THE INFLUENCE OF CORRUGATED PIPES PARAMETERS ON HEAT TRANSFER CHARACTERISTICS

by

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Original scientific paper
https://doi.org/10.2298/TSCI230709231Z

In order to investigate the influence mechanism of various corrugated structures on the heat transfer of continuous annular concave-convex corrugated pipes, this paper examines the influence of corrugation height (C_h = 1 mm, 1.5 mm, and 2 mm) and corrugation width (C_w = 1 mm, 1.5 mm, and 2 mm) on the flow pattern, turbulent kinetic energy, Nusselt number, friction coefficient, and performance evaluation factor. Then, the correlation equations for Nusselt number and friction coefficient are established with different corrugated structural parameters. The results show that with the increase of C_h, the vortex number, turbulent kinetic energy, and friction coefficient in the pipe increase while Nusselt number decreases. The maximum perfomance evaluation factor is 0.90 at C_h = 1 mm. However, with the increase of C_w, the vortex number and Nusselt number in the pipe increase, while turbulent kinetic energy and friction coefficient in the pipe do not change much. The maximum performance evaluation factor is 0.87 at C_w = 2 mm. Therefore, for this type of corrugated pipe, one should choose a small C_h and a large C_w.

Key words: *corrugated pipes, heat transfer, enhanced heat transfer, numerical simulation*

Introduction

The continuous annular concave-convex corrugated pipes are heat transfer mediums with self-cleaning functions, widely used in heat recovery, waste heat utilization, medical treatment, scientific research, and other fields [1-3]. For example, the heat transfer performance of the cooling tube used in the Synchrotron Radiation double crystal monochromator directly affects the stability of the light emitted [3]. Since the influence mechanism of corrugation-related structural parameters on the heat transfer of corrugated pipes has yet to be specified, it is urgent to optimize its structural parameters to improve the cooling system's heat transfer performance and structural stability.

In the current research, the active heat transfer enhancement technology generally relies on the external stirring and vibration. In contrast, passive heat transfer enhancement technology mainly increases the heat transfer coefficient by changing the structural configuration to enhance heat transfer [4, 5]. Laohalertdecha and Somchai [6] and Ma *et al*. [7] studied the flow, heat transfer, and pressure drop in smooth and corrugated pipes with different refrigerants by changing the

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corrugation pitch and gap number. The results show that the average heat transfer coefficient and pressure drop increase with mass flux and average mass increase. Increasing the number of corrugation gaps can improve friction loss and heat transfer performance more than increasing the number of corrugation pitches. Kown *et al.* [8], Jiang *et al.* [9], and Jaffal *et al.* [10] found that corrugated pipes show significantly better heat transfer performance with larger corrugation angles, higher corrugation heights, and spacing ratio at the Reynolds numbers, between 200 and 1200. Al-Obaidi and Alhamid [11], Al-Obaidi [12], and Al-Obaidi and Alhamid [13] conducted a numerical study to investigate the heat transfer performance of eight corrugated pipes with discontinuous and tiny corrugated structures. It was found that the heat transfer performance of the corrugated pipe increased with decreasing the distance between arc ring, corrugated arc ring angle, and distance between the corrugated ring, while the arc ring angle around the pipe, corrugated pipe ratio, number of the corrugated rings, corrugated diameter ring, and corrugated ring diameter showed the opposite trend. Han *et al.* [14] and Ajeel *et al.* [15] investigated the effect of four special-shaped corrugated structures on the heat transfer performance of corrugated pipes. The results show that the heat transfer performance of symmetrical corrugated pipes is higher than that of asymmetrical corrugated pipes and that optimizing the groove radius and groove profile can maximize the heat transfer performance.

Several researchers [16-21] investigated the heat transfer performance of spiral corrugated pipes with different parameters under constant heat flow and different Reynolds number. The results show that users can choose a high-roughened corrugated pipe when Reynolds numbere is low ($\mathrm{Re} \leq 4000$). When Reynolds number is high ($4000 < \mathrm{Re} < 12000$), it is the opposite. Increasing the concave degree of the pipe wall and the corrugation inclination angle, changing the depth ratio of the thread, and reducing the pitch ratio can increase heat transfer performance. Qi *et al.* [22] studied the heat and mass transfer flow characteristics of TiO_2-water nanofluids in stainless steel corrugated pipes and smooth pipes through experiments and numerical methods. It shows that the combination of corrugated pipes and TiO_2-water nanofluid exhibits excellent heat transfer performance, and the Nusselt number increases with the increase of Reynolds number, while the friction coefficient, f, is gradually decreasing. Some scholars [23-25] combined active and passive heat transfer enhancement techniques to study nanofluids' flow and heat transfer characteristics in different corrugated pipes and smooth pipes. Although the heat transfer performance has been significantly improved, the long-term use of active heat transfer enhancement techniques will cause waste. Changing the corrugation's rib shape and pitch ratio of the spiral corrugation can improve the heat transfer performance of corrugated pipes by destroying the temperature boundary-layer.

Researchers have extensively investigated heat and mass transfer enhancement mechanisms and structural parameters of corrugated pipe heat exchangers with tiny, discontinuous corrugated features [11-16, 23-25]. However, the heat transfer mechanism of continuous annular concave-convex corrugated pipes with large corrugation size configurations still needs to be clarified. Therefore, this research aims to establish a numerical analysis model to improve its heat transfer characteristics in the range of Re = 4000-12000 by changing the corrugation height, C_h, and corrugation width, C_w, of the continuous annular concave-convex corrugated pipes. Then, the correlation equations for Nusselt number and f are established with different corrugated structural parameters.

Corrugated pipe physical model description and boundary conditions

Physical pipes models descriptions

Figure 1 shows the physical model of the corrugated pipe. The length of the pipe is $L = 300$ mm, and the equivalent diameter of the corrugated pipe is $D_\mathrm{h} = 10$ mm. The parameters of the corrugated cooling tube used in the double crystal monochromator in SSRF are $C_h \approx 1.2$ mm and $C_w \approx 1.5$ mm [26]. However, the influence mechanism of the structural parameters of corrugated pipe on its heat transfer performance has not been specified. In order to specify this problem, the corrugation structures with C_h = 1 mm, 1.5 mm, and 2 mm and C_w = 1 mm, 1.5 mm, and 2 mm are chosen. In order to ensure that the fluid entering the corrugated pipes stage develops well into a turbulent flow, the length of the inlet smooth pipe is L_i = 100 mm, and that of the corrugated section is L_c = 180 mm, while the outlet length is L_o = 20 mm, in order to prevent the fluid backflow. It's assumed that the heat exchange only exists in the corrugated section.

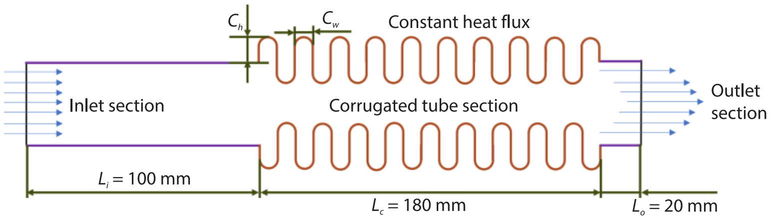

Figure 1. Physical model of corrugated pipe

Boundary conditions

In this numerical calculation, the working fluid is water, Reynolds number is controlled between 4000-12000, and the inlet water temperature is T_i = 293 K. A constant heat flux (Φ_q = 796 W/m^2) is applied on the corrugated section [11-13, 22], while the boundary conditions are velocity inlet and pressure outlet. Zero static pressure is maintained at the inlet and outlet, and a no-slip boundary condition is applied on the pipe wall. Table 1 shows the water's relevant physical parameters and the relevant boundary conditions of the simulation.

Table 1. The parameters of working fluid and boundary conditions

Properties	H_2O	Boundary conditions	
Density [kgm^{-3}]	998.2	Inlet velocity 1 [ms^{-1}]	0.419
C_p [Jkg^{-1}k^{-1}]	4182	Inlet velocity 2 [ms^{-1}]	0.603
Thermal conductivity [Wm^{-1}K^{-1}]	0.6	Inlet velocity 3 [ms^{-1}]	0.804
Viscosity [kgm^{-1}s^{-1}]	0.001003	Inlet velocity 4 [ms^{-1}]	1.005
Molecular weight [–]	18.0152	Inlet velocity 5 [ms^{-1}]	1.206
Standard state enthalpy [kJmol^{-1}]	285.830	Constant heat flux [Wm^{-2}]	796

Numerical calculation model

Computational analysis model

This numerical analysis calculation was executed with the steady-state solver in FLUENT software, using the RNG *k-e* turbulence model [11-13]. The convergence scales of the continuity, momentum, and turbulence equations are all at the magnitude of 10^{-6}, while the convergence scale of the energy equation is at the magnitude of 10^{-7}. Turbulent kinetic energy (TKE) and energy second-order upwind discrete equations are adopted. The relevant control equations can be seen in references [12, 13], which will not be detailed here.

Mesh generation and independence test

HyperMesh software is used to divide the asymmetrically structured mesh, as shown in fig. 2. It's necessary to conduct boundary-layer meshing at the walls to ensure the calculation accuracy.

Figure 3 shows the comparative analysis of the mesh independence verification of corrugated pipes meshes with mesh numbers of 3.3×10^6, 4.4×10^6, 5.5×10^6, 6.6×10^6, 7.7×10^6, 8.8×10^6, and keeping the Y^+ value of the wall less than 1. The objects of the verification f and Nusselt number. When the mesh number reaches to 7.7×10^6, the mesh can be considered independent, while the errors of f and Nusselt number are 1.612 % and 0.282 %, respectively.

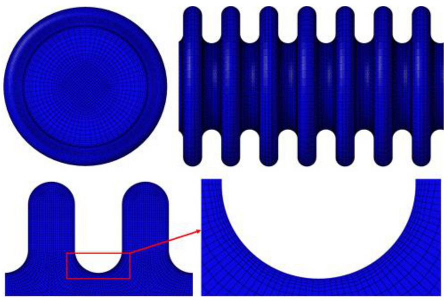

Figure 2. Mesh model of corrugated pipe

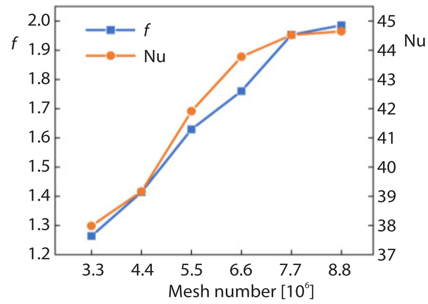

Figure 3. Mesh independence verification

Data reduction

The Reynolds number is a dimensionless number that can be used to characterize fluid-flow conditions, which can be expressed [27]:

$$\mathrm{Re} = \frac{\rho D_\mathrm{h} u}{\mu} \tag{1}$$

The heat transfer coefficient is defined:

$$h = \frac{\Phi}{A\left(T_\mathrm{wall} - T_\mathrm{ref}\right)} \tag{2}$$

where Φ, A, T_wall, and T_ref are the total heat rate, heating area, local wall temperature of the wall at the corrugation, and the overall temperature at the corrugation, respectively.

The local Nusselt number and f can be expressed:

$$\mathrm{Nu} = \frac{h D_\mathrm{h}}{\lambda} \tag{3}$$

$$f = \frac{2 D_\mathrm{h}}{\rho u^2} \frac{\Delta P}{\Delta L} \tag{4}$$

where λ is the thermal conductivity of the fluid and $\Delta P/\Delta L$ – the pressure drop per unit length.

The performance evaluation factor (PEF) is a significant parameter [27]:

$$PEF = \frac{\dfrac{\mathrm{Nu}}{\mathrm{Nu}_0}}{\left(\dfrac{f}{f_0}\right)^{1/3}} \tag{5}$$

where f_0 and Nu_0 are the smooth pipe's friction coefficient and Nusselt number.

Numerical validation

In order to verify the accuracy of the numerical analysis, Nusselt number and f of the corrugated pipes were compared with that of the experimental data from Q_i [22], measured at Reynolds number between 4000-12000. The numerical results in this research show good agreement with the experimental data of Q_i, which can be seen in fig. 4. The maximum deviation of Nusselt number is 4.5%, mainly occurring at Re = 5600, while the maximum deviation of f occurs at Re = 7000 with the value of 5.6 %.

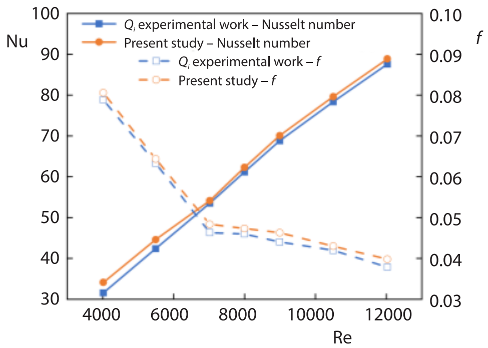

Figure 4. Numerical validation

Results and discussion

This section studies the effects of different corrugated structures parameters on the heat transfer of corrugated pipes, and the flow in the pipe, TKE, Nusselt number, f, and PEF are discussed completely. Firstly, the effect of C_h on heat transfer of corrugated pipes is investigated, C_w is fixed at 1.5 mm according to the size of the corrugated pipes used in Synchrotron Radiation [27]. Then, according to the parameter control method, C_h is fixed at 1.5 mm while studying the influence of C_w on the heat transfer performance of the corrugated pipes. Finally, the correlation equations for Nusselt number and f are established.

Effect of corrugation height

Figure 5 shows the CFD results of the pipe-line's velocity distributions and streamlines. With the increase of C_h, the velocity of the main flow area in the pipe-line will also increase. The flow velocity in the main flow area increases with the increase of pressure drop per unit length. The vortex will form inside the corrugated ring with different C_h, and a little secondary flow will form at the transition connection of the concave and convex corrugations. The continuous concave and convex corrugations inside the pipe change the flow direction from axial flow to radial flow. Thus, the size and number of vortexes also show differences with the increment of C_h, which can be seen in fig. 5. When C_w is determined, the larger C_h, there will be more vortexes. These vortexes and the secondary flow are also essential to improve the heat transfer characteristics of the corrugated pipes.

The TKE contour distribution map is shown in fig. 6 when C_h is 1 mm, 1.5 mm, and 2 mm, respectively. It can be observed that in the flow process of fluid, TKE in the pipe increases with the increase of C_h. The TKE at the corrugated connections is more significant than that in the main flow area, which means that the liquid between the corrugated ring area and the main flow area is more likely to produce circular flow and liquid mixing. Due to the liquid-flow direction, TKE near the right side of the corrugation is more significant than that on the left side. The impact of the liquid on the wall causes a vortex, resulting in a larger TKE. This phenomenon also disrupts the thermal boundary-layer, increasing the value of f and Nusselt number of fluid-flow in the pipe. When C_h is 2 mm, the maximum value of TKE is about 0.44 J/kg.

Figure 7(a) compares the local Nusselt number of the corrugated pipe and the smooth pipe when C_h is 1 mm, 1.5 mm, and 2 mm. It can be seen that Nusselt number increases as Reynolds number increases. At the same time, the local Nusselt number increases as C_h decreases. When C_h is 2 mm, Reynolds number is in the range of 4000-10000, Nusselt number is lower than that of the smooth pipe, while Reynolds is in the range of 10000-12000, Nusselt

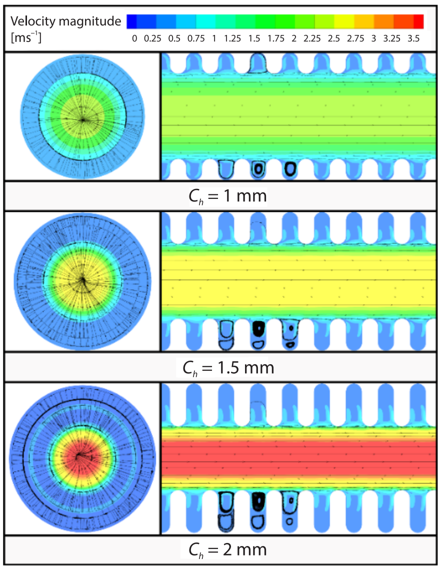

Figure 5. Velocity distributions with different C_h

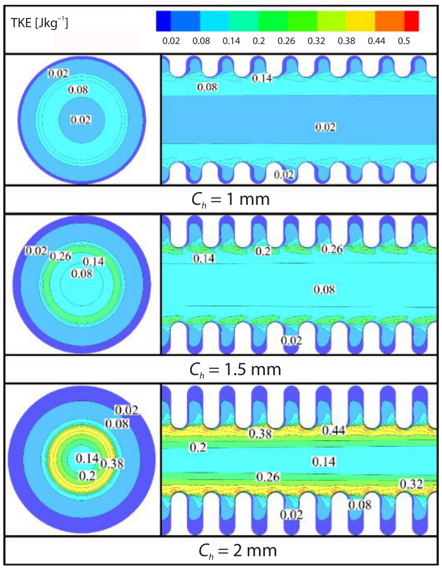

Figure 6. The TKE contour distribution map with different C_h

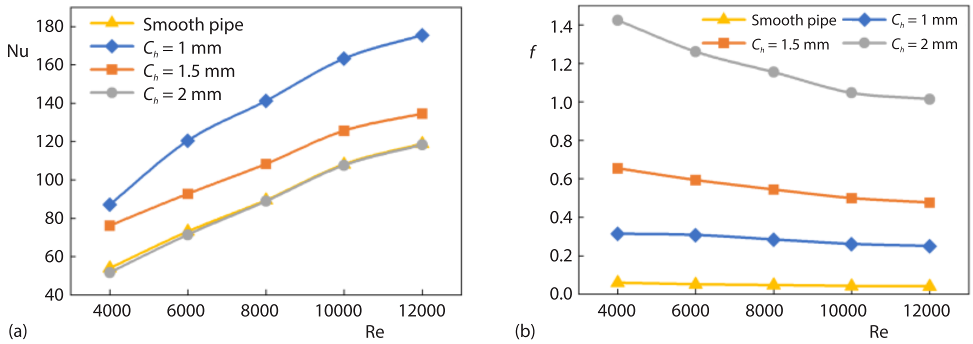

Figure 7. The influence of C_h on Nusselt number and *f*; (a) Nusselt number and (b) *f*

number is the same as that of the smooth pipe. When C_h is 1 mm and 1.5 mm, the difference of Nusselt number between corrugated pipes and smooth pipes is 39.8% and 7.5%, respectively. As C_h of the corrugated pipe increases, its heat transfer performance is reduced. Therefore, it is necessary to control C_h within a specific range.

Another critical factor describing convective-enhanced heat transfer is *f*, as shown in Figure 7(b). It can be seen that *f* has a certain degree of attenuation with the increase of Reynoilds number, while *f* increases as C_h increases since the corrugated structure will increase the pressure loss along the pipe-line. All the value of *f* under three different C_h are much higher than those of the smooth pipe, which are 513%, 1074%, and 2361% of the smooth pipe, respectively. It suggests that the increase of secondary flow and vortex at the corrugation leads to more energy dissipation, which increases *f*.

The PEF is used to evaluate the thermal performance of the corrugated pipe with different C_h. When PEF is greater than 1, it indicates that the enhanced heat transfer performance of the corrugated pipe with this structure parameter is better than that of the smooth pipe under certain boundary conditions, not *vice versa* [27]. As shown in fig. 8, when C_h is 1 mm, 1.5 mm, and 2 mm, PEF are all less than 1, which means that the heat transfer performance of the corrugated structure is not as good as that of the smooth pipe. However, when C_h is 1 mm and Reynolds number is 4000, PEF is about 0.90. It can be concluded that appropriate reduction of C_h may increase PEF.

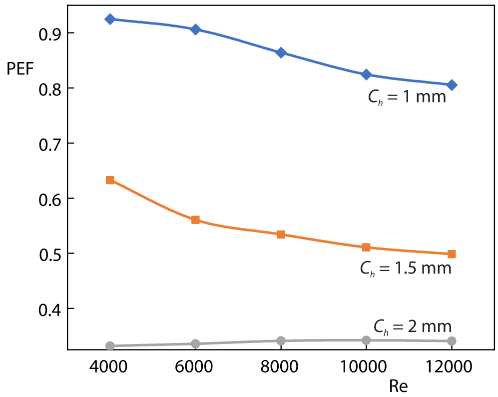

Figure 8. The value of PEF with different C_h

Effect of corrugation width

The CFD results of the velocity flow contour with the streamlines field in the pipeline is shown in fig. 9. It can be seen that when C_w changes, the velocity distribution does not change much. The reason is that the volume of the main flow region is unchanged, when C_w changes, its internal velocity distribution is unchanged, while its value is very high. The velocity near the corrugation is low. The vortex will form in the corrugated ring by the fluid-flowing through the corrugated ring striking the corrugated wall. The number of vortexes in the ring decreases as C_w increases. The vortexes close to the main flow area will stick to the wall due to the liquid-flow in the main flow area. When C_w is 1 mm and 1.5 mm, new vortexes will form on the outer side of the ring near the outer wall due to the TKE. The formation of these vortexes mainly relies on the energy provided by the central vortex close to the main flow area. Moreover, it will produce a particular wall-cleaning effect.

The contour map distribution of TKE inside the corrugated pipes with three different C_w is shown in fig. 10. It can be seen that the distribution of TKE with different C_w is consistent during axial flow, the value near the corrugated ring is high, while the value on the right side of

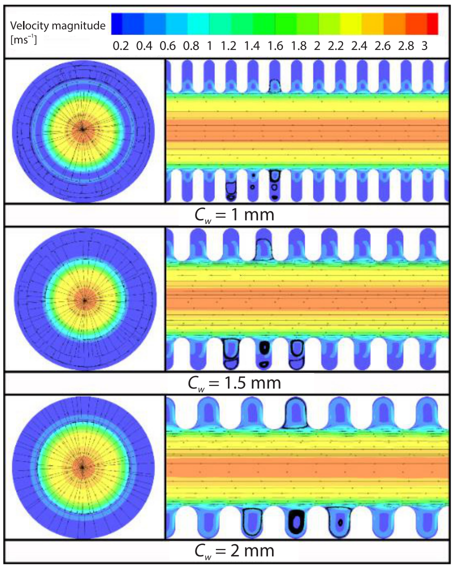

Figure 9. Velocity distribution in pipe with different C_w

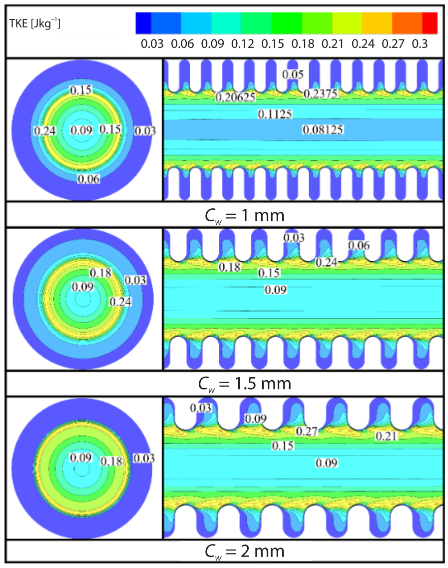

Figure 10. The TKE contour distribution with different C_w

the corrugated ring is higher than that on the left side which is caused by liquid striking the wall. Where TKE is high in the pipe, it is easier to form vortexes and flow circulation, destroying the thermal boundary-layer to improve the heat transfer performance. With the increase of C_w, the TKE value also increases, and the maximum value of TKE is about 0.27 J/kg when C_w is 2 mm.

Figure 11(a) compares the local Nusselt and Reynolds numbers for smooth and corrugated pipes with different C_w. It can be seen that Nusselt number increases as Reynolds number increases. When C_w is 2 mm, it has a higher Nusselt number. As C_w increases, Nusselt number also increases. Nusselt number of corrugated pipes is higher than that of smooth pipes, where Nusselt number of corrugated pipes with C_w of 1.5 mm and 2 mm are 7.5% and 40.6% higher than that of smooth pipes, respectively. However, when C_w is 1 mm, Nusselt number is lower than that of the smooth pipe. It shows that its heat transfer performance could be better than a smooth pipe at a given Reynolds number. Increasing C_w will improve the heat transfer performance.

It can be seen from fig. 11(b) that the value of f changes with Reynolds number and generally presents a downward trend for both corrugated pipes with different C_w and smooth pipes. In addition, it can be noticed that the value f of the corrugated pipes with different C_w are all higher than that of smooth pipes. When C_w increases, f also increases. However, the difference between the corrugated pipes with different C_w is 1028.7%, 1075.7%, and 1089.8% compared with the smooth pipes. Generally, the increase in C_w has little effect on f.

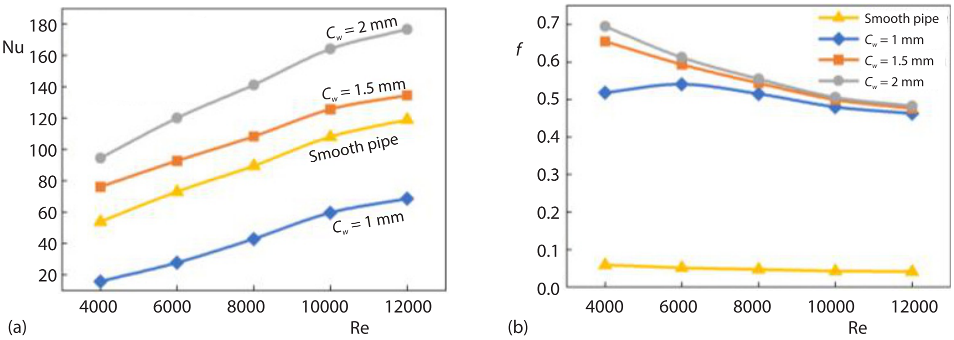

Figure 11. The influence of C_w on Nusselt number and f; (a) Nusselt number and (b) f

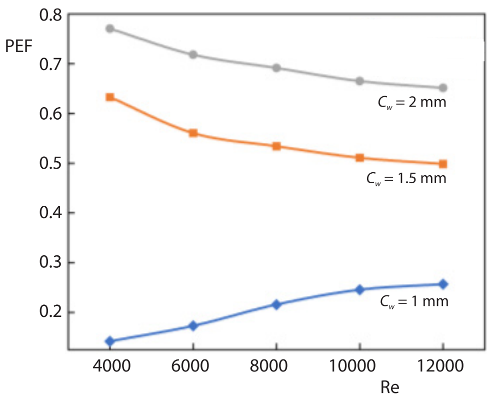

Figure 12. The PEF distribution with different C_w

Figure 12 illustrates the PEF of corrugated pipes with different C_w to evaluate the improvement of their thermal performance. The results show that C_w have a significant impact on the heat transfer performance. The PEF becomes larger as C_w increases, especially when Reynolds number is 4000, where the highest value reaches 0.77. In this case, the damage to the thermal boundary-layer of the corrugated pipes will be more severe. Therefore, when C_h is fixed, appropriately increasing C_w can improve PEF.

For the cooling corrugated pipes used in Synchronous Radiation Facility, it should be selected with a higher C_h and a smaller C_w. Since the PEF is less than 1, the friction loss increases in the pipe while the heat transfer coefficient decreases, which can avoid phase change of the coolant due to thermal load. For the heat transfer pipe-line of the monocrystalline silicon cooling system, a corrugated pipe with a C_h less than 1 mm and a C_w more than 2 mm is suggested. Because the PEF is greater than 1, its heat transfer performance is better than that of the smooth pipe, which can effectively improve the cooling efficiency.

Development of correlation equations for Nusselt number and friction coefficient

In order to study the influence of Reynolds number, C_h, and C_w on the heat transfer performance of corrugated pipes, the results of CFD numerical simulation were analyzed by the multiple non-linear regression, and the correlation equations for Nusselt number and f are established:

$$\mathrm{Nu} = 1.0951(\mathrm{Re})^{0.6296}\left(\frac{C_h}{D_h}\right)^{-0.73533}\left(\frac{C_w}{D_h}\right)^{1.333} \tag{6}$$

$$f = 1258.0649(\mathrm{Re})^{-0.29139}\left(\frac{C_h}{D_h}\right)^{2.4956}\left(\frac{C_w}{D_h}\right)^{0.19969} \tag{7}$$

The valid range of this relationship is within Reynolds number (4000-12000), corrugation height (C_h = 1-2 mm), and corrugation width (C_w = 1-2 mm). Moreover, the accuracy of the correlation coefficient, R^2, is above 94% and 95%. Figures 13 and 14 show the errors of Nusselt number and f, which are ±12.4% and ±13.3% compared with CFD results, respectively.

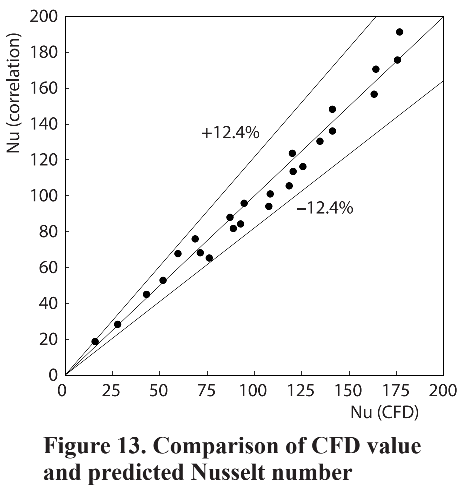

Figure 13. Comparison of CFD value and predicted Nusselt number

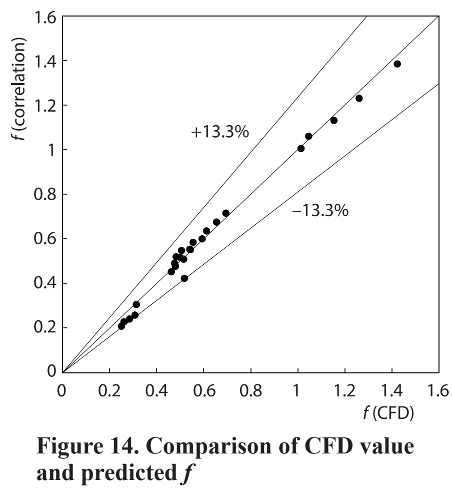

Figure 14. Comparison of CFD value and predicted *f*

Conclusions

- Increasing C_h and C_w will increase TKE. The main reason is that the liquid-flow strikes the wall of the corrugated pipes to form a large vortex, which increases TKE at the bottom connection of the corrugated ring and destroys the thermal boundary-layer.
- Nusselt number decreases with the increase of C_h. When C_h is 2 mm, Nusselt number is smaller than that of the smooth pipe, and when C_h is 1 mm and 1.5 mm, Nusselt number is more significant than that of the smooth pipe, with a difference of about 39.8 % and 7.5%. Therefore, C_h can be appropriately reduced to improve the heat transfer performance.

- Increasing C_w has little effect on f. When C_w is 1 mm, its heat transfer performance is not as good as that of a smooth pipe, and when C_w is 1.5 mm and 2 mm, Nusselt numner is larger than that of the smooth pipes, the differences are 7.5% and 40.6%, respectively. Therefore, C_w can be appropriately enlarged to improve the heat transfer performance.
- When C_h is 1 mm, 1.5 mm, and C_w is 1.5 mm, 2 mm, the PEF decreases with the increase of Reynolds number. It is found that in the low Reynolds number range, when C_h is 1 mm, PEF is 0.90. When C_w is 2 mm, PEF is 0.77. The accuracy of the equations related to Nusselt number and f reaches to 94% and 95%.

Acknowledgment

The authors appreciate greatly the financial support from the Open Fund of State Key Laboratory of Applied Optics, China (No. SKLAO2020001A09), and the Key Research and Development Plan of Shaanxi Province, China (No. 2023-YBGY-346).

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Paper submitted: July 9, 2023
Paper revised: September 18, 2023
Paper accepted: September 27, 2023