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EXPERIMENTAL STUDY OF TWO-PHASE AIR-WATER MIST JET IMPINGEMENT COOLING ON CYLINDER

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ABSTRACT

Experimental investigation on heat transfer mechanism of air-water mist jet impingement cooling on heated cylinder are presented. The cylinder is heated to non-boiling temperature under constant heat flux condition. Multiple experimental tests are conducted to investigate the effect of Reynolds number and mist loading fraction on heat transfer rate. In the experimental parametric study, the Reynolds number Re_d is defined on the outer diameter of the annular air jet,d and varied from 15000 to 30000. The mist loading fraction, f was varied from 0 to 1.0% and the nozzle-to-surface spacing was maintained at h/d=40. The experiment results indicate that increase in mist loading and Reynolds number increases the heat transfer rate. On further investigation, existence of secondary maxima Nusselt number was also noticed.

***Keywords*:** mist jet impingement, heat transfer, mist loading, heated cylinder.

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NOMENCLATURE

d	nozzle outer diameter, m
D	cylinder diameter, m
f	mist loading fraction
h	nozzle to target surface distance, m
L	length of cylinder, m
Nu	local Nusselt number
Nu_z	Axial Nusselt number
Nu_θ	Circumferential Nusselt number
Nu_{mix}	Stagnation Nusselt number of the air-mist mixture
Nu_{air}	Stagnation Nusselt number of air only condiition
Re_d	Reynolds number based on nozzle outer diameter
$\dot{m}_l$	mass flow rate of water, kg/s
$\dot{m}_a$	mass flow rate of air, kg/s
$\dot{m}_T$	total mass flow rate , kg/s
q''	heat flux, W/m^2
T	temperature, K

INTRODUCTION

Single phase jet impingement cooling has been widely implemented due to its effecient heat removal capability from heat generation sites. It has shown high heat transfer rate and has been extensively used in applications such as cooling of electronic compomponents, metal cutting processes and food processing unit etc. Sustantial work has been carried out in understanding the heat transfer rate for single-phase jet impingemnt cooling [1-5].

In recent trend, two-phase jet impingement cooling has been implemented, and superior heat transfer characteristic has been observed in such cooling with respect to single-phase jet impingment [6-8]. Mist is the entrainment of very fine liquid droplets in jet of air, and can also be considered as two-phase jet impingement process. Mist are formed due to shear atomization of liquid caused by high pressure annular air jet[9].

Mist jet impingement cooling on cylinder has yet to receive as much focus and significance as mist jet impingement cooling on flat plate. Experimental work has been carried out in boiling and non-boiling region. The area of mist jet impingement in non-boiling region can be further investigated, which gives the motivation of present study of air-water mist jet impingement cooling on cylinder.

Graham et al. [10] carried out experimental study on heat transfer characteristic for air-methanol and air-water mist jet impingement cooling on flat target surface. Experiments were conducted in the non-boiling region for jet Re in the range 6,000-10,000;m_l of methanol and water was varied from 3.7-4.1 ml/min and 2.4-6.9 ml/min respectively. Based on experimental study, they proposed an analytical model that predicts the liquid film thickness and rate of heat transfer. Kanamori et al.[11] studied experimentally the effect of air-water mist on heat transfer rate and flow characteristic on a flat target surface in the non-boiling region for Re = 12,500-50,000; m_l of water =1 - 3.33 ml/min; f=0.025-0.276, h/d = 2-16. They reported heat transfer rate enhanced with increasing the mass loading of liquid-to-air, increasing Re and decreasing h/d spacing. Lyons et al. [12] carried out experimental study to investigate the relation between heat transfer and fluid flow of air-water mist jet impingement on flat heated plate for h/d = 7.5 and 15.5, $Re_{d(hyd.)}$ = 4800, 7400 and 10300;m_l of water = 0.0046-0.45 ml/min and f = 0.02% -3.8% . They reported that atomization rate and amount of droplet entrainment affects the heat transfer rate, and that droplet diameter decreases with decrease in water flow rate and increase in air pressure. Also increase in heat transfer rate with increase in liquid loading fraction was observed. Quinn et al. [13] investigated the surface wetting and heat transfer phenomena at low, intermediate and high air-water mist loading condition on heated flat surface. Experiments were conducted in the non-boiling region for jet Re=4500 with m_l of water = 0.035-0.281 ml/min, f=0.003-0.024, h/d = 5 and 10. Three different regimes of mist loading fraction were reported, where at low mist loading fraction, the liquid droplets impinge on the surface producing small slugs on the surface. These slugs merge with each other in the intermediate loading conditions to form localized films. At high loading conditions these localized films merge to form a complete film of liquid.

Very limited literature is available on mist jet impingement cooling on cylinder. Buckingham et al. [8] investigated the heat transfer mechanism in spray cooling of air-water impingement on cylinder. Experiments were carried out in the boiling region for centre line droplet velocity of 15.3 m/s with m_l of water = 97-603 ml/min. Two distinct heat transfer regime were reported, in radiation dominated region the heat transfer is due to single phase flow. In covection dominated region, evaporation of droplets occurs near the boundary layer increasing the heat transfer rate. Lee et al. [14] also conducted experimental analysis to investigate air-water mist cooling and wetting phenomena on heated cylinder in the boiling region for droplet velocity ranging from 3.55-12.8 m/s with m_l of water = 5-22 l/min. They reported that surface wetting is a result of penetration of droplets into thermal boundary layer and striking the cylindrical surface. They also proposed analytical models which are in good agreement with the experimental data. Issa et al. [15] conducted both experimental and numerical analysis for air-water mist jet cooling of heated cylinder in the nucleate boiling region for air jet Re_d in the range1,000-3,000; m_l of water = 15.7 – 126.18 ml/min; f =1.6 – 15.5, h/d = 100 was maintained. A 2D transient heat transfer model was developed using ANSYS FLUENT for numerical analysis, with droplets modelled in Lagrangian frame and k-ε turbulence model was considered for continuous phase. They reported that the heat transfer coefficient is highest at the stagnation point, and gradually decreases over the cylinder. Based on simulation analysis, it was reported that a liquid film forms on the front portion of the cylinder under low droplet velocities and water flow rates.

To the best of our knowledge, heat transfer study on air-water mist jet impingement cooling on cylinder in non-boiling region is not available. Experimental analysis will be useful in understanding the heat transfer of mist jet impingement on cylinder. Hence, the objective of the present work is to conduct experimental study to investigate the effect of Reynolds number and mist loading fraction on the heat transfer mechanism on heated cylinder subjected to constant heat flux condition in non-boiling region. The variations in heat transfer coefficient along axial and circumferential direction have been reported.

EXPERIMENTAL SET-UP

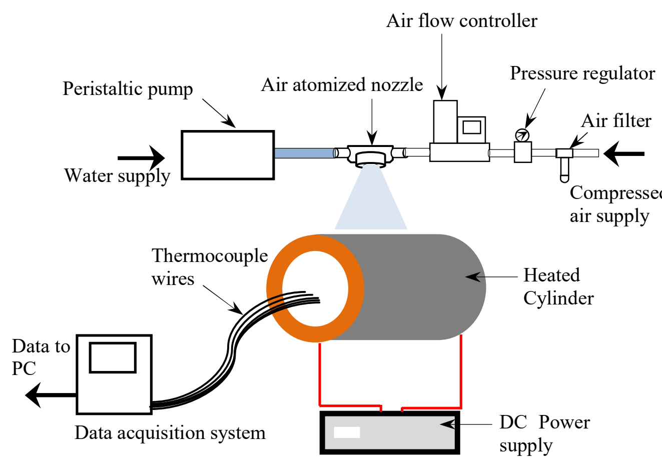

FIGURE 1. Experimental set up.

A schematic diagram of the present experimental set up is shown in Fig.1. Compressed air is supplied to the air atomizing nozzle through an air filter, pressure regulator and a flow control valve. Alicat-MCR-3000SLPM-D-PAR is used to control the air flow, which has an accuracy of ±1% over a range of 0-3000 SLPM. Ravelhiteks RH-P100VS-100 peristaltic pump is used to supply water to the atomizing nozzle. The pump delivers water at 0.1-99.9ml/min with an accuracy of ±1 %. Spraytech DASA 050 nozzle is used to generate the air-water mist.

The diameter and length of the cylinder are 50.5mm and 255mm respectively, and made from a smooth phenolic tube. A thin 25µm SS-304 foil is wound over the tube to obtain a constant heat flux condition.The temperature variation is measured with T-type thermocouples, which are located at specific locations along axial and circumferential direction of the cylinder as shown in Table 1. These thermocouples are then connected to NI Data Acquisition system (DAS) and the temperature readings are recorded in the computer. DC power is supplied to the end terminals of the cylinder, which heat up the SS foil.

TABLE 1: POSITION OF THERMOCOUPLES FROM STAGNATION POINT, T_0

Axial		Circumferential	
T_1	5 mm	T_{11}	30°
T_2	5 mm	T_{12}	45°
T_3	10 mm	T_{13}	60°
T_4	15 mm	T_{14}	90°
T_5	20 mm	T_{15}	120°
T_6	30 mm	T_{16}	135°
T_7	40 mm	T_{17}	180°
T_8	50 mm		
T_9	60 mm		
T_{10}	100 mm		

EXPERIMENTAL ANALYSIS

The mist loading farction (f) is defined as the ratio of mass flow of the liquid to that of the total mass flow of the system.

$$f = \frac{\dot{m}_l}{\dot{m}_T} \tag{1}$$

Where $\dot{m}_l$ is the mass flow rate of water and $\dot{m}_T$ is the total mass flow rate of the system, given by:

$$\dot{m}_T = \dot{m}_l + \dot{m}_a \tag{2}$$

The Reynolds number (Re_d) is defined based on the total mass flow and outer diameter of annular air jet opening:

$$\mathrm{Re}_d = \frac{4\dot{m}_T}{\pi d \mu_{eff.}} \tag{3}$$

Where$\mu_{eff.}$ is effective viscosity of mist calculated with the help of Taylor expansion [16].

The local Nusselt number is calculated as:

$$Nu = \frac{q''_{convection}.d}{(T_w - T_{ref.})k_{eff.}} \tag{4}$$

Where d is the outer diameter of annular air jet opening, T_w is the local surface wall temperature; $T_{ref.}$ is the temperature near the exit of the nozzle and $k_{eff.}$ is the effective thermal conductivity of the air-mist system.

The convective heat flux is calculated as:

$$q''_{convection} = q''_{all} - q''_{radiation} \tag{5}$$

$$q''_{all} = \frac{VI}{A} \tag{6}$$

Where V is the voltage, I is the current and A is the surface area of the cylinder.

$$q''_{radiation} = \varepsilon\sigma(T_w^4 - T_\infty^4) \tag{7}$$

The parametric study of the present study is shown in Table 2.

TABLE 2: PARAMETERS FOR PRESENT STUDY

h/d	40
f	0, 0.25%, 0.50%, 0.75% & 1.0%
Re_d	15000, 20000, 25000 & 30000

RESULT AND DISCUSSION

The Nusselt numbers along the cylinder were obtained by measuring the local temperature with thermocouples. The variation of heat ransfer rate with respect to mist loading fraction and Reynolds number are presented.

Effect of Reynolds Number

Fig. 2 illustrates the variation of local Nusselt number along the cylinder for $15{,}000 \leq Re_d \leq 30{,}000$ at h/d=40 and $0 \leq f \leq 1.0\%$. The heat transfer rate increases with the increase in Reynolds number.

The stagnation nusselt number is high for high Reynolds number. The local Nusselt number decreases along the axial and circulferential direction. A secondary maxima is observed at z/D=0.3 along axial direction and 30° along circumferential direction.

Effect of mist loading

Fig. 3 illustrates the variation of local Nusselt number along the cylinder for $0 \leq f \leq 1.0\%$ at h/d=40 and $15000 \leq Re_d \leq 30000$. The heat transfer rate increases with the increase in mist loading fraction.

The local Nusselt number increases from stagnation point to the secondary maxima and decreses monotonically along axial and circumferential direction. The significance of secondary maxima is neglisible for low mist loading, $f = 0$ and 0.25% with respect to higher mist loading.

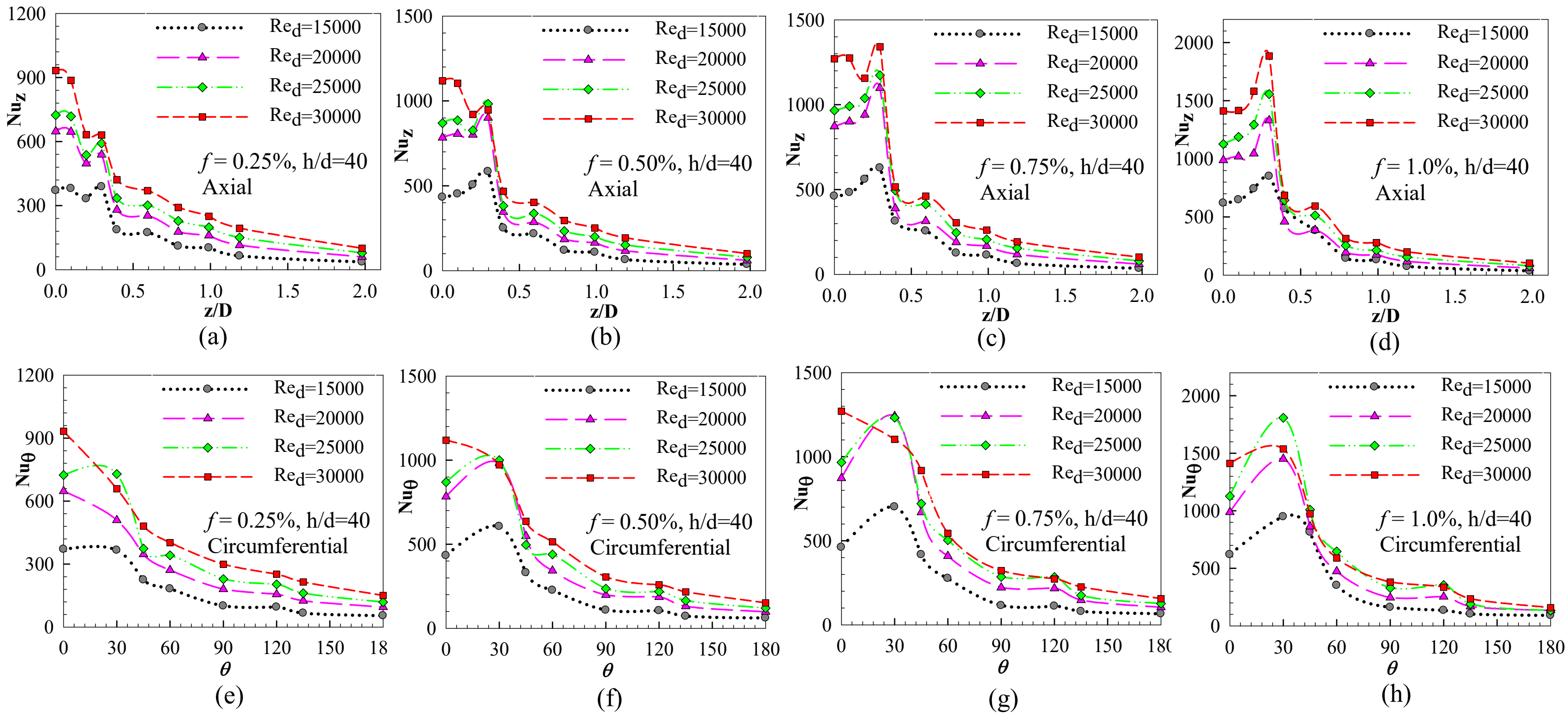

FIGURE 2. Effect of Reynolds number at h/d = 40 and Re_d = 15000, 20000, 25000 and 30000 for (a) f = 0.25%, axial (b) f = 0.50%, axial (c) f = 0.75%, axial (d) f = 1.0%, axial (e) f =0.25%, circumferential (f) f =0.50%, circumferential (g) f =0.75%, circumferential and (h) f =1.0%, circumferential.

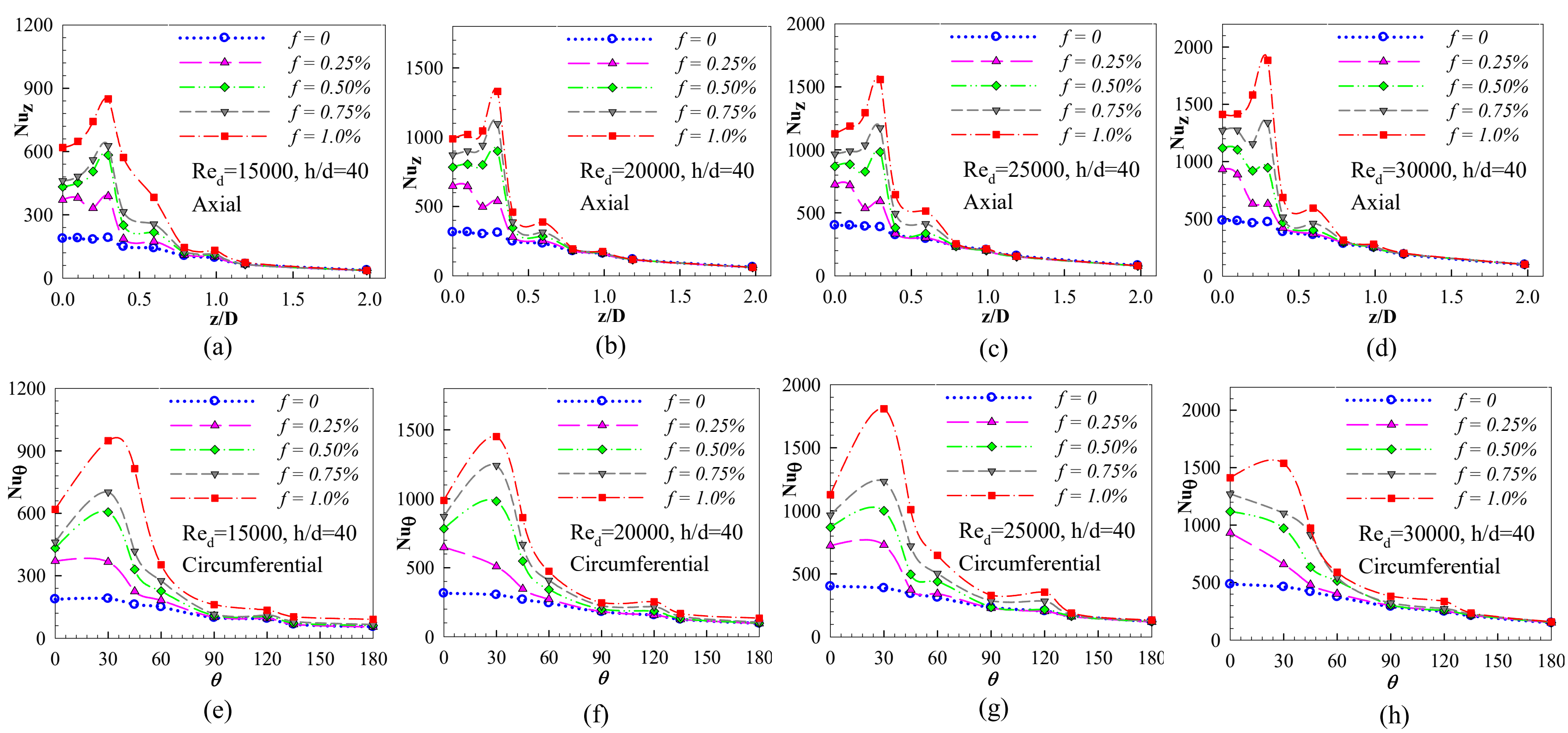

FIGURE 3. Effect of mist loading fraction at h/d = 40 and *f* = 0, 0.25%, 0.50%, 0.75% and 1.0% for (a) Re_d=15000, axial (b) Re_d=20000, axial (c) Re_d=25000 , axial (d) Re_d=30000, axial (e) Re_d=15000, circumferential (f) Re_d=20000, circumferential (g) Re_d=25000, circumferential and (h) Re_d=30000, circumferential.

The overall increment in the stagnation Nusselt number for the air-mist misture to that of single air condition is shown in Fig. 4. It is clear from the Fig. 4, that heat transfer in air-mist jet impingement is superior to that of single phase airjet impingement cooling.

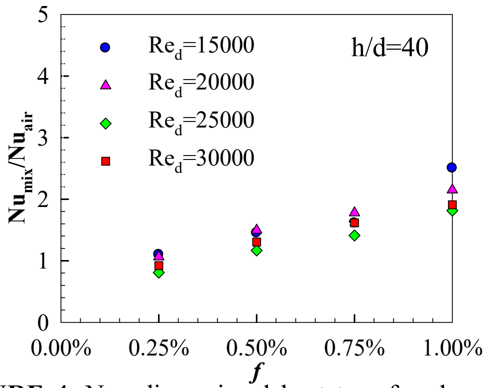

FIGURE 4. Non-dimensional heat transfer characteristic, $\frac{Nu_{mix.}}{Nu_{air}}$ at stagnation point for at f and Re_d.

CONCLUSION

Experimental investigation on the heat transfer rate over heated cylinder has been carried out. And effect of Reynolds number and mist loading fraction has been reported.

The following conclusions can be drawn from the present studies.

1. With the increase in Reynolds number, the heat transfer rate increases in both axial and circumferential direction.
2. Increase in mist loading fraction also leads to increase in heat transfer rate.
3. Secondary maxima in Nusselt number is observed in axial and circumferential direction. However , the significance of the secondary maxima is neglisible for low mist loading fraction.
4. A 1.0% mist loading increases the heat transfer rate by almost 200% with respect to single phase air impingemnt cooling.

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