EXPERIMENTAL INVESTIGATION OF THE INCREASING THERMAL EFFICIENCY OF AN INDIRECT WATER BATH HEATER BY USE OF THERMOSYPHON HEAT PIPE

by

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Natural gas must be preheated prior to pressure reduction in city gate stations. Indirect water bath heaters are employed in city gate stations for preheating, consume a considerable amount of natural gas for the process. This type of heater has a low efficiency therefore a significant amount of energy is wasted. Due to the high capacity of thermosyphon heat pipe, its utilization in city gate stations heater were investigated experimentally in this paper. For this purpose, a heater set-up was manufactured and its thermal efficiency was calculated. The thermosyphon heat pipe were then designed, manufactured, and utilized between the fire tube and the gas tube. Further the type of working fluid and the range of filling ratio were discussed and the most effective state was suggested. Moreover, the thermal efficiency of the heater in the presence of thermosyphon heat pipe was investigated. The obtained results showed that the thermal efficiency of the heater improved up to 13% with the addition of thermosyphon heat pipe. The most effective state of thermosyphon heat pipe was associated with the water as working fluid with 20% filling ratio in the front route and methanol as working fluid with 30% filling ratio in the back route of the fire tube.

Key words: *heater, thermosyphon heat pipe, working fluid, filling ratio, thermal efficiency*

Introduction

The energy crisis and increase in the energy demand have obligated scientists and engineers to seek ways to recover energy from useless sources. Natural gas (NG) is a significant source of energy and widely utilized in various applications. In most city gate stations (CGS), NG pressure is reduced through the regulator. In regulators due to the Joule-Thompson (J-T) effect, pressure drop causes temperature reduction. Therefore NG is heated by indirect water bath heater (IWBH) to prevent hydrate formation and liquefaction at the outlet station water is the intermediate fluid in the IWBH and plays the role of transmitting the thermal energy in a fire tube to a cool fluid by natural convection mechanism. The IWBH – referred to as *heater* throughout this paper at present, these heaters are widely employed in heating NG in the gas

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industry. The low thermal efficiency and high consumption of large amounts of fuel to provide the required heat are the main operational problems of these systems. Therefore, any decrease in gas consumption provides substantial economic benefit for gas companies.

Over the recent years, many investigations have been conducted on heaters and several methods have been proposed for recovering this dissipated energy. Azizi *et al.* [1] studied the energy loss from heater stack so as to reduce heater gas consumption. They suggested the use of a heat exchanger to preheat the gas prior to entering the heater, observing 11% improvement in heater thermal efficiency with a payback period of 1.2 years for heat exchanger installation. Farzaneh-Gord *et al.* [2] investigated the feasibility of using a solar set-up with uncontrolled heaters. They showed that the use of solar energy leads to 100 kWh of energy savings, amounting to a saving of $27000. Poživil [3] simulated a CGS regulator using the HYSYS software and presented a temperature reduction of about 4.5-6 °C for each 1 MPa gas pressure reduction in the regulator. Khalili *et al.* [4] calculated the heat loss and thermal efficiency of the Shahr-e Kord, Iran CGS heater. They revealed that over a year, more than 38% of the energy was wasted only through the exhaust gases from the heater stack. Ashouri *et al.* [5] computed the J-T coefficient of NG entering a CGS. They found that utilizing a controller to set the temperature of the inlet gas entering the station reduced the energy consumption of the heater by 43% with a payback period of less than 1 year. Howard *et al.* [6] examined the installation of a turboexpander in a small CGS in Canada. They concluded that the use of a turboexpander at a CGS with a capacity of 12000 SCMH would increase efficiency by almost 10%. Taheri *et al.* [7] calculated the NG energy lost through CGS regulators with a nominal capacity of 420000 SCMH. The results showed that by using the regulators, the annual amount of NG energy lost from the J-T valve was approximately 40 GWh. Farzaneh-Gord and Deymi-Dashtebayaz [8] did a wide feasibility study on generating the required electricity for the Khangiran, Iran gas refinery from its pressure reduction station. Their results indicated that the available energy overcome electrical demands requirement of the refinery. Zabihi and Taghizadeh [9] simulated the Akand, Iran CGS in HYSYS software in order to specify the amount of decrease and the economic benefits of installation of the new controller using Peng-Robinson (P-R) and SRK equations of state. The results indicated that P-R equation showed less deviation from the experimental data. Riahi *et al.* [10] optimized the combustion efficiency in the CGS heaters in Ardabil, Iran, observing that the regulation of burner and using barometric damper reduced heat losses and increased efficiency. Kostowski and Uson [11] proposed and evaluated an innovative system for exergy recovery from NG expansion based on the integration of an organic Rankine cycle and an internal combustion engine and peresented a favorable exergy efficiency of up to 52.6 % and maximum ratio of power generated to combustion (0.69-0.77%).

Heat recovery, as one of the energy saving technologies, not only conduces to saving energy but also reduce the adverse environmental impacts of waste heat by reducing the amount of CO_2 released to the environment [12]. Pandiyarajan *et al.* [13] investigated heat pipes in modern heat exchangers. This paper reviewed mainly heat pipe developments in the Former Soviet Union Countries. Srimuang *et al.* [14] reviewed the applications of heat pipe heat exchangers for heat recovery. This review article provided additional information for the design of heat pipe heat exchangers with optimum conditions in the heat recovery system. Weng and Leu [15] investigated on the performance of thermal thermosyphon solar water heater. They found experimentaly that such system can provide ample energy to satisfy the demand for hot water. The present experimental study utilized a THP to recover energy from CGS heaters. Heat pipes are ruggedly built and can withstand high levels of abuse [16]. Because of their great heat transfer capability, the application of heat pipes has been widely studied. The importance of heat pipes as devices for effective heat transfer has led to the development of various types such as loop heat pipes [17], wick type heat pipes [16, 17], THP [18], switchable heat pipes [16], pulsating heat pipes [19], variable conductance heat pipes [20]. The THP also called two-phase close thermosyphon, is a gravity-assisted wickless heat pipe, using the evaporation and condensation of the inner working fluid to transfer heat. The THP offers many advantages such as simple structure, low thermal conductance, high efficiency and low production cost [21]. Hence, it is widely used in a number of fields, including industrial heat recovery, electronic component and solar energy systems. Thermophysical properties of working fluid, filling ratio (FR), inclination angle, structure and geometry considerably influence the performance of THP [22, 23].

As seen, there are many studies in the literature focusing on theoretical and experimental energy performance and the thermal efficiency of CGS heaters. However, to the best of our knowledge, studies on the use of THP in CGS heaters are limited, hence the objective of this research to construct an experimental procedure to evaluate the effect of THP performancein heater of CGS.

Experimental set-up and test procedure

Fabrication of thermosyphon heat pipe

The THP shown in fig. 1(a) can be divided into three sections: evaporator, adiabatic, and condenser. When heat is added at the evaporator section, the working fluid inside the heat pipe vaporizes and carries heat from the heat source to the condenser section where heat is rejected to the heat sink. In this study a THP made of a smooth copper tube with a total length of 500 mm with inside and outside diameters of 7 and 8 mm, respectively, was designed and manufactured by considering the effective parameters on THP thermal performance. The details of experimental set-up for charging THP are shown in fig. 1(b). The THP was cleaned with the purified water to prevent the oxides formation. Also it was important to check for leaks at each connection. After THP fabrication, it was required to charge of working fluid. Before charging the THP with working fluid, the air and non-condensable gas was evacuated by a vacuum pump.

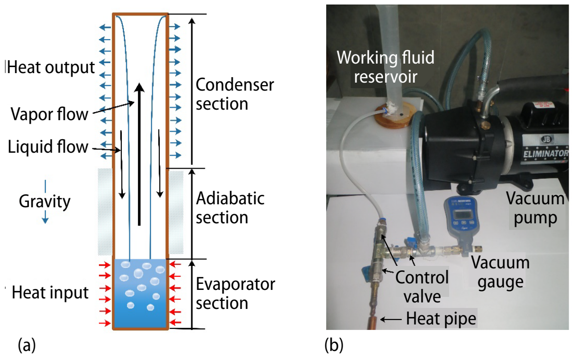


Figure 1. (a) Schematic diagram of typical THP, (b) detailed experimental set-up for charging working fluid

Explanation of experimental test device

A schematic and photograph of the experimental set-up used in the current study is represented in figs. 2 and 3. The set-up is contained of a heater, eight THP in different positions on the fire tube, a gas flow meter for measuring fuel consumption and 10 *K*-type thermocouples: 6 thermocouples were positioned at equal distances apart at the fire tube, T_{1-6}, two thermocouples in the inlet and outlet gas tube, $T_{7,8}$, one thermocouple in water, T_9, and one thermocouple in the stack, T_{10}. In order to measure air velocities, an anemometer was employed, figs. 2(b) and 2(c), illustrates the view of the THP in contact with the outer surfaces of the fire and gas tubes.

After preparing the experimental set-up, heat was applied to the device. For ideal combustion, it was necessary to use the excess air in the fueling route, hence used of an air com-

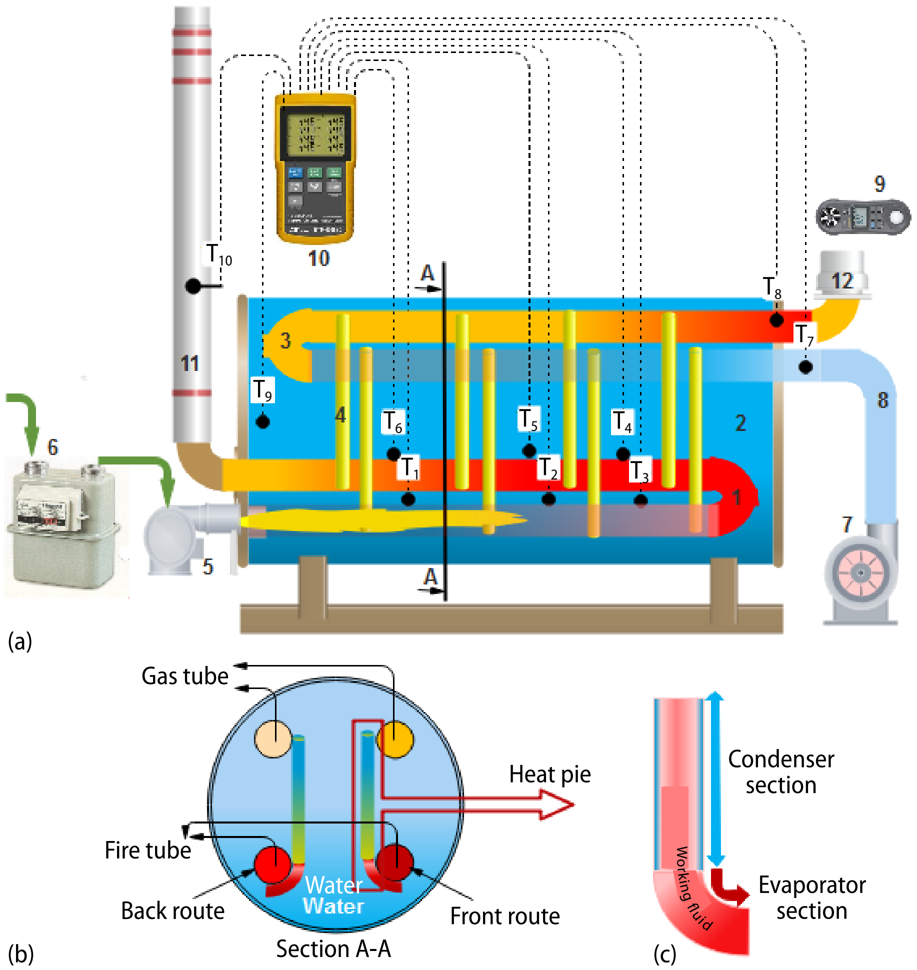


Figure 2. (a) Schematic of the experimental set-up: *1 – fire tube, 2 – water, 3 – air tube, 4 – THP, 5 – burner, 6 – fuel counter, 7 – blower, 8 – air inlet pipe, 9 – anemometer, 10 – temperature data logger, 11 – stack*, **(b) detailed of section A-A, and (c) schematic of used THP**

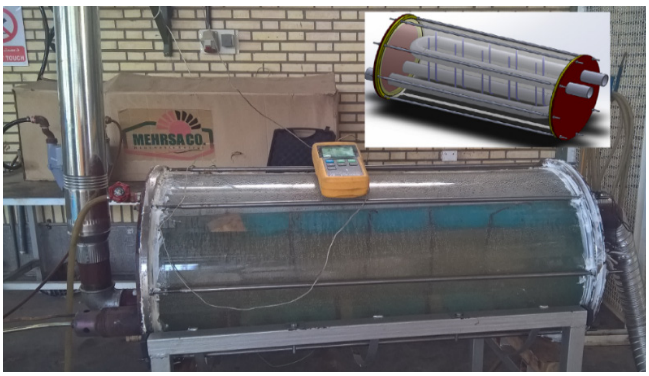


Figure 3. Photograph of the experimental apparatus

pressor parallel with the fuel path. During all experiments, combustion gas compositions were kept constant by adjusting the inlet air valve. For this purpose, combustion products were controlled by gas analyzer device in order to achieve the requirements of tab. 1. All tests were performed under the same experimental condition for the same range of heat input and to avoid measurement errors, the pressure and temperature of the fuel were kept constant during the tests. In all experiments, the mass-flow rate and temperature of inlet gas, T_7, were the same. Specifications of equipment and instrumentations used in the experiments presented in tab. 2.

Table 1. Limit values of combustion products for ideal combustion

Combustion product	Value	Unit
CO	< 100	[mgkW^{-1}hr^{-1}]
NO_x	< 170	[mgkW^{-1}hr^{-1}]
Excess air	40	[%]

Table 2. Specification of equipment and instrumentation

Name of instrument	Unit	Range of instrument	Model	Variable measured	Least division in measuring instrument	Min. and max. measured in experiment	Uncertainty [%]
Temp. data logger	[ms^{-1}]	100-1300 °C	Lutron/ BTM-42085D	Temperature	0.1	10-10.5	0.095
Anemometer	[ms^{-1}]	0.4-30	Lutron/ Lm9000	Air velocity	0.1	13.7-14.2	0.7
Vacuum gauge	[Pa]	0-1600	SVPCO/VGA	Absolute pressure	1	25-35	0.002
Gas flow meter	[m^{3}hr^{-1}]	0.5-6	Elster/G4	Fuel flow rate	0.01	0.5-2	0.005

Selection of working fluid

Selection of a working fluid is directly depending on heat transfer ability and compatible with the pipe material. It is further necessary to consider thermal stability, vapor pressure (not too high or low) over the operating temperature range, high latent heat and high thermal conductivity [24]. The THP normally uses water as it has a high working temperature range (50-200 °C) and high latent heat of vaporization (LHV) [25]. But in the present study, due to the change heat flux in the fire tube, it was necessary to select the appropriate working fluid. By increasing the heat input, a large amount of working fluid evaporates, resulting better heat transfer and consequently, decrease in thermal resistance [26]. Moreover, at low heat inputs, the thermal resistance increases. The optimal working fluid to ensure a small temperature difference between condenser and evaporator is one that can achieve the maximum rate of heat transfer at same temperatures. Therefore, it is necessary to experimentally determine the appropriate working fluid. As shown in fig. 2, six thermocouples T_1-T_6 were mounted on the fire tube at different positions. When the water temperature became steady, the temperatures in the vicinity of the fire tube wall were recorded, tab. 3. Noting that the mechanical vacuum pump can create an absolute pressure of 3 kPa, the saturation temperature of various working fluids corresponding to this pressure was investigated. Finally, distilled water and methanol were selected as working fluids (water in the front route and methanol in the back route of fire tube). It should be noted that the saturation temperatures of water and methanol corresponding to the absolute pressure of the THP in this experiment were 68 °C and 36 °C, respectively [27].

Table 3. Water temperature on the fire tube at a steady-state [°C]

T_{C1}	T_{C2}	T_{C3}	T_{C4}	T_{C5}	T_{C6}	T_{water}
102.4	92.1	75.1	66.5	55.0	45.6	60.1

Selection of proper filling ratio

The selection of a suitable FR is a major parameter in every type of heat pipe [28]. In this study, FR was defined as the ratio of the volume of charged liquid to that of the whole

THP. Several experimental investigations have suggested appropriate ranges of FR. Imura *et al.* [29] proposed a ratio of 1/5-1/3, and Hadara *et al.* [30] suggested 25-30% for FR. Noie [31] investigated the effect of the working fluid volume equal to 30%, 60%, and 90% of the evaporator volume on the performance of THP. From the foregoing experimental investigations, it can be concluded that the most optimal FR for any heat pipe depends on many factors such as geometry, heat input and type of liquid and operating conditions. Therefore, the suitable FR change from one heat pipe to another and choosing the best FR are required whenever any one of these parameters is changed. For this reason, a specific experimental analysis of THP is generally necessary for a better understanding of its operational characteristics. In this work, THP were vertically installed in the device, and their thermal performance was investigated in 9 different modes.

Thermal efficiency calculation

The output useful energy from the heater is difference in enthalpy between the inlet and outlet of gas. Input energy is the chemical energy released from fuel consumption. Therefore, thermal efficiency is computed:

$$\eta = \frac{\dot{Q}}{\dot{m}_f \text{lhv}} 100 \tag{1}$$

The rate of heat transfer to the NG is calculated:

$$\dot{Q} = \dot{m}_a C_p (T_{g,2} - T_{g,1}) \tag{2}$$

In eq. (1), mass-flow rate of the fuel is calculated:

$$\dot{m}_f = \frac{P\dot{V}}{Z\text{R}T} \tag{3}$$

In this study, for safety and prevention of energy waste, air was used in the gas tube instead of NG. The air entered in the gas tube by using a blower and the anemometer was used for measuring air velocity. Mass-flow rate is calculated:

$$\dot{m}_a = \rho VA \tag{4}$$

The values of the properties and coefficients used in eqs. (1)-(4) are given in tab. 4.

Table 4. Values of properties and coefficients used in formulas

ρ [kgm^{-3}]	A [m^2]	lhv [kJkg^{-1}]	P [kPa]	T [K]	R [Jkg^{-1}K^{-1}]	C_p [Jg^{-1}K^{-1}]	Z
1.23	0.0046	29175.38	101.305	288.15	286.9	1.005	0.94

Uncertainty analysis

It is no longer acceptable, in most fields, to present experimental results without describing the uncertainties involved. The uncertainty of the measured parametersis computable by knowing the accuracy of the measuring device [32]. In the present work, the aim was to calculate the thermal efficiency via eq. (1). Analysis of the measurement uncertainty was performed based on the root of sum squares method developed by Kline and Mclintoc [33]. In this study, different measuring devices were used with the specifications illustrated in tab. 2. The uncertainty of thermal efficiency can be calculated from the following equations:

$$U_\eta = \left\{ \left[\left(\frac{\partial \eta}{\partial \dot{m}_a} \right) U_{\dot{m}} \right]^2 + \left[\left(\frac{\partial \eta}{\partial \Delta T} \right) U_{\Delta T} \right]^2 + \left[\left(\frac{\partial \eta}{\partial \dot{m}_f} \right) U_{\dot{m}} \right]^2 \right\}^{1/2} \tag{5a}$$

$$\frac{U_\eta}{\eta} = \left[\left(\frac{U_{\dot{m}_a}}{\dot{m}_a} \right)^2 + \left(\frac{U \Delta T}{\Delta T} \right)^2 + \left(\frac{U_{\dot{m}_f}}{\dot{m}_f} \right)^2 \right]^{1/2} \tag{5b}$$

The uncertainties of the fuel mass-flow rate, $\dot{m}_f$, and air mass-flow rate, $\dot{m}_a$, according to eqs. (3) and (4) are:

$$U_{\dot{m}_f} = \left\{ \left[\left(\frac{\dot{V}}{ZRT} \right) U_\rho \right]^2 + \left[\left(\frac{P}{ZRT} \right) U_\mathrm{A} \right]^2 + \left[\left(-\frac{P\dot{V}}{ZRT^2} U_V \right) \right]^2 \right\}^{1/2} \tag{6a}$$

$$U_{\dot{m}_a} = \left\{ \left[\left(\frac{\partial \dot{m}}{\partial \rho} \right) U_\rho \right]^2 + \left[\left(\frac{\partial \dot{m}}{\partial \mathrm{A}} \right) U_\mathrm{A} \right]^2 + \left[\left(\frac{\partial \dot{m}}{\partial V} \right) U_V \right]^2 \right\}^{1/2} \tag{6b}$$

According to the maximum measured, the uncertainty of air mass-flow rate and the fuel mass-flow rate were 1.2% and 1.8%, respectively. In all the tests, the maximum value of thermal efficiency uncertainty was 3.8%. Also, the principle of the repeatability of the result was carried out in the experiments which is described in section *Heater in normal state (without THP)*.

Results and discussion

To determine the effects of THP on the thermal performance of the heater, the thermal efficiency of the device was investigated in two situations (with and without THP).

Heater in normal state (without THP)

All of the experiment and measurement were done in the steady-state condition. In fig. 4, it is shown that the temperature changed at different locations on fire tube until the water temperature became stable.

As seen in fig. 4, the fluid media (water) became steady after 215 minutes (on average) with a temperature of about 60 °C. In this condition, the temperatures of the different points of the fire tube also reached steady-state. Using the measuring equipment, the quantities required to calculate the thermal efficiency were recorded. The values of the various parameters in eqs. (1)-(5) are shown in tab. 5. To check the accuracy of the

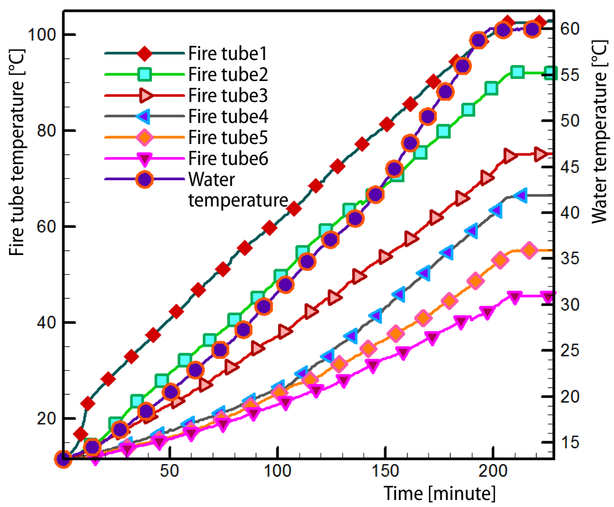


Figure 4. Distribution of water and fire tube temperature in normal state

Table 5. Calculation of the thermal efficiency of the heater in normal state

	V [ms^{-1}]	$\dot{m}_a$ [kgs^{-1}]	$\dot{V}$ [m^3]	t [hr]	$\dot{m}_f$ [kgs^{-1}]	$T_{g,1}$ [°C]	$T_{g,2}$ [°C]	η
Run 1	13.9	0.078	2.715	3.36	$2.2445 \cdot 10^{-4}$	11.6	41.2	35%
Run 2	14.1	0.080	2.680	3.34	$2.2289 \cdot 10^{-4}$	11.4	40.9	36%
Run 3	14.0	0.079	2.790	3.41	$2.2727 \cdot 10^{-4}$	11.8	41.4	35%
Run 4	13.8	0.078	2.740	3.38	$2.2518 \cdot 10^{-4}$	10.9	40.5	35%
Run 5	13.9	0.078	2.655	3.33	$2.2147 \cdot 10^{-4}$	11.1	40.9	36%

tests and evaluate the non-systematic errors, all tests were repeated five times. Different test runs led to almost the same heater thermal efficiency.

Heater with thermosyphon heat pipe

To determine the most effective mode of THP, different FR were investigated. In this regard, THP were charged by water and methanol as working fluids in the front and back routes of the fire tube, respectively. The THP were installed with different FR on the fire tube and in the space between fire tube and gas tube, then their effects were further studied. Experiments were conducted in three different modes to check all states for FR and working fluid:

The first mode: The heat pipes were installed in the front route of the fire tube with FR of 10, 20 and 30%, and in the back route with FR of 10%.

The second mode: The heat pipes were installed in the front route of the fire tube with FR of 10, 20 and 30%, and in the back route with FR of 20%.

Third mode: The heat pipes were installed in the front route of the fire tube with FR of 10, 20 and 30%, and in the back route with FR of 30%.

Water of heater was heated so that the temperature of it reached a steady-state, at this point, the temperature of the outlet gas, T_8, was measured and recorded on the data logger. The mass-flow rate of the fuel was similar to that of the normal heater (without THP). Thermal efficiency was ultimately calculated by eq. (1). As seen in tab. 6, the most optimal thermal efficiency occurred when THP FR in the front and back routes of the fire tube were 20% and 30%, respectively. In this condition, the thermal efficiency of the heater was 13% better than the normal state (without THP).

Table 6. The result associated with the thermal efficiency of heater with THP in different FR

Case	The FR in the front rout of the fire tube	The FR in the back rout of the fire tube	Fuel mass-flow rate	NG (Air) mass-flow rate	The exhaust gases temperature	Temperature of the exhaust gas from the stack	The temperature of the fire tube in various situations in a steady-state						Thermal efficiency [%]
							T_{C1}	T_{C2}	T_{C3}	T_{C4}	T_{C5}	T_{C6}	
1	10%	10%	$2.25 \cdot 10^{-4}$	0.079	46.07	74.4	98.9	90.5	73.7	64.9	53.7	44.4	42
2	20%	10%	$2.25 \cdot 10^{-4}$	0.081	44.56	75.2	99.6	90.8	74.1	65.1	53.9	44.6	41
3	30%	10%	$2.21 \cdot 10^{-4}$	0.078	42.75	79.3	101.6	91.5	74.8	65.7	54.4	45.1	38
4	10%	20%	$2.24 \cdot 10^{-4}$	0.080	46.35	73.8	98.5	90.2	73.4	64.8	53.6	44.3	43
5	20%	20%	$2.23 \cdot 10^{-4}$	0.079	47.56	72.9	981	90.1	73.1	64.7	53.5	44.2	44
6	30%	20%	$2.27 \cdot 10^{-4}$	00.77	44.30	76.1	100.1	91.2	74.4	65.3	54.2	44.8	40
7	10%	30%	$2.22 \cdot 10^{-4}$	0.078	47.02	71.4	97.1	89.8	72.7	64.6	53.3	44.1	45
8	20%	30%	$2.27 \cdot 10^{-4}$	0.080	51.03	68.2	96.8	88.6	72.0	64.1	52.9	43.8	48
9	30%	30%	$2.29 \cdot 10^{-4}$	0.080	48.89	71.4	97.9	89.8	72.7	64.6	53.3	44.1	45

As showen in fig. 5, for higher efficiency, the fire tube has lower temperatures, so that more heat is transferred to the gas tube. Figure 6 also shows that with higher efficiency, the temperature of the exhausted gases is lower and less heat is wasted from the stack.

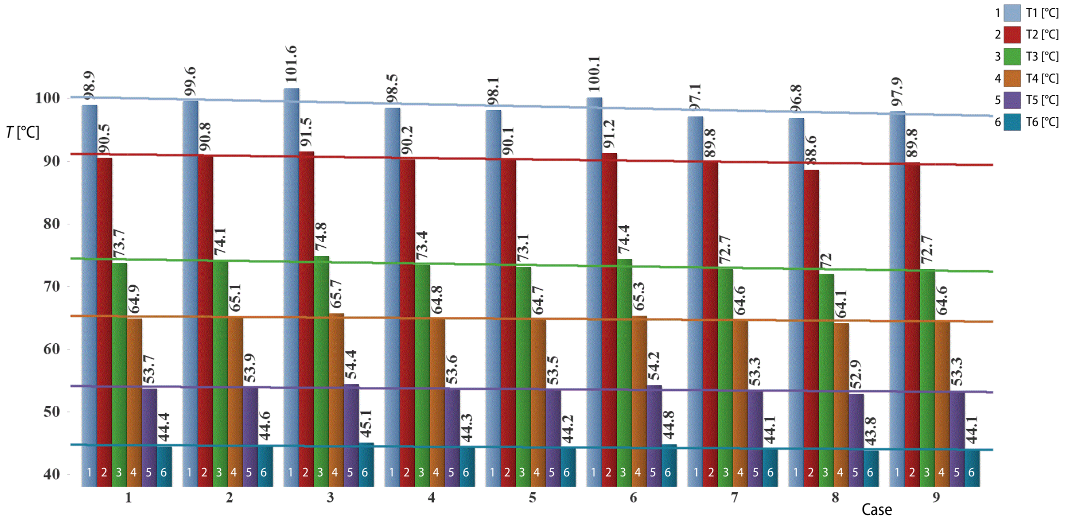
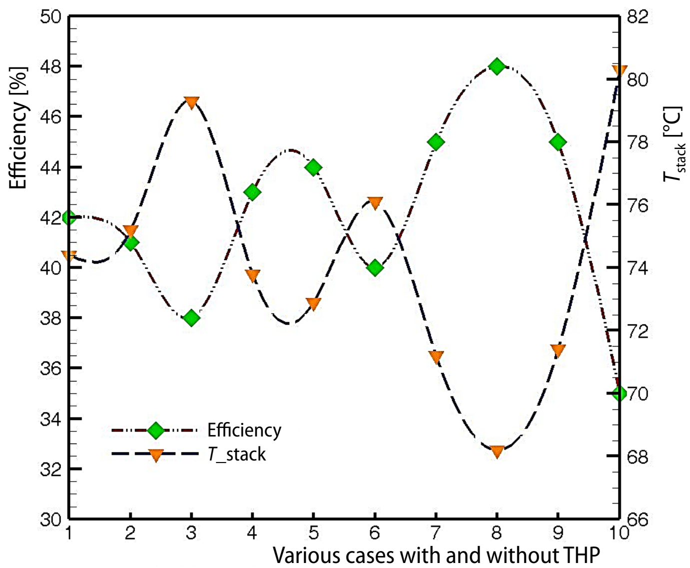


Figure 5. Distribution of temperature on a fire tube in different FR

Figure 6. The THP with different FR (1 to 9) and without THP (10)



Conclusion

Energy consumption in heaters of CGS is considerable and high-quality refined gas is employed for combustion in such heaters. The THP technology has been used for a variety of applications in several heat transfer devices. In this study, efforts were made to increase the heater thermal efficiency by use of THP. For this purpose, a heater was fabricated and its thermal efficiency was investigated. The THP were then designed according to all operational considerations and installed in the heater. In order to determine the most effective FR, the working fluid was injected into the THP with 10, 20, and 30% proportions. The THP with different FR were tested and thermal efficiency was calculated in each mode. The following main conclusions can be drawn from the present study:

- According to the saturated pressure and the associated boiling point, water and methanol, as THP working fluids, were suitable options for front and back routes of the heater fire tube, respectively.
- With the utilization of THP, the thermal efficiency of the heater increased from 3 to 13%.
- The maximum thermal efficiency reached to 48% that occurred with the water as working fluid with 20% FR in the front route and methanol as working fluid with 30% FR in the back route of the fire tube.

There is a direct relationship between energy use and environment. Obviously, by improving the thermal efficiency of the method proposed in this paper, and generalizing it to all CGS heaters, energy consumption and, consequently, greenhouse gas emissions will be reduced.

Acknowledgment

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Nomenclature

A – gas tube surface, [m^2]
C_p – specific heat, [Jkg^{-1}K^{-1}]
lhv – low heating value, [kJkg^{-1}]
$\dot{m}_a$ – air mass-flow rate, [kgs^{-1}]
$\dot{m}_f$ – fuel mass-flow rate, [kgs^{-1}]
P – pressure, [kPa]
R – universal gas constant, [Jkg^{-1}K^{-1}]
T – temperature, [°C]
$T_{g,1}$ – inlet NG temperature, [°C]
$T_{g,2}$ – outlet NG temperature, [°C]
t – time, [hours]
Z – compressibility factor

Greek symbols

η – thermal efficiency, [%]
ρ – density, [kgm^{-3}]

Acronyms

CGS – city gate station
FR – filling ratio
IWBH – indirect water bath heater
LHV – latent heat of vaporization
NG – natural gas
THP – thermosyphon heat pipe
SCMH – standard cubic meters per hour

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