Mist/Steam Cooling for Advanced Turbine Systems

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ABSTRACT

Internal cooling of turbine blades by steam is of current interest to several manufacturers for land-based designs. Enhancements in steam cooling can provide a less expensive cooling system. Previously, this project has demonstrated that a mild loading of mist, 1 to 5% by mass in the steam, allows enhanced heat transfer in straight tubular and bent tubular configurations of an attractive magnitude, being of the order of 100% in many of the observations. Conceptually the leading edge of each airfoil will experience the highest heating and thus will require the greatest cooling. The relatively more effective jet impingement will be given opportunity by designers to perform the cooling in this vicinity. This report summaries a four-year study and focuses on the jet impingement of mist onto a heated plate.

Steam is directed onto a segmented heater, instrumented and thermally insulated on the backside and heated electrically. Mist, provided by high pressure jets and in the size range of 10 µm in diameter, is administered in a mixer. The mixture generally provides a lower temperature of the heater at every location of the test section. A simply defined heat transfer coefficient is used to provide a value with mist and a value without mist; the ratio being the enhancement.

The enhancement was found to vary up to 200% with the addition of 2% mist at typical conditions. The unscaled mist effect is nearly independent of heat flux so that the enhancement is more at lower heat flux. The enhancement is dependent on but not quite proportional to the mist content. And the enhancement is mildly improved at higher Reynolds numbers.

A model for the overall behavior has been constructed of three superposed parts. The model combines the effect of steam alone plus the effect of having a mist near the plate which quenches the boundary layer and improves the heat transfer. To these effects is added the heat transfer by direct contact with the impacting particles. A fluids computational model is used to predict the velocity of impact and the density of particle impact on the target surface. A model of the wall to droplet heat transfer is constructed which completes this third component, direct heat flow. The possible enhancement by modification of the turbulence and the flow field is considered to be minimal based on other studies and the level of mist loading employed.

INTRODUCTION

The use of mist cooling technology is projected in concert with the possibility of using steam cooling and a bottoming cycle to increase station efficiency in advanced applications. The function of mist is to enhance the heat transfer beyond that expected for pure steam to serve as an alternative to other geometric schemes of increasing heat transfer that would be more expensive. The enhancement would need to be applicable to a range of geometric and thermal situations. Therefore four geometries have been chosen to provide evidence of a general improvement of heat transfer using mist. These geometries are a straight tube, a bent (180°) tube, a jet impinging on a straight wall, and a jet impinging on a concave wall. This report summaries results for a four-year study and focuses on the impingement onto the straight wall. The concave wall is yet to be operated.

EXPERIMENTAL

The conditions expected for internal passage cooling are pressures of 30 to 40 atmospheres, with superheated wall conditions above 2000°R. Heat fluxes in the range of 1MW/m^2 are expected in passages perhaps 0.25-inch characteristic diameter. The concentrations and size of mist to be used are not known in advance. It is anticipated, that because of the need in rotating blades to survive high accelerations relative to gravity, the size of droplet will need to be of the order of 10 µm. This has been anticipated in the design of the tests herein. Whereas the prototype may have Reynolds numbers of 300 000, the model used herein has a value of 10 000 to 30 000 but with velocities at or higher than those expected in the prototype.

A test facility has been set up as shown schematically in Figure 1. There are two filtered steam supplies; one at low pressure (about 20 psia) and one at high pressure (about 120 psia). Filtered water (not deionized) has been atomized by several methods; herein we used Mee atomizers furnished by the manufacturer for the bulk of testing conducted. Use of high pressure steam to atomize liquid also proved useful, but was not used for most of the testing. The atomizer is located in a chamber though which metered low pressure steam flows such that the mist is entrained by the steam and delivered to the discharge tube. Several mixers and modifications of mixer were used. The tube and bent tube used a small (4-inch diameter) mixer with horizontal exit flow. The jet impingement study used a 6-inch mixer section with vertically down exit flow. The reasons for these variations primarily depended on the test sections which were horizontal for the tubes and vertically down for the jets. Flow from the test section was routed immediately to a condenser. Traps were placed strategically to remove condensate in a way to prevent liquid carryover into the test section and to allow collection for mass inventory.

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Figure 1 Test Facility Schematic

The droplet analysis for the flows was conducted by use of Aerometrics phase Doppler particle analyzer, PDPA. The 488-nm lines from a argon laser were isolated by a TSI Color Burst separator and fed to the focusing lens through fiber optic cable. The detector with a 100µm slit was oriented at 30 degrees to the plane containing the beam pair. Optical admission to the tube was through small glass ports for the tube flows and large glass side plates for the impinging jet flows. The tube ports were located immediately downstream of the heated section vertically in the middle of the tube. The sensitivity of the PDPA to the small end of the size distribution was experienced in that at low power the small droplets are not detected. As power is increased, a greater fraction of the small droplets is detected, until further power increases do not cause greater detection. All the runs made herein are run with full detection.

Also, with the jet impingement configuration, large variations in the signal data rate are commonly experienced. For the most part the size distribution is unaffected, but the sample rate changes substantially. In this configuration, some of the mist is dispersed onto the glass sidewalls. There, either beam of the laser transmitter might be interrupted for the lifetime of the droplet. No useful signal can occur when there is a droplet in the path. Normally the signal-getting is done for 10 000 droplets and this requires about one second. Apparently the probability of having one second without a droplet in the beam path is small. Therefore the 10 000-droplet acquisition time is interrupted by one, two, or more lifetimes of droplets. It is not uncommon to have sequential data rates varying from 200 to 12 000. The procedure followed has been to heat the glass with an auxiliary heater to speed the evaporation and to make many observations. The highest data rate should be that interrupted by the droplets the least. Even higher data rates might be observable if more readings are taken. The data rate concentration agrees reasonably with the mass inventory data by the above method.

Figure 2 shows a size distribution for the atomizer operating into still air and the size distribution typical of testing. Into air, with 1000 psi supply pressure, the size distribution shows a most probable size of 5µm and a number average size of 10µm. When the nozzle is confined and the flow delivered through a tube, the larger droplets are selectively eliminated by impact with the wall. This results in smaller average and most probable size. Other details of the mist are included in Wang et al.(1999a).

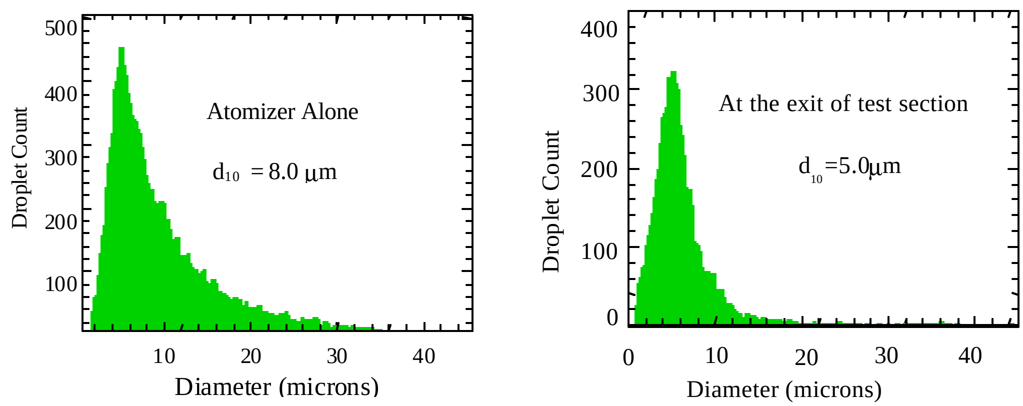

Figure 2 Spray Droplet Size Distribution

TEST SUMMARY FOR TUBE FLOWS

The first test sections were composed of 2-cm diameter tubes, electrically heated by DC current. The tubes were instrumented with thermocouples placed along the length and on both the bottom of the tube, sides, and top. The tube wall was heated a variable amount not over 350°C (660°F) and the steam flowed at velocities of up to 30 m/s. The Reynolds number was therefore between 10 000 and 35 000 and mist up to about 5% of the steam flow was added. Control of the amount of mist was done by adding more nozzles. Documentation of the mist content was by catch and weigh of liquid, direct measurement of steam and liquid flow rates, and by particle count. Details of the early test results are reported in Wang et al.(1999a, 1999b). Herein the results are summarized to emphasize major facts and important trends.

The wall temperature data for steam alone allows data reduction for single-phase heat transfer. Data from the test articles agree closely with conventional estimates for the expected heat transfer rates. When mist is added, the wall temperature is lowered; this means the heat transfer is augmented. This is true at all points on the tube, but less at the downstream portions, and less on the top than on the bottom. It is believed that the unheated inlet tubing accumulated a thin film of liquid which entered the test section for virtually all tube testing. The large enhancements associated with liquid film presence are not sustainable and will not be anticipated for any application. So, the following will focus on the interpretation of the flows far enough downstream for the effect of liquid film to be overcome.

Far downstream, the temperatures of the top and bottom of the tube are the same or virtually so. The enhancement depends primarily on the amount of mist in the flow and on the temperature of the wall. Figure 3 shows a plot of the cooling flux due to mist versus the liquid mass flow for the far downstream region. The total cooling experienced is reduced by the one-phase cooling to arrive at the cooling flux of mist. Here the one-phase cooling is the product of the single phase heat transfer coefficient and the superheat of the wall. In the figure, if the Reynolds number were shown, it would be clearly not systematically related to the result. What is shown is that the cooler walls have more effect of mist cooling than the higher wall temperatures allow. But, even for the higher temperature wall, the addition of more mist increases the cooling. This latter result, extended to superheat of 340°C (640°F), suggests that there can be control of cooling by adding mist. This is an important conclusion. It is not known whether there is a limit to this trend, either in temperature or mist concentration.

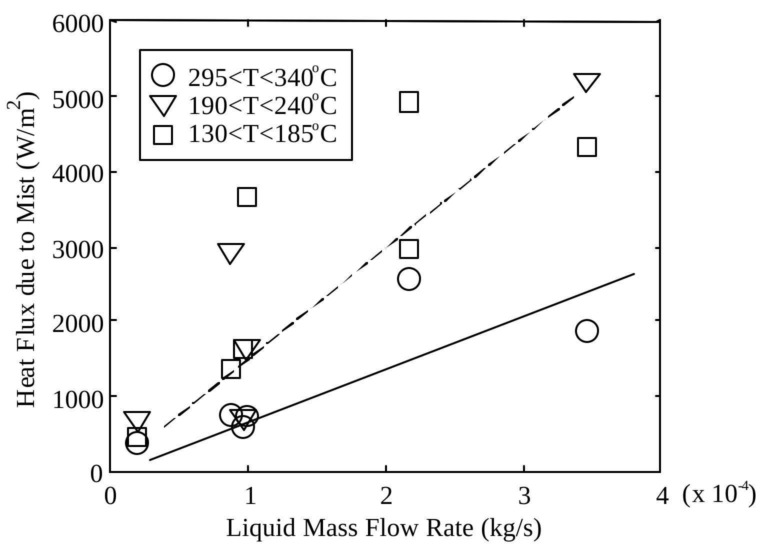

Figure 3 Cooling in Downstream Region

A potential concern also arises from Figure 3. There will be strong asymmetries in cooling demand within every anticipated channel of a cooled blade. The mist effect is stronger when the demand (indicated by wall temperature) is weaker. To determine the importance of this trend will require some modeling of the conductivity of the blade and the geometry of the cooling passages in addition to the cooling demands. It is a type of conjugate heat transfer problem.

Figure 4 shows data from the testing of the 180° bent tube. The data are for the two higher heating rates; the lower heating rate had much higher enhancements and is felt to be less appropriate for the application. The focus of the discussion is to address the fact that the inner wall has significant and even higher enhancements than the outer wall all through the bend region. This surely is a result of the paired vortical secondary flow which occurs in every bend. But with a centripetal acceleration of hundreds of g's the droplets are able to contact the wall for a significant cooling effect. It appears that the achievement of 1.5 enhancement is reasonable with the use of a mist in the range of 1 to 2%.

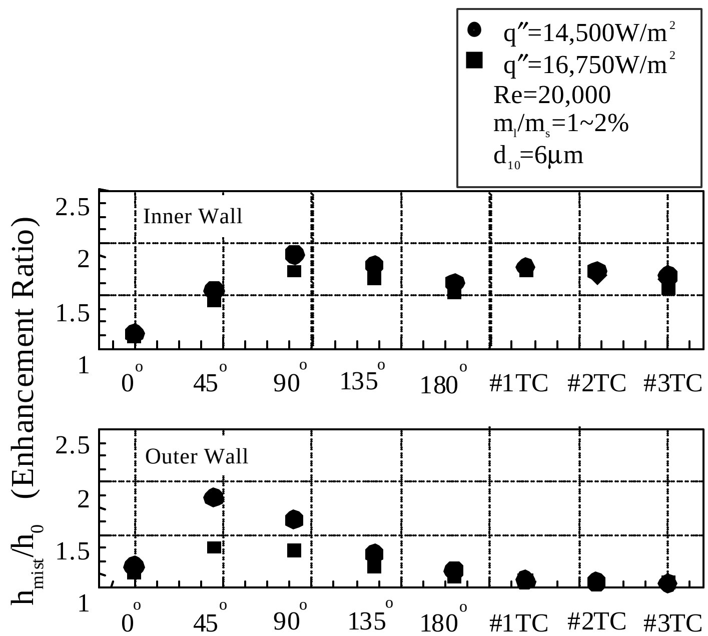

Figure 4 Cooling in a 180° Bent Tube

IMPINGEMENT TESTING

Figure 5 shows a schematic diagram of the test section used for jet impingement cooling. A summary of the results will be given for the slot jet and for the four discrete hole injection plate. The three-row jet plate results are in progress and the data have not been reduced at the time of this writing. The jet enters from above and impacts on the horizontal heated surface. The surface is heated at constant heat flux except for small variations due to temperature and resistance non-uniformity. Since this process was open to view, it is possible to report that the surface is always of an absolutely dry appearance. Rivulets of condensate from the entrance chamber do form on the side walls but do not reach the heated surface.

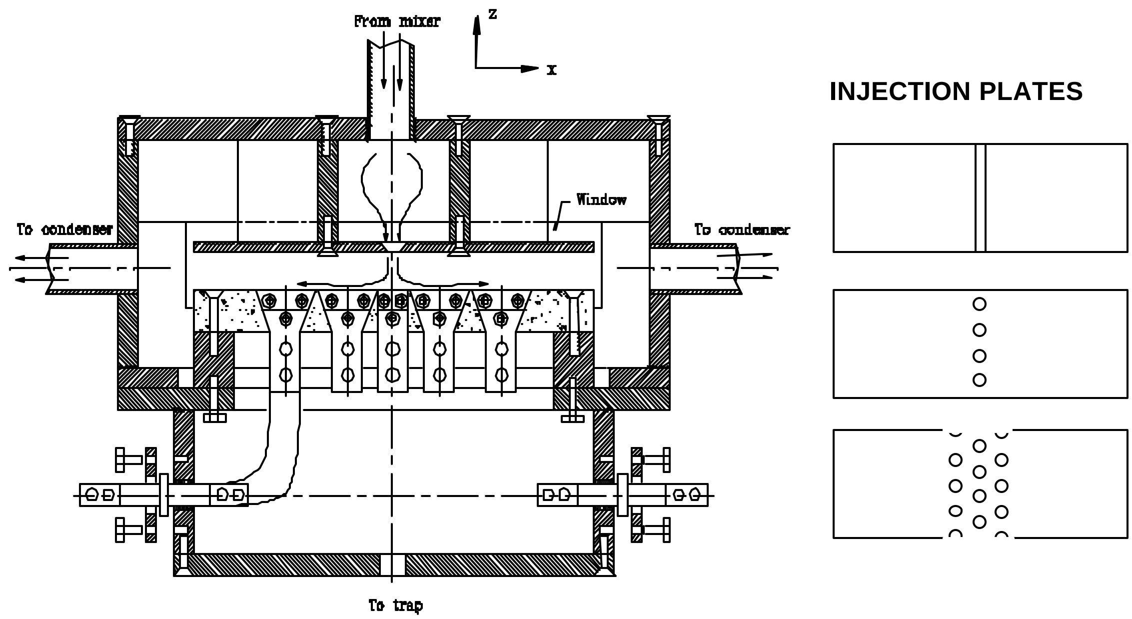

Figure 5 Jet Impingement Test Section

Figure 6 shows the typical information from testing for the slot jet case. The typical conditions are indicated in panel (c) and are a moderate heat flux and low mist amount at an intermediate steam velocity. Panel (a) indicates the temperatures at the thermocouple positions as a function of the distance from the impact line of the jet. It is shown clearly that the mist depresses the temperature and substantially so. The ratio of the heat transfer flux divided by the wall superheat (temperature minus saturation temperature) is the heat transfer coefficient, shown in panel (b). The heat transfer coefficient declines away from the stagnation point for the pure steam flow and actually increases before declining for the mist flow. The ratio of these coefficients is the enhancement, shown in panel (c). The enhancement is over 150% for 1.5% mist concentration.

The mist cooling is higher in this typical case at a point one slot width removed from the slot impingement center. This could be due to asymmetry, but a check for symmetry in velocity and temperature does not corroborate such a suspicion. The off-center peak cooling is noted for about half the cases, and is characteristic of the lower heat flux conditions and, slightly, to higher mist content. Modeling of the particle trajectory reveals that it is expected for particles to de-focus from the centerline. Also the particles larger than 6μm are expected to have more than one impact with the wall; i.e. they bounce. Both of these trends tend to have more mist impact away from the jet center, and consequently greater heat transfer.

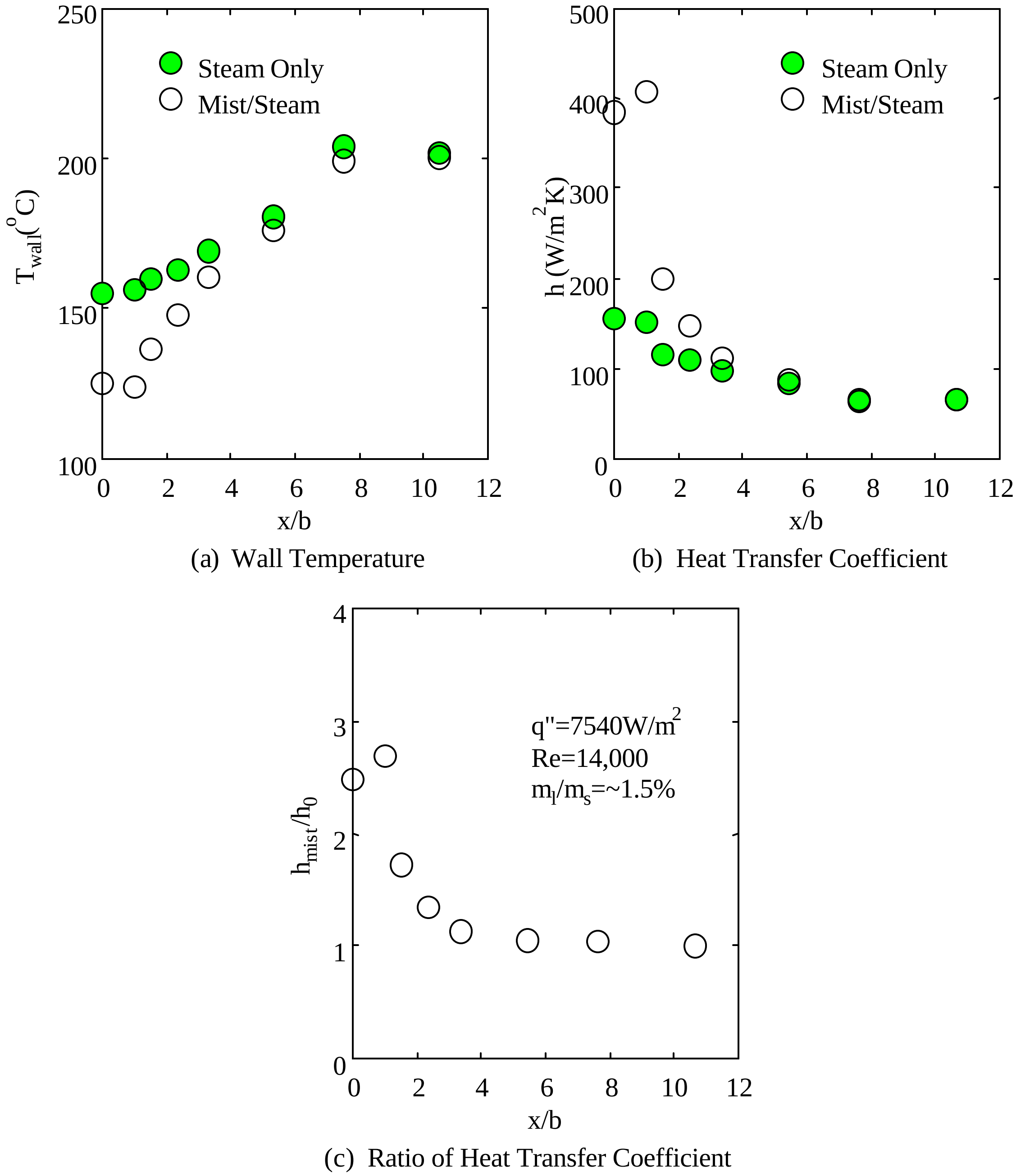

Figure 6 Results for a Typical Slot Jet Run

As the values of other parameters are changed, the heat flux is affected in systematic ways. Figure 7 shows the trend of enhancement with increasing wall temperature to be a general decline as was noted in the earlier testing of tubes. The data are shown at a relatively fixed value of mist concentration and steam flow (Reynolds number) and are representative of the trends at other combinations. The independent effect of Reynolds number is best estimated by treating the enhancement data to provide an index independent of mist content. The value of heat transfer coefficient h at any point, diminished by the single phase value, h_0, gives the amount of heat transfer due to mist per degree of wall superheat. When this mist effect is divided by the mass rate of mist, an index is produced which is basically that of the ordinate of Figure 8. This index can be considered a non-dimensional specific effect of mist. It becomes one when there is no effect so it is a type of enhancement. According to Figure 8, the effect of Reynolds number is positive on the mist enhancement index. The effect is fairly modest, but is positive. This fact yields some improved optimism on the prospect of moving to an order of magnitude greater Reynolds numbers than those of the current testing.

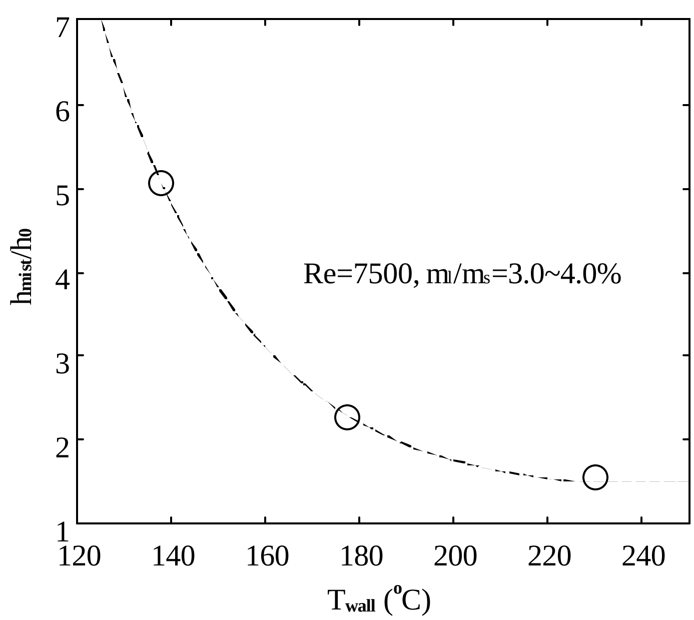

Figure 7 Effect of Wall Temperature

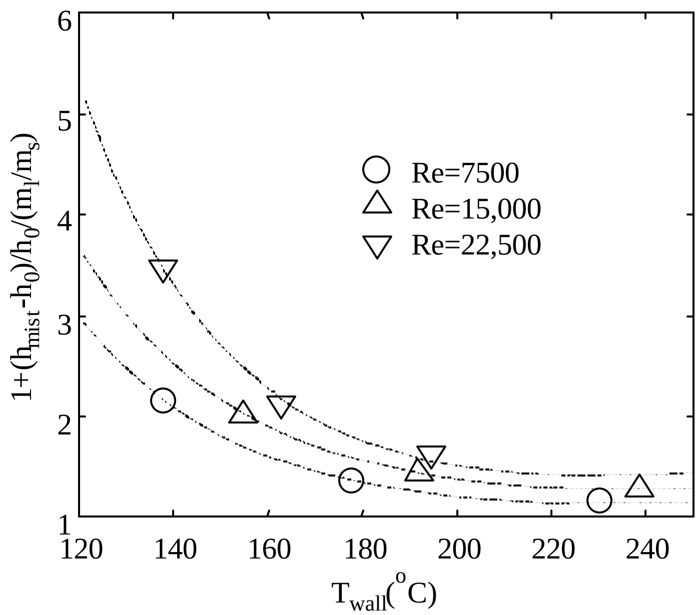

Figure 8 Effect of Reynolds Number

Similar results are obtained for the four discrete jet case. Figure 9 has wall temperature, heat transfer coefficient, and enhancement ratios for the single row test results at a typical case. There are two rows of thermocouples used; one is in line with one of the impingement holes, the other is midway between the holes. The patterns of wall temperature reduction and heat transfer enhancement are similar for both rows of thermocouples, but near the jet the row between holes shows better cooling. At two diameters downstream the difference has vanished and the heat transfer becomes essentially two-dimensional in appearance. This geometry, as in the slot jet, yields stagnation region enhancement of 150% or better with 2% mist addition. The effect wanes to a negligible amount beyond 4 slot widths.

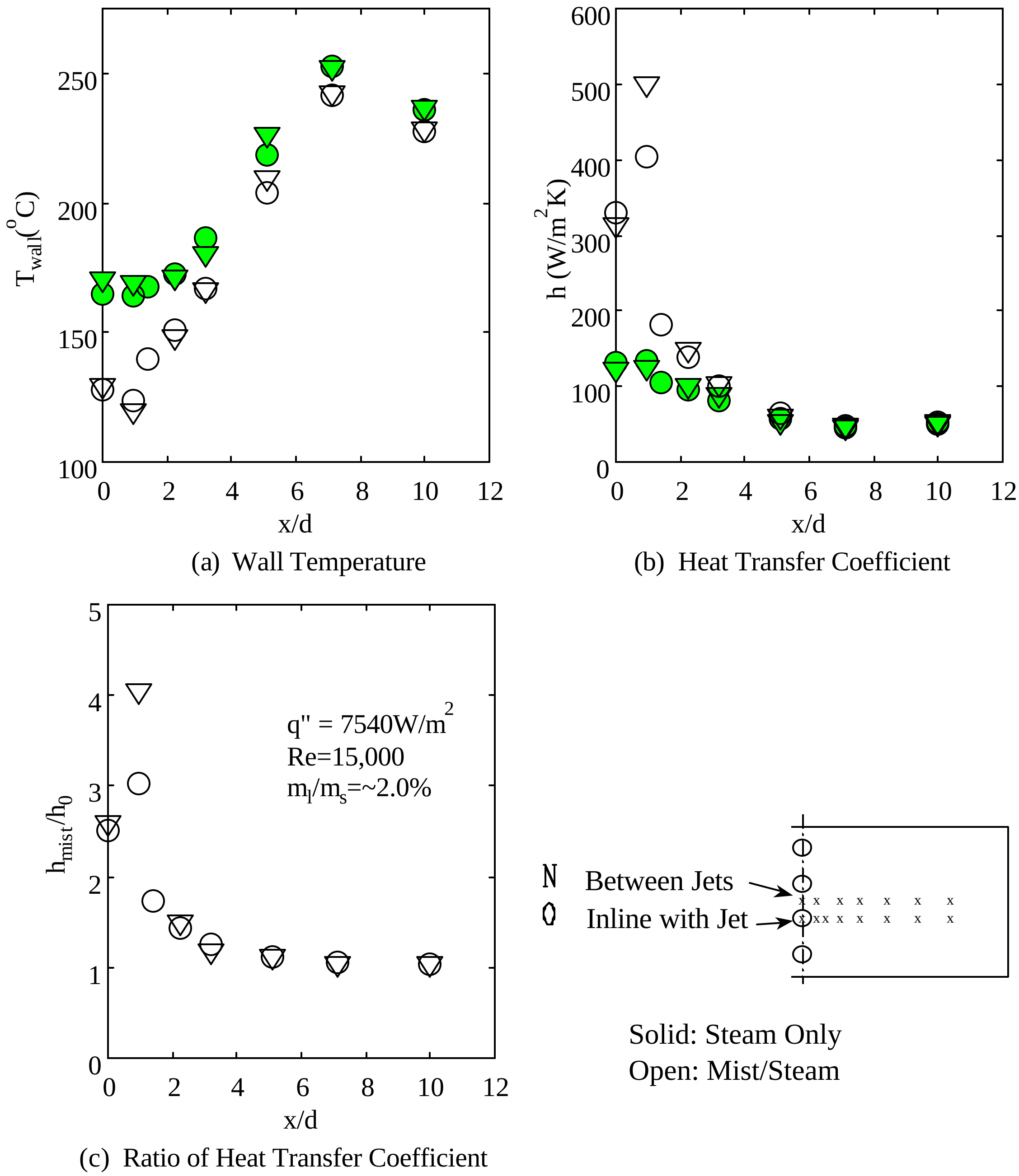

Figure 9 Results for a Typical Discrete Jet Run

MECHANISMS AND MODELS

A considerable time has been devoted to interpretation and modeling of the results obtained in testing. In particular a total heat transfer model is sought which would be the sum effect of a single phase convective model, modified by a turbulent and flow structure effect; a model of the quenching in the boundary layer due to the presence of mist; and the effect of direct impact of droplets on the wall. From studies of solid suspensions, it has been found that 80% solid particles by mass can cause heat transfer increases by 170%. If it is presumed that all of this effect is due to modification of the flow structure and a linear ratio of the mass effect is used, 2% mass addition would only produce about a 4% change in heat transfer. For this reason, the flow modification has been dismissed as unimportant. The effect of quenching in the boundary layer has been approached analytically using the particles as a heat sink in the layer. A simple model has been developed (Li, 1999) to calculate the effect of this quenching of the boundary layer. The model predicts that the quenching effect increases the heat flux in rough proportion to the mist concentration. The effect is relatively weak for the conditions of the present testing, representing an amount usually less than 5% of the total heat transfer. For the conditions of the typical case of Figure 5, the quenching effect adds about 2% of the total heat transfer.

With the effect of flow modification and quenching of the boundary layer determined to be of minor importance, the enhancement must be primarily due to direct contact heat transfer of droplets. Using the commercial code, FLUENT, we have modeled the flow from the jet entering the impingement zone. Then, in an interactive mode of calculation, droplets are allowed to enter with the steam flow. Primarily using the option of particle elastic rebound the trajectory of particles is computed. The principal force acting on the particle is the drag from relative motion of vapor and droplet as the vapor accelerates. FLUENT also computes the thermophoresis due to temperature gradients and the molecular-motion induced momentum interchange, but this is very small. Not included in the analysis are the lift due to asymmetric evaporation rate or any of the Saffman (slip-shear), Rubinow-Keller (slip-spin), or wall proximity forces. Also not included are the dispersive tendencies of particles due to turbulent flow, although FLUENT allows for random trajectories to be computed in a stochastic simulation of particle motion.

The near stagnation region including the first impact of droplets is expected to be computed with reasonable fidelity. The estimates of particle flux on the surface are believed to be accurate using FLUENT. What is believed to be faulty with the trajectory analysis is the downstream region. Here the droplets are computed to be concentrated near the wall in spite of the fact that the droplets do evaporate. Surveys by the PDPA instrument do not corroborate increased concentration near the heated target surface. Rather, the particle concentration is quite uniform except near the wall where evaporation is significant. So the predictions of FLUENT are not considered valid in the downstream region. But the region near impact is considered valid, including the bouncing of some droplets.

Using the flow analysis of FLUENT in the zone of jet impingement results in a density of particle impacts and a speed for each size. The density and size are input to a calculation of cooling effect per event. The overall experimental result insists that there is less heat transfer at higher wall temperature to the average droplet. If the contact time were constant, independent of temperature, the raising of wall temperature would result in increased heat flow, a trend denied by the data. Therefore the contact time must be larger at lower temperature. Purely fluid-mechanic analysis such as that of Fujimoto and Hatta (1996), are independent of wall temperature.

The hypothesis of Buyevich and Mankevich (1995, 1996) is that at low enough temperature all droplets will stick and evaporate completely, while at elevated temperature a vapor film restricts heat flow and the contact is brief. Use of the velocity criterion of Buyevich and Mankevich for our data and conditions has a large fraction of droplets becoming stuck and evaporating completely. The predicted heat fluxes are much more than the experimental data. Mikic and Rosenhow (1969) developed a criterion for the initiation of a vapor bubble in nucleate boiling. The criterion sets the energy superheat in the region of the to-be-formed bubble nucleus to exceed the energy needed to form the bubble surface energy. This criterion is expressed as a waiting time, dependent on wall temperature, after which the formation of the bubble is spontaneous. The development of Mikic and Roshenhow requires knowledge of a bubble radius for evaluation. Clearly the bubbles must be smaller than the droplets of our study, but no other size can be found to scale them. The times for this range of bubble size are both reasonable per the data and do increase at lower temperature.

We have preferred to express that our droplets will contact the wall long enough to produce a superheat sufficient to overcome a pressure estimated to restore a fraction of the dynamic pressure of impact and the pressure of surface distortion. This contact time criterion is

$$\alpha t/d^2 = dT/dP\ (\sigma/d)\ (1 + We/8)\ .$$

When a droplet assumes a spherical lens shape with 60° contact on a heated surface for the time given by the above equation, the contact heat transfer may be calculated by ordinary conduction models. Given the droplet impact density and velocity by size, the net heat transfer by impact is calculated to agree within 5% of the measured amounts. This agreement means that the droplet heat transfer model is valid within about 10% of its true value. Because the Weber number range is relatively small and the temperature range is also limited, the validity of the model can only be assessed reasonable. Its trends cannot be established by the operation of a single experiment.

CONCLUSIONS

Mist added to steam flow is generally observed to be capable of enhanced heat transfer. For plane surfaces without any liquid film, the enhancement depends on the turbulent dispersion to give particle contacts leading to increased heat transfer. These contacts are proportional to droplet density and are still effective, though less so, at wall temperatures of 200°C above the saturation state. In bends, where secondary flows can provide increased particle impact on the wall, the enhancement is increased.

Mist added to steam flow in impingement situations provide extreme enhancement in the stagnation zone and waning enhancement away from it. For both slot and discrete jets, enhancements of 150% are achieved with 1.5% of mist near the stagnation point. The enhancement fades to a vanishing amount at 4 diameters from the stagnation point.

Mechanism analysis suggests that the effect of modifying the flow structure is small in producing enhancement. Also, the quenching of the boundary layer by the presence of mist also has a small effect. The primary effect is due to direct contact of particles with the wall. for the known geometry of slot jet impingement, is possible to predict the deposition density and velocity. A model is given for the time of contact of particles before being rejected from the wall. This time can be used with heat conduction in contact to calculate the heat carried by the direct impact of mist. The model opens the door to predictions.

Many more items are essential to advance a successful design. Prediction of particle flux to the wall by turbulent dispersion is needed. Development of a blender for mixing mist and carrier steam is needed. Confirmation of the effects of larger droplets, different pressures, and larger heat fluxes are needed to develop confidence in the trends. If computational models are to be utilized an effort will be required to properly predict the dispersion of droplets in a turbulent flow and to allow a comprehensive model of the impact heat transfer process within the computational umbrella.

ACHIEVEMENTS

During the four-year duration of this study, the following notable achievements have occurred.

a. Completed the straight tube experiments, two papers were presented at the ASME Turbo Expo' 99 (Guo et al, 1999a and 1999b). The final report Volume 1 was completed.

b. Completed the 180-degree tube bend experiments, one paper was presented at the National Heat Transfer Conference, (Guo et al., 1999c). The final report Volume 2 was completed.

c. Completed the slot jet impingement cooling, one paper was accepted for presentation at ASME Turbo 2000 (Li et al. 2000). A paper for National Heat Transfer Conference (2000) is in preparation. The final report volume 3 was completed.

d. Two doctoral students graduated from this research program.

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