

A study on the design and analysis of a gas engine heat pump water heater using shower wastewater as heat source

Fengguo Liu, Zhenxi Ma and Rui Zhang

ABSTRACT

Shower wastewater contains large amounts of heat, and recovering and utilizing shower wastewater heat energy to heat shower water is of great significance for energy saving. This paper proposed a gas engine heat pump water heater (GEHPWH) using shower wastewater as the heat source.

Thermodynamic models were built to evaluate the performance of the GEHPWH and compare it with an electrical heat pump water heater (EHPWH). The results show that the GEHPWH has higher hot water outlet temperature and coefficient of performance (*COP*) than the EHPWH under the same conditions. Furthermore, the GEHPWH can maintain stable hot water outlet temperature and higher primary energy ratio (*PER*) in variable water flow mode, which resolves the issue that the conventional EHPWH needs an auxiliary heat source. Finally, based on a case, the GEHPWH reveals smaller energy consumption and lower operating costs compared with three other conventional hot-water schemes.

Key words | gas engine heat pump water heater, optimization, shower wastewater heat recovery, thermodynamic

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NOTATION

The following symbols are used in this paper:

<i>COP</i>	coefficient of performance
c_p	specific heat ($\text{kJ}/\text{kg}\cdot\text{K}^{-1}$)
h	specific enthalpy (kJ/kg)
<i>LMTD</i>	logarithmic mean temperature difference ($^{\circ}\text{C}$)
m	mass flow rate (kg/s)
P	shaft power of compressor (kW)
<i>PER</i>	primary energy ratio
Q	heat loads (kW)
T	temperature ($^{\circ}\text{C}$)
UA	heat transfer capacity (kW/K)

Subscripts

HR	heat recovery
GE	gas engine
GB	gas boiler
r	refrigerant
w	water

INTRODUCTION

The building sector accounts for about 40% of the total energy consumption and domestic hot water supply has become the main contributor to residential construction energy consumption (Li *et al.* 2015; Ju *et al.* 2018). For shower facilities, the shower temperature is in the range of 37–42 $^{\circ}\text{C}$, while the temperature of shower wastewater is in general more than

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30 °C (Chen *et al.* 2013). Therefore, there is a great deal of available heat in shower wastewater. In actual operation, however, shower wastewater is often discarded directly into the drainage system without the process of waste heat recovery, which results in tremendous energy waste.

A simple way of heat recovery of shower wastewater is that of installing a heat exchanger beneath the shower drain for preheating the tap water. Wong *et al.* (2010) and McNabola & Shields (2013) investigated the potential for shower water heat recovery from bathrooms and showed that 15–50% of shower wastewater heat can be recovered by an efficient heat exchanger.

As an improvement, heat pump technology applied to public bathrooms is an efficient method to save energy and increase overall energy efficiency (Willem *et al.* 2017). The electrical heat pump water heater (EHPWH) has good performance and high stability when using wastewater as the heat source (Dong *et al.* 2015). Various studies have been conducted on wastewater source EHPWH contributing to energy saving. These systems can be mainly classified into two categories, instantaneous heating mode and cyclic heating mode. For the instantaneous heating mode, Baek *et al.* (2005) evaluated the yearly mean coefficient of performance (COP) of EHPWH using shower wastewater as the heat source was higher than air source heat pump water heater (ASHPWH). Chen *et al.* (2013) indicated that the optimal COP of the whole EHPWH reached 4.97 when the condensing temperature was 51.5 °C and the evaporating temperature was 11.7 °C. For the cyclic heating mode, Liu *et al.* (2014) proposed an EHPWH assisted with solar energy used in public shower facilities for exhaust heat recovery. Shen *et al.* (2012) investigated an EHPWH with a dry-expansion evaporator to examine the effect of tube bio-fouling on the operating performance. In addition, Yang *et al.* (2013) and Hepbasli *et al.* (2014) indicated that the heat pump water heater in instantaneous heating mode has higher average COP than cyclic heating mode.

Previous researches also revealed that an auxiliary heat source appears necessary when the EHPWH is used in shower facilities, because the heat capacity of the heat pump is insufficient when using the shower wastewater as the single heat source. As a result, the overall energy efficiency of EHPWH is relatively low.

In recent years, the gas engine heat pump (GEHP) has received much attention around the world in terms of

overall energy utilization efficiency and energy-saving potential (Elgendy & Schmidt 2014). The GEHP has higher PER than the electrical heat pump (EHP), owing to the fact that the GEHP utilizes primary energy (natural gas) directly, and can recover the waste heat of the gas engine. In general, waste heat recovered from the gas engine is more than 55% of engine energy consumption (Liu *et al.* 2017, 2018). Hence, the GEHP technology applied in water heater equipment could be an available method to resolve the issue that the EHPWH needs an auxiliary heat source.

The objective of this work is to propose a novel gas engine heat pump water heater (GEHPWH) for the bathroom, which uses the shower wastewater as the heat source. Thermodynamic models will be established to investigate the system performance under different inlet water conditions. Furthermore, the energy saving potential and economic feasibility of GEHPWH will be investigated by comparing the energy consumption and total cost for four types of water heaters.

METHODS

Design consideration

For shower hot water, the supplied hot water temperature is mostly in the range of 40–50 °C (Shen *et al.* 2012; Liu *et al.* 2014). The objective hot water temperature is set as 45 °C and the shower wastewater temperature is set as 30 °C in this work. The water mass flow rate of tap water and wastewater are the same to ensure the water quantity balance.

Principles of the EHPWH

Figure 1 shows the schematic diagram of the general EHPWH, which includes a preheater, heat pump cycle, auxiliary heater, and hot storage tank. In the EHPWH, the compressor is driven by an electrical motor, and the preheater is set to preheat the tap water for reducing the energy consumption. The filled arrows denote the refrigerant cycle, and the blank arrows denote the streams of shower wastewater and tap water. On the wastewater side, the shower wastewater first flows into the preheater to preheat tap water, and then flows into the evaporator and releases heat to a low temperature refrigerant. On the tap water side, the temperature of tap water is heated to

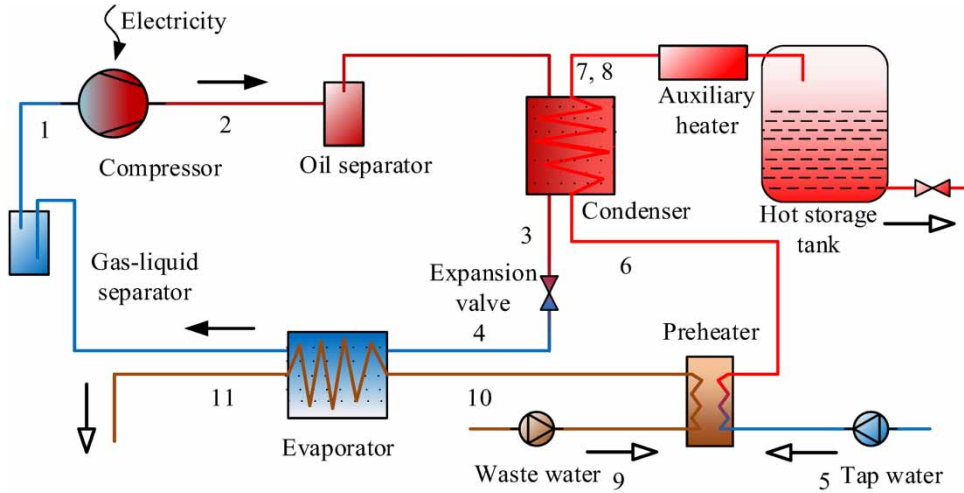


Figure 1 | Schematic diagram of EHPWH.

T_6 in the preheater, and heated to T_8 in the condenser. In addition, if the T_8 is below the objective hot water temperature, the auxiliary heater will switch on and provide extra heat.

Principles of the GEHPWH

Figure 2 shows the schematic diagram of the GEHPWH, which includes gas engine, engine heat recovery system, preheater, heat pump cycle, auxiliary heater, and hot storage tank. The heat pump cycle of the GEHPWH appears similar to the EHPWH, but the compressor is driven by the gas engine. In the pump cycle, the evaporator extracts low grade heat from the shower wastewater, and the condenser supply heat to tap water. The engine heat recovery system is equipped with an exhaust gas heat exchanger (HX1) and heat recovery heat exchanger (HX2). In detail, the internal cooling water recovers the waste heat from the HX1 and engine cylinder. Then, the internal cooling water releases the heat to tap water in the HX2. On the tap water side, tap water is heated in the preheater, condenser, and HX2 in sequence.

Modeling of the GEHPWH

Gas engine model

To determine the shaft power and heat recovery capacity of the gas engine, the thermal efficiency (η_m) and waste heat recovery efficiency (β) are used (Equations (1) and (2))

(Jeong *et al.* 2011):

$$\eta_m = \frac{P}{Q_{PE}} \quad (1)$$

$$\beta = \frac{Q_{HR}}{(1 - \eta_m) \cdot Q_{PE}} \quad (2)$$

where P is the shaft power of gas engine, Q_{PE} is the primary energy consumption of gas engine, β is the heat recovery efficiency of gas engine waste heat. In this paper, the η_m and the β are set to 30% and 80%, respectively.

Heat pump model

The graphic modeling method is used for modeling the compressor. The shaft power and the refrigerant mass flow of the compressor can be described by evaporating temperature (T_{eva}) and condensing temperature (T_{con}), which is calculated by Equations (3) and (4) (Liu *et al.* 2017):

$$P = \sum_{i=0, j=0}^3 a_{ij} \cdot T_{eva}^i \cdot T_{con}^j \quad (3)$$

$$m_r = \sum_{i=0, j=0}^3 b_{ij} \cdot T_{eva}^i \cdot T_{con}^j \quad (4)$$

where m_r is the mass flow rate of refrigerant, kg s^{-1} ; T_{eva} and T_{con} are the evaporating temperature and condensing

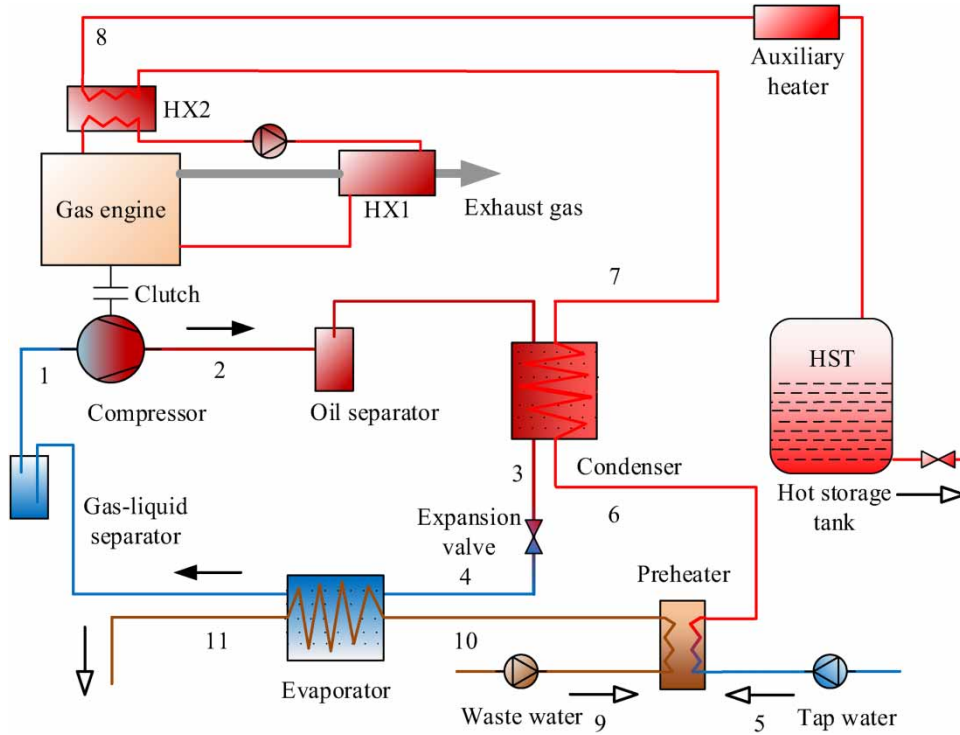


Figure 2 | Schematic diagram of the GEHPWH.

temperature, °C; a_{ij} and b_{ij} are the coefficients of polymerization, which were obtained from compressor selection software provided by the manufacturer, as shown in Table 1.

The lumped parameter method is chosen to establish the heat exchanger model based on the energy balance of each component (Equation (5)–(7)) (Shang et al. 2017; Wu et al.

2018):

$$Q = c_p \cdot m_w \cdot (T_{w,in} - T_{w,out}) \quad (5)$$

$$Q = m_r \cdot (h_{r,in} - h_{r,out}) \quad (6)$$

$$Q = UA \cdot LMTD \quad (7)$$

where Q is the heat duty, kW; m_w is the mass flow rate of water, kg s^{-1} ; T_{in} and T_{out} are the inlet and outlet temperature of water, °C; $h_{r,in}$ and $h_{r,out}$ are the inlet and outlet specific enthalpies of the refrigerant, kJ kg^{-1} ; UA is the product of heat transfer coefficient and heat transfer area; $\text{kW}^\circ\text{C}^{-1}$; $LMTD$ is the logarithmic mean temperature difference, °C. Technical parameters of the heat exchanger are shown in Table 2.

For the EHPWH and the GEHPWH, a gas boiler is chosen as the auxiliary heater. The heat provided by the gas boiler can be calculated by Equation (8):

$$Q_{GB} = \frac{c_{p,w} \cdot m_w \cdot (45 - T_8)}{\eta_b} \quad (8)$$

Table 1 | Coefficients of Equations (3) and (4)

ij	a_{ij}	b_{ij}
00	19.666	1.469
10	0.235	0.053
01	0.858	0.004
20	0.001	7.656×10^{-4}
11	0.007	8.171×10^{-5}
02	-0.003	-1.037×10^{-4}
30	-3.800×10^{-6}	5.414×10^{-6}
21	9.610×10^{-5}	5.756×10^{-8}
12	-1.322×10^{-4}	-7.050×10^{-7}
03	1.016×10^{-4}	4.003×10^{-7}

Table 2 | Technical parameters of the heat exchanger

Heat exchanger	UA (kW/k)	LMTD (°C)
Evaporator	31.40	8.46
Condenser	29.03	11.24
Preheater	16.80	9.53

Nominal condition: $T_s = 12\text{ }^\circ\text{C}$, $T_o = 30\text{ }^\circ\text{C}$, $m_w = 4.5\text{ kg/s}$.

where η_b is the boiler efficiency, which is typically 0.9 for a gas boiler.

The coefficient of performance (*COP*) and the primary energy ratio (*PER*) for the EHPWH and GEHPWH are calculated by Equations (9)–(11):

$$COP = \frac{c_{p,w} \cdot m_w \cdot (T_8 - T_6)}{P} \quad (9)$$

$$PER_{GEHPWH} = \frac{Q_{supply}}{Q_{GE} + Q_{GB}} \quad (10)$$

$$PER_{EHPWH} = \frac{Q_{supply}}{P/\eta_e + Q_{GB}} \quad (11)$$

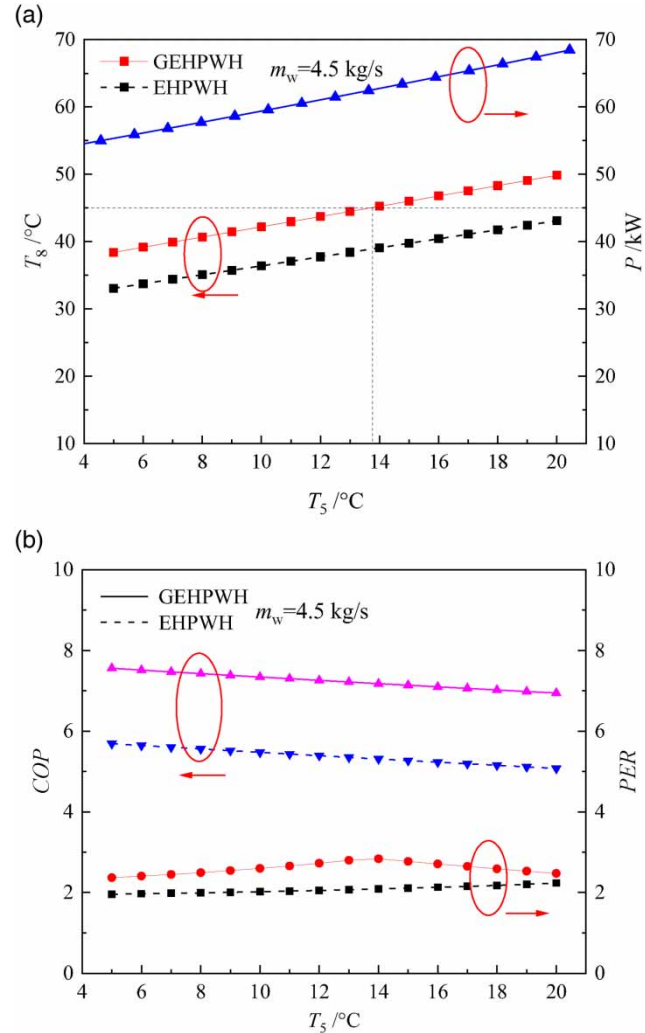
where Q_{supply} is the heat load which heats tap water from T_5 to supply water temperature, kW; η_e is the electricity generation efficiency, with an average value of 0.4 for natural gas generation plants, and the low heat value of natural gas is set to $36,750\text{ kJ/m}^3$.

The above models are solved using Engineering Equation Solver (EES) software package (Klein & Alvarado 2002).

RESULTS AND DISCUSSION

Influences of the tap water temperature

In order to study the influence of tap water temperature on performance, the outlet temperature hot water (T_8), P , *COP*, and *PER* at different tap water temperatures (T_5) are simulated when the water mass flow rate is 4.5 kg/s . The results are shown in Figure 3. Figure 3(a) shows that both the T_8 and P increase as the tap water temperature increases. The T_8 of the GEHPWH is higher than that of the EHPWH, owing to the waste heat recovery of the gas

**Figure 3** | Variation of performance with the tap water temperature.

engine. When the T_5 increases from 5 to $20\text{ }^\circ\text{C}$, the T_8 of EHPWH increases from 33.1 to $43.1\text{ }^\circ\text{C}$, and the T_8 of GEHPWH increases from 38.4 to $49.9\text{ }^\circ\text{C}$. T_8 of GEHPWH exceeds $45\text{ }^\circ\text{C}$ when the T_5 is higher than $13.7\text{ }^\circ\text{C}$. The T_5 increases results in the condensing temperature increases, which results in the increase of the compressor pressure ratio and shaft power of the compressor. The P increases 26.5% when the T_5 increases from $5\text{ }^\circ\text{C}$ to $20\text{ }^\circ\text{C}$.

Figure 3(b) illustrates the effects of T_5 on *COP* and *PER*. When the T_5 increases from 5 to $20\text{ }^\circ\text{C}$, the *COP* of EHPWH and GEHPWH decrease 10.8% and 8.1%, respectively. The *PER* of EHPWH increases as T_5 increases. This can be explained by the fact that the heat provided by the auxiliary

gas boiler decreases continuously as the hot water outlet temperature increases. For GEHPWH, the *PER* first increases and then decreases as T_5 increases. This is because when the T_8 is higher than 45 °C, the auxiliary gas boiler is not needed, and the higher hot water outlet water results in the worse performance of the heat pump cycle.

Influences of the water mass flow rate

In order to study the influence of m_w on performance, the T_8 , P , *COP*, and *PER* at different m_w are simulated when the tap water temperature is 12 °C. The results are shown in Figure 4. Figure 4(a) shows that both the T_8 and P decrease with the increase of the m_w . When the water mass flow increases from 2.5 kg/s to 6.5 kg/s, the T_8 of the

EHPWH decreases from 48.4 °C to 32.1 °C, and the T_8 of the GEHPWH decreases from 60.7 °C to 36.1 °C. The P decreases by 16.5% as the m_w increases from 2.5 kg/s to 6.5 kg/s.

Figure 4(b) shows the influences of the m_w on the *COP* and *PER*. At a certain T_5 , the *COP* of GEHPWH and EHPWH increases with the increase of m_w , and the former is about 1.3 times higher than the latter. The *PER* of EHPWH and GEHPWH first increases then decreases as m_w increases, which can be explained as follows. (1) When m_w is below 2.9 kg/s, the T_8 of EHPWH and GEHPWH is greater than 45 °C, and the auxiliary heat source is not needed. The increase of *COP* and the decrease of P causes the *PER* increases as the m_w increases. (2) When m_w is between 2.9 kg/s and 4.3 kg/s, the T_8 of GEHPWH is more than 45 °C, while the T_8 of EHPWH is less than 45 °C. The increase of heat demand from the gas boiler causes the *PER* of the EHPWH decreases. (3) When m_w is above 4.3 kg/s, the T_8 of the EHPWH and the GEHPWH are all below 45 °C, the increase of heat demand from the gas boiler causes the *PER* decreases as the m_w increases.

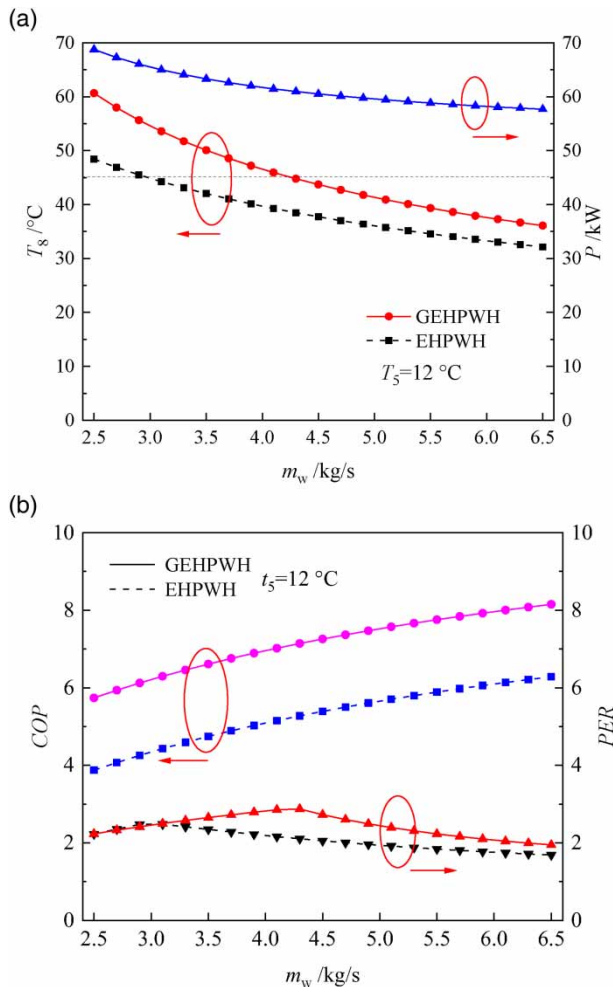


Figure 4 | Variation of performance with the water mass flow.

The performance of the GEHPWH in constant/variable water flow mode

The previous section presented the performance comparison between the EHPWH and the GEHPWH. Although the GEHPWH has higher hot water outlet temperature and *PER* than the EHPWH, the hot water outlet temperature of the GEHPWH still cannot reach set temperature under some conditions. In this section, the operational mode of the GEHPWH is optimized by adjusting the water flow rate. The performance of GEHPWH in variable water flow mode (mode 1) and constant water flow mode (mode 2) are analyzed and compared. In mode 1, the T_8 is maintained at 45 °C by adjusting the m_w according to the T_5 . In mode 2, the m_w is 4.5 kg/s.

Figure 5 describes the performances of the GEHPWH in mode 1 and mode 2. When the T_8 is maintained at 45 °C, the variation of the m_w with the T_5 is indicated in Figure 5(a). In mode 1, the m_w rises from 3.46 kg/s to 5.62 kg/s as the T_5 increases from 5 °C to 20 °C. Figure 5(b) indicates the influences of the T_5 on T_8 and P . When T_5 increases from 5 °C to 20 °C, the T_8 can be maintained at 45 °C in mode 1, while

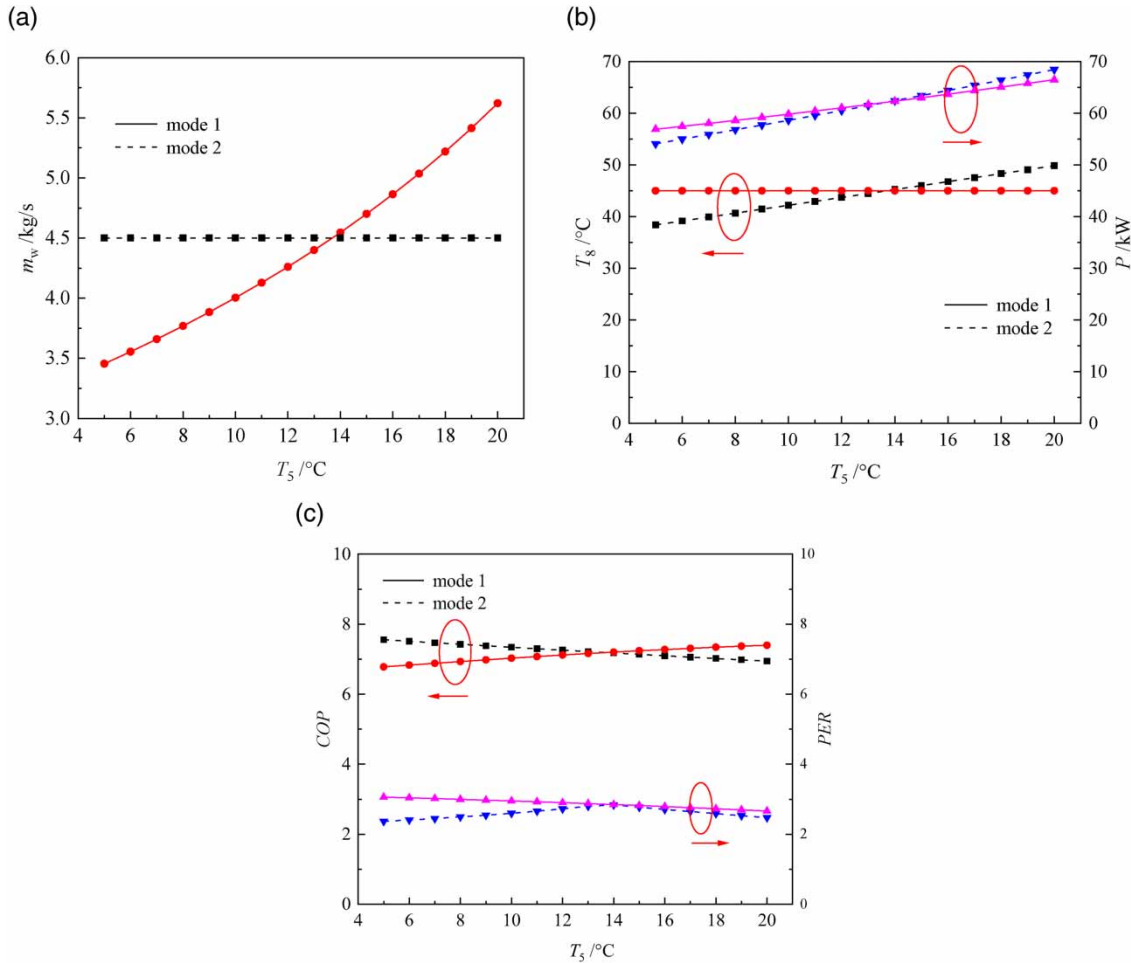


Figure 5 | The performance of the GEHPWH in constant/variable water flow mode.

for mode 2 it is from 38.4 °C to 49.9 °C. Owing to the m_w increases as the T_5 increases in mode 1, the growth rate of P in mode 1 is less than that in mode 2.

In Figure 5(c), it can be seen that with tap water temperature increases from 5 °C to 20 °C, the COP increases from 6.8 to 7.4 in mode 1, while the COP decreases from 7.5 to 6.9 in mode 2. This can be explained by the fact that the increase of m_w contributes to a lower pressure ratio of the compressor in mode 1, while the pressure ratio of the compressor increases as the T_5 increases in mode 2. In mode 1, the PER decreases slightly as the T_5 increases because the growth rate of Q_{PE} is more than the growth rate of Q_{supply} . The PER of the GEHPWH in mode 1 is always higher than that in mode 2 with the variation of T_5 . Through the above result, the GEHPWH operating in mode 1 has not only a stable hot water

outlet temperature, but also better performance and practicability.

Case studies

A college public bathroom in the Tianjin area was chosen to investigate the energy-saving potential of the GEHPWH. This bathroom generally consumes 200 tons of 45 °C hot water each day. Figure 6 illustrates the mean temperature of ambient air and tap water every month in Tianjin city.

Four hot-water schemes, GEHPWH, EHPWH, ASHPWH, and gas fired boiler, were selected to compare primary energy consumption and economic feasibility. Among the four hot-water schemes, the preheater was installed to preheat tap water. The PERs of ASHPWH and gas fired boiler were chosen according to Li et al. (2015).

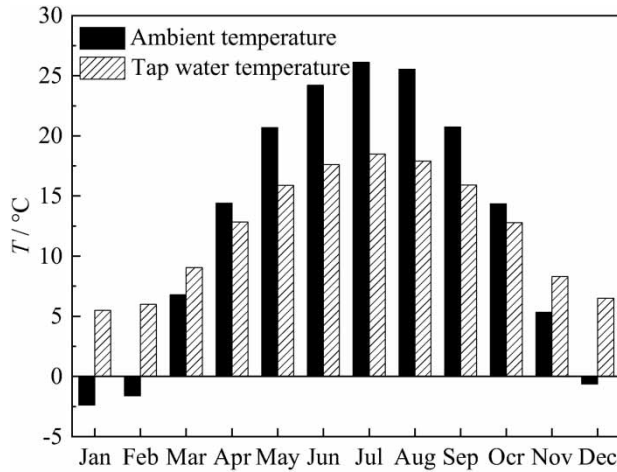


Figure 6 | Monthly mean temperature of ambient air and tap water in Tianjin.

The initial costs of the different hot-water schemes were chosen according to Liu *et al.* (2010, 2014). The price of natural gas was 2.8 Yuan/m³ and the price of electricity 0.8 Yuan/kWh.

Figure 7(a) depicts the monthly natural gas consumption comparison for the four water heating schemes. It can be seen that the natural gas consumption of the GEHPWH is lower and more stable all year around. The natural gas consumption of the ASHPWH changes significantly in different months because the *COP* of the ASHPWH is influenced directly by ambient air temperature. The natural gas consumption of the ASHPWH is much more than the gas boiler in December, while less than that of EHPWH in June.

Figure 7(b) presents the initial cost and the operating cost for ten years for the four hot-water schemes. The ranking of the initial costs is as follows: GEHPWH > ASHPWH > EHPWH > gas boiler. The GEHPWH has the highest initial cost due to its complicated structure, while initial cost is lowest for the gas boiler. Considering the projected operating cost for the estimated ten years of service, the ranking of the total costs of the four schemes is as follows: GEHPWH < EHPWH < ASHPWH < gas boiler. The gas boiler hot-water producing system is based on fuel burning, which presents the lowest energy efficiency and the highest operating cost. Compared with the EHPWH, ASHPWH, and gas boiler, the GEHPWH shows an enormous economic advantage due to 44.8%, 58.5%, and 61.8% of cost saving, respectively.

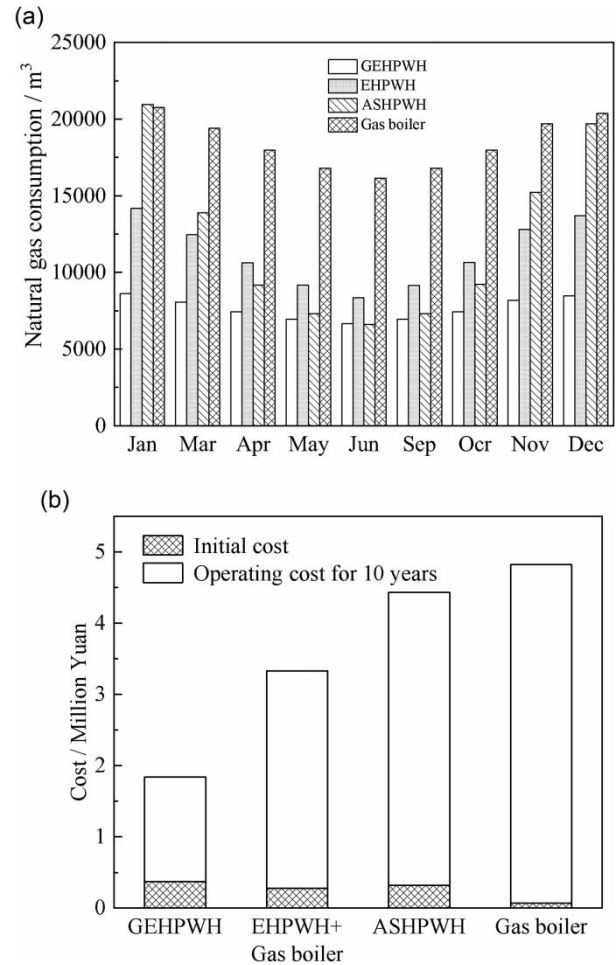


Figure 7 | Initial cost and operating cost of the four schemes.

CONCLUSIONS

A gas engine heat pump water heater (GEHPWH) was used in this study to find the feasibility and energy-saving potential for public shower facilities. The thermodynamic model of the GEHPWH was established to compare with EHPWH. The detailed performance of the GEHPWH and EHPWH were investigated under different tap water temperatures and water flow rates. The main conclusions can be drawn as follows:

1. When the inlet water conditions are the same, the GEHPWH has higher hot water outlet temperature than the EHPWH, and the *COP* of the GEHPWH is approximately 1.3 times higher than the EHPWH.
2. In the variable water flow rate mode, the GWHPWH can produce a stable hot water outlet temperature with high *PER*.

3. The GEHPWH has smaller energy consumption and lower operating cost compared with the conventional EHPWH, ASHPWH, and gas boiler. Thus, the GEHPWH can produce better economic and social benefits in bathroom application when recovering the heat of shower wastewater.

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