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Thermodynamic assessment and proposal of new configurations of an indirect water bath heater for a City Gate Station (a case study)

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ABSTRACT

This paper deals with thermodynamic modeling and proposal new configurations of an indirect water bath heater used in a city gate station (CGS) to preheat the natural gas (NG). In the transmission pipeline of natural gas before reducing NG pressure, it is necessary preheating to prevent hydrate formation. In the traditional systems, an indirect water bath heater is used for this purpose. Thermodynamic modeling of the proposed systems has been done in EES software, and an effective analysis based on proposed new configurations is conducted. The results indicate that using exhaust combustion products to preheat the combustion air can increase the thermal efficiency of the system from 46.36% (for base case) to 73.84%. In addition, with the use of an ORC to utilize the exhaust waste thermal energy, the system efficiency of 55.27% with an electrical output of 79 kW can be achieved. A comparison between configurations indicates that the second configuration (preheating combustion air) with 78.97 kg/h has the lowest emission rate.

1. Introduction

Natural gas passes long distances via transmission pipelines to receive a location from consuming by end-users. In order to have a suitable pressure in the consumption point, the pressure of the NG should be maintained at a relatively high value (about 1000 psi). Near the cities in a pressure reduction station, also known as City Gate Station (CGS), the pressure of the main line drops by throttle valves to a lower pressure (about 250 psi) which make the NG applicable for domestic and industrial use. Due to the Joule- Thomson effect, the NG pressure

reduction occurs along with temperature drop [1]. Since the NG contains suspended water droplets that become frozen at low temperatures (form hydrate), the stream of NG should be heated up to a temperature to prevent this phenomenon. Indirect water bath heaters (IWBHs) are the most commonly used heaters in gas stations. In an IWBH, the fire tube transfers the generated heat by a fuel gas-fired burner to an exothermic environment (such as water, a mixture of water and glycol, steam and salt) contained in the heater shell. Then, the required heat is transferred from the exothermic environment into the NG stream, which passes through a serpentine process coil comprising multiple "U" tubes. The thermal efficiency of the

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IWBHs is relatively low (0.35-0.55), and they require a high amount of the NG as fuel.

There are numerous studies that focus on energy recovery in CGSs, which can be categorized into two main groups. The first consists of research on the replacement of the gas-throttling process by an expansion process in a turbo-expander to convert the pressure of the NG into mechanical energy that can be transmitted to a loading device, for example, an electric generator, compressor, or pump [2-6]. The second group includes studies that attempt to reduce energy consumption in the heating process CGSs [7-11]. Farzaneh-Gord et al. [2] presented the NG properties at the inlets and outlets of all 29 Khorasan Province (Iran) CGSs for the entire year. They concluded that one could produce an average annual electrical power of over 9.9 MW using a turbo-expander. Sanaye and Mohammadi Nasab [3] proposed to replace the expansion valve with a combined heat and power (CHP) system consisting of an expander, gas engines, boilers, a pump, and a preheater. They used a genetic algorithm to find the maximum of an objective function named actual annual benefit ($), which was the sum of income (from selling electricity) and expenses (such as investment cost, operation and maintenance costs). Neseli et al. [4] investigated the power recovery from a CGS in Izmir (Turkey) for electricity generation using turbo-expanders. They evaluated the exergetic efficiencies of the system and components to specify their individual performances. It was reported that the maximum installed capacity and electricity that the system can produce are about 0.765 MW and 6366 MWh. Arabkoohsar et al. [5] proposed and evaluated a power recovery system for NG expansion, based on the integration of a solar heating system and a turbo-expander in order to reduce fuel consumption in Birjand CGS. They used solar engineering formulations and thermodynamics correlations to assess the locally available solar irradiation the NG availability, respectively. Net present value (NPV) analysis revealed that the payback period for the system is equal to 3.5 years. Tan et al. [6] offered a new NG liquefaction process to recover the energy contained in the high-pressure pipelines with pressure up to 10 MPa. The proposed process mainly included a turbo-expander driven booster, throttle valves, multi-stream heat exchangers, and separators. For the feed gas at the pressure of 10 MPa, they found that the highest value of total liquefaction rate and exergy utilizing rate at the optimal condition are 15.4% and 21.7%, respectively. Farzaneh-Gord et al. [7] evaluated the feasibility of using solar energy for preheating the NG in a CGS within Iran (Akand CGS) to decrease the amount of heater fuel consumption. The results revealed that the annual fuel saving is 24815$, and the payback ratio is 9 years. Ashouri et al. [8] presented a computational procedure for the calculation of the Joule-Thomson coefficient of NG to determine the minimum possible temperature of the NG entering to the pressure regulator of CGS. Using fundamental thermodynamic equations and AGA-8 equation of state, they proposed a temperature controller that could save energy consumption of heaters by 43%. Moreover, their economic analysis showed that the payback period is less than one year. Zabihi and Taghizadeh [9] investigated Akand City Gate Station, a relatively large station, to decrease fuel consumption as well as the cost of operation. They simulated the station in HYSIS software and proposed a temperature controller, which reduced energy consumption up to 430000 SCM. Ghezelbash et al. [10] offered an innovative system based on using a vortex tube and vertical ground heat exchanger. They used a vortex tube to separate the incoming high-pressure flow into two relatively lower pressure flows with temperatures higher and lower than the inlet flow. After leaving the vortex tube, the cold stream was heated by geothermal energy in a shell and tube heat exchanger. Then, the heated cold stream was mixed with the hot discharge steam from the vortex tube and went towards the heater at low pressure. Their results showed that the proposed system is capable of reducing energy consumption up to 88%, and the payback period is always less than 4.5 years. Farzaneh-Gord et al. [11] proposed a new system based on the vertical ground-coupled heat pump to eliminate in-situ fuel consumption at CGS stations. They

concluded that the system has the potential to eliminate in-situ fuel consumption of city gate stations.

In the present work, thermodynamic modeling of a CGS includes an IWBH to provide a better insight into its performance are performed. With the help of real data of four CGSs and analysis of their pressure and volumetric flow rate, different configurations for energy recovery of a case study are proposed. The aim of the present study is to introduce new arrangements to fully recover the waste energy of available gas-fired preheater of NG heater. The exhaust stream of NG heater with a high enthalpy level can be used to increase the air temperature required for combustion or used to produce electrical power. In one of the configurations, the exhaust flue as an input source is utilized to derive an organic Rankine cycle (ORC) from producing electricity. The other configuration includes an air preheater to enhance the air combustion temperature. In this way, a complete thermodynamic analysis is employed to examine various aspects of new proposed configurations and thermal and emission characteristics of systems investigated.

Nomenclature

a	theoretical air
c_p	constant pressure heat capacity (kJ/kg.K)
e	excess air
$\dot{E}_{fuel}$	fuel energy (kW)
$\dot{E}_{stack}$	stack energy loss (kW)
$\dot{E}_{surface}$	surface energy loss (kW)
emi	emission
$\bar{h}$	enthalpy (kJ/kmol)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (psi)
preh	preheater
$\dot{Q}$	thermal energy (kW)
T	Temperature (°C)
$\dot{W}$	output work rate (kW)
x	mole fractions

Subscripts

a	air
amb	ambient
ev	evaporator
f	formation
is	isentropic
prod	product
R	Rankine
Tur	Turbine

Greek symbols

α	molar fraction of methane
β	molar fraction of ethane
η	Efficiency

Acronyms

CF	confidence factor
CGS	city gate station
IWBH	indirect water bath heater
LHV	lower heating value (kJ/kg)
NG	natural gas
ORC	organic Rankine cycle
SCM	square cubic meters

2. Systems description

Figure 1 (a) illustrates a typical city gate station (CGS), which reduces natural gas (NG) pressure via the throttle process. The main flow of NG after passing through a dry gas filter is heated up by in an indirect water bath heater in order to increase its temperature to prevent hydrate and ice formation in the instruments passway. As it can be seen that three pass lines are provided in which two of them are in service with 50% capacity, and the other line is standby. In addition, Fig. 2 (b) represent two views of the Ilam city gate station.

As discussed earlier, two new configurations are proposed to recover the thermal energy of exhaust gas from IWBH, which dispatch to the environment in the conventional system. The first configuration (base case) includes a water bath heater, which indirectly heats the NG stream and exhaust gas in a high temperature leaves the heater. In the second configuration, the exhaust stream using a heat exchanger employed to preheat the air combustion. The third configuration includes an organic Rankine cycle to produce electricity from waste energy of exhaust gas. Figure 2 illustrates three arrangements for CGS and IWBH.
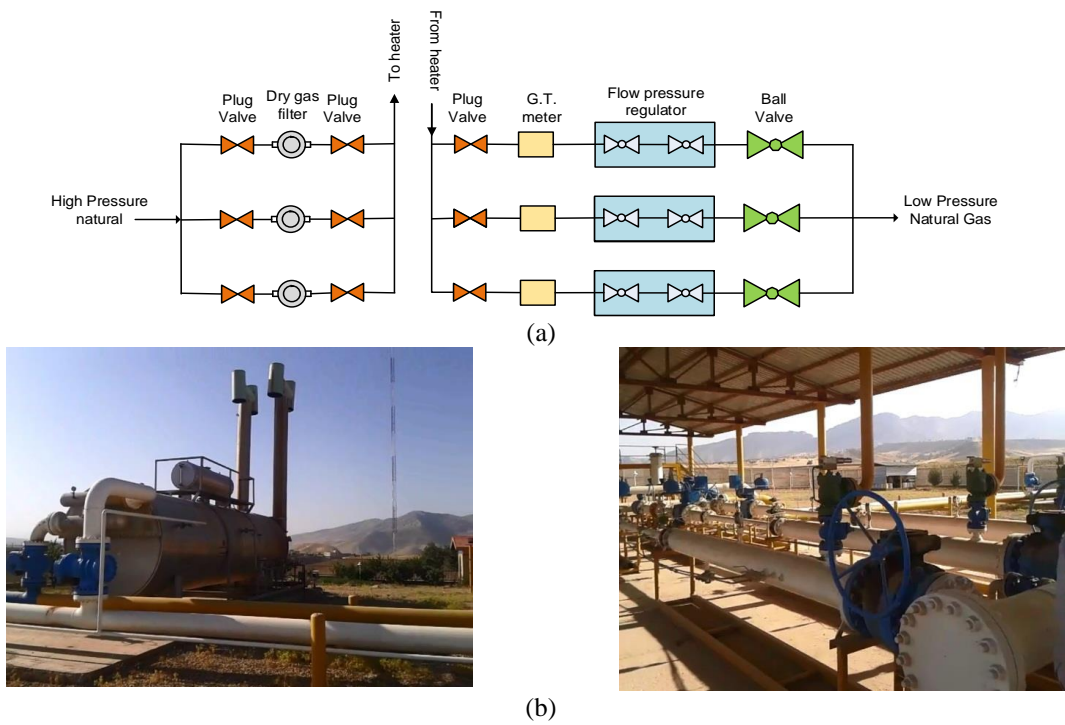
(a)

(b)

Fig. 1. (a) Schematic diagram of a typical city gate station (CGS), (b) Ilam City Gate Station
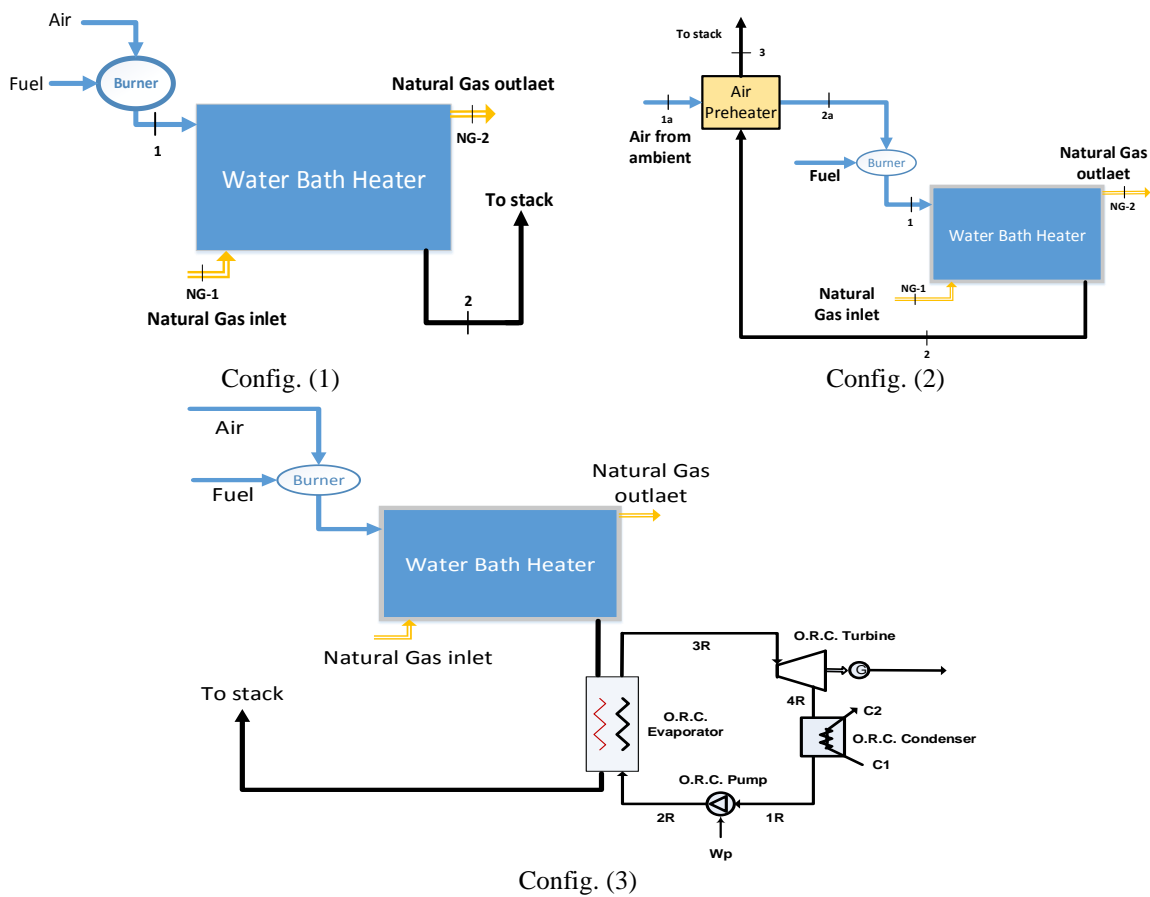

Config. (1)

Config. (2)

Config. (3)

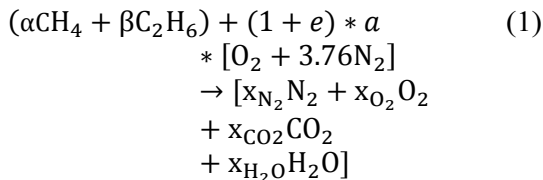**Fig. 2.** Schematic of proposed systems and base case to recover energy from the stack.

3. Modeling

3.1. General description

As discussed earlier, the current study focuses on various solutions to overcome low energy conversion efficiency in the CGSs, a location that has great potential to recover waste thermal energy. The mainstream of NG at the state NG-1 with high pressure (in a range between $\sim 5\ MPa$ and $\sim 7\ MPa$) and low temperature, depending on the environment, enters the IWBH to heat up before decreasing its pressure with a flow pressure regulator in the next stage of CGS. The NG warm up to a definite temperature, ensure preventing hydrate formation. As it can be seen in Fig. 1 (config. 1), the combustion product of natural gas after pass through the IWBH give its energy to the water and then dispatch to the stack. The experimental data shows the flue gas inlet to the stack at the temperature range between 360 and 480 ℃ [12]. Therefore, there is a suitable potential to use waste thermal energy. In the two proposed systems, this potential is used to increase the temperature of required combustion air and to produce more electrical energy.

• Combustion modeling

In the line heater, an atmospheric burner is employed to burn the natural gas as the source of energy. It is assumed it operates at the steady-state condition with constant pressure. Assuming complete combustion of natural gas with standard air, the general form of chemical reaction of combustion can be written as follows:

$$(\alpha CH_4 + \beta C_2H_6) + (1 + e) * a \qquad (1)$$
$$* [O_2 + 3.76N_2]$$
$$\rightarrow [x_{N_2}N_2 + x_{O_2}O_2$$
$$+ x_{CO2}CO_2$$
$$+ x_{H_2O}H_2O]$$

where α and β are a molar fraction of methane and ethane in natural gas, respectively, and e denotes excess air. Also, a is theoretical air necessary for complete combustion that can be obtained from the stoichiometric reaction. In order to obtain the exhaust gas compositions, an element balance has been used:

$$\alpha + 2\beta = x_{CO2} \qquad (2)$$

$$\alpha + 2\beta = x_{CO2} \qquad (3)$$
$$2(1 + e)a + 2\beta = 2x_{O_2} + 2x_{CO2} \qquad (4)$$
$$+ x_{H_2O}$$
$$3.76 * (1 + e)a = x_{N_2} \qquad (5)$$

From energy balance between reactant and product under steady-state condition without energy loss the required air for combustion can be calculated as bellow:

$$\left(\alpha \bar{h}^{\circ}_{f,CH_4} + \beta \bar{h}^{\circ}_{f,C_2H_6}\right) + (1 + e) * a \qquad (6)$$
$$* \left[(\bar{h}^{\circ}_{f,O_2} + \Delta\bar{h})\right.$$
$$+ 3.76(\bar{h}^{\circ}_{f,N_2} + \Delta\bar{h})\right]$$
$$= x_{N_2}(\bar{h}^{\circ}_{f,N_2} + \Delta\bar{h})$$
$$+ x_{O_2}(\bar{h}^{\circ}_{f,O_2} + \Delta\bar{h})$$
$$+ x_{CO2}(\bar{h}^{\circ}_{f,CO_2} + \Delta\bar{h})$$
$$+ x_{H_2O}(\bar{h}^{\circ}_{f,H_2O}$$
$$+ \Delta\bar{h})$$

which $\bar{h}^{\circ}_f$ is the molar enthalpy formation in terms of kJ/kmol, and $\Delta\bar{h}$ is enthalpy difference for the given state with reference state.

• Water bath heater

In the transmission pipeline of natural gas, the gas heater provides the required energy for the mainline before it expands in the expansion valve. The combustion products at high temperatures enter the IWBH and give their energy to water. The combustion product in the Point 2 (config. 1) leave the heater. The energy efficiency for an indirect water bath heater can be expressed as bellows:

$$\eta_{IWBH} = \frac{useful\ output\ energy}{input\ energy} \qquad (7)$$
$$= \frac{\dot{E}_{NG}}{\dot{E}_{fuel}}$$
$$= \frac{\dot{E}_{fuel} - (\dot{E}_{stack} + \dot{E}_{surface})}{\dot{E}_{fuel}}$$

Here, $\dot{E}_{Fuel}, \dot{E}_{stack}, \dot{E}_{surface}$ are energy of fuel, stack energy loss and surface energy loss. Using energy balance for inlets and outlets stream for heater control volume, the following equation can be written:

$$\dot{m}_{NG}(h_{NG-2} - h_{NG-1}) \qquad (8)$$
$$= \eta_{IWBH}[\dot{m}_{prod}(h_1$$
$$- h_2)]$$

where h_{NG-1} and h_{NG-2} are enthalpy inlet of cold natural gas to heater and enthalpy of the hot natural gas outlet of heater and h_1 and h_2 are enthalpies of combustion product inlet to heater and enthalpy of combustion product from heater.

• Air Preheater

In the second configuration, an air preheater is used to recover the waste energy of the heater. In the steady-state for the control volume of air preheater, the effectiveness of air preheater can be defined by [13]:

$$\varepsilon_{preh} = \frac{h_{2a} - h_{1a}}{h_2 - h_{1a}} \quad (9)$$

$$if \ (\dot{m}c_p)_{air} > (\dot{m}c_p)_{prod}$$

$$\varepsilon_{preh} = \frac{h_2 - h_3}{h_2 - h_{1a}} \quad (10)$$

$$if \ (\dot{m}c_p)_{air} < (\dot{m}c_p)_{prod}$$

Here, c_p is the thermal capacity at constant pressure and $\dot{m}$ denote the mass flow rate.

• Organic Rankine cycle

The organic Rankine cycle considered in this study includes four main parts. As it can be seen from Fig. 1 for configuration 3, the organic working fluid at state 1R enters the organic pump with an isentropic efficiency of $\eta_{is,p}$, after increase, its pressure leave the pump at the state 2R. Using the definition of isentropic efficiency, it can be written that [14]:

$$\eta_{is,p} = \frac{h_{2R,s} - h_{1R}}{h_{2R} - h_{1R}} \quad (11)$$

An organic evaporator is employed to recover the waste thermal energy of combustion product. In an organic evaporator, working organic fluid converts to vapor and reach the state of 3R. Due to heat transfer limitation and losses in the evaporator, it assumed that with η_{ev} the energy of hot stream exchange with the cold stream.

$$(h_{3R} - h_{2R}) = \eta_{ev}(h_3 - h_2) \quad (12)$$

3.2. Energy analysis

In the following section, an energy analysis for a typical CGS and modified stations has been done. The natural gas temperature in the inlet state of the gas heater directly depends on the ambient temperature. In fact, the transmission pipeline buried in the depth of 1.5 m underground and the NG temperature

equal to soil temperature at this depth. So, the inlet NG temperature can be expressed as follows [15]:

$$T_{NG-1} = 0.0048T_{amb}^2 \quad (13)$$
$$+ 0.3182T_{amb}$$
$$+ 11.403$$

Here, T_{amb} refers to ambient temperature in centigrade. To maintain the outlet temperature of natural gas in the desirable level, the heater outlet must be equal to:

$$T_{NG-2} = T_{hydrate} \quad (14)$$
$$+ \Delta T_{pressure \ drop}$$
$$+ \Delta T_{CF}$$

In which $T_{hydrate}$ and $\Delta T_{pressure \ drop}$ are respectively natural gas hydrate forming temperature and temperature drop due to throttle process, which is a function of inlet and outlet pressure of station and natural gas composition. Also, ΔT_{CF} is the confidence temperature difference which ensures natural gas temperature above temperature of hydrate formation. The confidence temperature difference is about 5 °C [15].

In order for a precise assessment of the systems, three criteria based on useful outputs is defined.

$$\eta_{system \ I} \quad (15)$$
$$= \frac{useful \ output \ energy}{input \ energy}$$
$$= \frac{\dot{m}_{NG}(h_{NG-2} - h_{NG-1})}{\dot{m}_{fuel}LHV}$$

$$\eta_{system \ II} \quad (16)$$
$$= \frac{useful \ output \ energy}{input \ energy}$$
$$= \frac{\dot{m}_{NG}(h_{NG-2} - h_{NG-1})}{\dot{m}_{fuel}LHV}$$

$$\eta_{system \ III} \quad (17)$$
$$= \frac{useful \ output \ energy}{input \ energy}$$
$$= \frac{\dot{m}_{NG}(h_{NG-2} - h_{NG-1}) + \dot{W}_{ORC}}{\dot{m}_{fuel}LHV}$$

Here $\dot{W}_{ORC}$ is the ORC work output and LHV is lower heating value of natural gas that is 48841kJ/kg for the gas compound in the studied region.

Organic Rankine cycle is usually used to recover energy from low-temperature waste heat. Working fluid selection directly affects the organic cycle performance. In order to select an organic working fluid, both physical

properties and environmental impacts should be considered. The organic cycle output is an important criterion. The mathematical model to calculate the generated work by ORC turbine with an isentropic efficiency of $\eta_{is,tur}$ can be expressed as [16]:

$$\dot{W}_{ORC} = \dot{m}_{ORC}(h_{3R} - h_{4R}) \qquad (18)$$
$$= \frac{\dot{m}_{ORC}(h_{3R} - h_{4Rs})}{\eta_{is,tur}}$$

The thermal efficiency of ORC can be employed as another criterion for fluid selection. The thermal efficiency of ORC is the ratio of ORC output work to the heated feed to the ORC system and can be written as follows:

$$\eta_{ORC} = \frac{\dot{W}_{ORC}}{\dot{Q}_{ev}} \qquad (19)$$

where, $\dot{Q}_{ev}$ is the amount of heat transfer from gas product to the organic working fluid.

3.3. Emission characteristics

The emission of CO_2 can be measured with the amount of total CO_2 mass exhaust to environment. The mitigation of CO_2 in proposed systems are normalized based on the useful product in the term of thermal and electrical produced energy as bellows:

$$\dot{m}_{CO_2} = x_{CO2} * \dot{m}_{prod} \qquad (20)$$

$$Emi_{CO_2, system\ I} \qquad (21)$$
$$= \frac{\dot{m}_{CO_2}}{\dot{m}_{NG}(h_{NG-2} - h_{NG-1})}$$

$$Emi_{CO_2, system\ I} \qquad (22)$$
$$= \frac{\dot{m}_{CO_2}}{\dot{m}_{NG}(h_{NG-2} - h_{NG-1})}$$

$$Emi_{CO_2, system\ III} \qquad (23)$$
$$= \frac{\dot{m}_{CO_2}}{\dot{m}_{NG}(h_{NG-2} - h_{NG-1}) + \dot{W}_{ORC}}$$

where Emi_{CO_2} is normalized CO_2 emission in kg/kWh.

4. Result and discussion

In this study, the feasibility of waste energy recovery from the water bath heater of Ilam CGS has been studied. Ilam Province is located in the western part of Iran with heavy showers or snow in winter and brutally hot, dusty in the summer. The average temperature, average high temperature, and average low temperature of the case studies region are presented in Fig. 3. Here the average maximum temperature and the average minimum temperature indicate the average value of daily maximum or minimum temperature over a month while maximum and minimum temperature show the highest and lowest value of temperature in a month.
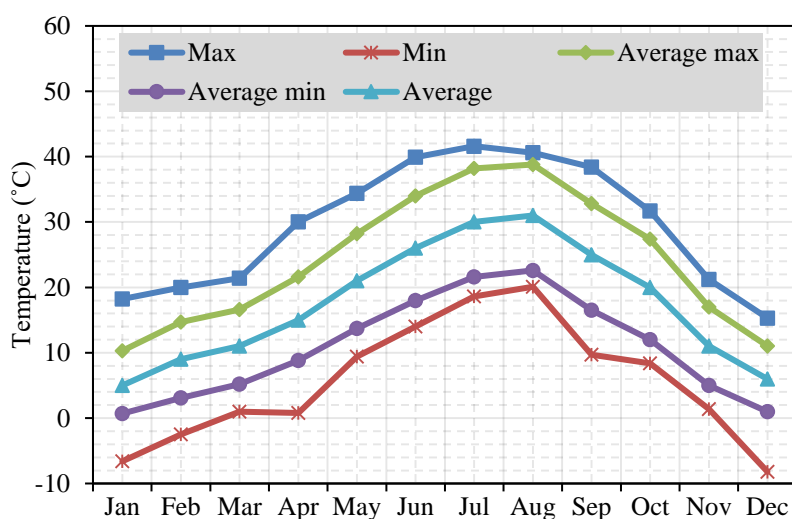

Fig. 3. The monthly average temperatures of Ilam city for 2015 [17].

This province has a precious resource of oil and natural gas. It is estimated that 11 percent of the oil and gas resource of Iran located in this area [18]. Widespread transmission pipelines in this area is a unique opportunity to use the potential of waste energy in the pipeline network.

Four stations are nominated to select one of them to study the modification on the water bath heater. Ilam, Ivan, petrochemical and cement factory considered as four case studies. Figure 4 represents the natural gas flow rate through these stations for a year. Analysis of available data shows that for Ilam

and Ivan as domestic use stations in the cold month, the flow rate is higher than the hot month. On the other side, petrochemical and cement factory as industrial stations in warm months have a higher consumption rate.

In addition, the variation of pressure inlet of CGS for case studies is illustrated in Fig. 5. As it can be seen except for cement factory station the pressure variation of all CGS are in the same range. The cement factory line pressure is 250 psi due to the fact that this line should directly feeds the industrial instrument with this pressure range.
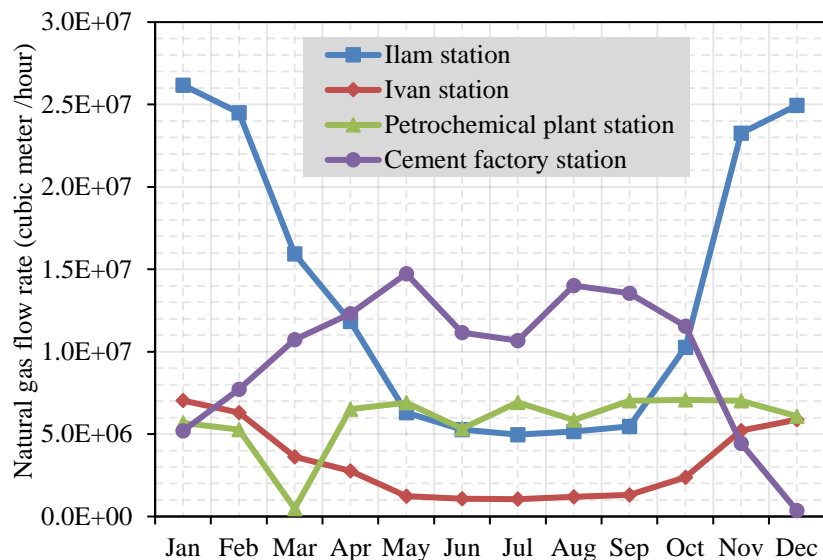

Fig. 4. The monthly natural gas flow rate of four CGS case studies 2015.
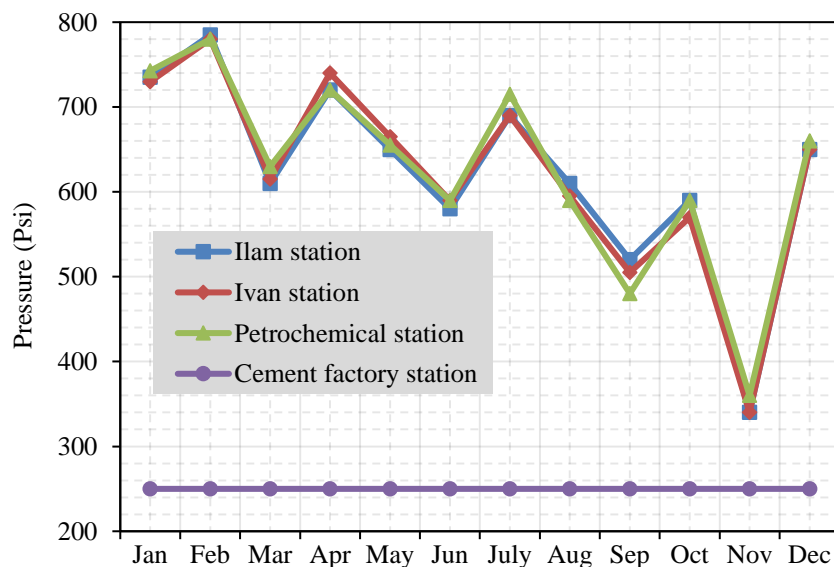

Fig. 5. Monthly pressure variation of four case studies in 2015.

Based on flow rate and pressure variation during a year, Ilam CGS is selected for detailed investigation. Table. 1 represents the main data for the reduction pressure of the Ilam city gate station.

Figure 6 represents the ambient temperature and natural gas inlet temperature obtained from Eq. (13). Results exhibit the natural gas temperature inlet to the station has a trend like ambient temperature due to the fact that it is a polynomial function of ambient temperature.

4.1. Configuration I (Basic case)

In the first configuration, which is the basic design for a water bath heater, after heat transfer with a water bath in the heater, the combustion products leave the heater in relatively high temperature between 360 ℃ and 480 ℃. As it can be seen from Fig. 7, the stack loss and required heat for the natural gas mainline with increased ambient temperature shows an increment. In addition, the result for CO_2 emission reveals that for the change the natural gas flow rate through a year, the CO_2 emission of the station can change between 45.69 kg/h and 240.8 kg/h.

Figure 8 represents the fuel consumption of the station for average monthly temperature in different months of year. As the results show, the fuel mass flow rate in August has the minimum value, which directly goes back to the ambient temperature and natural gas mainline inlet temperature. (See Fig. 6 for variation of ambient and natural gas temperature).

Table 1. Ilam city gate station characteristics.

	Capacity	Dry gas filter			Regulator valve			Water bath heater		
		Size	Working class	No. filters	Size	Working class	No. filters	Working pressure	Bath water capacity	No. of coil
Ilam CGS	100000 (SCM/h)	8 (inch)	600	3	4 (inch)	600	6	1050 (pisg)	38000 (lit)	7
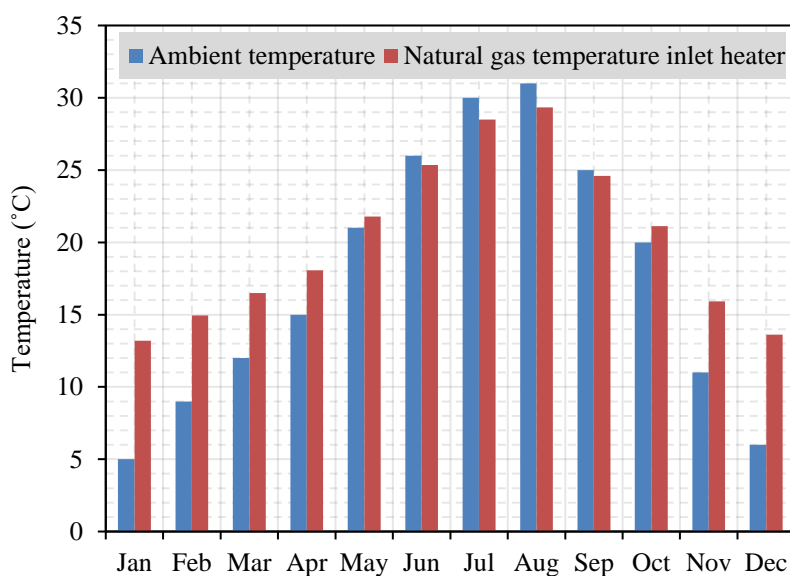
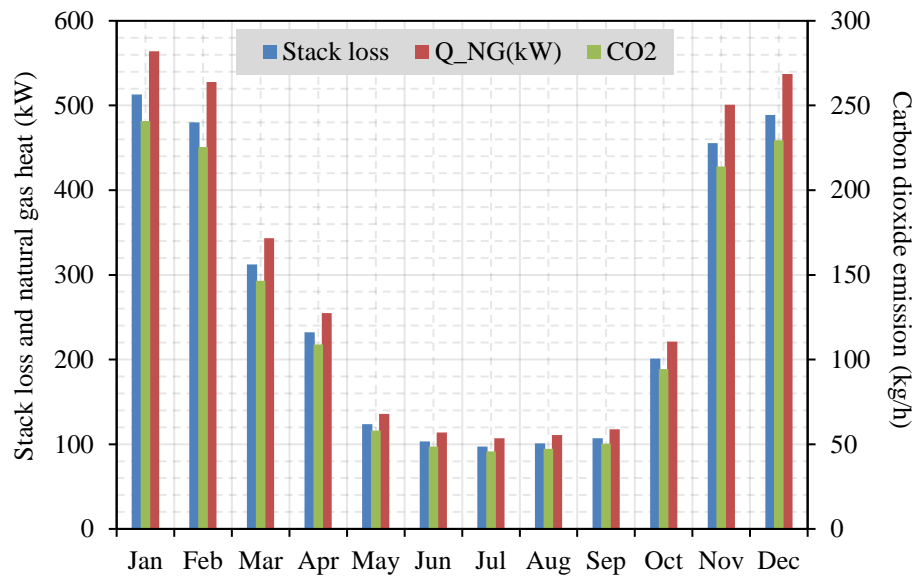**Fig. 6.** Temperature variation of inlet natural gas to Ilam CGS in 2015.

Fig. 7. Thermal and emission characteristics of IWBH in the first configuration.
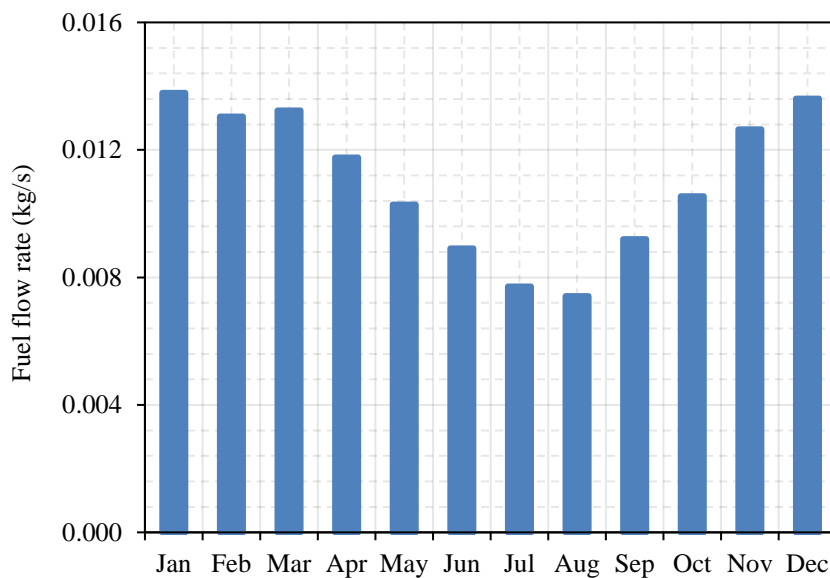

Fig. 8. Fuel combustion for Ilam CGS for different month and average high, low and mean temperature.

One of the substantial parameter, which controls the flame temperature and gas product temperature, is excess air. Figure 9 shows the change of two main outputs with combustion excess air change. As it can be seen, an increment in the excess air leads to a decrement in flame temperature. In addition, the results show that with increment of the excess air from 50% to 150% the stack loss increase from 115.2 kW to 232.1 kW. One of the main reasons to increase stack loss is an increment of combustion product mass flow rate. More analysis reveals that excess air increment in the range of 50% to 150% leads to an increase in the mass flow rate of combustion products from 0.236 kg/s to 0.49 kg/s, which substantially affect the stack heat loss.
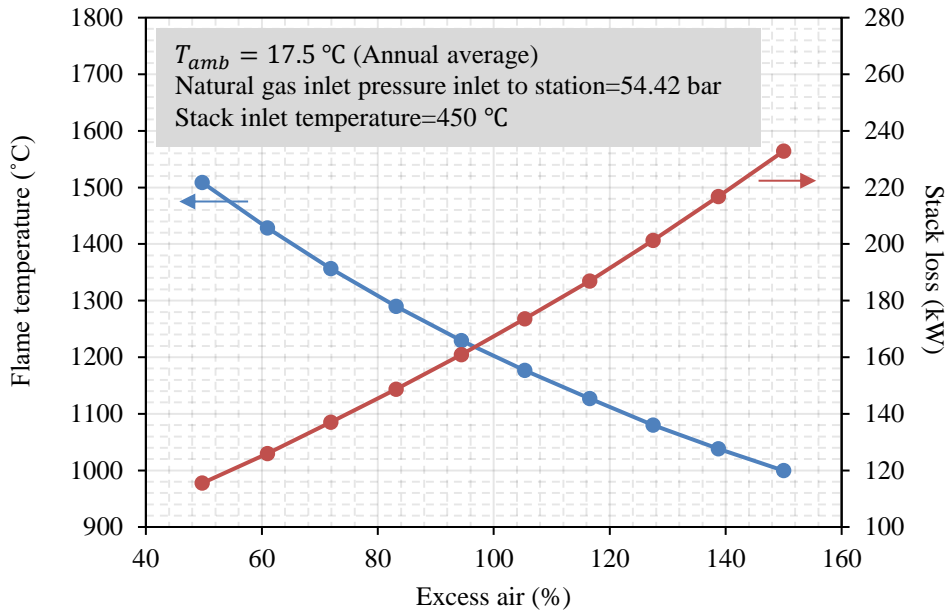
Fig. 9. Flame temperature and stack loss variation with the change of excess air.

The excess air also affects the system efficiency and emission characteristics of the system. As Fig. 10 shows at the temperature ambient of 17.5 °C and natural gas pressure inlet to the station of 5.442 Mpa, increase the excess air from 50% to 150% leads to decrease the efficiency of heater from 57.2% to 46.4%. In addition, the CO_2 emission of the system with the excess air changes can increase from 87 kg/h to 111 kg/h.

It can conclude that the lower excess air leads to decrease in heater efficiency while increases the stack loss and emissions. The stack decrement trend with increase excess air is more important when the exhaust combustion of gas product utilizes to derive ORC in configuration III.
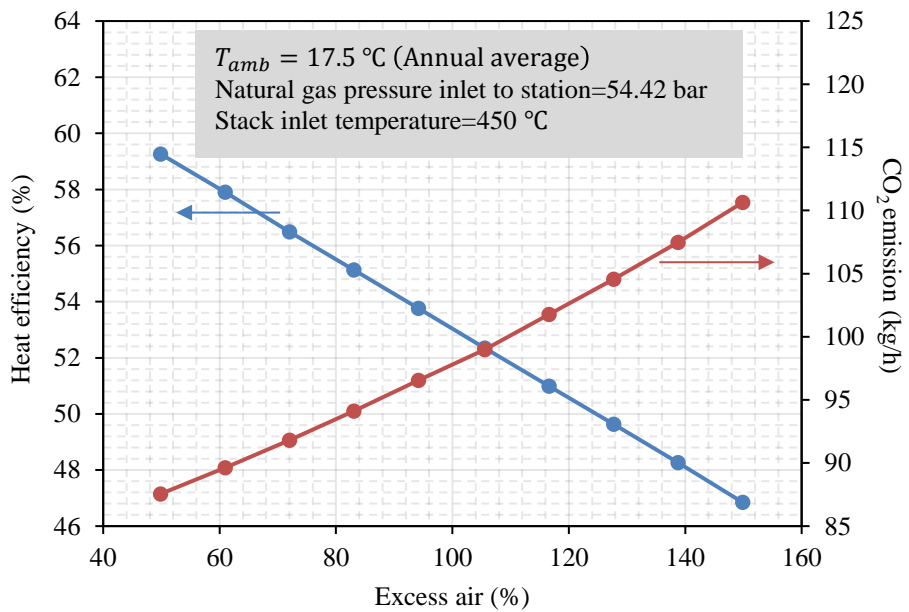

Fig. 10. Variation of heater efficiency and CO_2 emission with the change of excess air.

4.2. Configuration II

The stack loss is a significant parameter, which its effects are more considerable in cold weather. In the second configuration, the potential of stack waste heat to warm up the required combustion air is investigated. For annual average maximum, annual average minimum and annual average temperature, the variation of system efficiency is presented in Fig. 11.

As the results indicate in the second configuration, with increase excess air a decrement in the system efficiency takes place. It can be seen that for excess air increment from 50% to 200%, the stack losses increase from 4.534 kW to 4.668 kW and fuel mass flow rate from 20.99 kg/h to 21.65 kg/h. Moreover, it can be found that at higher

ambient temperature, the thermal efficiency of the system is higher.

Two main causes of efficiency drop in the lower ambient temperature are inlet NG temperature and stack losses. The NG inlet temperature to the station directly depends on the ambient temperature and experience an increment with the rise of ambient temperature (See Fig. 6). With a decrease the NG inlet temperature to the station, the higher energy requires to achieve a suitable temperature in the IWBH outlet and consequently, the fuel mass flow rate increase as presented in Fig. 12. As it can be seen from Fig. 12 the stack losses of system decrease from 14.31 kW in ambient temperature of 5.51 ℃ to 4.8 kW in ambient temperature of 24.22 ℃ which lead to increase the thermal efficiency of system.
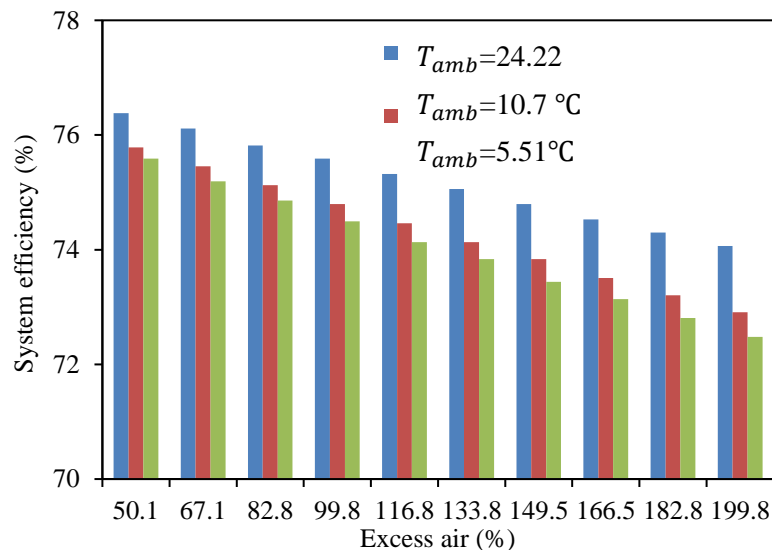

Fig. 11. System efficiency variation with excess air change for the annual average maximum, annual average maximum and annual average temperature.
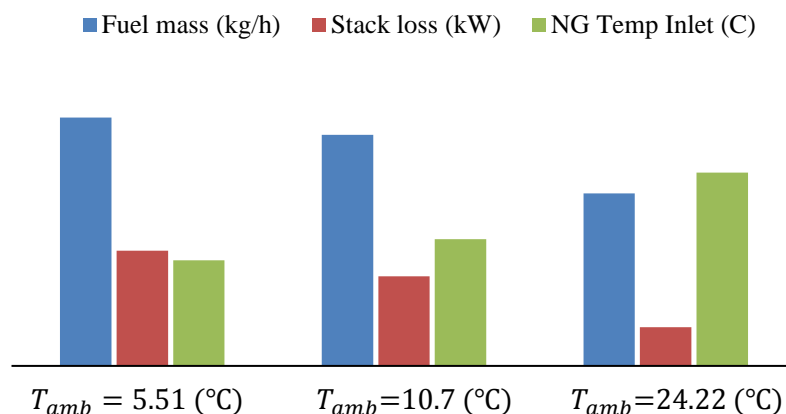

Fig. 12. Change of fuel mass flow rate, stack loss, and NG inlet temperature to station with ambient temperature.

4.3. Configuration III

In the third configuration, an ORC is employed to recover the waste heat energy of the water bath heater in CGS. Working fluid selection is so important for ORC performance and severely affect the efficiency and output of the system. The exhaust temperature and outlet stream enthalpy of gas products are two important values in working fluid selection. In this study it assumed that the ORC operates with a maximum pressure of 1000 kPa and condensation temperature of 40 ℃.

Figure 13 illustrates a schematic of T-S diagram for four selected working fluids. Because a wet fluid like water has a large latent heat of evaporation properties were not included here.

Also, the combustion product's exhaust temperature, which feed the ORC evaporator, is 450 ℃. The source temperature and working fluid pressure are two main criteria

in selecting a fluid for ORC system. Using these two facts, Table 2 represents the main properties of selected working fluids.

The higher critical temperature and pressure of working fluid is an advantage in selecting a working agent. However, here the net produced output work and ORC efficiency are employed to determine the best working fluid among different options. As it has been shown in Table 2, the net produced work and ORC efficiency for Toluene as the working fluid are 56.56 kW and 22.65%, respectively, which is higher than other fluids. Moreover, the high critical values of pressure and temperature is another benefit to select Toluene as a working medium in this study.

Figure 14 represents the net produced work of ORC. At the higher ambient temperature, the produced work by ORC decreases due to the stack loss, as it has been shown.
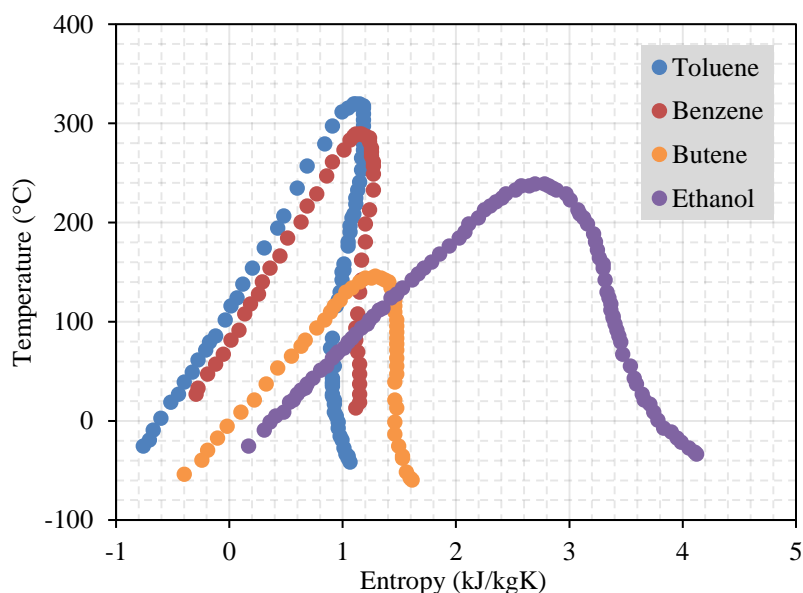

Fig. 13. Schematic of T-S diagram for selected organic working fluids.

Table 2. Properties of selected organic working fluids [19-23].

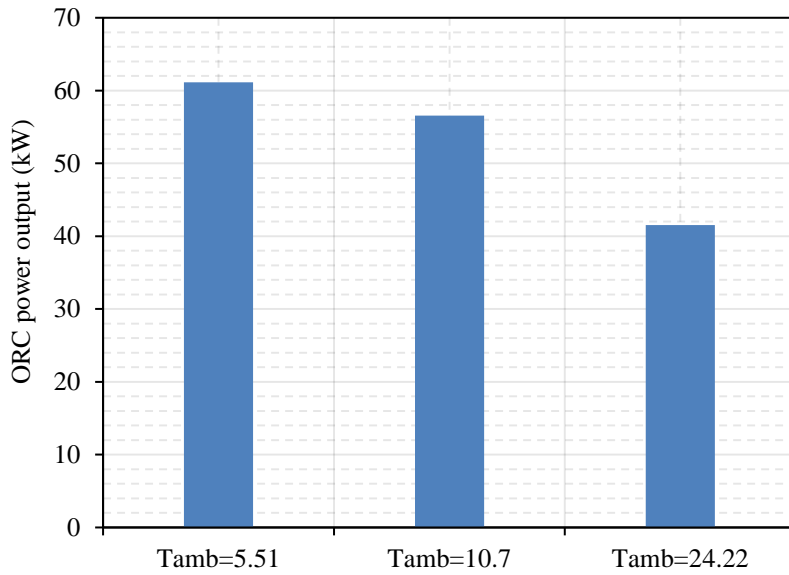| Fluid type	Molecular weight (kg/kmol)	Boiling point (℃)	Viscosity (mPa.s)	P_c (bar)	T_c (℃)	$\dot{W}_{ORC}$ kW	$\eta_{I,ORC}$ (%)
Benzene	78.11	80.08	0.604 @ 25 ℃	48.8	298	51.01	20.42
Toluene	92.14	110.6	0.56 @ 25 ℃	41.3	319	56.56	22.65
Butene	56.11	-6.47	0.186 @ 25 ℃	37.9	152	16.61	6.64
Ethanol	46.07	78.29	1.074 @ 20 ℃	40.6	241	46.12	18.47

Fig. 14. ORC net produced work in different ambient temperature.

In the third configuration, also the effects of excess air variation on the system performance investigated. As illustrated in Fig. 15, increase excess air leads to an increase both ORC net produced work and stack loss while decreasing the heater efficiency. However, increase excess air has a positive effect on the ORC output, it increases the stack loss and fuel mass flow rate that lead to a decrease the system efficiency. Results exhibit that with increase excess air from 50% to 200% the fuel mass flow rate increase from 35.97 kg/h to 52.95 kg/h.

4.4. Comparison of configurations

Figure 16 compares three configurations from system efficiency and stack heat loss point of view. As it expected, the first configuration, due to large part of stack heat loss (267.9 kW) has the lowest efficiency. In the second and third configuration, the thermal efficiency of the system due to recover waste heat energy increase to a reasonable value.
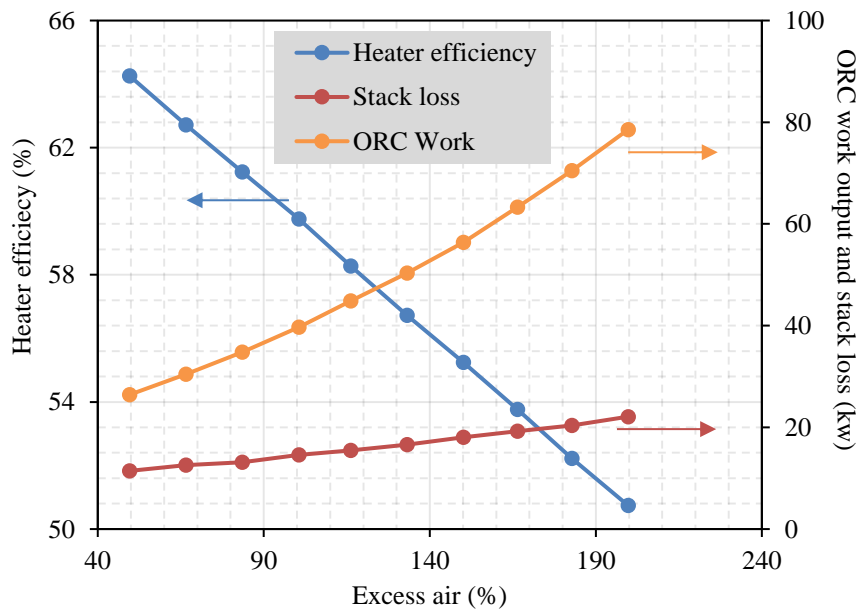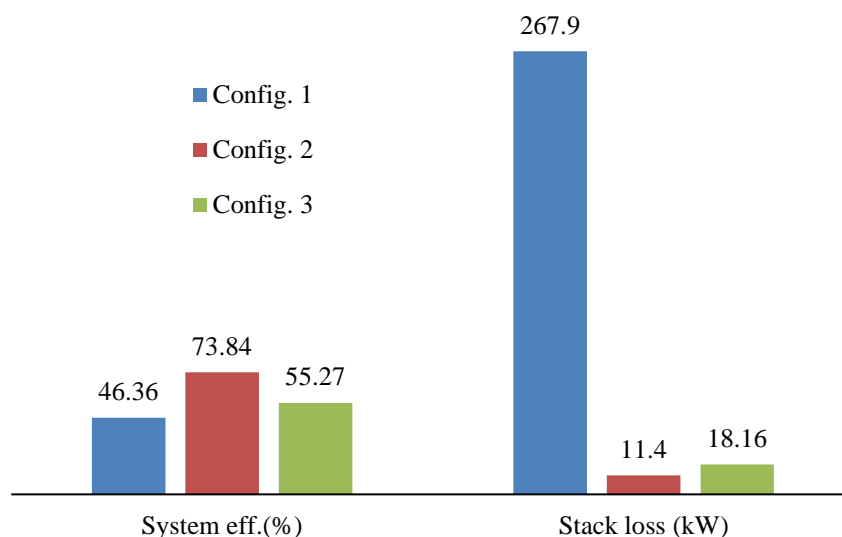

Fig. 15. Heater efficiency ORC output and stack loss with excess air variation.

Fig. 16. Variation of system efficiency and stack loss for different configurations of indirect water bath heater from.

In the second configuration, a large part of waste heat from the stack is used to heat up required air to combustion. Hence the fuel mass flow rate severely shows a decrement. As it can be seen in Fig. 17 the CO_2 emission also mitigate with preheating the combustion air effectively.

Although the emission per hour for the first configuration (base case) and third configuration has the same value, it should be noticed that in the third configuration, beside the NG heating with a fixed fuel mass flow rate, some work generated via ORC that indicate the comparative advantage of the third system.
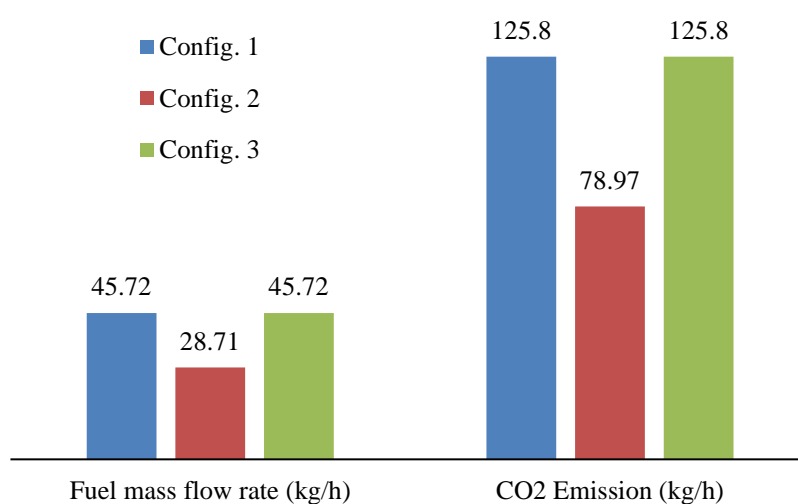
Fig. 17. Fuel mass flow rate and emission of CO_2 for different configurations.

5. Conclusion

In this work, real data of selected city gate stations has been analyzed and a station selected for more investigations. Three system configurations have been studied to find the best arrangement of an in-use CGS. The suggested systems have been utilized to assess revision, such as preheating the air combustion and stack loss energy recovery to produce electrical energy. The results indicated that for the first configuration (base case), second configuration (preheating air combustion), and a third configuration (ORC to recover stack loss), energy efficiency is 46.36%, 73.84% and 55.27%, respectively. Through the results analyses process, it conceives that for the base case, a large part of energy waste in the stack that severely reduce the thermal efficiency of system. Numerical calculation shows the waste thermal energy for base configuration is 267.9 kW. Therefore, the second and third arrangements are proposed based on this suitable potential to recover waste energy.

Excess air is one of the main parameters, which control the flame temperature and substantially affect the system performance. It concluded that with the increase the excess air, the stack loss for three configurations has been increased. The results of carbon dioxide emission represent that the second configuration with 78.97 kg/h has the lowest emission, which mainly due to preheating the combustion air.

In the third configuration, the working fluid selection procedure reveals that Toluene is the best working fluid medium based on work output and thermal efficiency of ORC. The analysis also shows that with the use of ORC to recover waste heat energy, in 200% excess air, 79 kW of electrical energy can be generated. Thermodynamic analysis of on the new proposed configurations and traditional configuration confirm the new proposed systems to utilize are more efficient.

Acknowledgment

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