



# Mathematical modelling and simulation of multiphase flow in a flat plate solar energy collector



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## ARTICLE INFO

### Article history:

Received 26 May 2015

Accepted 10 September 2015

Available online 26 September 2015

### Keywords:

Flat plate collector

Simulation

Multiphase flow

Heat transfer coefficient

## ABSTRACT

Non-conventional collectors where organic fluid or refrigerant experience a phase change have many advantages over conventional collectors which have either air or relatively high temperature boiling liquid. Increase in heat transfer coefficient and system efficiency, corrosion prevention and freeze protection are the main benefits of the first type. In this study, a detailed numerical model of a flat plate collector is developed to investigate the fluid mean temperature, useful heat gain and heat transfer coefficient along the collector tube. The refrigerant HFC-134a was used in the simulation as the working fluid of the collector. The model can both predict the location where the fluid undergoes a phase change in the tube and the state at the exit under given inlet conditions. The effect of boiling on the heat transfer coefficient of the fluid is also investigated. Simulations were performed at three different mass flow rates (0.001, 0.005 and 0.01 kg/s) and three different operating pressures (4, 6 and 8 bar) to be able to see the effect of mass flow rate and pressure on plate temperature, heat loss coefficient, efficiency of the collector and the heat transfer coefficient of the fluid. The simulation results indicate that the heat transfer coefficient of the fluid increases from 153.54 W/m<sup>2</sup> K to 610.27 W/m<sup>2</sup> K in multiphase flow region. In the liquid single phase region, the collector efficiency rises from 60.2% to 68.8% and the heat transfer coefficient of the fluid increases from 39.24 W/m<sup>2</sup> K to 392.31 W/m<sup>2</sup> K with an increased flow rate whereas the collector efficiency decreases from 72.5% to 62.3% as the operating pressure increases from 4 bar to 8 bar. In order to validate the simulation model an experimental test rig was built and the experiments were performed with HFE 7000 as working thermo-fluid. A new simulation model utilizing HFE 7000 has been developed and the outlet temperature of the fluid was compared with the measured outlet temperature. Both measured and simulated results have shown close conformity.

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## 1. Introduction

Utilization of fossil fuels has caused many problems such as the release of CO<sub>2</sub> to the atmosphere and its subsequent effects on our environment. This can be changed if our dependence is decreased on fossil fuels by using alternative renewable energy sources [1]. Due to its lower impacts on environment solar energy can be considered as one of the most favourable option to contribute to the energy demand with extensive applications in industry [2]. There has been an upward trend in various kinds of solar energy harvesting systems. A dynamic model of a solar pond was developed in order to investigate the effect of the sunny area ratios on the efficiency by [3]. Experimental and simulation studies on the thermal performance of a room heated with an attached sunspace were

conducted by [4]. The thermal behaviour of volumetric solar receiver with double-layer of porous media and the effects of geometry of each layer on the performance was numerically studied by [5]. Dehghan et al. [6] analysed the effect of radiation heat transfer on forced convective heat transfer mechanism through cellular porous media confined by two parallel plates. The plates subjected to constant heat flux and the Darcy–Brinkman equation was utilized to model the flow through the porous medium [6]. In another study the effects of thermal radiation on the forced convection through cellular porous media considering a combined conductive–convective–radiation heat transfer model were studied by [7].

Solar collectors which convert solar energy into heat produce either hot water or air depending on the working fluid of the collector [8]. Recently, many studies have focused on increasing the efficiency of solar water heaters. An experimental study to investigate the effect of using a mixture of ethylene glycol and copper nanoparticles as a working fluid on the collector efficiency was conducted by [9]. In another study it is reported that using

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## Nomenclature

|            |   |                      |  |
|------------|---|----------------------|--|
| $A_p$      | collector plate area, m <sup>2</sup>                                    | $U_T$                | total heat loss coefficient of the collector, W/m <sup>2</sup> K |
| $Bo$       | boiling number  | $W$                  | tube spacing, m  |
| $C_b$      | bond conductance  | $x$                  | vapour quality   |
| $C_p$      | specific heat of the working fluid, J/kg K                              | <i>Subscripts</i>    |  |
| $Co$       | convection number   | $a$                  | ambient  |
| $D$        | tube diameter, m  | $c$                  | glass cover  |
| $D_i$      | tube inner diameter, m  | $col$                | collector  |
| $D_o$      | tube outer diameter, m  | $cb$                 | convective boiling   |
| $f$        | friction factor   | $f$                  | fluid  |
| $F$        | fin efficiency  | $g$                  | gas  |
| $F_R$      | collector heat removal factor   | $in$                 | inlet  |
| $Fr$       | Froude number   | $ins$                | insolation   |
| $G$        | mass flux, kg/m <sup>2</sup> s  | $l$                  | liquid   |
| $h$        | heat transfer coefficient, W/m <sup>2</sup> K                           | $m$                  | mean   |
| $H$        | enthalpy, J/kg  | $mp$                 | multiphase   |
| $H_{fg}$   | heat of vaporization, J/kg  | $nb$                 | nucleate boiling   |
| $k$        | thermal conductivity, W/m K   | $out$                | outlet   |
| $L$        | Length, m   | $p$                  | plate  |
| $\dot{m}$  | mass flow rate, kg/s  | $sat$                | saturation   |
| $N$        | dimensionless parameter   | $sp$                 | single phase   |
| $N_c$      | number of glass cover   | $w$                  | wind   |
| $Nu$       | Nusselt number  | $wf$                 | working fluid  |
| $P$        | pressure, bar   | <i>Greek symbols</i> |  |
| $Pr$       | Prandtl number  | $\tau\alpha$         | transmittance-absorbance product                                 |
| $Q_{gain}$ | heat gain of the fluid, W   | $\beta$              | collector tilt angle, °  |
| $Q_l$      | heat loss, W  | $\delta$             | absorber plate thickness, m                                      |
| $Q_u$      | useful heat, W  | $\varepsilon$        | emissivity   |
| $Q''_u$    | useful heat rate, W/m <sup>2</sup>                                      | $\rho$               | density, kg/m <sup>3</sup>                                       |
| $Re$       | Reynolds number   | $\sigma$             | Stefan–Boltzmann constant, W/m <sup>2</sup> K <sup>4</sup>       |
| $S_{in}$   | incoming solar radiation, W/m <sup>2</sup>                              | $\psi$               | enhancement factor   |
| $T$        | temperature, K  | $\mu$                | dynamic viscosity, kg/m s  |
| $U_{back}$ | heat loss coefficient for the back of the collector, W/m <sup>2</sup> K | $\Phi$               | heat flux, W/m <sup>2</sup>                                      |
| $U_{edge}$ | heat loss coefficient for the edge of the collector, W/m <sup>2</sup> K | $\eta$               | efficiency, %  |
| $U_{top}$  | heat loss coefficient for the top of the collector, W/m <sup>2</sup> K  |                      |  |

Al<sub>2</sub>O<sub>3</sub>-distilled water nanofluid in the collector increased the thermal efficiency up to 11.7% [10].

Alternatively, collectors using organic fluid or refrigerant provide higher performance than conventional collectors where water or air is used. Because organic fluid or refrigerant undergoes a phase change, this phenomenon increases the heat transfer coefficient of the fluid and leads to an increase in the system performance [11]. Reduced parasitic energy use and freeze protection are the other benefits of such collectors [12].

Collectors using organic fluid or refrigerant can be used for further applications. Evacuated solar collector was used to generate vapour in the solar Rankine system where CO<sub>2</sub> was utilized as the working fluid [13]. Marion et al. [14] conducted both theoretical and experimental studies to indicate the potential mechanical energy generation by using solar thermal collectors which are combined with an organic Rankine cycle (ORC). The system was simulated for three organic fluids R134a, R227ea and R365mfc and the simulation model was validated against experiments using glycol–water mixture [14]. An optimization study of a solar organic Rankine cycle was conducted by [15]. In this study authors considered various models of stationary solar collectors such as a flat-plate collector, compound parabolic collector, and evacuated tube collector. Twelve substances, including dry, wet and isentropic organic fluids were taken into account as working fluids of the system and aperture area of the collector needed per unit of mechanical power output of the cycle was considered as a comparison criteria for different operating conditions of ORC

[15]. Wang et al. [16] carried out an experimental study of a low-temperature solar Rankine cycle system where flat plate collectors are used. Pure R245fa, zeotropic mixture of R245fa/R152a (0.9/0.1) and another mixture of R245fa/R152a, (0.7/0.3) were considered in the analyses [16]. An experimental study of a solar thermal system utilizing R245fa was conducted by [17]. Two stationary collectors which are evacuated tube and flat-plate collector were used in the experiments. Results showed that collector efficiencies of evacuated tube and solar collector were found 71.6% and 55.2% respectively [17].

Solar collectors using organic refrigerants also have been utilized in solar assisted heat pump systems. The thermal performance of direct expansion solar assisted heat pump system using two collector configurations which are bare collector and one cover collector were analysed in [18]. Several refrigerants were used to analyse the performance of the system. Authors reported that R-12 gives the highest performance value, followed by R-22 and R-134a [18]. Zhang et al. [19] studied the effects of refrigerant charge, solar collector area and solar collector thickness on the thermal performance of direct-expansion solar assisted heat pump system [19]. Solar assisted heat pump system for low temperature water heating application where solar collector is used as the evaporator of the heat pump was investigated by [20]. A simulation model in order to show the potential use of solar assisted heat pump system for hot water production was conducted by [21]. Authors found that the system can achieve a higher performance than conventional heat pump system [21].

Furthermore, various studies have been conducted to analyse solar collectors theoretically where the fluid undergoes a phase change.

An extended mathematical model of a boiling flat plate collector in which fluid and plate temperatures, heat transfer coefficient and vapour quality can be predicted along the collector tube was developed by [11]. Various organic fluids (R-113, n-pentane, methanol, acetone, diethyl ether) and water as a working fluid were used and the effect of transition from single phase flow to the boiling flow was taken into account in the study [11]. Results showed that no matter what working fluid was used, a higher thermal efficiency was obtained in a boiling flow as compared to single phase flow in the collector. Analysis of the boiling collector was studied by [12] where operating characteristics of condenser were also considered in their analysis. The refrigerant R-11 was utilized in a two-phase collector which operates in a thermo syphon mode. TRNSYS simulation program was used for modelling the boiling collector-condenser system which also accounted for heat losses and pressure drops in the vapour and liquid line [12].

Aziz et al. [22], conducted a numerical analysis of a solar collector which was employed as an evaporator of a heat pump cycle. In order to evaluate the size of a solar collector multiphase flow of (R-123–R134a) mixture is analysed, thermodynamic and heat transfer characteristics were calculated. The effect of various mass flow rate of the mixture, solar radiation and inlet pressure on the heat transfer coefficient and collector tube length was also taken into account. Authors concluded that both mass flow rate and solar radiation have important effects on the collector size and heat transfer coefficient of the mixture where operating pressure does not have any significant effect on the tube length of collector [22].

R134a is a hydrofluorocarbon (HFC) refrigerant and has been of interest to Sustainable Design Research Centre (SDRC), Bournemouth University in terms of industrial applications [23,24].

In this study, a detailed numerical model of a flat plate collector is developed to investigate mean temperature, useful heat gain and heat transfer coefficient of HFC-134a along the collector tube. The model can predict both the location where the fluid undergoes a phase change in the tube and the fluid exit state. The effect of mass flow rate of the fluid and operating pressure on the collector efficiency, outlet fluid temperature, absorber plate temperature and heat transfer coefficient of the fluid is also investigated. Furthermore, the model was utilized with novel-thermo fluid (HFE 7000) in order to compare the collector performance between two working fluids. Also, simulation results using HFE 7000 was validated against experimental results.

## 2. Collector specifications

Serpentine type flat plate collector was used to conduct this simulation and it consists of a cover, absorber, copper tube and insulation at the backend and edges of the collector. Cover (1) with 3 mm thick is used for reducing both radiation and convection loss from the collector. Absorber (2) is a coated stainless steel sheet, of which surface has dark-coloured paint for high absorption property. The thickness of the absorber plate is 1 mm. The fluid is circulated in the copper tubes (3) where the heat is gained from the absorber to the fluid. Conduction loss from the backend and edges of the collector is reduced by insulation (4). The schematic description of the collector is shown in Fig. 1.

## 3. Mathematical modelling

Developed analytical model of serpentine flat plate collector under a phase change is based on the model represented by [25]. The model is modified and used under following assumptions:

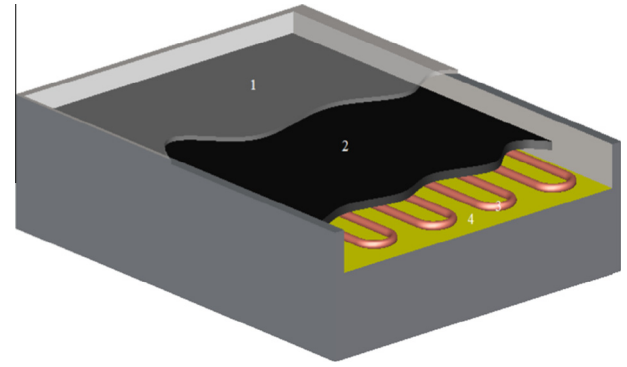


Fig. 1. Schematic of serpentine tube flat plate collector.

- Steady-state conditions.
- Absorber and glass cover thermal and radiation properties are constant (independent of temperature).
- Uniform heat flux conditions are granted instead of uniform wall temperature.
- Entry regions effects are neglected ( $L_{\text{tube}}/D = 7000$ ).
- The fluid undergoes a phase-change as the average fluid temperature reaches the boiling temperature of the fluid at corresponding pressure.
- Both single and multiphase heat transfer coefficient of the fluid varies along the tube.
- Serpentine tube considered as one flat tube to calculate heat transfer coefficients for both single and multiphase flows.

### 3.1. Heat loss calculation

There are mainly two types of losses occur in a flat plate collector which are optical and thermal losses, respectively. Optical losses can be shown as  $S_{\text{in}} (\tau\alpha)$ , where  $(\tau\alpha)$  is the transmittance-absorbance of the glass cover depending on the material properties. Thermal losses can be divided into three parts which are heat loss from the top of the collector  $U_{\text{top}}$ , from the back  $U_{\text{back}}$  and the edge of the collector  $U_{\text{edge}}$  respectively. During the calculations heat loss through the edges has been ignored. Heat loss from the top of the collector can take place by means of radiation and convection heat transfer mechanism from the glass cover to the atmosphere and from the absorber plate to the cover. Klein [26] generalized formula to calculate the top heat loss coefficient which is shown in Eq. (1).

$$U_{\text{top}} = \left( \frac{N_c}{\frac{C}{T_p} \left[ \frac{T_p - T_a}{N_c + f} \right]^e + \frac{1}{h_w}} \right)^{-1} + \frac{\sigma (T_p + T_a) (T_p^2 + T_a^2)}{(\epsilon_p + 0.00591 N_c h_w)^{-1} + \frac{2N_c + f - 1 + 0.133\epsilon_p}{\epsilon_c} - N_c} \quad (1)$$

where

$$f = (1 + 0.0889h_w - 0.1166h_w\epsilon_p)(1 + 0.07866N_c)$$

$$C = 520(1 - 0.000051\beta^2) \quad \text{for } 0^\circ < \beta < 70^\circ \text{ and if } 70^\circ < \beta < 90^\circ \text{ apply } \beta = 70^\circ$$

$$e = 0.430(1 - 100/T_p)$$

Heat losses from the back of the collector is

$$U_{\text{back}} = \frac{k_{\text{ins}}}{L_{\text{ins}}} \quad (2)$$

Therefore, total heat loss coefficient of the collector becomes

$$U_T = U_{\text{top}} + U_{\text{back}} \quad (3)$$

Absorbed solar energy on the surface of the plate can be written as

$$Q_p = A_p S_{\text{in}}(\tau\alpha) - Q_l \quad (4)$$

When plate temperature ( $T_p$ ) and heat loss coefficient ( $U_T$ ) are taken into account, then the absorbed solar energy on the plate surface can be calculated as;

$$Q_p = A_p [S_{\text{in}}(\tau\alpha) - U_T(T_p - T_a)] \quad (5)$$

### 3.2. Convective heat transfer to the fluid

Since there is incident solar energy on the absorber of the collector some portions of that will be transferred to the fluid by convection along the collector tube. This energy is called the useful energy and can be expressed as:

$$Q_u = A_p F_R [S_{\text{in}}(\tau\alpha) - U_T(T_{f,\text{in}} - T_a)] \quad (6)$$

In Eq. (6),  $T_p$  is replaced by fluid inlet temperature  $T_{f,\text{in}}$  and the new term  $F_R$  which is heat removal factor is introduced and is found to be as below;

$$F_R = \frac{\dot{m} C_p}{A_p U_T} \left[ 1 - \exp \left( \frac{-A_p U_T F'}{\dot{m} C_p} \right) \right] \quad (7)$$

( $F'$ ) is the collector efficiency factor and can be expressed as;

$$F' = \frac{(U_T)^{-1}}{W[U_T(D_o + (W - D_o)F)]^{-1} + (C_b)^{-1} + (\pi D_i h_f)^{-1}} \quad (8)$$

where  $C_b$  represents the bond conductance and it can be neglected ( $1/C_b = 0$ ) as it is assumed to be very large.

And  $F$  is the fin efficiency is

$$F = \frac{\tanh[m(W - D_o/2)]}{m(W - D_o/2)}, \text{ where } m = \sqrt{\frac{U_T}{k\delta}} \quad (9)$$

In Eq. (8),  $h_f$  represents the convective heat transfer coefficient for both single and multi-phase conditions. The determination of heat transfer coefficient in single and multiphase flows is analysed in the following section.

#### 3.2.1. Single-phase flow

For constant surface heat flux and fully developed single-phase laminar flow ( $Re < 2300$ ) in a circular tube, Nusselt ( $Nu$ ) number is constant and independent from Reynolds ( $Re$ ) and Prandtl ( $Pr$ ) numbers [27].

$$Nu = \frac{h_{sp} D}{k_{sp}} = 4.36 \quad (10)$$

For fully developed turbulent flow where  $0.5 < Pr < 2000$  and  $3 \times 10^3 < Re < 5 \times 10^6$  in a circular tube, Nusselt number can be obtained by Gnielinski equation which is a modification of Petukhov correlation [28].

$$Nu = \frac{\left(\frac{f}{8}\right)(Re - 1000)Pr}{1 + 12.7\left(\frac{f}{8}\right)^{0.5}(Pr^{2/3} - 1)} \quad (11)$$

#### 3.2.2. Multiphase flow

A large number of theoretical and experimental studies have been conducted to calculate saturated flow boiling coefficient since it is important to reduce cost and gain better design of evaporators, boilers and other multiphase process components [29]. Among many studies a general and reliable correlation was investigated

by [30]. However, it was limited to vertical flows. Another correlation which is valid for both horizontal and vertical flows was generated by [31]. In this study, the chart correlation for the estimation of saturated boiling heat transfer coefficient that includes a comparison with 800 data points from 18 experimental studies is represented. Due to the difficulties of graphical form of correlations researchers have presented equations which express these chart correlations [32].

Following the Shah's method the procedure to calculate two-phase heat transfer coefficient is represented in this section. The correlations mainly consist of four dimensionless parameters in order to estimate heat transfer coefficient. The Froude number is given by the following equation [31];

$$Fr_1 = \frac{G^2}{\rho_l^2 g D_i} \quad (12)$$

Froude number is calculated to determine if the surface is fully wet or not. For vertical tubes the surface is fully wet independently of Froude number. However, for horizontal tubes if  $Fr_1 > 0.04$  the surface is fully wet otherwise ( $Fr_1 < 0.04$ ) the surface is partly dry. Once Froude number is determined then the dimensionless parameter  $N$  can be calculated for two different conditions;

#### 3.2.3. Vertical and horizontal tubes

For vertical tubes at all conditions of  $Fr_1$  values and for horizontal tubes when  $Fr_1 > 0.04$  dimensionless parameter ( $N$ ) equals to the convection number ( $Co$ ).

#### 3.2.4. Horizontal tubes

For horizontal tubes when  $Fr_1 < 0.04$ , then the following relation is applicable.

$$N = 0.38 Fr_1^{-0.3} Co \quad (13)$$

The next dimensionless parameter is the convection number ( $Co$ ) and this can be calculated as [31];

$$Co = \left(\frac{1}{x} - 1\right)^{0.8} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \quad (14)$$

Another parameter is the enhancement factor ( $\psi$ ) and it represents the ratio of multiphase flow heat transfer coefficient ( $h_{\text{mp}}$ ) to the liquid phase heat transfer coefficient ( $h_l$ ). Dittus-Boelter equation is used to calculate liquid phase heat transfer coefficient in this analysis [31].

Then, the calculation is followed by the evaluation of both nucleate and convective boiling factor. For the calculation of nucleate boiling factor three applicability conditions of the dimensionless number ( $N$ ) are represented below [33];

For  $N > 1$

$$\psi_{\text{nb}} = 230 Bo^{0.5} \quad \text{If } Bo > 0.3 \times 10^{-4}$$

$$\psi_{\text{nb}} = 1 + 46 Bo^{0.5} \quad \text{If } Bo < 0.3 \times 10^{-4}$$

$\psi_{\text{nb}}$  and  $Bo$  represents nucleate boiling factor and boiling number respectively.  $Bo$  is calculated as [31];

$$Bo = \frac{\Phi}{G H_{\text{fg}}} \quad (15)$$

where  $\Phi$  is the heat flux,  $G$  is the mass flux and  $H_{\text{fg}}$  represents heat of vaporization.

Convective boiling factor  $\psi_{\text{cb}}$  is given by the following Eq. (16).

$$\psi_{\text{cb}} = 1.8 N^{-0.8} \quad (16)$$

As both nucleate boiling factor  $\psi_{nb}$  and convective boiling factor  $\psi_{cb}$  are calculated, the larger value is taken as enhancement factor and is used to calculate multi phase heat transfer coefficient ( $h_{mp}$ ).

For  $0.1 < N < 1$

$$\psi_{nb} = FBo^{0.5} \exp(2.74N^{-0.1}) \quad (17)$$

$\psi_{cb}$  is determined from Eq. (16) and  $\psi$  equals to the larger value of  $\psi_{nb}$  or  $\psi_{cb}$ .

For  $N < 0.1$

$$\psi_{nb} = FBo^{0.5} \exp(2.47N^{-0.15}) \quad (18)$$

$\psi_{cb}$  is evaluated from Eq. (16) and  $\psi$  equals to the larger value of  $\psi_{nb}$  or  $\psi_{cb}$ .

The constant  $F$  is as follows [33];

$Bo > 0.0011$ ,  $F = 14.7$ .

$Bo < 0.0011$ ,  $F = 15.43$ .

#### 4. Iteration procedure

During the research a numerical model is developed to simulate flat plate collector performance, fluid temperature and heat transfer coefficient along the tube. Collector specifications and operating parameters are shown in Tables 1 and 2.

Basically, the model relies on dividing the collector tube into small finite elements and calculating the fluid outlet temperature, plate temperature, the heat loss coefficient, the useful heat, the heat gain and the enthalpy values iteratively. As shown in Fig. 2 for each element the outlet temperature and enthalpy values are set equal to the inlet conditions of the next element till the end of the last element.

Initially, the model starts with calculating the heat loss coefficient by using Eq. (1). As it can be seen from Eq. (1) plate temperature ( $T_p$ ) is necessary to determine the heat loss coefficient ( $U_T$ ). Since  $T_p$  is an unknown an arbitrary value which is estimated as 5 °C higher than inlet fluid temperature is given in order to calculate  $U_T$ . Then, the model evaluates useful heat ( $Q_u$ ) by introducing the calculated  $U_T$  into Eq. (6). In order to determine heat transfer coefficient, the fluid flow is defined whether it is laminar or turbulence by calculating Reynolds ( $Re$ ) number. Depending on the flow type, Nusselt ( $Nu$ ) number is determined and heat transfer coefficient is calculated by using Eqs. (10) and (11). Fluid thermodynamic properties such as saturation temperature ( $T_{sat}$ ),

thermal conductivity ( $k$ ), density ( $\rho$ ), specific heat ( $C_p$ ), viscosity ( $\mu$ ) and enthalpy ( $H$ ) are calculated by developed regression equations. Data for the regression analysis is taken from REFPROP 9.1 programme [34] for various temperature values. Desired properties of the fluid are estimated by introducing fluid mean temperature into the regression equations. The estimated plate temperature is used to determine initial fluid mean temperature as the outlet temperature of the fluid is not known. Initial fluid mean temperature is defined as;

$$T_{f,m} = \frac{T_p + T_{f,in}}{2} \quad (19)$$

However, calculated  $Q_u$  represents the amount of useful heat for the whole collector tube. This can be represented by Eq. (20).

$$Q_u = Q_u'' \pi D_{in} L_{tube} \quad (20)$$

where  $Q_u''$  is the useful heat rate and  $\pi D_{in}$  is the surface perimeter. In other words, the useful heat rate can be obtained by dividing useful heat ( $Q_u$ ) by the surface area of the collector ( $\pi D_{in} L_{tube}$ ). To calculate heat gain for each small element the rate of useful heat is multiplied by the surface perimeter and the length of each element which is defined previously by dividing total collector length into small finite elements. Therefore, heat gain of each element can be evaluated by;

$$Q_{gain} = Q_u'' \pi D_{in} \int_0^L dx \quad (21)$$

Once the heat gain is evaluated for the first element the fluid outlet temperature is calculated by introducing  $Q_{gain}$  into Eq. (22).

$$T_{f,out} = T_{f,in} + \frac{Q_{gain}}{\dot{m} C_p} \quad (22)$$

Calculated fluid outlet temperature  $T_{f,out}$  is introduced into Eq. (23) to determine new fluid mean temperature. The new mean fluid temperature can be determined as;

$$T_{f,m} = \frac{T_{f,in} + T_{f,out}}{2} \quad (23)$$

The new mean temperature calculated by Eq. (23), is used for determining the new  $Q_u$ . After calculating the new  $Q_u$ , a new plate temperature is calculated as;

$$T_p = T_{f,in} + \frac{Q_u/A_p}{F_R U_T} (1 - F_R) \quad (24)$$

Then, the algorithm re-evaluates  $U_T$ ,  $Q_u$  and  $Q_{gain}$  by utilizing the new plate temperature and the new fluid mean temperature in Eqs. (3), (6) and (21). This process is repeated in the first loop until the difference between  $T_p$  and its value in the previous iteration and  $T_{out}$  and its value in the previous iteration is lower than convergence criteria. Convergence criterion of 0.01 °C was selected for the iterated indicators mentioned above and the results converge within 10 iterations.

Second loop checks if the outlet temperature of the element reaches the saturation temperature of the fluid at corresponding pressure. If the exit temperature does not provide saturation conditions then algorithm increases the number of element. In other words, the length of the tube is increased and the same calculations are performed for the next element. Once the outlet temperature reaches the saturation temperature then flow boiling occurs in the collector tube. In the flow boiling calculations same steps are followed with the exception of calculating single phase heat transfer coefficient, multiphase heat transfer coefficient is calculated. Since the fluid temperature is constant in boiling outlet enthalpy is determined as;

$$H_{f,out} = H_{f,in} + \frac{Q_{gain}}{\dot{m}} \quad (25)$$

**Table 1**  
Collector specifications.

|   |       |
|---|-------|
| Collector area (m <sup>2</sup> )              | 6.96  |
| Absorber plate thermal conductivity (W/m K)   | 50    |
| Absorber plate thickness (m)                  | 0.001 |
| Total length of tube (m)                      | 56    |
| Tube inner diameter (m)                       | 0.008 |
| Tube outer diameter (m)                       | 0.01  |
| Effective transmittance-absorance product (–) | 0.81  |

**Table 2**  
System parameters.

|  |                  |
|--|------------------|
| Fluid inlet temperature (K)                  | 278              |
| Fluid mass flow rate (kg/s)                  | 0.001–0.005–0.01 |
| Ambient temperature (K)                      | 275              |
| Operating pressure (bar)                     | 4–6–8            |
| Incoming solar radiation (W/m <sup>2</sup> ) | 500              |
| Wind velocity (m/s)                          | 2                |



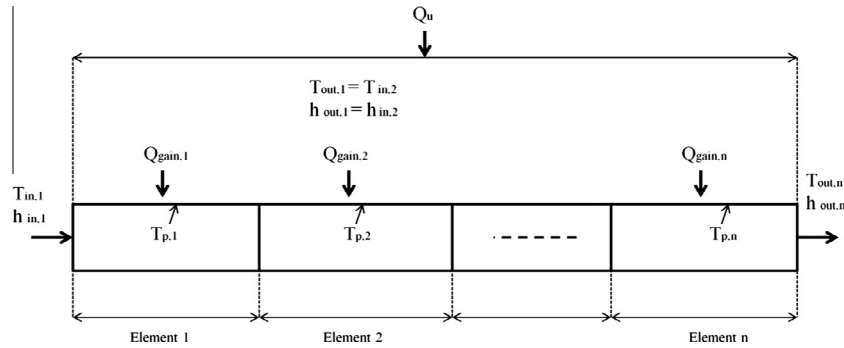


Fig. 2. Schematic of collector elements.

In boiling region vapour quality of the fluid is calculated at each element using Eq. (26).

$$x = \frac{H_{f,out} - H_l}{H_g - H_l} \quad (26)$$

where  $H_l$  and  $H_g$  represent saturated liquid and saturated vapour enthalpy of the fluid, respectively. The third loop also checks if the vapour quality reaches 1 at the end of each element. Multiphase flow calculations continue with the increase of elements since the fluid become superheated vapour ( $x > 1$ ). At this point same calculations are performed with the single phase heat transfer coefficient which was mentioned above. The only difference is superheated

vapour properties of the fluid are determined and used in the calculations.

A computer code is developed via Matlab software to perform iterative computations. Flowchart of the computer code is represented in Fig. 3.

## 5. Results and discussion

The simulation study is conducted with HFC-134a and its transport and thermodynamic properties can be found in [34]. Fig. 4 shows that the fluid enters to the collector at 5 °C temperature and 6 bar pressure. Its mean temperature increases along the col-

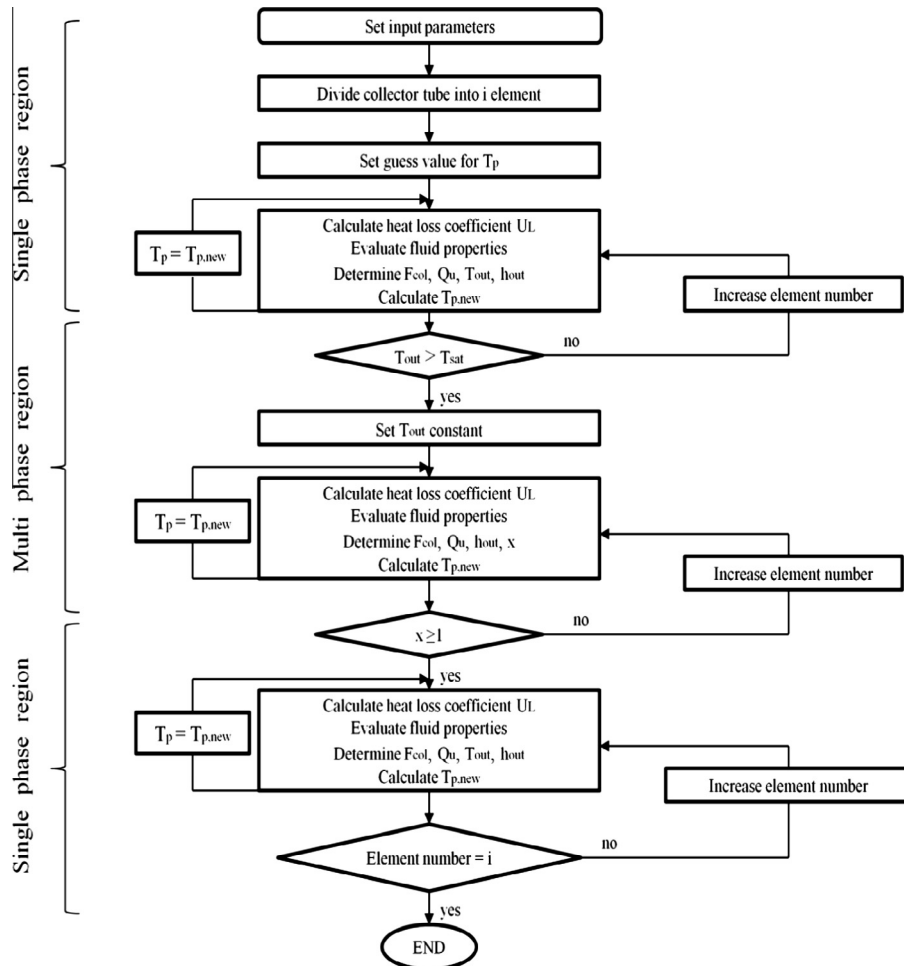


Fig. 3. Flow chart of the simulation study.

lector tube as the useful heat gain of the fluid raises. The increment in fluid mean temperature continues until it reaches the corresponding boiling temperature (22.2 °C). This region is called liquid single-phase flow which corresponds to 29% of total length of the collector tube. If the fluid reaches its saturation point, the fluid mean temperature remains constant and the fluid undergoes a phase-change. The region where saturated liquid turns into saturated vapour indicates multiphase flow or flow boiling. Boiling region occupies 58% of the tube. The heat gain of the fluid also increases during the flow boiling and this heat is used to generate vapour in the tube. Once the liquid completely turns into saturated vapour where the vapour quality equals to 1 represents the end of the boiling region. As the fluid continues gaining the heat its temperature increases again along the collector tube. This region is called vapour single-phase flow in which saturated vapour turns into superheated vapour. This phenomena proceeds till the end of the collector tube. The fluid temperature reaches 26.3 °C at the end of the tube and leaves the collector as a superheated vapour.

Fig. 5 represents the convective heat transfer coefficient of HFC-134a in both single-phase (liquid and vapour) and multiphase flows. In the liquid single phase region, the heat transfer coefficient ranged from 153.54 W/m<sup>2</sup> K to 173.93 W/m<sup>2</sup> K where as it changed from 375.2 W/m<sup>2</sup> K to 416.66 W/m<sup>2</sup> K in multiphase region. This indicates that in flow boiling the heat transfer coefficient is considerably higher than in single-phase region. Fig. 5 is generally in agreement with the studies [11,12]. In superheated vapour region, the heat transfer coefficient decreased to 73.4 W/m<sup>2</sup> K and remained almost constant.

The variation of flow boiling heat transfer coefficient versus vapour quality is shown in Fig. 6. Heat transfer coefficient increases

from 375.2 W/m<sup>2</sup> K to 498.23 W/m<sup>2</sup> K as the vapour quality increases. The maximum heat transfer coefficient (610.27 W/m<sup>2</sup> K) is observed at a quality of  $x = 0.8$  then it decreased gradually to 498.23 W/m<sup>2</sup> K.

This could be explained by the occurrence of dry out which results in a decrease in the heat transfer coefficient due to the low conductivity of dry steam [35]. Fig. 7 indicates that at constant pressure (6 bars), liquid single phase heat transfer coefficient and useful heat gain of the fluid increases with the increasing mass flow rate.

As the mass flow rate increases from 0.001 kg/s to 0.01 kg/s heat transfer coefficient and heat gain of the fluid increases from 39.24 W/m<sup>2</sup> K to 392.31 W/m<sup>2</sup> K and from 61.17 W to 535.13 W respectively. An increase in mass flow rate yields to a rise in Reynolds number, in other words a transition from laminar flow to turbulent flow in collector tubes. This results in a greater value of heat transfer coefficient and more heat which transferred to the working fluid. Instead of conventional fluid temperature absorber temperature is considered to develop efficiency equation for the collector. Collector efficiency  $\eta_{col}$  can be determined as follows:

$$\eta_{col} = \frac{Q_p}{S_{in}A_p} = \frac{A_p [S_{in}(\tau\alpha) - U_T(T_p - T_a)]}{S_{in}A_p} \quad (27)$$

It can be seen from Fig. 8 that the collector efficiency increases from 60.2% to 68.8% while absorber plate temperature decreases from 45.53 °C to 29.7 °C. This phenomenon indicates that the more heat is gained by the fluid the more heat which is absorbed on the absorber plate is transferred to the fluid. Increase in heat transfer from the absorber plate to the working fluid yields to a decrease

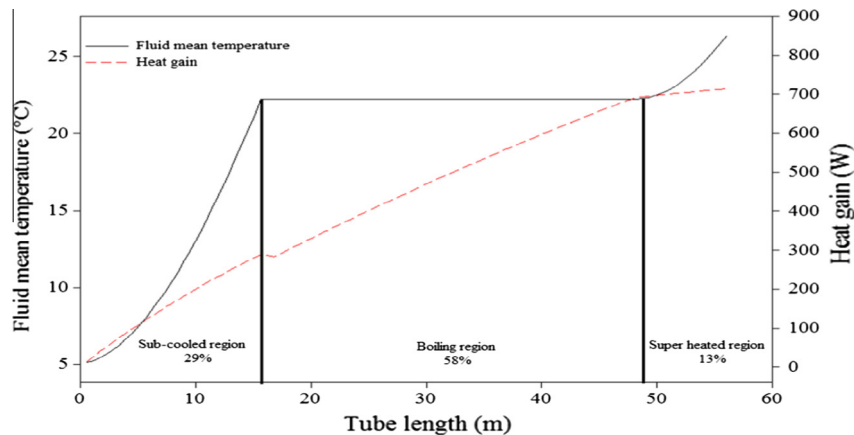


Fig. 4. Fluid mean temperature and heat gain variation along the collector tube ( $P = 6$  bar,  $m = 0.005$  kg/s).

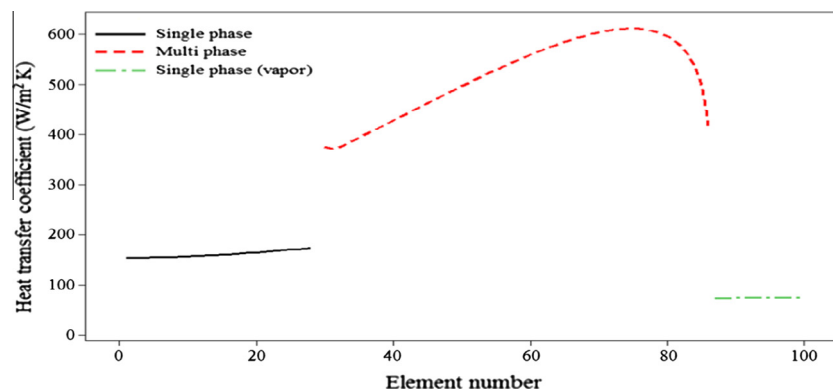


Fig. 5. Fluid heat transfer coefficient in various flow regions ( $P = 6$  bar,  $m = 0.005$  kg/s).

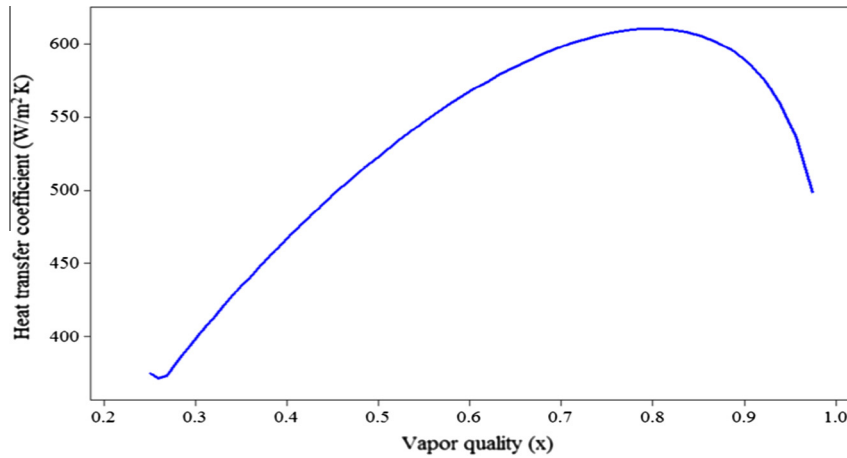


Fig. 6. Boiling heat transfer coefficient versus vapour quality ( $P = 6$  bar,  $m = 0.005$  kg/s).

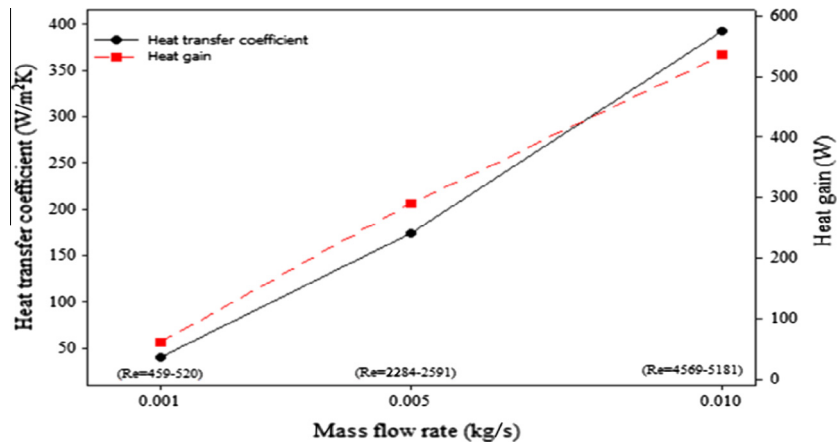


Fig. 7. Heat transfer coefficient and heat gain at various mass flow rates ( $P = 6$  bar).

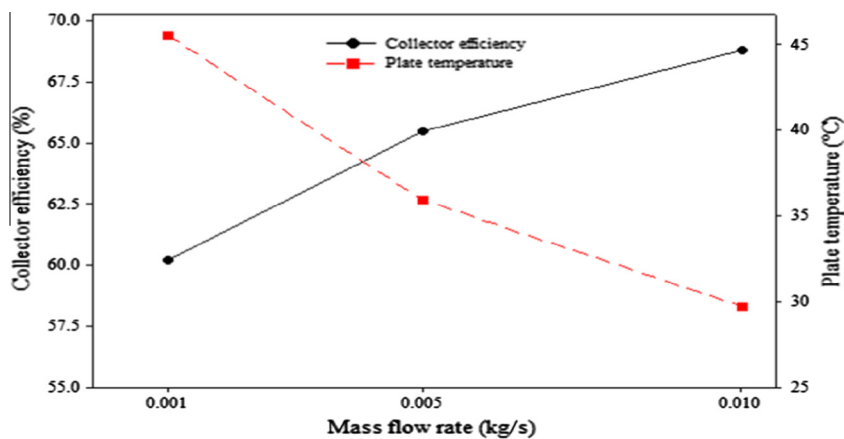


Fig. 8. Collector efficiency and plate temperature at various mass flow rates ( $P = 6$  bar).

in the absorber plate temperature which brings less heat loss from the collector plate to the surroundings. As a result higher collector efficiency value is observed.

Fig. 9 represents the effect of inlet pressure of the fluid on the plate temperature and collector efficiency at constant mass flow rate ( $\dot{m} = 0.005$  kg/s).

It can be seen from Fig. 9 that saturation temperature increases from 9.6 °C to 31.9 °C as the operating pressure of the collector increases from 4 bars to 8 bars. At saturation temperature of 9.6 °C plate temperature is found 22.1 °C whereas plate temperature is found 41.8 °C at 31.9 °C saturation temperature. Moreover, a decrease in the collector efficiency from 72.5% to 62.3% is due



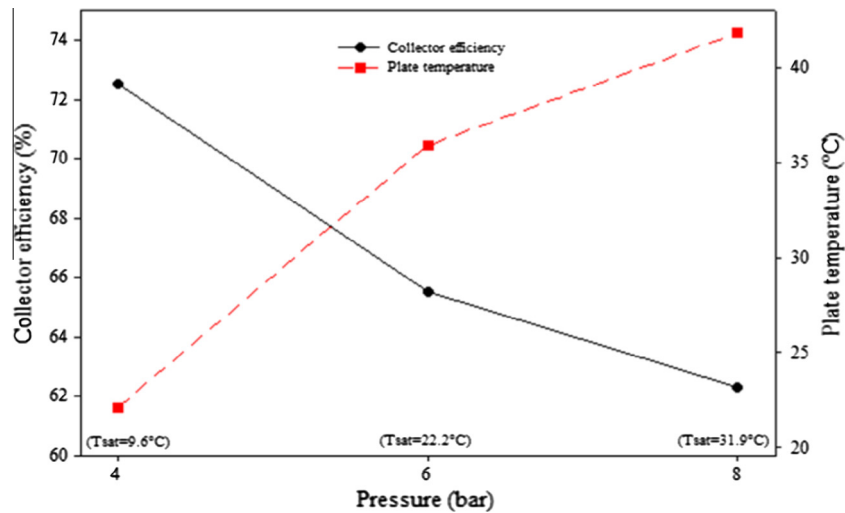


Fig. 9. Collector efficiency and plate temperature at various operating pressure ( $\dot{m} = 0.005$  kg/s).

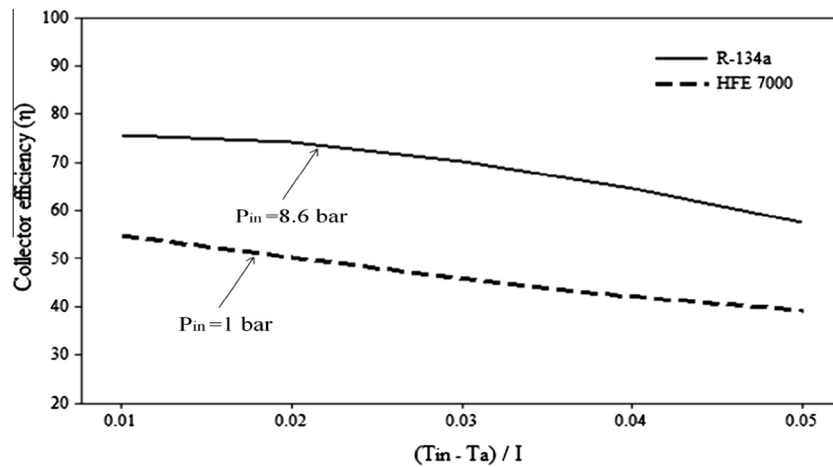


Fig. 10. Collector efficiency as a function of  $(T_{in} - T_a)/I$  with R-134a and HFE-7000 for the same saturation temperature condition.

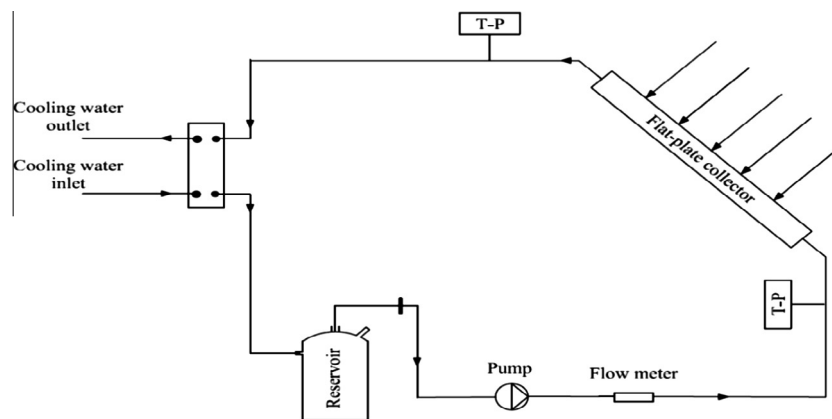


Fig. 11. Schematic of the experimental test rig.

to the increase in the plate temperature and thus causes thermal loss from the collector.

In Fig. 10 the efficiency of the flat-plate collector using both R-134a and HFE-7000 versus reduced temperature  $(T_{in} - T_a)/I$  is represented. In the analysis fluid saturation temperature for both

fluids are 15 °C higher than the fluid inlet temperature at corresponding collector operating pressures. As it can be seen from Fig. 10 that for the same incident radiation ( $I = 500$  W/m<sup>2</sup>), mass flow rate ( $\dot{m} = 0.005$  kg/s) and inlet temperature (20 °C) R-134a gives higher collector performance than HFE-7000 in both

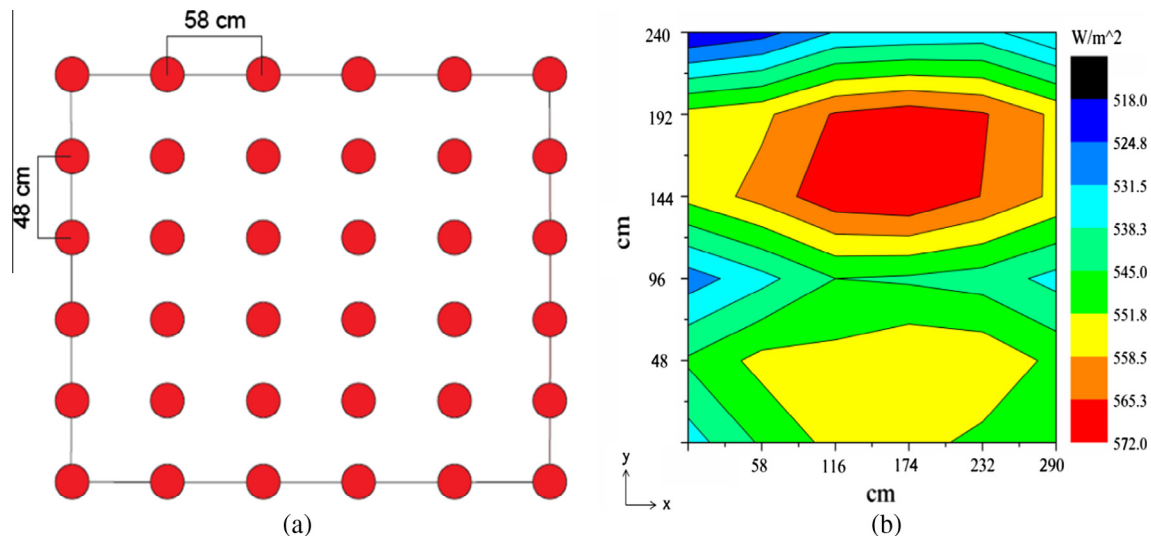


Fig. 12. (a) Measured points on the collector surface and (b) solar radiation distribution on the collector surface.

**Table 3**  
Experimental conditions.

| Parameter                   | Unit | Value       |
|-----------------------------|------|-------------|
| Collector inlet temperature | °C   | 28.4–47.9   |
| Ambient temperature         | °C   | 18          |
| Collector inlet pressure    | bar  | 1.4–2.1     |
| Fluid mass flow rate        | kg/s | 0.012–0.013 |
| Solar radiation             | W/m² | 550         |

non-boiling and boiling region. This could be explained by relatively higher thermal conductivity of R-134a ( $k = 0.083$  W/m K) than HFE-7000 ( $k = 0.0757$  W/m K) at given conditions.

## 6. Experimental validation

In order to validate the proposed mathematical and numerical models an experimental test rig was designed and commissioned. The test rig consists of a flat plate collector, a circulation pump, a reservoir and a heat exchanger (Fig. 11).

The flat plate collector specifications are given in Table 1 previously. A solar simulator was employed as an artificial source of radiation to provide steady radiant energy to the collector. Solar simulator consists of 12 glare lamps and each lamp can provide 1 kW heat output. In the experiments 6 kW heat is supplied from

the simulator to the collector. A pyranometer (Kipp&Zonen CMP3) was used to measure the irradiance on the collector surface. The solar simulator placed 2 m away from the collector surface during the measurements. Fig. 12(a) shows the measured points on the collector surface. It is found that the distributed heat flux ranged from 518 to 572 W/m² on the collector surface and the average irradiance over the collector surface ( $2.9 \times 2.4$  m) is 550 W/m² (Fig. 12(b)).

Although HFC 134a was used in the simulation study, in order to check the compatibility of the mathematical and the numerical models a novel working thermo-fluid (HFE 7000) was utilized in the experiment. The new simulation study using HFE 7000 was conducted and the fluid outlet temperature of the simulation was compared with the experimental data. The required thermodynamic and transport properties of HFE 7000 were taken from REFPROP 9.1 programme. Furthermore, HFE 7000 has a higher boiling point temperature (34 °C at 1 atm) than HFC 134a (−26 °C at 1 atm) and therefore it is easier to handle at atmospheric conditions. Initially 8 kg of HFE 7000 was charged in the system and it was compressed by the pump and sent to the collector. The absorbed heat which is converted from simulated solar radiation in the collector is transferred to the fluid via convective heat transfer mechanism. The fluid exits the collector with a higher temperature and reaches the heat exchanger. Some portion of its

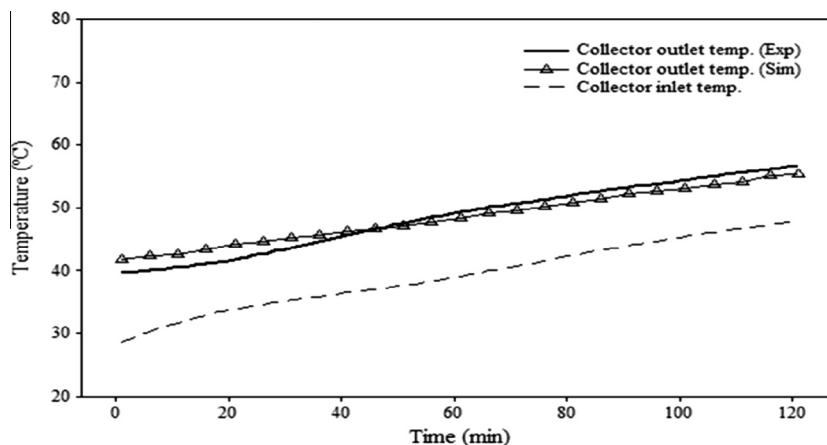


Fig. 13. Fluid temperatures at the inlet and outlet of the collector.

**Table A.1**  
Measured parameters and their uncertainties.

| Instruments                 | Measured parameters          | Range            | Uncertainty (%) |
|-----------------------------|------------------------------|------------------|-----------------|
| Thermocouples (°C)          | Collector inlet temperature  | 0–100 (°C)       | 2.5             |
|                             | Collector outlet temperature | 0–100 (°C)       | 2               |
| Pressure transmitters (bar) | Collector inlet pressure     | 0–10 (bar)       | 3.0             |
|                             | Collector outlet pressure    | 0–10 (bar)       | 3.5             |
| Flow meter (l/min)          | Fluid flow rate              | 0.12–1.6 (l/min) | 3.2             |

heat is rejected from the system and therefore its temperature decreases in the heat exchanger. Tap water is used to cool and condense the working fluid. The fluid is collected in the reservoir and is then compressed by the pump again to complete the cycle.

In order to measure temperature and pressure values of the fluid, K-type thermocouples with an accuracy of  $\pm 0.18\%$  and pressure transmitters with an accuracy of  $\pm 0.5\%$  are mounted at the inlet and outlet of the collector. A flow meter is placed 65 cm away from the pump to measure the volumetric flow rate of the fluid and the accuracy of the flow meter is  $\pm 2\%$ . Temperature, pressure and flow rate data of the fluid are recorded and transmitted to a computer by a data acquisition unit. Table 3 represents the environmental and the inlet conditions of the experiment.

It can be seen from Fig. 13 that the model can predict the outlet temperature of the fluid with a small range of deviation. The deviation for the outlet temperature of the fluid between the simulation and the experimental results ranged from 0.45% to 4.9%.

## 7. Conclusions

A numerical model for the flat plate collector was developed to simulate the collector performance under various conditions. The derived algorithm can solve the model and can iteratively evaluate the fluid mean temperature, fluid heat gain, absorber temperature and the heat transfer coefficient of the fluid at any point along the flow direction. The model is capable of calculating single phase heat transfer coefficient, as well as the multiphase heat transfer coefficient of the fluid if its temperature surpasses its boiling temperature in the collector. The algorithm was implemented by utilizing the MATLAB programme. The model was solved with HFC-134a refrigerant. The simulation results showed that in the flow boiling region the heat transfer coefficient ( $375.2\text{--}416.6\text{ W/m}^2$ ) is higher than in single phase liquid ( $153.54\text{--}173.93\text{ W/m}^2$ ) and single phase vapour ( $73.4\text{ W/m}^2$ ) regions. Effect of the mass flow rate and the pressure on the heat transfer coefficient, collector efficiency and heat gain of the fluid is also taken into account in the simulation study. It is found that heat transfer coefficient have shown a dependency on the flow rate of the flow. As the mass flow rate, as well as the Reynolds number of the flow increases, the flow becomes turbulent. In turbulent region higher heat transfer coefficient is obtained than laminar region which leads to an increase in the heat gain of the fluid and the efficiency of the collector. Analysis of the simulation results also showed that operating pressure, in other words saturation pressure of the fluid has an effect on the collector efficiency. In higher saturation pressure condition in the collector saturation temperature becomes further from the inlet fluid temperature and the efficiency of the collector decreases. Furthermore, the collector efficiency with two working fluids (R-134a and HFE-7000) is compared for the same inlet conditions. It is found that R-134a gives higher collector efficiency due to its superior properties compared to the HFE-7000 at given conditions. An experimental test rig was built in order to validate the simulation model against experimental results. HFE-7000 was utilized in the experiment and the simulation results for HFE-7000 show good agreement with the experimental results.

## Acknowledgment

This research is funded by Future Energy Source (FES) Ltd, UK. The authors acknowledge in-kind support provided by Mr Brian Camfield and Mr Tony Camfield.

## Appendix A. Uncertainty analysis

In order to evaluate the accuracy and the reliability of the measured parameters an uncertainty analysis is conducted. The measured instruments and their uncertainties are shown in Table A.1.

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