Flow resistance and heat transfer for single phase natural circulation flow in a vertical narrow rectangular channel

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Abstract: Experiment under natural circulation was performed to investigate the single phase flow and heat transfer characteristics of the one-side heating narrow rectangular channel. The experiment was conducted for pressure ranging from 0.2 to 0.3MPa, inlet sub-cooling temperature from 30K to 60K and heat flux from $30\,\mathrm{kW\cdot m^{-2}}$ to $90\,\mathrm{kW\cdot m^{-2}}$. The critical Reynold number in isothermal flow was about 2500. The experiment results indicated that the fraction factor can be well predicted by Shah & London correlation for laminar flow region and by Blasius correlation for turbulent flow regime. And the faction factor was found to decrease in turbulent region under natural circulation condition, when the test fluid was heated. Correlation for faction factors in natural circulation channel was carried out by modifying the Blasius correlation. Moreover, no obvious differences were found in heat transfer characteristics between natural circulation flow and force circulation flow.

Keyword: narrow rectangular channel; natural circulation; convective heat transfer; flow resistance

1 Introduction

The narrow rectangular channel has been widely used in plate fuel research nuclear reactors. And it has also been used in industrial applications, because of their higher heat transfer efficiency and better compactness.

As classified by Kandlikar[1], diameter ranging from $10\,\mu\mathrm{m}$ to $200\,\mu\mathrm{m}$ was miacrochannel, and diameter ranging from $200\,\mu\mathrm{m}$ to 3mm was minchannel, the others were called convenient channel. And for the rectangular channel of which gap size is smaller than 3mm can be regard as narrow channel. Kays & London[2] conducted their experiment in a narrow rectangular channel with a depth of 2mm in 1952, the experiment results showed that the flow resistance characteristics can be predicted with correlations suitable for convenient channel.

After Kays and London, lots of research has been done on narrow rectangular channel. The behavior of mini-channels, in terms of heat transfer and frictional pressure loss, had been found to be very close to that in conventional tubes in Agostini's work[7], and the value of the friction factor and heat transfer coefficients were well predicted by correlations established for conventional tubes in the range $500 < Re < 7000$. Moreover, Jung[8] conducted experiments in narrow rectangular channel, and the measured friction factor constants for the laminar regime were comparable to the theoretical values obtained from the correlation for the flow in narrow rectangular channel, whereas the convective heat transfer coefficients which were found to be linearly dependent on the wall temperature and the Reynolds number dependence on the Nusselt number is much higher than those for the corresponding flow in minchannels. Recently, Ma[9,10] performed experiments in narrow rectangular channel, and the correlations for the isothermal friction factors and the

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mean Nusselt numbers have been developed, and have a satisfactory agreement with the conventional theory.

Yukio Sudo[3,4] performed the experiment in a narrow rectangular channel with a gap of 2.5mm to study the natural convection. Their experimental results showed that the heat transfer coefficient higher than that predicted by the heat transfer correlation for turbulent forced convection. They explained this disagreement with the acceleration of the main flow through the development of the boundary layer.

The natural convection heat transfer of water in narrow rectangular vertical channel was studied experimentally by El-Morshedy E D [5] Their measured local Nusselt number values were compared with the predictions of two correlations for both natural and combined natural and forced convection regimes respectively, but did not agree well with both of the two correlations.

Most of the researches were conducted in isothermal condition or force circulation condition, experiments for natural circulation flow in narrow rectangular channel is infrequent. In this paper, the single phase natural circulation flow experiments were conducted to investigate the thermal-hydraulic characteristics in narrow rectangular channel.

2 Experiment apparatus

2.1 Natural circulation loop

The schematic of the natural circulation loop of this paper was shown in Fig.1. As shown in the figure, the natural circulation loop consisted of a preheater, a test section, a cooler, a main pump, an electromagnetic flow rate meter a pressurizer and some vales connected by pipes in different size.

The test water was firstly heated in the preheater to a certain temperature value, and then flowed to the test section to be uniformly heated in the narrow rectangular channel from one side. After heated in the test section, the water flowed to the condenser through the adiabatic riser. When the water arrived at the condenser, it was gradually cooled down in this apparatus by the secondary cold water. And finally, the cold test water flowed back to the preheater to take part in the next circle.

The starting up of natural circulation was difficult in our loop, so the assistant pump was put into use at the beginning, then the heater and condenser were put into use. The pump was shut down when the temperature difference was prominent. Lastly, the natural circulation was set up.

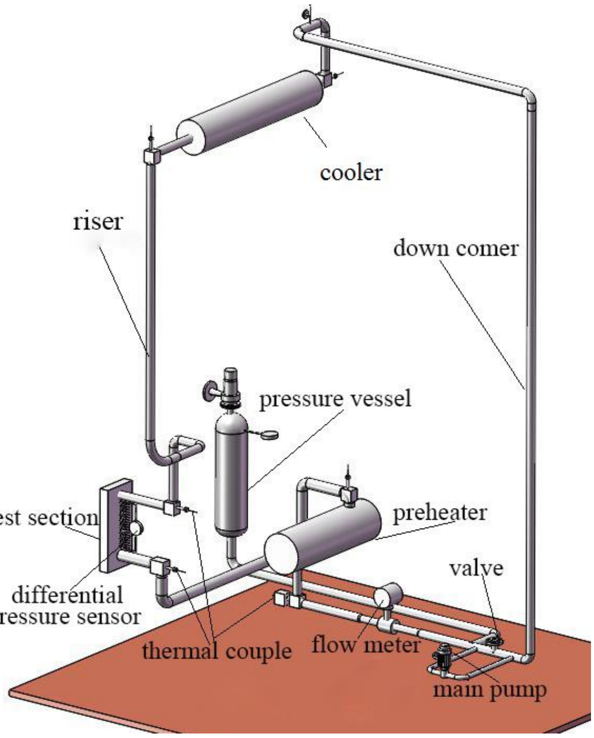


Fig.1 Schematic of natural circulation loop.

2.2 Test section

The test section consisted of a rectangular heating plate and coved with a quartz glass, which had been etched with a groove of 2mm in height and 40mm in width on the surface as shown in Fig.2. A silicon rubber “O” shape sealing was located between the edge of the groove and the heating plate to prevent the water from leaking. Thereby, the rectangular channel was structured between the heating plate and the quartz glass.

The narrow rectangular channel has a length of 670mm with 550mm heated and the cross section of the channel was a rectangular with an area of $40 \times 2mm^2$. Two pressure tap were drilled to measure the pressure drop along the pressure test part, each of which has a distance of 100mm from the inlet and the outlet. On the back of the heating plate, 17 “N” type thermal-couples with a diameter of 1mm were attached to measure the temperature of the plate.

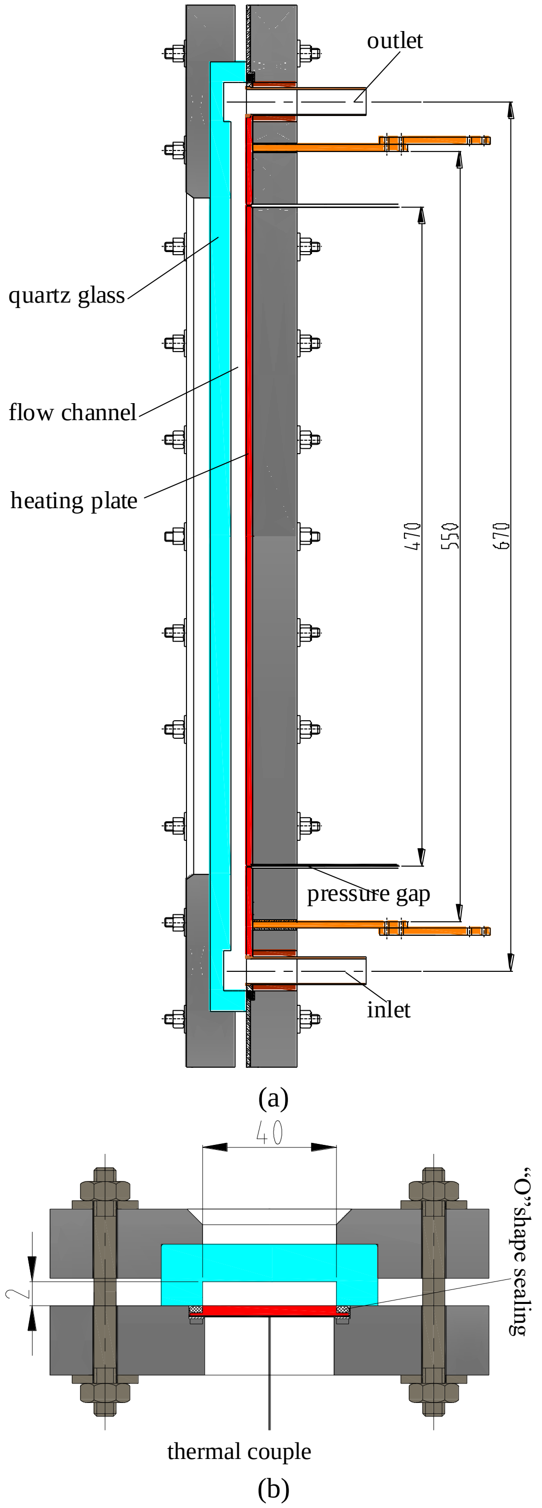


Fig.2 Visual one-sided heated test section.

3 Date reduction

The pressure drop along the test section can be expressed as follows:

$$\Delta P = \Delta P_g + \Delta P_f \tag{1}$$

And

$$\Delta P_g = \bar{\rho}_{ch} gLcos\theta \tag{2}$$

Where $\bar{\rho}_{ch}$ is the average fluid density in test section; *L1* is the length of the pressure test section; θ is the inclination angle;

Cause ΔP can get from the differential pressure sensor, the friction pressure drop could get with equation (1) and (2).

According to the Darcy friction factor equation, the friction factor could be calculated from the following equation

$$\lambda = \frac{2De\Delta P_f}{\bar{\rho}_{ch} u^2 L1} \tag{3}$$

Where u is the mean velocity of water in the test section.

The equivalent diameter *De* was defined as follows

$$De = \frac{2 \cdot w \cdot h}{w + h} \tag{4}$$

Where *w* and *h* were the width and the height of the narrow rectangular channel respectively.

For the thickness of the uniform heating plate was only 3mm, which is much smaller than the width (40mm) and the length (550mm)of the plate, the heat transfer process could be treat as an one dimension thermal conduction. The governing equation could be written as

$$\frac{\partial^2 T}{\partial y^2} + \frac{\Phi}{k_w} = 0 \tag{5}$$

With the constant heat flux boundary, the inner wall temperature of the heating plate can be calculated from the follow equation

$$T_{w,i} = T_{w,o} - \frac{1}{2}\frac{\Phi}{k_w} Y^2 \tag{6}$$

The k_w varied with the temperature

$$k_w = 14.85 + 0.0143371 T_w \tag{7}$$

The heating power Q was calculated with the enthalpy of the water in the inlet and outlet

$$Q = m \cdot \left(i_{out} - i_{in}\right) \tag{8}$$

Where *m* was the mass flow rate.

For the channel was uniform heated, the local bulk temperature of the flow was regarded to increase linearly.

So the heat transfer coefficient was gotten from the following equation

$$h = \frac{q''}{T_{w,i} - T_b} \tag{9}$$

Where q'' was the heat flux.

And the Nusselt number was calculate as

$$Nu = \frac{h \cdot De}{k_f} \tag{10}$$

4 Result and discussion

The isothermal flow and natural circulation flow experiments had been performed in this paper. In the isothermal flow experiment, the Reynolds number ranged from 500 to 10000. In the natural circulation experiment, the system pressure were 0.2MPa and 0.3MPa, with inlet subcooling ranging from 30K to 60K, and heating flux ranging from 30 $\mathrm{kW \cdot m^{-2}}$ to 90 $\mathrm{kW \cdot m^{-2}}$. During the experiments, the following result had been obtained.

4.1 Flow resistance characteristics

The isothermal experiment showed that the fraction factors in narrow rectangular channel agreed well with Shah & London predictions in laminar flow regime, as shown in Fig.3. And the Shah & London[11]correlation was define as flows

$$\lambda = 96 \times (1 - 1.3553\varepsilon + 1.9467\varepsilon^2 - 1.7012\varepsilon^3 + 0.9564\varepsilon^4 - 0.2537\varepsilon^5)/Re \tag{11}$$

Where ε was the channel aspect ratio.

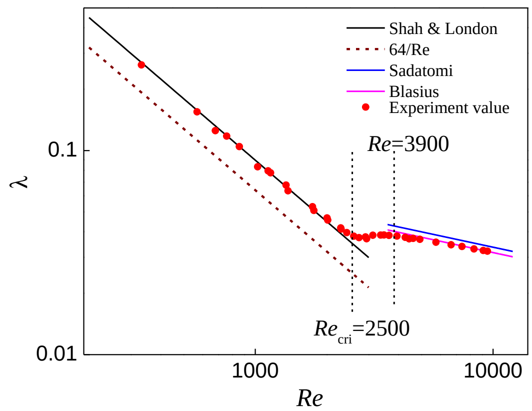


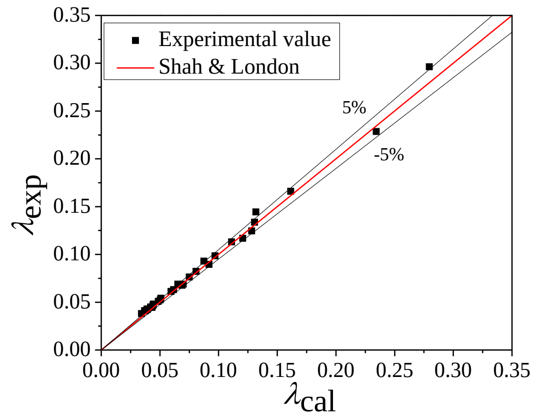
Fig.3 Variation of friction factors with *Re* in adiabatic channel.

The friction factors in turbulent region was found to can be predicted by Blasius correlation, while much smaller than the predictions of Sadatomi[12]correlation

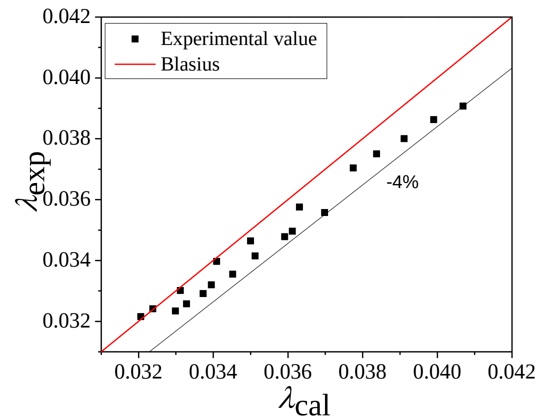
$$\lambda = 0.3164 Re^{-0.25} \cdot \left[(0.0154 C_{vl}/64 - 0.012)^{1/3} + 0.85\right] \tag{12}$$

which is reported to be suitable for non-circular channel in convenient size channel. Meanwhile Fig. 3 showed that the transitional Reynolds number was about 2500, which was a little larger than circular channel.

The relative errors between predictions and experiments results were showed in Fig.4 for laminar and turbulent flow regime respectively.



(a) laminar flow regime



(b) turbulent flow regime

Fig.4 Relative errors between experiment results and correlation predictions for fraction factors.

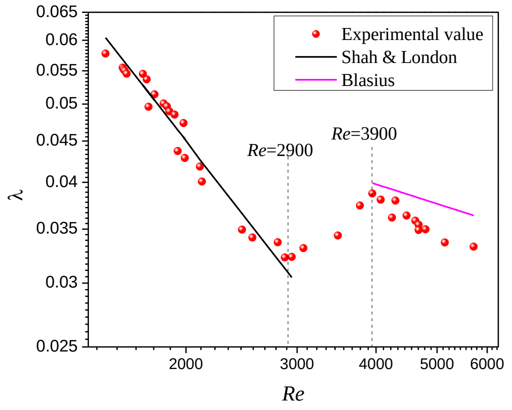


Fig.5 Variation of friction factors with *Re* in heated channel.

Figure 5 showed the friction factors in heated channel under natural circulation condition. It was found that the error range was larger, and the transition from laminar to turbulent might delayed. Moreover, the friction factors in turbulent region were significant smaller than the predictions of Blasius correlation.

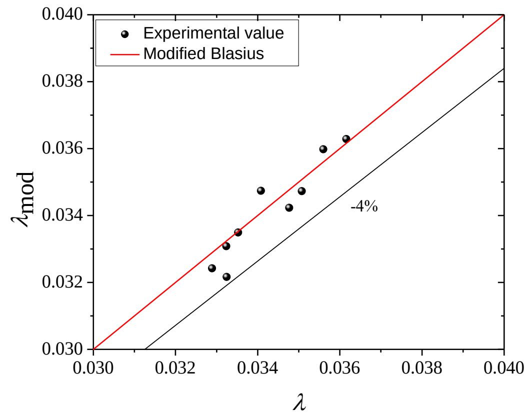


Fig.6 Relative errors between experiment results and modified correlation predictions.

For the channel was heated, the property of the fluid next to the heating surface was different from the bulk flow, especially the viscosity. It is known that the viscosity of water decrease with the increase of temperature. So the friction factors might be smaller. To modified the Blasius correlation, the viscosity under bulk flow temperature and heating plate surface temperature was taken in as follows

$$\lambda_{\mathrm{mod}} = 0.3164\,\mathrm{Re}^{-0.25}\left(\frac{\mu_{\mathrm{w}}}{\mu_{\mathrm{f}}}\right)^{n} \tag{13}$$

When *n* was 0.7, the predictions agreed well with the experimental results, as shown in Fig. 6.

4.2 Heat transfer characteristics

As shown in Fig.5, the critical Reynolds number from laminar flow to turbulent flow was about 2900, which is larger than the isothermal flow. In the study of heat transfer characteristics, the same result was obtained. Figure 7 showed the average *Nu* verse the *Re*. The *Nu* was slightly increase with the *Re* at low Reynolds number region, however, a sharply increase was found at the point of *Re* was about 2900. Combined with the phenomenon of the friction factors, we could take 2900 as the critical *Re* of the onset of turbulence. And conclusion was got that flow transition of heated natural circulation flow was delayed in narrow rectangular channel. This conclusion was also obtained by Ma[9] in his force circulation flow in a narrow rectangular channel.

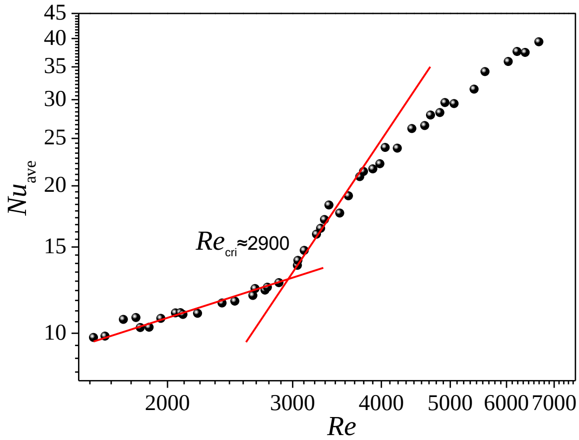


Fig.7 The average *Nu* through the channel verse *Re* in different flow regime.

The convection heat transfer coefficient was usual predicted by Dittus-Boelter (D-B)correlation and Gnielinski correlation in turbulent flow regime. The Gnielinski correlation was shown in the follow equation

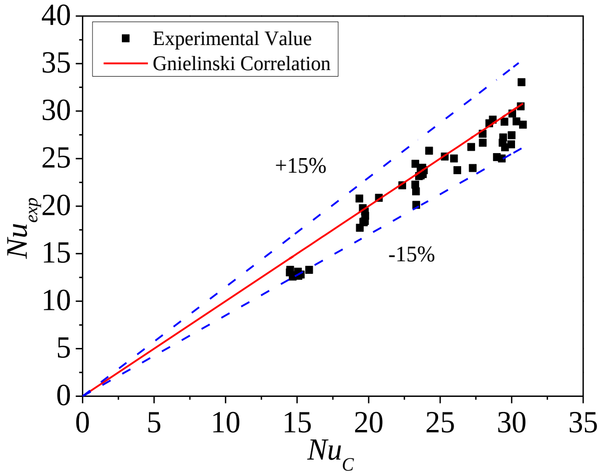
$$Nu_f = \frac{(f/8)(Re-1000)Pr_f}{1+12.7\sqrt{f/8}\,(Pr_f^{2/3}-1)}\left[1+(\mathrm{D_e}/\mathrm{L})^{2/3}\right]c_t \qquad c_t = \left(\frac{Pr_f}{Pr_w}\right)^{0.11} \tag{14}$$

In Gnielinski correlation, the Darcy coefficient was suggested to calculated with Filonenko correlation

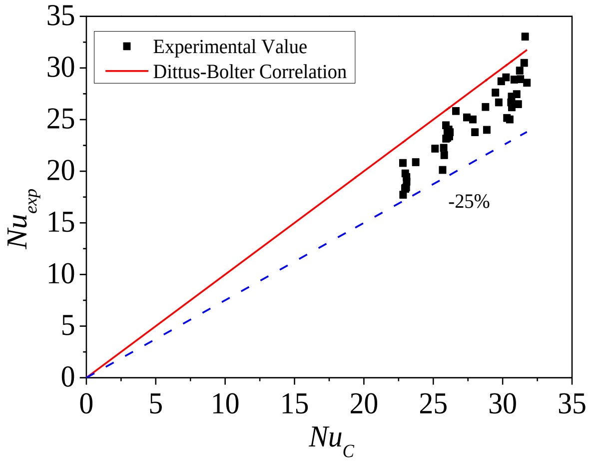
$$f = (1.82\lg Re - 1.64)^{-2} \tag{15}$$

However, the correlation was not suitable for the narrow rectangular channel. Finally, we chose the Darcy coefficient from our flow resistance results to modified the correlation. Then, the predictions agreed well with the prediction of modified Gnielinski with an error range of 15%, as showed in Fig. 8(a).

Whereas the heat transfer coefficient was found to lower than the prediction of D-B correlation, as showed in Fig. 8(b), and the error range was about 25%.



Gnielinski correlation



(b) D-B correlation

Fig.8 Relative errors between experiment results and correlation predictions for Nusselt number.

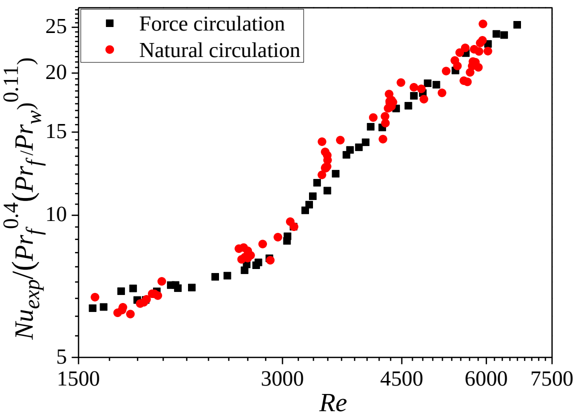


Fig.9 heat transfer characteristics along natural circulation and force circulation flow.

4.3 Comparison of natural circulation and force circulation

The difference of heat transfer characteristic between natural circulation and force circulation condition is still unclear. Yang[13, 14] and his co-works pronounced that the heat transfer coefficient of the natural circulation flow was smaller than that of the force circulation flow. To certify the conclusion, we preformed the force circulation experiments. However, little difference was found between the two kinds of circulation. The results were showed in Fig.9.

5 Conclusions

Experiments of single phase flow and heat transfer characteristics had been conducted in narrow rectangular channel in a natural circulation loop. With the measured pressure drop along the test section and the temperature of the heating plate and test fluid, the fraction factors and heat transfer coefficient had been figured out. Comparing with the correlations in convenient channel, most of the flow and heat transfer characteristics in narrow rectangular channel were similar but some slightly differences.

The Shah & London correlation can well predicted the fraction factors in laminar flow regime in both isothermal flow and natural circulation flow. However, fraction factors in turbulent flow regime showed a departure from Sadatomi correlation, and closed to Blasius correlation in isothermal flow, and when the test flow was heated, the Blasius showed be modified with the wall temperature.

The heat transfer coefficient in laminar flow regime was found to be greater than the theoretical solution of Kays & London, and could be predicted by Gnielinski correlation in turbulent flow regime, with the Darcy coefficient taken from our experiment result.

In addition, heating was found to delay the transition from laminar flow to turbulent flow, with *Re* equal to 2500 and 2900 respectively. No obvious difference in heat transfer characteristics was found between natural circulation flow and force circulation flow.

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