Investigation of Thermal Performance of Balcony Wall Flat-Plate Solar Collector

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Abstract. Integrated into balcony of the city building, the flat-plate solar collector becomes more widely used. However, a considerable part of balcony wall solar water heaters with flat-plate solar collectors has very low thermal efficiency. Hence, this paper developed an explicit and detailed model, to investigate the thermal performance of the collector and to give design suggestions. The model is based on the structure of commercial solar collector, and can output the detailed temperature distribution of the collector. The influence of major design factors, such as flow rate, unwelded length of tubes, and thickness of absorber, arrangement of tubes, inner fins of tubes and thermal conductivity of insulation layer are studied. The results show that it is not good for collector performance to install tubes along the short edge, and the unwelded length of tubes inlfluence the collector performance significantly. To improve the collector performance, it has little use to increase the flow rate and thickness of absorber pltate. And it is suggested to decrease the length of unwelded part of the absorber plate and tube interval, and use inner finned tubes.

Keywords: solar water collector, performance simulation, transient efficiency, thermal performance.

1. Introduction

Solar water collectors have been proved to be able to significantly reduce the energy consumption of domestic water heating system [1]. The flat-plate collector is widely used in China because of its high efficiency, reliability, conveniences of building integration [2]. Many research studied the performance of different collectors [3-7]. By decompose the two dimensional temperature distribution of absorber plate into independent temperature distribution in two axes, a simplified model of flat-plate solar collector (HWB model) has been proposed by Hottel and Whiller [8]. The fundamental theory and mathematical model of flat-plate solar collector were introduced by Duffy and Beckman [9]. The thermal performance of a flat-plate, sandwich-like solar collector with serpentine ducts was investigated by Alvarez et al. [10]. Riff at et al. performed theoretical investigation on a thin membrane heat-pipe solar collector [11]. A solar collector outfitted with honeycomb of different arrangements was tested and the influences of several parameters were evaluated with experiments.

In this paper, an explicit and detailed mathematical model is developed based on the structure of most widely used flat-plate solar collectors. The temperature distribution of each component of collector is revealed with this model. The influence factors of thermal performance, as incident angle, frame block, welding condition, tube structure, and insulation layer are evaluated. With the temperature distribution of collector, the mechanism of thermal performance is investigated, and the major and minor factors are determined to guide the optimization of flat-plate collector.

Nomenclature	
A	area of coee section, m^2
C	heat capacity, J/(Kg K)
D	diameter, m
I	irradiation, W/m^2
L	length, m
T	temperature, K
U	heat transfer coefficient, w/m
h	heat transfer coefficient, w/m^2
m	mass, Kg
$\dot{m}$	mass flow rate, Kg/s
DT	temperature differene between inlet temperature and air temperature, ^oC
Greek	
	absorptivity, -
	inclined angle of collector, degree
	thickness, m
	efficiency, -
	thermal conductivity, $W/(m^2 K)$
	density, kg/m^3
	transmittance, -
Subscripts	
a	air gap
c	collector
b	back plate
f	fluid
g	glass
L	loss
r	radiation
p	absorber plate
w	unwelded
cv	convection
in	inlet
out	outlet
tu	tube
ins	inslation

2. Configuration of the Balcony Wall Flat-Plate Solar Collector

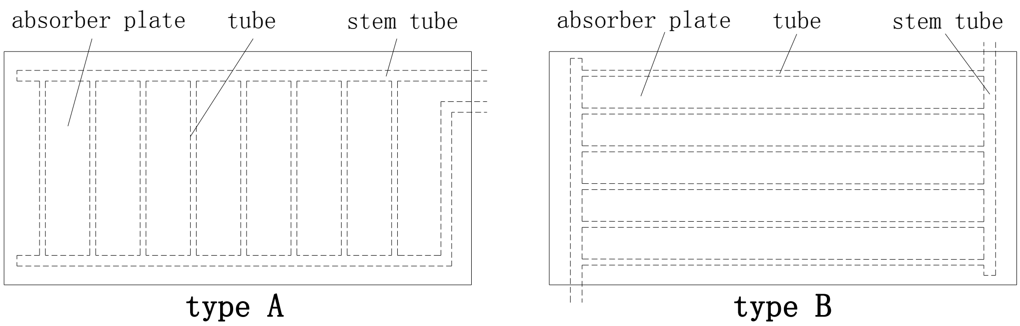

Fig. 1 Configuration types of Balcony Wall flat-plate solar collector.

The configurations of the most popular two types of the Balcony Wall Flat-Plate Solar Collector are show in Fig. 1. In type A, the tubes are installed along the short edge, while in type B the tubes are installed along the long edge. Since the inlet and outlet points are at the same side, the type A is easier to connect with tanks on balcony than type B. Since the tubes are installed along the long edge, type B uses less copper (tube & stem tube) than type A.

3. Mathematica model

The structure of flat-plate solar collector is shown in Fig. 2. Lw is the unwelded length. The aluminum absorber plate with selective absorber coating converts solar radiation into heat and transfers to the water in tubes. The glass cover is high permeable tempered glass of 3.8 mm thick. Because the glass is not contact with absorber plate directly, the temperature gradient of glass cover can be neglected. The energy equation of glass cover can be expressed as,

$$m_g C_g \frac{dT_g}{dt} = \alpha_g I + (T_{air} - T_g)h_{cv} + (T_{sky} - T_g)h_r + (T_p - T_g)(h_{\mathrm{r,a}} + h_{\mathrm{cv,a}}) \quad (1)$$

The flow rate in each branch tube can be assumed to be the same. Thus the temperature field of absorber plate symmetric to each tube is the same. The energy equation of absorber plate (0.4mm) is expressed as,

$$(\rho\delta C)_p \frac{\partial T_p}{\partial t} = \lambda_p \delta_p \left(\frac{\partial^2 T_p}{\partial x^2} + \frac{\partial^2 T_p}{\partial y^2}\right) + \alpha_a I(T_g - T_p)(h_{r,a} + h_{cv,a}) + \frac{(T_b - T_p)}{L_{\mathrm{ins}}}\lambda_{\mathrm{ins}} \quad (2)$$

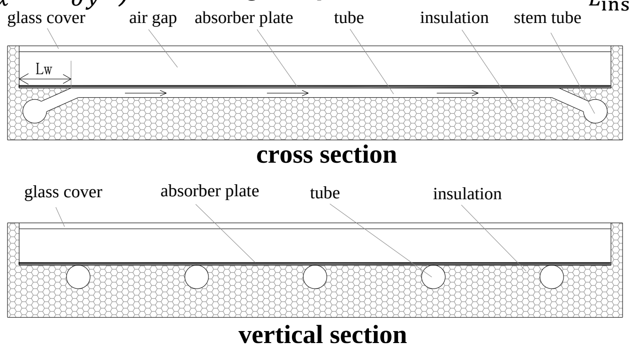

Fig. 2 Structure of flat-plate solar collector.

The temperature difference on cross section of branch tube can be neglected because of the high heat convection coefficient of copper. For each branch tube (thickness 0.5mm), the energy equation can be expressed as,

$$(\rho \mathrm{CA})_{tu} \frac{\partial T_{tu}}{\partial t} = \lambda_{tu} \mathrm{A}_{tu} \frac{\partial^2 \mathrm{T}_{tu}}{\partial y^2} + \pi \mathrm{D}_{tu} \mathrm{h}(T_{\mathrm{f}} - T_{tu}) + \mathrm{U}(T_a - \mathrm{T}_{tu}) \quad (3)$$

The energy equation of water in branch tube is given as,

$$(\rho \mathrm{CA})_f \frac{\partial T_f}{\partial t} = \dot{m} \mathrm{C}_f \frac{\partial T_f}{\partial y} + \lambda \mathrm{A}_f \frac{\partial^2 T_f}{\partial y^2} + \pi \mathrm{D}_f \mathrm{h}(T_{\mathrm{tu}} - T_f) \quad (4)$$

The transient thermal efficiency of collector is expressed as,

$$\eta = \frac{(T_{\mathrm{in}} - T_{\mathrm{out}})\dot{m}\mathrm{C}_f}{I \mathrm{A}_{c,surface}} \quad (5)$$

The part of absorber plate corresponding to one branch tube is discretized into a 50 by 50 grid to calculate the temperature distribution. The branch tube and water in it are discretized into 50 nodes. Implicit discretization and second order upwind scheme are utilized to ensure the accuracy. Conducting term is included in Eq. (4) because the heat conduct between each node of water cannot be neglected under low flow rate.

The thermal physical properties of collector components and fluid are calculated with initial temperature. Then the transmittance and absorption of glass cover and absorber plate is determined with incident angle. With the discrete equations, the equation of every nodes can be expressed as a matrix equation,

$$[\mathrm{M}]\vec{T} = \vec{b} \quad (6)$$

With Eq. (6), the temperature of every node of next time is got and used to update the matrix [M] and constant vector $\vec{b}$. With the stable outlet temperature after iteration, the thermal efficient is calculated by Eq. (5).

For conventional flat-plate solar collector, the instantaneous thermal efficiencies can be plotted as a function of reduced temperature, expressed as,

$$\eta = \eta_0 - U_L \frac{T_{\mathrm{in}} - T_{\mathrm{air}}}{I} \quad (7)$$

η_0 And U_L are two essentials of the thermal performance of collector. With these two factor, the transient efficiency under different conditions can be calculated easily. As specified in the national standard of China, thermal efficient under zero reduced temperature (η_0) should be over 72%, and heat loss coefficient (U_L) should be no bigger than 6 W/ (m^2·K). To make the figure clear and easy to be upstand, the heat loss coefficient is describe as the transient efficiency difference of two inlet temperature (T_{in}). In the simulation results, the heat loss coefficient loss can be expressed as,

$$U_L = \frac{\eta_0 - \eta_T}{T} \tag{8}$$

4. Results and discussions

4.1. Temperature distribution in collector.

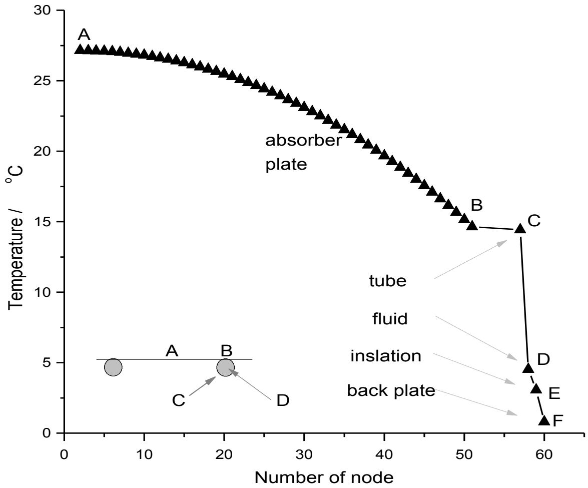

Fig.3 Temperature distribution of the collector.

The collector is set to be 2×1 m, with frame height of 3.5 cm. 8 branch tubes are placed evenly along the long edge (type B). The tubes are of 0.5 mm thick, 2 meter long and diameters are 10 mm. The unwelded length is 0 mm. The operating parameters are as follows: ambient temperature of 10 °C, collector inclined angle of 90°, solar radiation of 1000 W/m^2, incident angle of 0°, flow rate per unit area of 0.03 L/m^2, and inlet water temperature of 10 °C. The temperature distribution of upper cross section of collector are shown in Fig. 3. The average temperature of absorber plate is 22 °C higher than the ambient. The temperature of branch tube is 10 °C higher than water in it. We can see that the heat transfer is limited on absorber plate (A to B) and the fluid (C to D). This figure can be used to analyze the performance of solar collector.

4.2. Effect of operating parameters.

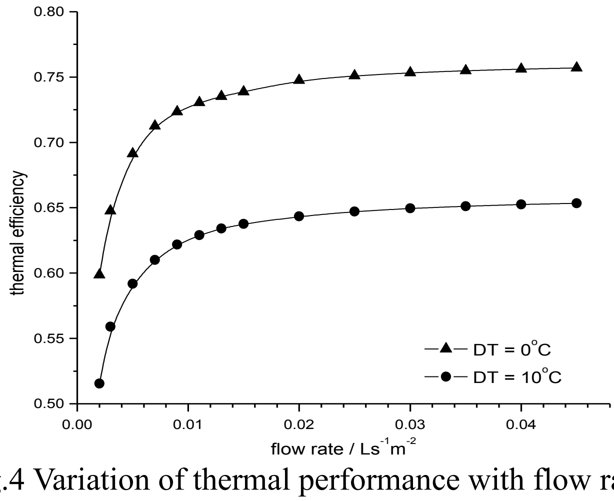

Fig.4 Variation of thermal performance with flow rate.

Fig. 4 shows the effect of flow rate on thermal performance of collector. It shows that the thermal efficiency increases with the ascending of flow rate. However, the increase of efficiency is insignificant when the flow rate per unit is over 0.015 L/(s·m^2). With low water flow rate, the increase of water temperature is the dominant factor of thermal efficiency increase. When the flow rate is high enough, the water temperature increase is not the dominant factor. When the inlet temperature increase, the efficiency decrease. The difference of the efficiency of two DT is nearly constant with flow rate over 0.015 L/(s·m^2), which is bigger than that with lower flow rate.

4.3. The effect of the structure parameters on the performance of the collector.

Fig. 5 shows that the intercept efficiency of the collector decreases with the increase of the length of the unwelded tube. In type A, the decrease of the intercept efficiency is more obvious. When the length of the unwelded tube is more than 7 cm, the intercept efficiency drops sharply. This shows that the heat absorbed by the absorber which the distance to the closest welding point is larger than 7 cm can nearly not be conducted to the water flow in the tube. So when the tube is set along the short edge (type A), the length of the unwelded tube should be decreased. The length of the tube of many current balcony wall flat-plate solar collector (type A) and the unwelded length of tube are 1 m and 8 cm, separately, and the distance between the edge and the closest tube is about 30 cm (right side of type A, the stem tube cannot be welded with absorber plate). According to the Fig. 5, it can be inferred that deceasing the length of the unwelded length of tube and the distance between the edge and the closest tube (right side of type A) can improve the performance of this type of collector significantly. Comparing the two lines, we can see type B is more suitable for the configuration of collector.

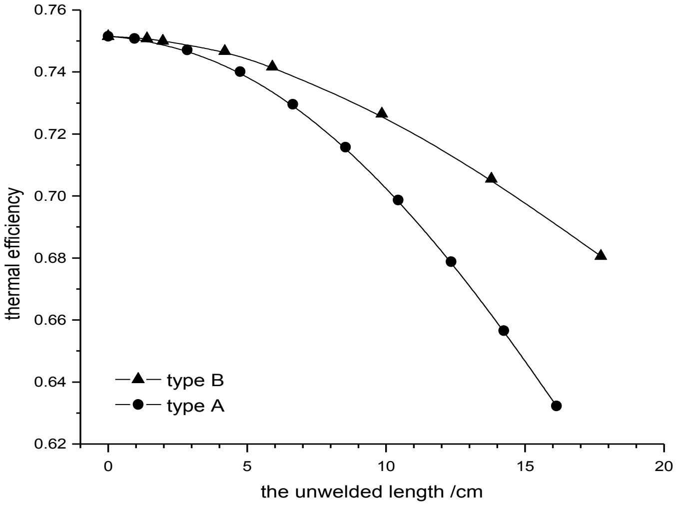

Fig.5 Variation of thermal performance with unwelded length.

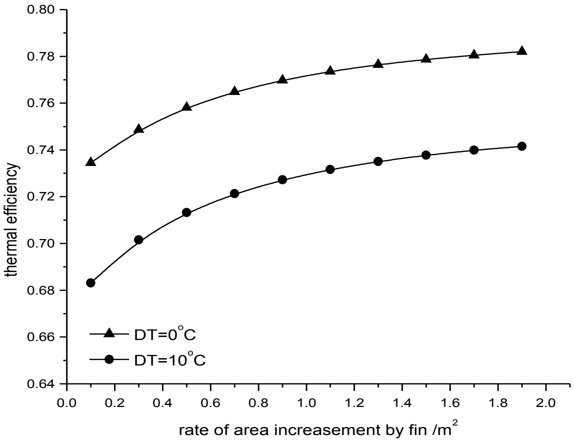

Fig.6 Variation of thermal performance with fin area.

In Fig. 3, the thermal performance of the collector can be improve by reducing the temperature difference of the absorber and the temperature difference between the tube and the water flow. Increasing the convective heat transfer coefficient and the heat exchange are effective ways to reduce the temperature difference between the tube and the water flow. However, under the normal condition of the collector, the internal flow cannot form turbulent flow. And with the limit effect of the laminar flow on the convective heat transfer coefficient, the increase of the flow rate has little influence on the convective heat transfer coefficient, as shown in Fig. 4. Hobbi et al. verified that the turbulence device can't improve the convective heat transfer coefficient in their experiments [12]. Fig. 6 shows that the thermal efficiency of the collector increases with the increase of the inner fin area of the tube, but the effect tends to diminish. From the comparison between the two curves, the fin area doesn't influence the heat loss coefficient.

As shown in Fig. 4, it reduces the heat transfer temperature difference on the absorber, that increasing the thickness of the absorber plate and decreasing the tube interval. Fig. 7 shows that with the increase of the thickness of the absorber plate, the thermal efficiency increases significantly at the initial stage then tends to be constant, and the optimum thickness of the absorber plate is 0.4 mm.

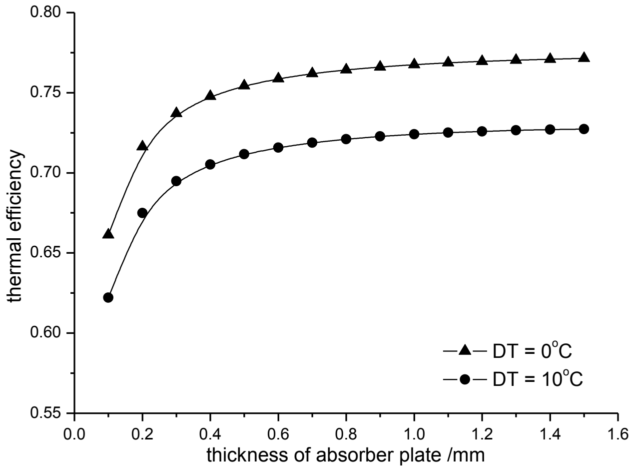

Fig.7 Variation of thermal performance with plate thickness (Lw=0cm).

The above results are gained with the assumption that the unwelded length of the tube and absorber plate is 0 cm. However, in real products, the unwelded length is usually about 10 cm. Fig 8 compared the thermal efficiency of the collector with the unwelded length of tube (Lw) of 0 cm and 8cm. The collector with Lw of 10 cm has significantly low thermal efficiency than that with Lw of 0 cm. The difference of the thermal efficiency with different DT decreases with the increase of the thickness of the plate. That is say that when Lw is about 10 cm, the absorber plate should be thicker to decrease the influence of the Lw. For the collector of type A, Lw can have much more significantly bad influence on the collector performance, which decreases the thermal efficiency significantly.

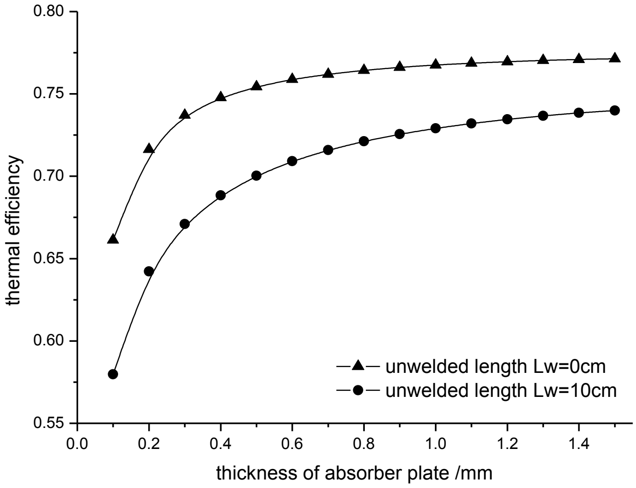

Fig.8 Variation of thermal performance with plate thickness (Lw =10cm).

Fig. 9 shows that when the tube interval decreases, the thermal efficiency of the collector increases significantly. Due to the decrease of the tube interval, the heat flow density on the absorber plate decreases, the temperature difference on the absorber plate and the temperature between the tube and water decrease, then the thermal efficiency of the collector rises. It can be say that reducing the tube interval can be an effective way to improve the thermal efficiency with the economy premise.

In practical application, the water vapor may condense or leak in the collector, and the condensing water will damp the insulation layer to weaken the heat preservation effect. Meanwhile, the heat capacity of the wet insulation increases, and the absorbed heat cannot be used, which will lead to the decrease of the thermal efficiency of the hot water system. Fig. 10 shows that the increase of the thermal conductivity of the collector will decrease the thermal efficiency of the collector significantly, and improve the heat loss factor. Therefore, it is necessary to cover the insulation layer with plastic layer before assembling into the collector.

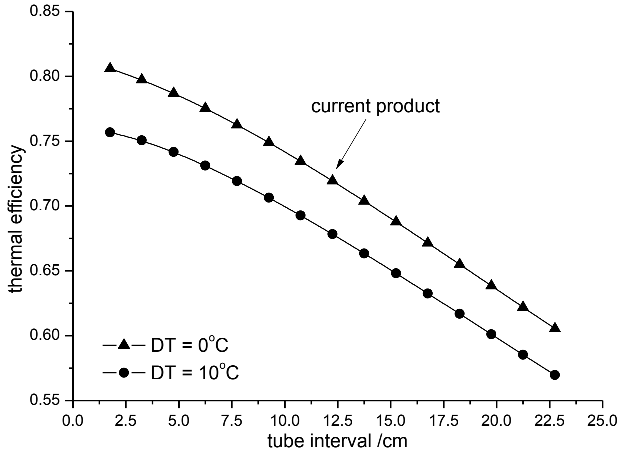

Fig.9 Variation of thermal performance with spacing of tubes.

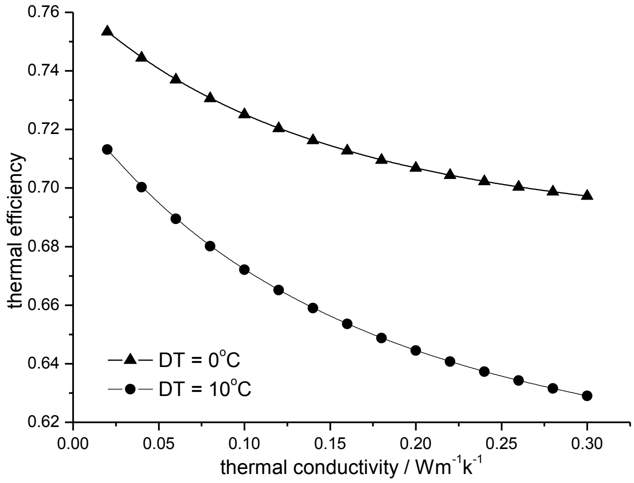

Fig.10 Variation of thermal performance with insulation thermal conductivity.

5. Conclusions

To investigated the thermal performance of the balcony wall collector and to give design suggestions, this paper build an explicit and detailed model based on the structure of commercial solar water collector. By the calculation and analysis of the temperature distribution of the collector, the main factors effecting the performance of the flat-plate collector are analyzed. On this basis, the thermal performance of collector is evaluated under different conditions, such as solar radiation, flow rates, unwelded length of tubes, and thickness of absorber, pitch of tubes, inner fins of tubes and thickness of insulation layer. The conclusions are as below:

The collector of type B (tubes installed along long edge) is a better design than that of type A (tubes installed along short edge). The design of type a costs more copper in tubes and it makes the unwelded length decrease the thermal efficiency more.

With flow rate per unit area of the flat-plate collector over 0.015 L/(s·m^2), the increase of the flow rate cannot enhance the thermal efficiency of the flat-plate collector significantly.

The unwelded length of tubes reduces the thermal efficiency of the flat-plate collector, and the thermal efficiency decrease more in type A than in type B. The minimum length between the point on the absorber plate and the welding point should be under 7 cm to insure the effectiveness of heat transfer in the absorber plate.

The thickness of current commercial solar collector is approximately 0.4 mm, on this base, the increase of the thickness can't increase the efficiency of the collector. However, for the collector that the unwelded length approaches to 10 cm, the thermal efficiency of the collector can be improved by increase the thickness of the absorber.

To improve the collector performance, it has little use to increase the flow rate and thickness of absorber pltate. And it is suggested to decrease the length of unwelded part of the absorber plate and tube interval, and use inner finned tubes.

The thermal efficiency of the collector will decrease with the improvement of the thermal conductivity of the soggy insulation layer. In the practical usage, it is advised to cover the insulation layer before assembling into the collector.

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