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The theoretical and experimental research on thermal performance of solar air collector with finned absorber

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Abstract

The theoretical analysis and experimental research on the thermal performance of the solar air collector with finned absorber are made in this paper. The effects of setting angle, medium flow and air inlet mode on solar air collector performance are analyzed , and the theory calculation model of it is confirmed by experimentation . The results show that the error between the theory calculation model and the experiment test maintains at around 9%. The results of calculation and analysis on thermal performance of this collector through theory calculation model indicate that setting angle has no influence on the thermal efficiency, while medium flow has a positive relation with it. In addition, heat-collecting efficiency can be improved by the negative pressure in the collector, but it is dropped along with the pressure going up. In sum, the fin plat solar air collector can be widely applied because it can be the medium-low temperature heat source for the solar drying systems.

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Keywords: solar air collector;thermal character; heat-collecting efficiency; theory calculation model

1. Introductions

It is very important to analyze the performance of solar energy flat plate air collector and the suitable condition for the development and improvement of .solar energy collector technology. Many scholars, domestic or abroad, have done a lot of research on . solar flat plate air heater. A hybrid PV/T solar flat plate air heater system with dual fin type has been designed by M.Y. Othman [1]. There were a few errors between the theoretical and experimental values based on. experiments and theoretical model made by him. He confirmed that dual fin flat plate solar air

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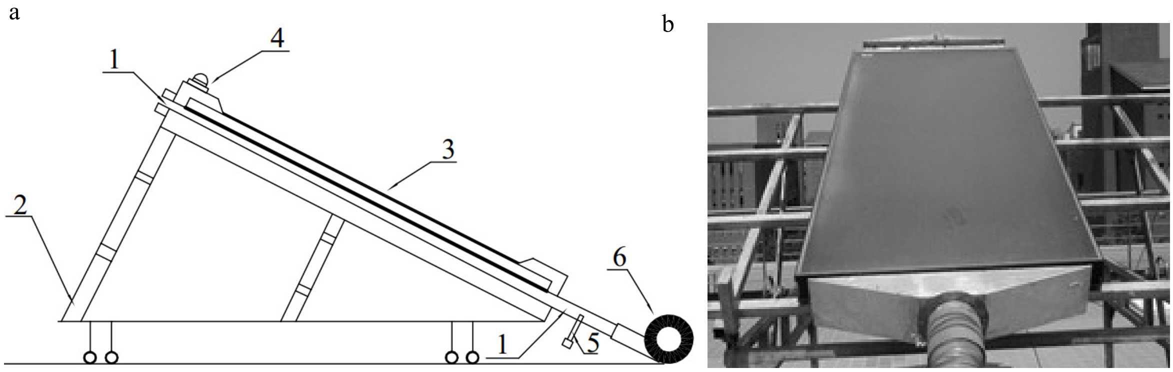
heater with PV/T solar cell hybrid system can improve the overall thermal performance. X.D. Yuan [2] built the theoretical models of v-type solar air collector which could easily get the efficiency of the collector by using the finite difference method. The error was within 5%, when compared with the experimental values. The thermal performance of solar flat plate solar air heater with a fin-type dual-channel has been researched by A. Fudholi [3]. Comparing the theoretical values based on his theoretical model with the experimental values, the error was within 7%. J. Ji has conducted researches on the light-heat property of a new type double effect solar flat plate collector. The results showed that the light-heat conversion efficiency is 40% higher than the conventional collector under the air heater mode and it reaches up to 49.7% under the hot water operating mode.

Through the analysis above, we can see that theoretical model of . thermal performance of flat plate solar air heater with single channel are rarely researched by scholars. In this paper, the theoretical mode is built and the thermal performance of the collector is determined by comparing the theoretical values with experimental values.

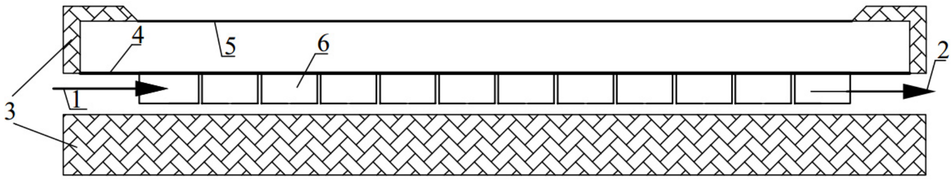
2. Brief and build of the experimental platforms

As shown in Fig. 1, a fin-plate solar air collector with the characteristic of the whole plate heat, high heat and small heat loss are provided in this experiment. The absorption rate of glass cover on solar radiation is up to 95% and launch rates is as low as 5% . Its total area is 2.00m^2 and contour day lighting area is 1.90m^2. The collector is made up of glass cover, absorber plate, insulating layer and bottom plate. The overall framework is 2000 mm×1000 mm×120 mm stainless steel, and the thermal insulation material of insulating layer is polyurethane of 50mm thickness. Glass cover is tempered glass, whose internal structure is tetrahedral, which aims to increase the area of accepting light and heat absorption then increase the efficiency of the collector.



1.Temperature sensor 2. Bracket 3.Flat plate collector 4. Total irradiance meter 5. Hot wire anemometer 6. Three-phase asynchronous air blower
Fig. 1. (a) Schematic of fin-and-flat plate solar air collector; (b) Experiment platform physical map of fin-and-flat plate solar air collector.

The flow process of working medium in the air collector. can be seen in fig. 2, which provides the sectional drawing of flat plate air collector with fin. The outside air is sent in from the collector entrance by the three-phase asynchronous air blower, then the working medium transfer heat through the absorber plate and fin. Finally, the gas with high temperature flows out from the export, which is advantageous for the actual industry and agriculture heat utilization.



1. Air-inlet 2. Air-outlet 3. Insulating layer 4.Absorber plate 5.Glass cover-plate 6.Fins
Fig. 2 The internal structure diagram of collectors.

3. Mathematical models

3.1 Simplified assumptions of model

In Fig. 3, a and b respectively shows the heat transfer coefficient of fin-and-flat plate solar air collector and the thermal resistance network structure diagram of each part within the collector. The collector of energy balance equations are listed according to the parameters. In order to solve it easily, the fin-and-flat plate solar air collector model is simplified in this paper on the premise of preserving accuracy and precision, making the follow assumptions [2,3]:

1）The heat transfer process is in a steady state.

2）Ignoring the heat transfer of the flank and the thermal radiation outward from the absorber plate.

3）Ignoring the convection heat resistance of glass cover-plate to absorber plate.

4）The heat conductivity coefficient of fins and absorber plate are constant.

5）The junction temperature of fin and absorber plate is equal to the temperature of fin at the bottom and top, namely $T_p = T_{top} = T_{bottom} = T_{fin}$.

a

b

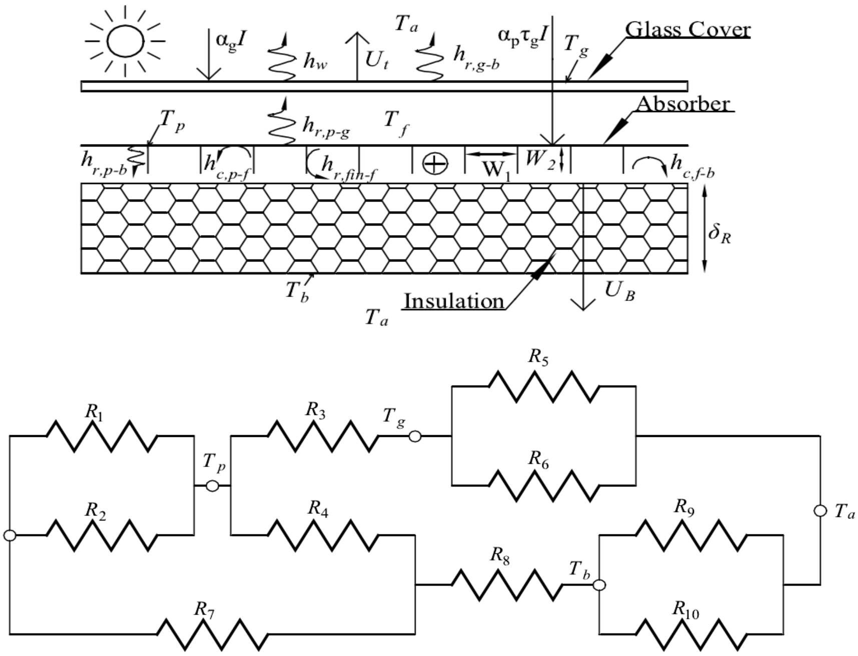


Fig. 3 (a) The heat transfer coefficient schematic diagram;(b) Thermal resistance network diagram of flat collector with fins.

Where: R_1 is the convection heat resistance of heat-absorbing surface to the working medium, $R_1=1/h_{c,p-f}$; R_2 is the convection heat resistance of fins to the working medium, $R_2=1/h_{r,fin-f}$; R_3 is the radiant heat resistance of absorber plate to the glass cover-plate, $R_3=1/h_{r,p-g}$; R_4 is the radiant heat resistance of absorber plate to the bottom plate, $R_4=1/h_{r,p-b}$; R_5 is the radiant heat resistance of glass cover-plate to the ambient, $R_5=1/h_{r,g-a}$; R_6 is the convection heat resistance of the glass cover to the ambient, $R_6=1/h_w$; R_7 is the convection heat resistance of working medium to the backboard of collector, $R_7=1/h_{c,f-b}$; R_8 is on the back surface to the lower surface of the convection heat resistance of back board from the upper surface to the lower surface; R_9、R_{10} are the radiant heat resistance and convection heat resistance of the back board to the ambient, $U_B=1/h_w+\delta_R/\kappa_R$.

Because of the linear relationship between the length of collector and the temperature of the internal working medium, it can be assumed that the internal working medium temperature can take the average value of the inlet temperature and outlet temperature of collector, namely [5]:

$$T_f=(T_{fi}+T_{fo})/2$$

3.2 The establishment of the equilibrium equation of flat plate collector model equation [6-8]

For glass cover plate:

$$\alpha_g I + h_{r,p-g}\left(T_p - T_g\right) = \left(h_w + h_{r,g-a}\right)\left(T_g - T_a\right) \tag{1}$$

For absorber plate:

$$\alpha_p \tau_g I = h_{r,p-g}\left(T_p - T_g\right) + h_{r,p-b}\left(T_p - T_b\right) + h_{c,p-f}\eta_p\left(T_p - T_f\right) \tag{2}$$

For back board:

$$h_{r,p-b}\left(T_p - T_b\right) + h_{c,f-b}\left(T_f - T_b\right) = U_B\left(T_b - T_a\right) \tag{3}$$

For working medium (air):

$$h_{c,p-f}\eta_p\left(T_p - T_f\right) = 2C_P\dot{m}\left(T_f - T_{fi}\right) \tag{4}$$

Simultaneous(1)-(4)equation can be obtained a 4×4 matrix[9]: [A]=[T][S]

$$\begin{bmatrix} S_{11} & S_{12} & 0 & 0 \\ S_{21} & S_{22} & S_{23} & S_{24} \\ S_{31} & 0 & S_{33} & S_{34} \\ S_{41} & 0 & S_{43} & S_{44} \end{bmatrix} \begin{bmatrix} T_p \\ T_g \\ T_f \\ T_b \end{bmatrix} = \begin{bmatrix} S_1 \\ S_2 \\ S_3 \\ S_4 \end{bmatrix} \tag{5}$$

Equation (5) can be solved by inverse matrix [T]=[A]-1[S] for the four unknown Tp，Tg，Tf，Tb ，hereinto：$S_{11}=h_{r,p-b}=-S_{22}$， $S_{12}=-h_{r,p-g}-h_w-h_{r,g-a}$， $S_{21}=h_{r,p-g}+h_{r,p-b}+h_{c,p-f}\cdot\eta_p$， $S_{23}=-h_{r,p-b}$， $S_{24}=-h_{c,p-f}\cdot\eta_p$， $S_{31}=h_{r,p-b}$， $S_{33}=h_{c,f-b}=S_{44}$， $S_{34}=-h_{r,p-b}-h_{c,f-b}-1/(1/h_w+\delta_R/\kappa_R)$ ， $S_{41}=h_{c,p-f}\cdot\eta_p$ ， $S_{43}=-2C_p\ \dot{m}\ -h_{c,f-b}-h_{c,p-f}\cdot\eta_p$ ， $S_1=-\alpha_g I-(h_w+h_{r,g-a})T_a$ ， $S_2=\alpha\tau I$ ， $S_3=-[1/(1/h_w+\delta_R/\kappa_R)]\cdot T_a$， $S_4=-2C_p\dot{m}\,T_{fi}$.

3.3 Determination of the related heat transfer coefficient in the model

The relative unknowns T_p, T_g, T_f, T_b in [T] can be worked out by using the iterative method based on MATLAB software program, then the related heat transfer coefficient are required to be determined at first:

Convection heat transfer coefficient of the glass cover plate [10]:

$$H_w=2.8+3.3\upsilon_w \tag{6}$$

Radiation heat transfer coefficient of glass cover plate to the sky:

$$h_{r,g-a} = \frac{\sigma\varepsilon_g\left(T_g + T_s\right)\left(T_g^2 + T_s^2\right)\left(T_g - T_s\right)}{T_g - T_a} \tag{7}$$

where, T_s is the sky temperature $$T_{sky} = 0.0552 T_a^{1.5} \tag{8}$$

The radiation heat transfer coefficient of absorber plate to the glass cover plate:

$$h_{r,\mathrm{p-g}} = \frac{\sigma\left(T_p^2 + T_g^2\right)\left(T_p + T_g\right)}{1/\varepsilon_p + 1/\varepsilon_g - 1} \tag{9}$$

The radiation heat transfer coefficient of absorber plate to back board:

$$h_{r,p-b} = \frac{\sigma\left(T_p^2 + T_b^2\right)\left(T_p + T_b\right)}{1/\varepsilon_p + 1/\varepsilon_b - 1} \tag{10}$$

The efficiency of absorber plate with fins in fin type air collector [5]:

$$\eta_p = 1 - \frac{A_{fin}}{A_{ab}}\left(1 - \eta_{fin}\right) \tag{11}$$

where, $\eta_{fin} = \dfrac{\tanh\left(mW_2\right)}{mW_2}$, $m = \left(\dfrac{h_{c,p-f}}{\kappa\sigma}\right)^{1/2}$;

The convection heat transfer coefficient of working medium and absorber plate, and the convection heat transfer coefficient of working medium and baseboard [11,12]:

$$h_{c,p-f} = h_{c,f-b} = Nu\frac{\kappa}{D} \tag{12}$$

Where：$D=WH_g/(W+H_g)$；

If the fluid changes in the channel are the laminar flow, the related coefficient of the model [13]:

$$Nu = 0.664\,\mathrm{Pr}^{1/3}\,\mathrm{Re}^{1/2}, \quad (\mathrm{Re} \le 5\times10^5, \mathrm{Pr} \ge 0.5) \tag{13}$$

If the fluid changes are turbulent in the channel [14]:

$$Nu = 0.033\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.33}, \quad (5\times10^5 < \mathrm{Re} < 10^8, 0.6 < \mathrm{Pr} < 60) \tag{14}$$

When the temperature of the working medium is in the range of 280K~470K, some empirical formula can be estimated by using some parameters, so the main design parameters of the whole fin-and-flat-plate air collector and some parameters in the formula are as follows [15]:
$A_P \approx 1.91m^2$, $A_{fin} \approx 1.75m^2$, T_a=25°C, W_2=0.018m, W=0.965m, H_g=0.013m, δ_R=0.03m, $C_p \approx 1000J/(kg\cdot K)$, I=800W/m^2, υ_w=1m/s, κ_R=0.4$W/m\cdot°C$, $\dot{m}$=0.05kg/s, $\dot{m}$=0.05kg/s, $\sigma=5.57\times10^{-8}W/(m^2\cdot K^4)$, ε_g=0.9, α_g=0.05, τ_g=0.92, ε_p=0.94, ε_p=0.95, α_b=0.95, $\kappa=(0.0015215+0.097459T-3.3322\times10^{-5}T^2)\times10^{-3}$.

3.4 The solving method of model

The heat transfer coefficient of flat plate air collector with fins is the function of temperature and cannot be solved directly. It is worked out by using the iterative method based on Matlab software programming in this paper [16] :

1）Input the structural parameters, and enter the parameters and environmental conditions.

2）An initial temperature [T] which can calculate each heat transfer coefficient for absorber plate, glass cover plate, back board and internal working medium, is demonstrated as the correlation coefficient in matrix [A].

3）A new matrix [T'] can be calculated by matrix [A], the initial temperature given by the astringency of calculated value of temperature is a criterion. If it's convergent, the program stops and the outlet temperature T_{fo} of collector is obtained. If it is not convergent, the calculated value [T'] will continue to iterate as the new investitive initial value until the error is less than 0.01℃, and at which point the outlet temperature T_{fo} can be obtained.

4) The efficiency of collector can be calculated under the available type:

$$\eta = \frac{C_p \dot{m}\left(T_{fo} - T_{fi}\right)}{A_{ab} I} \quad (15)$$

4. The analysis of experimental results

The experiment is taken in Kunming • China, where the approximate latitude is 25 degrees, the experimental conditions are maintained in a (quasi) steady-state, and the irradiance is at 800 ± 50 W/m^2 or less, the float of inlet temperature is within 1° C. This experiment focuses on the three important parameters impacting on the performance of the collector from analog computation and comparative analysis. The experiments are detailed as follows:

4.1 The impact of installation angle of flat plate collector on its performance

The efficiency and the import-export temperature difference can be seen in fig. 4(a). The import - export temperature difference basically keeps a linear relationship with the collector efficiency. The installation angle has little effects on the efficiency of the collector when it varies from 5 ° to 50 °.

It can be seen that the change of the collector inclination angle has a little impact on the efficiency of collector from Fig. 4(a), the difference between the maximum value and the minimum value is only 3%, the maximum efficiency appears in 25°, which is close to the local latitude value; the difference value of import and export temperature between the maximum and the minimum is only 1.6℃. The error of the theoretical values and experimental values of collector efficiency is within 3.5% which is within the range of permissible errors.

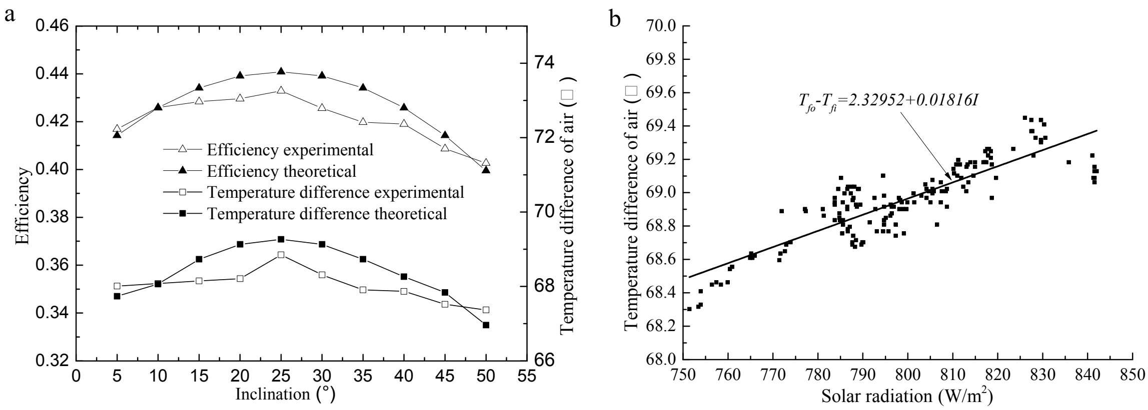


Fig. 4 (a) The value of import and export temperature difference of collector efficiency in different inclination; (b) The value of import and export temperature difference of collector under a certain inclination.

Fig. 4(b) shows the relationship between the import-export temperature difference and the solar irradiance of 800 ± 50W/m^2 when the installation angle is 25 °. The import-export temperature difference increases with the increasing level of irradiance. according to the fitting experimental data: T_{fo}-T_{fi}=2.32952+0.01816I, the fitting coefficient is 0.75324. It presents a linear relationship.

4.2 The impact of working medium flow on the performance of collector

Nine different experiments are carried out in the irradiance of 800W/m^2, controlling the mass flow of working medium in 0.01kg/s-0.09kg/s in the collector entrance. The experimental results are shown in Fig. 5, showing that the collector efficiency increases with the increasing of the air flow, but the import and export temperature difference decreases with it. As is shown in Fig. 4(a), the collector efficiency increases gradually from 41.4% to 78.6%, but the import and export temperature difference falls from 63.2℃ to 13.3℃in irradiance of 800W/m^2, when the air mass flow changes from 0.01kg/s to 0.09kg/s. The increase of air mass flow make the import and export temperature difference drop slowly, so the collector efficiency is improved. The collector efficiency increases slowly with the increasing of the working substance mass flow, because the reason is that the air speed of flow has a large impact on the air and the convective heat transfer coefficient of absorber plate when the air flow , and conversely the air flow speed has small impact on them when the speed is high. The difference of collector average efficiency is 4% through analysis of theoretical values and experimental values, in which the efficiency of maximum error is 8.3%, the maximum error for imports and exports temperature difference is 8.1%.

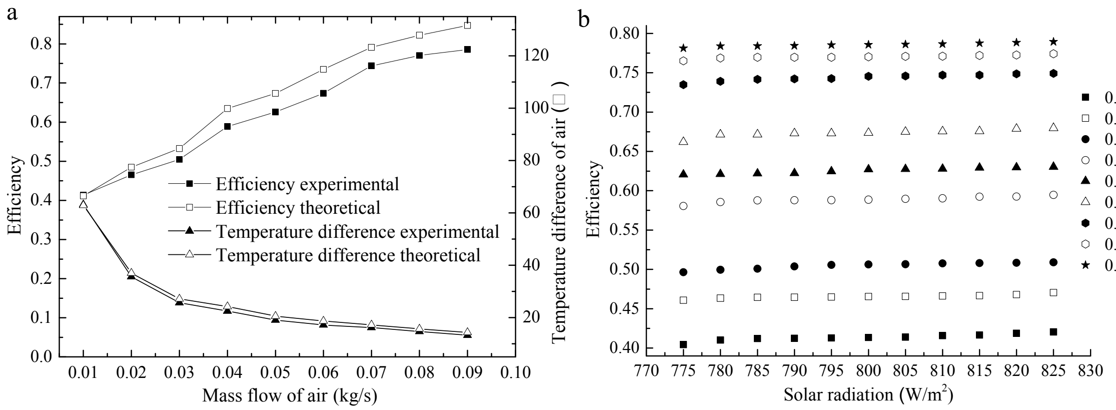


Fig. 5 (a) The impact of air mass flow on the collector efficiency and temperature difference; (b) The impact of air mass flow and irradiance on the collector efficiency.

Fig. 5(b) shows that when the irradiance change is small, the efficiency difference of collector is also small under identical mass flow. The average efficiency of the collector is given by Table 1 under different irradiance and identical working medium flow, when irradiance varies from 775W/m^2 to 825W/m^2. Due to small irradiance change, collector efficiency increases only by 1.4% from 61.1% to 62.4%. The average efficiency of collector is given by Table 2 under different mass-flow and the same irradiance, when the working medium mass flow varies from 0.01kg/s to 0.09kg/s. The collector efficiency increases by 37.5% from 41.1% to 78.6%, which demonstrates an obvious effect. By analysis of the above data, it can be known that working medium mass flow has a greater impact on the collectors.

Table 1 Average efficiency under the different irradiance and the same working medium mass flow.

Irradiance (W/m^2)	775	780	785	790	795	800	805	810	815	820	825
Average efficiency	0.611	0.615	0.617	0.617	0.618	0.619	0.620	0.620	0.621	0.622	0.624

Table 2 Average efficiency under the different medium mass flow and the same irradiance.

Air mass flow（kg/s）	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
Average efficiency	0.414	0.466	0.505	0.589	0.626	0.674	0.744	0.770	0.786

4.3 The impact of internal working medium pressure of collector on its thermal performance

The internal air pressure of collectors is tested by using two ways of placing the fan in experiment, one is above the collector, another is at the bottom of the collector. When the wind machine is placed above the collector to exhausts outward, the internal working medium of the collector will be in a negative pressure. When the wind machine is placed below to blow inward, the internal working medium will be in a normal state. Neither of the two ways change the flow direction of working medium, but they change its internal pressure status of working medium.

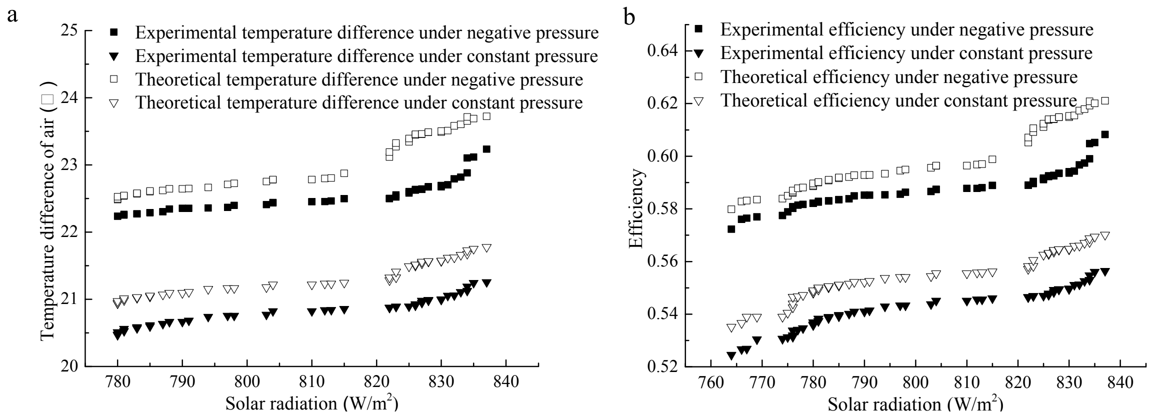


Fig. 6 (a) Import and export temperature difference of internal working medium of collector under different conditions ;(b) The efficiency of internal working medium of collector under different conditions.

Fig. 6 shows that the efficiency of the internal working medium operating under negative pressure is higher than it under normal pressure. The reason is that the gas under negative pressure can maximally use the atmospheric pressure to increase the heat transfer between the working medium and the absorber plate and fin, thereby enhancing the efficiency of the collector.

The experiments are carried out in the condition that the air flow of 0.04kg/s and the irradiance of 800±50W/m^2. From Fig. 6(a), it is observed that the import and export temperature difference when the internal of collector is under negative pressure is larger than it under normal pressure, and the maximum temperature difference between theoretical value and experimental value under negative pressure is 0.82℃ where the absolute error of temperature difference is 3.5%, and the maximum temperature difference between theoretical value and experimental value under normal pressure is 0.6℃ where the absolute error of temperature difference is within 3.0%. No matter when internal working medium of the collector is under whether negative pressure or normal pressure, its theoretical value is greater than experimental value, as the theoretical calculation ignores the radiation outward from side wall of collector and absorber plate. The efficiency of the collector is respectively 59% and 54% when the internal working medium is under the stage of negative pressure and normal pressure. Thus, The fan should be placed above the collector in the future test, keeping the internal working medium under negative pressure, which can contribute to the heat transfer of the air in the collector flow channel and get higher outlet temperature to promote the efficiency. It is helpful for the application and promotion of the fin plate air collector in solar heat utility, especially in the solar drying field as the stable medium and low temperature heat source.

5. Conclusion

The thermal performance and the suitable conditions of fin flat plate air collector are summarized through comparatively analyzing theory model of air the collector with experimental value in this paper. The results are as follows:

1) The installation angle of the fin plate air collector has little effect on the efficiency. The efficiency difference is only 3% in different angles, the error is only with 3.5% which is in permitted range.

2) The efficiency of fin plate air the collector varies significantly in different air mass flows, especially when the working medium mass flow is small and increases slowly.

3) The efficiency and temperature difference of inlet and outlet when the wind machine placed on the top are both higher 5% on average than those when placed at the bottom. In actual application, the fan can be installed on the top of fin flat plate collector, keeping the internal working medium under negative pressure, which can enhance the export temperature and improve the heat utility of the collectors.

Nomenclature

T_s	Sky temperature ℃	T_a	Environment temperature ℃
T_{fi}	Working medium temperature of Collector at the entrance ℃	T_{fo}	Working medium temperature of Collector at the exit ℃
C_p	The specific heat of air at constant pressure J/kg · K	I	The solar irradiance W/m^2
α_g	Absorption rate of glass cover plate	τ_g	Transmissivity of glass cover plate
α_p	Absorptivity of absorber board	ε_b	Emissivity of baseboard
ε_g	Emissivity of glass cover plate	υ_w	Environmental wind speed m/s
ε_p	Emissivity of absorber board	σ	Stefan-Boltzmann constant，5.578×10^{-8}W/(m^2 • K)
$\dot{m}$	Air mass flow kg/s	δ_R	Thickness of back insulation layer m
κ_R	Insulation thermal conductivity W/(m^2 • K)	κ	Air thermal conductivity W/(m^2 • K)
W_1	Width of single flow channel m	W_2	Height of the fin m
A_{fin}	Aarea of the fin m^2	A_p	Effective heating area of collector m^2
D	Equivalent diameter of collector m	W	Width of collector m
H_g	Average distance of base plate and endothermic board m	Nu	Nusselt number
Re	Reynolds number	Pr	Pratt number

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