NUMERICAL INVESTIGATION OF THERMAL HYDRAULIC PERFORMANCE OF AN AUTOMOBILE HEAT TRANSFER TUBE WITH ELLIPSOID DIMPLES

by

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The heat transfer tube is one of the key components affecting the heat transfer performance of automobile radiators. Proposes a new kind of heat transfer tube with ellipsoidal dimples based on the elliptical heat transfer tube. The effects of the arrangement and pitch ellipsoidal dimples on the turbulent heat transfer and flow resistance of the heat transfer tube are further studied by numerical simulation in the range of Reynolds number in 4080-24480. The results show that the ellipsoid dimple arrangement makes the near-wall fluid produce different flow forms, which enhances the turbulence degree of the tube fluid and thus improves the convective heat transfer capacity of the tube. Among them: vertical parallel arrangement (Case 1) causes convergent flow, oblique parallel arrangement (Case 2) causes helical flow, and diagonal symmetrical arrangement (Case 3) causes cross-helical flow. The cross-helical flow causes the most significant degree of turbulence, followed by the helical flow. Furthermore, the arrangement and pitch of the ellipsoidal dimples also have an essential influence on the heat transfer performance and flow resistance of the heat transfer tube. The Nusselt number and friction factor of Cases 1-3 increase successively. But the Nusselt number and friction factor gradually decrease with the rise of the pitch of ellipsoidal dimples. However, under different ellipsoidal dimples pitch, the comprehensive performance of Cases 1-3 is better than that of smooth elliptical tubes. Among them, Case 3 has the best performance when P = 15 mm, and the performance evaluation criteria value is up to 1.39.

Key words: *heat transfer tube, ellipsoidal dimples, numerical simulation, arrangement, pitch*

Introduction

When the engine temperature is too high, the power and torque of the engine will be reduced, resulting in insufficient power. In addition, the piston will expand excessively, and the movement resistance will increase, leading to increased wear and even damage between components. In severe cases, the parts can be burnt and stuck together, and the engine cannot run. The function of the automobile radiator is to transfer the heat from the coolant in the engine to the external environment through the heat transfer tube and fins, and the cooled fluid returns to the engine-it keeps circulating like this-so that the engine can work in a suitable temperature range, thus ensuring the efficient operation of the engine [1-4]. Therefore, as one of the critical

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components of the automobile radiator, improving the heat transfer rate of the heat transfer tube is crucial to the regular operation of the automobile engine. The most effective way to enhance the heat transfer performance of the heat transfer tube is to optimize the structural design of the heat transfer tube to increase the heat transfer area of the heat transfer tube or enhance the turbulence intensity of the fluid [5].

Currently, the tube types widely used in automobile radiators are flat tubes, round tubes, elliptical tubes, waist tubes*, etc.* Still, the heat transfer performance of these smooth tubes is low. Generally, the technologies to improve heat transfer performance can be divided into passive and active ones, and using the protrusions and concaves of the pipe wall or inserting a turbulator in the tube is a passive technology that is easy to apply, because its maintenance cost and required energy are relatively low [6-10]. However, compared with inserting a turbulator into the tube, making dimples on the tube wall is a more effective way to improve the heat transfer performance-because the pressure loss caused by dimples is lower than that of the turbulator. Therefore, many researchers have focused on improving the heat transfer performance by machining dimples on the heat transfer tube wall. Dagdevir *et al.* [11] proposed a circular tube with trapezoidal dimple, fig. 1(a), and the maximum Nusselt number value of the horizontal tube with trapezoidal dimple was 2.1 times that of the traditional circular tube. Xie *et al.* [12, 13] arranged ellipsoidal dimples on the pipe wall in a spiral and cross way, figs. 1(b) and 1(c), and studied the influence of the number, depth and radius of dimples on the heat transfer performance. It is found that since the dimple changes the direction of the fluid, Nusselt number and friction factor, f, increase with the increase of dimple radius and depth, so the performance of traditional heat transfer tubes is significantly enhanced. Wang *et al.* [14] proposed a three-starting torsional tube combined with elliptical dimple, fig. 1(d). It is found that the dimple makes the longitudinal vortex in the tube, which has a significant influence on the fluid mixing. Wang *et al.* [15, 16] and Zheng *et al.* [17] studied spherical dimple tube and ellipsoidal, figs. 1(e) and 1(f). They reported that ellipsoidal dimple tubes had better performance than spherical dimple tubes tube experiments and simulations. Kumar *et al.* [18] designed a heat transfer tube with raised dimples, fig. 1(g). It was found that the Nusselt number and f values of convex tubes are significantly higher than those of traditional tubes. Zheng *et al.* [19] studied the heat transfer tubes with discrete double elliptical cavities by numerical simulation, fig. 1(h). The results show that the vortex generated in the heat exchanger tube has a significant impact on the heat transfer performance of the heat exchanger tube. Xie *et al.* [20] designed a new type of strengthened tube, using indentation and protrusion improve heat transfer, fig. 1(i). The traditional common tube extrusion forming, special reinforced tubes with dents and bumps, it was found that compared with ordinary tubes, the enhancement with concavity and convex has significant advantages in improving heat transfer rate and performance evaluation criteria (PEC) value. Li *et al.* [21, 22] performed experiments and simulations on reinforcement tubes with dimples, fig. 1(j). It was found that the dimple on the pipe wall caused the indentation on the boundary-layer, which led to the periodic separation and mixing of the fluid in the tube.

A large number of studies have found that concave-convex heat exchanger tubes have lower f and higher Nusselt number. However, the studies' large inner diameter and thick wall are unsuitable for automotive automobile radiators with compact requirements. In addition, there is no report on the research of an ellipsoidal dimpled tube applied to an automobile radiator. To improve the performance of automobile heat transfer tubes, dimples are used to elliptical tubes, the heat transfer mechanism and flow characteristics of dimples on the tube wall are studied. The influence of dimple arrangement and pitch on the performance of heat transfer tubes is analyzed, which provides some potential application directions for enhancing the performance of heat transfer tubes.

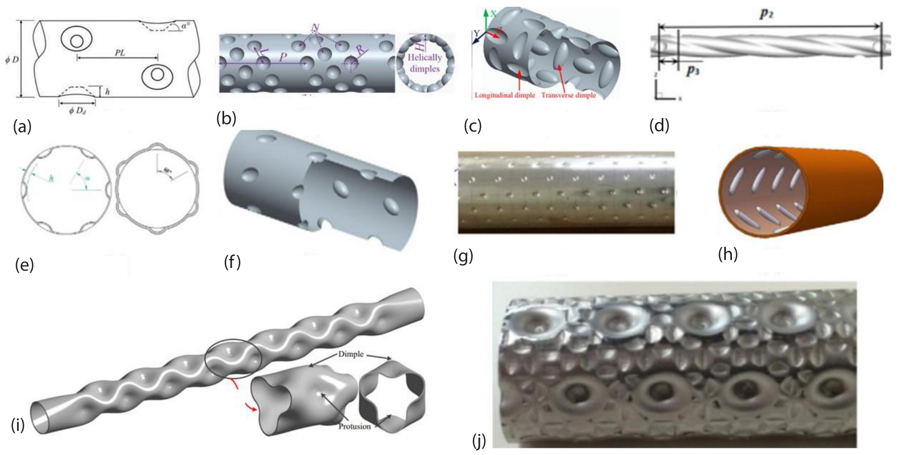

Figure 1. Application of dimples on heat transfer tube: (a) [10], (b) and (c) [11, 12], (d) [13], (e) [14, 15], (f) [16], (g) [17], (h) [18], (i) [19], and (j) [20, 21]

Model description

Physical model and boundary conditions

The geometric model of the smooth elliptical tube and the ellipsoidal dimpled tube is shown in fig. 2. The heat transfer and flow characteristics of elliptical tubes with vertical parallel ellipsoidal dimples (Case 1), inclined parallel ellipsoidal dimples (Case 2), and inclined symmetric ellipsoidal dimples (Case 3) were investigated by using smooth elliptical tubes (Case 0) as a comparison. In addition, the influence of ellipsoidal dimples pitch is also studied. The pitch of ellipsoidal dimples P = 5-25 mm. Ellipsoidal dimples are disposed on smooth elliptical tubes with a major axis of 10 mm and a minor axis of 8 mm. Heat transfer tubes' total length and wall thickness are 600 mm and 2 mm, respectively. The major axis of the ellipsoidal dimple is 4 mm, and the minor axis is 2 mm.

To more accurately simulate the fluid-flow in the tube, the whole tube length area is used as the computational domain in this study. Figure 3 shows the computational domain and boundary conditions of numerical simulation. The coolant inlet is the velocity inlet boundary condition with a temperature of 353 K, the coolant outlet is the pressure outlet boundary condition, and the outer wall temperature of the heat transfer tube is constant at 293 K. In the numerical calculation, liquid water is taken as the coolant, assuming that the fluid is incompressible.

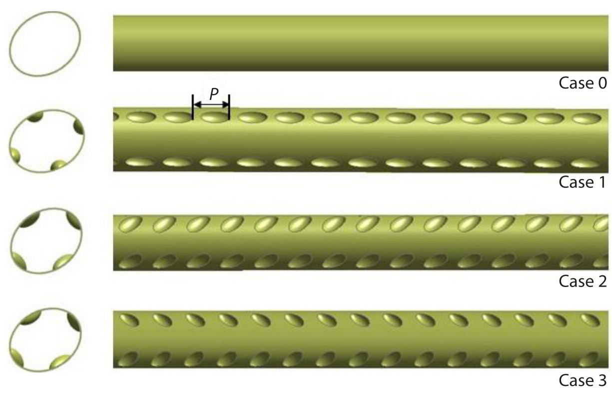

Figure 2. Geometric model of the smooth elliptical tube and ellipsoidal dimpled tube

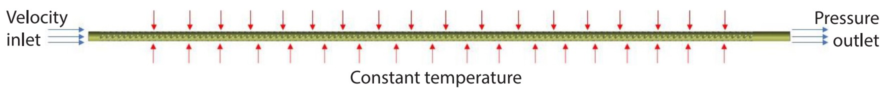

Figure 3. Computational domain

Governing equations

In this study, the commercial software FLUENT 20.0 on CFD was used to solve all equations. The finite volume method was selected to discretize the control equations. The pressure-velocity coupling in steady-state is solved by semi-implicit method for pressure linked equations consistent (SIMPLEC). The diffusion term and convection term are discretized by the QUICK scheme. When the residual of the energy equation solution is less than 10^{-6} and the residual of other equations is less than 10^{-3}, the numerical solution result is considered convergent. The governing equation in the state condition is [23]:

– Continuity equation

$$\frac{\partial}{\partial x_i}\left(\rho u_i\right)=0 \tag{1}$$

– Momentum equation

$$\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right)=\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left(\mu+\mu_\mathrm{t}\right)\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right) \tag{2}$$

– Energy equation

$$\frac{\partial}{\partial x_i}\left(u_i T\right)=\frac{\partial}{\partial x_i}\left[\left(\frac{\mu}{\mathrm{Pr}}+\frac{\mu_\mathrm{t}}{\mathrm{Pr_t}}\right)\frac{\partial T}{\partial x_i}\right] \tag{3}$$

The transport equations for shear stress transport (SST) k-ω model can better predict ellipsoidal dimpled tubes' wall flow and heat transfer. Therefore, the SST k-ω turbulence model simulates this study's computational domain. Equations (4) and (7) are the governing equations of SST k-ω model:

$$\frac{\partial}{\partial t}\left(\rho k\right)+\frac{\partial}{\partial x_i}\left(\rho k u_i\right)=\frac{\partial}{\partial x_j}\left(\Gamma_k\frac{\partial k}{\partial x_j}\right)+\hat{G}_k-Y_k+S_k \tag{4}$$

$$\frac{\partial}{\partial t}\left(\rho \omega\right)+\frac{\partial}{\partial x_i}\left(\rho \omega u_i\right)=\frac{\partial}{\partial x_j}\left(\Gamma_k\frac{\partial \omega}{\partial x_j}\right)+\hat{G}_\omega-Y_\omega+S_\omega+D_\omega \tag{5}$$

where k is the indicates the turbulence kinetic energy, ω – the specific dissipation rate, $\hat{G}_k$ – the turbulence kinetic energy, $\hat{G}_\omega$ – the generation of ω, Γ_k – the effective diffusivity, D_ω – the cross-diffusion, Y_k and Y_ω are the turbulence dissipation of k and ω, respectively, and S_k and S_ω are source terms.

Data reduction

Reynolds number is expressed:

$$\mathrm{Re}=\frac{\rho D_\mathrm{h} u}{\mu} \tag{6}$$

Friction factor, f, is applied to evaluate pressure loss during fluid-flow, and is defined:

$$f = \frac{2\Delta p}{\rho u_i^2}\frac{D_h}{L} \tag{7}$$

The Nusselt number is expressed:

$$\mathrm{Nu} = \frac{hD_h}{\lambda} \tag{8}$$

$$h = \frac{\phi}{A\left(T_{\mathrm{wall}} - T_{\mathrm{ref}}\right)} \tag{9}$$

In addition, while improving the heat transfer performance, the tube often sacrifices the pressure loss. Therefore, while increasing the heat transfer performance, the pressure loss cannot be ignored. The PEC cannot only focus on the heat transfer enhancement, but also take into account the influence of pressure loss. The PEC can be obtained [24]:

$$PEC = \frac{\dfrac{\mathrm{Nu}}{\mathrm{Nu}_0}}{\left(\dfrac{f}{f_0}\right)^{1/3}} \tag{10}$$

The y^+ is the indicator of near-wall mesh resolution that depends on Reynolds number and size of wall-adjacent cell [25]:

$$y^+ = y\left(\frac{u^*}{v}\right) \tag{11}$$

where y^+ is the distance from the wall to centre of wall-adjacent cell, friction velocity u^* can be calculated using:

$$u^* = \left(\frac{f}{8}\right)^{1/2}\overline{v} \tag{12}$$

where f is the friction factor and $\overline{v}$ is the average velocity.

Grid mesh independence study and grid details

Under the condition of Re = 8160, the mesh independence is verified using smooth elliptical tubes. Four models with different mesh numbers (1280893, 1542908, 1837325, and 567740) were studied. Table 1 shows the test results of grid independence. With the increase in the number of grids, the Nusselt number and f gradually decreases, and the minimum errors are 0.1% and 0.2%, respectively. Considering the computer's computing performance and computing time, the grid model with 1837325 grids was selected for numerical simulation, and grid generation methods for other tube types refer to the grid model with grid number of 1837325. Figure 4 shows the details of the polyhedral mesh model of heat transfer tubes with ellipsoidal dimples. The boundary-layer mesh is divided at the interface between the fluid and the solid domain, making the calculation results more accurate. In the selected grid model, note the resultant $y^+ < 1$. It ensures that the mesh size is sufficient to address near-wall effects.

Table 1. Grid independence test

Number of cells	f	Difference [%]	Nu	Difference [%]
1280893	0.0312	10.8%	76.74	3.1%
1542908	0.0336	4%	78.93	0.3%
1837325	0.0349	0.2%	79.12	0.1%
5677400	0.0350	Baseline	79.21	Baseline

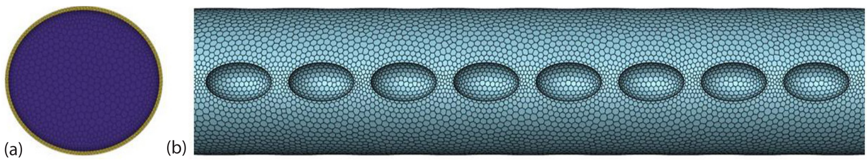

Figure 4. The mesh model of ellipsoidal dimpled tube; (a) inlet section and (b) outside the wall

Results and discussions

Model validation

To ensure the reliability of the numerical method used in this study, the turbulence model used in the numerical simulation should be verified first. It is most appropriate to select the Nusselt number correlation formula of Gnielinski [26] and the f correlation formula of Petukhov [27] to verify the calculation results of smooth elliptic tubes in this study, and the relevant equations of Nusselt number and f are shown in eqs. (13) and (14), respectively. However, the research object of this paper is the elliptic dimple tube proposed based on the smooth elliptic tube. Therefore, to make the numerical method used in this study more convincing, the numerical simulation results of the elliptical dimpled tube must be verified. The most appropriate correlation formula to verify the numerical simulation results of the elliptical dimpled tube is Sabir, *et al.* [28]. The empirical correlation formula for the elliptic dimpled tube is presented in eqs. (15) and (16). The comparison between numerical simulation results and correlation results is shown in figs. 5 and 6, from which it can be seen that the numerical simulation results in this paper are in good agreement with the calculation results of empirical correlation formula. Therefore, the turbulence model used in this paper can accurately predict the performance of ellipsoid dimple tubes:

$$\mathrm{Nu}=\frac{\left(\frac{f}{8}\right)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\left(\frac{f}{8}\right)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)}\left[0.5\le\mathrm{Pr}\le 2000, 3000\le\mathrm{Re}\le 5\cdot 10^{6}\right] \tag{13}$$

$$f=\left(0.79\ln(\mathrm{Re})-1.64\right)^{-0.2}\left[2300\le\mathrm{Re}\le 10^{6}\right] \tag{14}$$

$$\mathrm{Nu}=a\mathrm{Re}^{b\left(1+\frac{0.3\alpha}{2\pi}\right)}\quad\left\{\begin{array}{l}a=-0.0006617\alpha+0.13474]\\ b=0.00051111\alpha+0.7412\end{array}\right\} \tag{15}$$

$$f=\left[a\ln\left(\mathrm{Re}+\frac{\alpha}{2\pi}\right)+b\right]^{-3/2}\quad\left\{\begin{array}{l}a=-0.0079747\alpha+1.2462\\ b=0.059878\alpha-5.3636\end{array}\right\} \tag{16}$$

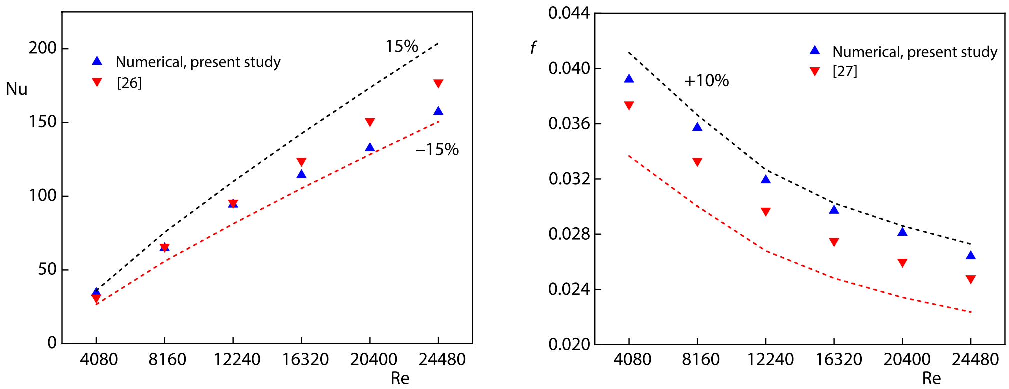

Figure 5. Validation results of smooth elliptical tubes

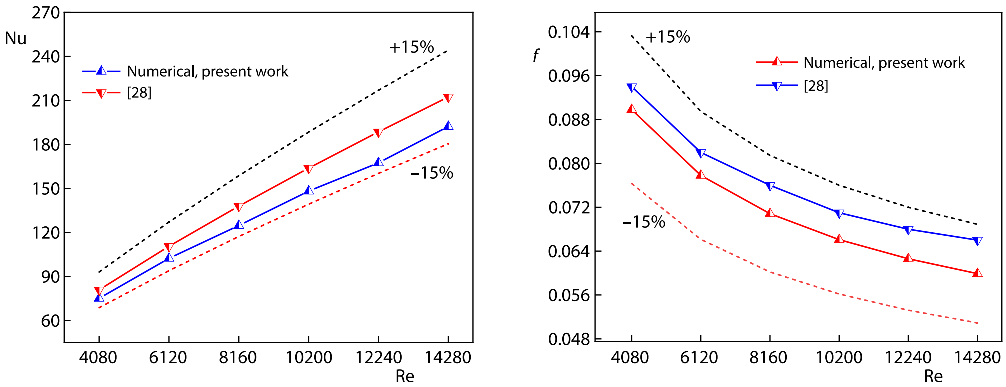

Figure 6. Validation of the results of ellipsoidal dimpled tubes

Comparison of heat transfer performance under different arrangements

Figure 7 shows the temperature contours of the smooth elliptical tube (Case 0) and ellipsoidal dimpled tube with ellipsoidal dimple pitch of 10 mm at Reynolds number of 4080. Wherein, Case 1 is a vertical parallel ellipsoidal dimpled tube, Case 2 is an inclined parallel ellipsoidal dimpled tube, and Case 3 is an inclined symmetric ellipsoidal dimpled tube. When the high temperature liquid water flows through the low temperature tube wall, the average temperature of the liquid water gradually decreases. The fluid temperature in the ellipsoidal dimpled tube will first reduce, and then tend to be stable-compared with the smooth elliptical tube. However, when the ellipsoidal dimple is inclined and symmetrical, the liquid water temperature in the outlet section of the ellipsoidal dimpled tube is the lowest. Therefore, the existence of ellipsoidal dimples enhances the heat transfer performance of the smooth elliptical tube. When the ellipsoidal dimples are inclined and symmetrically arranged around the tube wall, the heat transfer rate is the fastest.

The Nusselt number of the smooth elliptical tube and ellipsoidal dimpled tube vary with different Reynolds number as shown in fig. 8. The pitch of ellipsoidal dimples is 10 mm. With the increase of Reynolds number, turbulence intensity increases, convective heat transfer is amplified, and Nusselt number shows an increasing trend. In addition, under different Reynolds number, the Nusselt number of Cases 1-3 increase in turn and are greater than those

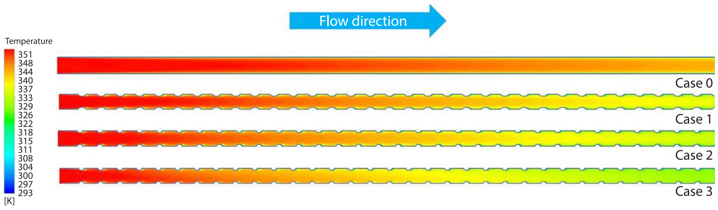

Figure 7. Temperature contours of the middle section of smooth elliptical tube and ellipsoidal dimpled tube (Re = 4080, *P* = 10 mm)

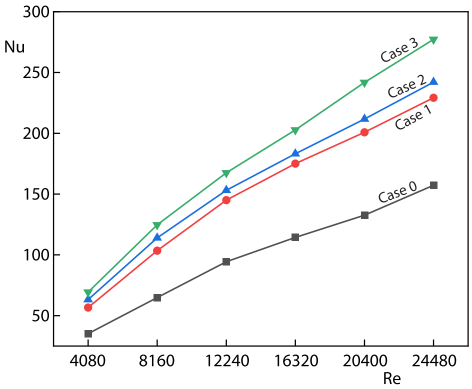

Figure 8. Nusselt number values for smooth elliptical tubes and ellipse fluted tubes (*P* = 10 mm)

of smooth elliptical tubes. When Re = 24480, the Nusselt number of Cases 1-3 ellipsoidal dimpled tube are 61%, 79.6%, and 97.2% higher than those of smooth elliptical tubes (Case 0), respectively. This is because the concave ellipsoidal dimple on the tube wall improves the flow mixing, which makes the fluid in the tube form a periodic impinging stream, and induces the generation of vortex behind the ellipsoidal dimple, thus realizing enhanced heat transfer. In addition, the flow direction of the fluid is affected by the arrangement of the ellipsoidal dimple. The arrangement of Case 1 causes a horizontal secondary flow of the fluid in the tube, and then converges to the centre of the tube. The arrangement of Case 2 causes spiral flow in the tube, while the arrangement of Case 3 causes cross-spiral flow of the fluid in the tube, as shown in the 3-D streamlines of fluid the tube in fig. 9. However, the secondary flow in different directions increases the turbulence intensity of liquid water, thus improving the heat transfer rate. In the three arrangements, the turbulence intensity caused by the inclined symmetrical arrangement is the largest, followed by inclined parallel arrangement, and the vertical parallel arrangement is the lowest. This shows that ellipsoidal dimples can enhance heat transfer rate of heat transfer tube, and the best effect is when the ellipsoidal dimples are arranged symmetrically and obliquely.

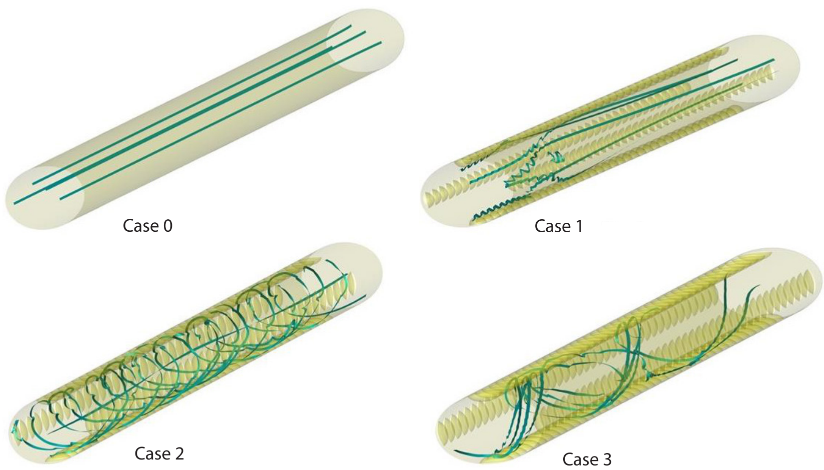

Figure 9. The 3-D streamlines of fluid in the tube

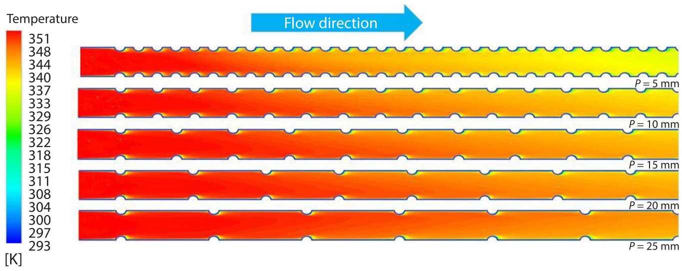

Figure 10. Temperature contours of the middle section of Case 3 (Re = 4080)

Comparison of heat transfer performance under different pitches

The pitch of dimples increases gradually, and the dimple on the tube wall decreases accordingly. To quantify the effect of pitch of ellipsoidal dimples on the heat transfer performance of heat transfer tube, five ellipsoidal dimpled tubes with different pitch sizes were tested. The test pitch of dimples is 5 mm, 10 mm, 15 mm, 20 mm, and 25 mm, respectively. In addition the pitch of ellipsoidal dimples, other geometric parameters such as depth of ellipsoidal dimples, arrangement and shape remain unchanged. The temperature contours of symmetrically arranged ellipsoidal dimple tubes at Re = 4080 is shown in fig. 10. It can be observed from the figure that the high temperature range of the liquid water in the ellipsoidal dimpled tube becomes wider and wider as the pitch of the ellipsoidal dimple gradually increases. This is due to the large pitch of ellipsoidal dimples reducing the turbulence degree of the coolant, thus reducing the heat transfer performance of the ellipsoidal dimpled tube. The Nusselt number changes of Cases 1-3 at different pitches are shown in fig. 11. The Nusselt number decreases with the pitch of the ellipsoidal dimple increasing gradually. At P = 5 mm, Cases 1-3, increased by 17.4%, 30.9%, and 39.5%, respectively, compared with that at P = 25 mm. Therefore, the greater the pitch of the ellipsoidal dimple, the lower the heat transfer performance of the ellipsoidal dimpled tube.

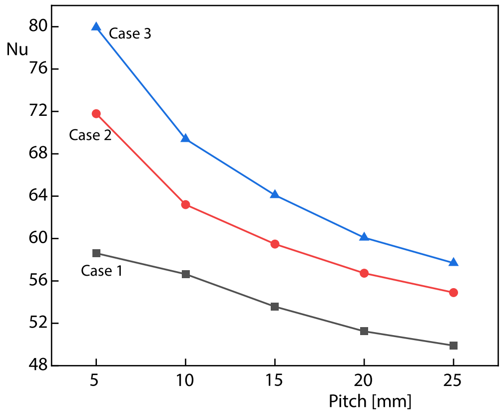

Figure 11. Nusselt numbers of Cases 1-3 at different pitches (Re = 4080)

Comparison of flow characteristic under different arrangements

Figure 12(a) shows the velocity distribution of the middle section of the smooth elliptical tube and ellipsoidal dimpled tube at Re = 4080. At the ellipsoidal dimple tube, the flow rate of liquid water in the tube increases significantly due to the reduction of the equivalent diameter of the tube. However, on the section without ellipsoidal dimples, the velocity decreases. The increase or decrease of velocity leads to the periodic change of speed along the fluid-flow direction, forming a periodic impinging stream, realizing enhanced heat transfer, and leading to a significant pressure drop. It is consistent with the observation of Cheraghi *et al.* [29].

When optimizing the design of heat transfer tubes, the geometry of the heat transfer tubes will increase the friction between the fluid in the tubes and the tube wall, thus changing the pressure drop of the fluid in the tubes and increasing the parasitic power of the cooling sys-

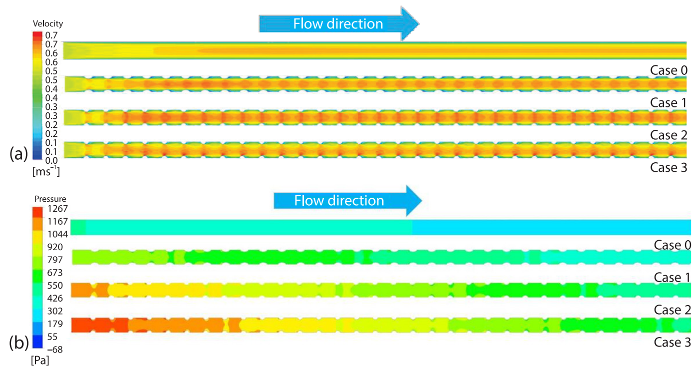

Figure 12. Comparison of smooth oval tube and ellipsoidal dimpled tube; (a) velocity distribution and (b) pressure distribution

tem. In general, the change of cross-section greatly influences the friction between the liquid in the tube and the tube wall. Figure 12(b) shows the pressure contours of the smooth elliptical tube and ellipsoidal dimpled tube. As is shown in figure, the pressure drop in the ellipsoidal dimpled tube is significantly higher than that in the smooth elliptic tube, and the inclined ellipsoidal dimple makes the pressure drop of the heat transfer tube larger. This is because the ellipsoidal dimples on the tube wall reduces the equivalent diameter of the heat transfer tube, which makes the fluid-flow through the ellipsoidal dimple to form vortex, thus making the turbulence degree of the fluid in the tube larger. Figure 13 shows the streamlines of an ellipsoidal dimpled tube, vortices are generated behind the ellipsoidal dimple along the flow direction. In addition, the inclined ellipsoidal dimple changes the flow direction of the fluid, resulting in a more significant pressure drop of the fluid in the inclined ellipsoidal dimpled tubes (Cases 2 and 3). Compared with the inclined parallel ellipsoidal dimples, the inclined symmetric ellipsoidal dimples have greater turbulence intensity. Therefore, the pressure drop of Case 3 is the largest.

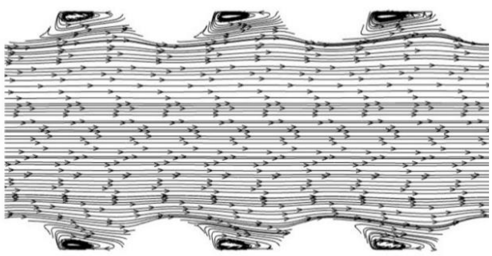

Figure 13. Streamlines of the middle section of Case 3

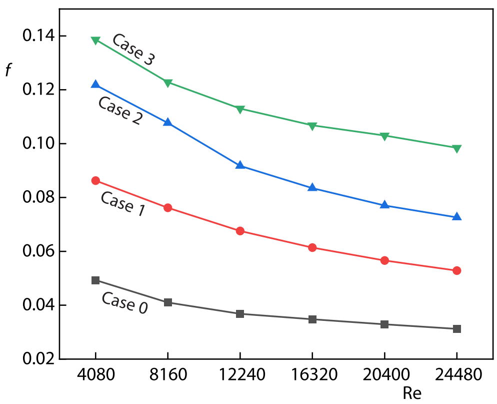

Figure 14. Comparison of *f* of smooth elliptical tubes and ellipsoidal dimpled tube at different Reynolds numbers

The pressure drop is mainly caused by the friction between the liquid water in the tube and the tube wall for smooth elliptical tubes and ellipsoidal dimple tubes. Figure 14 shows the *f* of the smooth elliptical tube and ellipsoidal dimpled tube at different Reynolds number. The pitch of ellipsoidal dimples is 10 mm. Because the ellipsoidal dimple reduces the equivalent diameter of the tube, increases the *f* between the fluid in the tube and the wall, resulting in a more significant pressure drop, the *f* of the ellipsoidal dimpled tube is higher than those of the Case 0. However, the arrangement of ellipsoidal dimples also has a significant impact on the

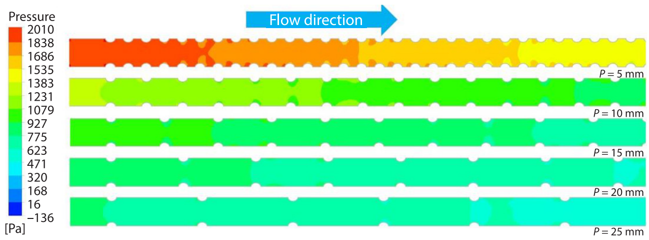

Figure 15. Pressure distribution in the middle section of Case 3 at a different pitch

friction factor of ellipsoidal dimpled tubes. The f of inclined symmetric ellipsoidal dimpled tubes (Case 3) is the largest, followed by inclined parallel ellipsoidal dimpled tubes (Case 2). When Re = 4080, the f of Cases 1-3 ellipsoidal dimpled tube increase by 72%, 140%, and 180%, respectively, compared with smooth elliptical tubes. The results show that the existence of ellipsoidal dimples not only increases the heat transfer performance of the heat transfer tube, but also leads to a greater pressure drop in the tube.

Comparison of flow characteristics under different arrangements

Figure 15 shows the pressure distribution of Case 3 at different pitches when Re = 4080. The areas of high pressure are mainly distributed on the starting position of the ellipsoidal dimple because the ellipsoidal dimple blocks the flow channel. In addition, with the increase of the pitch of the ellipsoidal dimple, the pressure drop of the fluid in the tube decreases gradually. When P = 5 mm, the ellipsoidal dimple is the largest and the pressure drop is the largest. Figure 16 shows the variation of f of Cases 1-3 at a different pitch. With the increase of the pitch of ellipsoidal dimples, the f gradually decreases. This is because with the rise of the ellipsoidal dimple pitch, the ellipsoidal dimple on the heat transfer tube wall must be reduced under the limited tube length, so the number of vortices along the flow direction decreases, reducing the turbulence degree of liquid water in the tube. Therefore, the f at a particular Reynolds number is reduced. When P = 25 mm, the f of Cases 1-3 are reduced by 23.5%, 47.9%, and 59.0%, respectively, compared with P = 5 mm. The results show that although the smaller dimple pitch brings the more significant heat transfer performance, it also increases the parasitic power of the cooling system.

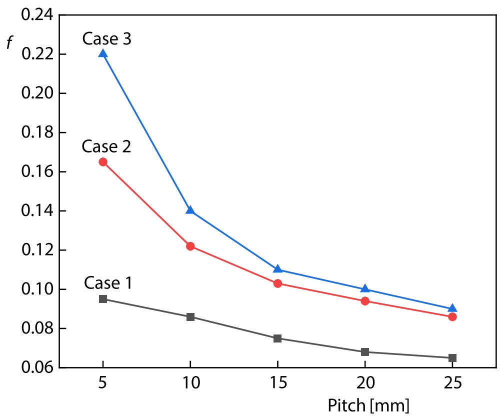

Figure 16. Variation of f of Cases 1-3 at a different pitch

Performance evaluation criteria

While improving the heat transfer performance, the heat transfer tube often sacrifices the pressure loss. Therefore, must seek a balance between heat transfer performance and pressure drop. The PEC is introduced in this paper, as shown in eq. (12). The value of PEC is more than 1, which manifests that the performance of the ellipsoidal dimpled tube is better than that of Case 0. Figure 17(a) shows the PEC values of Cases 1-3 at different Reynolds number. It is observed in the figure that the PEC values of the ellipsoidal dimpled tube are all greater than 1, and that of Case 2 is the lowest. At low Reynolds, Case 3 produces the highest heat transfer performance, so the PEC value of Case 3 is higher than that of Case 1. However, with the increase of Reynolds, the pressure drop caused by inclined symmetrical ellipsoidal dimples is more significant than its contribution heat transfer enhancement. It can be seen that at high Reynolds, the overall performance of Case 1 is superior to Case 3. The Re = 4080, Cases 1-3 all show the best comprehensive performance, which is 33.6%, 32.9%, and 39.7% higher than smooth elliptical tube, respectively. The results show that the ellipsoidal dimpled tube has a great application prospect for improving the comprehensive performance of automobile radiator.

Figure 17(b) shows the PEC values of Case 3 at different ellipsoidal dimple pitches at Re = 4080. The PEC values of Cases 1 and 2 decrease with an increasing pitch of the ellipsoidal dimple. However, PEC value of Case 3 tends to increase when pitch is less than 15 mm. The ellipsoidal dimple pitch tends to decrease when it is larger than 15 mm. This is because at smaller pitches, the increased turbulent intensity caused by inclined symmetrical ellipsoidal dimples has more significant support for heat transfer enhancement than for pressure drop. Conversely, when the pitch of the ellipsoidal dimple is large, the supporting effect on heat transfer enhancement is lower than that on pressure drop.

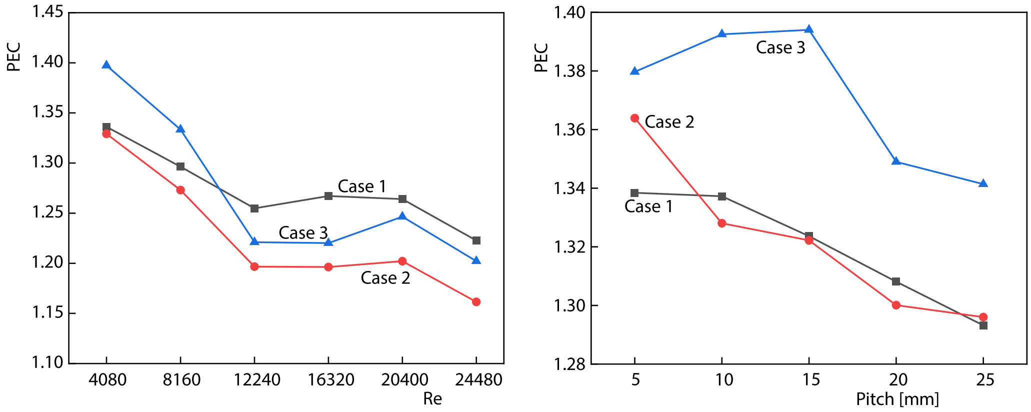

Figure 17. The PEC values of ellipsoidal dimpled tubes; (a) with different Nusselt numbers and (b) with different pitch

Conclusions

This paper studies a new type of automotive heat transfer tube (ellipsoidal dimpled tube) structure through numerical simulation. Three different arrangements of ellipsoidal dimples (vertical parallel, inclined parallel, and symmetrically) were simulated under different Reynolds number and compared with smooth elliptical tubes' heat transfer and flow characteristics. The influence of pitch of ellipsoidal dimple tubes is analyzed. Finally, the PEC of an ellipsoidal dimpled tube under different conditions was compared. The research conclusions are as follows.

- Ellipsoidal dimpled tube has significant advantages in improving heat transfer performance, compared with smooth elliptical tube. The ellipsoidal dimples structure on the tube wall improves the flow state, causing periodic impingement of the fluid in the tube, which induces vortex behind the ellipsoidal dimple, thus realizing heat transfer enhancement.
- The arrangement of ellipsoidal dimples and pitches have an important influence on the heat transfer rate and flow characteristics of heat transfer tubes. Case 3 has the largest Nusselt number and *f*, followed by Cases 2 and 1 the smallest. However, the Nusselt number and *f* of

the ellipsoidal dimpled tube decrease with the increase of the pitch of the ellipsoidal dimple. When Re = 4080 and P = 25 mm, the Nusselt number of Cases 1-3 were 14.8.4%, 23.5%, and 27.8% lower than those of P = 5 mm, respectively, and the f were 23.5%, 47.9%, and 59.0% lower than those of P = 5 mm, respectively.

- Under different pitches of the ellipsoidal dimple, the PEC value of the ellipsoidal dimpled tube is greater than 1, and the ellipsoidal dimpled tube arranged symmetrically and obliquely has the best comprehensive performance. When P = 15 mm, the PEC value reaches 1.39. The results show that the ellipsoidal dimpled tube can significantly improve the comprehensive performance of automobile radiator.

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Nomenclature

A – heat transfer surface area, [m^2]
D_h – hydraulic diameter, [m]
f – friction factor
f_0 – friction factor of smooth elliptic tube
h – heat transfer coefficient, [W^{-2}K^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
L – length, mm
Nu – Nusselt number
Nu$_0$ – Nusselt number of smooth elliptic tube
P – pitch, [mm]
Pr – Prandtl number
Re – Reynolds number
T – temperature, [K]
T_{wall} – wall temperature, [K]
u – velocity, [ms^{-1}]
y^+ – near-wall distance, [m]

Symbols

μ – dynamic viscosity, [Pa·s]
μ_t – turbulent viscosity
ρ – density, [kgm^{-3}]
ϕ – heat flux, [Wm^{-2}]

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