The study of flow and heat transfer characteristics of impinging jet array mounting air-induced ducts

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Abstract. Impinging jet is widely employed in thermal industrial applications due to having high heat transfer coefficient in impingement region. One method to increase heat transfer on an impingement surface is to increase turbulence intensity in jet flow. The mounting of an air-induced duct at nozzle outlet is a passive method to increase entrainment air resulting on increasing turbulence intensity. The aim of this research is to study flow and heat transfer characteristics of array of impinging jets mounting air-induced ducts. The investigation model was jets discharging from pipe nozzle having an inner diameter of d=17.2 mm and a length of 200 mm. Nozzle arrangement were inline configuration having 5 rows x 5 columns. A jet-to-jet distance (S) was S=6d, 8d and a jet-to-plate distance (H) was H=6d. The inner diameter (D) and the length (L) of the air-induced ducts were D=4d and L=4d, respectively. The Reynolds number was fixed at Re=20,000. In addition, the impinging jets without mounting the air-induced ducts were also investigated for benchmarking with the case of mounting the air-induced ducts. In the study, a thin foil technique was used to measure heat transfer on the impingement surface, and a computational fluid dynamic (CFD) using ANSYS, Fluent (V.15.0) was also adopted. The results showed that the effect of mounting air-induced duct can enhance entrainment air into the jet flow resulting on increasing of heat transfer of impinging jets on target surface, and the effects of mounting air-induced duct on increasing heat transfer in case of larger jet-to-jet distance (S/d=8) was more effective than that of smaller jet-to-jet distance (S/d=6).

1. Introduction
This century, efforts have been made to produce more efficient thermal equipment by employing various methods of heat transfer enhancement. Heat transfer enhancement techniques can be separated into two categories: passive and active methods. Passive method requires no external power while active method requires external power to activate fluid flow which causes to enhance heat transfer.

Heat transfer on a surface is principle transferring of thermal energy between fluid and solid surface. More commonly, an impinging jet is used in thermal engineering application or thermal industrial processes due to having high heat transfer coefficient in impingement region. Previous works have been extensively studied on flow and heat transfer characteristics of impinging jet. Heat transfer rate on an impinged surface is governed by momentum of jet impinges on a target surface and a turbulence intensity of jet just before impingement. Generally, after jet discharging from nozzle, the
spreading jet results in the reducing of axial velocity and the increasing of turbulence intensity in a jet flow. The optimal matching between axial velocity and turbulence intensity can be found at the end of potential core occurring in the rage of 5 to 8 time of nozzle diameter from jet outlet, depending on nozzle shape and jet Reynolds number. Subsequently, the maximum heat transfer at stagnation region is gained [1-5].

Many researchers have devoted to enhance heat transfer on an impingement surface by increasing turbulence intensity of jet flow such as attaching mesh screens [6] or triangular tape [7] at the jet outlet and inserting twisted tape [8-11] or guide vane into nozzles [12, 13]. So, an important factor to enhance turbulence intensity of jet flow is to increase entrainment of ambient fluid.

An expanding of jet outlet is a simple method to increase the entrainment of ambient fluid into the jet flow [14-19]. Generally, this method is adopted to increase the mixing and spreading of jet in combustion of industrial applications [20, 21]. However, it is rare to adopt for enhancing heat transfer on impingement region. Keawchoothong et al. [22, 23] studied a single impinging jet associated with short pipe called “air-induced duct” to increase heat transfer on impingement surface. They illustrated an ambient air was increasingly sucked through air-induced duct for enhancing turbulence intensity into a jet flow. These results can be confirmed that the heat transfer on an impingement surface increased due to using of air-induced duct. However, the applying of air-induced duct in an array of impinging jets is yet to study, and it is very interesting to adopt for heat transferring in large surface.

The aim of this research is to experimentally and numerically investigate flow and heat transfer characteristics of an impinging jet array mounting air-induced ducts. The results will be compared to the case of conventional impinging jet under the same of mass flow rate.

2. Experimental setup and method
2.1. Experimental model and parameters
The experimental model of impinging jet from nozzle with air-induced ducts is shown in Figure 1. The jets discharged from nozzle pipes and perpendicularly impinged on a target surface. The air-induced ducts were assembled at the end of pipe nozzle. Both centerlines of pipe nozzles and air-induced ducts were concentric. Nozzle arrangements were inline configuration having 5 rows x 5 columns. An origin of the Cartesian coordinates was located at the center of the jet exit. The Z-axis was on the axial of jet; X-axis and Y-axis were normal to the axial of jet in horizontal and vertical direction, respectively. The details of experimental parameters are summarized in Table 1.
2.2. Experimental setup
The diagram of experimental setup is shown in Figure 2. The blower accelerated the air which flow through the orifice flow meter and temperature controlled chamber equipped with 12-kW heater. The temperature of air jet was controlled by a temperature controller and a power controller at 27 °C. The flow rate of air jet was controlled by adjusting rotating speed of blower with an inverter. The turbulent jet discharging from round pipe with inner diameter of d=17.2 mm and length = 200 mm impinges on the wall. The pipe length was long enough to ensure the flow being fully developed flow at pipe exit. Air-induced ducts were assembled at pipe nozzle outlet. Three layers of mesh plate were mounted in jet chamber to uniform the jet temperature at pipe exit. The pipe nozzle and impingement wall were movable to adjust required positions.

![Diagram of experimental setup](image)

**Figure 2.** The diagram of experimental setup

2.3. Data reduction
The impingement plate was made of plastic plate (830 mm × 830 mm and 15 mm in thickness) and was opened with rectangular hollow on the center. A stainless steel foil with 300 mm × 300 mm and 0.03 mm in thickness was used to be impingement surface. It was tightly stretched between two copper bus bars over the rectangular hollow.

The temperature distributions on the impingement surface were measured from the rear side of impingement surface using an infrared camera. The temperatures on both impingement side and measurement side can be considered being the same since the stainless foil was sufficiently thin.
The air jet discharging from the pipe nozzle impinged on the heated surface with constant heat flux for cooling. The wall temperature on the impingement surface was measured by using the infrared camera, and the local heat transfer coefficient can be subsequently evaluated from

\[ h = \frac{\dot{q}_{\text{input}} - \dot{q}_{\text{loss,r}} - \dot{q}_{\text{loss,c}}}{T_w - T_{aw}} \]  

where \( \dot{q}_{\text{loss,r}} = \sigma \varepsilon (T_w^4 - T_s^4) \) and \( \dot{q}_{\text{loss,c}} = h_{\text{loss,c}} (T_w - T_s) \) were the heat losses to the environment by radiation and natural convection from the rear side of jet impingement surface, respectively; \( T_w \) was the local wall temperature; \( T_{aw} \) was the local adiabatic wall temperature (without heat flux); \( \sigma \) was the Stefan-Boltzman constant; \( \varepsilon \) was the emissive coefficient of captured side; \( T_s \) was the surrounding temperature; and \( h_{\text{loss,c}} \) was the heat transfer coefficient calculated from natural convective heat transfer from the vertical surface to the surrounding.

The input heat flux was calculated from

\[ \dot{q}_{\text{input}} = IV / A \]  

where \( I \) was the electrical current from supplier unit, \( V \) and \( A \) were the voltage across the bus bars and the area of the stainless steel foil, respectively.

The local Nusselt number was calculated from

\[ Nu = \frac{h d}{k} \]  

where \( d \) was the inner diameter of the pipe nozzle and \( k \) was a thermal conductivity of the air jet.

The average Nusselt number on impingement surface can be calculated from

\[ \overline{Nu} = \frac{\overline{h} d}{k} \]  

where \( \overline{h} \) was the average heat transfer coefficient calculated from wall average temperature.

3. Numerical Simulation

3.1. Computational model and boundary conditions

ANSYS FLUENT (version 15.0) was adopted in the numerical simulation. The computational model divided in three sections and boundary types are shown in Figure 3. The first section was included the main pipe nozzles diameter \( d = 17.2 \) mm, and length is 200 mm. The air-induced ducts were assembled at the end of main pipe nozzle in the second section. The length (L), inner diameter (D) of air-induced duct and the jet-to-plate distance (H) were fixed at \( L=4d \), \( D=4d \) and \( H=6d \), respectively. An impingement surface of impinging jet was included in the third section. The jet-to-jet distance (S) was varied at \( S=6d \) and \( 8d \). The details of boundary conditions were summarized in Table 2.

| Boundary condition          | Define                          |
|----------------------------|--------------------------------|
| Constant velocity inlet    | 27.21 m s\(^{-1}\) (Re=20,000) |
| Front and side surface     | Symmetry                        |
| Top and bottom surfaces    | Wall                            |
3.2. Grid generation and grid dependency

The rectangular grid was mainly applied in this numerical model. The grids in region having high velocity gradient as near impingement surface were finely controlled. The generating grid on Z-X plane at center of nozzle is shown in Figure 4. The number of generated grid was varied to achieve an accurate solution by considering the dimensionless wall distances \( (y^+) \) of the first node less than 1, calculated from

\[
y^+ = \frac{y_1 u_t}{v}
\]

where \( y^+ \) was the distance of the first node to the wall, \( u_t \) was shear velocity, \( y_1 \) is the distance to the nearest wall and \( v \) is the kinematic viscosity of air.

The effects of grid dependency on \( y^+ \) distribution on the target surface passing jet centerline are shown in Figure 5. In this model, the number of grids at 5,495,941 elements was selected to numerical simulation.
3.3. Calculation method and algorithm

Computations were conducted by solving Reynolds averaged continuity and Navier-Stokes equations under existing boundary conditions. The SST k-ω turbulence model was selected for applying in this simulation. Since, this turbulence model has been adopted in solving many numerical simulations of jet impingement problems [24]. It excellently predicted the solutions of jet impingement problems with moderate computation cost. The SIMPLE algorithm was used with second order upwind scheme for all spatial discretization. The convergence of iterative solution was insured when the residual of all the variables was less than the specified values. The specified value was $1 \times 10^{-5}$ for continuity and momentum equations.

4. Results and discussion

4.1. Flow characteristics

The streamlines on Z-X plane passing the center of nozzle are shown in Figure 6. For the case of conventional jet at S/d=6 as shown in Figure 6(a), the ambient air was entrained and mixed in the jet flow, and the wall jet turned around becoming circulation flows above impingement surface. The size
of these circulation flows became larger when jet-to-jet distance was larger (S/d=8) as shown in Figure 6(b). For the case of jet with air-induced ducts at S/d=6 as shown in Figure 6(c), the ambient air turned into the air-induced ducts. This entrainment of ambient air into the jet flow was more than the case of conventional jets. In addition, for the case of S/d=6 as shown in Figure 6(c), circulation flows occurred between air-induced ducts. These circulation flows seem that they blocked the entrainment of ambient air into the wall jet flow. When jet-to-jet distance become larger (S/d=8) as shown in Figure 6(d), the size of these circulation flows became smaller and the ambient air was entrained into wall jet flow that is similar with that case of conventional jet (Figure 6(b)).

From this flow study, it can be illustrated that the effect of mounting air-induced duct can enhance the entrainment of ambient air into the jet flow more that the case of jet without mounting air-induced duct (Conventional jets). For jet mounting air-induced duct at larger jet-to-jet distance (S/d=8), the ambient air entrained into the wall jet flow more than the case of smaller jet-to-jet distance (S/d=6). It would case to increase the heat transfer on the surface that would be discussed.

4.2. Heat transfer characteristics

The contours of Nusselt number distributions on the impingement surface are shown in Figure 7(a)-7(d). Small circle marks were identified the locations of nozzles, and the larger ones were identified the locations of air-induced duct. For all cases, Nusselt numbers were high in stagnation regions and were low in areas of interval jets. Generally, Nusselt number distributions for the case of S/d=6 were higher than the case of S/d=8. This is from the effect of interference of adjacent jets for the case of smaller jet-to-jet distance resulting on increasing turbulence intensity of jet before impingement as mentioned in previous work [25]. For the case of jet with air-induced duct for both S/d=6 and 8, areas of high Nusselt number around impingement regions were larger than those for conventional jets. This can be attributed that the mounting of air-induced duct can increase the entrainment of ambient air into the jet flow resulting on increasing of turbulence and heat transfer.

![Figure 7](image-url)

**Figure 7.** (Left) Nusselt number contours on the impingement surface and (Right) local Nusselt number distributions along X/d axis at jet center and interval jets (Experimental results, Re=20,000)

Nusselt number distributions along X/d are shown in Figure 7(e) and 7(f). The peak of Nusselts number for every nozzle took place at the stagnation points and gradually decreased along radial direction. For S/d=6 as shown in Figure 7(e), the Nusselt number distributions of jet with air-induced
duct were almost similar when compare with the case of conventional jet whereas for the case S/d=8, the Nusselt number distributions of jet with air-induced duct were higher than those of conventional jets.

Average Nusselt numbers calculated using equation (4) by averaging temperature in area of \(-6 \leq \text{Y/d} \leq 6\) and \(-6 \leq \text{X/d} \leq 6\) for S/d=6 and in area of \(-8 \leq \text{Y/d} \leq 8\) and \(-8 \leq \text{X/d} \leq 8\) for S/d=8 are shown in Figure 8. Generally, the average Nusselt numbers for all cases of S/d=6 were higher than those of S/d=8. The average values for the cases of jet with air-induced duct were higher than those of the conventional jets: 1.12% for S/d=6 and 5.41% for S/d=8. It can be noticed that the effect of mounting air-induced duct on increasing of heat transfer in case of larger jet-to-jet distance (S/d=8) was more effective than that of smaller jet-to-jet distance (S/d=6).

![](image)

**Figure 8.** Average Nusselt number on the impingement surface (Experimental results, Re=20,000)

5. Conclusions
In this paper, flow and heat transfer characteristics of impinging jet array mounting air-induced duct were studied experimentally and numerically. The results can be concluded as follow:

- The effect of mounting air-induced duct can enhance entrainment air into the jet flow resulting on increasing of heat transfer of impinging jets on target surface, especially in area of around stagnation regions.
- The effects of mounting air-induced duct on increasing heat transfer in case of larger jet-to-jet distance (S/d=8) was more effective than that of smaller jet-to-jet distance (S/d=6).

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