NUMERICAL SIMULATION OF CONCENTRATING SOLAR COLLECTOR P2CC WITH A SMALL CONCENTRATING RATIO

by

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Solar energy may be practically utilized directly through transformation into heat, electrical or chemical energy. A physical and mathematical model is presented, as well as a numerical procedure for predicting thermal performances of the P2CC solar concentrator. The demonstrated prototype has the reception angle of 110° at concentration ratio CR = 1.38, with the significant reception of diffuse radiation. The solar collector P2CC is designed for the area of middle temperature conversion of solar radiation into heat. The working fluid is water with laminar flow through a copper pipe surrounded by an evacuated glass layer. Based on the physical model, a mathematical model is introduced, which consists of energy balance equations for four collector components. In this paper, water temperatures in flow directions are numerically predicted, as well as temperatures of relevant P2CC collector components for various values of input temperatures and mass flow rates of the working fluid, and also for various values of direct sunlight radiation and for different collector lengths. The device which is used to transform solar energy to heat is referred to as solar collector. This paper gives numerical estimated changes of temperature in the direction of fluid flow for different flow rates, different solar radiation intensity and different inlet fluid temperatures. The increase in fluid flow reduces output temperature, while the increase in solar radiation intensity and inlet water temperature increases output temperature of water. Furthermore, the dependence on fluid output temperature is determined, along with the current efficiency by the number of nodes in the numerical calculation.

Key words: *solar radiation, thermal, solar collector, concentration ratio*

Introduction

The critical energy situation forces the mankind to reconsider thoroughly all possibilities given by renewable energy sources, and our present knowledge and technology. Unfortunately, many factors have decided that between renewable and ecological, and non-renewable energy sources, the first choice has always been non-renewable energy sources. The device which is used to transform solar energy to heat is referred to as solar collector. Depending on the temperatures gained by them, the solar thermal collector (STC) can be divided into low, middle and high temperature systems. Mid-temperature systems are applicable for

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refrigeration systems and industrial processes. The advantage of an STC used in middle temperature conversion of solar energy (100-400 °C) compared to a high-STC is that it does not require a sharp focus and accurate tracking of the apparent orbit of the Sun. It can be stationary or running with the occasional collector or absorber. Concentrating collectors for concentration ratio greater than 10 can use only direct sunlight, while the diffusion does not respond. Most often, compound parabolic concentrators are used that have two parabolic reflectors with the solar absorber between them.

Numerous theoretical and experimental researches of solar collectors for the mid-temperature conversion of solar radiation into heat via a liquid as a working fluid have been conducted. Tchinda *et al.* [1] investigated heat transfer in a P2CC collector and developed a model which during numerical analysis of heat transfer takes into account axial heat transfer in a CPC collector. The impact of various parameters has been investigated, such as input temperature and the value of the flow of working fluid on dynamical behavior of the collector. Norton *et al.* [2] developed a theoretical heat transfer model, which describes steady state heat behavior of a symmetric P2CC collector, and investigated the impact of the tilt angle of the collector to the P2CC collector performance. Oommen *et al.* [3] experimentally investigated a complex parabolic collector (CPC), with water as a working fluid. They investigated the efficiency of the collector for various input temperatures of water, as well as the efficiency of the collector in the conditions of steam generation. Gata Amaral *et al.* [4] worked on the development of solar concentrating collectors based on the shape known as a CPC of solar energy, which allowed the utilization of maximal solar energy ratio. They followed the efficiency of P2CC collector with refrigeration systems during the year. P2CC collectors were of different length and with different concentrating ratios. In this paper, physical and mathematical models are given, as well as the numerical calculation of a solar collector P2CC. This is a solar concentrator, designed for the mid-temperature conversion [5-8]. The paper gives a change of fluid temperate and collector components in the direction of the flow, as well as the influence of the input fluid temperature, radiation intensity, mass flow rate to the heating behavior and current efficiency of the collector. The change of fluid temperature was given in the direction of the fluid flow for six collectors connected in a row.

Basic features of the concentrating STC

The device which increases the density of solar radiation on the absorber of the receiver above the environmental level is called a concentrator. A frequent name for solar energy concentrator is also a "focusing STC". One of the major features of the focusing systems is the concentration ratio CR, which represents the ratio of the effective surface of the aperture and the surface of the absorber of STC. The concentration is improved by using reflecting and refracting parts. They focus solar flux into a point or a line. Because of this, their names are either a point-concentrator or a line-concentrator. The first can provide much higher temperatures than the second. They belong to a group of STC for high-temperature conversion of solar radiation into heat (indirectly in electric energy), but they demand more complex optical and moving devices. Practically, when it operates, every focusing STC requires a change in position to get the maximal amount of heat from sun radiation. This radically increases the price of the STC. Rotation of the STC can be done manually, when terms of use allow it. The shortcoming of mobile focusing collectors is that they use only direct sun radiation, and in (most cases) they do not use any diffuse solar radiation. The investigated concentrator has the concentration ratio $CR = W/(2\pi r_{r,0}) = 1.38$.

Physical model

In this paper, a laminar flow of the fluid (l) through the absorber (2) is assumed. The pipe of the absorber is subjected to solar radiation. Solar rays go through the transparent cover (5) on the collector apparatus. A part of the solar radiation falls directly to the absorber after going through the transparent cover of the pipe (4) and evacuated area (3). Another part of the solar radiation goes through the transparent cover layer of the pipe, after reflection from the reflector (6), and falls at the pipe absorber. The look of a P2CC module is presented in fig. 1(a), while the scheme of its cross section is given in fig. 1(b).

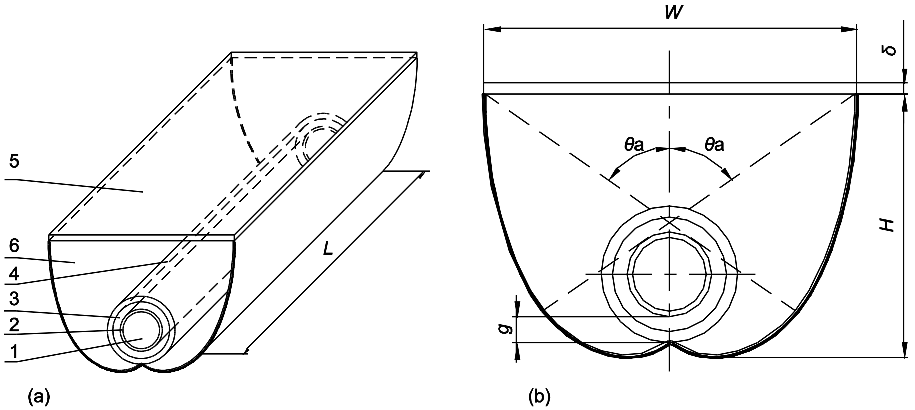

Figure 1. (a) P2CC collector module, (b) cross-section of a P2CC collector

The P2CC module consists of a complex cylindrical-parabolic reflection surface and a copper tube with ∅15 mm outside diameter used as the absorber. The investigated concentrator labelled as P2CC has a tubular receiver and two identical reflectors, one on each side of the tubular receiver. Figure 1(b) shows a cross-section of the P2CC, where the tubular receiver is presented by two concentric circles, while each reflector is presented by a curve. The tubular receiver is a concentric tube consisting of a metal tube (absorber) inside a glass tube. The cylindrical copper absorber is surrounded by a glass layer for lowering convective loss from the collector pipe. The area between the pipe absorber and the glass surrounding layer is evacuated. Conversion of solar energy into heat is conducted on the pipe collector. The pipe absorber is colored with a selective color of high absorbing properties and low emissivity (ε_r). The area of the reflector has high reflection ratio (ρ_m). Between the pipe absorber and the reflector there is a gap that stops heat transfer from the collector pipe to the reflector. Water is the working fluid, with a laminar flow through the pipe of the collector. The apparatus of the collector is covered by transparent cover layer made of plexiglas, so that the reflector area could be saved from wear and to lower the heat loss from the assembly pipe absorber – surrounding pipe layer. The collector consists of six P2CC modules which are connected parallel to the flow paths, and flow properties inside can be considered equal.

Mathematical model

Based on the physical model represented by the P2CC collector, through whose absorber pipe a laminar flow of water occurs, and which is subjected to solar radiation, a mathematical model is proposed. The mathematical model consists of the energy balance equations for four P2CC module components: (1) working fluid, (2) collector pipe-absorber, (3) surrounding pipe layer, and (4) transparent cover. During the set-up of the mathematical model only one module of P2CC collector was analyzed. There is a theoretically considered laminar flow through one module of the P2CC collector. We used flow rates that provide laminar fluid flow (Re < 2320). The objective of this research was to start from a simple model for laminar flow, no matter how this was applicable for the real flow of the receiver. The study of turbulent flow through the receiver will be a subject of our following research in this field.

Basic assumptions upon which the model is defined

The following assumptions were introduced for the definition of the mathematical model:

(1) heat transport in the transparent cover, surrounding cover layer, collector surrounding layer, pipe absorber, and the fluid is transient,
(2) thermo-physical properties of the P2CC collector component material (ρ^*, c, λ), as well as optic properties ($\rho, \tau, \varepsilon, \alpha$) of the components of P2CC collector, do not depend on coordinates, temperature, or time,
(3) fluid flow shape is the same in every axial cross-section, thus the tangential velocity component w_φ does not exist,
(4) radial velocity component w_r is neglected as a value of smaller order and thus the convection in radial direction is also neglected, and
(5) conduction in the axial direction has a negligibly small contribution to the resulting heat transport compared to the convection.

P2CC module is constructed with ideal geometry, and the concentration ratio is:

$$C_a = (\sin\theta_a)^{-1} = A_c A_r^{-1} \tag{1}$$

Energy balance equations

The mathematical model consists of equations of energy balance for all of the four components of the P2CC module, relations for determining heat transfer coefficient, and relations for radiation absorbed by the relevant system components. In order to gain a unified solution from the system of equations, initial and boundary conditions are defined.

Energy balance equation for working fluid

Energy balance for an elementary fluid volume of length dz in the axial direction, after sorting, may be written as:

$$\rho_f^* c_{pf} A_f^* \frac{\partial T_f}{\partial t} = -\rho_f^* c_{pf} A_f^* w_z \frac{\partial T_f}{\partial z} + U_{r/f}(T_r - T_f) 2\pi r_{r,o} \tag{2}$$

where ρ_f^*, c_{pf}, and w_z represent the density, specific heat, and velocity in the axial direction of the elementary fluid volume, respectively, and $A_f^* = r_{r,i}\pi$ is the area of the cross section of the elementary fluid particle. The second term on the right hand side of eq. (2) represents the heat gained from the outer side of the collector pipe.

Initial conditions

It was assumed, for the initial conditions (equation 3), that in the initial time point the temperature field in all components was equal to the environment temperature T_a, while the fluid temperature at the inlet was constant in time:

$$t = 0,\ T_f = T_r = T_e = T_c = T_a; \quad z = 0,\ T_f = T_{in} \tag{3}$$

Energy balance equation for the collector absorber pipe

Energy balance for elementary part of the collector of length dz in the axial direction may be written as:

$$\rho_r^* c_r \frac{\partial T_r}{\partial t} A_r^* \mathrm{d}z\mathrm{d}t = \frac{\partial}{\partial z}\left(\lambda_r \frac{\partial T_r}{\partial z}\right) A_r^* \mathrm{d}z\mathrm{d}t - (h_{c,r/e} + h_{r,r/e})(T_r - T_e)2\pi r_{r,o}\mathrm{d}z\mathrm{d}t -$$

$$- U_{r/f}(T_r - T_f)2\pi r_{r,o}\mathrm{d}z\mathrm{d}t + (q_{b,r} + q_{d,r})2\pi r_{r,o}\mathrm{d}z\mathrm{d}t \tag{4}$$

where λ_r, ρ_r^*, and c_r are the head conduction coefficient, density and specific heat for elementary pipe volume, respectively, and $A_r^* = (r_{r,0}^2 - r_{r,i}^2)\pi$ is the area of the cross section of elementary part of the collector pipe.

The first term on the right hand side of eq. (4) represents heat transport by conduction in the axial direction of the collector, the second is the heat lost due to convention and radiation between the collector pipe and the surrounding layer of the collector pipe, the third term represents the heat given to the working fluid, and the fourth represents the heat gained via solar radiation.

Boundary conditions for this equation are:

$$\text{for } z = 0,\ \frac{\partial T_r}{\partial z} = 0;\ \text{ for } z = L,\ \frac{\partial T_r}{\partial z} = 0 \tag{5}$$

Energy balance equation for the transparent cover of the pipe collector

Energy balance for the elementary part of the surrounding layer of the collector pipe of length dz is written as:

$$\rho_e^* c_e A_e^* \mathrm{d}z \frac{\partial T_e}{\partial t}\mathrm{d}t = (h_{c,r/e} + h_{r,r/e})(T_r - T_e)2\pi r_{r,o}\mathrm{d}z\mathrm{d}t -$$

$$-(h_{c,e/c} + h_{r,e/c})(T_e - T_c)2\pi r_{e,o}\mathrm{d}z\mathrm{d}t + (q_{b,e} + q_{d,e})2\pi r_{r,o}\mathrm{d}z\mathrm{d}t \tag{6}$$

where ρ_e^* and c_e are the density and specific heat of the surrounding layer of the tube receiver, and $A_r^* = (r_{e,0}^2 - r_{e,i}^2)\pi$ is the area of the cross section of elementary part of the surrounding layer of the collector. The second term on the right hand side of eq. (6) is the heat lost due to radiation from the surrounding layer of the collector to the transparent cover.

Energy balance equation for the transparent cover

Energy balance for the elementary part of the transparent cover of the concentrating collector, with length dz in the z-direction and width W, may be written as:

$$\rho_c^* c_c A_c^* \mathrm{d}z \frac{\partial T_c}{\partial t}\mathrm{d}t = (h_{c,e/c} + h_{r,e/c})(T_e - T_c)2\pi r_{e,o}\mathrm{d}z\mathrm{d}t - h_{c,c/a}(T_c - T_a)W\mathrm{d}z\mathrm{d}t -$$

$$-h_{r,c/s}(T_c - T_s)W\mathrm{d}z\mathrm{d}t + (q_{b,c} + q_{d,c})2\pi r_{r,o}\mathrm{d}z\mathrm{d}t \tag{7}$$

where ρ_e^* and c_e are the density and specific heat of the surrounding layer of the collector, respectively, and $A_c^* = W\delta$ is the area of the cross section of elementary part of the transparent cover of the collector. The second term on the right hand side of eq. (7) is the heat lost due to convection from the transparent cover to the environment, the third term is the heat lost due to radiation between the transparent cover and the sky.

Heat transfer coefficients

Heat transfer coefficient for convection from the cover to the environment, caused by air flow (wind) $h_{c,c/a}$, is given by the following relation [9]:

$$h_{c,c/a} = (5.7 + 3.8v)A_c A_r^{-1} \tag{8}$$

Total heat transfer coefficient $U_{r/f}$ between the outside surface of the collector pipe and the working fluid is:

$$U_{r/f} = \frac{1}{\dfrac{r_{r,o}\ln\left(\dfrac{r_{r,o}}{r_{r,i}}\right)}{\lambda_r} + \dfrac{A_{r,o}}{h_f A_{r,i}}} \tag{9}$$

where $r_{r,o}$ and $r_{r,i}$ are the outside and inside diameter of the pipe of the collector, λ_r is the heat conduction coefficient of the collector pipe, and h_f is the coefficient of heat transfer by convection from the internal surface of the pipe of the collector to the working fluid.

Coefficient of heat transfer h_f, is given as:

$$h_f = \mathrm{Nu}_f \lambda_f d^{-1} \tag{10}$$

Numerical model

Dividing the calculation area into control volumes

The numerical model is based on dividing every component of the P2CC collector, with length L,on a number of control volumes, thus making each control volume surrounding a single node. Differential balance equations are integrated for every control volume of the relevant component. N number of nodes in z-direction is taken. Nodes 1 and N are positioned on the system boundary. Control volumes surrounding these nodes are of $\Delta z/2$ in length, whilst control volumes surrounding internal nodes are of Δz in length. Figure 2 presents a divided working fluid with N control volumes. Such division was also made for other P2CC collector components.

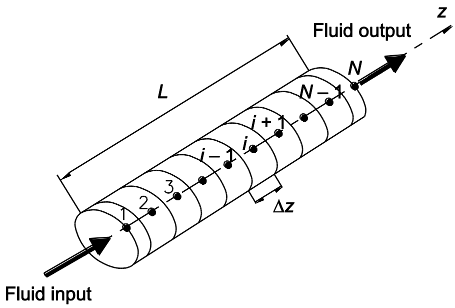

Figure 2. Grid for the working fluid of P2CC collector

Discretization of the energy balance equation for the working fluid

In order to create a discretized equation for the working fluid, a control volume around node i was taken, as shown in fig. 3. Node i is marked by P, and the neighboring nodes, node $i-1$ and $i+1$ are marked by W and E, respectively. Since the time is a one way co-ordinate the solution is obtained by "marching" through time beginning with initial temperature field. Temperature T is set for the nodes in the time step t, and T values in the time step $t + \Delta t$ should be found. "Old" (set) values for T in the nodes are marked $T_{Fp}{}^{0}$, $T_{FE}{}^{0}$, $T_{FW}{}^{0}$, and "new" (unknown) values in $t + \Delta t$ time step are marked $T_{Fp}{}^{1}$, $T_{FE}{}^{1}$, and $T_{FW}{}^{1}$. Discretization of the differential equations of energy balance for the working fluid can be conducted by its integration for the control volume, as shown in fig. 3.

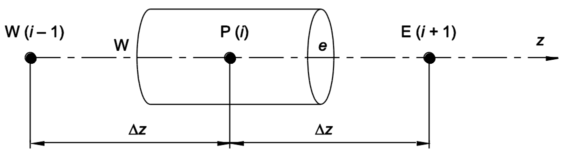

Figure 3. Control volume around a node points for working fluid

$$\rho_f^* c_{pf} A_f^* \int_w^e \int_t^{t+\Delta t} \frac{\partial T_f}{\partial t} \mathrm{d}t\mathrm{d}z = -c_{pf} A_f^* \int_t^{t+\Delta t} \int_w^e \rho_f^* w_z \frac{\partial T_f}{\partial z} \mathrm{d}z\mathrm{d}t + \int_t^{t+\Delta t} \int_w^e U_{r/f} (T_r - T_f) 2\pi r_{r,o} \mathrm{d}z\mathrm{d}t \quad (11)$$

By sorting eq. (11), an equation for node (i) for the fluid in the first time step is:

$$T_{f,i}^1 = \frac{\Delta t}{\rho_f^* c_{pf} A_f^* \Delta z} \left[A_1 T_{f,i-1}^0 + A_2 T_{f,i}^0 + A_3 T_{f,i+1}^0 + A_4 T_{r,i}^0 \right] \quad (12)$$

Discretization of the energy balance equation for the collector pipe

By integrative differential equation of the energy balance for the collector pipe for the control volume in the time interval from t to $t + \Delta t$ eq. (13) can be written.

Re-arranging the balance eq. (14) gives the equation for temperature $T_{r,i}{}^{1}$ nodal point (i) the receiver tube in the next instant of time (^{1}), where $\lambda_{re} = \lambda_{rw} = \lambda_r$.

$$T_{r,i}^1 = \frac{\Delta t}{\rho_r^* c_r A_r^* \Delta z} \left[b_1 T_{r,i-1}^0 + b_2 T_{r,i}^0 + b_3 T_{r,i+1}^0 + b_4 T_{f,i}^0 + b_5 T_{e,i}^0 + b_6 \right] \quad (13)$$

$$\rho_r^* c_r A_r^* \int_w^e \int_t^{t+\Delta t} \frac{\partial T_r}{\partial t} \mathrm{d}t\mathrm{d}z = A_r^* \int_t^{t+\Delta t} \int_w^e \frac{\partial}{\partial z}\left(\lambda_r \frac{\partial T_r}{\partial z} \right) \mathrm{d}z\mathrm{d}t -$$

$$- \int_t^{t+\Delta t} \int_w^e (h_{c,r/e} + h_{r,r/e})(T_r - T_e) 2\pi r_{r,o} \mathrm{d}z\mathrm{d}t -$$

$$- \int_t^{t+\Delta t} \int_w^e U_{r/f} (T_r - T_f) 2\pi r_{r,o} \mathrm{d}z\mathrm{d}t + \int_t^{t+\Delta t} \int_w^e (q_{b,r} + q_{d,r}) 2\pi r_{r,o} \mathrm{d}z\mathrm{d}t \tag{14}$$

Discretization of the energy balance equation for transparent cover of the collector

In the same fashion, by integrating differential equation for the energy balance equation for transparent cover collector for the control volume in the time interval t to $t + \Delta t$, an equation for temperature of the node (i) is gained for the cover of the collector in the first time step:

$$\rho_c^* c_c A_c^* \int_w^e \int_t^{t+\Delta t} \frac{\partial T_c}{\partial t} \mathrm{d}t\mathrm{d}z = \int_t^{t+\Delta t} \int_w^e (h_{c,e/c} + h_{r,e/c})(T_e - T_c) 2\pi r_{e,o} \mathrm{d}z\mathrm{d}t -$$

$$- \int_t^{t+\Delta t} \int_w^e h_{c,c/a} (T_c - T_a) W \mathrm{d}z\mathrm{d}t - \int_t^{t+\Delta t} \int_w^e h_{r,c/s} (T_c - T_s) W \mathrm{d}z\mathrm{d}t + \int_t^{t+\Delta t} \int_w^e (q_{b,c} + q_{d,c}) 2\pi r_{r,o} \mathrm{d}z\mathrm{d}t \tag{15}$$

$$T_{e,i}^1 = \frac{\Delta t}{\rho_e^* c_e A_e^* \Delta z} \left[c_1 T_{e,i}^0 + c_2 T_{r,i}^0 + c_3 T_{c,i}^0 + c_4 \right] \tag{16}$$

Discretization of the energy balance equation for transparent cover

By integrating the energy balance differential equation for the transparent cover for the control volume in the time interval t to $t + \Delta t$, an equation for the temperature of the node (i) is obtained for the transparent cover for the first time step:

$$T_{c,i}^1 = \frac{\Delta t}{\rho_c^* c_c A_c^* \Delta z} \left[d_1 T_{c,i}^0 + d_2 T_{e,i}^0 + d_3 \right] \tag{17}$$

The method for solving algebraic equations

The discretization equation of energy balance components for the corresponding receiver modules P2CC gets a set of coupled linear algebraic equations of the form eqs. (14), (15), (16), and (17), with the total number of equations equaling the number of nodal points in the axial direction for all four components.

Since the thermophysical properties of working fluid (water) depend on the temperature, there is a need for an iterative method of solving. The temperature of all components of nodal points of equal temperature environment is taken for the initial temperature field. Nodal point $i = 1$ for the working fluid at the entrance to the pipe receiver has a constant temperature T_{in} for each time instant. The solving procedure continues until the temperature difference between two successive iterations, for all nodal points, reaches the pre-set errors ε_e. After several tests, we found that $\varepsilon_e = 10^{-5}$ °C and $\Delta t = 0.0005$ s. These values were used for all the results of the budget, which are presented in this paper. The procedure began with entering the calculation of basic geometric characteristics of P2CC receiver, number of nodal points, thermophysical and optical properties of system components, working fluid inlet temperature, intensity of solar radiation, and wind velocity.

The results of numerical simulations

During the calculation, it was assumed that the solar rays fall normal to the aperture plane. It was assumed that all off the radiation falls to the aperture directly. Table 1 shows dimensions and thermophysical properties for the P2CC module, for which the numerical simulation was conducted.

Table 1. P2CC module data

Dimensions	$L = 1000$ mm, $H = 465$ mm, $W = 650$ mm, $r_{r,i} = 65$ mm, $r_{\mathrm{r},0} = 5$ mm, $r_{e,i} = 10$ mm, $r_{e,0} = 12$ mm, $\delta = 5$ mm
Optical properties	$\varepsilon_r = 0.05$, $\varepsilon_e = \varepsilon_c = 0.85$, $\rho_e = \rho_c = 0.05$, $\rho_r = 0.15$, $\rho_m = 0.85$, $\tau_e = \tau_c = 0.90$, $\alpha_r = 0.95$, $\alpha_e = \alpha_c = 0.05$
Thermo-physical properties	$\rho_e = \rho_c = 2700$ kgm^{-3}, $\rho_r = 8390$ kgm^{-3}, $\mathrm{C_e} = \mathrm{C_c} = 840$ Jkg^{-1}K^{-1}, $\mathrm{C_r} = 383$ kg^{-1}K^{-1}, $\lambda_\mathrm{r} = 395$ Wm^{-1}K^{-1}

During the calculation, the environment temperature $T_a = 28$ ºC and wind speed $v = 2$ m/s were kept constant. For direct solar radiation $I_b = 950$ W/m^2, inlet fluid temperature $T_{ul} = 32$ ºC and different flow values, the values of local temperature of the working fluid in the direction of the flow are given in fig. 4 (a), which shows that by increasing the fluid flow rate, the output temperature is reduced. Figure 4 (b) shows the change of temperature of the collector pipe, working fluid, surrounding collector pipe layer, and transparent cover in the flow direction, for the direct solar radiation $I_b = 950$ W/m^2, inlet temperature of the fluid $T_{ul} = 32$ ºC and working fluid flow 0.00162 kg/s.

Figure 4(a), shows the impact of the direct radiation intensity on the working temperature change in the flow direction, for the inlet fluid temperature $T_{ul} = 32$ ºC and fluid flow 0.00162 kg/s. Figure 4(b) depicts the impact of the value of local inlet temperature of the fluid on the local fluid temperature for the flow rate of 0.00162 kg/s and direct radiation of $I_b = 950$ W/m^2.

Figure 5(a), depicts the impact of inlet fluid temperature on the current heat efficiency, for direct radiation $I_b = 950$ W/m^2 and flow rate 0.00162 kg/s. The graphs shows that by raising inlet temperature of the fluid, heat efficiency drops. This is explained by the fact that solar flux and environment temperature remain constant, and the difference between inlet fluid temperatures is reduced, thus reducing useful energy.

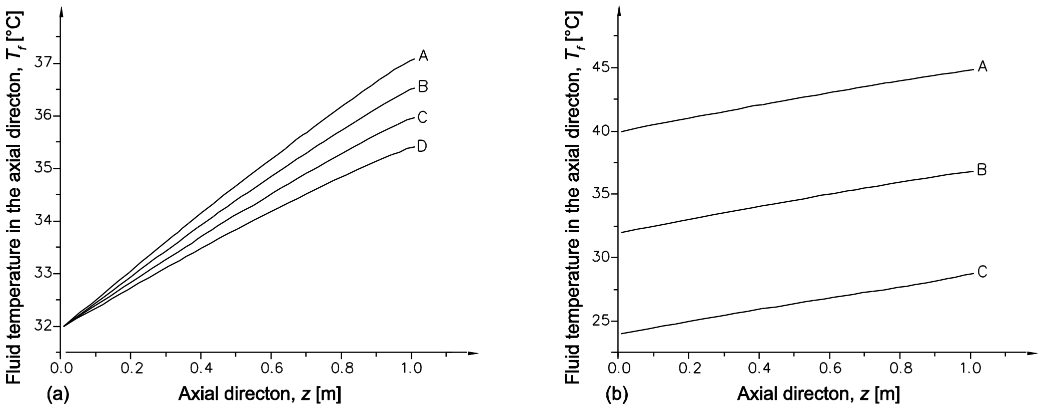

Figure 4. (a) Impact of the direct radiation value on the local temperature in the flow direction; A – I_b =1000 W/m^2, B – I_b = 900 W/m^2, C – I_b = 800 W/m^2, D – I_b = 700 W/m^2, (b) Impact of the inlet fluid temperature value on the local fluid temperature in direction flow; A – T_{ulf} = 40 °C, B – T_{ulf} = 32 °C, C – T_{ulf} = 24 °C

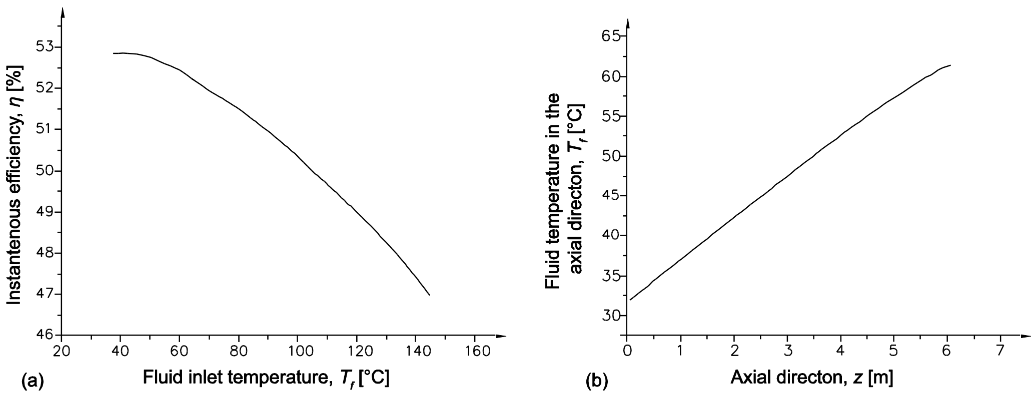

Figure 5. (a) Impact of the input fluid temperature value on the current heat efficiency of the P2CC collector, (b) Impact of the length on the value of the output temperature

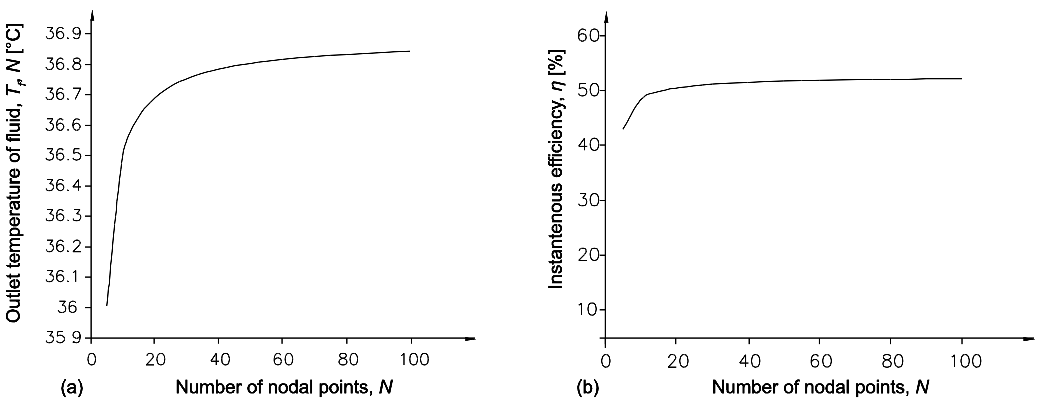

Figure 6. (a) Impact of the number of nodes in the network on the value of the outlet temperature of fluid, (b) Impact of the number of nodal points on the value of current efficiency of the receiver

Figure 5(b) shows the impact of collector length on the output fluid temperature, for the P2CC module length L = 6 m, I_b = 950 W/m^2, input fluid temperature T_{ul} = 32 ºC and flow rate 0.00162 kg/s. Figures 6(a) and 6(b) show the effect of the number of nodes on the output value of the fluid temperature and the current performance of the receiver, respectively. The number of nodes in the network ranged from 5 to 100. In these calculations, the ambient temperature of T_a = 28 ºC and wind velocity of v = 2 m/s were both taken as constants. Direct radiation considered as input was I_b = 950 W/m^2, inlet temperature T_{ul} = 32 ºC and effective mass flow rate $\dot{m}$ = 0.00162 kg/s.

For the implementation of numerical procedures for solving the mathematical model of fluid flow in the pipe P2CC receiver, and thermal behavior of other components of the receiver, the program was developed in FORTRAN-77, and its algorithm is shown in fig. 7.

An important feature of the program FORTRAN-77 is that it has a simple user interface that allows a direct input of all parameters of interest using the keyboard. This opens wide possibilities of numerical simulations of real experiments and detailed numerical analysis of the impact of geometric and operating parameters on the mechanism of transformation of solar energy into heat energy.

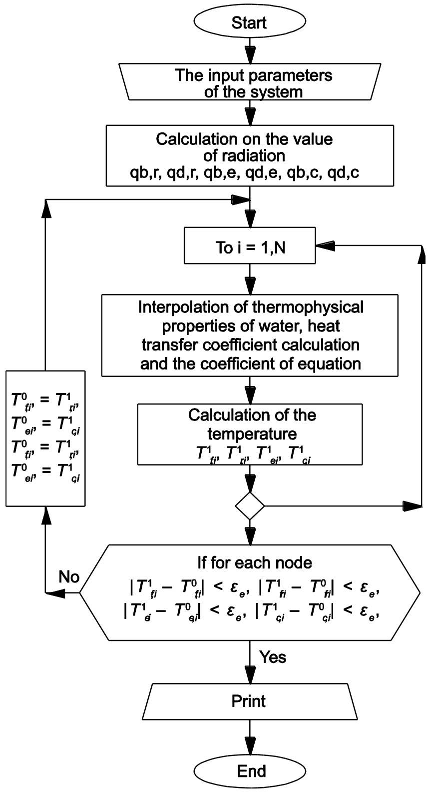

Figure 7. The algorithm of calculation

Discussion and conclusions

This paper provides numerical estimated changes of temperature in the direction of fluid flow for different flow rates, different solar radiation intensities, and different inlet fluid temperatures. The increase in fluid flow reduces output temperature, while the increase in solar radiation intensity and inlet water temperature increases output temperature of water. A prediction of the change of temperature of the P2CC module components in the flow direction is also given. The change in efficiency is predicted with the change in input temperature of water. With the increase in inlet water temperature, the current efficiency of the collector decreases. All of the predictions are given for a 1 m long module. In addition, a numerical estimation is also conducted for the change of the fluid temperature in the direction of the flow for a 6m long module connected in a row, where a higher output temperature is achieved compared to the 1 m long module.

Nomenclature

A – the area, [m^2]
A^* – area of cross section, [m^2]
A_c – area of aperture, [m^2]
A_r – area of tube receiver, [m^2]
G – gap between the receiver tube and the top reflector, [m]
H – height of the backing plate, [m]
h – heat transfer coefficient, [Wm^{-2} °C^{-1}]
I – incident radiation, [Wm^{-2}]

N – number of nodes, [–]
Nu – Nusselt number, [–]
P – loss correction coefficient of gaps, [–]
q – the heat flux based on a unit receiver area, [Wm^{-2}]
r – radius, [m]
Re – Reynolds number, [–]
$r_{e,0}$ – external radius of the transparent pipe, [m]
$r_{e,i}$ – internal radius of the transparent pipe, [m]
$r_{r,0}$ – external radius of the receiver, [m]
$r_{r,i}$ – internal radius of the receiver, [m]
T – temperature, [°C]
t – time, [s]
Δt – time interval, [s]
$U_{r/f}$ – total heat transfer coefficient, [Wm^{-2}K^{-1}]
W – width of aperture cover, [m]

Greek symbols

α – absorptivity, [–]
δ – thickness of the transparent cover, [m]
ε – emissivity, [–]
ε_e – specify the error between two consecutive iterations, [°C]
η – the instantaneous efficiency, [%]
θ_a – semi-angle reception, [°]
ρ^* – density, [kgm^{-3}]
ρ_m – reflectivity, [–]
τ – transsmitivity, [–]

Subscripts

a – environment
b – direct radiation
c – transparent cover, convection
d – diffuse radiation
e – transparent layer tube receiver
f – working fluid
i – ordinal number of the node
in – fluid input
m – mirror
out – fluid output
r – tube receiver – absorber
s – sky
tot – total radiation
z – axial direction

Superscripts

0 – old time instant
1 – new time instant

Acronyms

CPC – compound parabolic collector
P2CC – parabolic-and-circular collector
PTC – parabolic trough collector
STC – solar thermal collector

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