NUMERICAL STUDY ON THE FLOW AND HEAT TRANSFER OF WATER-BASED Al₂O₃ FORCED PULSATING NANOFLUIDS BASED ON SELF-EXCITED OSCILLATION CHAMBER STRUCTURE

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In this study, the flow and heat transfer characteristics of the forced pulsating Al₂O₃/water nanofluid were numerically studied. The pulsating excitation of the nanofluid is provided by the Helmhertz self-excited oscillating cavity. The large eddy simulation method is used to solve the equation, and the local Nusselt number and heat transfer performance index are used to analyze the heat transfer characteristics of the nanofluid in the self-excited oscillation heat exchange tube. In addition, the effect of different downstream tube diameters on heat transfer enhancement is discussed. The research results show that the existence of the countercurrent vortex can increase the disturbance of the near-wall fluid, thereby improving the mixing degree of the near-wall fluid and the central mainstream. As the countercurrent vortex migrates downstream, pulse enhanced heat transfer is realized. Furthermore, it was also found that when the downstream tube diameter \( d_2 = 1.8d_1 \), the periodic effect of the local Nusselt number of the wall is the best and the heat transfer performance index has the most stable pulsation effect within a pulsation cycle. But when \( d_2 = 2.0d_1 \), the change curve of heat transfer performance index in a pulsating period is the highest, the maximum value is 3.95.

Key words: self-excited oscillation; pulsating nanofluid; enhanced heat transfer; Large Eddy Simulation; countercurrent vortex

1. Introduction

In recent years, in order to realize more efficient and energy-saving heat exchange process in heat exchange equipment, passive enhanced heat exchange technology is widely used because it does not require external power. Changing the geometric structure and using nanofluid as a heat transfer medium are common methods to improve heat transfer enhancement. It is reported that the low concentration of nanoparticles greatly improves the heat transfer efficiency of the heat exchange working fluid, thereby significantly increasing the local Nusselt number of heat exchange[1]. In this study, the effects of changing the geometry and using the nanofluids on the flow and heat transfer characteristics are investigated simultaneously. Since the application of forced pulsation nanofluids is
in compact heat exchanger equipment, the purpose of this research is to further increase the enhanced heat transfer rate of compact heat exchanger under the premise of energy saving.

Some studies have shown that the use of nanofluid as a working medium in the heat exchange tube can increase the heat transfer coefficient, and the heat transfer enhancement increases with the increase of the nanoparticle volume fraction[2-4]. However, Bayat and Nikseresh[5] showed through numerical studies that although nanofluids can increase the convective heat transfer rate, it also require greater pumping power. Similarly, studies have shown that the application of nanofluids in non-straight pipes is more conducive to improving the efficiency of heat transfer enhancement[6-8]. And Manca et al.[8] also found that the pumping power also increased with the increase of nanoparticles.

Therefore, whether nanofluids are used for heat transfer in straight or non-straight pipes, there is a problem of increased pumping power. On the basis of using nanofluids to enhance the heat transfer rate, in order to reduce the pumping power or keep the pumping power constant, some researchers consider using pulsating nanofluids for heat transfer. Li et al.[9] found through numerical research that increasing the pulsation frequency to a certain extent would be more helpful to improve the local and average heat transfer. Sivasankaran and Narrein[10] conducted a numerical analysis on the thermal-hydraulic characteristics of the pulsating water-based Al₂O₃ nanofluid in the porous medium spiral microchannel radiator. Their research results show that the presence of porous media produces a better heat transfer enhancement effect for sinusoidal velocity inlet conditions compared to steady flow conditions. Naphon and Wiriyasart[11,12] combine four enhanced heat transfer technologies: pulsating flow, nanofluid, magnetic field, and curved tube (or spiral corrugated tube) to enhance heat transfer. Their experimental results show that thermal boundary layer disturbance and nanoparticles have significant promoting effects on heat transfer enhancement, and the Nusselt number of pulsating flow is also higher than that of continuous flow. In addition, Zufar et al.[13] numerically and experimentally studied the effect of mixed nanofluids on the heat transfer performance of pulsating heat pipes. Experimental results show that, nanofluids can initiate pulses at lower heating power. A review article on the use of nanofluids in heat pipes by Alhuyi Nazari et al.[14] pointed out that the use of graphene oxide/water nanofluids in pulsating heat pipes can reduce thermal resistance by up to 42%. It can be seen from the above literature that the pulsating nanofluid is more conducive to heat exchange in the channel. Therefore, it is necessary to study the heat transfer performance of the pulsating nanofluids in heat exchange channels.

At present, researchers have conducted numerical and experimental studies on self-excited oscillation pulse enhanced heat transfer. Wang et al.[15] analyzed the feasibility of enhancing heat transfer in the self-excited oscillation shell and tube heat exchanger. The results proved that the application of the self-excited oscillation cavity in the shell and tube heat exchanger can not only effectively prevent fouling, but also conducive to the structural transformation of the existing shell and tube heat exchanger. Zheng et al.[16] conducted an experimental study on the enhancement of pulsation heat transfer. And the pulsation of the fluid is generated by a self-excited oscillation cavity. Their research results show that the hydraulic parameters of the fluid and the structure of the self-excited oscillating cavity have a significant effect on the enhancement of fluid pulsation heat transfer. It can be seen that the self-excited oscillation cavity structure not only causes the fluid to pulsate, but also facilitates the structural design according to the different applications of the compact heat exchanger.

Therefore, this paper adopts the Large Eddy Simulation (LES) method to analyze the influence
of the Al₂O₃ nanofluid in the flow and heat transfer characteristics of the self-excited oscillation cavity. The flow characteristics of Al₂O₃ nanofluid in the self-excited oscillation heat exchange tube are discussed through the distribution of velocity cloud, temperature cloud and vorticity cloud. The changes in local Nusselt number and nanofluid heat transfer performance indicators are used to evaluate the heat transfer performance of Al₂O₃ nanofluids in self-excited oscillation heat exchange tubes. In addition, the influence of different downstream pipe diameters on heat transfer enhancement is discussed. The final research results will provide a certain theoretical basis for the optimal design of compact heat exchangers.

2. Numerical model

2.1. Physical model and boundary conditions

Fig. 1 shows the schematic diagram of the two-dimensional geometric structure of the self-oscillating heat exchange tube. The geometric structure of the self-excited oscillation heat exchange tube studied in this paper consists of three parts: the upstream tube, the self-excited oscillation chamber and the downstream tube. In order to ensure the stability of the inlet flow state of the chamber, the geometry of the upstream channel is consistent with the Helmhertz nozzle inlet structure studied by Liu et al.[17]. The upstream pipeline inlet diameter \( d_0 = 13 \text{mm} \), the upstream pipeline convergence angle is \( \alpha = 14^\circ \), and the upstream pipeline outlet diameter \( d_1 = 5.9 \text{mm} \). At the same time, in order to enable the nanofluid to obtain sufficient pulsation excitation sources, the chamber structure parameters are selected according to the optimal parameter ratio range of Wang et al.[18]. The chamber diameter \( D = 12d_1 \), the chamber length \( L = 6d_1 \), and the downstream channel diameter \( d_2 \) are 1.6\(d_1 \), 1.8\(d_1 \), 2.0\(d_1 \), and 2.2\(d_1 \), respectively. In addition, according to the research of Liu et al.[19], it is known that the chamber impingement angle \( \theta = 120^\circ \) is the best. According to the considered inlet Reynolds number \( (Re_m=40000) \), the upstream pipeline inlet is set as the velocity inlet boundary condition. And the downstream pipeline outlet is set as the pressure outlet boundary condition. The walls of the pipeline are all set as non-slip boundary conditions and have a uniform and constant temperature of \( T_w = 400 \text{K} \). At this time, the upstream pipeline inlet temperature is set to \( T_{in} = 298.15 \text{K} \).

![Fig. 1 Schematic diagram of the geometric structure of the self-oscillating heat exchange tube](image)

2.2. Numerical procedures and governing equations

2.2.1 Numerical procedures

In this study, ANSYS ICEM CFD 17.0 is used to grid the computational domain, and ANSYS FLUENT 17.0 code is used to conduct a two-dimensional numerical simulation of the self-oscillation
heat transfer tube model. The governing equations are solved by the finite volume method (FVM). Both the convection and diffusion terms adopt the second-order upwind scheme. In this numerical simulation process, a large eddy simulation turbulence model based on self-similarity theory is used. This turbulence model is based on the idea of only calculating turbulent large eddies and filtering out small eddies. Subgrid-Scale Models are used to describe large vortices, and Wall-Adapting Local Eddy-Viscosity (WALE) Model is used to solve eddy viscosity. The SIMPLE algorithm is used to solve the velocity-pressure coupling equation, and the second-order discretization is used to obtain a higher precision numerical solution. Furthermore, in order to achieve a compromise between numerical solution time and accuracy, the maximum residual is set to $10^{-6}$.

2.2.2 Governing equations

First make the following assumptions about the working conditions of the computational domain: (i) nanoparticles are uniformly distributed in the base fluid and are stable; (ii) nanofluids are incompressible; (iii) a constant wall temperature is applied to all walls; (iv) the thermophysical properties of the fluid and channel materials are not affected by temperature.

Since the flow boundary has little effect on the small-scale vortices, the filtering method can be used to eliminate the small-scale vortices. After filtering, the incompressible Navier-Stokes equation is written as [20]:

**Continuity equation:**

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u_i}{\partial x_i} = 0 \quad (1)$$

**Momentum equation:**

$$\frac{\partial u_i}{\partial t} + \frac{\partial (u_j u_i)}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} - \frac{2}{3} \rho \frac{\partial u_i}{\partial x_j} \frac{\partial u_j}{\partial x_j} \quad (2)$$

Where, $\rho$, $u$, and $\rho_p$ are the density, velocity and pressure of the working fluid, respectively. $\sigma_{ij}$ is the stress tensor, defined by molecular viscosity as:

$$\sigma_{ij} = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \rho \frac{\partial u_i}{\partial x_j} \delta_{ij} \quad (3)$$

In the WALE model, the eddy viscosity is given by:

$$\mu_i = \rho L_s^2 \left( \frac{S_i^d S_i^d}{S_i^d S_i^d} \right)^{3/2} \quad (4)$$

Where $L_s$ and $S_i^d$ in the WALE model are defined, respectively, as:

$$L_s = \min(k d, C_s V^{4/3}) \quad (5)$$

$$S_i^d = \frac{1}{2} \left( \frac{\overline{g_i^2}}{\overline{g_i^2}} + \frac{1}{\overline{g_i^2}} \right) - \frac{1}{3} \delta_{ij} \overline{g_i^2} \overline{g_j^2} = \frac{\overline{u_i u_i}}{\partial x_i} \quad (6)$$

The value for the WALE constant, $C_w$, is 0.325.
2.3. Data reduction

In numerical simulation, the following formulas can be used to calculate Reynolds number, Nusselt number, pressure drop and friction factor[21]:

\[ Re = \frac{\rho_{nf} U_m D_h}{\mu_{nf}} \]  
(7)

\[ Nu = \frac{hD_h}{\kappa} = \frac{\dot{m}C_p(T_{in} - T_{out})D_h}{(T_b - T_w)\kappa} \]  
(8)

\[ f = \frac{2}{L/D_h} \frac{\Delta p}{\rho_{nf} U_m^2} \]  
(9)

\[ \Delta p = \bar{p}_{inlet} - \bar{p}_{outlet} \]  
(10)

Where, \( T_{in} \) and \( T_{out} \) are the inlet and outlet temperatures of the tube, \( T_b \) is the bulk mean temperature (the average of the inlet and outlet temperatures), \( T_w \) is the tube wall temperature, \( D_h \) is the hydraulic diameter (m), \( \kappa \) is the thermal conduction coefficient, \( \dot{m} \) is the mass flow rate, and \( C_p \) is the specific heat at constant pressure.

In order to characterize the heat transfer performance of Al\(_2\)O\(_3\) nanofluid in the self-excited oscillation heat exchange tube, the heat transfer performance index (HTPI) is defined. In this study, Al\(_2\)O\(_3\) nanofluid is regarded as a single-phase fluid in which nanoparticles and base fluid are uniformly mixed, so the required heat transfer performance standard is derived from the following formula:

\[ HTPI = \left( \frac{E_{Nu}}{(E_f)^{\frac{1}{3}}} \right) = \frac{Nu_{nf}}{Nu_{bf}} \frac{f_{nf}}{f_{bf}} \]  
(11)

Where, \( Nu_{nf} \), \( f_{nf} \), \( Nu_{bf} \), and \( f_{bf} \) are Nusselt numbers and friction factors of the enhanced (nanofluid flow inside the self-oscillating tube) and non-enhanced (base fluid flow inside the straight tube) conditions, respectively. When the value of HTPI is greater than 1, the enhanced heat transfer is high and the pressure drop loss is low. So the index is the bigger the better the parameter.

2.4. Thermophysical properties of nanofluid

Correctly specifying the properties of nanofluids is essential to obtain accurate numerical results. This study used a nanofluid with spherical Al\(_2\)O\(_3\) nanoparticles with a diameter of 38 nm uniformly distributed in water as the working medium, because Al\(_2\)O\(_3\) nanoparticles are more commonly used and relatively inexpensive. Tab. 1 shows the thermophysical properties of water and Al\(_2\)O\(_3\) nanoparticles. In this paper, a single-phase model is used for numerical simulation, and the following equations are used to calculate the thermophysical properties of nanofluids[6].

Density:

\[ \rho_{nf} = \sum_{k=1}^{n} \phi_k \rho_k = (1 - \phi)\rho_{nf} + \phi \rho_{np} \]  
(12)

Specific heat:

\[ (\rho C_p)_{nf} = (1 - \phi)(\rho C_p)_{nf} + \phi(\rho C_p)_{np} \]  
(13)
Viscosity:

$$\mu_{nf} = \frac{\mu_{bf}}{(1 - \phi)^{2.5}}$$  \hspace{1cm} (14)

Thermal Conductivity:

$$\kappa_{nf} = \frac{k_{np} + 2k_{bf} - 2\phi(k_{bf} - k_{np})}{k_{np} + 2k_{bf} + \phi(k_{bf} - k_{np})}k_{bf}$$  \hspace{1cm} (15)

### Table 1 Thermophysical properties of alumina oxide nanoparticles (Al$_2$O$_3$) and water[22]

<table>
<thead>
<tr>
<th>materials</th>
<th>$\rho$ [kg m$^{-3}$]</th>
<th>$C_p$ [J kg$^{-1}$ K$^{-1}$]</th>
<th>$\mu$ [Pas]</th>
<th>$\kappa$ [W m$^{-1}$ K$^{-1}$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Al$_2$O$_3$</td>
<td>3880</td>
<td>773</td>
<td>-</td>
<td>36</td>
</tr>
<tr>
<td>Base fluid (water)</td>
<td>998.2</td>
<td>4182</td>
<td>998×10$^6$</td>
<td>0.597</td>
</tr>
</tbody>
</table>

### 2.5. Model verification

#### 2.5.1 Grid independence test

When $Re_{in}=40000$, water is used as the working medium, and the self-excited oscillation heat transfer pipes with structural parameters $D=12d_1$, $L=6d_1$ and $d_2=1.8d_1$ is tested for grid independence to analyze the grid size influence of numerical results. Four groups of grids are considered, namely 43395 nodes, 77308 nodes, 172982 nodes, and 354206 nodes.

Eq. (16) is used to calculate the relative error of Nusselt number under different grids, where $M_1$ and $M_2$ represent the average value of Nusselt number under the first and second grids respectively. Tab. 2 shows the relative errors of the Nusselt numbers of the above four grids. Through the comparison of the errors, the grid size of 172982 nodes has been adopted to achieve proper coordination between calculation time and calculation accuracy.

$$e = \left| \frac{M_2 - M_1}{M_1} \right| \times 100\%$$  \hspace{1cm} (16)

#### 2.5.2 Turbulence model verification

In order to prove the credibility of the numerical results, the numerical model used in this study should be verified. Among the current turbulence models, the direct numerical simulation (DNS) turbulence model requires higher computer requirements, the Large Eddy Simulation (LES) turbulence model mainly uses different models according to the turbulence structure scale, and the Reynolds Average (RANS) turbulence model focuses on the flow.
The field vortex structure can only provide steady flow field results. Since this study needs to consider the vorticity distribution in the turbulence and the computational cost, the LES turbulence model is selected for this simulation. When Re_{in}=40000, the water-based Al_{2}O_{3} nanofluid is used as the working medium to verify the numerical simulation results of the self-excited oscillation heat exchange tube with chamber structure parameters D=12d₁, L=6d₁ and d₂=1.8d₁. As shown in Fig. 2, within one second after the turbulent pulsation reaches a stable state, the relative error range of the average Nusselt number of the current result and Dittus-Boelter empirical formula is 0.11% and 7.8%. And the error is less than 10%, indicating that the current numerical simulation results are within the acceptable range of the empirical formula.

Dittus-Boelter's empirical formula[23]:

\[ Nu = 0.023Re^{0.8}Pr^{0.4} \]  \hspace{1cm} (17)

3. Results and discussion

In this section, the particle volume fraction \( \phi \) of the nanofluids used in the study are all 4%. The flow and heat transfer characteristics of the Al_{2}O_{3}-water nanofluid in the self-excited oscillation pulse heat exchange tube and the basic liquid in the straight tube are analyzed by numerical methods.

3.1. Distribution of velocity and temperature

Velocity can effectively represent the effect of drag reduction and heat transfer enhancement. Reverse flow in the downstream tube of self-excited oscillating heat exchange tube is considered to be the key to enhanced heat transfer. The velocity distribution of the nanofluid in the self-excited oscillation pulse heat exchange tube within a pulsation period is shown in Fig. 3. It is obvious that after the nanofluid flows through the self-excited oscillation chamber, a pulsating flow will be generated in the downstream pipe. It can also be observed that low-speed countercurrent zones and high-speed extrusion zones alternately appear on the upper and lower pipe walls, and the low-speed countercurrent zone is the key to drag reduction. And the velocity near the wall surface in the low-velocity countercurrent zone is lower, while the velocity near the wall surface in the high-speed extrusion zone will increase. As a result, this increase and decrease in speed will cause the flow velocity in the downstream heat exchange channel to change periodically.

![Fig. 3 The velocity distribution of the nanofluid in the self-excited oscillating pulse heat exchange tube in a pulsation period](image)

Fig. 4 shows the temperature distribution of the working fluid in the self-excited oscillation heat exchange tube in a pulsation period. Compared with Fig. 3, it can be seen that the fluid temperature in
the low-speed countercurrent zone is higher, while the temperature in the high-speed extrusion zone is relatively low. According to the isotherm in Fig. 4, the high temperature area of the downstream pipeline is located on both sides of the low-speed countercurrent zone, depending on the strength of the countercurrent. And the temperature gradient near the wall reaches the maximum at the countercurrent.

![Fig. 4 The temperature distribution of the nanofluid in the self-excited oscillating pulse heat exchange tube in a pulsation period](image)

In order to further analyze the periodic changes of the velocity and temperature of the nanofluid in the downstream pipeline, four reference lines and five reference points are set. Among them, the four monitoring lines are the downstream pipeline cross-sections, and the distances from the downstream pipeline inlet are 0mm, 37.5mm, 75mm and 112.5mm respectively, and they are marked as line1-line4 in sequence. The five reference points are evenly distributed on the reference line line3. From the upper wall to the lower wall, they are marked as point1-point5.

Fig. 5 shows the velocity and temperature distribution of the downstream pipeline at the four reference lines. From the figure, it can be found that the low-speed recirculation zone and the high-speed extrusion zone alternately appear along the tube wall. In terms of temperature distribution, the temperature gradient near the wall decreases with the axial movement. The thermal boundary layer at the inlet of the downstream tube is the thinnest, and the closer to the tube outlet, the thicker the thermal boundary layer. Due to the existence of the countercurrent vortex, the temperature near the wall is significantly higher than that of the central mainstream.

![Fig. 5 The velocity distribution (left) and temperature distribution (right) of the downstream pipeline on different reference lines](image)
Fig. 6 shows the velocity and temperature changes over time at different reference points. It can be seen from the velocity distribution that point1 and point5 are located near the wall of the pipeline, so their velocity is relatively low, and the velocity fluctuations of these two points are opposite. Point3 is located on the central axis, so its speed is the highest. The velocity fluctuations of point2 and point4 present an axisymmetric trend. When analyzing the temperature of each point, it is found that the temperature of point1 and point5 is higher and the fluctuation range is larger, while the temperature changes of point2, point3 and point4 are basically the same, and the fluctuation range is smaller. This is because point2, point3, and point4 are located in the central mainstream area, where the flow velocity is faster and the temperature gradient is lower.

3.2. Vortex distribution in downstream tube

The vorticity can clearly reflect the flow pattern of the nanofluid in the self-excited oscillation pulse heat exchange tube. When the nanofluid reaches the collision wall, a part of the shear flow flows along the collision wall and forms vortices in the chamber. It is the periodic pulses caused by these vortex structures. Other parts of the shear flow flows along the tube wall of the downstream pipeline. This part of the fluid has a large velocity gradient with the fluid in the central main flow zone, and countercurrent vortexes near the wall of the downstream tube. The vorticity distribution in the downstream tube is shown in Fig. 7. It can be observed that at t=T/4, the shear flow is about to reach the collision wall, and the shear flow of the previous cycle has gradually dissipated in the downstream pipeline; at t=T/2, the shear flow has reached the collision wall and is divided into two parts by the collision sharp corner, forming a countercurrent vortex near the downstream pipe inlet; at t=3T/4, the countercurrent vortex falls off the chamber and migrates downstream, near the wall near x=0.11m The disturbance is the largest; at t=T, the countercurrent vortex in the downstream pipeline has gradually dissipated, and the shear flow in the chamber is preparing for the next collision and separation. In addition, it can be seen from the figure that both sides of the counter-current vortex near the wall of the downstream tube show lower vorticity, which is caused by the high-speed extrusion micelles between two adjacent counter-current vortices. As the countercurrent vortex migrates downstream, the flow velocity near the wall of the tube exhibits periodic fluctuations, which indicates that the existence of the countercurrent vortex increases the volatility of the disturbance near the wall, thereby enhancing the mixing of hot and cold fluids in the tube.
3.3. Heat transfer changes in downstream tubes

3.3.1 Wall shear stress and Nusselt number

The greater the wall shear stress, the greater the flow velocity of the fluid near the wall and the stronger the disturbance. The left image of Fig. 8 compares the wall shear stress curve of self-excited oscillation heat exchange tubes with different downstream tube diameters and smooth tubes. It can be seen from the figure that in a smooth round tube without a self-excited oscillation cavity structure, the wall shear stress decreases linearly along the wall and changes little. In the self-excited oscillation heat exchange tube, the wall shear stress presents a periodic change trend due to the collision and separation of the shear flow. Among them, when $d_2/d_1=1.6$ and 2.0, the wall shear stress has a higher pulsation amplitude, but its fluctuation stability is relatively poor; when $d_2/d_1=1.8$ and 2.2, the fluctuating amplitude of wall shear stress is smaller than the first two cases, and its periodicity is relatively stable. The best periodic wall shear stress is obtained when $d_2/d_1=2.2$. In general, all the self-excited oscillating heat exchange tubes studied are higher than the wall shear stress of smooth round tubes, and when $d_2/d_1=1.8$ and 2.2, the wall shear stress is higher and the periodicity is more stable, so the self-excited oscillation heat exchange tubes with $d_2/d_1=1.8$ and 2.2 have larger disturbances near the wall.

The larger the Nusselt number, the better the heat transfer enhancement effect. The right image of Fig. 8 shows the variation of the Nusselt number of the self-excited oscillation heat exchange tube and smooth tube with different downstream pipe diameters. It can be observed that the local Nusselt number of the smooth circular tube wall has the same changing trend as the wall shear stress. In the self-excited oscillating heat exchange tube, the local Nusselt number on the wall also shows a periodic change trend. Moreover, the local Nusselt number of the wall of the self-excited oscillation tube is generally better than that of the smooth round tube, indicating that the enhanced heat transfer effect of the nanofluid in the self-excited oscillation tube is significantly better than that of the base fluid in the smooth round tube. It is also found that when $d_2/d_1=2.0$, the fluctuation peak of the Nusselt number of
the wall gradually decreases along the wall, indicating that the pulsation at this time is not continuous. When $d_2/d_1 = 1.6$, 1.8 and 2.2, the periodic pulsation effect of the wall Nusselt number is better, so the downstream pipe diameter at this time is more conducive to the enhancement of heat transfer.

![Fig. 8](image) Wall shear stress (left) and local Nusselt number (right) of the self-oscillation heat exchange tube and smooth tube in the downstream pipeline

3.3.2 Heat transfer performance index

Fig. 9 shows the change of HTPI of self-excited oscillation heat exchange tubes with different downstream pipe diameters in a pulsation period. It can be seen that HTPI is greater than 1 in all studies, indicating that the existence of the self-oscillation structure can effectively improve the heat transfer performance. Among them, when $d_2/d_1 = 1.6$, 1.8, 2.2, and 2.2, the maximum value of HTPI in a pulsating period is 1.86, 1.94, 2.64, and 3.95, respectively. In general, when $d_2/d_1 = 1.6$, 1.8, HTPI fluctuates more stably within a pulsating cycle; when $d_2/d_1 = 2.0$, the periodicity of HTPI is the worst, indicating that the enhanced heat transfer is unstable; When $d_2/d_1 = 2.2$, the periodicity of HTPI fluctuation is relatively stable and the value is relatively high. The maximum HTPI value is 3.95, which shows that the heat transfer enhancement effect is the best at this time.

4. Conclusion

In this study, the flow and heat transfer of water-based Al$_2$O$_3$ nanofluids in a heat exchange pipe with a self-excited oscillation cavity structure were studied by large eddy simulation. This numerical results obtained are compared with base fluid in the straight pipe. In addition, the influence of the dimensionless structure parameter $d_2/d_1$ of the chamber on the heat transfer performance at different values is also studied. The main conclusions of this study are as follows:
In the downstream pipeline, the flow near the wall can be divided into two areas: low-speed countercurrent and high-speed extrusion. The temperature in the low-speed countercurrent zone is relatively high, while the temperature in the high-speed extrusion zone is relatively low. And the low-speed recirculation zone and the high-speed extrusion zone alternately appear along the wall. Moreover, the shear flow generated by the self-excited oscillation cavity is the key to the formation of countercurrent vortices in the downstream pipeline. There is a large velocity gradient between the shear flow near the wall of the downstream pipeline and the fluid in the central mainstream zone, which causes the formation of a countercurrent vortex near the near wall of the downstream pipeline. As the countercurrent vortex migrates downstream, periodic pulsating heat transfer is realized.

In addition, through the analysis of wall shear stress and Nusselt number, when \( \frac{d_2}{d_1} = 1.8 \) and 2.2, the disturbance near the wall of the downstream pipeline is the largest and the Nusselt number periodicity is more obvious. And HTPI fluctuates most stable when \( \frac{d_2}{d_1} = 1.8 \), and the maximum value reaches 3.95 when \( \frac{d_2}{d_1} = 2.2 \). Therefore, these two structures are most conducive to the enhanced heat transfer of nanofluids.

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Nomenclature

- \( C_p \): specific heat, \([Jkg^{-1}K^{-1}]\)
- \( d_0 \): diameter of upstream tube inlet, \([m]\)
- \( d_1 \): diameter of upstream tube outlet, \([m]\)
- \( d_2 \): diameter of downstream tube, \([m]\)
- \( d_2/d_1 \): dimensionless parameter, [-]
- \( D \): diameter of chamber, \([m]\)
- \( D/D_1 \): dimensionless parameter, [-]
- \( D_h \): hydraulic diameter, \([m]\)
- \( \Delta p \): pressure drop, \([Pa]\)
- \( f \): friction factor, [-]
- \( h \): heat transfer coefficient, \([Wm^{-2}K^{-1}]\)
- \( L \): length of chamber, \([m]\)
- \( L/d_1 \): dimensionless parameter, [-]
- \( Nu \): Nusselt number(=\(hD_h/\kappa\)), [-]
- \( Pr \): Prandtl number(=\(C_p\mu/\kappa\)), [-]
- \( Re \): Reynolds number(=\(\rho UD_h/\mu\)), [-]
- \( T \): temperature, \([K]\)

- \( U \): velocity, \([m/s]\)
- \( \alpha \): convergence angle of upstream tube, \([^\circ]\)
- \( \theta \): impingement angle of chamber, \([^\circ]\)
- \( \kappa \): thermal conductivity, \([Wm^{-1}K^{-1}]\)
- \( \mu \): dynamic viscosity, \([Nsm^{-2}]\)
- \( \rho \): density, \([kgm^{-3}]\)
- \( \phi \): volume friction of nanoparticles, [%]

Subscripts

- \( f \): friction factor, [-]
- \( bf \): base fluid
- \( h \): heat transfer coefficient, \([Wm^{-2}K^{-1}]\)
- \( in \): inlet
- \( L \): length of chamber, \([m]\)
- \( m \): mean
- \( nf \): nanofluid
- \( np \): nanoparticle
- \( out \): outlet
- \( w \): wall

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13


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