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Numerical heat transfer simulation of solar-geothermal hybrid source heat pump in Kazakhstan climates

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A numerical energy balance model has been proposed in this work for predicting the thermal performances of solar-geothermal hybrid source heat pumps in winter climates of Kazakhstan. The numerical simulation was performed for the year round continental climate conditions. The energy balance model has been developed based on first law of thermodynamics. The proposed heat pump configuration is working in solar mode during sunshine hours and geothermal model during off sunshine hours. Moreover, the system is operating in solar-geothermal hybrid model to meet the evaporator load during insufficient availability of solar and geothermal sources. The energy performance comparison between conventional geothermal source and solar-ground hybrid source heat pump configurations has been made. The influences of solar intensity, ambient temperature, heat pump operating temperatures are discussed.

Keywords: Solar-geothermal source hybrid heat pump; Heat transfer; Numerical simulation.

INTRODUCTION

Heat pump is an energy efficient device due to its capability to deliver more heat output than the work input [1]. The performance of the heat pump systems are improved using renewable energy sources such as, ambient, solar, geo-thermal and its hybrid forms [2]. The continental climate regions are facing with large fluctuations of annual ambient temperature during the day time in winter and drops below -20 ºC. The availability of solar radiation and length of sunshine is not sufficient during winter in the continental climatic conditions like, Kazakhstan, Russia, China, Ukraine, Uzbekistan and North America. Hence, it is necessary to integrate the solar and geothermal hybrid sources of energy with the heat pump systems to improve the performance of heat pump systems. Many research and developments have been progressed with solar-geothermal heat pump system in the continental climatic regions, which are summarized in earlier literature reviews [3-5].

From recent studies in the field of solar-geothermal hybrid source heat pumps, it can be noted the following papers [6-14]. In [6] an optimization method for the design and operation of a hybrid solar geothermal source heat pump was described. The system studied consists of solar thermal collectors, borehole heat exchanger, a heat pump, natural gas boiler and a stratified daily storage tank. In [7] 2000 sets of performance data collected from solar-assisted geothermal source heat pump systems that keep operating over 20 years were simulated. The thermal performance of a solar-geothermal source heat pump system operated in different dual heat source coupling modes were studied experimentally [8]. The average unit COP and collecting efficiency for the solar and geothermal source combination mode are 3.61 and 51.5%. The suitable collector area and geothermal source heat exchangers number are found to be 80 m^2 and 9, respectively. To eliminate the effect of geothermal thermal imbalance and minimizing system lifetime cost, paper [9] focuses on combining a geothermal source heat pump system with a solar thermal array. Three buildings were investigated for use with the solar-assisted geothermal source heat pump system, which had heating-to-cooling load ratios of 20.4:1, 8.6:1, and 1.2:1. Numerical modelling of transient soil temperature distribution for horizontal geothermal heat exchanger of geothermal source heat pump was presented in [10]. When the system is operated using the geothermal source heat pump at the end of the 10-year heating period, the

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average soil temperature in solution domain and soil temperature around the pipe are found to decrease by approximately 2.5 ºC, 0.7 ºC for Q_{max}=21 W/m, respectively. The decrease in these temperatures will increase if the heat drawn from the soil per unit pipe length increases. A solar photovoltaic/thermal-geothermal source heat pump was proposed in [11] to solve the problems of energy shortage and high energy-consuming buildings. According to the results photovoltaic/thermal-geothermal source heat pump system can reduce the temperature of photovoltaic/thermal modules as high as 10 ºC, and improve the efficiency of electricity production by 25%, simultaneously with average COP equal to 3. Combined operation of solar thermal and geothermal source heat pump was applied in [12]. According to the results the assistant of solar thermal, heat pump could work temporarily, with a COP of 5.2 in the day. A hybrid solar geothermal source heat pump system was presented in [13]. 32% of the electrical energy consumption in the hybrid solar geothermal source heat pump system could be saved if the load circulation pump was turned off when no fan-coil was running, rather than always keeping it running at all times. Exergy analysis of a hybrid ground-source heat pump system was conducted in [14]. The true exergy performance of the system in heating mode (~ 30%) is twice as high as for cooling mode (~15%), while the natural exergy performance is considerably better in cooling mode (~26% to~3%).

The cited literature confirmed that, many research and development initiatives have been already progresses on solar-geothermal heat pump systems. However, there is no specific work has been reported on Kazakhstan climatic conditions. Hence, a numerical heat transfer model has been developed for solar-geothermal hybrid heat pump system for the climatic conditions of Almaty in Kazakhstan. In addition, the influence of solar radiation, ambient temperature and heat pump operating temperatures (condenser and evaporator) were discussed.

DESCRIPTION OF THE HYBRID SYSTEM

The schematic illustration of solar-geothermal source hybrid heat pump (SGSHHP) used for space heating and domestic hot water applications is shown in Fig. 1. The SGSHHP consists three circuits namely, solar and geothermal hybrid source circuit I, heat pump circuit II and water heating III. The heat pump circuit II consists of all the basic components such as, compressor 6, condenser 3’, expansion valve 7 and plate type evaporator 3. In addition, the accessories such as, sight glass, liquid receiver, filter-drier have been used to enhance the performance of the system. The pressure and temperature controls have been used to control the system operating parameters. The solar-geothermal circuit I consists of two solar thermal collectors 1, geothermal heat exchangers 4, two centrifugal pumps 5, 5’ and solenoid flow control valves 2, 2’, 2’’. The hot water circuit III consists of pump 5’, hot water storage tank 8, plate type heat exchanger 3’, room radiators 9 and hot water utilization 9’.

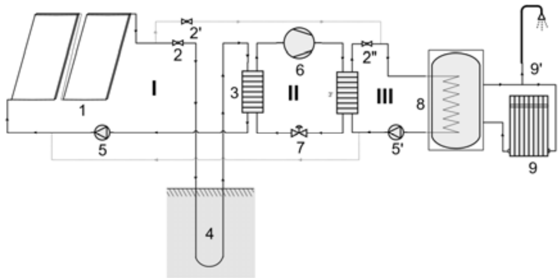

Fig.1. Solar-geothermal hybrid source heat pump

The system is operating in following two modes: (i) solar thermal mode and (ii) solar-geothermal hybrid heat pump mode. In solar thermal mode, the harvested solar energy through the solar collectors is directly transferred and stored in hot water storage tank for domestic water heating and space heating applications using ethylene glycol. In the solar-geothermal hybrid heat pump mode, the harvested solar energy is enhanced with the geothermal source heat exchanger. In heat pump mode, the ethylene glycol is circulated through the solar collectors for harvesting the solar energy and the circulated through the geothermal heat exchangers for extracting the geothermal energy. The combined solar-geothermal hybrid source is used in the evaporator of a heat pump. The compression heat pump absorbs the solar-geothermal heat and enhance to the room space heating requirements.

During peak sunshine hours, the solar energy is harvested by solar collectors and transferred directly to the storage tank. During lean and off sunshine hours, the hybrid forms solar energy (harvested in the solar collectors) and geothermal heat energy (energy extracted by geothermal heat exchangers) are utilized in the evaporator of a heat pump for further enhancement. The flow of ethylene glycol through the solar collectors and geothermal heat exchangers are controlled by a solenoid direction control valves. In the daytime, when solar radiation is available, the excess heat obtained by the working fluid from solar collectors heats the soil around the ground source heat exchanger, thereby increasing the thermal potential of the soil. At night, in the absence of solar radiation, the working fluid after the evaporator 3 enters the ground source heat exchanger through the solar collectors, where the heat loss due to good insulation of the absorber is insignificant.

MATHEMATICAL MODEL

The mathematical model contains three basic equations: two-dimensional heat transfer equation for a solar thermal collector (1), which includes a main panel and a copper coil; energy conservation equations for the flow of a heat transfer fluid (antifreeze) in a solar collector tube (8) and in a vertical type geothermal source heat exchanger (9).

The two-dimensional heat transfer equation of a thermal collector. The thermal collector consists of two elements: copper coil, copper plate (absorber). By neglecting the thermal resistance at the contact surface, the temperature of these two elements is assumed to be the same on any reference volume. The equation of thermal conductivity of each control volume with a concentrated mass m_c, specific heat C_c and thermal conductivity λ_c is written in the form:

$$\begin{aligned} m_c C_c \frac{\partial T_c}{\partial t} &= G(\tau\beta)_c A_c + \alpha_{a-c} A_c (T_a - T_c) + \\ &+ \alpha_{r,a-c} A_c \left(T_{sky} - T_c\right) + \alpha_f A_f \left(T_f - T_c\right) + \\ &+ A_c \frac{T_a - T_c}{R_b} + \lambda_{c,y} l_{c,y} A_c \frac{\partial^2 T_c}{\partial y^2} + \lambda_{c,z} l_{c,z} A_c \frac{\partial^2 T_c}{\partial z^2} \end{aligned} \tag{1}$$

where G is solar radiation; $(\tau\beta)_c$ - effective absorptivity of the thermal collector; α_{a-c} and $\alpha_{r,a-c}$ - coefficients of convective and radiative heat transfer between the collector and the environment; R_b – thermal resistance between the back side of the heat collector and the environment; $l_{c,y}$ and $l_{c,z}$ - effective thickness along the Y and Z directions, respectively.

The above parameters are given by:

$$(\tau\beta)_c = \frac{\tau_c \beta_c}{1 - (1 - \beta_c) \cdot r} \tag{2}$$

$$\alpha_{a-c} = 2.8 + 3.0 \cdot u_{wind} \tag{3}$$

$$\alpha_{r,a-c} = \varepsilon_c \sigma \left(T_c^2 + T_{sky}^2\right)\left(T_c + T_{sky}\right) \tag{4}$$

where $(\tau\beta)_c$ - absorptivity of the solar collector absorber; τ_c and r - transmissivity and reflective capacity of the absorber; u_{wind} - wind speed; T_{sky} - sky temperature, which defined as

$$T_{sky} = 0.0552 T_a^{1.5} \tag{5}$$

$$\tau_c = \frac{1 - r}{1 + r} \tag{6}$$

$$r = \frac{\sin^2(\theta_1 - \theta_2)}{\sin^2(\theta_1 + \theta_2)} + \frac{\tan^2(\theta_1 - \theta_2)}{\tan^2(\theta_1 + \theta_2)} \tag{7}$$

where θ_1 and θ_2 - the angles of incidence and refraction of sunlight.

Equations of energy conservation for working fluid (antifreeze) flow. The process of transfer of the working fluid in the coil of the evaporator and in the tube of the geothermal heat exchanger can be described by a mathematical system of partial differential equations, which is based on the laws of conservation of energy of the fluid. The following assumptions were made:

- working fluid flow is one-dimensional and incompressible;
- the change of the kinetic and potential energy in the energy equation is neglected.

With the above assumptions, a simplified model of the flow of working fluid can be written as:

Energy conservation equation of working fluid

$$m_f C_f \frac{\partial T_f}{\partial t} + m_f C_f u \frac{\partial T_f}{\partial z} = A_f \alpha_f \left(T_c - T_f\right) \tag{8}$$

$$m_f C_f \frac{\partial T_f}{\partial t} + m_f C_f u \frac{\partial T_f}{\partial z} = A_f \frac{\left(T_g - T_f\right)}{R_{g-f}} \tag{9}$$

where m_f – fluid mass, C_f – specific heat, α_f – heat transfer coefficient in the solar collector tube, $R_{g\text{-}f}$ – thermal resistance between the working fluid and the environment (ground).

$$R_{g-f} = \frac{l_{U-pipe}}{\lambda_{U-pipe}} \qquad (10)$$

where $l_{U\text{-pipe}}$ – the thickness of pipe wall, $\lambda_{U\text{-pipe}}$ – thermal conductivity of pipe wall. The working fluid and ground temperature change along the U-pipe heat exchanger. Instead of separately calculating the heat balance equation for the pipe wall, the influence of the pipe wall on the heat exchange between the soil and the working fluid was taken into account through the thermal resistance coefficient (10).

METHOD OF SOLUTION

In order to solve the above system of equations (1)-(10) it is necessary to set the initial temperature distribution of the heat collectors. In addition, it is necessary to set the input data for solar insolation, ambient temperature, physical properties of used materials and working fluid flow. In order to investigate the dynamic behavior of the thermal evaporator and working fluid a computer program in C++ has been developed.

The program begins with setting the necessary initial conditions for the physical parameters (Tab.1).

Table 1. System parameters

Density of absorber	ρ_c	8920 kg/m^3
Heat capacity of absorber	C_c	385 J/kg·K
Thermal conductivity of absorber	λ_c	401 W/m·K
Emissivity coefficient of absorber	ε_c	0.05
Absorptivity coefficient of absorber	β_c	0.95
Stefan-Boltzman coefficient	σ	$5.67\cdot10^{-8}$ W/m^2·K^4
Heat transfer coefficient of working fluid	α_f	5.6 W/m^2·K
Density of working fluid	ρ_f	1055 kg/m^3
Heat capacity of working fluid	C_f	3620 J/kg·K
Thermal conductivity of U-pipe material	$\lambda_{U\text{-pipe}}$	2.5 W/m·K

As the initial conditions for the solar collector temperature T_c, working fluid temperature T_f the ambient temperature was taken. On the ground surface, the temperature of soil is equal to ambient temperature. From the ground surface, the temperature of the soil varies up to 10 meters depth depending on the ambient temperature. After 10 meters of depth, the temperature of the ground is assumed equal to 10 ºC.

Then the subprograms for calculating of solar collector absorber and working fluid temperatures distribution, for heat transfer coefficients and physical properties are called. This process is repeated until iteration is established.

RESULTS AND DISCUSSION

Using the above algorithm, the temperature of the heat collector and working fluid was calculated for the climatic conditions of Almaty, Kazakhstan. In the calculations, the corresponding data on solar radiation and ambient temperature were taken into account.

In Fig.2 monthly average solar radiation and ambient temperature are shown. From the data shown in Fig.2, it can observe that the average ambient temperature in Almaty is reached 273.15 K in the month of January and has its maximum value of 303.15 K in July (the dotted line in Fig.2). The average solar irradiance varies from 101.55 W/m^2 in December to 544.35 W/m^2 in June (the solid line in Fig.2).

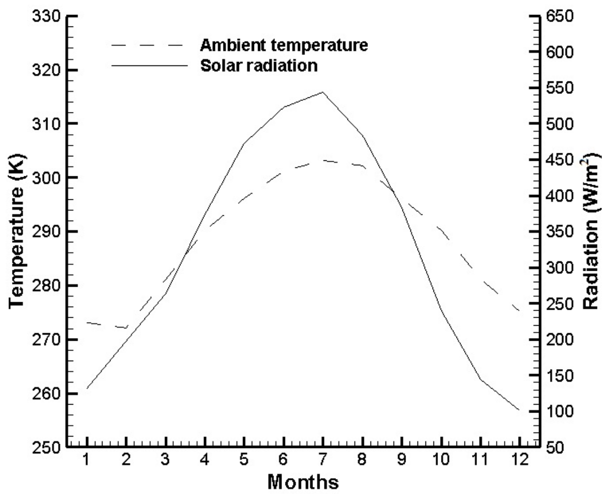

Fig.2. Seasonal solar radiation and ambient temperature data

The authors of the research have installed Vantage Pro2 Plus 6162C weather station, which provide meteorological data (wind speed, wind direction, air temperature, humidity, barometric pressure, rainfall, rainfall intensity, UV, solar radiation). In Fig.3 the daily solar radiation and ambient temperature variation for different days are shown.

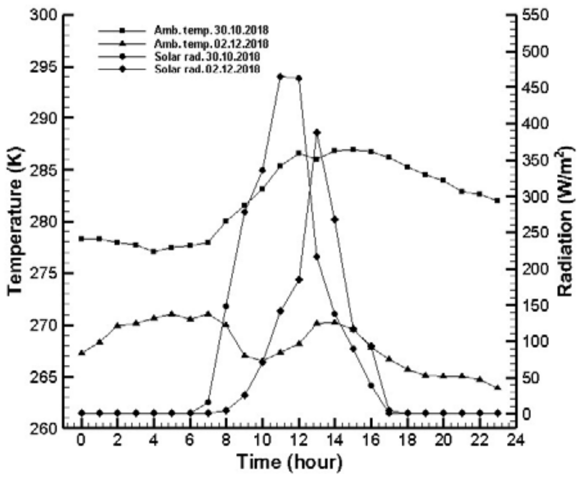

Fig.3. Daily solar radiation and ambient temperature data

Fig.4 shows the temperature variation of the solar collector as a function of the months according to equation (1).

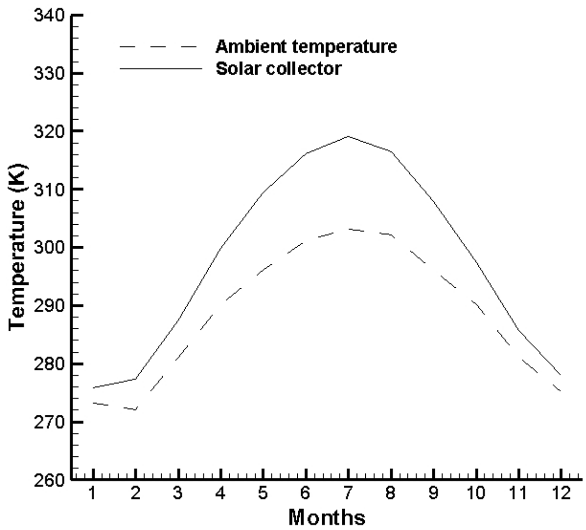

Fig.4. Seasonal temperature variation in the solar collector

From the numerical results presented in Fig.4 the maximum temperature of the solar thermal collector is achieved in the summer season. The behavior of temperature profiles is similar to the behavior of solar radiation, shown in Fig.2. The figure shows that the temperature of the thermal collector increases in summer, reaching a maximum in July 319.06 K. The difference between the temperatures of the absorber and the environment is 2.77 K in the month of January and 15.91 K in the month of June. This effect is due to the different values of solar radiation in different season.

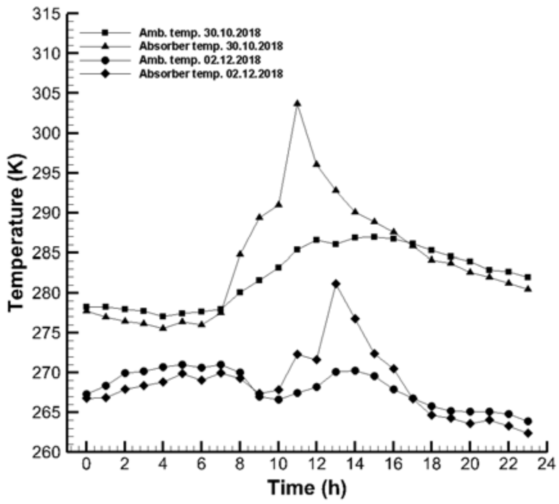

Fig.5. Daily temperature variation in the solar collector

According to the Fig.5 for the daily absorber plate temperature variation same behavior as in seasonal case is observed. During nighttime, without solar radiation and with the cold sky temperature, absorber plate temperature is 1-2% lower, than ambient temperature. This is an indicator that, in the absence of solar radiation, heat losses from the solar collector are small.

Fig.6 shows the daily temperature variation of the working fluid at the outlet of the solar collector tube and the geothermal heat exchanger depending on the hours according to the system of equations (8) - (9) for 30.10.2018.

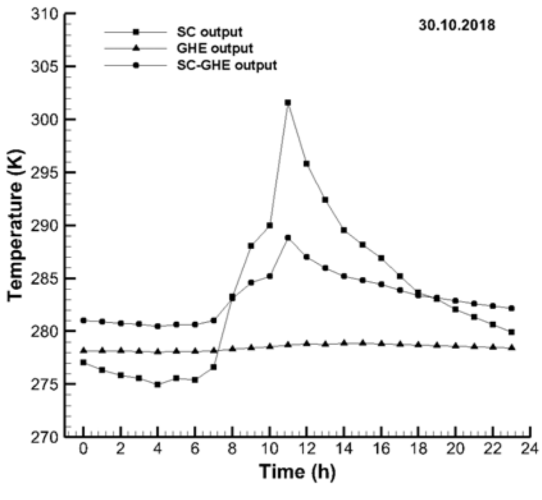

Fig.6. The temperature of working fluid in solar and geothermal collectors (30.10.2018)

Fig.7 shows the same daily temperature variation for 02.12.2018.

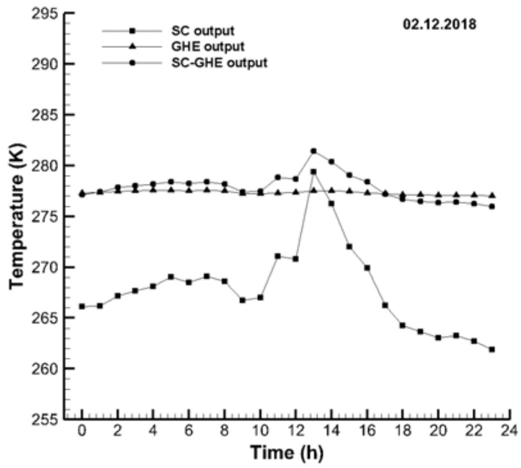

Fig.7. The temperature of working fluid in solar and geothermal collectors (02.12.2018)

From Fig.6-7 it can be seen that the working fluid temperature at the outlet of the solar collector depends on the ambient temperature and solar radiation. At the same time, the temperature of the working fluid at the outlet of the geothermal source heat exchanger is independent of the ambient temperature. During the daytime with the consistent use of a ground collector and solar collector, excessive solar energy is transferred to heat the ground, and in the absence of solar energy (nighttime), this heat from the ground is used for the heat pump evaporator. To obtain high heat pump COP and +50 - +60 ºC in heating circuit (space heating and DHW) with a high volumetric compressor efficiency, a positive temperature of +5 - +10 ºC is sufficient for the refrigerant evaporation. Therefore, the combined use of two low-potential heat sources – solar and ground collectors is recommended for a heat pump evaporator in continental climatic conditions. In the proposed SGSHHP configuration of the heat pump, the ground heat potential is constantly maintained by solar collectors.

Also, according to the SGSHHP configuration the system can operate in energy efficient mode, where with sufficient solar radiation, the heat pump circuit can be switched off, and system operate as direct solar water heating.

Two-dimensional results of the temperature distribution in the solar thermal collector absorber plate were also presented in Fig.8. The grid size is 50x100 (Fig.8a).

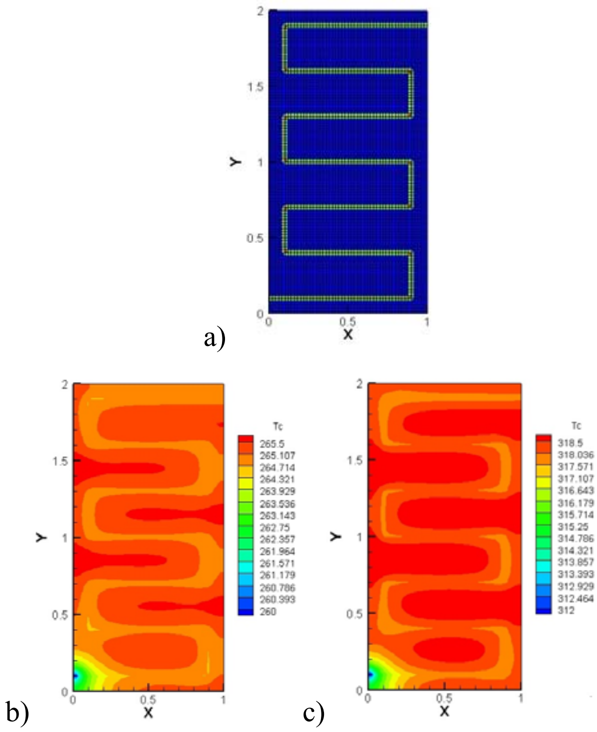

Fig.8. Temperature distribution in the solar thermal collector absorber plate (a – numerical grid, b –January month, c – July month)

CONCLUSIONS

Numerical estimation of hybrid solar-geothermal source heat pump thermal performance in terms of solar thermal collector and geothermal source heat exchanger operation for meteorological conditions of Almaty, Kazakhstan has been conducted. A mathematical model, numerical algorithm and computer program for calculating of solar collector and geothermal source heat exchanger performance parameters has been developed. Numerical calculations showed the efficiency of using a hybrid solar-geothermal heat source for heat pump operation.

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NOMENCLATURE

A	-	surface area, m^2
C	-	specific heat, J/(kg K)
G	-	solar irradiance, W/m^2
l	-	effective thickness, m
m	-	mass, kg
R	-	thermal resistance, K/W
r	-	reflectivity, –
T	-	temperature, K
t	-	time, s
u	-	flow velocity, m/s

Subscripts

a	-	air; ambient
b	-	back
c	-	thermal collector
r	-	radiation
f	-	working fluid
g	-	ground

Greek letters

α	-	heat transfer coefficient, $W/(m^2\ K)$
β	-	absorptivity, –
ε	-	emissivity, –
θ	-	angle, degree
λ	-	thermal conductivity, W/(m K)
ρ	-	average density, kg/m^3
σ	-	Stefan–Boltzman constant, $W/(m^2\ K^4)$
(τβ)	-	effective absorptivity

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