Direct Numerical Simulation of Active-Controlled Impinging Jets

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Abstract
In order to improve the heat transfer on the wall, impinging jets are used in various industrial applications, and have been investigated experimentally and numerically so far. However, it is not enough to make clear the detail of vortical structure contributing to the heat transfer. In the present paper, direct numerical simulations (DNS) of the impinging jet are conducted through the control of vortical structure in order to investigate the heat transfer. The discretization in space is performed with a hybrid scheme in which Fourier spectral and 6th order compact scheme are adopted. As the control parameter, two cases of perturbations are superposed on the inflow boundary conditions. These excitations contribute to the generation of coherent vortical structures, resulting in the enhancement of mixing away from the impinging wall. However, the heat transfer at the wall is not vitalized in comparison to the no excitation case. The reason why no enhancement of the heat transfer occurs are considered, based on both the balance of the heat flux and the snapshot of flow. It is found that the excitation strengthens the upward flow, which disturbs the heat transfer, and that the upward lifting of coherent vortical structures make the inactive state in the vicinity of the impinging wall.

Key words: Impinging Jet, Heat Transfer, DNS , Coherent Structure, Turbulent Mixing

1. Introduction

Impinging jets are widely used in industrial applications, such as heating on surfaces, cooling electronic devices and drying coating surfaces. Their characteristics are reviewed by a few notable literatures(1) – (3). So far, a few or several ten thousand Reynolds number, defined with inlet nozzle diameter, are mainly targeted in laboratory research. However, according to the downsizing of electronic equipments, several one hundred of Re numbers are going to be dealt with(4). Impinging jets are subdivided into three regions, such as the free jet region, the stagnation region and the wall jet region(3). Their performance of heat transfer on the impinging wall is depending on the Re number, the shape of nozzle, the number of nozzle, the distance between the nozzle and the impinging wall and so on(3). In order to establish highly effective control of heat transfer, it should be understood the details of the heat transfer, as well as the characteristic features of these parameters should be systematically investigated. Recently significant advances of computer power lead to the realization of fluid phenomena including miniature vortices through the direct numerical simulation (DNS). Under the circumstances, we found the effectiveness of the active control of a round jet for the mixing enhancement(5). DNS of impinging jets are conducted so far(6) – (8) and highly resolved LES ( large eddy simulation ) (9) have demonstrated the flow structure and the statistical properties of heat transfer. However, to our knowledge, there is no DNS evaluating the heat transfer under the active-controlled jets. In the present paper, in order to develop the effective method of heat transfer for the impinging jet, we examine the effect of modulation of turbulence structures using the active control. We have already developed the DNS code calculating a free
jet\(^{(5)}\). In the present study, further this code is improved to be able to calculate the impinging jet by adding the boundary conditions realizing the solid wall. Both the statistical properties and instantaneous flow structures are demonstrated to make clear the mechanism of the heat and momentum transfer.

2. Numerical method

2.1. Governing equations and their discretization

Under the assumption of incompressible flow, the governing equations are as follows:

\[
\begin{align*}
\frac{\partial u_i}{\partial x_i} &= 0 \quad (1) \\
\frac{\partial u_i}{\partial t} + h_i &= -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{1}{Re} \frac{\partial^2 u_i}{\partial x_i^2} \quad (2) \\
\frac{\partial T}{\partial t} + \frac{\partial u_i T}{\partial x_i} &= \frac{1}{Re Pr} \frac{\partial^2 T}{\partial x_i^2} \quad (3)
\end{align*}
\]

where the convective terms are described as rotational form. Above equations are normalized by both the diameter of the inlet jet, \(D\) and the inlet velocity, \(V_0(= V_{1} - V_{2}\) in Fig. 1). \(Re\) number and \(Pr\) number are defined as \(Re = V_0 D/\nu\) (\(\nu\): dynamic viscosity) and \(Pr = \nu/\alpha\) (\(\alpha\): thermal conductivity), respectively. In some previous papers\(^{(10)}\), the effect of natural convection has been considered. However, since the low Grashof number flow is assumed in the present simulation, thus the buoyancy effect is neglected. Figure 2 shows the computational volume and coordinate systems. The computational volume is the rectangular box. The origin of axes is set at the impinging position on the wall. The wall-normal direction, \(y\) and two horizontal directions, \(x\), \(z\) are set, and the velocity component for each direction denote \(u\), \(v\) and \(w\), respectively. Similarly, the radial direction, \(r\), the azimuthal direction, \(\theta\) are set, and the velocity components for two directions denote as \(u_r, u_\theta\), respectively.

Fig. 1 Inflow velocity conditions.

Fig. 2 Computational domain and coordinate systems.
The spatial discretization is performed with a Fourier series expansion in x and z directions and 6th-order compact scheme\(^{(11)}\) in the streamwise direction. In order to remove the numerical instability due to the nonlinear terms, the 2/3-rule is applied for the horizontal directions and an implicit filtering for the wall-normal direction is conducted with 6th order compact scheme. For the time advancement, third order Adams-Bashforth method is used. The well-known MAC method is employed for pressure-velocity coupling, which results in a Poisson equation for the pressure. After the Poisson equation is Fourier transformed in x and z directions, the independent differential equations are obtained for each wave number and then is discretized with sixth order compact scheme. Finally, the penta-diagonal matrix is deduced for each wave number. In the present simulation code, these matrixes are solved using the LU Decomposition method. Because the periodic boundary conditions are imposed for the horizontal directions, the jet is not able to flow into the surroundings after the jet impingement to the wall. To solve the difficulty, then the flow is sinked around the impinging wall (Fig.2), where the width of this region is set as \(\Delta z = 0.23D\). The sink flow is assumed to be a parabolic profile and its strength is determined such that the flow rate of sink flow equals to that of inflow.

2.2. Calculation conditions

Figure 2 is schematic drawing of the flow field. The inlet velocity distribution is assumed to be top-hat type, which is given as following:

\[
V_b = \frac{V_1 + V_2}{2} - \frac{V_1 - V_2}{2} \tanh \left( \frac{1}{4} \frac{r}{\theta_0} \left( \frac{r}{R} - \frac{R}{r} \right) \right) \tag{4}
\]

where, \(V_1\) and \(V_2\) are the center velocity and co-flowing velocity, respectively. \(\theta_0\) denotes the initial momentum thickness, \(R = D/2\) is radius of an inlet jet. These parameters are selected by referring to the literature\(^{(12)}\): \(V_1 = 1.075V_0\), \(V_2 = 0.075V_0\) and \(R/\theta_0 = 20\). In the present simulations, the distance to the impinging wall, \(H\) is \(H/D = 4\). Computational conditions such as the size of computational domain, the grid number, the Reynolds number, the Prandtl number are \((H_x, H_y, H_z) = (12D, 4D, 12D)\), \((N_x, N_y, N_z) = (256, 200, 256)\), \(Re = 1500\) and \(Pr = 0.71\), respectively. The grid spacing of wall-normal direction is densely populated near wall region. The inflow temperature, \(T_0\) and the ambient temperature, \(T_a\) are assumed to be higher than the wall temperature, \(T_w\), i.e., \(T_0 = T_a > T_w\). The statistical properties are averaged over the time and azimuthal direction. The mean quantity is denoted with bar (\(\bar{\cdot}\)) and fluctuating components, by prime (\(\cdot'\)). In order to acquire the statistical properties, the physical quantities are averaged between \(t' = 100\) and \(t' = 300\).

2.3. Excitation types

In order to enhance the mixing using active control, two types of excitation, i.e. axisymmetric \((V_a)\) and helical \((V_h)\) excitations are considered. In each excitation, the following perturbation velocity and a random perturbation having 1% strength of the inflow velocity are superposed on the inlet velocity, \(V_b\).

\[
V_a = 0.01 \sin(2\pi St_a t^*)V_b \tag{5}
\]

\[
V_h = 0.01 \sin(\theta - 2\pi St_h t^*)V_b \tag{6}
\]

where \(t^*\) means a nondimensionalized time, \(t^* = tV_0/D\), and \(\varepsilon_{a,h}\) is the strength of excitation. \(\theta\) is the azimuthal angle shown in Fig. 2. The Strouhal number, \(St_{a,h}\) is defined as \(St = fD/V_0\) (where \(f\): frequency). According to the excitation, it is well-known that the peculiar instability mode is induced near field of the jet; In the axisymmetric excitation, the column of vortex rings is formed upstream, and in the helical excitation, the helical-like vortical structures appear. The frequency of instability mode associated with the generation of large-scale structures induced by the column instability near the end of potential core of the jet, is so-called ’preferred mode’. Since the preferred mode is influenced by the shape of nozzle or the boundary layer thickness near the nozzle exit, the preferred mode, \(St_p\) becomes \(0.25 < St_p < 0.5\).
In the present simulation, the excitation frequency are set to $St_a = St_h = 0.4$, the strength of excitation, $\varepsilon_a = \varepsilon_h = 0.05$.

3. Results and discussion

3.1. Evaluation of numerical accuracy

The Reynolds number of inlet jet flow, $Re = 1500$ is categorized into transition or semi turbulent state in the literature\(^{(3)}\). Due to low speed flow, there is quite a few experimental data enable to compare the present results, thus we demonstrate the validity of our simulation as far as we can.

As shown in previous paper\(^{(5)}\), the data of free jet is in good agreement with the experimental data. Therefore, we demonstrate only the data on the impinging wall.

Figure 3 shows the wall-normal distribution of mean radial velocity at different positions, those are normalized with both the maximum velocity, $u_{r_{max}}$ and the half width, $b_{0.5}$. As shown in the figure, The resulting data collapse into a single curve at $r/D = 1.5 \sim 3.5$, well representing a self-similarity of the wall jet. However, at $r/D = 4$ the similarity is not established in the vicinity of the impinging wall, due to the influence of the sink flow around the impinging wall. The variation of the distance from the impinging wall versus local Nusselt number at the stagnation point($r = 0, y = 0$) is shown in Fig. 4. In the figure, the symbols denote the experimental data measured by Yokobori et al.\(^{(14)}\). Double circles which stay at $H/D = 4$ and 8, denote the present simulation under the coarse grid ($128 \times 100 \times 128$). These data demonstrate that the present data are in the scatter of the experimental data. Figure 5 shows the radial distribution of the Nusselt number, $Nu$ defined with $Nu(r) = q_w/(T_0 - T_w)$ ($q_w$ : the heat flux on the wall). In the figure, symbols denotes the experimental data of Angioletti et al.\(^{(15)}\)(Re = 1000, 1500) and the coarse grid case of the present simulation($128 \times 100 \times 128$), respectively. The present results are in good agreement with the experimental data.
3.2. Flow properties of controlled jet

3.2.1. Instantaneous of vortical structures

In order to visualize the coherent vortical structures, the iso-surfaces of the second invariance of velocity gradient tensor, $Q$ value\(^{(16)}\), are demonstrated in Fig. 7. In spite of the short distance from the wall, $H/D = 4$, the vortical structures in the free jet region before the impingement behaves similar to that of unbounded free jets, i.e., demonstrating that axisymmetric excitation (Fig. 7(b)) induces the formation of vortex rings, and that the helical excitation (Fig. 7(c)) induces the formation of the helical-like structures before the impingement. These structures responded to the excitations are agreement with previous literatures\(^{(5),(17)}\). In the no excitation case (Fig. 7(a)), large-scale vortex rings are continuously formed on the impinging wall and move toward the radial direction. In the axisymmetric excitation (Fig. 7(b)), the generated vortex ring before the impingement move for the impinging wall and then turn to the radial direction. Downstream a lot of quasi-streamwise vortices, of which the axis of rotation direct radially, are to a large degree formed between different large-scale vortex rings. The reason is considered that the stretching due to large-scale vortex rings on the impinging wall contributes to the formation of quasi-streamwise vortices. In the helical excitation (Fig. 7(c)), the helical like structures in the vicinity of the nozzle are formed and impinging to the wall. Unlike the axisymmetric excitation, quasi-streamwise vortices less generated downstream. Figures 8 show the bottom view of the coherent vortical structures. Unlike the perspective view, regardless with or without the excitation, the large-scale vortex rings are concentrically formed from the
impinging point.

3.2.2. Statistical properties The position where potential core breaks down is $8 \sim 10$ times nozzle diameters from our previous results\((5)\) in DNS of active-controlled unbounded free jet. Not shown here, in the present simulations it is found that the break-down position is nearly same regardless with or without the excitation. This characteristic is quantitatively consistent with the fact that the distances from onset of reduction of centerline velocity to the impinging wall is $1.2D^{(2)}$.

Figures 9 show the wall-normal distribution of various mean flow properties. In Fig. 9(a), the radial mean velocity profiles are similar at $r/D = 1$ regardless with or without the excitation. At $r/D = 2$, in the axisymmetric excitation, the peak value is away from the wall, and the recirculation zone is visible further away. At $r/D = 3$, both the helical and axisymmetric excitation cases demonstrate the similar trend. At $r/D = 4$ and 5 there are no obvious difference between all cases. For turbulent kinetic energy profile($k = \frac{1}{2} u'^2$) shown in Fig. 9(b), as getting away from the impinging position, the peak values of all cases move away from the impinging wall and then weaken downstream. From this figure, it obviously demonstrates the effectiveness due to the excitation, in particular, at $r/D = 2$ axisymmetric
excitation induces the strong turbulence. Similar to the trend of radial mean velocity, the turbulence energy is sufficiently small downstream. Thus the excitation influences only around the impinging point. From the Reynolds stress profile shown in Fig. 9(c), the Reynolds stress is enhanced in the high-shear region away from the impinging wall, but the generation of Reynolds stress is inactive in the vicinity of the impinging wall as getting away from the impinging position.

Figures 10 show the vorticity intensity. The radial vorticity intensity, which corresponds to the quasi-streamwise vortices, has the peak value in the vicinity of the impinging wall. Generally, in wall turbulence, the streamwise vorticity intensity corresponding to the quasi-streamwise vortices, has the peak value on the wall. That is inevitably caused due to the non-slip conditions, and does not mean the existence of coherent vortical structures on the wall. That of impinging jet also occurs by the same reason. Corresponding to the visualized coherent vortices, the radial vorticity intensity away from the wall is strong in order of the axisymmetric-, the helical- and no excitation. Furthermore, the profile of azimuthal vorticity
intensity (Fig. 10(b)) behaves as well as that of the radial vorticity intensity, and the azimuthal vorticity is the dominant component than the other. These features are consistent with that of the visualized coherent vortical structures.

3.3. Thermal properties of controlled jet

3.3.1. Statistical properties

Figures 11 show the wall-normal distribution of thermal properties. Mean temperature profile (Fig. 11(a)) in the vicinity of the impinging wall is similar regardless with or without the excitation, but the temperature away from the wall decreases by the effective mixing through the excitations. While the strong temperature gradient in the vicinity of the impinging wall contributes to the occurrence of the temperature fluctuation as shown in Fig. 11(b). Turbulent heat flux shown in Fig. 11(c), which reflects on the flow properties mentioned before, in particular, at \( r/D = 2 \) is vitalized by the excitation.

Figure 12 shows the radial distribution of the local Nusselt number. Corresponding to the mean temperature profile in the vicinity of the wall, the heat transfer enhancement does not occur on the wall in spite of the appearance of the obvious excitation effect. On the other hand, since the turbulent heat flux is augmented due to the structure modulation, these results are contradictory to each other.

In order to clarify the reason why the heat transfer enhancement does not occur, we investigate the balance of heat flux along the wall normal direction.

\[
\begin{align*}
- \frac{1}{PrRe} \left. \frac{\partial \bar{T}}{\partial y} \right|_{y=0} &= \int_{0}^{y} \left\{ \frac{1}{r} \frac{\partial \bar{u} \bar{T}}{\partial r} + \frac{1}{r} \frac{\partial \bar{u} \bar{T}'}{\partial r} \right\} dy \\
&- \int_{0}^{y} \left\{ \frac{1}{PrRe} \frac{1}{r} \frac{\partial T}{\partial r} \right\} dy - \frac{1}{PrRe} \frac{\partial \bar{T}}{\partial y} \\
&+ \bar{v} \bar{T} + \bar{v}' \bar{T}' 
\end{align*}
\]
The above equation is derived from the integration of time-averaged energy equation. In the equations, the l.h.s. means the local heat flux on the impinging wall. The first, second and third terms of r.h.s express the heat flux from radial direction due to the mean flow, turbulence and molecular diffusion, respectively. While the fourth, fifth and sixth terms express the heat flux from the wall-normal direction due to the molecular diffusion, mean flow and turbulence, respectively. Note that in a fully developed pipe flow or a wall turbulence, the inhomogeneous direction is only one, i.e., in the wall-normal direction, fourth and sixth terms in the above equation only contribute to the heat transfer. However, in the case of impinging jet several terms affect the heat transfer.

Figures 13 show the balance of heat transfer in the vicinity of the wall at $r/D = 2$ for both the no excitation and helical excitation. In the figures the primary terms contributing to the heat flux are first, fourth, fifth and sixth term. In particular, regardless with or without the excitation, the heat transfer due to the radial mean flow (square) is primary, while the wall-normal
mean flow (▽) suppresses the heat transfer. Although the turbulent heat flux in the no excitation case (Fig.13(a)), is weak, instead, the radial mean flow governs the heat transfer. On the other hand, in the helical excitation (Fig.13(b)), turbulence heat flux is strong in comparison to the no excitation case, however, the wall-normal mean flow strongly suppresses the heat transfer, resulting in no occurrence of strong heat flux on the wall. The factors of heat-transfer improvement due to the impinging jet are pointed out by Kataoka(1) as follows: (i) increasing the impinging velocity to the impinging position; (ii) increasing of turbulence intensity; (iii) surface renewal effect due to large-scale eddies. From Fig. 13, it is found that the radial convection is importance near the impinging position as well as the impinging velocity related to the factor (i). The large-scale eddies being imagined by Kataoka is not that of low Reynolds number flow such as present simulation, but that of high Reynolds number flow. If the large-scale flow corresponds to the vortex ring structures over the impinging wall, it is obviously that the excitation yields the effect to the flow field as the factor(ii) and (iii). However, it does not always get the improvement of performance on the heat transfer. There is a substantial difficulty in the heat-transfer control through the impinging jet.

3.3.2. Instantaneous vortical structures and heat transfer

We reconsider the heat transfer based on the instantaneous vortical structures. Figures 14 show the iso-contours of the
azimuthal vorticity and the contour lines of the temperature on the plane thorough the origin ($r = 0$). In the figure, white colored regions mean the negative azimuthal vorticity (the counter clockwise rotation in the figure); black colored regions means positive one (the clockwise rotation); white lines are the contour of temperature. In the no excitation case(Fig.14(a)), the strong shear layer exists in the free jet region ($r = 0.5D$), and then turn to the radial direction on the impinging wall. As not clearly observed, the strong positive vorticity (black colored region), which occurs due to the no-slip condition, stably distributes in the vicinity of the impinging wall. Near the $r/D = 2$ upstream-convected vorticity develops and moves away from the wall. At the same time near the $r/D = 3$ the counter-rotating vorticity lifts up from the wall and develops downstream. The above mentioned results are similar to the story in which a single vortex ring impinging a wall and developing over the wall, is demonstrated in the literature\cite{18}. Further, since the temperature distribution follows the development of vorticity, the vortical structures having the azimuthal axis of rotation contribute to the heat transfer on the impinging wall. Note that in order to improve the heat transfer performance, it needs that the both positive and negative vortices do not blow up from the wall but blow down to the wall. However, the vortices lift up from the wall, losing the supply of vorticity from the high-shear layer over the wall. Thus the vortices continue to diffuse, and less contribute to the heat transfer on the wall. In the axisymmetric excitation(Fig14(b)), vorticity distribution, which relate to the upstream vortex rings, are visible. The vortex structures which starts to develop earlier than that of no excitation case, promote to generate the counter-rotating vortices and enhance both the heat and momentum transfer away from the wall, but less effect to the wall. In the helical excitation, near the $r/D = 1$ and 2 the growing vortices are observed, and then trace the same process as well as the axisymmetric excitation. In the present simulation, since the viscous sublayer is thick due to the low Reynolds number flow, the vortices away from the wall are difficult to activate the near wall layer. However, since the viscous sublayer is
thin under the high Reynolds number flow, it is expected that the excitation improve the heat transfer performance. In future we try to investigate the Reynolds number effect and to find out the effective control parameters by reference to a forced laminar wall jet\(^{(19)}\).

4. Conclusions

We investigate the effect on the heat transfer of impinging jet through the two type excitations using the newly developed high-resolution DNS code. Conclusions are as follows:

(1) According to the excitation pattern, the peculiar vortical structure are produced in the free jet region before the impingement. After the impingement, the large-scale vortex rings are formed on the impinging wall regardless with or without the excitation.

(2) From the profile of turbulence energy, the Reynolds stress and the local Nu number, it is found that the vortical structures generated by the excitation, strongly mixing further away from the wall, however do not contribute to the heat transfer enhancement on the impinging wall.

(3) In the no excitation case, since the vortical structures is less generated from the free jet region to the impinging region; the diffusion due to the jet is suppressed, the convective heat transfer in the vicinity of the wall is considerably maintained. On the other hand, in the excitation the turbulent heat flux is markedly enhanced by the vortical structures. However, the convective heat transfer in the vicinity of the wall is suppressed by the wall-normal mean flow, resulting in the no enhancement of heat transfer on the wall.

(4) From the instantaneous view, the mixing enhancement away from the wall and no enhancement of heat transfer in the vicinity of the wall are caused by that strong excited vortical structures which moves toward the radial direction and lift up away from the wall.

References