An Empirical Investigation of a Modified Gas Engine Heat Pump in Heating and Cooling Mode for the Residential Application

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ABSTRACT: *Gas Engine Heat Pump (GEHP) is found as an effective machine among energy conversion systems. However, further improvement of GEHP performance is still indispensable. In the current study, to recycle efficiently the flue gases waste heat of GEHP unit, it is suggested to utilize a condensing heat exchanger. To assess the performance of the proposed GEHP, an equipped test room was designed. The nominal capacity of the current GEHP is about 80 kW for heating and 71 kW for cooling. The experimental investigation was implemented for two common strategies, i.e. heating and cooling modes. The effect of outlet dry bulb temperature of 0 °C, 5 °C, 10 °C and 15 °C for heating, and 25 °C, 30 °C, 35 °C and 40 °C for cooling as well as water outlet temperature of 40 °C, 45 °C and 50 °C for heating and 8.5 °C, 11 °C, 14 °C for cooling on the performance of the GEHP is investigated. The 95% confidence interval has been considered in statistical analysis. In heating mode, results show that the efficiency of the condensing heat exchanger increases during the reduction of water out temperature and it leads to a Coefficient of Performance (COP) enhancement. Additionally, observation indicates that the decrement of the ambient air temperature from 5 to 0 ° C leads to the severe enhancement of fuel consumption of the gas engine. The performance indicators are re-evaluated using the non-condensing heat exchanger. It is concluded that the COP of GEHP with the condensing heat exchanger decreases by the steeper slope versus the reduction of the ambient dry bulb temperature compared to the non-condensing heat exchanger.*

KEYWORDS: *Gas engine heat pump; Condensing heat exchanger; Heating and cooling mode; performance map.*

INTRODUCTION

The first studies about the GEHP have been started in about 40 years ago [1-3] and it was introduced to the market about 1985 [4]. Since then, these systems are most commonly used for heating and air conditioning system [5, 6]

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1021-9986/2020/5/321-332 12/$/6.02

and sometimes integrated for industrial applications like the dehumidifying process [4, 7]. Using the GEHP in residential application have some distinguished advantages [6, 8] as follows: (1) reduce the electric-grid peak demand leads associated with conventional, all electrical system, (2) recovery of waste heat from flue gases, (3) Adjust the compressor speed with the required refrigerant level (4) reach a high efficiency in a wide range of the outdoor temperature (5) increase the part load efficiencies (6) no need to have defrosted cycle (7) reduce emissions from combustion. Therefore, GEHP has come to be one of the accessible patterns of the combination of cooling and heating for environmentally friendly buildings.

It is notable that the standard GEHP system is not an optimal system and there are still capabilities to further development of the system performance. Therefore, some investigations have been done to modify the efficiency of GEHP by suggesting the different flow diagrams [4, 9, 10], different refrigerant [11-13], adding an extra system [14-17] and improvement of the control strategy [18-20]. Recently, *Shang et al.* [15] proposed a two-driven mechanism, in which the compression cycle is driven by a gas engine for heating mode and by electrical motor for cooling mode. It is shown that the primary energy ratio (PER) of the GEHP for cooling and heating modes is about 28.5% to 51.2% and 15.8% to 25.3% lower than that of their proposed system, respectively. *Lio et al.* [17] proposed a hybrid compression-absorption system for GEHP. In this system, the waste heat of the gas engine is used to generate the cooling via the absorption refrigeration cycle. The comparison shows that their proposed hybrid system can effectively recover the heating waste energy at the low air temperature. *Wang et al.* [20] adjust the capacity of GEHP by developing a control system to reduce the rotary speed fluctuations caused by the instant load. The experimental results show that the performance of the professional controller is better than the performance of the traditional system. The variation of the engine rotary speed of an expert control system is smooth and there are no any overshoots with an acceptable response through the capacity adjustment process.

The behavior of the GEHP with different types of variables has been investigated experimentally in two studies [21, 22]. *Liu et al.* [21] show that the Coefficient of Performance (COP) and PER of the GEHP system increased with growing ambient air temperature and condenser water inlet temperature, however, it is decreased with the increasing of the rotary speed of the engine. *Hu et al.* [22] find that the characteristic performance of the system is considerably impressed by the rotary speed of the engine and the outdoor air temperature. They realize that the water inlet temperature acts slightly on heating capacity, while it had some effect on the PER of the GEHP.

Applying the condensing heat exchanger can be taken into account as a way to improve the performance of the GEHP unit. It recovers efficiently the wasted heat, which is latent in the steam part of the flue gases of the engine. This paper aims to study the performance characteristics of GEHP using the condensing heat exchanger. To apply this change, it is necessary to modify the path of the cooling flow of the engine. In most of the past studies [5, 15, 17, 21, 23, 24], water first absorbs the heat of the engine body and then come into the exhaust heat exchanger, while in the present system the water first recover the wasted heat within the condensing heat exchanger and then enter the engine. This change leads to reduce the inlet water temperature of the condensing heat exchanger and increase the condensation rate, GEHP's heating capacity, and COP during the heating mode.

THEORETICAL SECTION

The system of the current GEHP

The view of the process flow diagram of the examined GEHP system is illustrated in Fig. 1. In this flow diagram, three flow paths have been identified. The first one, which is colored in blue and named the heat recovery subsystem, is related to the water coolant circuit of the engine. In the current study, the heat recovery system uses a condensing heat exchanger, which gets well the waste heat energy of exhaust gas by recovering the latent heat of the water vapor. The second path, which has a green color, shows the refrigerant movement route through the heat pump circuit. The vapour compression of the refrigerant cycle is moved by a compressor within a circuit, which consists of typical equipment like expansion valve, evaporator, and condenser. The last path, which is shown with black color, is depicted as the accessories of the gas engine (combustion air and flue gas subsystem and oil and fuel subsystem). As it is shown in Fig. 1, the gas engine drives the scroll compressor to obtain the required pressure.

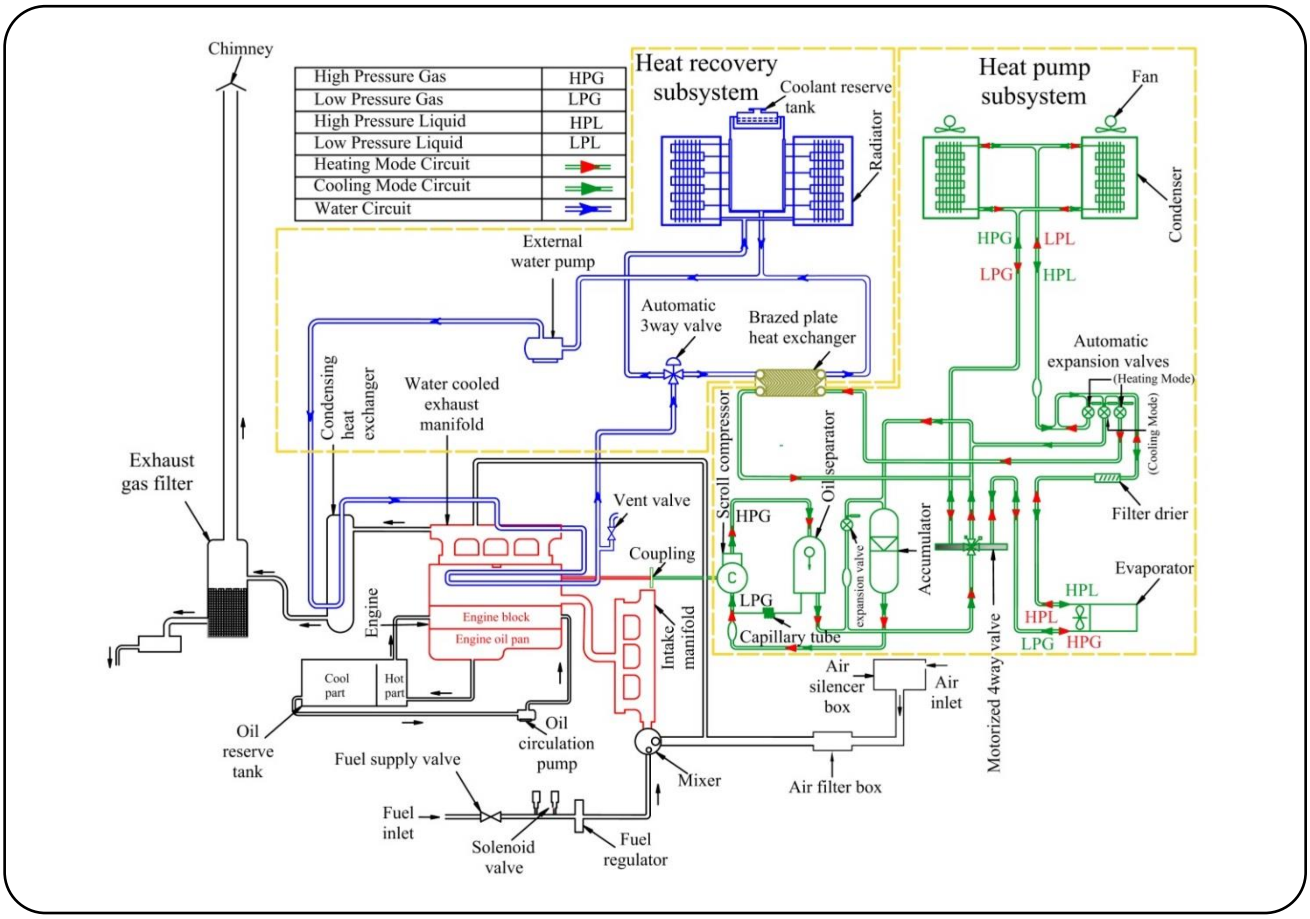

Fig. 1: The schematic of the GEHP unit with the heat recovery subsystem.

In the present study, the gas engine is controlled by a smart control system. This system tries to optimize for the engine lifetime and the response rate of the system load requirement.

In the current study, the gas engine heat pump can be used in several operating modes namely; cooling and heating mode, which is shown in Fig. 1. The red arrows and green arrows indicate the direction of refrigerant movement in the heating and cooling system, respectively. As it is illustrated, in the heating mode, the engine coolant water accumulates the heat waste from the engine, which it transfers to the refrigerant via a brazed plate heat exchanger. Waste heat is the heat which is dissipated from the exhausted flue gases and engine block. Within the cooling mode, this heat can be used for domestic hot water or can be dissipated in Radiator. In the current study, as is depicted, the radiator is selected and is assumed according to experimental investigation. The energy absorbed by the refrigerant can be transferred to the water which is circulated within indoor units by the evaporator heat exchanger.

The parameters which are used in the sensitivity analyses are defined in the following sentences. It is notable that all of the presented parameters are considered as the main factors of the GEHP units. According to the standard EN 437 [25-27], the heating value released from the input fuel can be estimated from Eq. (1).

$$\dot{Q}_{in} = \frac{P_{atm} + P_{gas}}{101.325} \frac{288.15}{273.15 + T_{gas}} H_{gas} \dot{V}_{gas} \tag{1}$$

P_{gas}, P_{atm}, $\dot{V_{gas}}$ and T_{gas} are natural gas and atmospheric pressures (kPa), volumetric flow rate (m^3/s) and natural gas temperature ($°C$), respectively. The above mentioned parameters are read within the experimental tests. However, the H_{gas} is the heating value of the natural gas (kJ/m^3) and is obtained from Table 1. In the current GEHP unit, a part of the wasted heat is used to enhance the temperature of the water which is circulated inside of the test room. Then, $\dot{Q}_{delivered}$ is estimated considering the below relation:

Table 1: Composition of Iran natural gas used as engine fuel during the experiments [34].

Composition	Value
Methane	89.124 %
Ethane	4.5862 %
Propane	1.0029 %
iso-Butane	0.1558 %
n-Butane	0.2045 %
iso-Pentane	0.0682 %
n-Pentane	0.0487 %
n-Hexane	0.0389 %
n-Heptane	0.0191 %
Nitrogen	3.7098 %
Carbon dioxide	1.0419 %
Lower heating value	34616.83 kJ/m^3

$$\dot{Q}_{delivered} = \dot{m}_{water,in} cp_{water} \left(T_{water,out} - T_{water,in}\right) = \rho_{water} \dot{V}_{water} cp_{water} \left(T_{water,out} - T_{water,in}\right) \quad (2)$$

$T_{water,out}$, $T_{water,in}$, $\dot{V}_{water}$ and $\dot{m}_{water,in}$, are delivered temperature, input temperature, volumetric and mass flow rate of water, respectively. The above mentioned parameters are read within the experimental tests. However, the density of water (ρ_{water}) and specific heat capacity of water (cp_{water}) are calculated at the inlet temperature and average of inlet and outlet temperatures, respectively. This density correlation is selected because of the flow meter is located in the inlet water. The below equations (Eqs. (3) and (4)) are considered for calculating the density and isobaric heat capacity of water, at the atmospheric pressure and range of 5–90°C. These accurate correlations are obtained from the curve fitting scheme which is introduced in Refs. [28, 29].

$$\rho_{water} = -0.0037\, T_{water}^2 - 0.072\, T_{water} + 1000.9\ (kg\ m^{-3}) \quad (3)$$

$$cp_{water} = (-1.3273\, e-7) T_{water}^3 + (3.1895e-5) T_{water}^2 - 0.0018108\, T_{water} + 4.2078\ \ (kJ\ kg^{-1}\ K^{-1})) \quad (4)$$

Additionally, the Coefficient of Performance (COP) of the unit is calculated as the following equation:

$$COP_{heating} = \frac{Q_h}{Q_h - Q_c} \quad (5)$$

$$COP_{cooling} = \frac{Q_c}{Q_h - Q_c} \quad (6)$$

Where Q_h and Q_c are the heat supplied to the hot reservoir and heat removed from the cold reservoir, respectively. Furthermore, the uncertainties of the parameters which are going to be examined within the sensitivity analyses are informed in Table 2. As it is known, for the considered parameters, the law of propagation of uncertainty [30-32] is applied. Uncertainty considers as one of the most main parameters in measurements as well as modelling processes [33].

EXPERIMENTAL SECTION

The testing room and measuring instruments

In the current investigation, a prepared test stand is engaged to assess the designed GEHP system. It is displayed in Fig. 2. The width, length, and height of the outer of the test room are 3.530 m, 3.612 m, and 4.080 m, respectively, while its inner dimensions are 3.000 × 3.000 × 3.000 m^3. The inner and outer sides of the walls are made by the wavy steel panels. The polyurethane is used to insulate the walls. It stuffs the space between both sides of the walls. A tank with a volume of 60 L is considered to remove any disturbance in recording the process water temperature. Afterward, an air handling unit is designed to consume the energy which is produced in the GEHP system. This energy is transferred to the air handling system via the two parallel plate heat exchangers. A surge tank with a volume of 20L is assumed in the return pass. The process flow diagram of the designed test room with the water loops is shown in Fig. 3***Fig. 3:***. To obtain the temperature of the specified points, five thermocouples are considered. The specifications of these thermocouples as well as the other measuring instruments are depicted in Table 3. The uncertainty of each instrument is also specified in Table 4.

To manage the GEHP unit and reach to operator requests need to design a smart control system based on the gathering data. Accordingly, some sensors are embedded in different parts of the GEHP system, including the gas engine, heat recovery section, heat pump subsystem and accessories of a gas engine. The required data is transmitted from the sensors via several cables. During the test, the smart control system through these

Table 2: Uncertainties of the parameters which are going to be investigated through the sensitivity analyses.

Parameter	Uncertainty
Exhaust temperature	±0.050 °C
In/out water temperatures difference	±0.10 °C
Heat output	±0.020 kW
Power input	±0.025 kW
The coefficient of Performance for heating	±0.025 %
The coefficient of Performance for cooling	±0.030 %

Fig. 2: The manufactured test room with the air handling unit.

cables is connected to the monitoring system of the test room. The typical sensors which are generally located in the gas engine, are captured the fuel flow, coolant water temperature, and engine speed. Therefore these parameters can be observed in the monitoring system of the test room. In developing the current smart control system, engine lifetime increment has been especially important. Therefore, In addition to the mentioned parameters, temperature and pressure of the lubrication system are also observed and controlled in the current study.

The sensors which are located in heat recovery and heat pump subsystems have been used to detect the system fault and estimate the performance indicators of the GEHP unit. In addition to the mentioned sensors, the test room monitoring system is also equipped with temperature, pressure and flowmeter sensors. Among these sensors can be referred to the temperature sensor at the inlet of the flue gas-water heat exchanger, which is used to evaluate the steady condition of the results. The steady condition will result when the amplitude of the fluctuations in the gathered temperature data of this sensor is less than 0.05 °C. During the test, the volumetric flow of water which is circulated in the test room circuit was kept constant at 135 liters per minute. In order to achieve a steady condition, the monitoring system of the test room adjusts the speed of the air conditioner's fan so that the amount of heat dissipation reaches a constant value and does not change versus the time. It must be noted that the steady condition is considered as a main assumption in the current study.

RESULTS AND DISCUSSION

In the current section, the characteristic curve of the GEHP unit is obtained during the sensitivity analyses. The two main operational strategies, heating, and cooling modes are considered here. Condensing and non-condensing heat exchangers have been considered to obtain the performance curves of the GEHP unit. The composition of Iran natural gas, which is assumed as fuel, is identified in Table 1. Furthermore, the tests were implemented at the mean atmospheric pressure of 902.1 mbar. The equipment which is used for measurement, have been revisited precisely. The data which are obtained from the experimental test is recorded via the data logger in the testing room. Statistical analysis is done considering the average of the indicators quantities which has been perceived. Assuming the normal distribution and 95% confidence interval by means of an average of the observed data, upper and lower bounds are specified during the test and shown in the following Figures.

Pressure drop in condensing heat exchanger

The effect of water condensing in the heat exchanger on engine performance must be assessed. Accordingly, it is required to study the pressure drop due to the condensation across the condensing heat exchanger. This is measured during the 90 seconds period of the engine operation. The results are shown in Fig. 4 for the output power of the engine of about 15.7 (kW). As can be seen, the value of the pressure drop is between 0.63 (kPa) and

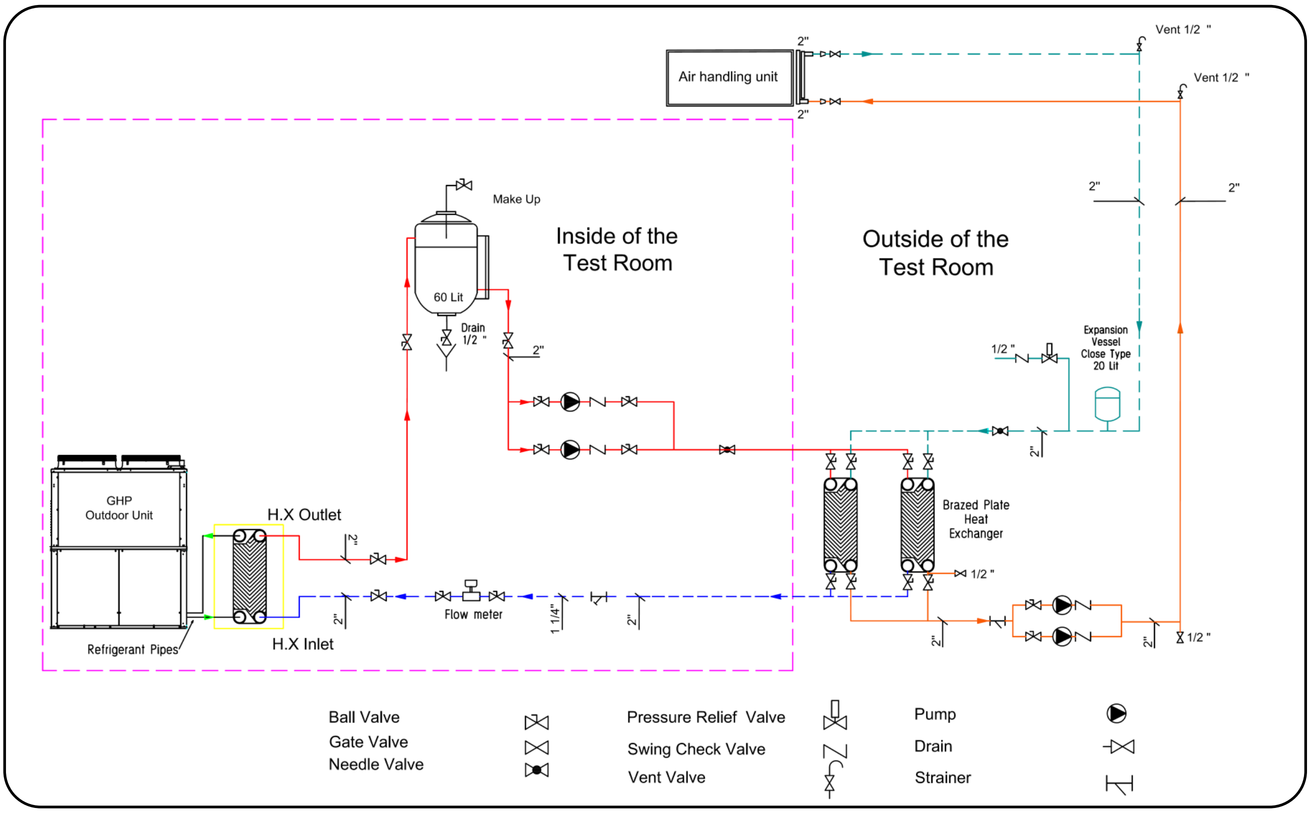

Fig. 3: The process flow diagram of the test room.

0.84 (kPa), which is less than the allowable value provided by the engine manufacturer.

Sensitivity analyses I: the strategy of following the heating mode

In this case, the experiments were done under three conditions, 40 °C, 45 °C and 50 °C of the outlet water temperature of GEHP system with condensing heat exchanger. The outlet water is the water which is delivered to the consumer. For these conditions, the effects of variation of outdoor dry bulb temperature on the heating capacity, power input, and COP of the system are shown in Fig. 5. As it is shown, when the control system operates at the outdoor dry bulb temperature range between 5 °C to 15 °C, the power input to the gas engine remains nearly constant in the minimal value. Practically reducing the power input leads to a reduction in fuel consumption of GEHP with condensing heat exchanger. However, when the ambient temperature decrease from 5 ° C to 0 ° C, a relatively sharp increment happen in the fuel consumption of the gas engine. It is due to the reduction of the evaporator heat exchanger ability to absorb the heat from the ambient cold air at 0 °C temperature. The enhancement of fuel consumption leads to the rise of the value of waste heat and ultimately prevent the inappropriate reduction of the COP. additionally, decrement of the ambient temperature from 5 ° C to 0 ° C makes relatively high increment on the mass flow rate of condensing water in the heat exchanger. This leads to a further enhancement in the heat absorbed from the exhausted gases and a relatively significant increase on the COP of GEHP with condensing heat exchanger during the mentioned interval of temperature.

The COP results of GEHP with condensing heat exchanger in Fig. 5(C) show that the outdoor dry bulb temperature enhancement can lead to increase the condenser efficiency. Also, in a constant outdoor dry bulb temperature, the increment of the outlet water temperature from 40 °C to 50 °C results in the reduction of the COP. It is due to the efficiency enhancement of the condensing heat exchanger within the reduction of the water out temperature.

The engine performance indicators of GEHP with the non-condensing heat exchanger were revisited. Observation

Table 3: The specifications of the investigated GEHP system.

Performance(kW)		
Cooling capacity(Standard)		71.0 kW
Heating capacity (Standard)		80.0 kW
Heating capacity (low temp.)		78.0 kW
Electrical Rating		
Cooling	Running Amperes (A)	6.22
	Power input (kW)	1.33
	Power factor (%)	93
Heating	Running Amperes (A)	3.92
	Power input (kW)	0.83
	Power factor (%)	92
Starting Amperes(A)		30
Gas Type		
Gas Band	P	Propane gas(G31)
	H	Natural Gas (G20)
	L	Natural Gas (G25)
Compressor		
Cooling Oil (L)		7.5 (HP-9)
Crankcase heater(W)		30
Engine		
Displacement (L)		2.488
Rated output (kW)		15.7
Oil quantity (L)		43
Starter motor		12 V DC, 2 kW
Engine cooling water		
Concentration		50% Water-50% Ethylene glycol
Freezing temperature		-35ºC
Quantity (L)		29
Water pump input power(kW)		0.16
Refrigerant		
Type		HFC (R410A)
Quantity (kg)		11.5
Ventilation System		
Type		Propeller fans (x2)
Air flow rate (m^3/min)		380
Rated input (kW)		0.70×2
Drain heater (W)		40
External Dimensions & Weight		
Height (mm)		2,273
Width (mm)		1,650
Depth (mm)		1000(±80)
Weight (kg)		805
Operating noise (dB)		60

Table 4: Measurement devices and their uncertainties.

Parameter	Measurement device	Operation range	Uncertainty
Water flow rate	Magnetically inductive flow meter	0–1000 m^3h^{-1}	±1%
Gas flow rate	Magnetically inductive flow meter	0–1000 m^3h^{-1}	±1%
Pressure	Pressure sensors	0–4 bar	±0.08 bar
Temperature	Temperature sensors	−20 to 100 °C	±0.10 °C

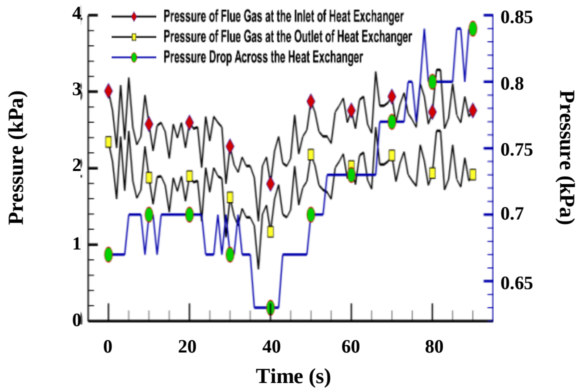

Fig. 4: The pressure drop of condensing heat exchanger against the time in seconds.

shows that the unwanted condensation increase within the decrement of the ambient dry bulb temperature. In outlet water temperature of 40 °C, the mass flow rate of condensing water is so high that it produces a severe pressure drop in the path of the flue gas within the heat exchanger. This pressure drop disrupts the engine's performance, therefore at mentioned temperature, there is no result for non-condensing heat exchanger. In outlet water temperature of 45 °C and 50 °C, the difference between the heating capacities of the GEHP using the condensing and non-condensing heat exchangers is increased by an enhancement of the ambient dry bulb temperature. Fig. 5 (C) illustrates a steady decrease in the COP of the GEHP with non-condensing heat exchanger versus the decrement of the dry bulb temperature.

In the designing of a GEHP system, when the highest possible outlet temperature is selected, the heat transfer area must be reduced to the lowest value. However, after selecting the components, it does not have any effect on it. Furthermore, enhancement of the delivered temperature leads to reduce the condensation of water vapor and decrement of the efficiencies of heat exchangers. On the other hand, as discussed earlier, the efficiency of the condensing heat exchanger increases within the reducing of the water out temperature. Seeing all of the above debates, it is established that the lowest feasible temperature is the best rate of the heating mode of the GEHP unit. It leads to increase the life time and a maintenance interval of the system due to the reduction of the operating temperature of the heat exchangers, surge tank, pipelines, and recirculation pump. Consequently, this temperature decrement has efficiency as well as technical and economic advantages.

Sensitivity analyses II: the strategy of following the cooling mode

In this case, the experiments were done under three conditions, 8.5 °C, 11 °C and 14 °C of the outlet water temperature of the GEHP system. For these conditions, the effects of variation of outdoor dry bulb temperature on the cooling capacity, power input, and COP of the system are shown in Fig. 6. As it is illustrated, in a constant outlet water temperature, increasing outdoor dry bulb temperature leads to a decrease in cooling capacity almost linearly. The slope of decrement remains approximately constant with an increase in the outlet water temperature. However, the power input is exponentially changed in terms of outdoor dry bulb temperature. At 40 °C of outdoor dry bulb temperature, the significant enhancement in power input has led to an impressive reduction in the amount of the COP. It is due to the decrease of the condenser efficiency during the dissipation of the heat to the hot air at 40 °C temperature.

CONCLUSIONS

In the current study, the experiments were done at six conditions, 40 °C, 45 °C and 50 °C of the outlet water temperature for heating mode, and 8.5 °C, 11 °C and 14 °C of the outlet water temperature for cooling mode using the condensing heat exchanger. The 95% confidence interval has been assumed in statistical analysis. At these

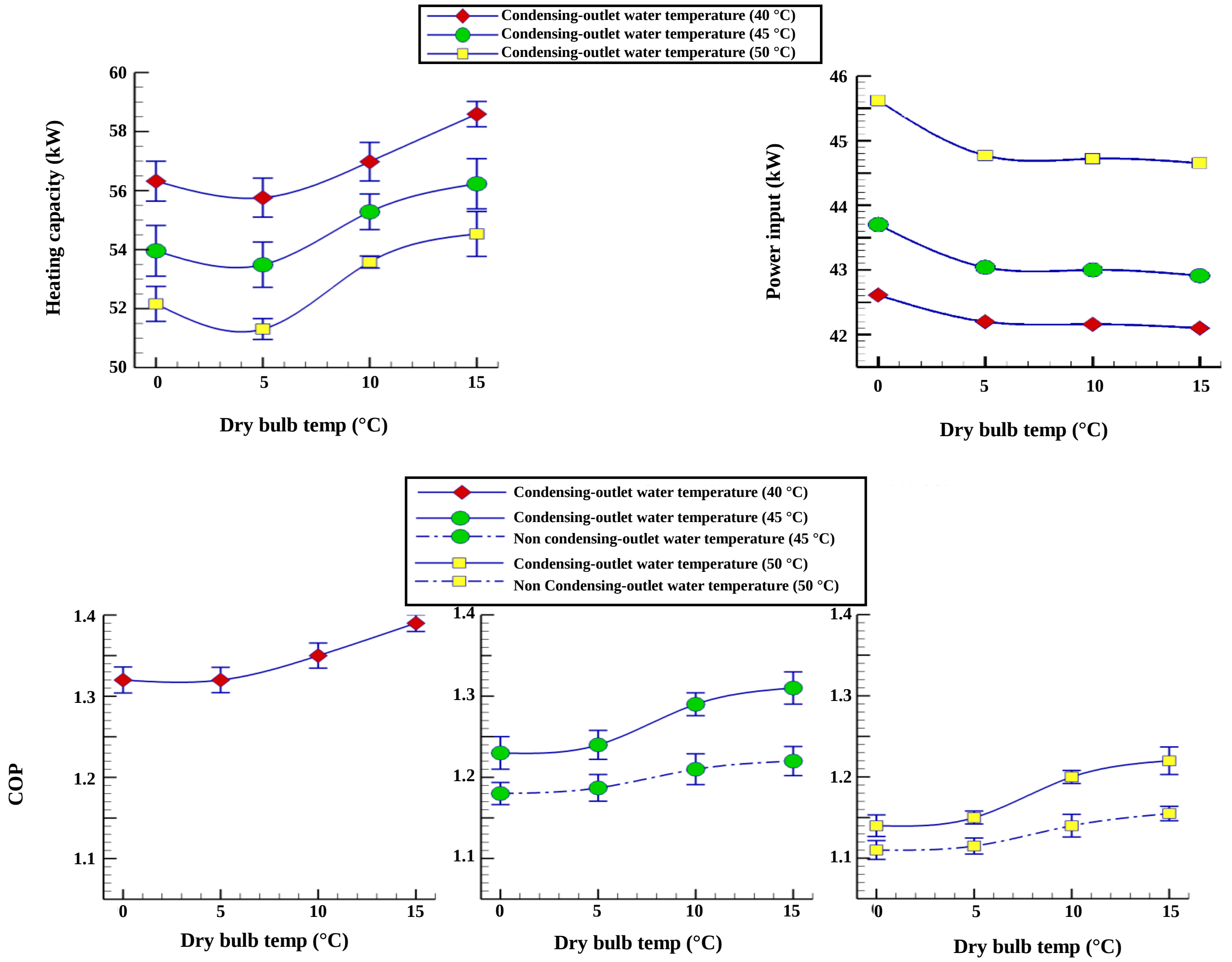

Fig. 5: The impacts of variation of outdoor dry bulb temperature on (A) heating capacity (B) power input (C) COP for the outlet water temperature of 40 °C; 45 °C; 50 °

conditions, the effects of variation of outdoor dry bulb temperature on the heating or cooling capacity, power input and COP is investigated. In heating mode, results show that when the system operates between 5 °C to 15 °C for outdoor dry bulb temperature, the fuel consumption remains nearly constant. However, at 0 °C, a relatively sharp increment happens. Results show that the efficiency of the condensing heat exchanger enhances during the reduction of water out temperature and it leads to COP enhancement. In cooling mode, figures show that the behavior of the GEHP unit is related to the condenser efficiency. In addition, the significant enhancement in power input has led to an impressive reduction in the value of the COP because of the condenser efficiency reduction near the 40 °C temperature of outdoor dry bulb temperature. The results in heating mode are revisited by analyzing the GEHP unit using the non-condensing heat exchanger. Comparison between these two data sets illustrates that the difference between the GEHP's heating capacities using condensing and non-condensing heat exchangers increases by an increment of the ambient dry bulb temperature. Observation in heating mode shows that the reduction of outlet water temperature leads to the onset of unwanted condensation in a non-condensing heat exchanger. It results in an enhancement of pressure drop in the flue gas path within the heat exchanger and disrupts the engine performance at 40 ° C. Finally, it is obtained that the behavior of the GEHP in cooling mode is absolutely related to the condenser efficiency.

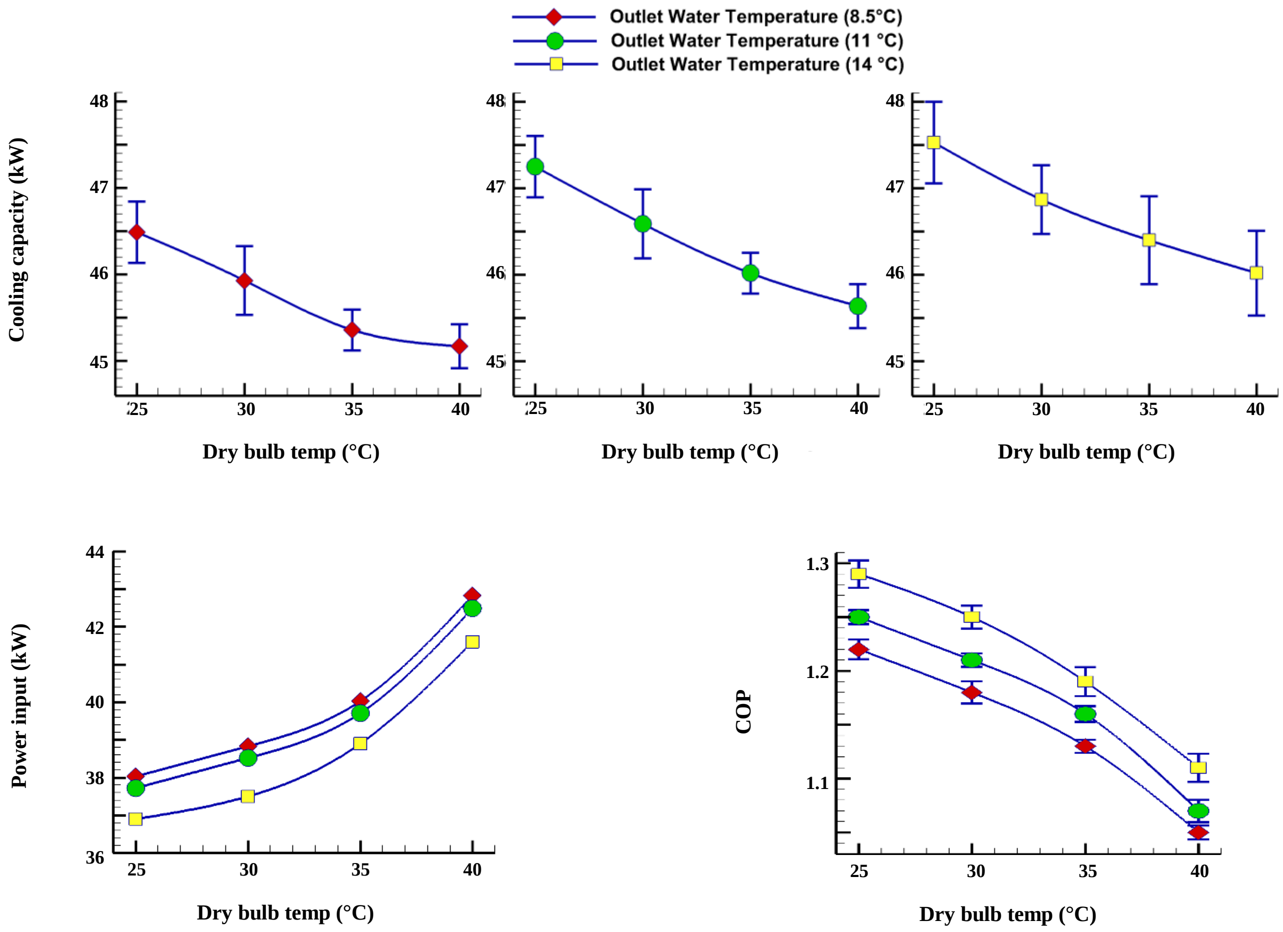

Fig. 6: The impacts of variation of outdoor dry bulb temperature on (A) cooling capacity (B) power input (C) COP for the outlet water temperature of 8.5 °C; 11 °C; 14 °C.

Nomenclature

$\dot{Q}_{in}$ The heating value released from the input fuel, kW

P_{gas} The natural gas pressure, kPa

P_{atm} The atmospheric pressure, kPa

$\dot{V}_{gas}$ The natural gas volumetric flow rate, m^3/s

T_{gas} The natural gas temperature, °C

H_{gas} The heating value of natural gas, kJ/m^3

$\dot{Q}_{delivered}$ The wasted heat which is delivered to the circulating water in the test room, kW

$T_{water,out}$ The outlet water temperature in the test room, °C

$T_{water,in}$ The input water temperature in the test room, °C

$\dot{V}_{water}$ The volumetric flow rate of the circulating water in the test room, m^3/s

$\dot{m}_{water,in}$ The mass flow rate of the circulating water in the test room, kg/m^3

COP The Coefficient of Performance

Q_h The heat supplied to the hot reservoir, kW

Q_c The heat removed from the cold reservoir, kW

Received : Feb. 23, 2019 ; Accepted :Jun. 10, 2019

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