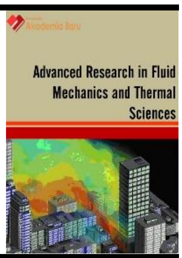
Penerbit Akademia Baru

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences

Journal homepage: www.akademiabaru.com/arfmts.html
ISSN: 2289-7879



Energetic Optimization of Solar Water Heating System with Flat Plate Collector using Search Group Algorithm

Open Access

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ARTICLE INFO

Article history:
Received date 8 April 2019
Received in revised form date 13 June 2019
Accepted date 2 July 2019
Available online date 30 September 2019

Keywords:
Solar water heating system; flat plate solar collector; energetic optimization; search group algorithm

ABSTRACT

Solar water heating system has a vital potential for low and medium thermal applications. This study proposes a metaheuristic optimization technique, namely Search Group Algorithm (SGA), for energetic optimization of solar water heating system using flat plate collector (SWH-FPC). For this purpose, the following parameters are considered as design variables: mass flow rate, fluid inlet temperature, absorber plate thickness, riser tube outer diameter, tube spacing, and insulation thickness. In this study, SGA is applied to find optimal values of such parameters for maximum energy efficiency of flat plate solar collector. Moreover, the impact of each design variable on energy efficiency is also analyzed. The simulation results show that energy efficiency is improved by 4.904 % compared to the base case, which emphasizes the effectiveness and robustness of SGA to achieve high performance of solar thermal collector.

1. Introduction

Global energy demand is rapidly progressing due to global population growth, technological development, commercial and industrial activities [1, 2]. Energy is an extremely important factor for economic operation of the country [3]. Fossil fuels including oil, coal and natural gas currently account for a large portion of total energy consumption [4, 5]. The excessive usage of conventional fossil fuels and nuclear power would cause detrimental effects on the natural environment like emitting harmful emissions that exacerbate to global warming. These emissions also aggravate pollution which is the contributing element for smog, ozone depletion, acid rain, nuclear waste, and other environmental contaminants [6, 7]. Therefore, reducing energy consumption and using renewable energy have always been a concern of worldwide governments.

Renewable energy which has the advantages of being sustainable and eco-friendly provides a perfect solution for limiting fossil fuel consumption and environmental issues from combustion

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processes for energy generation [8]. Solar energy is endless and a reliable source of energy that delivers sunlight freely for everyone [9]. Solar thermal system is one of the potential applications of renewable energy utilization. Of all the applications of solar thermal system, solar water heating system (SWH) is an extremely efficient technology to convert solar energy into thermal energy, which currently contributes an essential role significantly for the domestic and industrial area. Solar thermal collector is a critical component of SWH in which flat plate solar collector (FPC) is popularly used for low and medium thermal applications. One of the most significant barriers and obstacles to the development of SWH is the low energy efficiency of FPC. Thus, the enhancement in the performance of FPC has considerably drawn the attention of researchers in this field.

In the literature, many meta-heuristic algorithms have been developed and applied for assessing optimal thermal performance of FPC and gained impressive results. Siddhartha [10] indicated that the efficiency of smooth flat plate solar air heater (SFPSAH) improves by raising the heat transfer rate, number of glass cover. The performance came out at 72.42% by using Particle Swarm Optimization (PSO). Hajabdollahi [11] analyzed a multi-objective PSO technique for optimizing cost and efficiency of SWH-FPC. The outcomes indicated that better values of thermal-economic were achieved at a lower rate of heat transfer. Bornatico [12] used PSO to define the optimum value of the key components of the solar thermal building installation. In [13], Genetic Algorithm (GA) was implemented for estimating the yield and the costs of the SWH with the lowest payback time. The GA method in [14] and a Simulated Annealing algorithm (SA) in [15] is applied to predict the minimum efficiency of SFPSAH. In general, by increasing Reynolds number, tilt angle, and number of covers, the efficiency was enhanced for all the cases considered. Sahin [16] applied Artificial Bee Colony Algorithm (ABC) and GA to investigate the correlation between different parameters of SFPSAH. The outcomes displayed that efficiency of solar air collector obtained by ABC is slightly higher than the GA method. The optimal value of thermal efficiency reached about 74.98%. Yildirim [17] analyzed the optimal thermohydraulic condition of the single pass solar air heaters examining the channel depth and air flow rate. For further analysis, numerical simulation has been carried out for an in-depth analysis of the impact of design parameters on the performances of FPC in [18]. Sopian [19] and Rosli [20] investigated recent applications and advancement in high efficiency and cost-effective photovoltaic thermal (PV/T) collectors. Taloub [21] studied the impact of albedo and wind velocity on the solar collector operating in forced circulation. The results found that solar radiation obtains optimum values in the desert regions. In [22], two computer programs were carried out using Engineering Equation Solver (ESS) to determine the monthly and seasonal optimum tilt angle of solar collector for Nigeria. Kinan [23] studied to determine the performance of a circular and square collector for the Solar updraft tower power plant. Kalogirou [24] investigated a technique to obtain the maximum the storage tank mass and area of collector for maximizing the lifetime circle saving of solar energy system by developing a combination of GA and Artificial Neural Networks (ANN). Kalogirou [25] proposed ANN to predict parameters of FPC, and obtained results were compared to experimental data. Krause [26] improved the performance of solar heating systems by optimizing the operating parameters. The results revealed that solar heat cost decreased by 18% compared to the conventional system. Several studies adopted hybrid optimization techniques to solve the problem of FPC, namely a combination of PSO and Hooke-Jeeves in [27], a combination of GA and binary search method [28]. However, there is always requires a new algorithm which enables for optimization of solar thermal collector effectively.

Search Group Algorithm (SGA) is one of the most recent metaheuristic algorithms, which was developed by Gonçalves *et al.,* [29]. The primary purpose of SGA is to generate an appropriate balance between exploration and exploitation of the design domain to obtain feasible solutions. For this reason, SGA has been achieved promising results for different engineering problems, namely truss structures optimization [29, 30], optimization of planar steel frames [31], networked control systems [32], automatic generation control [33, 34], power systems [35, 36].

Most related research has only determined optimal design parameters (i.e. depth or length) and optimal operating parameters (i.e. mass flow rate or fluid inlet temperature) and has not concerned about the geometric parameters of FPC including absorber plate thickness, tube diameter, tube spacing, and insulation thickness. This research presents the first attempt of the SGA technique for energetic optimization of SWH-FPC considering six design variables, comprising mass flow rate, fluid inlet temperature, absorber plate thickness, tube diameter, tube spacing, and insulation thickness. The objective of this work is to find optimal values of design variables for maximum energy efficiency of FPC. Furthermore, the trend and impact of each design variables on the variation of energy efficiency are also implemented. The obtained outcomes emphasize the effectiveness and robustness of SGA for energetic optimization of solar thermal collector.

Section 2 describes the mathematical formulation of FPC. Section 3 provides an overview of the SGA algorithm and application of SGA for optimizing the efficiency of FPC. Section 4 discusses the trend of change in efficiency with the effect of design variables, and the results after optimization using SGA. Finally, Section 5 presents the main conclusions of this research.

2. Problem Formulation

The energy efficiency of FPC can be evaluated and predicted by applying the empirical correlations for heat transfer coefficients and other parameters, namely the heat removal factor, total heat loss coefficient. This section presents a thermal analysis of FPC for water heating system under steady state condition.

2.1 Thermal Modelling of Flat Plate Solar Collector
2.1.1 Collector energy balance

Solar radiation absorbed by FPC is estimated as follows [37]

$$S = (\tau\alpha)I_T \tag{1}$$

where I_T is the total solar radiation intensity and $(\tau\alpha)$ is the effective transmittance-absorptance.

The thermal energy balance of FPC can be expressed as follows

$$Q_u = Q_{ab} - Q_{loss} \tag{2}$$

where Q_u is the useful energy, Q_{ab} is the absorbed energy and Q_{loss} is the total heat loss.

Absorbed energy represents solar radiation energy which passes the glass cover and is absorbed by the collector. Hence, it can be stated as

$$Q_{ab} = A_p S = A_p(\tau\alpha)I_T \tag{3}$$

where A_p is the area of the absorber plate.

2.1.2 Collector energy loss

In a simplified way, total heat loss is the lost energy from the collector to the surrounding can be computed by [38]

$$Q_{loss} = U_L A_c \left(T_{pm} - T_a\right) \tag{4}$$

where U_L is the overall heat loss coefficient, T_{pm} is the mean temperature of absorber plate, and T_a is the ambient temperature.

The overall heat loss coefficient is composed of the top, edge and back loss coefficient

$$U_L = U_t + U_e + U_b \tag{5}$$

The top loss coefficient is computed by following the empirical formulas of Klein [39] as follows

$$U_t = \left[\frac{N}{\frac{C}{T_{pm}}\left[\frac{(T_{pm}-T_a)}{(N+f)}\right]^e} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(T_{pm}+T_a)(T_{pm}^2+T_a^2)}{(\varepsilon_p+0.00591Nh_w)^{-1}+\frac{2N+f+1+0.133\varepsilon_p}{\varepsilon_g}-N} \tag{6}$$

In Eq. (6), f, C, e and h_w are presented by the following equations

$$f = \left(1 + 0.089h_w - 0.1166h_w\varepsilon_g\right)(1 + 0.07866N) \tag{7}$$

$$C = 520(1 - 0.000051\beta^2), \begin{cases} 0 < \beta < 70^{\circ} \\ \beta = 70^{\circ}\ if\ \beta > 70^{\circ} \end{cases} \tag{8}$$

$$e = 0.430\left(1 - \frac{100}{T_p}\right) \tag{9}$$

$$h_w = 5.7 + 3.8v \tag{10}$$

where σ is the Stefan-Boltzmann constant; N is the number of glass cover; β is the collector tilt; ε_g is the emissivity of glass cover; ε_p is the emissivity of absorber plate; v is wind speed of ambient air; h_w is the heat transfer coefficient of wind.

The edge and back loss coefficients are calculated as

$$U_e = \frac{k_e}{\delta_e} \times \frac{A_e}{A_c} \tag{11}$$

$$U_b = \frac{k_b}{\delta_b} \tag{12}$$

where A_e is the area of edge heat transfer surface; k_e and δ_e are the thermal conductivity and thickness of edge insulation, respectively; k_b and δ_b are the thermal conductivity and thickness of back insulation, respectively.

The mean temperature (T_{pm}) is computed by assuming an initial value to estimate U_L and Q_u. The next value of T_{pm} is calculated according to the below equation, and the initial value is modified through each iteration [40]

$$T_{pm} = T_i + \frac{Q_u}{A_p F_R U_L}(1 - F_R) \tag{13}$$

where the heat removal factor (F_R) can be expressed as

$$F_R = \frac{\dot{m}C_p}{A_p U_L}\left[1 - exp\left(-\frac{F' U_L A_p}{\dot{m}C_p}\right)\right] \tag{14}$$

in which $\dot{m}$ is the mass flow rate, C_p is the special heat capacity, and T_o is fluid outlet temperature.

The collector efficiency factor (F') is

$$F' = \frac{\frac{1}{U_L}}{W\left(\frac{1}{U_L[D_o + (W - D_o)F]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}}\right)} \tag{15}$$

where D_o and D_i are the outer and inner diameter of riser tube, respectively; W is the tube spacing; C_b is the thermal conductance of bond; h_{fi} is the convection heat transfer coefficient between fluid and tube wall.

F is the standard fin efficiency is given by

$$F = \frac{tanh[m(W - D_o)/2]}{m(W - D_o)/2} \tag{16}$$

$$m = \sqrt{\frac{U_L}{k\delta}} \tag{17}$$

where k and δ are the thermal conductivity and thickness of absorber plate, respectively.

2.1.3 Useful heat output and energy efficiency

In steady-state conditions, the useful energy gain can be obtained from Eq. (2)-(4)

$$Q_u = A_p[S - U_L(T_{pm} - T_a)] \tag{18}$$

Furthermore, useful heat energy output can be rewritten as follows

$$Q_u = A_p F_R[S - U_L(T_i - T_a)] \tag{19}$$

Finally, the energy efficiency of FPC is determined by

$$\eta = \frac{Q_u}{A_c I_T} \tag{20}$$

2.2 Process of Thermal Modelling

An iterative process is implemented to compute the energy efficiency of FPC as follows

i. At the start of the iteration process, the mean temperature of absorber plate (T_{pm}) is assumed from the inlet temperature of fluid as: $T_{pm} = T_i + 10$.
ii. Top loss coefficient (U_t), edge loss coefficient (U_e), back loss coefficient (U_b), and as a result overall loss coefficient (U_L) are calculated according to Eq. (5)-(12).
iii. By applying the obtained overall heat loss coefficient, heat removal factor (F_R) and useful energy output (Q_u) are computed by using Eq. (14)-(18).
iv. Then the new mean temperature of absorber plate is adjusted using Eq. (13).
v. This new value of T_{pm} is compared to the previous value. If the difference is within the acceptable boundary, the procedure is stopped and move to Step 6; if the difference exceeds the specified limit, the new value of T_{pm} is adopted as replaced value and back to Step 2.
vi. When the correct value of T_{pm} is obtained, energy efficiency of FPC is calculated by Eq. (20).

2.3 Objective Function of the Energetic Optimization

In this study, the energy efficiency of FPC in Eq. (20) is selected as the objective function of energetic optimization and to be maximized. Hence, the optimization problem is expressed as follows

Find

$$x^* = [\dot{m}, T_i, \delta, D_o, W, \delta_b] \tag{21}$$

Maximize

$$\eta(x) = \frac{\dot{Q}_u}{A_c I_T} \tag{22}$$

Subject to

$$0.01 \leq \dot{m} \leq 0.1 \tag{23}$$

$$20 \leq T_i \leq 40 \tag{24}$$

$$0.0002 \leq \delta \leq 0.001 \tag{25}$$

$$0.008 \leq D_o \leq 0.02 \tag{26}$$

$$0.06 \leq W \leq 0.2 \tag{27}$$

$$0.01 \leq \delta_b \leq 0.1 \tag{28}$$

where mass flow rate ($\dot{m}$), fluid inlet temperature (T_i), plate thickness (δ), outer diameter of riser tube (D_o), tube spacing (W), and insulation thickness (δ_b) are selected as design variables as well as the control variables in the optimization procedure.

3. Search Group Algorithm

SGA is a population-oriented algorithm. This algorithm is classified into two stages (global stage and local stage), both stages comprising generation, mutation, and selection procedures [30]. To attain solutions from the optimization problem in Section 2.3, SGA created search groups that explore the promising regions in global search and exploit the best design of these promising domains in local search [31]. The principal goal of this algorithm is to balance between the exploration stages and exploitation stages [29]. The procedures of SGA are described as below.

3.1 Population Initialization

In the initialization procedure, SGA generates a random initial population **P** on the search space

$$P_{ij} = x_j^{min} + \left(x_j^{max} - x_j^{min}\right)U[0,1] \tag{29}$$

for $j = 1, \ldots, n,\ \ i = 1, \ldots, n_{pop}$,

where P_{ij} is the jth design variable of the ith entity of **P**; $U[0,1]$ is a stochastic variable between a range $[0,1]$; x_j^{max} and x_j^{min} are the upper and lower limit of the jth design variable, respectively; n is the sum of design variables; and n_{pop} is the sum of **P**.

3.2 Selection of the First Search Group

After the generation of the population **P**, SGA calculated the objective function of the entire entity. A benchmark tournament selection is performed to create a search group **R** by choosing n_g entities from **P**.

3.3 Mutation of the Search Group

In this step, n_{mut} entities of the search group **R** are replaced by new entities to enhance the global search ability by Eq. (30)

$$x_j^{mut} = E\left[R_j\right] + t\varepsilon\sigma\left[R_j\right] \text{ for } j = 1, \ldots, n \tag{30}$$

where x_j^{mut} is the jth design variable of a mutated entity; E is the mean value; σ is the standard deviation operators; ε is the convenient stochastic variable; t is the value that manages how far the new entity is formed; and $R_{:,j}$ is the jth column of the search group matrix.

The possibility of the individual can be displaced depending on its rank of the present search group. An inverse tournament selection is employed to implement this procedure.

3.4 Formation of the Families

'Each member of the search group is considered as a family leader. The family is a set of a family leader and the entities that it generated. Therefore, each one of the family leaders creates a family as follows

$$x_j^{new} = R_{ij} + \alpha\varepsilon \text{ for } j = 1, \ldots, n, \tag{31}$$

in which α adjusts the size of the perturbation. α is reduced as follows

$$\alpha^{k+1} = b\alpha^k \tag{32}$$

3.5 Selection of the New Search Group

The optimization process consists of two stages: the global stage and local stage. In the global stage, the best member of each family is selected to create a new search group to explore most of the search domain. In the local stage, best n_g entities of all the families are selected to create the new search group to exploit the domain of the current best design.

3.6 Overall Procedure of SGA for Energetic Optimization of SWH – FPC

The objective function to be maximized using SGA for energetic optimization of FPC is defined in Eq. (21). Each entity of the population **P** characterizes a set of control variables. These design variables are mass flow rate ($\dot{m}$), fluid inlet temperature (T_i), plate thickness (δ), outer diameter of riser tube (D_o), tube spacing (W), and insulation thickness (δ_b), which is described as follows

$$x_i = [\dot{m}, T_i, \delta, D_o, W, \delta_b] \text{ for } i = 1, \dots, n_{pop}, \tag{33}$$

The overall procedures of the implementation of SGA for energetic optimization of FPC are described as follows

i. Set the parameters of SGA: $k = 0, it^{max}, it^{max}_{global}, \alpha^k, \alpha_{min}, b, n_{pop}, n_g, n_{mut}, h, t, \varepsilon, \upsilon$.
ii. Determine design variables within their lower and upper boundary.
iii. Randomly initialize the population P using Eq. (29).
iv. Initialize the first search group $\boldsymbol{R}^k$ choosing n_g entities from P adopting a benchmark tournament selection.
v. Substitute n_{mut} entities by new candidates generated as defined in Eq. (30).
vi. Create the families $\boldsymbol{F}_i$ following Eq. (31)
vii. Select the new search group as below
If $k < it^{max}_{global}$: search group $\boldsymbol{R}^{k+1}$ is created by the best member of each family.
Else: search group $\boldsymbol{R}^{k+1}$ is created by the best n_g entities of P.
viii. Update α^{k+1} using Eq. (32).
ix. Set $k = k + 1$, if $k = k + 1$, move to Step 10, else back to Step 5.
x. Solution: $\boldsymbol{x}^* = \boldsymbol{R}_1$.

4. Numerical Results
4.1 Validation of the Thermal Modelling

In this research work, the simulation of SWH-FPC is implemented in MATLAB platform. The specification of FPC is referenced from the technical details of TitanPower-ALDH29-V3 premium flat plate solar collector [41]. Aluminum sheet with the emissivity of 0.05 and the thermal conductivity of 240 (W/m K) is used as the absorber plate. The characteristics and test conditions of FPC are summarized in Table 1.

The simulation results of this research are verified by comparing with experimental results of [41], which is presented in Table 2 for the same input values stated in Table 1. As shown in Table 2, the difference in percentage values for the results of the simulation and experimental are acceptable.

Table 1
Specifications and test conditions of flat plate solar collector

Parameter	Value
Gross dimensions	1.158 m × 2.368 m × 95 mm
Emissity of glass cover (ε_g)	0.9
Absorber dimensions	1.11 m × 2.32 m × 0.5 mm
Emissivity of absorber plate (ε_p)	0.05
Outer diameter of tube (D_o)	9.0 mm
Inner diameter of tube (D_i)	8.6 mm
Tube Spacing (W)	105 mm
Back insulation thickness (δ_b)	50 mm
Edge insulation thickness (δ_e)	25 mm
Mass flow rate ($\dot{m}$)	0.04 kg/s
Total solar radiation intensity (I_T)	1000 W/m2
Effective transmittance-absorptance ($\tau\alpha$)	0.874
Slope of collector (β)	45°

Table 2
Comparison of experimental results of Ref. [41] and simulation results

Output Parameter	Ref. [41]	Base case of this work	Difference
Useful heat gain (W)	2019	2115	4.75 %
Collector efficiency (%)	74.7	77.16	3.29 %

This comparison can be visually verified in Figure 1, which shows that the efficiency curve with the temperature difference between the fluid inlet temperature (T_i) and ambient temperature (T_a) under various solar radiation intensity. Similar trends are observed in the efficiency curve for both Ref and this work. Therefore, thermal modeling of this work is suitable for the simulation of FPC.

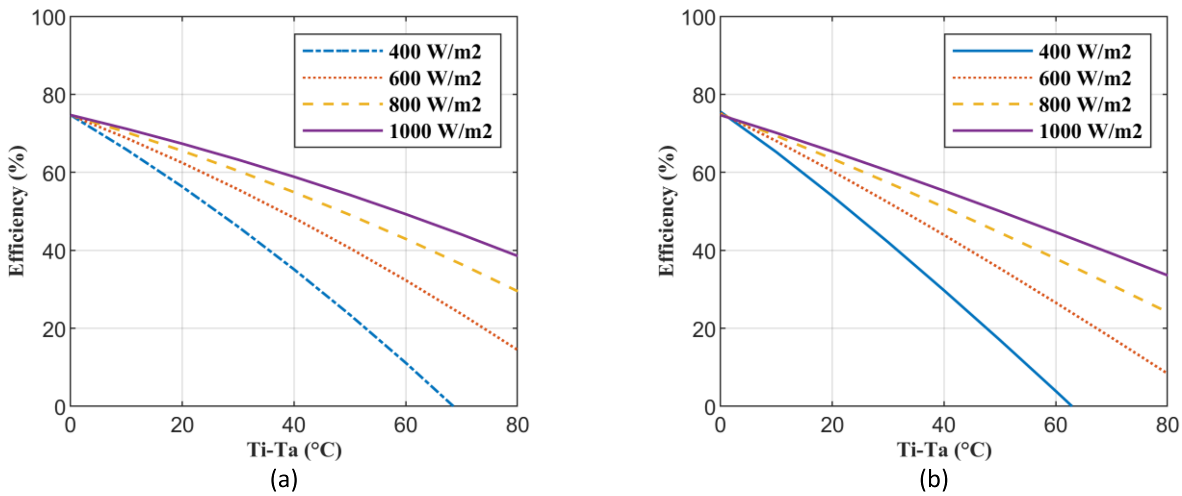

Fig. 1. Efficiency curve with temperature difference under various solar radiation intensity (a) Experimental results of [41]; (b) Simulation Result of this work

4.2 Effect of Design Variables on Energy Efficiency

In this section, numerical studies to determine the effect and sensitivity of energy efficiency to variation in the design parameters of SWH-FPC including mass flow rate, fluid inlet temperature, plate thickness, riser tube outer diameter, tube spacing, and insulation thickness were analyzed as follows

4.2.1 Mass flow rate

Figure 2 presents the trend of energy efficiency versus mass flow rate as can be observed that the mass flow rate rises, the efficiency is enhanced. Factually, by increasing this parameter, heat removal factor and the useful heat gain rise; therefore, the efficiency also increases.

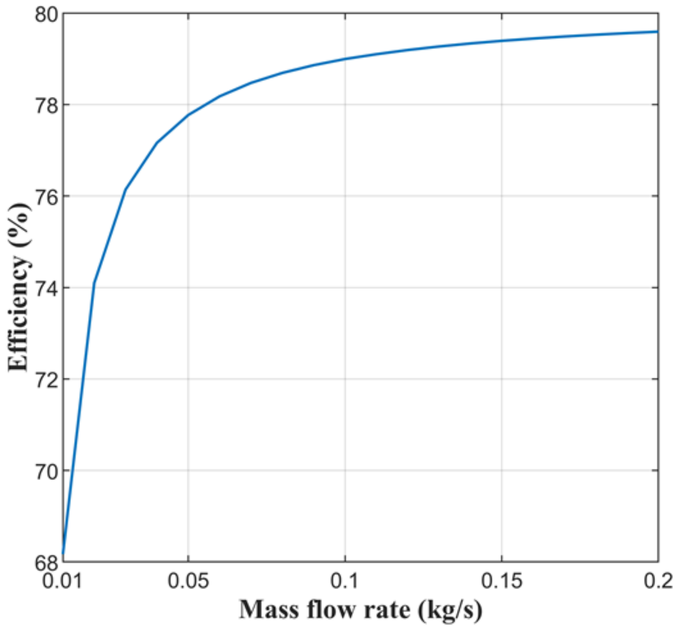

Fig. 2. Variation of energy efficiency versus mas flow rate

In general, when mass flow rate varies from 0.01 kg/s to 0.06 kg/s resulting in a significant climb in energy efficiency from 68.17 % to 77.77 % (a value of 9.6 %). Additionally, the efficiency shows a slight upward trend from 77.77 % to 79.59 % (a value of 1.82 %) as increasing the mass flow rate from 0.06 kg/s to 0.2 kg/s. This reveals that the impact of mass flow rate on energy efficiency is more obviously at the low level of mass flow rate. Maximum efficiency of 79.59 % achieved at the highest mass flow rate (0.2 kg/s) and the minimum efficiency of 68.17 % achieved at the lowest mass flow rate (0.01 kg/s).

4.2.2 Fluid inlet temperature

As it is shown in Figure 3, fluid inlet temperature increases result in a reduction in energy efficiency. This is mainly due to the difference between the fluid inlet temperature and the ambient temperature $(T_i - T_a)$ which increases overall heat losses. When the ambient temperature is constant, the inlet temperature increases from 20°C to 40°C shows a clear downward trend in efficiency from 79.23 % to 70.31 %.

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences
Volume 61, Issue 2 (2019) 306-322

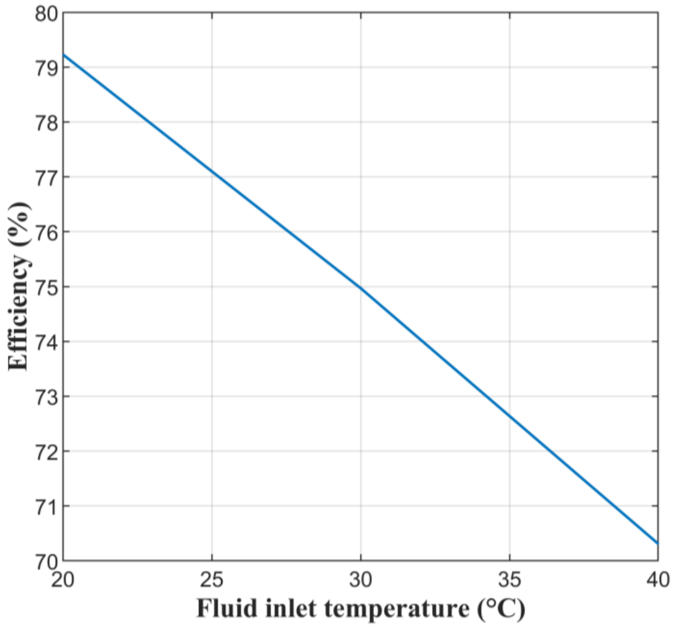

Fig. 3. Variation of energy efficiency versus fluid inlet temperature

4.2.3 Absorber plate thickness

Energy efficiency increases by increasing absorber plate thickness as shown in Figure 4. This is due to increasing plate thickness leads to an increase in fin efficiency, collector efficiency factor, and overall efficiency.

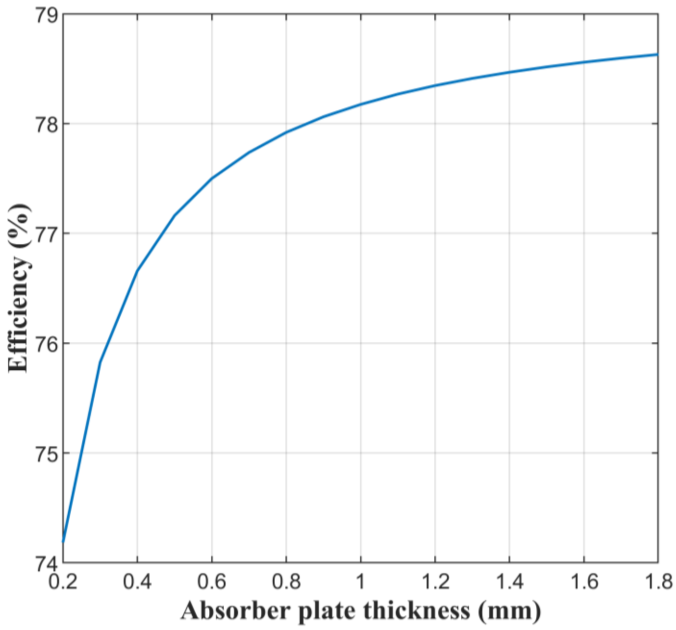

Fig. 4. Variation of energy efficiency versus absorber plate thickness

Energy efficiency increases from 74.18 % to 77.92 % (a value of 3.74 %) as increasing the thickness of absorber plate from 0.2 mm to 0.8 mm. Additionally, energy efficiency is enhanced from 77.92 % to 78.63 % (a value of 0.71 %) when the plate thickness value varies from 0.8 mm to 1.8 mm. This indicates that when the absorber plate is comparatively slim (δ < 0.8 mm), energy efficiency can be dramatically improved through increasing the absorber plate thickness.

4.2.4 Riser tube diameter

As shown in Figure 5, energy efficiency increases by rising tube diameter, but the change in efficiency is insignificant. In fact, by raising this parameter, the heat transfer coefficient decreases result in useful heat and efficiency increase. By varying tube diameter from 0.008 to 0.02 m, energy efficiency tends to increase slightly by 0.24 %.

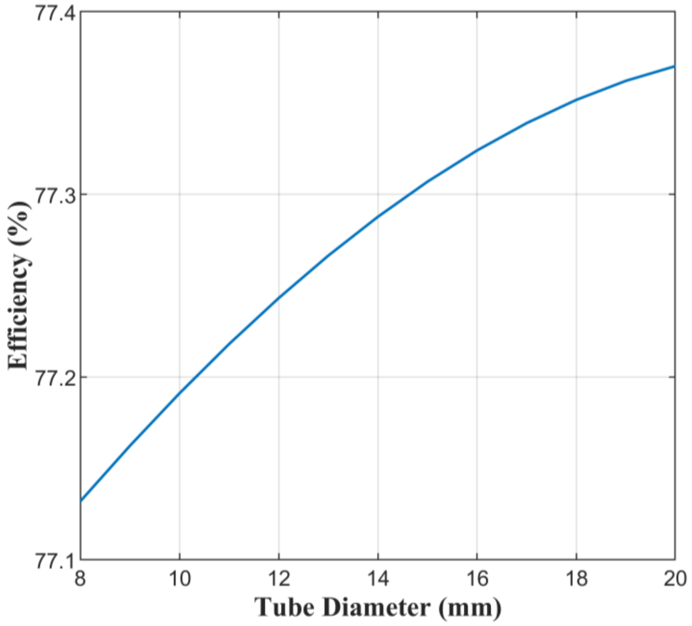

Fig. 5. Variation of energy efficiency versus tube diameter

4.2.5 Tube spacing

By increasing tube spacing, energy efficiency decreases as shown in Figure 6. Factually, by increasing tube spacing, collector fin efficiency, heat removal factor, and consequently useful heat decrease, resulting in reduced efficiency. When tube spacing varies from 60 mm to 200 mm, efficiency is reduced from 78.86 % to 70.75 % (a value of 8.11 %); therefore, varying the tube spacing has a dramatic effect on energy efficiency.

4.2.6 Insulation thickness

Figure 7 shows that energy efficiency increases by increasing the thickness of back insulation. This is mainly due to the rise in insulation thickness which can enhance thermal conduction resistance of insulation. Hence, the heat loss to the back decreases and energy efficiency simultaneously increases.

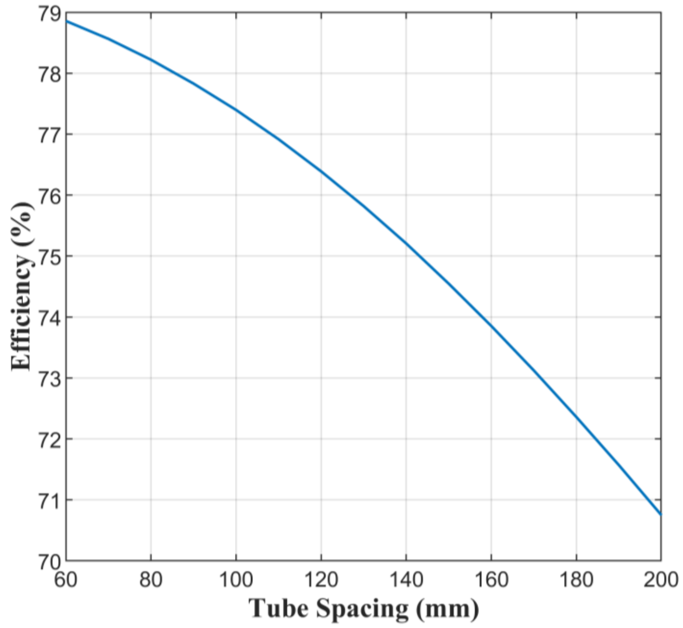

Fig. 6. Variation of energy efficiency versus tube spacing

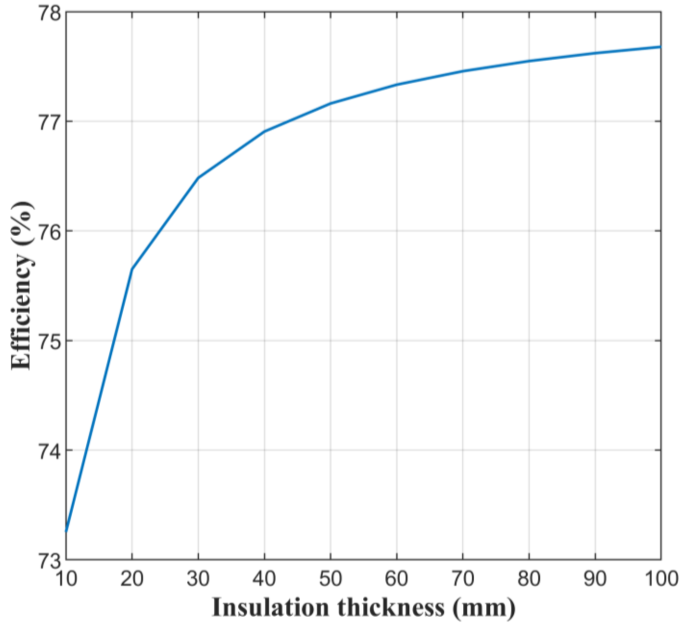

Fig. 7. Variation of energy efficiency versus insulation thickness

When the thickness of insulation increases from 20 mm to 40 mm, energy efficiency is improved by 3.66 %. By changing the thickness of insulation from 40 mm to 100 mm, energy efficiency rises by 0.77 %. When the thickness of insulation is comparatively thin (δ_b < 40 mm), increase insulation thickness has a significant effect on energy efficiency. When the insulation thickness is approximately thick (δ_b > 40 mm), the change in insulation thickness has an insignificant effect on efficiency.

4.3 Results of the Optimization Procedure

For the detailed input data in Table 1, the optimization procedure using the SGA method leads to the optimal value of the objective function is implemented. The obtained results include optimal design variables, maximum efficiency, overall heat loss coefficient, and useful heat gain after optimization are recorded in Table 3 in which the maximum efficiency is (82.064 %). Comparing the optimized efficiency with the base case from Table 1 shows that it is dramatically improved from 77.16 % to 82.064 %, i.e., by 4.904 %.

Table 3
Optimal solution of design variables and output parameters

Parameter	Value
Mass flow rate ($\dot{m}$)	0.1958 kg/s
Fluid inlet temperature (T_i)	22.4079°C
Absorber thickness (δ)	0.0006 m
Tube diameter (D_o)	0.0130 m
Tube spacing (W)	0.0604 m
Insulation thickness (δ_b)	0.0825 m
Overall heat loss coefficient (U_L)	1.7635
Heat removal factor (F_R)	0.9946
Useful heat gain (Q_u)	2250.3 W/m2
Energy efficiency (η)	82.064 %

Moreover, the evolution of the objective function over iterations is described in Figure 8. Energy efficiency shows a tendency toward convergence when the number of iterations is 20. In other words, the SGA finds a feasible solution for this system after 20 iterations.

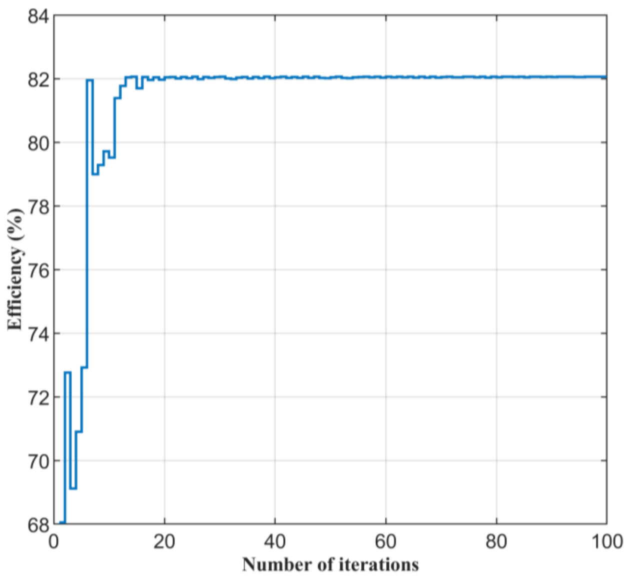

Fig. 8. Convergence characteristic for energetic optimization of FPC

In order to assess the robustness of the proposed algorithm, SGA was run independently 50 trials, and the outcomes are tabulated in Table 4. From this table, the average values are approximate to the best ones. Furthermore, the history of 50 runs from the proposed SGA technique for this system is given in Figure 9. Therefore, the SGA is very robust in solving this problem.

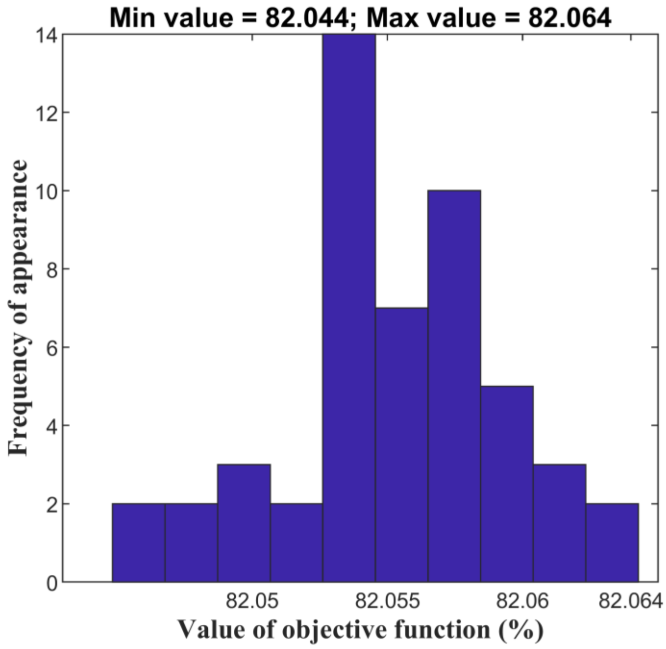

Fig. 9. History of 50 runs by SGA for energetic optimization of FPC

Table 4
Results of 50 runs for energetic optimization of FPC

Min	Average	Max	Standard deviation	Computational time (s)
82.044	82.055	82.064	0.004	0.31

4. Conclusions

The principal conclusion derived from this paper is that SGA is an efficient and robust algorithm for obtaining the optimal value of design variables at which the energy efficiency of FPC is maximum. It has good convergence characteristics and can achieve a feasible solution with fast convergence speed. Moreover, the effect of design variables on the energy efficiency of FPC is also carried out. The algorithms proposed here can assist the manufacturer and engineers in improving the energy efficiency of SWH-FPC in solar thermal system.

Acknowledgement
This research was sponsored by YUTP – FRG and Graduate Research Assistantship (GRA) Scheme of Universiti Teknologi PETRONAS.

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