J Ther Eng, Vol. 9, No. 2, pp. 551–564, March 2023

Journal of Thermal Engineering
Web page info: https://jten.yildiz.edu.tr
DOI: 10.18186/thermal.1285281

Research Article

A theoretical analysis on the operating and design parameters affecting the performance of a sewage wastewater sourced heat pump system

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ARTICLE INFO

Article history
Received: 4 June 2021
Accepted: 28 October 2021

Keywords:
Heat Pump; Sewage Wastewater; Indirect Type Heat Exchanger; COP

ABSTRACT

Sewage wastewater heat exchanger (SWHE) has a significant role in the performance of sewage wastewater sourced heat pump (SWSHP) system as it provides to transfer the energy of wastewater to intermediary fluid or working fluid. Thus, a theoretical analysis of the SWSHP system was carried out to investigate the effects of SWHE design parameters on the system's performance. For this purpose, a simulation program based on the proposed mathematical model of the SWSHP system was developed in MATLAB. Afterward, the indirect type SWSHP system that can meet 50 kW heating load was theoretically designed. The influences of SW temperature, its mass flow rate, the inner diameter of the heat exchanger tube, and intermediary fluid mass flow rate on the performance of the designed SWSHP system were analyzed. The results indicate that variation of SW temperature affects the COP_{sys} more than the variation of SW mass flow rate. Considering the ranges of parameters investigated, the COP_{sys} raises from 2.56 to 4.51 and 2.89 to 4.27 with the variations of SW temperature and SW flow rate, respectively. Moreover, an increase in the intermediary fluid mass flow rate provides an improvement on the COP_{sys} and COP_{unit}. However, SWSHP performance is adversely affected by the increasing value of the inner diameter of the tubes. As a result, small changes in the design parameters of the SWHE directly affect the system performance and system operating conditions.

Cite this article as: Ercan D, İsmail S, Özgür B. A theoretical analysis on the operating and design parameters affecting the performance of a sewage wastewater sourced heat pump system. J Ther Eng 2023;9(2):561–574.

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This paper was recommended for publication in revised form by Regional Editor Younes Menni

Published by Yıldız Technical University Press, İstanbul, Turkey

INTRODUCTION

With the developing technology and increasing world population, the energy demand is gradually growing. Most of this energy demand is met by fossil fuels such as crude oil, natural gas, and coal. However, these energy resources are non-renewable, create economic problems if they are imported, and lead to negative impacts on the environment. Therefore, shifting the energy demand from fossil fuels to domestically produced renewable energy should be a high priority for the governments.

The share of heating load plays an important role in the residential and public buildings' total energy demand which is generally relied on fossil fuels. Therefore, heat pump (HP) technology could be a promising alternative to reduce and possibly eliminate the reliance on fossil fuel for meeting the heating load of buildings due to its cost-effective, energy-efficient, and environmentally friendly characteristics, and it has been successfully used in many commercial and industrial applications, recently [1–3]. In HP systems, water, air, and soil are generally used as low-temperature heat sources [4] . Unlike these heat sources, the systems assisted by sewage wastewater (SW) have also been developed in recent years. The sewage wastewater-sourced heat pump (SWSHP) systems were first developed in Norway in the early 1980s and later, a 3.3 MW system was installed in Sweden [5]. SWSHP systems have not been studied extensively in the literature and have not received enough attention. However, some of the studies on the heat recovery from sewage waste or the SWSHP designs in the literature are stated as follows. Zhou and Li [6] investigated the characteristics of SW, and the HP system installed in a wastewater treatment plant in China in terms of economical and technical aspects. The authors reported that SW temperature and flow rate do not show significant variation during the year, and it is an important resource for HP systems due to its cost-effective feature. Cipollo and Maglionico [7] monitored the daily and seasonal alteration of the temperature and flow rate of the SW in Bologna, Italy. The authors reported that the SW temperature varies from 11 °C to 16 °C in winter and the thermal power potential of Bologna's sewage system is 74 kW with a 5.9°C temperature drop and 3 L/s flow rate. In general, the researchers have acknowledged that energy recovery from wastewater has a salient potential, and its utilization can greatly reduce carbon emissions and pollutants released into the environment [8–11]. Oh et al. [12] dynamically analyzed the thermal storage tank of the SWSHP in a vertical water treatment building using the TRNSYS simulation program. Both single and double HP systems have been studied by taking into account different heating zones of the building. The average COP values for single and double HP systems were calculated as 4.79 and 4.92 for heating, and 3.76 and 3.68 for cooling, respectively. Cho and Yun [13] reported that the raw wastewater HP system designed with the flow rate of 190 L/min which is approximately 3.17 kg/s for the heating and cooling of the control room in a treatment plant is more efficient compared to the air-sourced HP system; in addition, it was found that the average COP value is 3.3 and 7.2 for heating and cooling seasons, respectively. Li and Li [14] compared the performance of the SWSHP system with the performance of the geothermal water sourced HP system. It was reported that the efficiency of the SWSHP system is higher than that of the geothermal-water-sourced HP system under the same operating conditions. It was also mentioned that the exergy losses occurred mostly in the compressor. Additionally, in the literature, two different design approaches, namely direct and indirect have been used to transfer heat from SW to the refrigerant fluid. In the direct systems, SW enters the evaporator and releases its heat to the refrigerant. On the other hand, in the indirect type systems, the SW enters an external heat exchanger and the heat is transferred from the SW to the intermediate fluid. Then, the intermediate fluid enters the evaporator and transfers its heat to the refrigerant. Qian [15] emphasized in his study that the COP value of the directly designed system is 0.3 higher and the SW temperature affects the performance linearly in both system designs. The direct type design shows higher performance, but in order to implement it on the system, SW must be treated before entering the evaporator since heat transfer is negatively affected by the presence of dirt and other particles in the SW. Therefore, directly designed systems are equipped with filtering devices to prevent the stated problem[16]. However, filtering devices cannot completely remove particles smaller than 2 mm; hence, these small particles enter the heat exchanger and cause fouling [17]. Fouling adversely affects the performance and life of the system, and thanks to that, many researchers have conducted studies to investigate the consequences, reduction, and prediction of fouling. While generally artificial wastewater is used in experimental studies, Song et al. [18] used the real SW in their study and developed a mathematical model to calculate the asymptotic fouling resistance. In the developed model, the author expressed the asymptotic fouling resistance as a function of the initial velocity of the SW and time constant and it can be used in the theoretical design of heat exchangers. Many researchers have conducted studies on the various type of sewage heat exchangers [19–21] using different materials like steel, plastic [22], and composite [23]. However, Culha et al. [24] reported that most academic studies and commercial applications occur on the exclusive design of shell and tube heat exchangers. Shen et al. [25] conducted experimental and theoretical studies on a direct type shell and tube heat exchanger called DESTE. They reported that increasing the wastewater mass flow rate from 0.45 L/s to 1.43 L/s approximately increases the COP from 2.45 to 3.5. Also, they reported that the capacity of the heat exchanger decreases when the thermal resistance -due to fouling- is greater than 2.5x10. Liu et al. [26] experimentally and numerically tested the heating performance of the actual indirect HP system using untreated SW with an inlet temperature of 11°C as a source and reported that the thermal resistance caused by convection and pollution constitutes 80% of the total thermal resistance. Qin and Hao [27] theoretically analyzed the performance of a direct sewage source heat pump when the temperature of SW at the inlet is between 12 °C and 13 °C. It was pointed out that the system's COP is approximately 3.75.

In the available literature, it is observed that the effect of fouling resistance on the performance of sewage heat exchangers is generally neglected or evaluated by using the estimated values of fouling resistance. In addition, numerical studies investigating the temperature and flow rate effects on the system components are scarce, or not reported in detail. In this framework, unlike to open literature, a mathematical model of the SWSHP system including the fouling effect in an indirect type SW heat exchanger is proposed by benefiting from up-to-date literature, and then a simulation program is built in MATLAB within the framework of this model to analyze the several SW heat exchanger design parameters and operating conditions in detail. Afterward, the indirect type SWSHP system that can meet 50 kW heating load is theoretically designed. Finally, the effects of the temperature of the SW, its flow rate, the inner diameter of the heat exchanger tube, and the intermediary fluid mass flow rate on the performance of the SW heat exchanger, fouling resistance, total pumping power, evaporator temperature, refrigerant's mass flow rate and its quality at the inlet of the evaporator and overall system performance are numerically analyzed.

MODELLING

The SWSHP systems consist of three cycles: end-user, HP, and SW cycle. The HP cycle consists of an evaporator, compressor, condenser, throttling valve, and auxiliary equipment. The SW side includes a specially designed heat exchanger and intermediary fluid section to transfer the waste heat to the evaporator. In the designed system, the end- user cycle was not considered, and a mathematical model was developed for the combined HP and SW cycles. The schematic view of the SWSHP system and the *P-h* diagram of the HP cycle are given in Fig 1a. and 1b.

Heat Pump Cycle

The developed thermodynamic model for the HP cycle is presented as follows. The compressor's power required to raise the refrigerant pressure from point 1 to 2 was evaluated

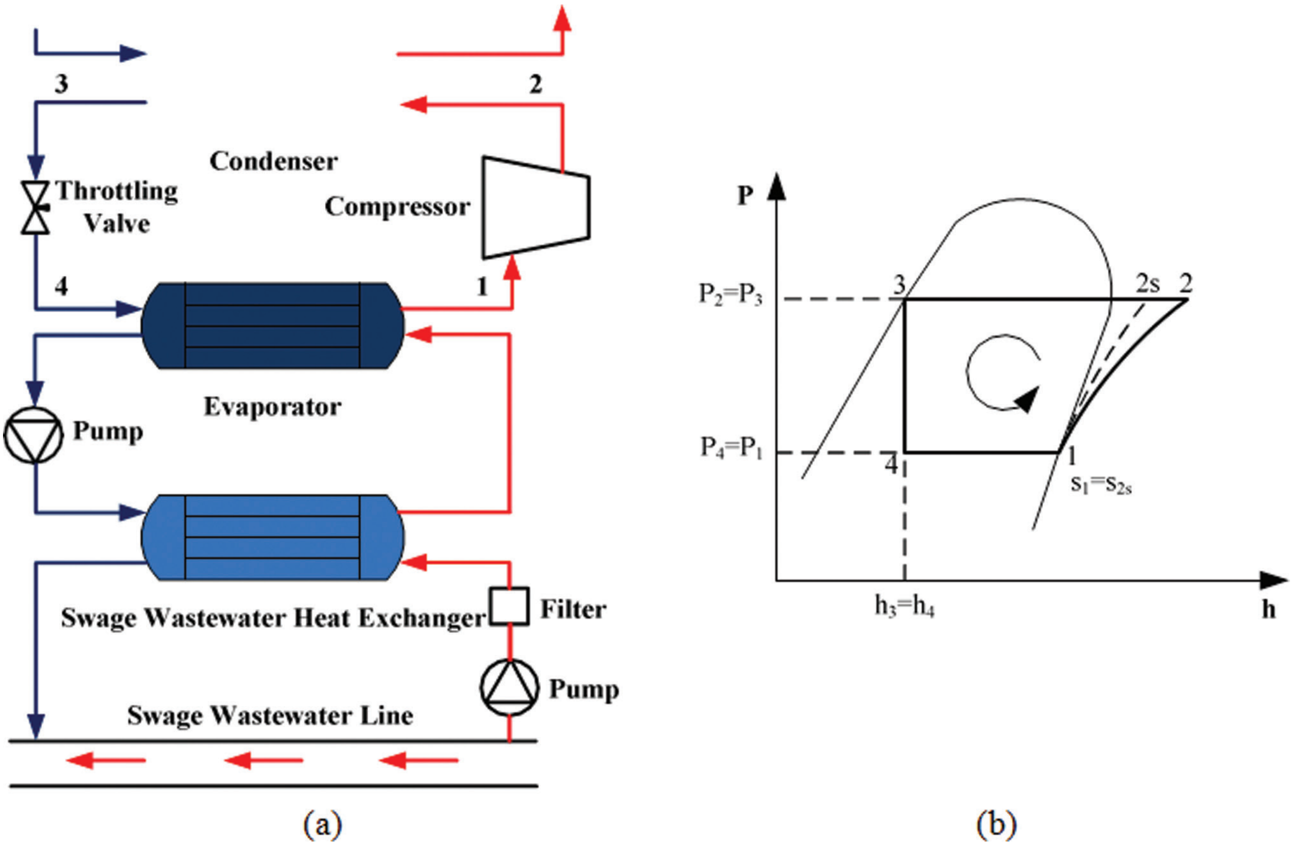


Figure 1. a) the schematic view of the SWSHP system, **b)** the *P-h* diagram of the HP cycle.

by taking its isentropic, electrical, and mechanical efficiencies into account and computed by Eq. (1).

$$W_{comp} = \frac{m_r.(h_{2s} - h_1)}{\eta_m.\eta_{el}.\eta_{is}} \quad (1)$$

It was assumed that there is no heat loss in the condenser, and hence, the Eqs. (2) and (3) can be written.

$$m_r.(h_2 - h_3) = m_a.c_{p\_a}.(T_{a\_o} - T_{a\_in}) \quad (2)$$

$$m_r.(h_3 - h_4) = m_w.c_{p\_w}.(T_{w\_in} - T_{w\_o}) \quad (3)$$

The rate of heat transfer from refrigerant to air in the condenser and SW to the refrigerant in the evaporator can be easily calculated by the logarithmic mean temperature difference (LMTD) approach.

$$Q_c = U_c.A_c.\Delta T_{im\_c} \ \mathit{and} \ Q_e = U_e.A_e.\Delta T_{im\_e} \quad (4)$$

Where,

$$\Delta T_{lm\_c} = \frac{(T_3 - T_{a,o}) - (T_c - T_{a,in})}{\ln[(T_3 - T_{a,o})/(T_c - T_{a,in})]} \ \mathit{and}$$
$$\Delta T_{lm\_b} = \frac{(T_{w,o} - T_4) - (T_{w,in} - T_1)}{\ln[(T_{w,o} - T_4)/(T_{w,in} - T_1)]} \quad (5)$$

The refrigerant enthalpy was assumed to be the constant through the throttling valve ($h_3=h_4$) because of the negligible heat transfer.

Sewage Wastewater Cycle

Special attention needs to be given to the design of SWHE because of the fouling problem adversely affecting the heat transfer. In this study, the SW part of the system was equipped with the most preferred shell and tube heat exchanger in the literature. The heat transfer model for the SWHE is presented below. The maximum rate of heat that can be benefited from SW through the SWHE is expressed by Eq. (6). The amount of heat transferred to the intermediary fluid from the SW is specified by Eq. (7).

$$Q_{sw} = m_{sw}.c_{p\_sw}.(T_{sw\_in} - T_{sw\_o}) \quad (6)$$

$$Q_w = m_w.c_{p\_w}.(T_{w\_o} - T_{w\_in}) \quad (7)$$

The overall heat transfer coefficient (U_{SWHE}) for the SWHE is evaluated by Eq. (8). In this equation, the fouling resistance on the intermediary fluid side was neglected since it is very small compared to the other thermal resistances.

$$U_{SWHE} = 1/\left(R_{f\_sw} + \ln(d_o/d_i)/(2.\pi.L.k_t) + 1/\alpha_{sw} + 1/\alpha_w\right) \quad (8)$$

Fouling resistance is defined as a function of the time and the initial velocity of the SW and it approaches an asymptotic value over time [18]. In this work, the asymptotic value of the fouling resistance of the SW side ($R_{f,\_sw}$) was taken into account and evaluated by Eq. (9) that can be used in the range of wastewater velocities from 0.82 m/s to 1.60 m/s for SW velocity.

$$R_{f\_sw} = (12.9 - 4.75V_{sw\_initial}).10^{-4} \quad (9)$$

In the modeling, it was assumed that the SW flows through the tubes and the intermediary fluid passes through the shell side in SWHE. The convective heat transfer coefficients for the tube and shell side are given in Eqs. (10) and (11), respectively.

$$\alpha_{sw} = 0.012.(\mathrm{Re}^{0.87} - 280).\mathrm{Pr}^{0.4}\left[1 + (d_i/L)^{2/3}\right].(k_{sw}/d_i) \quad (10)$$

$$\alpha_w = 0{,}7.0{,}35.\mathrm{Re}^{0.6}.\mathrm{Pr}^{0.36}.(k_w/d_o) \quad (11)$$

The velocity of the SW passing through the tubes was computed by Eq. (12) [28].

$$V_{sw} = (4.m_{sw}.s/\pi.d_i^2.\rho_{sw}).\left[1/C.((D_{shell} - 0.02)/d_o)^n\right] \quad (12)$$

Eqs (13) includes a set of equations. It was first used to design the shell part of the heat exchanger in question, andthen, to evaluate the velocity of intermediary fluid which flows through the shell side [28,29].

$$S_t = 1.25d_o,\ D_{ed\_sq} = 1.27/d_o.(S_t^2 - 0.785d_o^2),$$
$$D_{ed\_tr} = 1.1/d_o.(S_t^2 - 0.917d_o^2)$$

$$A_{shell\_ca} = (S_t - d_o).e.D_{shell}/S_t,\ e = L/(N_b + 1)$$

$$V_w = m_w/\rho_w.A_{shell\_ca} \quad (13)$$

The Number of Transfer Units (NTU) method is applied to estimate the outlet temperatures of the SW and intermediary fluid. The effectiveness of the SWHE was evaluated by the Eq. (14) employed for one shell and n tube pass crossflow heat exchangers without mixing fluids.

$$\varepsilon = 2\left\{1 + C_r + (1 + C_r^2)^{0.5} / \frac{1 + \exp\left[-NTU(1 + C_r^2)^{0.5}\right]}{1 - \exp\left[-NTU(1 + C_r^2)^{0.5}\right]}\right\}^{-1} \quad (14)$$

Eqs. (15) and (16) were used to calculate the outlet temperatures of the SW and intermediary fluid.

$$T_{sw\_o} = T_{sw\_in} - \varepsilon . C_{\min} . \left(T_{sw\_in} - T_{w\_in} \right) \big/ m_{sw} . c_{p\_sw} \quad (15)$$

$$T_{w\_o} = T_{w\_in} - \varepsilon . C_{\min} . \left(T_{sw\_in} - T_{w\_i} \right) \big/ m_{w} . c_{p\_w} \quad (16)$$

Pressure Drop in SWHE

The pressure drop (ΔP_t) in the SWHE's tubes was calculated by Eq. (17) [29]. The f_t expresses the Fanning friction coefficient and can be computed by the Eq. (18) [30].

$$\Delta P_t = \rho_{sw} \left(2.f_t.N_t.L.V_{sw}^2 \big/ d_i + 1{,}25.N_t.V_{sw}^2 \right) \quad (17)$$

$$f_t = 0.079 \big/ \mathrm{Re}_{sw}^{0.25} \quad (18)$$

The pressure drop in the shell side (ΔP_{shell}) was evaluated by Eq. (19)[31]

$$\Delta P_{shell} = 8.j_{f\_k}.\left(D_{shell} / D_{ed\_t} \right).\left(L/e \right).\left(\rho_w .V_w^2 /2 \right) .\left(\mu_w / \mu_{wall} \right)^{-0.14} \quad (19)$$

The following correlation proposed by Kızılkan is utilized to evaluate the dimensionless pressure factor (j_{f_k}) [31]. In the below equation, x represents the ln(Re).

$$\ln j_{f\_k} = \frac{-0.1292x^5 + 6.332x^4 - 136.3x^3 + 1257x^2 - 5232x + 7703}{x^3 + 45.12x^2 - 604.9x + 1136} \quad (20)$$

μ_{wall} needs to be evaluated at wall temperature (T_{wall}) that can be estimated by Eq. (21).

$$T_{wall} = 0{,}5.\left[\left(T_{sw\_in} + T_{sw\_o} \right) /2 + \left(T_{w\_in} + T_{w\_o} \right) /2 \right] \quad (21)$$

Performance Analysis of the System

The coefficient of performance of the heat pump cycle (COP_{unit}) and the whole system (COP_{sys}) are defined as in Eqs. (22) and (26), respectively.

$$COP_{unit} = Q_{load} \big/ W_{comp} \quad (22)$$

$$COP_{sys} = Q_{load} \big/ \left(W_{comp} + W_{p\_t} + W_{p\_shell} \right) \quad (23)$$

where,

$$W_p = \Delta P.m \big/ \rho.\eta_p \quad (24)$$

The correlations proposed in the literature were utilized to calculate the thermophysical properties of SW, intermediary fluid (clean water), and refrigerants [25,32,33].

In the next subsection, a general designing of SWSHP to conduct the parametric study was done by a MatLab simulation program based on the proposed mathematical model of the SWSHP system.

Design of the HP Unit and SWHE

Firstly, by benefiting the literature [33–35], the following operating conditions were taken into consideration to the size the capacity of HP parts for various refrigerants such as R134a, R404a, R410a, R502, R22, and R12.

- The heating load or capacity of the condenser was assumed to be 50 kW.
- The ambient temperature was constant at 23°C.
- Condenser and evaporator temperatures were accepted to be 35°C and 1°C, respectively.
- Air enters the condenser at 20°C and goes out at 30°C.
- Intermediary fluid enters the evaporator at 8°C and comes out at 5°C.
- The compressor's isentropic, mechanical, and electrical efficiencies are 0.70, 0.85, and 0.90, respectively.

Fig 2a shows the algorithm of HP part design and, algorithm for SWSHP system performance analyses is given in Fig 2b. Under the conditions presented above, the capacity of the HP system's parts for various refrigerants was determined by benefiting from the developed simulation program in MATLAB based on the proposed thermodynamic model above. The obtained results from this analysis are presented in Table 1. It is clear from Table 1 that R134a and R12 refrigerants have a superior impact on the COP_{unit}. In this study, R134a was chosen to be the refrigerant since it is more environmentally friendly and leads to a higher COP value.

Due to its characteristics, SW causes significant fouling in the heat exchanger. This fouling adversely affects the performance of the heat exchanger and hence, the performance of the overall system. Therefore, special attention needs to be given to the design of the SWHE. In this work, the SWHE was re-designed by taking the following recommended criteria in the literature into consideration [35,36] The velocity of the SW, the inner diameter of the tubes and the difference between the inlet and outlet temperatures of the SW were taken to be around 0.9-1.1 m/s, 16-20 mm and 2-4 °C, respectively. According to the proposed mathematical model, a MATLAB program based on a trial and error approach was developed to design the SWHE in question. In the SWHE design, it was assumed that the SW enters the SWHE with a flow rate of 4.50 kg/s at 14°C, while the intermediary fluid enters the SWHE at a flow rate of 3.28 kg/s at a temperature of 5°C and leaves at a temperature of 8°C. The values of the output parameters under the assumed operating conditions are presented in Table 2.

Parametric Analysis on the SWHE and SWSHP System

Up to this point, the HP and SWHE are designed according to the assumed design input parameters given in section 2. In fact, the parameters such as the flow rate

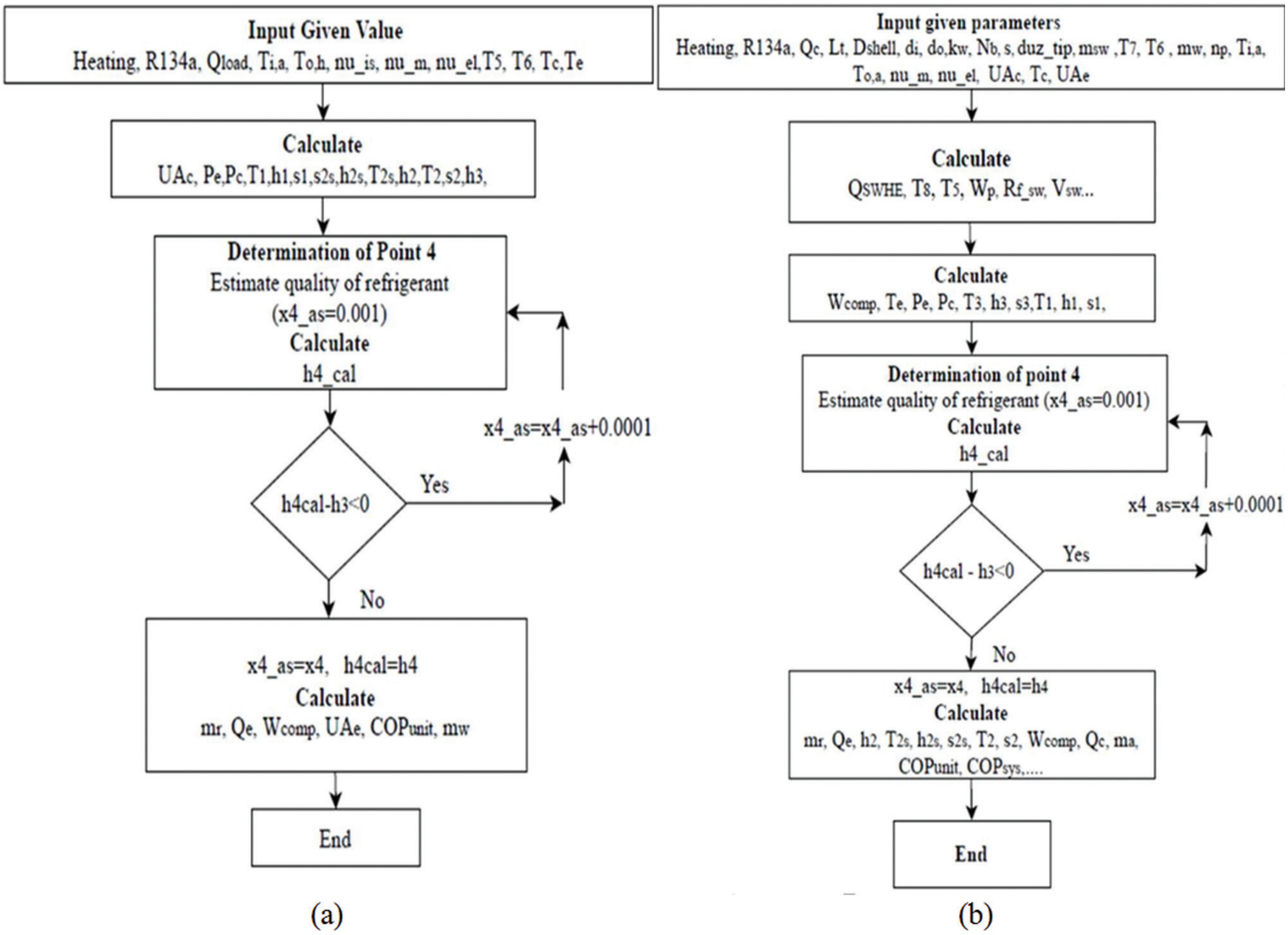


Figure 2. a) Algorithm for the HP part design **b)** Algorithm for the SWSHP system performance analyses.

Table 1. The capacity of HP system's parts for various refrigerants

	Refrigerant					
	R134a	**R404a**	**R410a**	**R502**	**R22**	**R12**
COP_{unit}	4.40	3.97	3.91	4.03	4.16	4.41
Q_e (kW)	41.31	40.35	40.22	40.50	40.81	41.35
P_e (bar)	3.03	6.11	8.27	5.58	5.12	3.19
$(UA)_e$ (kW/C)	7.70	7.53	7.50	7.55	7.61	7.71
Q_c (kW)	50	50	50	50	50	50
P_c (bar)	8.85	15.74	21.42	14.46	13.46	8.53
UA_c (kW/C)	5.49	5.49	5.49	5.49	5.49	5.49
W_c (kW)	11.36	11.35	12.78	12.41	12.01	11.30
x	0.2437	0.3110	0.2603	0.2713	0.2086	0.2158
m_r (kg/s)	0.2757	0.3555	0.2466	0.3679	0.2527	0.3453
m_w (kg/s)	3.28	3.20	3.19	3.21	3.24	3.28
m_a (kg/s)	4.98	4.98	4.98	4.98	4.98	4.98

Table 2. Design parameters of SWHE

INPUT DESIGN PARAMETERS		OUTPUT DESIGN PARAMETERS			
Geometrical Parameters		Geometrical Parameters		Heat Transfer Values	
L_t (m)	2.43	N_t	38.00	U_{SWHE} (W/m°C)	584
d_i (m)	0,017	S_t (m)	0.0263	UA_{SWHE} (W/m)	7.11
d_o (m)	0,021	e (m)	0.0714	Q_{SWHE} (kW)	41.35
D_{shell} (m)	0.240	D_{ed} (m)	0.0149	T_{sw_o} (°C)	11.89
N_b	13	$A_{shell\_ ca}$ (m^2)	0.0034	T_{w_o} (°C)	8.00
Arrangement	Triangle	A_{hta} (m)	12.184		
s	2	Flow Properties		Circulation Pump Load	
Intermediary Fluid		V_{sw} (m/s)	1.05	ΔP_t (kPa)	317.66
m_w (kg/s)	3.28	V_w (m/s)	0.96	ΔP_{shell} (kPa)	97.36
T_{w_in} (°C)	5.00	Re_{sw}	5879	W_{p_t} (kW)	1.79
SW		Re_w	9386	W_{p_shell} (kW)	0.40
m_{sw} (kg/s)	4.50	Thermal Resistances		Convection Coefficients	
T_{sw_in} (°C)	14.00	R_t (m°C/ W)	0.00023	α_{sw} (W/m°C)	2318
Other		R_{f_sw} (m°C/ W)	0.00079	α_w (W/m°C)	3887
η_p	0.80				
k_t (W/m.°C)	60.00				

and temperature of the SW may vary in seasonal or daily periods and hence, the SWSHP system's performance. Additionally, SWHE's tube inner diameter and intermediary water mass flow rate have great importance on the SWHE performance. Therefore, a parametric analysis has been conducted to examine the influence of these parameters on the fouling resistance, rate of heat transfer in the SWHE, total pumping power, evaporator temperature, the intermediary fluid temperature at the evaporator inlet, steam quality, refrigerant mass flow rate, COP_{unit} and COP_{sys}.

RESULTS AND DISCUSSION

The Effect of Investigated Parameters on the SWHE Performance

The effects of the investigated parameters on the V_{sw} and the R_{f_sw} are presented in Fig. 3. It is seen that the V_{sw} increases linearly as the m_{sw} is increased. As it is expected that when the m_{sw} is decreased, the V_{sw} decreases and the R_{f_sw} increases accordingly. An increase in d_i results in a decrease in the V_{sw} and hence, an increase in the R_{f_sw}. The temperature of SW and m_w are a negligible influence on the V_{sw} and R_{f_sw}.

Fig. 4 shows the influence of investigated parameters on the Q_{SWHE} and U_{SWHE}. The U_{SWHE} is positively affected by the increasing value of m_{sw}. The reason behind this behavior can be explained by the fact that an increase in the m_{sw} causes a decrease in the R_{f_sw} which creates an enhancement on both U_{SWHE} and Q_{SWHE} in SWHE. It is also clear that SWHE with the U_{SWHE} are adversely affected as the d_i is increased. Since, an increase in d_i causes a decrease in the V_{sw} and hence, an increase in the R_{f_sw}. The influence of SW temperature on the U_{SWHE} is very small as it is compared to Q_{SWHE} in SWHE. This is due to the fact that the physical properties of the SW do not vary significantly in the range of SW temperature investigated, and thus, the U_{shell} almost remains the constant. Additionally, it is stated above that R_{f_sw} variation with SW temperature is also very small. The seen positive effect on the Q_{SWHE} in SWHE is sourced from an increase in temperature difference between fluids. A significant enhancement on the Q_{SWHE} in SWHE is also observed as the m_w is increased.

The influence of investigated parameters on the W_{p_total} is shown in Fig. 5. The W_{p_total} increases with the increasing value of m_{sw} since it causes higher ΔP_t. The variation in the SW temperature does not affect the V_{sw} and the thermos-physical properties of the SW, and hence, the W_{p_total} nearly remains constant. An increase in d_i leads to a decrease in the V_{sw}, and hence, W_{p_total}. The increasing value of m_w creates an adverse influence on the W_{p_total} because of the increasing pressure drop.

The effect of considered parameters on the HP_{unit} performance

In indirect type SWSHP systems, the amount of energy extracted from the SW directly affects the working conditions and capacities of the HP parts. The effects

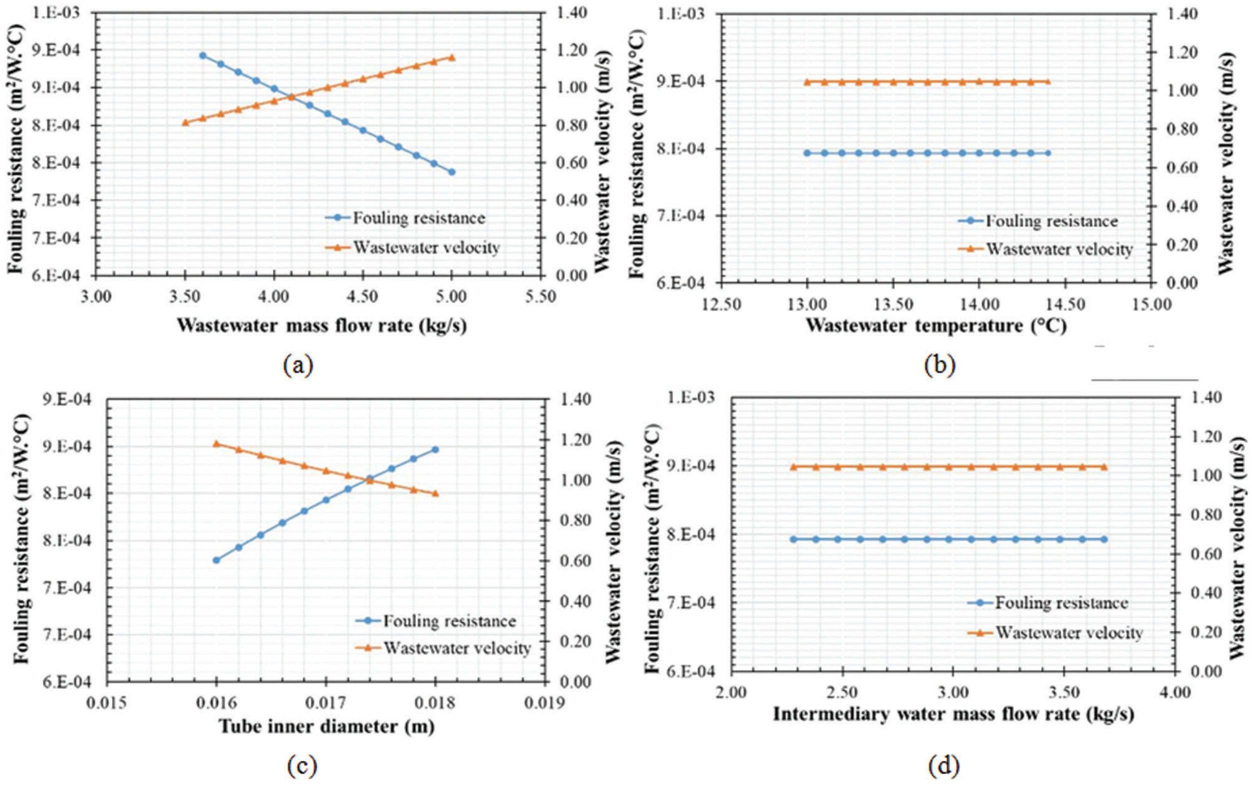


Figure 3. The effect of investigated parameters on SW velocity and fouling resistance. **a)** wastewater flow rate, **b)** wastewater temperature, **c)** tube inner diameter, **d)** intermediary water mass flow rate.

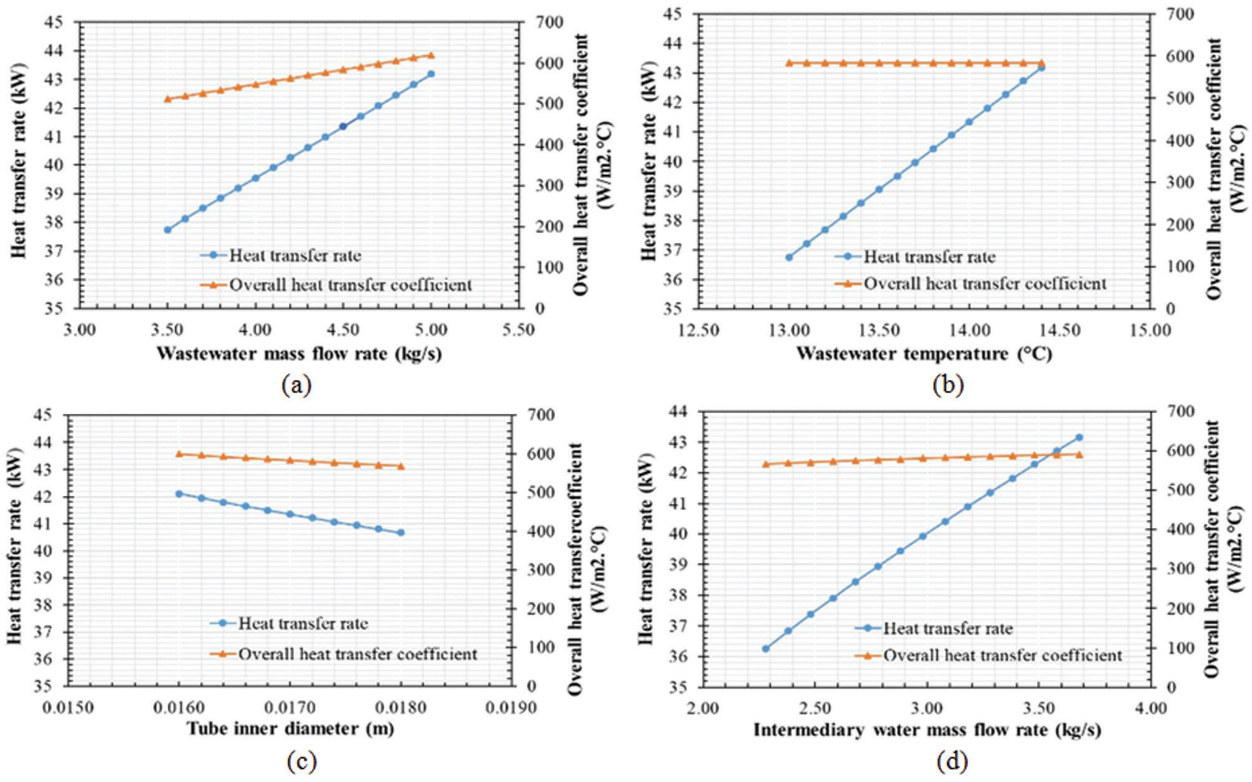
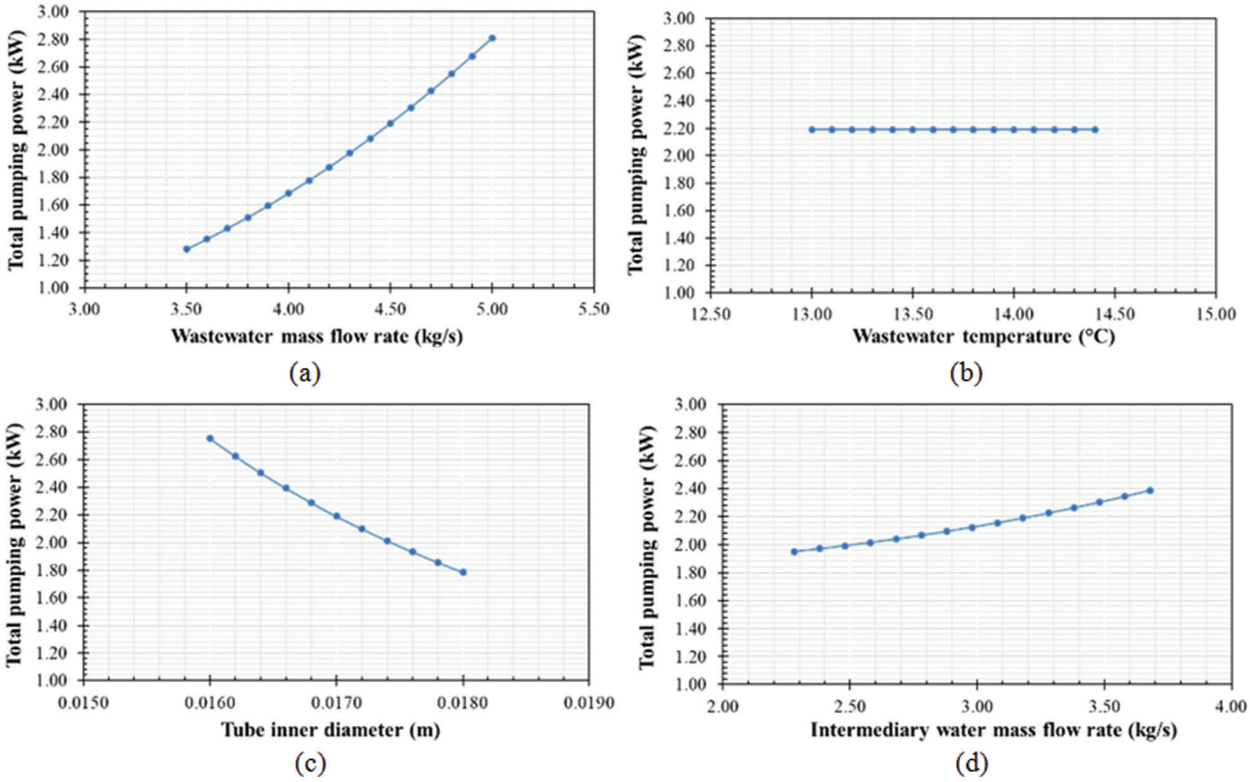


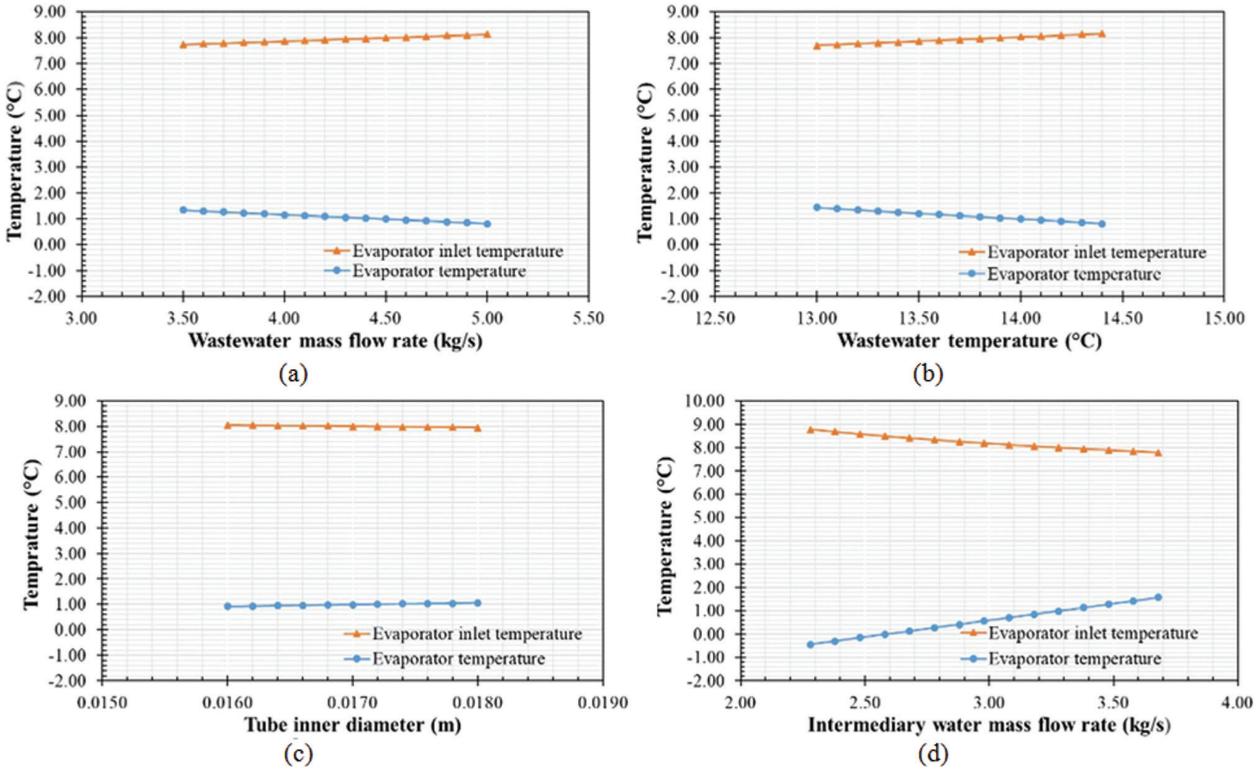
Figure 4. The effect of investigated parameters on the Q_{SWHE} in SWHE and U_{SWHE}. **a)** wastewater flow rate, **b)** wastewater temperature, **c)** tube inner diameter, **d)** intermediary water mass flow rate.



(a) (b)

(c) (d)

Figure 5. The effect of the investigated parameter on the $Wp_{\_total..}$ **a)** wastewater flow rate, **b)** wastewater temperature, **c)** tube inner diameter, **d)** intermediary water mass flow rate.



(a) (b)

(c) (d)

Figure 6. The effect of investigated parameters on the T_e and $T_{i\_e\_w}$ **a)** wastewater flow rate, **b)** wastewater temperature, **c)** tube inner diameter, **d)** intermediary water mass flow rate.

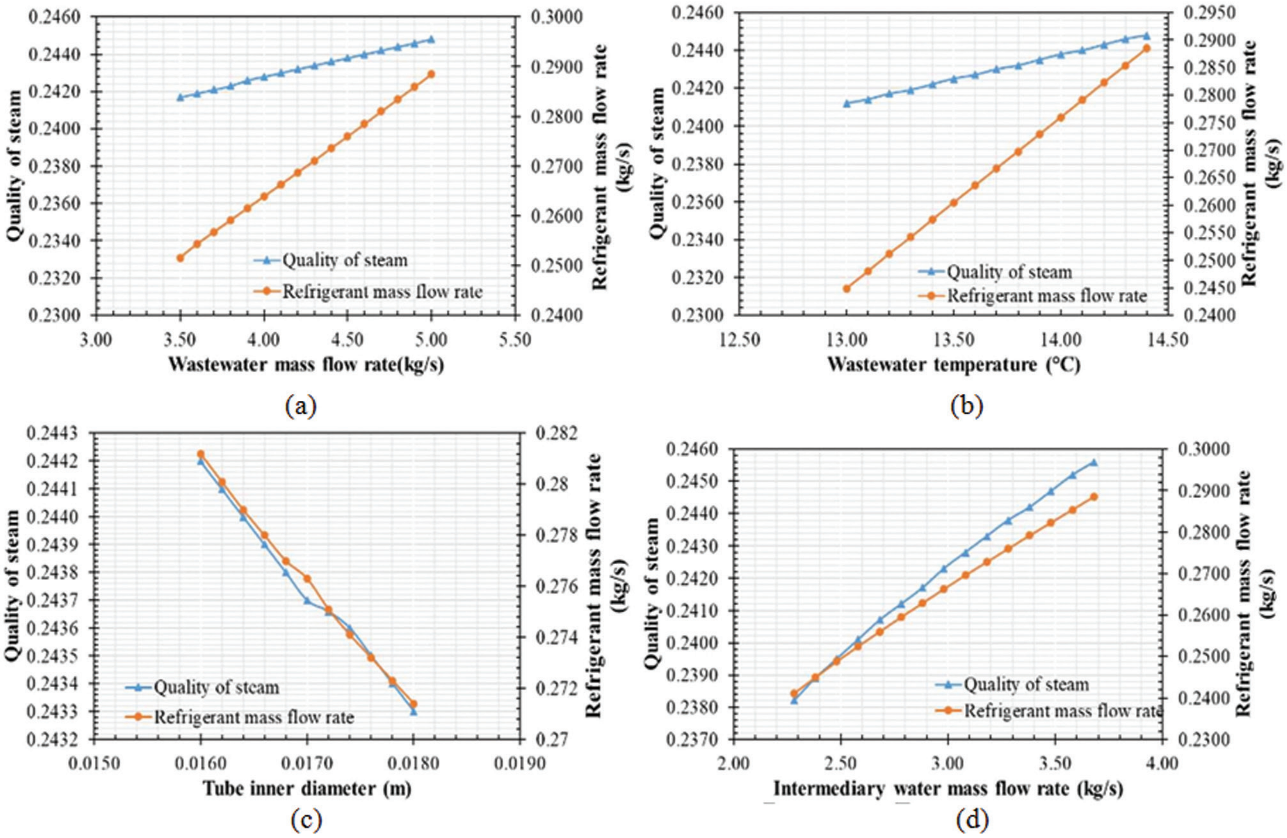


Figure 7. The effect of investigated parameters on x and m_r. **a)** wastewater flow rate, **b)** wastewater temperature, **c)** tube inner diameter, **d)** intermediary water mass flow rate.

of investigated parameters on T_e and $T_{w\_in\_e}$ are presented in Fig. 6. The $T_{w\_in\_e}$ increases from 7.74 °C to 8.13 °C with the increasing value of m_w. That change occurs due to the increasing Q_{SWHE} in SWHE which results in an increase in $T_{w\_in\_e}$ as the m_w is increased. It is also clear from Fig. 6 that the T_e decreases from 1.34 °C to 0.81°C with the increase in Q_{SWHE} in SWHE and the $T_{w\_in\_e}$. The observed effect of SW temperature on T_e and $T_{w\_in\_e}$ is almost the same as that of the m_w. An increases in d_i and m_w increase the T_e while causing a decrease in the $T_{w\_in\_e}$. However, the effect of d_i on the T_e and $T_{w\_in\_e}$ are very small.

The influences of investigated parameters on the x and m_r are shown in Fig. 7. An increase in m_{sw}, SW temperature, and m_w slightly increases the x and m_r, while an increase in the d_i causes a decrease in x and m_r due to the decrease in Q in SWHE, and hence, Q_e.

The effects of investigated parameters on the COP_{unit} and COP_{sys} are given in Fig. 8. It is clear that the COP_{unit} and COP_{sys} are around 3 and very close to each other at low m_{sw} values (m_{sw} = 3.5 kg/s). The COP_{unit} and COP_{sys} generally increase with the increasing value of m_{sw}. However, when the m_{sw} is increased, the COP_{sys} increases less than the COP_{unit} due to the increase in the ΔP_t. It is also obvious that the COP_{unit} and COP_{sys} at low SW temperatures are almost the same and increase as the SW temperature is increased. Also, the effect of SW temperature on the COP_{unit} and COP_{sys} is more remarkable than that of m_{sw}. On the other hand, an enhancement on COP_{unit} and COP_{sys} can be achieved when the m_w is increased. Besides, the increasing value of the d_i creates a deterioration on COP_{unit} and COP_{sys}.

Some design parameters with COP_{sys} results for indirect and direct type SWSHP are given in Table 3. From the table, SW flow rate can be a design parameter at both high and low flow rates depending on the quantity of heating or cooling load, however, SW temperature range generally is preferred from 11 to 15°C due to its characteristic property. It is obvious from Table 3 that the presented results for COP_{sys} are well-agreed with each other. On the other hand, the observed deviation between the COP results can be sourced from heat exchanger design parameters, heat pump operating conditions and used refrigerant, etc. It can be stated that the most important parameters affecting the performance of SWSHP systems are the fouling resistance and the SW temperature. As stated in Ref. [27], the thermal resistance increases significantly over time due to the fouling of the heat exchanger and thus, the efficiency

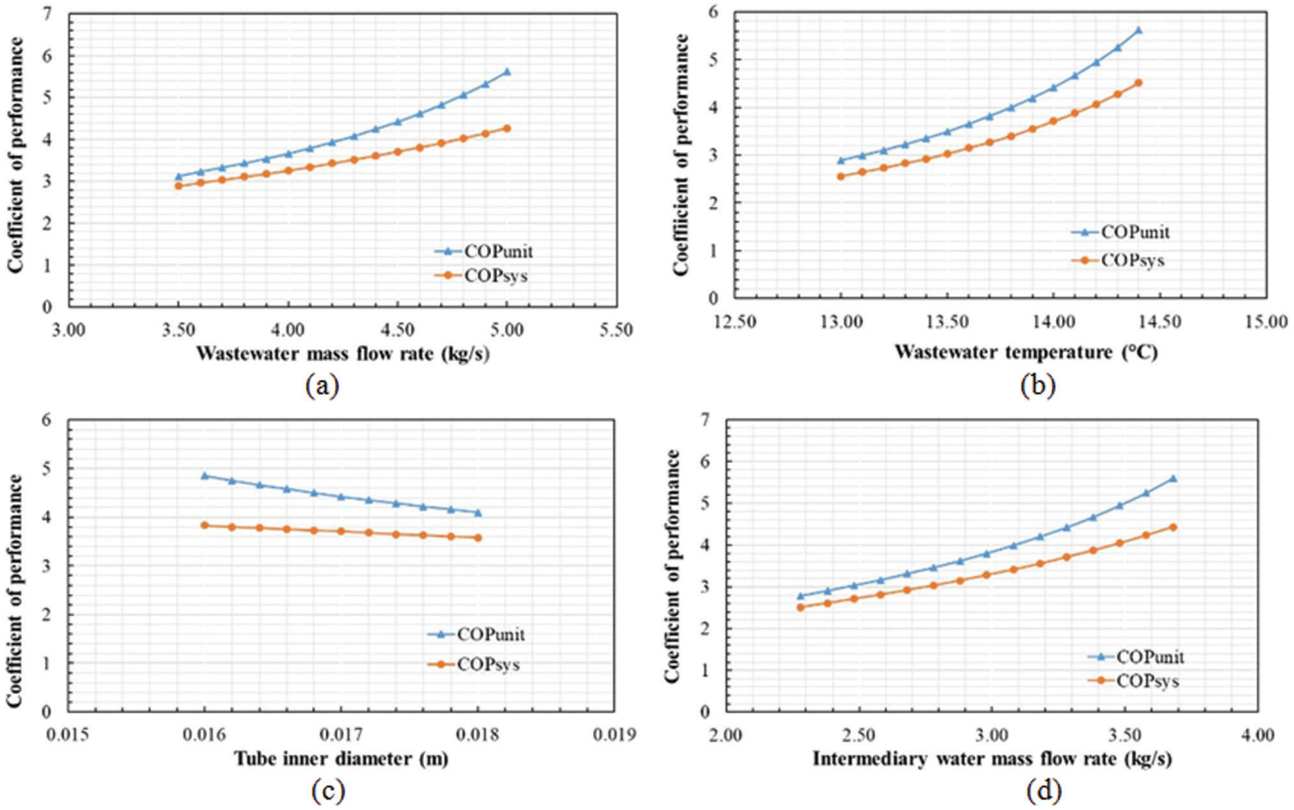


Figure 8. The effect of parameter changes on coefficients of performance. **a)** wastewater flow rate, **b)** wastewater temperature, **c)** tube inner diameter, **d)** intermediary water mass flow rate.

Table 3. A comparison between the COP_{sys} of the present study and literature

Reference	Type of SWSHP	Design Parameters		Average COP_{sys}
		SW mass flow rate (kg/s)	SW temperature (°C)	
Cho and Yun [13]	Indirect	~ 3.17	3.0-15.0	3.00
Liu et al. [26]	Indirect	~ 36.11	11.0	3.60
Qin and Hao [27]	Direct	0.63	12.0-13.0	3.75
Qian [15]	Direct	N/A	9.0-11.0	3.50
	Indirect	N/A	9.0-11.0	3.20
Present study	Indirect	4.50	14.00	3.80

of the heat exchanger is severely reduced. Therefore, a significant decrease in the COP_{sys} of the SWSHP system was observed. Also, it is emphasized in Ref. [13] that the COP_{sys} value varies between 1.9 and 3.2 with the variation of SW temperature. The present study showed that temperature variations have a greater impact on the system performance than variations in mass flow rate. As a result, heat exchanger design parameters have a significant influence on SWSHP system performance. Focusing on the usage of different types of heat exchangers in SWSHP systems and conducting extensive parametric investigations both experimentally and numerically will contribute to come up with more effective designs and hence, the development of SWSHP systems.

CONCLUSION

In this study, a numerical study was carried out on the indirect type SWSHP system designed for space heating. The paper researches how changes on SW temperature, mass flow rate, the inner diameter of the heat exchanger tube, and intermediary fluid mass flow rate affect the performance of the SWSHP system. The results show that the effect of SW temperature on COP_{sys} and COP_{unit} is more remarkable compared to the SW mass flow rate. Considering the specified parameter value ranges, the system performance coefficient increased from 2.56 to 4.51 for the changes on wastewater temperature while it increased from 2.89 to 4.27 for changes on wastewater flow rate. Also, an enhancement on COP_{unit} and COP_{sys} can be achieved when the intermediary mass flow rate is increased, and an increase in the tube's inner diameter unfavorably influences the COP_{sys} and COP_{unit}. In conclusion, SWE design influences the indirect SWSHP system performance and its operating condition, and consequently more theoretical and experimental studies are carried out on sewer heat exchanger designs to improve SWSHP system performance.

NOMENCLATURE

COP	Coefficient of performance
HP	Heat Pump
NTU	Number of transfer units
SW	Sewage wastewater
SWSHP	Sewage wastewater sourced heat pump
SWHE	Sewage Wastewater heat exchanger
A	area (m^2)
C, n	coefficients for heat exchanger design
c_p	specific heat, (kJ/kg.K)
C_r	heat capacity ratio
d	tube diameter, (m)
D_{shell}	shell diameter, (m)
e	distance between baffles, (m)
f	friction coefficient
h	enthalpy, (kJ/kg.K)
$J_{f,k}$	dimensionless pressure factor
k	thermal conductivity, (W/m.°C)
L	length, (m)
m	mass flow rate, (kg/s)
N	number
P	pressure, (bar, kPa)
ΔP	pressure drop
Pr	Prandtl number
R	thermal resistance, $(W/m)^{-1}$
Re	Reynolds number
s	tube pass number
S_t	tube pitch, (m)
T	temperature, (°C)
ΔT	temperature difference, (°C)
Q	heat transfer rate, (kW)
U	overall heat transfer coefficient, (W/m.°C)
x	quality of steam
V	velocity, (m/s)
W	work, power, (kW)

Greek Symbols

α	convective heat transfer coefficient,(W/m.°C)
μ	dynamic viscosity, (Pa.s)
η	efficiency
ρ	density, (kg/m)
ε	efficiency coefficient

Subscripts

a	air
b	baffle
c	condenser
ca	cross sectional area
comp	compressor
e	evaporator
ed	equivalent diameter
el	electrical
f	fouling
hta	heat transfer area
i	inner
in	inlet
is	isentropic
lm	logarithmic mean
m	mechanical
o	out
p	pump
r	refrigerant
sq	square
sw	sewage wastewater
SWHE	sewage wastewater heat exchanger
sys	system
t	tube
tr	triangle
w	water

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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