Numerical Simulation Study on Turbulent Heat Transfer Characteristics of Supercritical Fluid in Micro-Fin Tubes

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Abstract. In consideration of the technical requirements for compact and efficient heat-exchange equipment in industrial fields, such as sludge incineration heat utilization, solar thermal power generation and aerospace, this paper analysed the supercritical nitrogen flow and heat transfer processes in square straight micro-fin tube, triangle straight micro-fin tube and smooth circular tube under different Re numbers, and obtained the changes in velocity field, local turbulence intensity, average Nusselt number and resistance coefficient of supercritical fluid in micro-fin tubes. Moreover, it was found that the introduction of micro fin was able to improve the turbulent heat transfer performance of supercritical fluid in tubes within certain Re number range.

1. Introduction

With the rapid development of sludge incineration heat utilization and solar thermal power generation technologies, heat exchangers are becoming more and more compact with higher performance. Particularly, the heat-exchange equipment that takes supercritical fluid as the heat transfer medium has become a research direction for the heat-exchange equipment in sludge incineration heat utilization. Currently, micro turbulators, such as micro fin, metallic foam and twisted strip, are able to improve the heat transfer performance of heat exchangers, thereby achieving efficient, compact and light weight design objective [1-4].

Micro-fin tube has many short fins of certain spiral angle on its inner surface, which has some geometrical similarities with inner fin tube. Its major parameters include fin height e, spiral angle α and fin number ηf. Since it has some similarities with internal fin tube and spiral inner fin tube, Brognaux [5] defined the typical dimensions of micro-fin tube: 0.02≤e/di≤0.04, 1.52≤p/e≤2.5, 13°≤α≤30°, where: p=πDi/ηf, D<sub>i</sub> represents inner diameter, ηf represents number of circumferential fins, and p/e represents dimensionless fin spacing. Jensen [6] defined the dimensionless parameters of micro-fin tube based on fin heights: e/d<0.05, ηf>30. Hall [7], Jackson [8], Koshizuka [9], Anderson [10], Pioro [11], Chen [12,13], Xu and Guo [14] et al. performed a great number of studies on the heat transfer of supercritical water and CO<sub>2</sub> in vertical and horizontal circular tubes, and obtained many empirical correlations on the basis of Dittus-Boelter correlation. However, such studies and correlations have considerable limitations. There are great differences among the empirical correlations and the theoretical analysis results proposed by different researchers. Even the experiment data obtained by the same researcher under similar conditions is inconsistent. Qin [15-17] et al. of Harbin Institute of...
Technology made study on the performance of liquid hydrogen under supercritical pressure with numerical computation method. In their study, they only considered changes in density under supercritical pressure, and other physical property parameters were only temperature functions. Du et al. studied the heat transfer value of supercritical methane in a vertical tube, including the impact on heat transfer coefficient from mass flow, flow direction and density change.

At present, micro-fin tube has been widely used in refrigeration industry. Nowadays, about 30% of overseas refrigeration and air-conditioning equipment is equipped with micro-fin tubes, the majority of whose fin height is within 0.1-0.2 mm, and number of fins is within 50-60. Micro-fin tube is featuring good heat transfer effect and low flow resistance. However, there are few reports of the studies on the supercritical fluid in small-diameter micro-fin tubes up to now due to its complex flow characteristics and heat transfer process.

To sum up, there are few experimental results and inadequate numerical analysis researches on single-phase convective heat transfer in micro-fin tubes by now, and the mechanism of enhanced heat transfer is still not very clear. For this reason, it is necessary to analyze convective heat transfer of supercritical fluid in micro-fin tubes based on the study results of different researchers, so as to learn the heat transfer mechanism of turbulent flow in micro channels, and perform comprehensive analysis of the impact on the enhanced heat transfer effect and the flow pressure loss in micro-fin circular tubes from all parameters with thermal performance evaluation factor $PEC$, providing the basis for the extended application of micro-fin tubes in enhanced heat transfer field.

2. Numerical Model and Method

2.1. Physical Model

A heat exchange tube with the outer diameter of 2 mm, wall thickness of 0.05 mm and length of 100 mm is selected for numerical simulation study in this paper (as shown in Fig. 1), of which, cold fluid flows from the inside out along the positive direction of Axis Z.

In this paper, the fin height of the micro-fin tube is $h = 0.05\text{-}0.4\text{ mm}$, and the number of fins is $N = 8, 16\text{ and } 32$ respectively.

2.2. Governing Equations

Governing equations includes three-dimensional steady state incompressible continuity equation, Reynolds Average Navier-Stokes equation and energy equation.

Continuity equation:
\[
\nabla \cdot (\rho u) = 0
\]

(1)

Where, $\rho$ represents fluid density and $u$ represents fluid velocity.

Momentum conservation equation:
\[
\frac{\partial}{\partial x_i} \left( \rho u_i u_j \right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ (\mu + \mu_t) \right]
\]

(2)

Energy equation:
\[ \frac{\partial}{\partial x_j} (r \rho u_j e_i) + \frac{1}{r} \frac{\partial}{\partial r} (r \rho u_i e_r) = \frac{\partial}{\partial x} \left[ \lambda_{\text{eff}} \frac{\partial T}{\partial x} \right] + \frac{1}{r} \frac{\partial}{\partial r} \left[ r \lambda_{\text{eff}} \frac{\partial T}{\partial r} \right] \]  

(3)

Where: \( e_i \) represents internal energy of fluid and \( \lambda_{\text{eff}} \) represents effective heat conductivity coefficient.

2.3. Boundary Conditions

Nitrogen inlet velocity in the tube: 1-8 m/s\(^{-1}\), inlet pressure: 100 bar, inlet temperature: 188 K, wall temperature: 300 K, range of \( Re \) number: 16000-152500, outlet: pressure outlet, no slip boundary conditions, and buoyancy lift is ignored. Other settings are as follows:

1) The fluid in the tube is subject to single-phase heat transfer without phase change;
2) The gas in the tube is real gas, and NIST physical property model is used;
3) Gravity influence is ignored;
4) The fluid is under the convective heat transfer condition of fully developed turbulent flow.

Equivalent diameter:

\[ D_h = \frac{4A}{P} \]  

(4)

Where: \( A \) represents flow cross-sectional area/m\(^2\); \( P \) represents wetted perimeter/m.

Average heat transfer coefficient \( h_m \):

\[ h_m = \frac{q}{(T_w - T_f)A} \]  

(5)

Average Nusselt number \( Nu_m \):

\[ Nu_m = \frac{h_m D_h}{\lambda} \]  

(6)

Where, \( q \) represents the heat flux of the entire wall surface/w-m\(^{-2}\), \( T_w \) represents the wall surface temperature/K, \( T_f \) represents the average fluid temperature/K, \( D_h \) represents the hydraulic diameter/m, and \( \lambda \) represents the heat conductivity coefficient of fluid/W-m\(^{-1}\)-K\(^{-1}\).

Comprehensive enhanced heat transfer evaluation factor \( PEC \):

\[ PEC = \frac{Nu / Nu_0}{\sqrt{f / f_0}} \]  

(7)

Where, \( Nu \) represents Nusselt number of micro-fin tube, \( Nu_0 \) represents Nusselt number of smooth tube, \( f \) represents resistance coefficient of micro-fin tube, and \( f_0 \) represents resistance coefficient of smooth tube.

Friction coefficient \( f \):

\[ f = \frac{2 \Delta p D_h}{\rho u^2 L} \]  

(8)

Where, \( \Delta p \) represents the average pressure drop in the computational domain of micro-fin tube/Pa.

3. Experimental Results and Discussions

3.1. Analysis of Heat Transfer Characteristics

Fig. 2 shows the calculation results of the average \( Nu \) number in the micro-fin tube obtained along with changes in Reynolds number \( Re \). According to Fig. 2, \( Nu \) of the square straight micro-fin tube increases obviously along with the increase of \( Re \) number, and it increases by 43.67 \% compared with smooth tube when \( Re \) number is 74000. However, \( Nu \) of the triangle tube increases up to 14.76 \% compared with smooth tube. It can be learnt that the heat transfer effect of the square straight micro-fin tube and the triangle micro-fin tube has been improved significantly when \( Re > 35000 \). In addition, according to the change trend in the figure, the heat transfer effect of the square straight micro-fin tube
and the triangle micro-fin tube has been improved by 9.97% and 9.71% respectively compared with that of the smooth circular tube when $Re<35,000$. Furthermore, $Nu$ of the triangle straight micro-fin tube is 5.52% higher than that of the square straight micro-fin tube, and the heat transfer effect improvement is not obviously when $Re=16000$. With the increase of $Re$ number, improving speed of the heat transfer effect of the triangle straight micro-fin tube decelerates compared with that in the low Reynolds number domain, while the heat transfer effect of the straight micro-fin tube improves obviously. When $Re$ is 150000, the $Nu$ number of the straight micro-fin tube is 1.42 that of the smooth circular tube and 1.36 that of the triangle straight micro-fin tube. Therefore, it can be learnt that the straight micro-fin tube with $N=16$ and $h=0.05$ mm under fully developed turbulent flow is able to improve heat transfer effect significantly, while the triangle micro-fin tube cannot improve heat transfer effect significantly, which indicates that the main reason why the micro-fin tube is able to enhance heat transfer is not the increase of heat transfer area (the difference between the heat transfer area of the triangle micro-fin tube and that of the straight micro-fin tube is 10%).

3.2. Analysis of Flow Characteristics

Fig. 3 shows the changes in resistance coefficient under different $Re$ numbers. According to the figure, resistance loss decreases slowly along with the increase of $Re$ number, but the resistance loss of the smooth circular tube decreases quickly; the frictional resistance deceleration in the micro-fin tube keeps dropping along with the increase of $Re$ number, but the resistance coefficient of supercritical nitrogen in the micro-fin tube barely changes any more when $Re$ number is greater than 94000. Furthermore, the resistance coefficient of the micro-fin tube changes very slowly along with the increase of Reynolds number. When $Re=16000$, the difference between resistance coefficients is small. Fig. 4 is the distribution of the turbulence intensity on the outlet sections of different tubes at $v=2$ m/s and $Re=35000$. Based on the figure, the turbulence intensity of the square straight micro-fin tube is obviously lower than that of the triangle straight micro-fin tube and the smooth tube, the turbulence intensity change at the fin valley is greater than that of the triangle straight micro-fin tube, the contour lines distributed at the fin tip is denser, and the turbulence intensity gradients at the fin valley and the fin height are obviously greater than that of the triangle straight micro-fin tube. The overall turbulence intensity of the smooth circular tube is similar to that of the triangle straight micro-fin tube. The difference is that the turbulence intensity distribution of the smooth circular tube is more uniform, while the turbulence intensity distribution of the triangle micro-fin tube on the wall surface is denser, i.e. the closer to the wall surface, the more drastic the turbulence intensity change is. The turbulence intensity distribution of the square straight micro-fin tube and the triangle straight micro-fin tube is similar to the cross-sectional shape of the fin. To be specific, it changes along with the changes in fin height and fin valley, but the turbulence intensity is smaller, and that at the fin height is closer, which
leads to a larger contact surface between the high-turbulence intensity fluid on the entire cross section and the wall surface, so the heat transfer effect is improved.

![Image](a) Smooth Tube  (b) Square Straight Micro-fin Tube  (c) Triangle Straight Micro-Fin Tube

Fig. 4 Distribution Nephogram of Turbulence Intensity on the Outlet Sections of different Tubes at \(v=2 \text{ m·s}^{-1}\)

3.3. Analysis of Enhanced Heat Transfer Characteristics

Table 1 shows the enhanced heat transfer comprehensive performance evaluation factors under different \(Re\) numbers. It can be learnt from the table that when \(Re<35000\), the \(PEC\) number of the square micro-fin tube is slightly smaller than that of the triangle micro-fin tube with the difference of 0.02 approximately. However, the enhanced heat transfer comprehensive performance evaluation factors (\(PEC\)) of the square straight micro-fin tube and the triangle straight micro-fin tube are keeping improving along with the increase of \(Re\) number. But when \(Re\) number reaches 94000, the \(PEC\) numbers of the triangle micro-fin tube and the square straight micro-fin tube no longer increase, which indicates that the introduction of micro fin is able to improve the turbulent heat transfer performance of the supercritical nitrogen in tubes within certain range of \(Re\) numbers.

**Table 1 Enhanced Heat Transfer Comprehensive Performance Evaluation Factors under different \(Re\) Numbers**

| \(Re\)   | \(PEC\) of square straight micro-fin tube | \(PEC\) of triangle straight micro-fin tube |
|----------|------------------------------------------|-------------------------------------------|
| 16000    | 1.06                                     | 1.08                                      |
| 35000    | 1.16                                     | 1.09                                      |
| 54000    | 1.24                                     | 1.11                                      |
| 74000    | 1.28                                     | 1.08                                      |
| 94000    | 1.28                                     | 1.05                                      |
| 113000   | 1.28                                     | 1.04                                      |
| 133000   | 1.28                                     | 1.04                                      |
| 152500   | 1.28                                     | 1.05                                      |

**Table 2 Logarithm Mean Temperature Differences between Inlet and Outlet under different Fin Shapes**

| \(Nu\)   | Smooth tube  | Triangle  | Straight fin-tube |
|----------|--------------|-----------|-------------------|
| 160.13   | 244.60       | 162.71    |
| 0.03     | 0.04         | 0.04      |
| 1.00     | 1.28         | 1.08      |

4. Conclusions

This paper studied the flow and heat transfer of supercritical nitrogen in micro-fin tubes under different \(Re\) numbers by changing fluid inlet velocity, and analyzed the changes in fluid velocity field, local turbulence intensity, average Nusselt number and resistance coefficient in micro-fin tubes under different \(Re\) numbers. The main conclusions obtained are as follows:
1) For supercritical nitrogen, the $Nu$ number increases along with the increase of $Re$ in the micro-fin tube with the diameter of 2 mm, the average temperature of the fluid in the low $Re$ number domain rises quickly, but the temperature field increases slower and slower in the higher $Re$ number domain.

2) The frictional resistance coefficient of the flowing supercritical nitrogen in micro-fin tubes keeps dropping along with the increase of $Re$ number, and it enters a very gentle domain when $Re$ number reaches 113,000 where the increase of $Re$ number gets less and less along with the changes in resistance coefficient. When $Re=16,000$ in the low Reynolds number domain, the resistance coefficient of the triangle micro-fin tube is almost equal to that of the smooth circular tube, and the dropping speed is very high.

3) However, the enhanced heat transfer comprehensive performance evaluation factors ($PEC$) of the square straight micro-fin tube and the triangle straight micro-fin tube are keeping improving along with the increase of $Re$ number. But when $Re$ number reaches 94,000, the $PEC$ number of the micro-fin tube no longer increases, which indicates that the introduction of micro fin is able to improve the turbulent heat transfer performance of the supercritical fluid in tubes within certain range of $Re$ numbers.

4) The turbulence intensity distribution of the flowing supercritical nitrogen on the cross section in the micro-fin tubes is similar to the cross section shape, namely, it is more concentrated in the area close to the wall surface than in the smooth circular tube, and the gradient change of the turbulence intensity along the fin height is greater than that of the smooth circular tube. With the shape fluctuation of fin height and fin valley, the turbulence intensity in the middle fin valley is less than that on both sides due to fluid viscosity. The main reason why supercritical fluid is able to reinforce heat transfer in micro-fin tubes is that the effective turbulence intensity on the wall surface is increased.

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References
[1] Chen K, Guo L, Xie X, et al. Experimental investigation on enhanced thermal performance of staggered tube bundles wrapped with metallic foam[J]. International Journal of Heat & Mass Transfer, 2018, 122: 459-468.
[2] Chen K, Guo L, Wang H. A review on thermal application of metal foam [J]. Science China Technological Sciences, 2020, https://doi.org/10.1007/s11431-020-1637-3
[3] Chen Kang, Chen Pengfei, Li Wensheng et al. Permeation characteristics of electrodeposited metal foam [J]. Journal of Xi'an Jiaotong University, 2020, 54(5): 87-94.
[4] Chen Kang, Guo Liejin, Zhao Liang et al. Experimental investigation on fluid flow and heat-transfer properties of metallic foam tube-bundle [J]. Journal of Engineering Thermophysics, 2016, 37(4): 770-774.
[5] Brognaux LJ, Webb RL, Chamra LM, et al. Single-phase heat transfer in micro-fin tubes[J]. International Journal of Heat and Mass Transfer. 1997, 40(18): 4345-4357.
[6] Jensen M K, Vlakancic A. Technical Note Experimental investigation of turbulent heat transfer and fluid flow in internally finned tubes[J]. International Journal of Heat & Mass Transfer, 1999, 42(7):1343-1351.
[7] Yoshida S, Mori H, Hong H, et al. Prediction of heat transfer coefficient for refrigerants flowing in horizontal evaporator tubes[J]. Transactions of the Japan Society of Refrigerating and Air Conditioning Engineers, 2011, 11: 67-78.
[8] Hall W B. Heat transfer near the critical point[J]. Advances in Heat Transfer, 1971, 7: 1-86.
[9] Jackson J D, Hall W B, Fewster J, et al. Heat transfer to supercritical pressure fluids[J]. UKAEA, AERER, 1975, 8158.
[10] Koshizuka S, Takano N, Oka Y. Numerical analysis of deterioration phenomena in heat transfer
to supercritical water[J]. International Journal of Heat and Mass Transfer, 1995, 38(16): 3077-3084.

[11] Licht J, Anderson M, Corradini M. Heat transfer to water at supercritical pressures in a circular and square annular flow geometry[J]. International Journal of Heat and Fluid Flow, 2008, 29(1): 156-166.

[12] Pioro I L, Duffey R B. Experimental heat transfer in supercritical water flowing inside channels (survey)[J]. Nuclear engineering and design, 2005, 235(22): 2407-2430.

[13] Chen Tingkuan, Xu Jinliang et al. Study on waterwall hydrodynamics of supercritical vertical tube coils [J]. Power Engineering, 1993,13 (5): 19-25.

[14] Chen Tingkuan, Luo Yushan, Hu Zihong and Yin Fei. Investigation on the heat transfer characteristics of supercritical pressure boiler spirally water wall tube [J]. Journal of Engineering Thermophysics, 2004, 25 (02): 247-250.

[15] Xu Feng and Guo Liejin. Mixed convective flow and heat transfer of water in a helicoidal pipe under supercritical pressure [J]. Journal of Xi'an Jiaotong University, 2005, 39 (09): 978-981.

[16] Bao W, Qin J, Zhou W, et al. Parametric performance analysis of multiple re-cooled cycle for hydrogen fueled scramjet[J]. international journal of hydrogen energy, 2009, 34(17): 7334-7341.

[17] Qin J, Bao W, Zhang S, et al. Comparison During a Scramjet Regenerative Cooling and Recooling Cycle[J]. Journal of Thermophysics and Heat Transfer, 2012, 26(4): 612-618.

[18] Du Zhongxuan, Lin Wensheng, Gu Anzhong and Gu Min. Numerical simulation of cooling heat transfer to supercritical methane in vertical circular tube [J]. Journal of Chemical Industry and Engineering (China), 2009, 60: 63-67.