Control of Impingement Heat Transfer Using Mist*

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Abstract

Impingement heat transfer from a circular orifice jet by using latent heat of water mists was studied experimentally. The amounts of mists of about Zauter's mean diameter $14\text{ }\mu\text{m}$ were from 60 to 200 g/h within a range where liquid films were not formed on the target plate and mists were added near the orifice edge. Experiments covered Reynolds numbers from 12,500 to 50,000 and a heat flux is $1,400\text{ W/m}^2$. The experimental results indicate that adding mists had little influence on free jet mean velocity profiles and target plate pressure coefficients. On the other hand, mists had a strong influence on temperature and humidity profiles of a free jet and they also influenced Nusselt number distributions on the target plate. Increases of mists and Reynolds number caused increases in Nusselt number on the developed region. In addition, we investigated influence of the way mists were added and these results showed that Nusselt number was influenced not only by the amounts of mists but also by the adding method. Local Nusselt number profiles with mists were closely related to temperature distributions of the free jet at the location corresponding to the target plate.

Key words: Impingement Jet, Heat Transfer, Water Mist Cooling

1. Introduction

One of the most challenging energy problems is the effective use of energy in the improvement of heat exchanger efficiency, especially in the case of low quality energy. Impingement jet⁽¹⁻⁴⁾ has been used widely in industrial fields such as manufacturing processes of with glass or steel requiring cooling, heating and drying, because it provides a high heat transfer coefficient.

However, in general control of impingement heat transfer is very poor and only a few control methods, such as the changes in velocity profiles at the exit of a nozzle by using coaxial dual nozzles (Shakouchi⁽⁵⁾), a non-circular nozzle and a three-dimensional nozzle (Colucci⁽⁶⁾, Oyakawa⁽⁷⁾) have been reported.

Recently, the cooling method using vapor latent heat of small amounts of water mist has been considered as an active heat transfer control method. Graham⁽⁸⁾ studied water or methanol/air mist jet impingement cooling using a coaxial jet atomizer under the condition that surface-averaged heat fluxes could be dissipated with the mist. Wang⁽⁹⁾ investigated the mist/steam heat transfer of three rows of circular jet impingement in confined channel. The mist/steam flow was mixed before jets.

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In this study, the work-saving cooling mechanism and the possibility of a heat transfer enhancement from a circular orifice jet with water mists which were added near the orifice edge have been studied experimentally.

Nomenclature

A : area of the orifice
 C_p : wall pressure coefficient
 c : specific heat
 d : diameter of the orifice
 L : distance between orifice and target plate
 m : weight flow rate of the jet
 Nu : local Nusselt number on the target plate
 Nu_c : conversion Nusselt number (defined by Eq. (5))
 Nu_d : normalized maximum Nusselt number with mists (defined by Eq. (3))
 Nu_{max} : maximum Nusselt number with mists
 Nu_n : maximum Nusselt number without mist
 $\overline{Nu}$: averaged Nusselt number on the target plate (defined by Eq. (4))
 p : static pressure
 Q : latent heat
 Q_m : the amount of added mists
 q : heat flux
 Re : Reynolds number = $U_0 d / \nu_{air}$
 R_h : relative humidity
 r : radial coordinate on the target plate
 T : temperature
 U : velocity of the jet
 x : coordinate along the jet central
 y, z : coordinate along the target plate (see Fig. 2)
 λ : thermal conductivity
 ν : kinematics viscosity
 ρ : density

Subscript

air : dry air
 w : target plate
 0 : orifice exit

2. Experimental Apparatus and Method

Figure 1 shows the experimental apparatus and Fig. 2 shows the coordinate systems and symbols used in this report. An orifice is made in a brass plate 2 mm in thickness and 51 mm in diameter, d . Experimental Reynolds numbers, Re , are $(5.0, 2.5, 1.25) \times 10^4$ using a characteristic length, d , and characteristic velocity at the orifice exit, U_0 measured with an I-type hot wire anemometer.

In free jet experiments, temperature was measured with a sheath K-type thermocouple and humidity was measured with a capacity type polymer hygrometer.

In impingement experiments, heat transfer was conducted under a constant heat flux, q_w of $1,400 \text{ W/m}^2$ using direct current to a thin stainless foil of $20 \text{ }\mu\text{m}$ thickness on a target plate. A local Nusselt number on the target plate was defined;

$$Nu = \frac{q_w \times d}{\lambda_{air} (T_w - T_0)} \quad (1)$$

where λ_{air} [W/mK] is the heat conductivity of dry air, T_0 is the temperature of the air jet at the orifice edge and T_w is the temperature on the target plate. T_w was measured with Cu-Co thermocouples soldered on the back of the stainless foil. Static pressures on the target plate, p_w [kg/m²], were measured using immersed bell type precision manometers and wall pressure coefficients on the target plate, C_{pw} , defined as;

$$C_{pw} = \frac{2p_w}{\rho_{air} \times U_0^2} \quad (2)$$

where ρ_{air} [kg/m³] is the density of dry air.

As shown in Fig. 2, water mists were generated by an ultra-sound wave type humidifier, and they were added at the ambient temperature to the jet from the z -axis side near the orifice edge. Zauter's mean diameter ZMD was about 14 μ m. The amounts of added mists, Q_m , were 60, 120, and 200 g/h, a region where liquid films were not formed on the target plate. In $Re = 5.0 \times 10^4$, the amounts of mists were converted to mist densities [ppm], that is the ratio of water mist mass flow rate to air mass flow rate are as follows; 60 g/h to 415 ppm, 120 g/h to 831 ppm and 200 g/h to 1380 ppm.

Fig. 1 Experimental apparatus

Fig. 2 coordinate systems and main symbols

3. Experimental Results and Considerations

3.1 Free Jet

Figure 3 shows mean velocities, U / U_0 , in the streamwise direction of the free jet at special sections, x / d , are 8, 12 and 16. Q_m , are zero and 200 g/h for a constant $Re = 5.0 \times 10^4$. Both results show nearly the same profile indicating that mists did not have a significant influence on the free jet velocity distribution.

Figure 4 shows the local temperature, T [K] and profiles of the free jet with mists in the y direction. Re and x / d are the same as those given in Fig. 3. Q_m are 0, 60, 120 and 200 g/h. Compared with profiles without mist, temperatures near the axis center of the free jet were the lowest when mists were added. Moreover, an increase of Q_m brought a drop in temperatures in the free jet. On the downstream side, the variations of temperatures were smaller and the influences of mists were not as large.

Figure 5 shows the local relative humidity, R_h [%], distributions in the y direction of the free jet with the added mists. Q_m , Re and x / d are the same as those given in Fig. 4. R_h near the axis center of the free jet with mists was higher than without mist. Moreover, as larger mists were added, the jet became more humid. At the downstream side, the influence of mists was weak.

As mentioned previously the addition of mists to the free jet in this experimental ZMD hardly influenced its mean velocity profiles. On the other hand, mists had a strong influence on temperature and humidity profiles in the free jet.

Fig. 3 Influence of mists on mean velocity profiles ($Re = 5.0 \times 10^4$)

Fig. 4 Influence of mists on local temperature profiles ($Re = 5.0 \times 10^4$)

Fig. 5 Influence of mists on local relative humidity profiles ($Re = 5.0 \times 10^4$)

3.2 Impingement Jet

3.2.1 Pressure coefficient profile

Figure 6 shows C_{pw} profiles in y and z directions. Orifice-to-target plate spacing, L/d , is 2, 4, 8, 12 and 16 and a constant Re is 5.0×10^4 . Q_m are zero and 200 g/h. Both Q_m profiles showed the same trend indicating that the amounts of mists in this experimental range did not affect the velocity distributions of an impinging jet region. Each profile of y/d and z/d did not have an apparent difference. These results show that the direction of added mists has little influence on the pressure distributions.

3.2.2 Local Nusselt number profile

Figures 7(a) to (c) show Nu profiles on the target plate in y and z directions. L/d , are 8, 12 and 16 and Re are $(5.0, 2.5, 1.25) \times 10^4$. Q_m , are 0, 60, 120 and 200 g/h. Compared with the profiles without mist, for $L/d = 8$ and $Re = 5.0 \times 10^4$, the variations of Nu near the

Fig. 6 Influence of mists on local wall pressure coefficients profiles ($Re = 5.0 \times 10^4$)

center axis of the free jet are the largest for the maximum amount of added mists. However, in $|y/d| > 1$ and $|z/d| > 1$, Q_m had only small differences in Nu distributions. The influence of added mists on Nu decreases as Q_m decreases. In the downstream, the variations of Nu with Q_m are small. When Re are $(2.5 \text{ and } 1.25) \times 10^4$, the lower Re produces small influences of mists on Nu . The result that Nu profiles of y/d are the same as those of z/d shows that the influence of the direction of added mists on Nu profiles was small.

3.2.3 Iso-Nusselt contour maps

To recognize the influence of mists on plane variations of Nu on the target plate, Figs. 8(a) and (b) show iso- Nu contour maps for $L/d = 8$ and 16 and $Re = (5.0, 1.25) \times 10^4$, respectively. The upper and lower parts of these figures show results without mist and with mist of $Q_m = 200 \text{ g/h}$, respectively. An iso- Nu contour line with mists is wider than that without mist for both L/d . And the iso- Nu contour area increased when L/d decreased or Re increased.

3.2.4 Consideration of maximum Nusselt number

In Fig. 7, Nu near the axial center, which is a maximum Nu in each cross section, increases with the increases in Q_m and Re . Therefore, Fig. 9 shows relations between a normalized maximum Nu with mists, Nu_d , and Re . Nu_d is defined as follows;

$$Nu_d = \frac{Nu_{max} - Nu_n}{Nu_n} \quad (3)$$

where Nu_n is the maximum Nu without mist and Nu_{max} is the maximum with mists. Q_m are 60, 120 and 200 g/h. In the cases of $L/d = 16$ and 12 and $Re = 1.25 \times 10^4$, Nu_d remained unchanged regardless of Q_m or Re . In the case of $L/d = 8$, the increases of Q_m and Re brought on increases in Nu_d . For $Re = 5.0 \times 10^4$ and $L/d = 8$ and $Q_m = 200 \text{ g/h}$, Nu_d had the largest value.

Figure 10 shows relations between Nu_{max} and Q_m which were indicated as mist densities for $Re = 5.0 \times 10^4$ and $L/d = 8$. A dramatically increased Nu_{max} was observed near the border of 1000 ppm, that is, $Q_m = 145 \text{ g/h}$.

Fig. 7 Influence of mists on local Nusselt number profiles

(a) $Re = 5.0 \times 10^4$, (b) $Re = 2.5 \times 10^4$, (c) $Re = 1.25 \times 10^4$

3.2.5 Consideration of averaged Nusselt number on the target plate

Figure 11 shows the averaged Nu on the plate, $\overline{Nu}$, in order to estimate the improvement of heat transfer on the impingement jet with mists. $\overline{Nu}$ was defined as follows;

$$\overline{Nu} = \frac{I}{4\pi d^2} \int_0^{2d} 2\pi r Nu(r) dr \quad (4)$$

Fig. 8 Iso-Nusselt number contour maps

Fig. 9 Relation between normalized maximum Nusselt and Reynolds numbers

Fig. 10 Relation between maximum Nusselt number and mist density

where $Nu(r)$ was determined from the Nu profile in Fig. 7 and where r is the radial distance from the orifice center. For all L/d , when Re increased $\overline{Nu}$ increased. Especially for $L/d = 8$, $\overline{Nu}$ increased dramatically in the case of $Q_m = 200$ g/h.

Here, the result and consideration of §§ 3.1 and 3.2 are summarized. First, as shown in Fig. 6, the pressure distribution did not change with mists addition. Therefore, the kinetic energy of the jet did not change with mists so that it was supposed from the free jet velocity distribution in Fig. 3. From this result, we considered that thermal energy change of the jet by latent heat of mists was one of the causes of the increasing of Nu with mists as shown in Fig. 7.

Next, compared with the result in Figs. 9 and 11, in the case of $Re = 5.0 \times 10^4$ and $Q_m = 200$ g/h, the Nu_{max} was 2.2 times and $\overline{Nu}$ was 1.8 times as much as without mist. It was suggested that the added mists did not influence on the whole jet but had a strong influence on near the axis center of the jet. In this result, it might be relevant to the followings, the temperature and humid distributions with mists in the cross section of the free jet stream

direction were not uniform and then the decreasing of temperature near the axis center of the free jet with mists was large.

Moreover, as shown in Figs. 9 and 11, Nu increased with increasing Q_m , increasing Re and decreasing L/d . We suggested that it was because the increasing of Q_m brought the increasing of latent heat and the increasing of Re brought the increasing of mass flow rate, that is, the flow with decreased temperature by latent heat of mists and included mists reached downstream.

Fig. 11 Relation between averaged Nusselt and Reynolds number

3.3 Evaluation of the added mist

As mentioned in §3.1, added mists in the free jet hardly influenced its mean velocity profiles. On the other hand, the mists had strong influence on temperature and humidity profiles in the free jet. Considering these points, two types of conversion were tried. A conversion temperature was T_c was calculated by (1) the vapor latent heat of added mists and (2) the local temperature distribution in the free jet at the location corresponding to the target plate. Moreover a conversion Nu , Nu_c was defined as follows;

$$Nu_c = \frac{q_w \times d}{\lambda_{air}(T_w - T_0 - T_c)} \quad (5)$$

3.3.1 Evaluation by vapor latent heat

We supposed that mists added to the jet vaporized perfectly. The temperature of the jet decreased uniformly by the latent heat, Q [kJ/s], and converted to a conversion temperature, T_c .

$$Q = c_{air} m_{air} T_c \quad (6)$$

where c_{air} [kJ/(kgK)] is the specific heat of dry air, and m_{air} [kg/s] was the weight flow rate of the dry air jet showed by the following equation;

$$m_{air} = \rho_{air} A_0 U_0 \quad (7)$$

where A_0 is the area of the orifice. It is assumed that temperature equilibriums in the entrained ambient air are neglected and mists did not influence the jet flow behavior.

Figures 12(a) and (b) show the evaluated results of Nu in the cases of $L/d = 8$ and 16, and $Re = 5.0 \times 10^4$. Dark and open symbols are Nu and Nu_c , respectively. In the case of $L/d = 8$, compared with Nu without mist, Nu_c increased a little near the center of the jet and decreased in the far center area. In the case of $L/d = 16$, the converted value is smaller than that without mist. Moreover, in all y/d this conversion was lower than without mist and this value was an over estimation.

Fig. 12 Converted Nusselt number by latent heat ($Re = 5.0 \times 10^4$)

3.3.2 Evaluation by temperature distribution in a free jet at the location corresponding to the target plate

The effect of added mists on the temperature distribution in the free jet did not have the uniformity shown in Fig. 4. The former estimation in §3.3.1 did not take account of these facts. Then, added mists gave local temperature decreases as a result of vaporizing mists and its mixing with the surrounding fluid. In this section, it is supposed that a jet flow with the non-uniform temperature profiles at the location of the target plate in the free jet directly impinged on the target plate as an oncoming flow. Namely, a modified temperature $T_c(r)$ is;

$$T_c(r) = T(r) - T_0 \quad (8)$$

where $T(r)$ is the local temperature in the free jet at the location corresponding to the target plate.

Figures 13(a) and (b) show the evaluated results in $Re = 5.0 \times 10^4$ for $L/d = 8$ and 16, respectively. In the case of $L/d = 8$, converted Nu profiles agreed with Nu profiles without mist very close to the center of the jet. In the case of $L/d = 16$, the converted values are large in all areas and this value was a little under estimation.

Compared with the former estimation based on latent heat, in §3.3.1, this estimation is in good agreement with the result without mist. From these estimations, we presumed that Nu distributions with mists were closely related to temperature distributions in the free jet at the location corresponding to the target plate.

3.4 Difference in the Addition Method of Mists

3.4.1 Free jet

We also studied the influence of the way mists were added in order to confirm our conclusions in §3.3.2. Figure 14 shows the mechanics of adding mists at the same Q_m . Q_m was a constant 240 g/h in all cases and the mists were added from (a) one pipe, (b) two pipes and (c) three pipes. In the above three cases for $Re = 5.0 \times 10^4$ and $x/d = 8$, Figs. 15(a) and (b) show temperature and humidity profiles, respectively.

Fig. 13 Converted Nusselt number by local temperature distribution in free jet ($Re = 5.0 \times 10^4$)

When mists were added from one pipe, differences between the free jet and ambient temperatures were the largest both at the y - and z -axes. Moreover increases in the number of pipes produced a decrease in the temperature differences between jet and ambient ones. Humidity distributions showed the same trend. We assume that increasing the number of pipes reduced the density of mists at an added location and that the mists evaporated easily and diffused towards the upstream side.

Fig. 14 The mechanics of adding mists

3.4.2 Impingement jet

Figure 16 shows Nu profiles of impingement jets in the case of $L/d = 8$ with the same conditions of Q_m and Re as those in Fig. 15. Nu was greatest with mists added from one pipe while the temperature distribution of the free jet on the cross section was the smallest as shown in Fig. 15(a). From these results, the effects of mixture diffusions by added mists appear to be related to temperature distributions in the free jet. Figure 17 shows the results estimated by the same method as in §3.3.2. These results show a fairly good estimation of the effect of additional mists in all cases accounting for the accuracy of the local temperature measurement in the free jet.

Fig. 15 Profiles with same amount of mists 240 g/h ($Re = 5.0 \times 10^4$)

(a) Local temperature, (b) Local relative humidity

Fig. 16 Local Nusselt number profiles with same amount of mists 240 g/h ($Re = 5.0 \times 10^4$)

Fig. 17 Converted Nusselt number by local temperature distribution in free jet ($Re = 5.0 \times 10^4$)

However, several factors which contributed to evaporation such as a mist particle diameter or an ambient temperature and humidity might have great influence on the heat transfer. For example, it is considered that the evaporation time of mists becomes shorter when the mists particle diameter is smaller. Therefore, the mists diameter and distribution depend heavily on the temperature of the free jet. Moreover, in the case of the ambient temperature and humidity near saturation, it is suggested that the liquid film is formed on the target plate for large L/d .

4. Conclusion

When mists about Zauter's mean diameter $14\mu\text{m}$ were added to the jet from near the orifice edge within a range where liquid films were not formed on the target plate for $L/d = 8$ to 16 and $Re = (5.0, 2.5, 1.25) \times 10^4$, the following conclusions were obtained;

- Added mists did not influence the free jet mean velocity profiles and the target plate pressure coefficients.
- Added mists varied the local temperature and humidity distributions in the free jet and these distributions were not uniform, and Nu distribution on the target plate.
- In the same method of mists addition, Nu increased with increasing the mist amounts, increasing Re and decreasing L/d .
- The mechanism of adding mists at the same amounts of mists brought the difference of temperature and humidity distributions in the free jet and Nu distribution on the target plate.

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References

- (1) Shakouchi, T., Jet Flow Engineering, (2004), Morikita Publishing Co., Ltd.
- (2) G. N. Abramovich, The Theory of Turbulent Jets, (1963), The M. I. T. Press
- (3) N. Rajaratnam, Turbulent Jets, (1976), Elsevier Scientific Publishing Company
- (4) P. M. Sforza, et al., Studies on Three-Dimensional Viscous Jets, AIAA J., Vol. 4, No. 5 (1996), pp. 800-806.
- (5) Shakouchi, T., et al., Flow Characteristics and Control of Turbulent Annular Jet (Effect of Coaxial Round Jet), JSME (Ser. B) (in Japanese), Vol. 63, No. 614 (1997), pp. 3278-3286.
- (6) Colucci, D. W. and Viskanta, T., Effect of Nozzle Geometry on Local Convective Heat Transfer to a Confined Impinging Air Jet, Experimental Thermal and Fluid Science, Vol. 13 (1996), pp. 71-80.
- (7) Oyakawa, K., et al., Effects of Nozzle Configuration on Impingement Heat Transfer, Proc. of ASME / JSME Thermal Eng. Conf., Vol. 1 (1995), pp. 377-383.
- (8) K. M. Graham and S. Ramadhyani, Experimental and Theoretical Studies of Mist Jet Impingement Cooling, Journal of Heat Transfer, Vol. 118 (1996), pp. 343-349.
- (9) T. Wang, et al., Mist/steam Heat Transfer of Multiple Rows of Impinging Jets, Int. J. of Heat and Mass Transfer, Vol. 48 (2005), pp. 5179-5191.