Heat transfer and flow in an atomizing mist jet: a combined hot film and shadowgraph imaging approach

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Abstract. This paper presents research in the area of heat transfer and fluid dynamics in an impinging atomizing air/water mist jet. Time averaged and fluctuating local surface heat transfer results obtained by microfoil and hot film sensors are correlated with flow field measurements of droplet diameter and velocity obtained by shadowgraph imaging and droplet tracking velocimetry. This paper seeks to understand the linkage between the atomization process in the nozzle, the two-phase flow dynamics and the surface heat transfer characteristics.

1. Introduction

Impinging air jets have long been known to achieve superior heat transfer coefficients, with the variation in their local heat transfer coefficients also lending itself to application in areas of large temperature gradients. Their ability to achieve effective cooling rates has led to the implementation of jet cooling in many situations including the replacement of lubricants in some machining operations. Previous work in the research group investigated their effect on grinding temperatures [1]. The addition of a liquid phase in the form of a mist leads to a further improvement in performance. Lee et al. [2] state that at droplet diameters of 30-80 μm, a "superbly effective cooling scheme" is present. Convective heat transfer coefficients can increase by up to 10 times, through evaporation of an "ultra-thin" liquid film (50-100 μm). A spray is obtained by pressurising the water in the nozzle in order to atomize it. Since mist jets use the air pressure to atomize the water, they allow smaller droplet size [3] and the liquid flow can be controlled with less atomization constraints.

Much research has been undertaken in both air jet and spray cooling, whereas fewer studies have been undertaken in mist jet cooling. These studies have been spread across water mist/steam [4], various liquids/air [5] and water/air [6] and generally involve high wall temperatures; the effect of droplet size has not been the focus of investigation. There is some disagreement over the effect of droplet velocity on heat transfer. Sozbir et al. [6] suggests that droplet velocity does not significantly influence the water mist heat transfer, while Chen et al. [7] note a strong dependence of heat transfer on droplet velocity for their water spray. A preliminary study of heat transfer enhancement in mist jets has been reported by Lyons et al. [8], in which the fluctuating component of the heat transfer coefficient has been compared to that for an air only flow, whereas limited flow visualisation measurements were presented by Lyons et al. [9] for the same nozzle as used in the current study.

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Misting nozzles that employ shear driven atomization of a liquid into a gaseous medium typically employ annular exit conditions so that gas is co-flowing annularly around a central jet of liquid. Excluding bistability and hysteresis [10], annular nozzles experience recirculation of the flow at low nozzle to plate spacings, H/D, whereas at H/D ratios above 6 Lyons et al. [11] report that the annular jet behaves like a jet of circular exit geometry. Ko and Chan [12] and Chan and Ko [13] showed that the annular nozzle behaves rather like a rectangular nozzle wrapped into an annulus. Taking a section through the centre of the jet as shown in figure 1, the jet displays as two separate nozzles, each with their own potential core and entrainment effects. Between the two jets, there exists a recirculation zone where the jets initially start mixing. In an annular jet used for atomization, the water droplets are “injected” into the flow within the initial mixing region and are entrained into the flow.

![Figure 1. Annular Nozzle: Left: Spray Profile of an annular nozzle with co-flowing water and air streams. Middle: Definition of flow structures, [12]: 1, 2, 3, initial, intermediate and fully developed merging zone; 4, recirculation zone; 5, potential core; 6, tore; 7, stagnation point; 8, reattachment point. Right: Schematic of test apparatus](image)

A spray or mist can be defined as a collection of liquid droplets, moving in a controlled fashion along with a gaseous phase, containing a variety of droplet sizes. Fluid properties, including viscosity, surface tension and density, affect how a spray is formed [14]. Atomization is the breakup of a continuous body of fluid into many small bodies of fluid with new surface areas. By breaking up a volume of fluid into smaller bodies with the same combined volume, the overall potential energy increases because the overall gas/liquid interface surface area increases. This energy is typically supplied by strong shear forces acting on the liquid flow, e.g. by entrainment of liquid in a fast-moving gas jet [15]. The typical energy cascade is as follows: potential energy stored as pressure in liquid and gas reservoirs is converted into kinetic energy in the nozzle, the breakup of liquid bulk into small droplets with surface tension converts this back into potential energy; some kinetic energy will most likely be dissipated as heat by friction. In an annular atomisation nozzle, the liquid is injected into an annular co-flowing gas stream, which breaks up due to gas-liquid momentum transfer, or air-blast atomization [16]. An understanding of the atomization process is not only of fundamental importance, but also relevant in combustion, power, medical and manufacturing processes.

This paper seeks to understand the linkage between the atomization process and the heat transfer characteristics of the nozzle. The overall aim of the research is to characterize the heat transfer from a heated plate to an impinging mist jet and to correlate it with the two-phase flow dynamics.

2. Materials and Methods
The research has been undertaken using a combination of techniques for two-phase flow visualisation and heat transfer measurement. The test surface is an instrumented isothermally heated copper plate. A thermopile heat flux sensor was used to provide time averaged local heat transfer coefficients, denoted by h [W/m²K]; heat transfer coefficients associated with the jet are calculated using the equation $h = q''_{conv}/(T_s - T_{jet})$, where $q''_{conv}$ is the convective heat flux, $T_s$ is the plate surface temperature and the reference temperature $T_{jet}$ is the temperature of the air jet flow. Constant temperature anemometry (CTA) in conjunction with a surface mounted hot film sensor provides a
fluctuating component of the heat transfer signal. Flow visualisation and measurement is performed using high speed shadowgraph imaging with the fluctuating component of the heat transfer being matched to the shadowgraph data. The PIV system comprises of a Quantronix Darwin-Duo Nd:YLF twin cavity laser (maximum pulse energy of 10 mJ at a repetition rate of 1000 Hz) and a Photron Fastcam SA1 (Lavision High Speed Star 6) CMOS camera, (1024 x 1024 pixels, 12 bit) with a 105mm macro lens. The shadowgraphy technique provides excellent droplet imaging in the diameter range 10µm-100µm. Visualization of droplet shape and spray morphology, such as the observation of break-up regions close to the nozzle exit, is possible with this technique [17].

The nozzle used is a Spraying Systems 1/8VAU-SS+SUV152-SS nozzle. The structure of the nozzle provides an inner water jet, surrounded by an annular co-flowing jet of air; the hydraulic diameter of the air jet is 0.43mm. Variation of either the air or water pressure controls the droplet size. The nozzle is a shear driven atomizing nozzle, i.e. the primary mode of atomization is by momentum transfer from the gas to the liquid flow or shearing of the water jet by the gas stream. The nozzle and the resulting mist spray can be seen in figure 1.

3. Results

3.1 Local convective heat transfer coefficient profiles

![Figure 2](image-url)

**Figure 2.** Heat transfer profiles for Reynolds numbers of 4800, 7400 and 10300; fixed water flow rate of $7.75 \times 10^{-8}$ kg/s giving water loading fractions of 0.04%, 0.03% and 0.02%; a) H/D = 7.5, b) H/D = 15.5. The radial displacement, r, and the nozzle to plate spacing, H, are both normalised by the outer diameter, D, of 1.726mm in all cases.

In all cases, for the Reynolds number the characteristic length is the hydraulic diameter, as per previous work within the department [11] and the work of Ichimiya [18] as this choice takes into account the flow conditions at the nozzle exit; the characteristic velocity for the Reynolds number is the velocity of the air jet flow as measured by an Alicat Scientific flow meter with a range of 0-500 SLPM.

Previous work by the authors [8, 9] has shown the increased cooling performance of a mist jet over that of an air only flow, with enhancement levels of up to an order of magnitude. In the stagnation zone, the increase in air Reynolds number causes an increase in heat transfer, which diminishes with increasing radial distance. This is evident from figures 2 and 3. However, for all but the lowest water flow rate and jet to plate spacing (figure 2(a)), the heat transfer appears to level off at the intermediate Reynolds number tested.
All profiles show the typical bell shaped curve, although the drop-off in heat transfer rate appears much sharper than is typical for air only flow. There exists a rather steep drop at a radial distance from the nozzle centre of 4-5 diameters. After this drop, the heat transfer rates merge regardless of air or water flow rate. Typically at a radial distance of 6-8 diameters the mist jet creates pooling of water which impacts on the jet’s effectiveness. In this pooled water zone, the energy of the air flow is no longer great enough to propel the water along the surface, leading to the heat flux sensor being submerged under a thin (1mm) layer of water.

Figure 3. Heat transfer profiles for Reynolds numbers of 4800, 7400 and 10300; fixed water flow rate of $7.5 \times 10^{-6}$ kg/s giving water loading fractions of 3.8%, 2.5% and 1.9%; a) H/D = 7.5, b) H/D = 15.5.

From figures 4 to 6 the higher water flow rate of $7.5 \times 10^{-6}$ kg/s outperforms that of $7.75 \times 10^{-8}$ kg/s flow rate although the level of enhancement achieved at the lowest water flow rate is still very significant. In fact as shown in figure 5a, the maximum heat transfer coefficient is even greater for the lowest water flow rate. More experiments at intermediate mass fractions will need to be performed to fully explain this behaviour. For r/D greater than around 6, augmentation again drops away, due to a drop off in water droplets at the outer radial distances. For r/D greater than 8 this drop-off can be
attributed to the pooling of water discussed previously. At H/D of 7.5 compared to 15.5, the higher water flow rate maintains effective heat transfer for greater radial distances than the lower water flow rate. This can be explained by the shadowgraph imaging results in section 3.2 (figure 9), which show that there exist a greater number of larger droplets in these regions for a higher water flow rate. The repeatability of the testing procedure has been verified to within 9% for the $7.75 \times 10^{-8}$ kg/s water flow rate and 8% for the $7.5 \times 10^{-6}$ kg/s water flow rate both in terms of the stagnation heat transfer coefficient.

Figure 5. Heat transfer profiles for water flow rates of $7.75 \times 10^{-8}$ kg/s and $7.5 \times 10^{-6}$ kg/s; Reynolds number = 7400; a) H/D = 7.5, b) H/D = 15.5.

Figure 6. Heat transfer profiles for water flow rates of $7.75 \times 10^{-8}$ kg/s and $7.5 \times 10^{-6}$ kg/s; Reynolds number = 10300; a) H/D = 7.5, b) H/D = 15.5.

Figure 7 shows a plot of fluctuating heat transfer (a root mean square representation (RMS) of the variation in heat transfer) with respect to radial distance from the nozzle exit. An off-centre peak is typically attributed to the shear layer not penetrating the potential core [19], however in this case it is more than likely due to the atomization profile outlined in the shadowgraphy section below (figure 9). In the stagnation zone, mist particles exist within the potential core. There exist significant numbers of particles and their atomization frequency is fairly constant. With increasing radial distance from the
nozzle centre, the number of particles drops and far away from the stagnation zone, the particles are
atomized at a random frequency. This contributes to the peak in fluctuating heat transfer coefficient at
approximately two diameters. Outside of this zone the previously mentioned pooling phenomenon
probably contributes to the reductions in fluctuations.

![Fluctuating Data Component](image)

**Figure 7.** A root mean square plot of the fluctuating heat transfer component in terms of RMS values,
for Re=10300, H/D=30 and a water flow rate of 7.75 x 10^-8 kg/s

3.2 Mist flow characteristics based on shadowgraph imaging

![Nozzle Exit, Water flow rate of 7.75 x 10^-8 kg/s](image)

![Nozzle Exit, Reynolds number of 7400](image)

**Figure 8.** Droplet Diameter and Particle number variation with air and water flow rates

At the lower air flow rates there is less energy available for atomization and thus it takes longer for
droplets to be stripped off the water flow; the flow stays as a coherent body of fluid to greater
distances from the nozzle exit. From figure 9a it can be seen that the particle size tends to decrease as
the air flow is increased, while figure 9b shows increasing particle sizes with water flow rate. This
occurs because the process of atomization is shear driven atomization, whereby particles tend to
increase with decreasing slip velocity. For low air flow rates, the energy available is not sufficient for
the larger droplets to be broken into smaller droplets. An obvious consequence of this larger droplet
distribution is that for a constant water flow rate, the particle intensity will decrease, evident from
figure 8a. At greater radial distances, droplet diameter and intensity both decrease as seen from figure
9b. At a radial distance of 10 mm, the droplets are probably outside the mist spray core, and thus
fewer droplets will exist; there exists a zone where particles appear suspended in the ambient air. Smaller particles require less energy to be thrown out of the core and thus it is more probable for smaller particles to appear in this region.

Figure 9. Shadowgraphy data with vertical and radial distance from nozzle exit

As can be seen from figure 9a particle diameters and particle number densities or intensities vary with both vertical distance from the nozzle exit and radial distance from the centre of the nozzle body. Where the vertical distance is varied, the radial location is that of the geometric centre, and where radial distance is varied, this is at a vertical distance of 10mm from the nozzle exit. N is the number of particle recorded per interrogation window [4mmx4mm]. For a Reynolds number of 7400 and a water flow rate of $7.75 \times 10^{-8}$ kg/s, the particles in the first shadowgraph have a sauter diameter of 70 $\mu$m dropping to 60 $\mu$m and 40 $\mu$m at 6 and 12 diameters away respectively. The particle intensity ranges from 5,000 to 14,000 to 35,000 for the same values. At the nozzle exit the water has only just been exposed to the gas stream; there has been insufficient time for the momentum of the gas stream to be transferred to the liquid flow and for the interfacial instabilities to build up. With increasing distance from the nozzle, the particles have been exposed to more of the co-flowing gas stream, resulting in the growth of the instabilities and more particles being stripped from the initial filament. There is more energy available to break up the droplets further resulting in a decrease in particle size. Once the initial filament has broken up one might expect the droplet intensity to decrease with increasing distance from the nozzle, since the particles will be spread out to greater radial distances. Countering this effect is the breakup of droplets into smaller droplets, and thus the increase in the droplet intensity. For low to medium exit distances, this is the predominant driving mechanism behind droplet intensity.

Outside of this mist spray core, where most of the droplets are present, there exists a fine mist flow where a much smaller concentration of mist exists. The particle sizes are smaller in this region and the velocities lower and more turbulent, possibly due to entrainment effects of the flow. In a similar manner to that of air only flow, air outside a potential core is entrained into the ambient fluid. Since the mist particles, especially the smaller particles, predominantly follow the air flow, particles outside of this central region where flow velocity is maintained behave in a much more turbulent manner. This can be seen from the lower numbers of particles in the outer regions, figure 9.

The data provided on droplet sizes and droplet intensities shows the variation with radial and vertical distance (figures 9a & b). This is evident in the heat transfer profiles (figures 2-6) which can be partly attributed to this variation. The flow characteristics also impact on the heat transfer, although this impact is less clear.
4. Conclusions

A study has been carried out to characterise the convective heat transfer of an impinging air/water mist jet. Heat transfer profiles follow a typical bell shaped pattern with increasing water flow rates contributing to higher heat transfer coefficients, although the level of enhancement achieved at a very low water flow rate is still very significant. The results suggest that the frequency of atomization as well as the droplet intensity contributes to the heat transfer performance. The effect of droplet size is a little less clear and further investigation is needed.

Data show that the droplet formation follows typical shear driven atomization; droplet diameters decrease with increasing air and decreasing water flow rates, while droplet number density increases with increasing air and decreasing water flow rates. Droplet sizes decrease with increasing vertical and radial distance from the nozzle exit, although this is due to different underlying mechanisms. The droplet intensity increases with increasing vertical distance from the nozzle while falling with increasing radial distance from the geometric centre.

Further analysis of the flow field, both at nozzle exit and at impingement, is important, while further measurement of fluctuating heat transfer parameters is expected to provide additional information on the characteristics of the atomizing nozzle.

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5. References

[1] O'Donovan T, Murray D and Torrance A 2006 Proc. of the Inst. Mech. Eng. Part C: J. Mech. Eng. Sci. 220 837-45
[2] Lee S, Yang Z and Hsyua Y 1994 J. Heat Transfer 116 167-72
[3] Lee S, Park J, Lee P and Kim M 2005 Heat Transfer Engineering 26 24-31
[4] Li X, Gaddis J L and Wang T 2001 J. Heat Transfer 123 1086-92
[5] Ferrand V, Bazile R, Borée J and Charnay G 2003 Int. J. Multiphase Flow 29 195-217
[6] Sozbir N, Chang Y and Yao S 2003 J. Heat Transfer 125 70
[7] Chen R H, Chow L C and Navedo J E 2004 Int. J. Heat and Mass Transfer 47 5095-9
[8] Lyons O F P, Persoons T and Murray D B 2010 14th Int. Heat Transfer Conf. (Washington DC USA)
[9] Lyons O, Persoons T and Murray D 2010 23rd Annual Conf. Liquid Atomization and Spray Systems (Brno Czech Republic)
[10] Trávníček Z and Tesař V 2008 5th European Thermal-Sciences Conf. (The Netherlands) p 186-7
[11] Lyons O F P, Murray D B, Byrne G and Persoons T 2009 Proc. IMECE2009-11244 (Lake Buena Vista Florida USA: ASME)
[12] Ko N and Chan W 1978 J. Fluid Mechanics 84 641-56
[13] Chan W and Ko N 1978 J. Fluid Mechanics 89 515-33
[14] Lefebvre A H 1989 Atomization and Sprays (Hemisphere Pub. Corp.)
[15] Gorokhovski M and Herrmann M 2008 Annu. Rev. Fluid Mechanics 40 343-66
[16] Chigier N 1991 Proc. ILASS 5th ICLASS-91, (Gaithersburg) p 49-64
[17] Berg T, Deppe J, Michaelis D, Voges H and Wissel S 2006 ICLASS’06 (Kyoto Japan)
[18] Ichimiya K 2003 Int. J. Heat and Mass Transfer 39 545-551
[19] O’Donovan T S and Murray D B 2007 Int. J. Heat and Mass Transfer 50 3291-301