and water thermal capacity respectively.

Thermal resistance inside the borehole includes the thermal resistance of fluid convection, R_{conv}, and that of solid conduction in the pipe, R_{cond}, and grout, R_{grout}. Conductive Thermal resistance of pipe and convective thermal resistance of fluid are obtained by Eqs. (19) and (20), respectively [27]:

$$R_b = \frac{1}{2}(R_{conv} + R_{cond}) + R_{grout} \tag{18}$$

$$R_{cond} = \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k_{pipe}} \tag{19}$$

$$R_{conv} = \frac{1}{\pi d_i h} \tag{20}$$

h is a convective heat transfer coefficient and a function of the Nusselt number (Nu). Nu is determined according to the flow regime. By increasing the turbulence of flow, thermal convection resistance decreases and consequently heat transfer between soil and VGHX enhances [11].

$$\begin{cases} Nu = 0.023Re^{0.8}Pr^{0.4}, & Re > 10^4 \\ Nu = 4.36, & Re < 2300 \\ Nu = \frac{\left(\frac{f}{8}\right) \times Re \times Pr}{1.07 + 12.7\left(\frac{f}{8}\right)^{0.5} \times (Pr^{0.67} - 1)}, & 2300 < Re < 10^4 \end{cases} \tag{21}$$

To calculate grout thermal resistance, Paul's model [43] was used. Paul used the so-called shape factor correlations, which were created by the experimental data and simulation results.

$$R_{grout} = \frac{1}{k_{grout}\beta_0 \left(\frac{d_b}{d_o}\right)^{\beta_1}} \tag{22}$$

where β_0 and β_1 are dimensionless equation fit coefficients. d_b and d_o are diameter of borehole and outer diameter of pipe, respectively. Values of β_0 and β_1 are variable depending on position of the tube in the grout. In this study, typical values of $\beta_0 = 20.100377$ and $\beta_1 = -0.94467$ are considered for a typical 3.2 mm U-tube shank spacing [39].

The energy consumption of the pump to maintain the water cycle between the VGHX and shell and tube heat exchanger is computed by the following equation which gives the exact value of pumps work in a geothermal system [15]:

$$W_p = \frac{\dot{m}_b \cdot H_T \cdot \Delta P}{\rho_b \cdot \eta_p} \tag{23}$$

where ΔP is the amount of pressure drop along the vertical pipes which is considered to 0.4 $\frac{kPa}{m}$ [44]. ρ_b is water density and equal to 1000 $\frac{kg}{m^3}$ and η_p is the pump efficiency which is considered to be 80%. H_T is also the total effective length of pipes in the boreholes and can be given by:

$$H_T = 2N(H + D) \tag{24}$$

4.3. Line heater

Both the typical CGS and proposed system (Figs. 1 and 2) have line heater in common. Therefore, the heater energy balance for two systems can be written as follows:

$$m_w \cdot C_{pw} \frac{dT_W}{dt} = \dot{Q}_h - \dot{Q}_{NG} \quad (25)$$

In which, m_w, C_{pw} and T_w are the heater water mass, heat capacity and water temperature. $\dot{Q}_h$ is the rate of thermal energy generated by the burner (in kW), and computed in time step of 1 h [13]:

$$\dot{Q}_h = \frac{m_w \cdot C_{pw}(T_{w(i+1)} - T_{w(i)})}{3600} + \dot{Q}_{NG(i)} \quad (26)$$

The mass flow rate of fuel (m^3/h) to provide the amount of energy is defined as follow [13].

$$\dot{m}_f = \frac{m_w \cdot C_{pw}(T_{w(i+1)} - T_{w(i)})/3600 + \dot{Q}_{NG(i)}}{\eta_h \cdot \text{LHV}} \quad (27)$$

4.4. Energy analysis of the proposed system

In the second system, as the previous one, thermodynamic properties of the inlet gas should be determined. To this aim, the outlet gas temperature of shell and tube heat exchanger should be specified. This gas stream is heated by geothermal energy. The energy balance for shell and tube heat exchanger and vertical heat exchanger is presented as below.

$$m_w \cdot C_{pw} \frac{dT_w}{dt} = \dot{Q}_{VGHX} - \dot{Q}_{NG} \quad (28)$$

$$T_{w(i+1)} = T_{w(i)} + \frac{(\dot{Q}_{VGHX(i)} - \dot{Q}_{NG(i)}) \times 3600}{m_w \cdot C_{pw}} \quad (29)$$

where $\dot{Q}_{VGHX}$ is the rate of thermal energy absorbed by VGHX from the ground. All variables on the right and left hand side of equal sign, in Eq. (28), are unknown. By assuming an initial temperature of water in shell and tube heat exchanger, Eq. (28) can be solved. The assumed temperature causes that the inlet water temperature of VGHX becomes a known variable in first time step. Then by solving the following two equations with two unknowns, Eq. (30), temperature of water outgoing from VGHX, and rate of thermal energy absorbed from the ground is determined [15].

$$\begin{cases} T_{out} = T_{bw} + q \cdot R_b - \frac{q \cdot H}{2\dot{m}_b C_b} \\ q = \frac{\dot{m}_{bw} C_{bw}(T_{out} - T_{in})}{H} \end{cases} \quad (30)$$

Eq. (30) must be solved at successive time steps to specify the temperature of water in shell and tube heat exchanger at the time step of $i + 1$. Here, time step of 1 h is considered. Total thermal energy absorbed by VGHX from the ground is as follows.

$$\dot{Q}_{VGHX} = q \times N \times H \quad (31)$$

In shell and tube heat exchanger, water temperature is known and temperature of NG at the outlet could be obtained. Parameter y is calculated by Eq. (33) and thermodynamic properties of NG at the entrance of heater are obtained by Eq. (6).

$$T_{NG} = T_w[1 - \exp(y)] + T_{cold} \times \exp(y) \quad (32)$$

$$y = \frac{-\pi D_{OC} L_C U_C}{\mu_C \cdot \dot{m}_{NG} \cdot C_{pNG}} \quad (33)$$

The rate of thermal energy absorbed by NG is given by Eq. (34) while passing through the shell and tube heat exchanger. By specifying this parameter, calculation of energy and fuel consumption of the heater is determined by Eqs. (26) and (27) respectively.

$$\dot{Q}_{NG} = \mu_c \cdot \dot{m}_{NG} \cdot C_{p,NG}(T_{NG} - T_{cold}) \quad (34)$$

5. Methods of economic evaluation

In order to decide whether to accept or reject capital budgeting projects, it is necessary to perform assessments using some methods like DPP and net present value (NPV). The DPP is the time in which the initial cash outflow of an investment is expected to be recovered from the discounted cash inflows. This period is usually expressed in years. Easy to calculate and understand are the advantages of this method. Eq. (35) is used for calculating DPP of the project:

$$\text{DPP} = A + \frac{B}{C} \quad (35)$$

A is the last period with a negative discounted cumulative cash flow; B is the absolute value of discounted cumulative cash flow at the end of the period A, C is the discounted cash flow during the period after A.

The difference between the present value of cash inflows and the present value of cash outflows is called NPV. It is one of the most reliable measures used in capital budgeting because it accounts for the time value of money by using discounted cash inflows. In addition, this parameter is used to analyze the profitability of an investment or project. Project with negative NPV is rejected and positive or zero NPV is accepted:

$$\text{NPV} = \sum_{n=0}^{N} \frac{C_n}{(1+r)^n} \quad (36)$$

where n, C_n and r refer to the number of years, cash flow in the project in the corresponding year and the inflation rate.

6. Case study

Foe studying the performance of the offered system, Semnan station is selected. Semnan is located in the central northern portion of the Iran. It is selected due to low need and no need to line heater running in warm seasons. So heater will be run only at the three coldest months of the year, i.e. January, February and December.

To ensure that the gas temperature after leaving the station does not fall below the hydrate temperature (see Table 3), the outlet gas temperature leaving the station is assumed to be kept at 15 °C. Then by considering Eq. (1), inlet NG temperature to the throttle valve computed to be 38 °C. Inlet pressure to the CGS is 6895 kPa and gas leaves the station at a pressure of 1724 kPa.

Table 3
NG composition and properties of the Semnan CGS.

Compositions	Mole fraction (%)
CH_4	87.7
C_2H_6	4.7
C_3H_8	1.74
n-C_4H_{10}	0.42
i-C_4H_{10}	0.37
n-C_5H_{12}	0.1
i-C_5H_{12}	0.13
C_6H_{14}	0.08
N_2	4.7
CO_2	0.06
LHV (MJ/kg)	45.85
ρ_{NG} (kg/m^3)	0.7756
$C_{p,NG}$ (J/kg K)	2594
T_{hyd} (°C)	6

Table 4
Properties of the line heater [13].

Natural gas inlet pressure	7 MPa
Natural gas outlet pressure	1.7 MPa
Surface area of fire tube heater	88.1 m^2
Water capacity	38 m^3
Diameter of coil	0.1 m
Length of coil	280 m
Heater maximum working temperature	88 °C
Heater maximum heating duty	1750 kW

Another important parameter related to the offered system is the undisturbed ground temperature. As mentioned at Section 1, ground temperature at the depth below than 15 m is relatively constant. This constant value is equal to yearly average ground surface temperature. Thus, Semnan undisturbed ground temperature was determined to be 17.8 °C [45].

The geometry and other parameters associated with a common line heater in Iran are presented in Table 4.

To illustrate the performance of the system, four sets of VGHX with 16 and 25 boreholes are used in square and L shaped configurations. Fig. 4 shows two examples of borefield configurations. Distance between boreholes is B. Other parameters have been specified in Table 5.

7. Results and discussion

The existence of the heater in CGS is accompanied with two risky problems. The first is related to the short distance of fire flame from NG passing through it, and another problem is high fuel consumption in cold seasons. In this paper, the possibility of the simultaneously using geothermal heat exchanger and VT is investigated.

7.1. No hydrate formation zone for VT

Fig. 5 shows the temperature of cold stream according to changes in pressure ratio and cold mass fraction for VT. It is observed that the cold stream temperature reaches down to 0 °C when the cold mass fraction is low and pressure ratio is high. In this condition, due to high-pressure reduction, there is a possibility of hydrate formation. Consequently, to prevent hydrate formation, the proper cold mass fraction should be selected for different pressure ratios. It can be seen for a cold mass fraction of 0.8, cold stream temperature will be always higher than 1.8 °C. According to Fig. 5, for a pressure ratio of 3, cold stream temperature will

Table 5
Parameters used in simulation of VGHX.

Parameter	Value	Parameter	Value
Pipe properties		*Soil properties*	
d_i (cm)	2.04	T_{gr} (°C)	17.8
d_o (cm)	2.5	k_{gr} (W/m K)	3
k_{pipe} (W/m K)	0.42	α (m^2/day)	0.1178
Borehole properties		*Fluid properties*	
d_b (cm)	15	k_b (W/m K)	0.5626
K_{grout} (W/m K)	1.25	C_b	4193
H (m)	100	ρ_b (kg/m^3)	1000
B (m)	5	Pr (–)	10.5
N	16 and 25	μ_b	0.001409
D (m)	5	m_b (kg/s)	0.2615

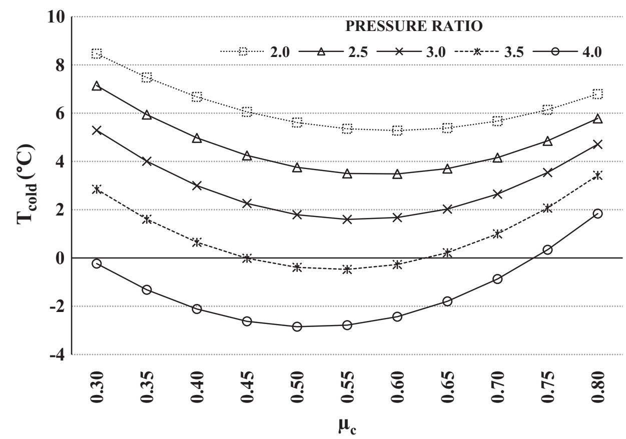

Fig. 5. Variation of cold stream temperature with cold mass fraction and pressure ratio.

be always higher than 1.5 °C at different values of cold mass fractions. Fig. 6 illustrates the pressure ratio of Semnan station over one year. The average pressure ratio over the year is about three. Therefore, the cold mass fraction of 0.8 is selected for studying the performance of the proposed system.

7.2. Detailed thermal and economic performance of the proposed system

Fig. 7 illustrates NG mass flow rate at the CGS inlet and mass flow rate of the cold and hot stream at the exit of the VT. Due to the high amount of cold mass fraction (0.8), cold stream mass flow

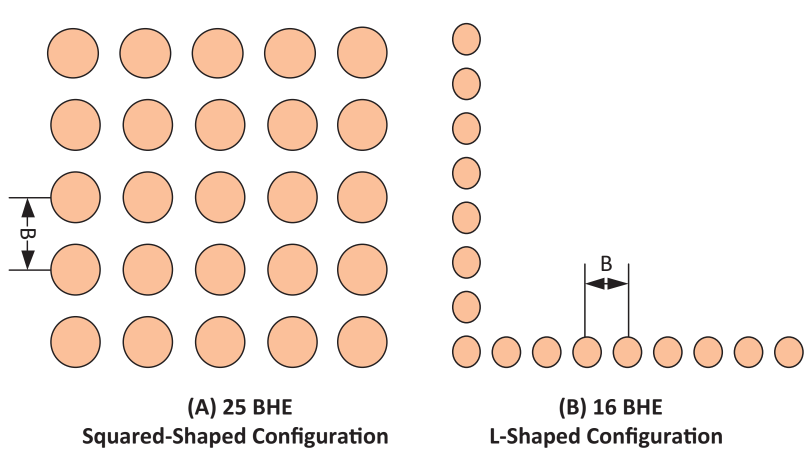

Fig. 4. Borehole heat exchanger configurations.

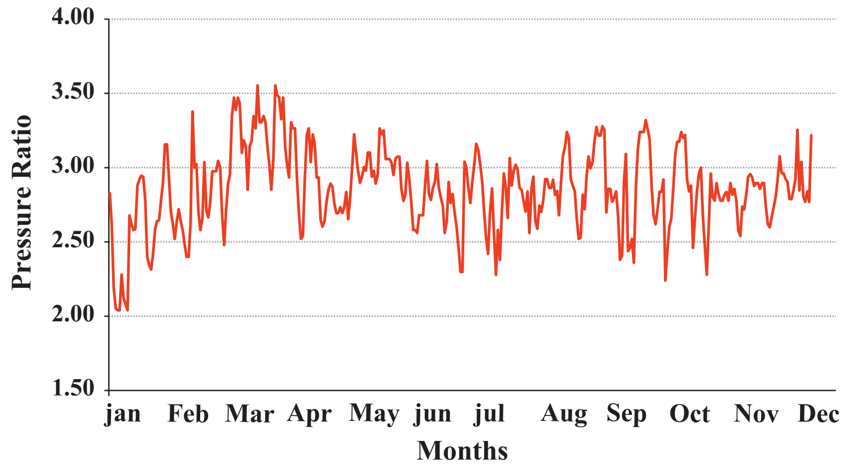

Fig. 6. Pressure ratio at Semnan CGS during a year.

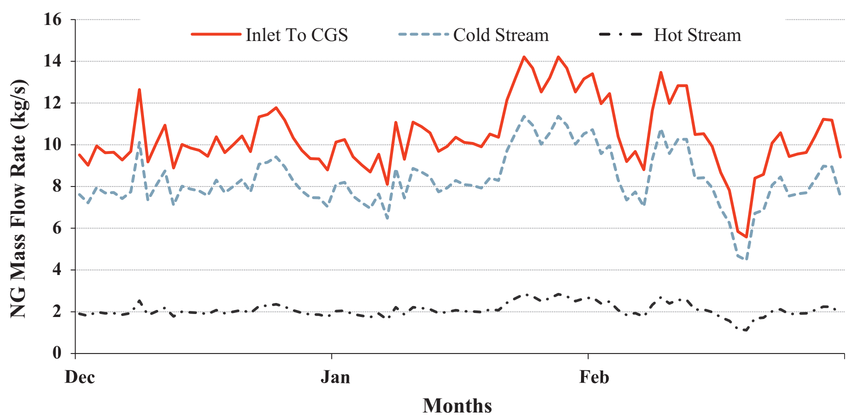

Fig. 7. NG mass flow rate at three different points: inlet to CGS, cold and hot stream at the VT exit.

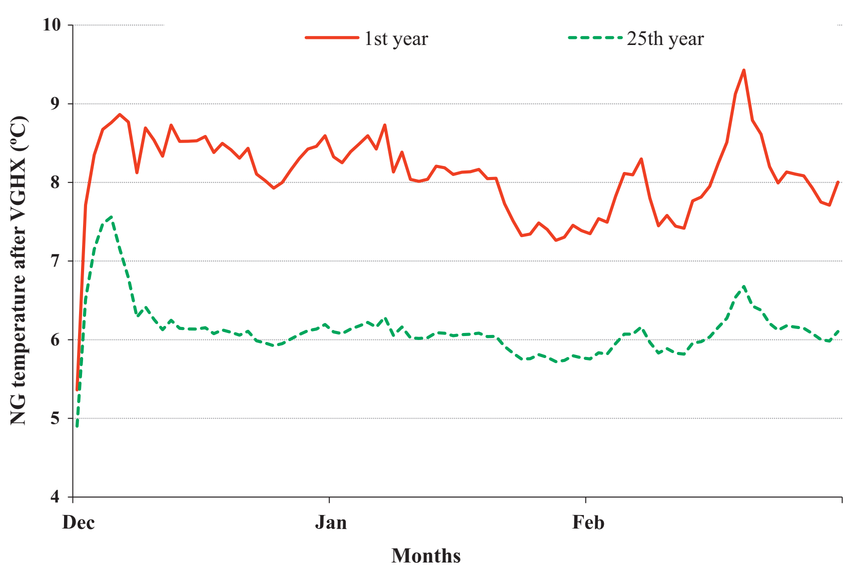

Fig. 8. NG temperature at the exit of the shell and tube heat exchanger.

rate which should be pass the shell and tube heat exchanger is higher than hot one. It is clearly obvious the peak NG mass flow rate occurs on late days of the January and reaches to lowest value on February.

Fig. 8 shows the heated NG temperature by the geothermal heat exchanger on the 1st and 25th year. Inlet NG temperature and pressure to the VT is 10 °C and 6895 kPa. VT separates the incoming stream into cold and hot stream with the same pressure of the

2298.3 kPa (pressure ratio of 3). But they have different temperature of 4.7 °C and 31.2 °C respectively. Then cold stream enters the shell and tube heat exchanger and receive the geothermal heat collected by the VGHXs. At the exit point of the shell and tube heat exchanger NG temperature (T_{NG}) reaches to the range of (5.4–9.4) °C and (4.9–7.6) at the 1st and the 25th year of the operation respectively. The decreasing trend in NG stream temperature outgoing from the shell and tube heat exchanger is because of the decrease in the absorbed geothermal energy by the VGHXs. Thermal interaction between boreholes and temperature drop around the boreholes environment after a long time operation are responsible of fall in the absorbed geothermal energy.

At the next step heated NG via geothermal energy proceeds towards the mixing with hot stream outgoing from the VT, after mixing process, the NG temperature at the line heater inlet (T_{NG-1}) increases, Fig. 9.

The NG temperature at the line heater inlet in the proposed and conventional system is in the range of (8.5–13.8) °C and 10 °C respectively. The corresponding pressures at this point are also 2298.3 kPa and 6895 kPa. The heater receives the incoming stream and raise the NG temperature to 17.9 °C (offered system) and 38 °C (conventional system) in order to keep the NG temperature constant (15 °C) at the throttle valve exit. Fig. 10 displays the heating duty of the conventional system and offered system on the first and last year operation of the system. As can be seen, line heater heating duty of the conventional system decreased sharply as the geothermal system is employed. Based on obtained results the share of geothermal in energy consumption reduction is up to 10% and the remainder comes from the VT.

It is worth noticing that the water bath temperature in the heater, Fig. 11, also decreases by employing the offered system. It decreases approximately from 41.5 °C to 18.5 °C (average temperature). The low temperature water bath will be lowered maintaining costs of the line heater in reality.

Fig. 12 shows the total daily fuel consumption of the both systems. The conventional CGS consumes 0.42% of the total NG passing through the station. While by employing the geothermal system the average NG consumption in the whole life of the system reaches to 0.079% of total NG passing through the station.

Table 6 presents descriptive statistics of the pressure ratio (r_p), inlet NG mass flow rate to CGS, NG temperature after warming up via VGHX (T_{NG}), NG temperature at the inlet of line heater (T_{NG-1}), water in heater temperature (T_w) and daily fuel consumption of heater. For example, daily fuel consumption before and after proposed system, at first and final year, is 4892.3, 764.22 and 992.36 m^3 respectively. In case of old system, Daily fuel consumption is lower than 4732.6 m^3 at 50% of operating time (median),

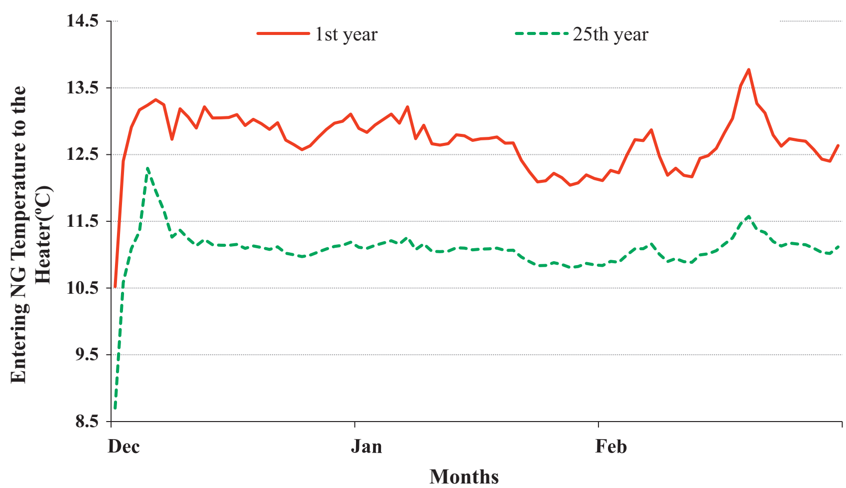

Fig. 9. NG temperature at the entrance of the conventional and proposed line heater.

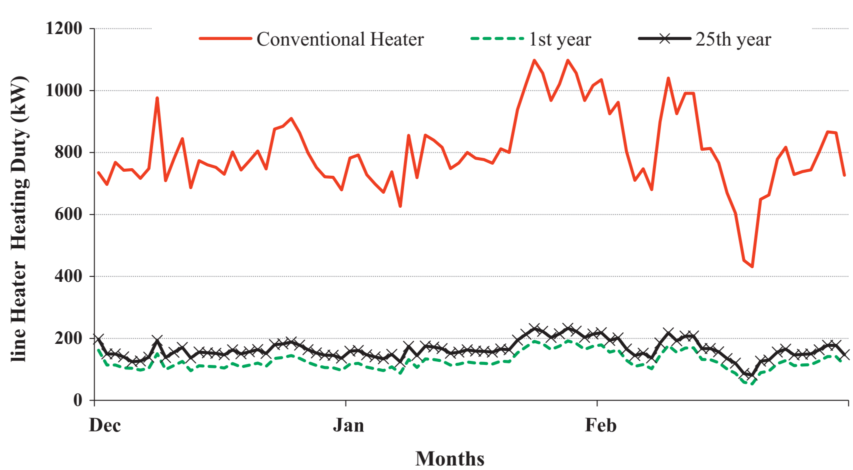

Fig. 10. Heating duty of the line heater at the conventional CGS and the proposed system.

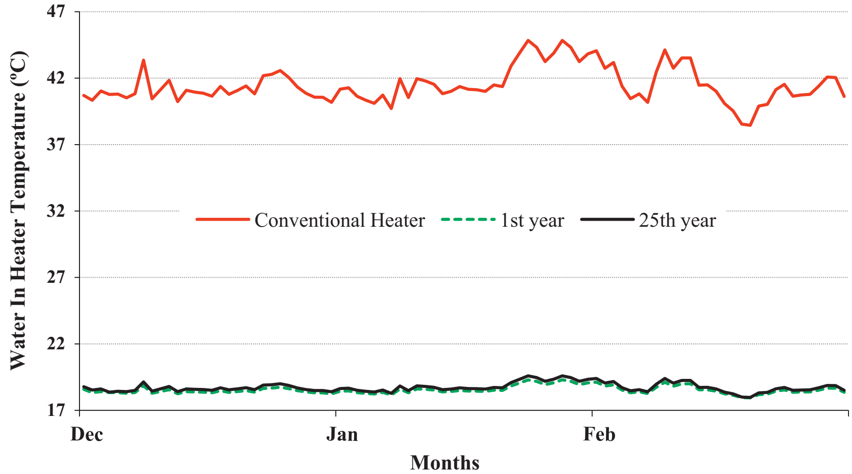

Fig. 11. Water bath temperature of the conventional and offered system.

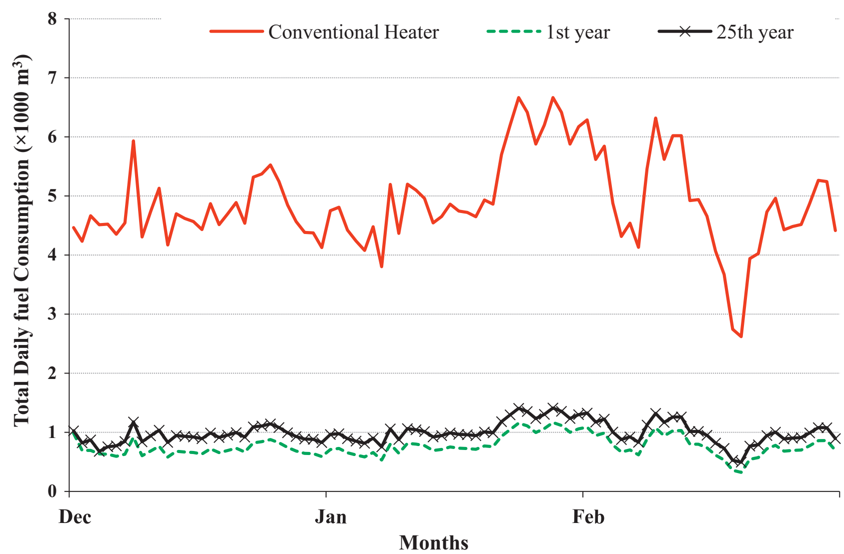

Fig. 12. Total daily fuel consumption of heater before and after the employing proposed system.

Table 6
Descriptive statistics of Figs. 6–12 (except Fig. 10).

	r_p	$\dot{m}_{NG}$	T_{NG}		T_{NG-1}		T_w (heater)			Daily fuel consumption (m^3)		
			1st Year	25th Year	1st Year	25th Year	Old system	1st Year	25th Year	Old system	1st Year	25th Year
Range	1.51	8.6	4	2.66	3.25	3.59	6.39	1.36	1.63	4047.5	843.37	917.09
Mean	2.86	10.43	8.1	6.11	12.71	11.08	41.5	18.53	18.73	4892.3	764.22	992.36
Variance	0.076	2.7	0.288	0.125	0.184	0.115	1.78	0.086	0.118	594,720	29,047	33,264
Std. deviation	0.27	1.64	0.54	0.35	0.429	0.34	1.33	0.293	0.343	771.18	170.43	182.38
Min	2.04	5.58	5.36	4.9	10.5	8.7	38.45	17.94	17.98	2619.3	321.83	494.6
25% (Q1)	2.69	9.44	7.79	5.96	12.47	10.99	40.64	18.34	18.5	4431.1	655.21	885.68
50% (median)	2.86	10.1	8.13	6.08	12.74	11.09	41.14	18.45	18.64	4732.6	723.58	961.73
75% (Q3)	3.02	11.25	8.44	6.16	12.98	11.16	42.1	18.67	18.88	5278.5	859.16	1084.5
Max	3.55	14.2	9.43	7.56	13.78	12.29	44.84	19.3	19.6	6666.7	1165.2	1411.7

this parameter is 723.58 and 961.73 m^3 for the proposed system at first and final year respectively. As could be seen, daily fuel consumption standard deviation of the proposed system is very small compared to the old system; it means heater controller equipment could be uninstalled if there is no extra budget for the project, because it could be adjusted daily or weekly by an operator.

Fig. 13 illustrates the total annual obtainable benefit from the proposed system. Evidently, for the first year of performance the annual benefit is approximately 100 thousands USD. Although, the annual benefit of the system decreases over time, even after 25 years, the system still works impressively by the total annual benefit about 95 thousand USD.

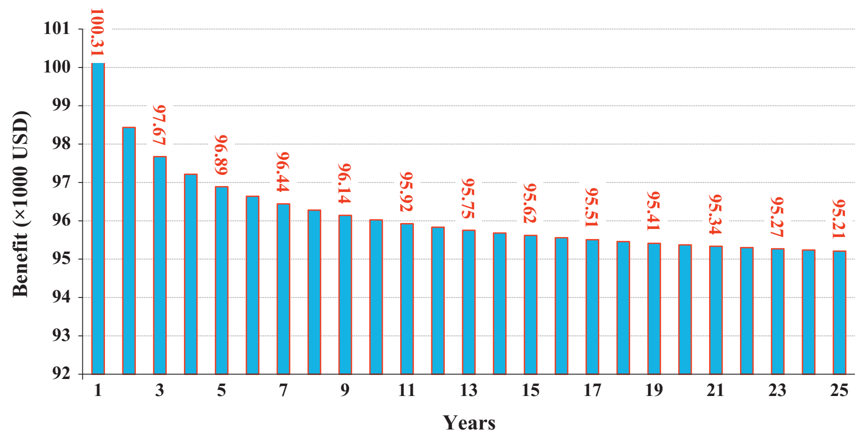

Fig. 13. Total annual obtainable benefit from the proposed system.

7.3. Performance of the system at different pressure ratio and borehole configurations

The Performance of the system is investigated at different pressure ratios and in a cold mass fraction of 0.8. Results are shown in Fig. 14. The amount of reduction in energy consumption is expressed as a percentage of reference state. The reference state represents the energy consumption of the typical CGS heater.

The reduction in energy consumption is proportional to the pressure ratio. With the further reduction of pressure in VT, the outgoing cold stream temperature get colder, and temperature difference increases between gas stream and water in shell and tube heat exchanger. The high temperature difference causes more heat to be absorbed by gas flow passing through the shell and tube heat exchanger. Outgoing stream temperature from shell and tube heat exchanger increases in comparison with the low-pressure ratio case. Thus, the thermal load of heater reduces. As an example, in a pressure ratio of 4 with 16 VGHX in L-shaped configuration, energy consumption reduction is 86.42.

7.4. Economic aspects of the proposed system

7.4.1. Discounted payback period

The payback period for each considered case were calculated and are shown in Fig. 15. It is observed that the payback period by increasing the total borehole length goes up. It is due to the high cost of drilling and installing VGHX. Here, costs of VGHX are considered to be 40 USD/m. As Fig. 15 shows by an increase in pressure ratio, payback period reduces. Minimum and maximum Payback period occurs in pressure ratio of four, and two, respectively.

7.4.2. NPV analysis

Fig. 16 shows net present value or profitability of the proposed systems. Due to the overlapping of 16 and 25 VGHX graphs in the L and square shaped configurations, only the results of L-shaped heat exchangers are presented. Systems with 16 VGHX are profitable than systems with 25 VGHX. In addition, it reveals this fact that increasing numbers of VGHX although could decrease energy consumption of the heater, but the high capital cost of VGHX system has an opposite effect on profitability of the proposed system.

Some uncertainties involved in analysis are measured quantities, NG price, vortex tube cost and discount/inflation rate, but among them, measured quantities impact on results is insignificant. So, the impact of discount rate, NG price and vortex tube cost on results (NPV and DPP) were investigated. In Table 7 six set of discount rate, NG price and VT cost can be observed. It could be concluded from the table that NG price could be vital in analyzing of such systems, at a minimum discount rate of 5%, the NG price of 0.03 USD and VT cost of 7000 USD the DPP was computed about 19 years, however on the other side, the DPP was computed about

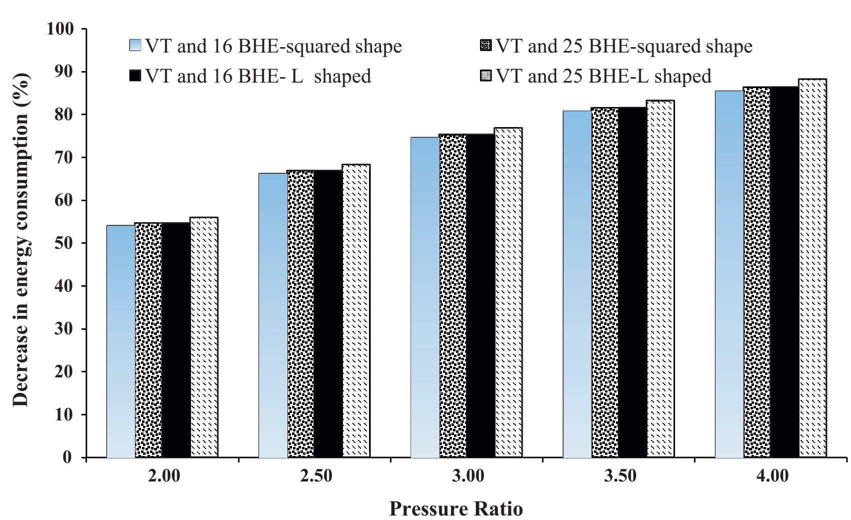

Fig. 14. Reduction in energy consumption of heater at different pressure ratios.

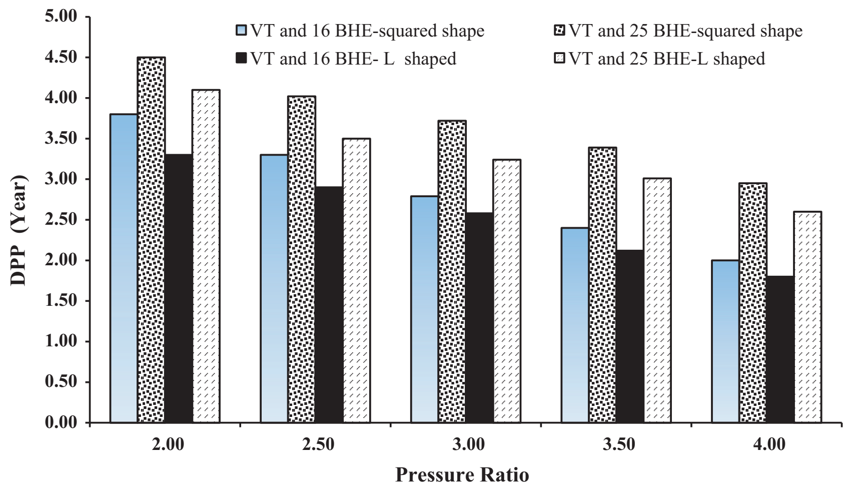

Fig. 15. Discounted payback period of proposed systems.

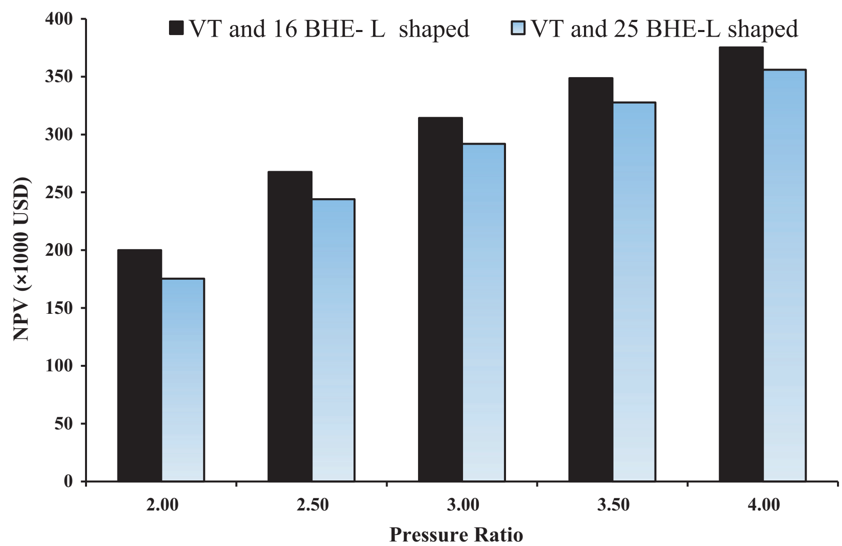

Fig. 16. Net present value of the proposed systems.

Table 7
Impact of uncertainties in NPV and DPP analysis.

	Discount rate (%)	NG price (US$)	VT cost (US$)	NPV (US$)	DPP (year)
SET1	5	0.03	7000	22,183	18.8
SET2	5	0.05	7000	122,972	9.1
SET3	15	0.08	25,000	39,198	10.3
SET4	10	0.1	12,000	191,755	4.8
SET5	15	0.1	15,000	95,748	6.9
SET6	20	0.15	25,000	120,965	4.3

Table 8
Optimum total borehole length and NPV.

r_p	N_b	H (m)	H_T (m)	NPV (USD)
2.00	6	120	720	238450.3
2.50	8	110	880	283441.6
3.00	11	100	1100	325840.2
3.50	12	130	1560	353673.1
4.00	14	120	1680	365236.4

4 years at a maximum discount rate of 20, the NG price of 0.15 USD and VT cost of 25,000 USD. It could be understood that even though increasing discount rate and VT cost have a negative impact on payback period, but high NG price could lower payback period.

For such systems, naturally, increasing the number of boreholes leads to more obtainable energy and higher cost of capital simultaneously. In such cases, the capital cost value must be selected equal to the total annual cost of the system pivotal parameter (here, the annual price of fuel). The point in which the fuel cost line crosses its respective capital cost line reveals the optimum total borehole length. This method was used in different pressure ratios with different number and length of boreholes. In determining the optimum borehole length, the range of borehole length (H) is 100–150 and number of boreholes also were changed from 5 to 30. Table 8 presents the optimum total borehole length and NPV.

8. Conclusion

Temperature of inlet NG to CGS is low in cold seasons and large amount of NG passes through CGS. Due to this fact, line Heaters must consume considerable amount of fuel to prevent Hydrate formation. On the other hand, water bath heaters have a low thermal efficiency and cause to dramatically fuel consumption. The purpose of the study was to investigate the fuel consumption in CGS with the proposed system. For this purpose, employing a VT and VGHX was proposed. The system is installed before the heater.

It was observed that in pressure ratio of four and a cold mass fraction of 0.5, temperature of NG cold stream reaches below 0 °C. If the CGS continuously operates in this condition, the hydrate formation probability will be high and damage to CGS will be Imminent. For preventing this condition, cold mass fraction of 0.8 be selected to keep the cold stream temperature above 1 °C always.

For studying the effect of VGHX on energy conservation, 25 and 16 boreholes in square and L-shaped configurations were considered. The results showed that L-shaped configuration thermal performance is better than square shape. In other words, square shape boreholes absorb low heat relative to L-shape borehole configuration. Squared-shape boreholes are compact in comparison with L-shape ones and thermal effect of boreholes on each other will be higher than L-shaped configuration in a long time. The proposed system potential in reducing energy consumption was calculated up to 88%, and the payback period is always less than 4.5 years.

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