AREA GOODNESS FACTOR OF FLOW IN A PLAIN CIRCULAR TUBE WITH TWISTED TAPE INSERT AND SQUARE-EDGE ENTRY IN THE TRANSITIONAL FLOW REGIME

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ABSTRACT

The transitional flow regime in the pressure drops of tube with twisted tape inserts is presented in this experimental investigation. This region of transition has not been previously identified for friction factors with twisted tape insert. This study employed the method of area goodness factor in order to establish this region with twisted tape insert. This experimental study was conducted at a constant heat flux boundary condition of 2 kW/m^2 using water as working fluid and a twisted tape insert with twist ratio of 5. The friction factors in a plain circular tube and tube with twisted tape insert across the fully developed flow region were determined from the measured pressure drops over a bulk Reynolds number range of 560 to 10 404 and Prandtl number range of 3.81 to 6.67. The area goodness factor used to identify the region of transition with the twisted tape insert was evaluated by dividing a fully developed local Colburn *j*-factors at a length-to-diameter ratio of 141 by the corresponding friction factors obtained at the same mass flow rate. This method established a diversion of data in the laminar flow regime from the data in the turbulent flow regime. The data within the diversion was classified as the transitional flow regime which commenced at a bulk Reynolds number of 1 029 and ended at a bulk Reynolds number of 1 505 for the tube with the twisted tape insert of twist ratio of 5. A comparative analysis of the friction factors against Reynolds numbers in the plain tube showed that the transitional flow regime commenced at a bulk Reynolds number of 3 041 and ended at a bulk Reynolds number of 3 292.

INTRODUCTION

One of the methods of identifying the transitional flow regime in the variation of friction factors with Reynolds numbers is the diversion of the data in this regime from that of the data in the laminar flow regime and data in the turbulent flow regime. In plain tube heat exchangers, this diversion is easily observed as reported in the work of Everts [1], Tam et al. [2] and Ghajar and Madon [3].

However, with a tube enhanced with twisted tape inserts, such diversion indicating the transitional flow regime is not easily visible or identifiable when the friction factors are plotted against Reynolds numbers.

Previous studies have opined that the transition from the laminar to turbulent flow regime is smooth as the friction factors in twisted tape inserts vary with Reynolds numbers. In the isothermal friction factor results of Seymour [4], as the twist ratios reduced, a discontinuation was not obtained from the laminar flow regime to the turbulent flow regime. The reason for the smooth transition was that the onset of the turbulent mixing was suppressed and delayed by the secondary motion [5]. The challenge of characterizing the transitional flow regime was reported in Manglik and Bergles [6], it was suggested that using a linear curve fit, the three flow regimes could be identified and that any flow with Reynolds number greater than 10 000 could be regarded as fully turbulent. It was also reported that the turbulent flow regime could also be achieved at lower Reynolds numbers in some instances.

It is evident that no previous study has been able to characterise the transitional flow regime of pressure drops in a tube with twisted tape insert. This means that a method is needed in order to characterize the transitional flow regime of pressure drops in a heat exchanger with twisted tape inserts.

In a heat transfer experiment, the data of the heat transfer and that of the pressure drops are obtained at the same mass flow rate. This is true of the results obtained and reported in the plain tube heat exchangers mentioned earlier. This means that the region of transitional flow regime obtained in the heat transfer analysis should correspond to that of the pressure drop since they were measured and evaluated at the same mass flow rate, but not necessarily the same Reynolds numbers. This is because some results are evaluated on local basis and others are evaluated on average basis.

One of such methods to obtain the transitional flow regime of pressure drops using this "same mass flow rate principle" is evaluation of the area goodness factor *(AGF)*.

The area goodness factor is one of the two methods used in the design of compact heat exchangers for different surface geometries. This method is used to make an immediate comparison between the Colburn *j*-factors and the corresponding friction factors. The *AGF* is evaluated as the ratio of the Colburn *j*-factor to friction factor. The *AGF* is a useful tool when the surfaces of heat exchangers with a fixed pressure drop of a fluid are compared [7, 8]. The decision on the best surface geometry is made based on how high or low the AGF is; a higher area goodness factor shows that the surface geometry is better, as the heat exchanger will need a reduced area of flow and frontal area [9].

The focus of this study was to characterize the pressure drops of flow in a heat exchanger in terms of non-isothermal friction factors in the three regimes of laminar, transitional and turbulent flow. The particular attention was on the identification of the transitional flow regime in a plain circular tube heat exchanger with twisted tape insert of twist ratio of 5. Water used as the working fluid made a square-edge entry into the heat exchanger and the boundary condition was maintained at constant heat flux of 2 kW/m^2. The range of Reynolds number was 560 to 10 404.

NOMENCLATURE

A_c	[m^2]	Cross sectional area
AGF	[-]	Area goodness factor
D	[mm]	Diameter of the test section
f	[-]	Friction factor
j	[-]	Colburn j-factor
k	[W/m.K]	Thermal conductivity
L	[m]	Length of the test section
$\dot{m}$	[kg/s]	Mass flow rate
p	[Pa]	Pressure drop
Pr	[-]	Prandtl number
Re	[-]	Reynolds number
RTD	[°C]	Resistance temperature detector
T	[°C]	Temperature
U	[m/s]	Velocity
x	[m]	Distance from tube inlet
y	[-]	Twist ratio
Greek symbol		
ρ	[kg/m^3]	Density of the working fluid
μ	[kg/m.s]	Dynamic viscosity
Special characters		
Δ	[-]	Change
Subscripts		
b		Bulk
e		Exit
h		Heating
i		Inlet, inner
m		Mean or average

EXPERIMENTAL SETUP

The experimental setup for the measurement of the pressure drop is shown in Figure 1. This setup comprised of a 1 000 ℓ cold water reservoir (1), a 15 kW chiller unit (2), an electronically controlled positive displacement pump (3), an accumulator (4), valves (5), two coriolis flow meters (6), inlet mixer, calming section and inlet section (7), test section (8) on which two pressure taps were installed to measure the pressure drops using the pressure transducer (9), power supply (10) connected to heating wires, exit mixer (11), hot water reservoir (12) and hot water pump (13).

EXPERIMENTAL PROCEDURE

The water circulated during this experiment was from a 1 000 ℓ cold water reservoir (1), the water in the reservoir was maintained at a temperature of 20°C by the 15 kW chiller unit connected to this reservoir. The water was pumped using the electronically controlled positive displacement pump (3), this pump had a maximum volume flow rate of 1 344 ℓ/h. The water then flowed through a 1 ℓ standard valve accumulator (4), with a maximum volume flow rate of 14 000 ℓ/h. This accumulator is equipped with bladders that contained nitrogen. The accumulator was used to reduce the pulsation accompanied with the electronically controlled positive displacement pump and to also ensure a constant pressure process.

As the water flowed through the pipe network the valves (5) were installed to bypass the flow in order to achieve a lower mass flow rate and to also switch between each of the two coriolis flow meters (6).

The two coriolis flow meters (6) were of different capacities and were connected in parallel. The first coriolis flow meter could be operated to a maximum volume flow rate of 108 ℓ/h and was used during the measurement of lower mass flow rates. The second coriolis flow meter, capable of a maximum volume flow rate of 2 180 ℓ/h, was used during the higher mass flow rate measurements.

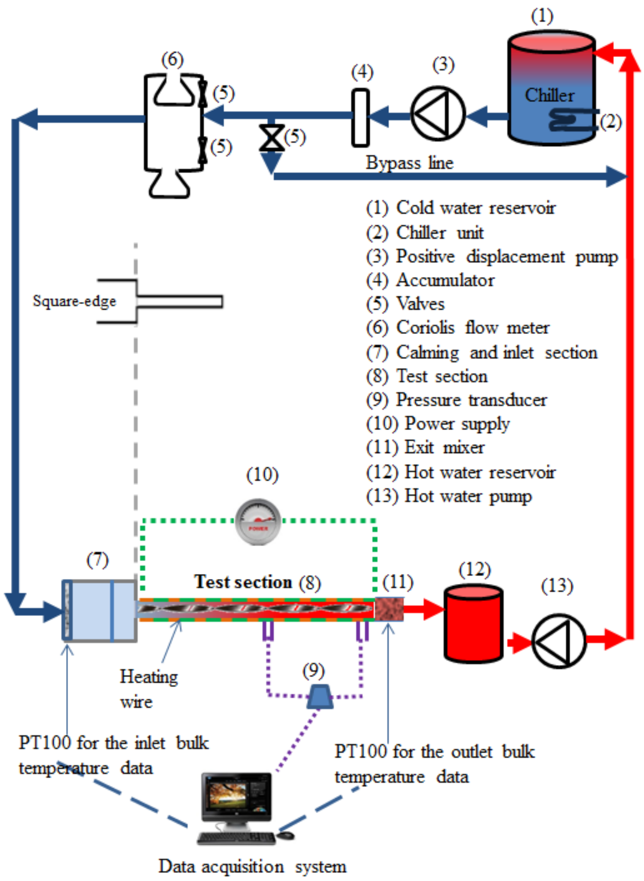

Figure 1 The schematic diagram of the pressure drop setup

As the working fluid exited either of the coriolis flow meters, it flowed through an inlet mixer, calming section and square-edge inlet section (7), before entering the test section (8). A resistance temperature detector *(RTD)* was installed in the inlet mixer to measure the bulk temperature of the fluid entering the calming section and the test section. This test section was made from a plain copper tube with an inner diameter of 19 mm, outer diameter of 22 mm and length of 5.27 m. Two experiments were conducted on the test section, the first was for the plain tube and was used for the purpose of validation. The second was conducted by inserting a typical twisted tape into the plain tube test section. The length of the insert was the same as that of the test section.

On the test section were installed two pressure taps. The first tap was installed at a distance of 2.52 m from the inlet of the test section. The second tap was installed at a distance of 2.45 m away from the first and in axial direction of flow. The installation of these two taps was achieved by soldering a copper tube with a diameter of 4.6 mm on each of the points and then drilled a hole of 1.6 mm into these small copper tubes. The diameter of each of the holes was less than 10% of the inner diameter of the test section, this choice of the hole drilled into these copper tubes was made in order to avoid the formation of eddy that could influence the pressure drop measurement on the test section as the fluid flowed through the taps [10, 11]. The upstream and downstream of the taps were then connected to a pressure transducer (9) using pneumatic fittings and 3.17 mm hoses. The pressure transducer contained a diaphragm with a maximum pressure of 1.4 kPa.

The non-isothermal process on the test section was achieved by the two heating wires wound round the outer surface of the test section. The wires were connected in parallel and then to a power supply (10) on which the current and voltage were set to achieve the heat transfer. These heating wires were used to achieve a constant heat flux boundary condition.

As the working fluid flowed out of the test section after the pressure drops across the two pressure taps have been measured at different mass flow rates, the fluid flowed through an exit mixer (11) where an *RTD* was installed to measure the bulk exit temperature of the working fluid.

The hot water leaving the test section then flowed into a hot water reservoir (12). The hot water was then pumped by a positive displacement pump (13) back into the cold water reservoir (1) and then cooled by the chiller unit (2). This procedure was repeated for all the range of mass flow rates considered in the two experiments carried out in this study.

A detailed uncertainty analysis was conducted on all the instrumentations and variables such as the friction factors and the Reynolds numbers. The uncertainty was obtained as the square-root of the sum of the squares of the bias and the precision of all the instrumentations. The bias was taken as the uncertainty of the instrumentation provided by the manufacturers. The precision was evaluated according to the work of Dunn [12]. This was obtained as the product of the standard deviation of the 400-sample data and the student't values at the 95% confidence level. The accuracies of the instrumentations: *RTDs*, mass flow rates and pressure drops were 0.06℃, 9.87×10^{-5} kg/s and 3.5 Pa respectively. The uncertainties of the friction factors are 27% and 0.5% at the minimum and maximum mass flow rates. The uncertainties of the Reynolds numbers reduced from 0.5% to 0.3% as the mass flow rate increased.

DATA REDUCTION

The inlet and outlet temperatures measured using the *RTDs* located at the inlet and outlet mixers were respectively logged into the data acquisition system. The values of temperatures were then substituted into Eqn. (1) to obtain the bulk temperature of the fluid at each of the mass flow rates. The bulk temperature of the working fluid was obtained at a distance of 3.769 m from the entrance of the tube. This location was the midpoint of the two pressure taps. The bulk temperature was calculated using

$$T_b = T_i + \left[\frac{T_e - T_i}{L_h}\right] * x \qquad (1)$$

The bulk Reynolds number was obtained for each of the mass flow rates measured from the flow meters, other variables used in the calculation of the bulk Reynolds number are the inner diameter of the test section, the dynamic viscosity of the water and the cross-sectional area of the test section. The Reynolds number was evaluated using

$$Re = \frac{\dot{m} D_i}{\mu A_c} \qquad (2)$$

The properties of the working fluid at each of the mass flow rates were then obtained by substituting the values of the bulk temperatures; located between the two pressure taps at a length-to-diameter ratio of 198, into the equations of the fluid properties provided by Popiel and Wojtkowiak [13]. These properties of the fluid are the density, the specific heat capacity, the thermal conductivity, the dynamic viscosity and the Prandtl number.

The friction factor was then evaluated from the measured pressure drops across the two pressure taps installed on the test section, the inner tube diameter, the fluid density, the distance between the two taps and the mean velocity of the flow using the equation of Darcy-Welsbach, given as

$$f = \frac{2 D_i \Delta p}{\rho L_{\Delta p} U^2} \qquad (3)$$

Then the area goodness factor, AGF, was obtained as the ratio of the Colburn j-factor and the friction factor at each of the mass flow rates as

$$AGF = \frac{j}{f} \qquad (4)$$

The comprehensive approach by which the Colburn j-factor was evaluated is presented in a companion paper on heat transfer [14]. The reason for combining these two dimensionless numbers in this paper is to be able to establish the region of transition in the friction factors, since both dimensionless numbers were obtained at the same mass flow rates. Therefore, the flow regime of the Colburn j-factor should correspond to that of the friction factor.

VALIDATION WITH LITERATURE

The results of the variation of the non-isothermal friction factors against Reynolds numbers are shown in Figure 2.

These results are presented in this section for the plain tube and for the purpose of validation with literature. These results are presented for the three flow regimes of laminar, transition and turbulent.

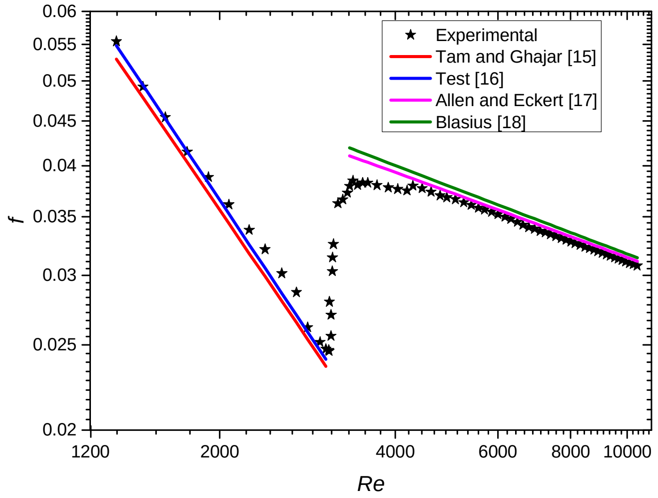

Figure 2 The variation of the friction factors with the bulk Reynolds numbers in plain tube.

The present non-isothermal friction factors are compared with known equations of Tam and Ghajar [15] and Test [16] in the laminar flow regime. In the turbulent flow regime, the present experimental results of the friction factor are compared with the equations of Allen and Eckert [17] and Blasius [18] This comparison is shown in Figure 2.

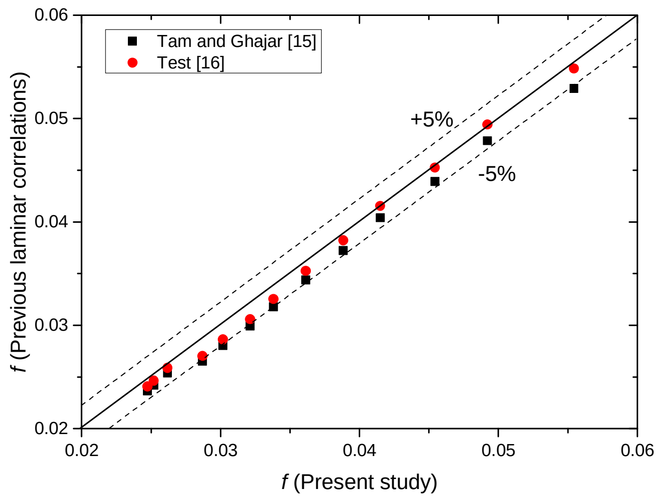

Figure 3 Deviation of the present friction factor experimental data with previous equations in the laminar flow regime

The friction factors in plain tube as presented in Figure 2 were obtained and plotted over the range of Reynolds numbers of 1 331 to 10 404 from laminar to turbulent flow regime.

The percentage deviation of the present non-isothermal friction factors in the laminar flow regime are shown in Figure 3. These results did not deviate so much from the equations of the previous studies and thus showed an excellent agreement with the literature to which they have been compared. As presented in Figure 3, the experimental data has an average deviation of 4.7% from the equation of Tam and Ghajar [15] and an average deviation of 2.3% from the correlation Test [16].

As shown in Figure 2, in the plain tube, the transitional flow regime is easily identifiable, and thus no need of using the *AGF* method to identify the transitional flow regime in the plain tube. From the smooth tube results, the critical Reynolds number of the friction factor was evaluated as 3 041. This regime progressed until the bulk Reynolds number of 3 292.

The comparison of the present experimental data in the turbulent flow regime is shown in Figure 4 with the previous turbulent correlations. The results showed that the present friction factor experimental data is lower than the one predicted by the equation of Allen and Eckert [17] by 2%. A similar trend was obtained with the correlation of Blasius [18] with an average percentage deviation of 3.5%. It was observed that the deviation increased in the region in which the present data diverted from the turbulent to the transitional flow regime.

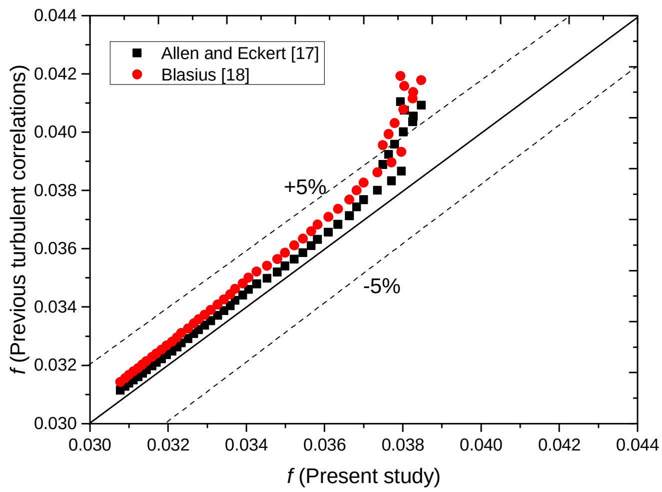

Figure 4 Deviation of the present friction factor experimental data with previous equations in the turbulent flow regime

The excellent agreement obtained in the comparison of the non-isothermal friction factors from the present experimental setup in plain tube heat exchanger with literature has shown that the setup was accurate and could be relied on for the experimental tests with the twisted tape inserts presented in the next section.

RESULTS AND DISCUSSION

The variation of the non-isothermal friction factors results against the bulk Reynolds numbers is presented in Figure 5 for the twisted tape insert with the twist ratio of 5 and at the heat flux boundary condition of 2 kW/m^2.

These friction factors are also compared with the correlation of Manglik and Bergles [19] obtained at uniform wall temperature boundary condition. The present friction factor is lower but compared well with this correlation.

As shown in Figure 5, the discontinuity of the laminar from the turbulent data is not that very obvious compared to the friction factor results in plain tube (in Figure 2). However, linear curve fits were placed on the data as shown to distinguish the three flow regimes. It was concluded from the linear curve fits that the slopes of the data characterising each of the flow regimes were different even as the adjusted R^2 value of each of the regimes was very close to unity. The values of the gradients of the linear curve fits for the laminar, transitional and turbulent flow regimes are evaluated as -0.463, -0.582 and -0.561.

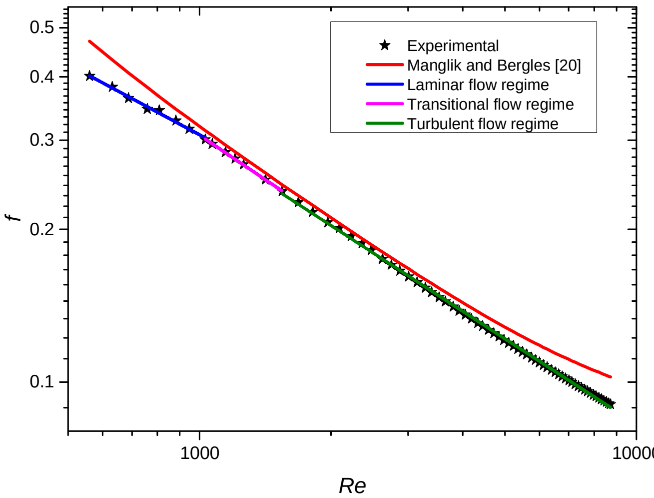

Figure 5 The variation of the friction factors with Reynolds numbers with the twisted tape insert with a twist ratio of 5.

In order to properly distinguish between the three flow regimes in the friction factor results, a method adopted in this study was the evaluation of the area goodness factor. This is the ratio of the heat transfer results in terms of Colburn *j*-factors and pressure drop calculated in terms of the friction factors. This area goodness factor was obtained using Eqn. (4).

The results of the area goodness factor as presented in Figure 6, clearly showed the discontinuation between the data in the laminar flow regime and the data in the turbulent flow regime.

The set of data in-between these two extreme regimes is referred to as transitional flow regime.

In the laminar and transitional flow regimes, the pattern of the data in Figure 6 against Reynolds number, is similar to that of the friction factors in tube tubes. In the laminar flow regime, the area goodness factor increased as the Reynolds numbers reduced. In the transitional flow regime, the area goodness factor increased as the Reynolds numbers increased.

With this method, the critical Reynolds number, which signifies the beginning of the transitional flow regime with the twisted tape insert was obtained at a bulk Reynold number of 1 029, this regime progressed until the bulk Reynolds number of 1 505. The turbulent flow regime then commenced from the end of the transitional flow regime with increased Reynolds numbers and very small decrease in the area goodness factor.

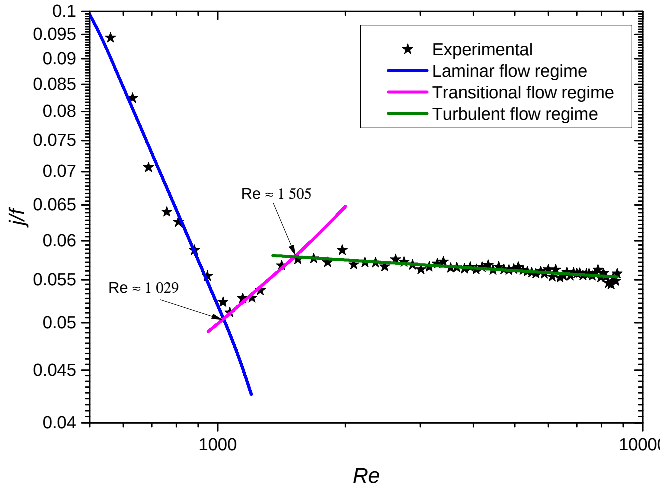

Figure 6 The variation of the area goodness factor with Reynolds numbers and the identification of three flow regimes

It could be seen that the same number of data points were characterised by each of the flow regimes when Figure 5 and Figure 6 are compared. As also shown in Figure 6, it was also possible to linearly curve fit the data in each of the flow regimes. This therefore substantiate the methodology and choice of using the area goodness factor to identify the transitional flow regime with the results of the friction factors in twisted tape inserts.

CONCLUSION

An experimental investigation conducted in a plain copper tube and in a tube with twisted tape insert was reported for the identification of the transitional flow regime. These experiments were conducted under a constant heat flux boundary condition of 2 kW/m^2 using water as working fluid and a square-edge inlet. The experimental setup was validated by comparing the present friction factors in plain tubes with previous correlations. An excellent agreement was obtained with this comparison.

It was possible to use the area goodness factor method to identify the three flow regimes of laminar, transition and turbulent with the friction factor results of the twisted tape insert. The reason for using this method was because, each of the heat transfer and pressure drop data was obtained at the same mass flow rate.

The results also showed that the range of the transitional flow regime with the twisted tape insert was obtained at lower Reynolds numbers compared with that of the plain tube.

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