THE INFLUENCE OF CORRUGATED PIPES PARAMETERS ON HEAT TRANSFER CHARACTERISTICS

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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 \( (C_h = 1, 1.5 \text{ and } 2 \text{ mm}) \) and corrugation width \( (C_w = 1, 1.5 \text{ and } 2 \text{ mm}) \) on the flow pattern, turbulent kinetic energy (TKE), Nusselt number (Nu), friction coefficient (f), and performance evaluation factor (PEF). Then, the correlation equations for Nu and f are established with different corrugated structural parameters. The results show that with the increase of \( C_h \), the vortex number, TKE, and f in the pipe increase while Nu decreases. The maximum PEF is 0.90 at \( C_h = 1 \text{ mm} \). However, with the increase of \( C_w \), the vortex number and Nu in the pipe increase, while TKE and f in the pipe do not change much. The maximum PEF is 0.87 at \( C_w = 2 \text{ mm} \). 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

1. 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]. Suriyan L. and Ma et al. [6-7] studied the flow, heat transfer, and pressure drop in smooth and corrugated pipes with different refrigerants by changing the 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 and Jiang et al. [8-10] found that corrugated pipes show significantly better heat transfer performance with larger corrugation angles, higher corrugation heights, and spacing ratio at the Reynolds numbers \((Re)\) between 200 and 1200. Al-Obaidi A.R. [11-13] 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 and R.K. Ajeel et al. [14] 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 \(Re\). The results show that users can choose a high-roughened corrugated pipe when \(Re\) is low (\(Re \leq 4000\)). When \(Re\) is high (\(4000 < Re < 12000\)), 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 \((Nu)\) increases with the increase of \(Re\), while the friction coefficient \((f)\) is gradually decreasing. Some scholars [23] 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. 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 \(Nu\) and \(f\) are established with different corrugated structural parameters.

2. Corrugated pipe physical model description and boundary conditions

2.1. 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 $D_h = 10$ 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 $C_h = 1, 1.5, \text{ and } 2 \text{ mm}$ and $C_w = 1, 1.5, \text{ and } 2 \text{ mm}$ 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.

2.2. Boundary conditions

In this numerical calculation, the working fluid is water, $Re$ is controlled between 4000-12000, and the inlet water temperature is $T_i = 293$ K. A constant heat flux ($\Phi_q = 796 \text{ W/m}^2$) 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>
<thead>
<tr>
<th>Properties</th>
<th>H₂O</th>
<th>Boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density [kgm⁻³]</td>
<td>998.2</td>
<td>Inlet velocity 1 [ms⁻¹]</td>
</tr>
<tr>
<td>$C_p$ [Jkg⁻¹K⁻¹]</td>
<td>4182</td>
<td>Inlet velocity 2 [ms⁻¹]</td>
</tr>
<tr>
<td>Thermal conductivity [Wm⁻¹K⁻¹]</td>
<td>0.6</td>
<td>Inlet velocity 3 [ms⁻¹]</td>
</tr>
<tr>
<td>Viscosity [kgm⁻¹s⁻¹]</td>
<td>0.001003</td>
<td>Inlet velocity 4 [ms⁻¹]</td>
</tr>
<tr>
<td>Molecular weight [-]</td>
<td>18.0152</td>
<td>Inlet velocity 5 [ms⁻¹]</td>
</tr>
<tr>
<td>Standard state enthalpy [kJmol⁻¹]</td>
<td>285.830</td>
<td>Constant heat flux [Wm²]</td>
</tr>
</tbody>
</table>

3. Numerical calculation model

3.1. 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 $10^{-5}$, 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.

3.2. 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 $3.3 \times 10^6$, $4.4 \times 10^6$, $5.5 \times 10^6$, $6.6 \times 10^6$, $7.7 \times 10^6$, $8.8 \times 10^6$, and keeping the Y+ value of the wall less than 1. The objects of the verification $f$ and $Nu$. When the mesh number reaches $7.7 \times 10^6$, the mesh can be considered independent, while the errors of $f$ and $Nu$ are 1.612 % and 0.282 %, respectively.

![Figure 2. Mesh model of corrugated pipe](image)

![Figure 3. Mesh independence verification](image)

3.3. Data reduction

$Re$ is a dimensionless number that can be used to characterize fluid flow conditions, which can be expressed as [27]:

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

(1)

The heat transfer coefficient is defined as:

$$h = \frac{\Phi}{A(T_{wall} - T_{ref})}$$

(2)

Among which, $\Phi$, $A$, $T_{wall}$, 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 $Nu$ and $f$ can be expressed as:

$$Nu = \frac{hD_{h}}{\lambda}$$

(3)

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

(4)

Where, $\lambda$ represents the thermal conductivity of the fluid.

$\frac{\Delta P}{\Delta L}$ is the pressure drop per unit length.

The performance evaluation factor (PEF) is a significant parameter [27]:
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\[ PEF = \frac{Nu / Nu_0}{(f / f_0)^{1/3}} \]  

(5)

\( f_0 \) and \( Nu_0 \) represent the smooth pipe's friction coefficient and Nusselt number.

4. Numerical validation

In order to verify the accuracy of the numerical analysis, \( Nu \) and \( f \) of the corrugated pipes were compared with that of the experimental data from Qi [22], measured at \( Re \) between 4000 and 12000. The numerical results in this research show good agreement with the experimental data of Qi, which can be seen in Fig. 4. The maximum deviation of \( Nu \) 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](image)

5. 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, \( TKE, Nu, f, \) 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 \( Nu \) and \( f \) are established.

5.1. Effect of corrugation height (\( C_h \))

Figure 5 shows the CFD results of the pipeline's velocity distributions and streamlines. With the increase of \( C_h \), the velocity of the main flow area in the pipeline 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, 1.5, 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 $Nu$ of fluid flow in the pipe. When $C_h$ is 2 mm, the maximum value of TKE is about 0.44 J/kg.

**Figure 5.** Velocity distributions with different $C_h$

**Figure 6.** TKE contour distribution map with different $C_h$

Figure 7(a) compares the local $Nu$ of the corrugated pipe and the smooth pipe when $C_h$ is 1, 1.5, and 2 mm. It can be seen that $Nu$ increases as $Re$ increases. At the same time, the local $Nu$ increases as $C_h$ decreases. When $C_h$ is 2 mm, $Re$ is in the range of 4000-10000, $Nu$ is lower than that of the smooth pipe, while $Re$ is in the range of 10000-12000, $Nu$ is the same as that of the smooth pipe. When $C_h$ is 1 mm and 1.5 mm, the difference of $Nu$ 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 Fig. 7(b). It can be seen that $f$ has a certain degree of attenuation with the increase of $Re$, while $f$ increases as $C_h$ increases since the corrugated structure will increase the pressure loss along the pipeline. 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 $f$.

**Figure 7.** The influence of $C_h$ on $Nu$ and $f$: (a) Nusselt number ($Nu$); (b) friction coefficient ($f$)
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, 1.5, 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 $Re$ 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](image)

5.2. Effect of corrugation width ($C_w$)

The CFD results of the velocity flow contour with the streamlines field in the pipeline are 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 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 $Nu$ and $Re$ for smooth and corrugated pipes with different $C_w$. It can be seen that $Nu$ increases as $Re$ increases. When $C_w$ is 2 mm, it has a higher $Nu$. As $C_w$ increases, $Nu$ also increases. $Nu$ of corrugated pipes is higher than that of smooth pipes, where $Nu$ 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, $Nu$ 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 $Re$. Increasing $C_w$ will improve the heat transfer performance.
It can be seen from Fig. 11(b) that the value of $f$ changes with $Re$ 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 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. $PEF$ becomes larger as $C_w$ increases, especially when $Re$ 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 $PEF$. 

**Figure 9.** Velocity distribution in pipe with different $C_w$

**Figure 10.** $TKE$ contour distribution with different $C_w$

**Figure 11.** The influence of $C_w$ on $Nu$ and $f$: (a) Nusselt number ($Nu$); (b) friction coefficient ($f$)

**Figure 12.** $PEF$ distribution with different $C_w$
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 pipeline 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.

5.3. Development of correlation equations for \( Nu \) and \( f \)

In order to study the influence of \( Re \), \( C_h \), and \( C_w \) on the heat transfer performance of corrugated pipes, the results of CFD numerical simulation were analyzed by the multiple nonlinear regression, and the correlation equations for \( Nu \) and \( f \) are established, which are as follows:

\[
Nu = 1.0951 \left( Re \right)^{0.6296} \left( \frac{Ch}{D_h} \right)^{0.73533} \left( \frac{Cw}{D_h} \right)^{1.333}
\]

\[
f = 1258.0649 \left( Re \right)^{-0.29139} \left( \frac{Ch}{D_h} \right)^{2.4956} \left( \frac{Cw}{D_h} \right)^{0.19669}
\]

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

6. Conclusion

1) 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.

2) \( Nu \) decreases with the increase of \( C_h \). When \( C_h \) is 2 mm, \( Nu \) is smaller than that of the smooth pipe, and when \( C_h \) is 1 and 1.5 mm, \( Nu \) 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
3) 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 and 2 mm, $Nu$ 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.

4) 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 $Re$. It is found that in the low $Re$ 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 $Nu$ 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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Submitted: 09.07.2023
Revised: 18.09.2023
Accepted: 27.09.2023