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EXPERIMENTAL INVESTIGATION OF MIXED CONVECTION HEAT TRANSFER CAUSED BY FORCED-JETS IN LARGE ENCLOSURE

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ABSTRACT
This research investigates experimentally mixed convection and heat transfer augmentation by forced jets in a large enclosure, at conditions simulating those of actual passive containment cooling systems and scales approaching those of actual containment buildings or compartments. The experiment was designed to measure the key parameters governing the heat transfer augmentation by forced jets and investigate the effects of geometric factors, including the jet diameter, jet injection orientation, interior structures, and enclosure aspect ratio. The tests cover a variety of injection modes leading to flow configurations of interest that contribute to reveal the nature of mixing and stratification phenomena in the containment under accident conditions of interest. By nondimensionalizing the governing equations, the heat transfer of mixed convection can be predicted to be controlled by jet Archimedes number and geometric factors. Using a combining rule for mixed convection and appropriate forced and natural convection models, the correlations of heat transfer augmentation by forced jets are developed and then tested by experimental data. The effects of jet diameter, injection orientation, interior structures, and enclosure aspect ratio on heat transfer augmentation are illustrated with analysis of experimental results.

INTRODUCTION
Passive containment cooling systems (PCCS) provide a new safety design for the nuclear industry that provides the safety-related ultimate heat sink for a new generation of inherently safe reactors such as Westinghouse's AP600 and AP1000 and General Electric's SBWR and ESBWR. It uses only natural forces, such as gravity, natural circulation, and a small number of automatic valves to make the system work. The steel containment vessel itself provides the heat transfer surface that removes heat from inside the containment and transfers it to the atmosphere. Heat is removed continuously from the containment vessel by a natural circulation flow of air. However, since the passive safety system represents a totally new concept in safety reliance, a more extensive testing program has to be undertaken with these reactors. Passive Containment Cooling System Test Program is one of the key aspects of the test and analysis program to obtain integral-systems test data on the thermal-hydraulic performance of this system to support code validation.

Mixed convection flows have received considerable attention since the late 1970s. Catton and Edwards [2] investigated effects of side walls on natural convection between horizontal plates, and found that, for large Rayleigh numbers extending into the turbulent regime, the size and thermal conductivity of the horizontally confining surfaces did not affect the mean heat transfer coefficient. Churchill studied combined free and forced convection around immersed bodies [3] and in channels [4]. Osborne and Incropera [5] studied laminar mixed convection heat transfer for flow between horizontal parallel plates with asymmetric heating, and found significant asymmetries might be associated with convection heat transfer at top and bottom surfaces. They also found [6] although buoyancy effects could significantly enhance heat transfer for laminar forced convection flows, enhancement was typically negligible if the forced flow was turbulent. Kuhn et al. [7] studied gas mixing process and heat transfer augmentation by a forced jet in a large cylindrical enclosure and gave a correlation as a function of Archimedes number, fluid property factor, and geometric factor, to evaluate the mixed convection heat transfer augmentation inside the enclosure heated from below.

Buoyant jets have been studied extensively, as summarized by Gebhart et al. [8]. Tenner and Gebhart [9] studied upward low-momentum laminar buoyant axisymmetric jets of fresh water into linearly stratified saltwater. The buoyant jet induced the flow of a toroidal cell around itself, drawn up by the viscous shear of the jet. Photographs of the developing region of a buoyant jet or plume were presented by List [10], which showed large coherent structures near the source. Quinn [11] reported the results of detailed mean flow and turbulence

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measurements made with hot-wire anemometry in the flowfield of a turbulent free jet of air issuing from a sharp-edged rectangular slot into still air surroundings. Chen and Rodi [12] reviewed extensive experimental investigations of turbulent buoyant plumes.

Impingement cooling is an effective way to generate a high cooling rate in many engineering applications. In steel or glass industry impinging jets are used to cool down the products after rolling. In gas turbine engines impinging jets are applied to the cooling of turbine blades or vanes. In laser or plasma cutting processes, the application of jet impingement cooling can reduce thermal deformation of products. Besides the above applications, impinging jets are also adopted to enhance industrial drying processes or electronic cooling [13]. Jambunathan et al. [14] did a detailed survey on the impingement cooling of a single air jet. Bouchez and Goldstein [15] investigated the impingement cooling of a circular jet with/without a cross flow. An extensive review of available convection coefficient data for impinging gas jets has been performed by Martin [16].

Stratification is a state characterized by strata, or horizontal layers, of different density. Stratification is stable when the lower layers are increasingly dense due to composition and/or temperature. Static stratification occurs if the upper horizontal boundary of the domain is maintained at a higher temperature than lower boundary. Stratification also occurs if the concentration of heavy components is low in the mixture in the upper portion of the domain. Heat transfer is predominantly governed by conduction, while mass transfer is driven by diffusion. The formed fluid layers are stable and communicate only with the neighboring upper and lower layers. The resulting heat and mass transfer rates are low.

When strongly stratified, an enclosure's ambient temperature and concentration distributions can be considered one-dimensional, with negligible horizontal gradients except in narrow regions occupied by jets. Jaluria [17], Goldman and Jaluria [18], and Jaluria and Cooper [19] have exploited the one-dimensional temperature distribution to develop zone-mixing models for enclosure fires. Under zone mixing, the ambient enclosure temperature distribution is divided by a sharp horizontal interface. Mixing between the regions is then generated by buoyant plumes from fires and by wall jets, which form on cold structure surfaces. From plots of the temperature variation along the vertical direction, the height of the hot-cold interface can be quantitatively determined. Following Steckler et al. [21], the height of the interface is estimated from the temperature profile data as the position of rapid temperature change between the lower and the upper portions of the cavity. These temperature profiles illustrate how diffusion and mixing preclude a sharp designation of the hot-cold interface. Consequently, the interface height could only be determined to within ±10~25% accuracy.

Peterson et al. [20] studied the transient thermal stratification in pools with shallow buoyant jets and presented a detailed experimental and numerical investigation of such two-dimensional transient stratification. Fox et al. [22] and Smith et al. [23] found that experimental results for transient stratification of boiling water reactor (BWR) pressure suppression pools could also be predicted using numerical solutions of one-dimensional differential equations describing the effect of buoyant jets on the vertical temperature distribution. Peterson et al. [24] showed that convenient scaling parameters for design of scaled experiments for reactor containment simulation can be derived for stratified conditions. Peterson and Gamble [25] presented a more advanced scaling method that could provide the basis for the design of scaled experiments for studying jet-induced heat and mass transfer in large enclosures. Peterson [26] showed that large enclosures mixed by buoyant plumes and wall jets could normally be expected to stratified, and provided a criteria for assessing when the momentum injected by forced jets would break down stratification in large enclosures.

As mentioned above, mixed convective transport is of interest and importance in a wide variety of engineering applications. However, this area has received less attention than its practical importance warrants. Much of the effort in the past has been directed at delineating the transport regimes, mostly to determine when buoyancy effects are negligible in a forced flow circumstance and when the externally imposed flow may be neglected in a natural convection process. Similarly, the mixed-convection in large rectangular enclosure has not been investigated at great length. Limited work has been performed on natural-convection augmentation by forced jets. Few experimental data have been obtained on mixing and stratification phenomena inside large three-dimensional enclosure agitated by break-jet flows. Much of the research has concerned laminar flow in simple configurations and geometries. Many experiment geometries are simplified into parallel-plate channels or rectangular cavities and the thermal boundary conditions are set to be symmetrical. The extensive additional mixing experiments in large containment enclosure are thus needed to improve key scaling, experimental, and modeling tools for predicting mixing and transport in passive containments and confinement structures. There is a need for large-scale and multidimensional tests, supporting the development of a new generation of computer codes for safety analysis [27]. Indeed, it was noted [28] that, "in relation to scaling effects, the simulation of real-scale plant behavior with models and system codes validated on experimental data from small or reduced-scale test facilities may suffer from scaling distortions whose effects have to be assessed; model validation based on large scale tests will become more important in this respect."

This experimental research studies mixed-convection and heat transfer augmentation by forced jets in various directions inside a large enclosure with a vertical cooling surface. The experiment is designed to measure the key parameters governing the heat transfer augmentation by forced jets, and to investigate the effects of geometric factors, including the jet diameter, jet injection orientation, and enclosure aspect ratio. Both scaling and modeling of stratified mixing in large enclosures require detailed and accurate empirical models for wall and free jets. The research effort provides experimental results to support the development of a new, computationally efficient model for mixing under the stratified conditions that characterize large volumes in passive systems.

NOMENCLATURE

Ar_j jet Archimedes number
b width parameter
C constant
d jet diameter

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D_t	diameter of the block tube
Gr_L	enclosure Grashof number
H	height of enclosure
L	enclosure characteristic length
M	momentum flux
Nu	Nusselt number
Q	volume flow rate
r	radial coordinate
Re	Reynolds number
u	x-velocity
U	velocity
z	vertical coordinate

Greek symbols

α_T	Taylor's jet entrainment constant
μ	dynamic viscosity
ρ	density

Subscripts

0	entrance value
bj	free buoyant jet
C	centerline
D	drag
j	jet
L	using enclosure characteristic length
m	mixed convection
n	natural convection
w	wall

EXPERIMENTAL FACILITIES AND PROCEDURES

Design of Experiment

The experiment was designed to study the combined natural and forced convection heat transfer in a large rectangular enclosure mixed by injected jets. It can be used to quantify the augmentation of the heat removal rate under natural convection, generated by forced convection during blow down. The tests were designed to measure the key parameters governing the heat transfer augmentation by forced jets, and to investigate the effects of geometric factors, including the jet diameter, injection orientation, jet location, and enclosure aspect ratio. A schematic diagram of the system is shown in Figure 1. It shows an open loop composed of steam/air supplies, a heating system, a test section with a large insulated rectangular enclosure, a cooling system, and a set of measure system.

The large rectangular enclosure was constructed with the size of 2.29m×2.29m×2.29m. One of the walls was made of a 0.32-cm-thick copper plate. Cooling water was circulated through copper tubes, soldered to the backside of the copper plate. All the walls, ceiling, and floor were surrounded with insulation materials. The wall opposite the vertical cooling surface can be moved to change the enclosure aspect ratio. The jet tubing was horizontally inserted into the enclosure through this wall at different locations. The air/steam entered the enclosure from the jet and left through the exit on the bottom of the movable wall. Compressed air was heated by helical heaters before injected into the test section.

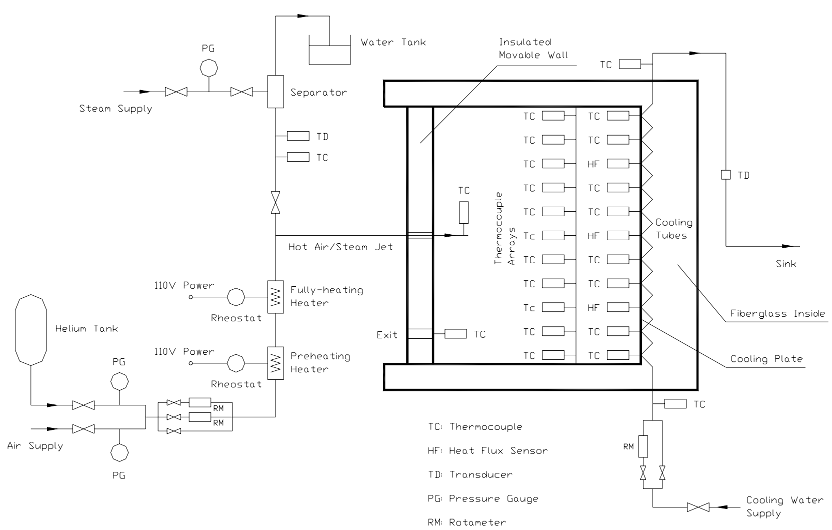

Figure 1. Schematic diagram of experimental system

Nine sheathed J-type thermocouples and three built-in K-type heat flux sensors were embedded in the copper plate to measure the temperatures and heat flux of the cooling surface, respectively. Four T-type thermocouples were mounted in the inlet of the cooling water and another four in the outlet, providing the temperature difference of the cooling water for the calculation of total heat loss from the cooling plate. A calibrated orifice with a differential pressure transducer was installed in the cooling water loop to measure the water flow rates. Heat removal rates by injected air were calculated from the temperature difference between the thermocouples mounted in the air inlet and outlet, and the air flow rates was measured by flow meters at different scales. The ambient temperatures inside the enclosure were measured with three vertical arrays of T-type thermocouples. Each array had eleven thermocouples from the bottom to the top and each horizontal line had three

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thermocouples put in both sides and middle point, giving the vertical temperature distribution as well as the mean temperature inside the enclosure. Figure 2 shows the full view of the experimental system.

Figure 2. Full view of the experimental system

Experimental Procedures

In preparing the equipment for a run, all valves were checked to ensure that only the valves on the short circuit were open. The computer was turned on first and all channels were checked and recorded as the initial conditions. Atmospheric pressure from a barometer and room temperature were recorded as well on a data sheet. Then open the cooling water valve and let water go through the short circuit rather than the rotameter. A loop leakage was detected prior to any test from which data were obtained.

To start the experiment, the PVC ball valve was opened slowly to let the water enter the flow meter. Meanwhile, the short circuit valve was closed slowly. Excessive stress on the meter could result from water hammer or surges when flow is suddenly began or stopped. Liquid surges are particularly damaging to flowmeters if the pipe is originally empty. To avoid damaging surges, the fluid lines were kept full (that is why the short circuit is employed) and valves were opened slowly. In the same way, the air valves were opened. When the desired air and water flow rates were reached, all channels were again recorded.

Heater power was turned on and checked regularly. The transformer was then set in low voltage output. All the heaters were heated up at low power. The heater outlet temperature was checked carefully to avoid overheating. When the temperatures inside the enclosure were constant or changed little for 15 minutes, the system was deemed at steady state. Data were recorded for each channel every ten seconds, for a one-minute period.

A second test could be run by increasing the heating power to another power level or adjusting the air or water flow rate. The air flow rate was increased first. Then the heating power was increased to obtain the same jet outlet temperature. After that, the cooling water flow rate was increased until a similar inside temperature distribution was reached. Running two tests consecutively avoided repetition of the long startup procedure. Due to the suprisingly long time consumption, seven was the maximum number of steady-state conditions that could be achieved in one workday.

Experiments were first performed to investigate the natural convective heat transfer in the enclosure, and the results provided the basis as a reference to further evaluate the heat transfer augmentation from combined natural and forced convection. For mixed convection heat transfer research, the geometric factors investigated in the present experiments included the use of three different enclosure aspect ratios, four different diameters of jets, four different injection orientations, and two different jet locations.

A final heating power check was recorded before system shutdown. The heater power was turned off first. The air valve was opened in the short circuit to the maximum flow to speed up the cooling of the heaters. The jet outlet temperature was checked and when it dropped to the normal temperature, all of the water and air valves were closed. The computer was shut down after the data were stored. An isothermal condition of the loop was achieved by the next morning, at which time a new experiment could be started.

EXPERIMENTAL RESULTS AND ANALYSIS

Effects of Injection Orientation and Jet Diameter on Heat Transfer Augmentation

To investigate how injection orientations and jet diameters affect heat transfer augmentation, air is injected into the enclosure with three directions: horizontal injection normal to the cooling plate, 45° upward injection, and vertical/up injection. Three different size jet heads are employed for each injection direction.

As illustrated in Figure 3, the experimental data can be well correlated by

$$\frac{\overline{Nu_m}}{\overline{Nu_n}} = \left(1 + C_1 Ar_j\right)^{1/3} \tag{1}$$

where the jet Archimedes number is defined as

$$Ar_j = \frac{\mathrm{Re}_j^{\,2}}{Gr_L} \tag{2}$$

Equation (1) is the correlation of forced-jet augmentation of natural convection heat transfer, which can be derived using a combining rule for mixed convection and appropriate forced and natural convection models [1]. It is a function of the jet Archimedes number and the coefficient C_1, which include the effects of fluid properties, jet mode, injection orientation, and enclosure aspect ratio.

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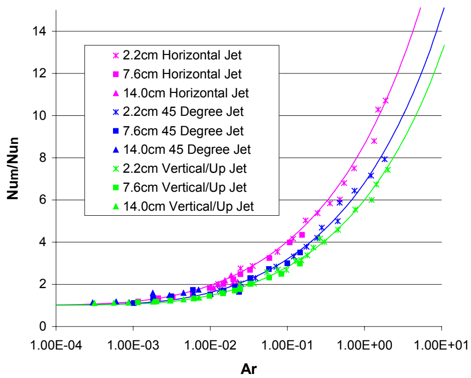

Figure 3. Effects of injection orientation and jet diameter

The heat transfer augmentation decreases from horizontal injection to vertical/up injection. This means compared to vertical/up injection, horizontal injection has better mixing effects, and moreover, since it is injected towards cooling plate, it can increase the wall jet velocity which will in turn raise heat transfer greatly. The data for all injection orientations asymptotically approach 1 at low Archimedes number, where heat transfer is dominated by natural convection. It is found (more clearly for vertical/up injection) that the experimental data, regardless of different jet diameter, are clustered into groups of trend lines in accordance with their injection orientations, which implies that the effect of jet diameter is weak.

The effect of jet diameter on heat transfer augmentation can be analyzed by considering the recirculation speed near the vertical cooling plate. In mixed convection, the heat transfer rate is basically controlled by the flow velocity across the cooling plate, which is induced by forced and natural convection. Defining the characteristic Reynolds number on the cooling plate for forced convection as:

$$\mathrm{Re}_w = \frac{\rho u_w H}{\mu_w} \tag{3}$$

where H is the height of the cooling plate and u_w is the characteristic recirculation speed near the cooling plate induced by the forced jet.

For a forced jet, neglecting gravity and buoyancy, the momentum flux is conserved along the path of the jet, such that

$$M = M_0 \tag{4}$$

where the momentum flux M and M_0 can be expressed in terms of volumetric form assuming the fluid to be incompressible. Thus

$$M \approx Qu = \frac{\pi d_{bj}{}^2 u^2}{4} \tag{5}$$

$$M_0 = Q_0 u_0 = \frac{\pi d_{bj0}{}^2 u_0{}^2}{4} \tag{6}$$

where d_{bj} is the local diameter of the jet, u the local streamwise mean velocity, and u_0 the uniform jet exit velocity, $u_0=U_j$. In large volumes, free forced jets can be expected to be turbulent. For turbulent forced jets, List [10] provided an empirical relationship for jet volumetric entrainment rates which is useful for scaling purposes:

$$Q_{bj}{}' = \frac{dQ_{bj}}{dz} \approx \frac{Q - Q_0}{L} = \alpha_T \sqrt{8\pi M} \tag{7}$$

where α_T is Taylor's jet entrainment constant, typically taking a value around 0.05, and L is the length between the vertical cooling plate and the movable wall.

Based on experimental observation, the velocity of the jet while impinging the wall is of the same order of magnitude with the recirculation speed near the wall. For scaling purposes, it is reasonable for us to use the impinging velocity as the characteristic recirculation velocity near the cooling plate. Then from Equations (4) to (6) we have

$$u_w = \frac{u_0 d_{bj0}}{d_{bj}} \tag{8}$$

Equation (3) becomes

$$\mathrm{Re}_w = \frac{\rho u_0 d_{bj0}/\mu_j}{d_{bj}/H}\left(\frac{\mu_j}{\mu_w}\right) = \frac{\mathrm{Re}_j}{d_{bj}/H}\left(\frac{\mu_j}{\mu_w}\right) \tag{9}$$

With algebraic manipulation of Equations (4) to (7),

$$d_{bj} = 4\sqrt{2}\alpha_T L + d_{bj0} \tag{10}$$

Plug Equation (10) into (9),

$$\mathrm{Re}_w = \frac{\mathrm{Re}_j}{4\sqrt{2}\alpha_T \frac{L}{H} + \frac{d_{bj0}}{H}}\left(\frac{\mu_j}{\mu_w}\right) \tag{11}$$

The second term d_{bj0}/H controlled by the jet diameter in the denominator of Equation (11) is much smaller than the first term $4\sqrt{2}\alpha_T L/H$. So the jet diameter has little effect on the characteristic Reynolds number on cooling plate related to the forced jet, and therefore little effect on heat transfer augmentation.

Effects of Jet Modes on Heat Transfer Augmentation

To investigate the effects of jet modes, experiments are performed with a purely impinging jet (horizontal), a purely buoyant jet (vertical/up), and two mixed jet modes at 30° and 60° injections. Figure 4 shows the comparison of jet mode effects.

The experimental data for 30° and 60° injections, which are mainly controlled by the buoyant jet with some impinging jet effects, are a little higher than the data of the vertical/up buoyant jet. However, the purely impinging jet has significantly greater augmentation than the purely buoyant jet. Under the same air-flow rate, the big size jet has much smaller outlet velocity. In this case, the jet is no longer an impinging jet. So

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the experimental data of big size jets will fall on the buoyant jet curves.

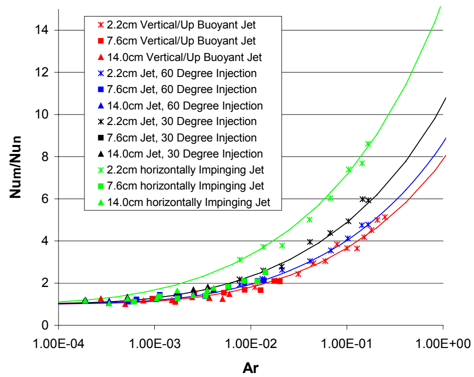

Figure 4. Effects of jet modes on heat transfer augmentation

Effects of Enclosure Aspect Ratio on Heat Transfer Augmentation

Figure 5 gives a comparison of the effects of enclosure aspect ratio. Experiments are performed separately in a large enclosure (2.11m×2.27m×2.18m) and a tall narrow enclosure (1.30m×2.27m×2.18m). The vertical/up injection orientation is used to avoid the effects of impinging jets. It can be seen the heat transfer augmentation increases with decreasing enclosure aspect ratio. Some experiments [7] have shown that the circulation speed induced by injection increases with decreasing enclosure aspect ratio. It can be used to explain the current experimental results. After the buoyant jet impinges on the insulated ceiling, it transforms to a ceiling jet flow spreading out with decreasing velocity. In a large enclosure, due to the friction, the ceiling jet flow may dissipate completely before reaching the vertical cooling plate. In the narrow enclosure, however, the ceiling jet flow may have enough energy to reach the corner and then turns down to accelerate the cooling wall jet. So it has higher heat transfer augmentation with small aspect ratio.

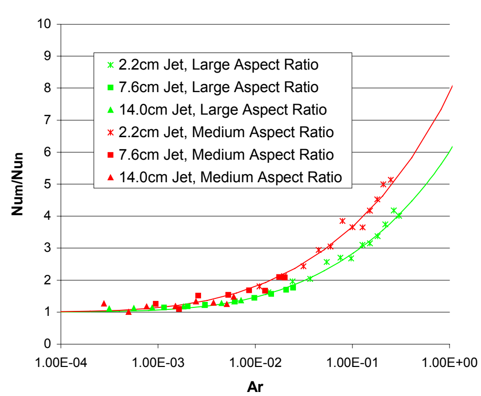

Figure 5. Effects of enclosure aspect ratio on heat transfer augmentation

Effects of Structures inside the Enclosure

The objectives of the research are to study the mixed convection in multidimensional, "clean" geometries, at scales approaching those of actual containment buildings or compartments, under a variety of well established conditions. However, the experimental data from a simplified, empty enclosure will inevitably be quite different from those from a real containment with complex interior structures. To investigate the effects of structures, a tube with 4.2-cm O.D. was placed as a blocking structure between the jet exit and the cooling plate. Experiments were performed to measure the variation of heat transfer under the condition of forced jet directly impinging on the tube or not.

If the jet hits the blocking structure before reaching the cooling plate, the loss of jet momentum can significantly affect the effectiveness of mixing the whole volume, and thus reduce the heat transfer coefficient. Except for the short region near the jet exit or impinged surface, the jet will develop into a linear decay region if not impinge on a blocking structure [1]. In this region, the distribution of the streamwise velocity across a free expanded jet has been given by List [29]:

$$u(r) = U_C e^{-(r/b)^2} \tag{12}$$

where U_C is the local streamwise velocity at the centerline, and b is the width parameter. The fractional momentum loss can then be evaluated by

$$\frac{M_{loss}}{M} = \frac{2\int_0^\infty C_D \dfrac{\rho u^2(r)}{2} D_t dr}{\int_0^\infty \rho u^2(r) \cdot 2\pi r dr} \tag{13}$$

where the drag coefficient $C_D \approx 0.9$ in the range of Reynolds number of the experiments for flowing transversely across the tube, D_t is the diameter of the blocking tube, ρ is the air density assumed to be constant. Integrating Equation (13) yields

$$\frac{M_{loss}}{M} = \frac{C_D D_t}{\sqrt{2\pi} b} \tag{14}$$

For a given self-similar distribution of mean velocity and pressure, the width parameter b is related to the local jet expanded diameter d_{bj} [29]:

$$b = \frac{d_{bj}}{2\sqrt{2}} \tag{15}$$

where d_{bj} can be obtained from Equation (10),

$$d_{bj} = 4\sqrt{2}\alpha_T x + d_{bj0} \tag{16}$$

Then from Equation (14) to (16) the fraction of momentum loss can be estimated in terms of the jet nozzle diameter, block size, and the distance between the jet origin and the blocking tube, as

$$\frac{M_{loss}}{M} = \frac{2C_D D_t}{\sqrt{\pi} d_{bj}} = \frac{2C_D D_t}{\sqrt{\pi}\left(4\sqrt{2}\alpha_T x + d_{bj0}\right)} \tag{17}$$

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This equation is only valid in the linear decay region far from jet exit and cooling plate. For the regions near the jet exit or cooling surface, the flow field has large spatial and temporal variations that make it complicated to estimate momentum loss. Equation (17) shows the fraction of jet momentum loss increases with the projected frontal size of the blocking structure and decreases with the jet nozzle diameter and the distance from the jet origin.

Figure 6 shows the experimental result under the condition that 2.2-cm I.D. jet nozzle was used and the distance between the jet exit and the 4.2-cm diameter blocking tube was 30 cm, and a comparison of heat transfer augmentation by an impinging jet with and without a blocking structure. The fraction of jet momentum loss can be calculated by

$$\frac{M_{loss}}{M} = \frac{2C_D D_t}{\sqrt{\pi}\left(4\sqrt{2}\alpha_T x + d_{bj0}\right)}$$

$$= \frac{2\times 0.9\times 4.2[cm]}{\sqrt{\pi}\left(4\sqrt{2}\times 0.05\times 30[cm] + 2.2[cm]\right)} = 39.9\%$$

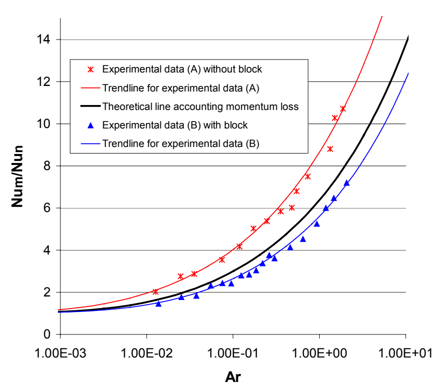

Figure 6. Effects of block structure on heat transfer augmentation and comparison of experimental data with theoretical line

The heat transfer augmentation by the forced jet clearly decreases if the jet does not impinge on the cooling plate directly. The reduction of the heat transfer rate by other structures can also be predicted by the loss of jet momentum. The heat transfer augmentation is a function of the jet Archimedes number,

$$Ar_j = \frac{{\mathrm{Re}_j}^2}{Gr_L} = \frac{\rho^2 {u_j}^2 {d_{bj0}}^2}{\mu^2 Gr_L} \tag{18}$$

The jet momentum flux is conserved along the path of the free jet and can be expressed as

$$M \equiv \frac{\rho\pi {d_{bj0}}^2 {u_j}^2}{4} \tag{19}$$

Equation (18) and (19) imply that, if density variation is neglected, the jet momentum is proportional to the jet Archimedes number, and thus

$$\frac{M_{loss}}{M} \propto \frac{Ar_{j,loss}}{Ar_j} \tag{20}$$

Figure 6 shows the theoretical curve predicted by Equation (1) with 39.9% reduction of jet Archimedes number for an unblocked jet, compared to the experimental data with 39.9% jet momentum loss due to the blocking structure. Both curves have good agreement, but the theoretical prediction is a little greater than the experimental data. This is because the theoretical curve, although has been accounting for the reduction of the jet Archimedes number, still comes from the impinging jet model, while the experimental jet, after impinging on a structure, will change the flow direction and can no longer totally impinging on the cooling plate directly.

CONCLUSIONS

The experimental studies have investigated heat transfer under combined natural and forced convection with a variety of jet injection modes. The heat transfer augmentation by forced jets is controlled by jet Archimedes number, fluid properties, jet mode, injection orientation, and enclosure aspect ratio. The experimental data are well correlated by developed correlations for heat transfer augmentation. The jet Archimedes number is the important parameter in characterizing mixed convection heat transfer. The jet injection orientation has a substantial effect on heat transfer while the effect of the jet diameter is very weak. For vertical cooling surfaces, an impinging jet can achieve more effective heat transfer than a buoyant jet. The heat transfer augmentation increases with the reduction of enclosure aspect ratio. If the jet impinges on a structure before reaching the cooling surface, the loss of jet momentum can significantly affect the effectiveness of mixing the whole volume, and thus reduce the heat transfer coefficient. The loss of jet momentum from impingement on structural objects is proportional to the loss of jet Archimedes number. Thus the reduction of heat transfer augmentation can be estimated in terms of jet momentum loss.

Future safety assessment of reactor systems will be based on integrated thermal hydraulics modeling, where uncertainty in model predictions will be characterized with high precision. Phenomena identification and ranking, combined with results from well-scaled separate effects and integral experiments, will ensure that all sources of uncertainty have been identified, characterized, and their effects quantified. To use small-scale tests to large equipment design, scale-up methods and criteria are important to match the key governing parameters. The insights for large-volume mixing found here can be generalized to provide generic rules for designing reactor safety systems that can be modeled with lower uncertainty.

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