Parametric Study of Negatively-Buoyant Wall Jets and Air Curtains

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Study and Control of Refrigerated Air Curtain Entrainment
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Abstract

This objective of this work was an extensive study of the parameters that affect ambient entrainment in refrigerated air curtains used in medium temperature vertical multi-deck display cases. The purpose of this study is to increase the understanding of the air-curtain physics and entrainment through a systematic parametric study. Experiments were conducted using flow visualization, cinematic Particle Image Velocimetry and thermo-couple measurements at various locations in the flow cycle. These techniques were used to investigate various effects including variations in: air jet width, velocity profile of the jet, velocity of the jet, temperature difference between the air curtain and to the ambient air and geometry of the display case (e.g. fully-stocked configuration with back-flow perforations vs. an idealized wall jet). Thermal entrainment and velocity distributions were obtained for several test conditions to understand these effects.

The air curtain was found to be highly turbulent at the inlet. Hence, the observed air curtain dynamics were found to be largely different from previous wall-jet experiments and simulations which employed laminar inflow conditions. The results indicated that the entrained mass flow rate was found to reduce nearly linearly with decreasing Reynolds number (jet velocity), but the effect of product heat flux would need to be taken into account before lower velocity levels can be recommended. In addition, the air curtain entrainment was found to reduce when decreasing the air curtain width and/or using a stepped profile instead of a uniform profile. However, increasing the Richardson number tended to increase mixing which somewhat unexpected since negative-buoyancy effects were thought to act as a stabilizer. As such, the increased mixing was attributed to modifications in the downstream velocity profile shape, where decreased jet temperatures led to increasing necking effects, which subsequently increased the velocity gradients and thus led to more mixing of the air curtain. With respect to geometry, the fully stocked configuration with perforations open was found to give the minimum entrainment of all configurations studied, even less than that from the solid-wall configuration.
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Chapter 1. Introduction

1.1 Motivation

This objective of this work was an extensive study of the parameters, which affect ambient entrainment in refrigerated air curtains used in medium temperature vertical multi-deck display cases. The role of the air curtain is to keep out the warm ambient air from interacting with the colder product. Refrigerated display cases are extensively used in supermarkets and grocery stores. For frozen foods, they are usually equipped with glass doors but mostly the open type is used for chilled products. For these open display cases, recirculated cold air is used to keep the case contents at a desired temperature, while allowing customers unhindered access to refrigerated product. Electricity consumption in large supermarkets represents a substantial share of the national electric use, about 4% in both the United States and Europe. This corresponds to an annual electricity consumption of about 400 kWh/m$^2$. According to estimates, refrigeration accounts for about 50% of the overall supermarket electricity consumption. Adams (1985) estimated that 75% of refrigeration load comes from air curtain entrainment. Hence, reducing ambient entrainment can be significant in terms of possible energy savings and is therefore the motivation for this study.

In terms of fluid physics, the air curtain is similar to a negatively buoyant wall turbulent jet. As a consequence of the ambient being a still environment some of the warm ambient air infiltrates the cold air curtain. Due to the turbulent nature of the jet, mixing occurs at the shear layer. Mass balance across the fans means that some cold air is spilled over into the ambient while warm ambient air is ingested into the return grille. The entrained ambient air causes the temperature of the recirculated air to increase. Warmer air collected at the return grille passes across the evaporator, thereby adding refrigeration load to the case, after which it is reinjected. Further, higher humidity levels in the return air (due to entrainment) lead to increased frost deposition rates on the evaporator coil which not only increases the pressure drop but also decreases the efficiency of heat transfer at the evaporator coil.

Thermal entrainment is expected to depend on various parameters such as the 1) air curtain inlet jet width, 2) velocity of the jet at the inlet, 3) velocity profile of the issuing jet (related to the Reynolds number of the flow), 4) temperature difference between the air curtain to the ambient (related to Richardson number), and 5) turbulence levels in the jet, etc. Refrigerated display case design has generally been empirical in nature since the influence of these parameters has not been a subject of a detailed fundamental investigation. A study to help understand the basic fluid flow physics associated with these five parameters and the variation of ambient entrainment with different air curtain parameters is thus important to proper industrial design, and is the motivation for the present study.

Considerable effort has been made into the understanding of air curtains through CFD (computational fluid dynamics) simulations. However, these studies have been mostly successful in a qualitative sense since the validations have been qualitative. Part of the lack of agreement with practical display case performance may be attributed to the complex turbulent conditions at the jet exit. Hence an experimental study utilizing advanced non-intrusive diagnostics assumes importance. In particular, experiments of an air curtain that will quantify the thermal entrainment, as a function of different parameters will help devise guidelines for the efficient design of refrigerated display cases, and is the motivation for the current study.

A further motivation for this study is the fact that wall jets have significant engineering importance in automobiles and gas turbine engines. In particular, wall jets are used for mass transfer applications in automobile
defrosters, while in aero engines and gas turbines they are used for cooling of combustion chamber walls and film cooling of turbines. In such cases, understanding the mixing of wall jet with the ambient flow and momentum transfer is desirable to maintain the protective film cooling layer as far downstream as possible. Such an understanding of the wall jet fluid dynamics is also useful for airfoils in high-lift configurations, where it is necessary to supply momentum to the boundary layer to avoid separation. At lower Reynolds numbers, the inflow is typically laminar but the shear layer and boundary layer instability soon dominate the dynamics.

1.2 Previous Wall Jet/Air Curtain Research

1.2.1 Transitional and Turbulent Wall Jet Studies

The fully turbulent wall jet has a two-scale character (Figure 1.1). The inner layer is similar to a turbulent boundary layer. And the outer layer is a free shear layer. The interaction of the large-scale structures of the outer shear layer with the small-scale structures in the wall boundary layer determines the development of the wall jet.

Figure 1.1: Two-scale nature of the wall jet from Karlsson et al. (1998)

Bajura & Catalano (1975) used hot-film flow visualization studies to investigate the mechanism of transition in forced and unforced wall jets in the Reynolds numbers ranging from 100 to 600 with laminar inflow condition. They found that the Kevin-Helmholtz instability of the free shear layer was primarily responsible for the transition of the wall jet. The initial stages of transition were found to be two-dimensional and dominated by the mechanism of vortex pairing. They described the transition process involving different stages such as formation of discrete vortices in the free shear layer, coalescence of adjacent vortices, eruption of these vortices into the ambient fluid, dispersion of the organized flow pattern by three-dimensional turbulent motions and relaminarization of the upstream flow until vortex pairing occurs again.

Hsiao & Sheu (1996) studied the flow transition in a wall jet in a range of Reynolds numbers from 300 to 30,000 using hot-wire anemometry. They concluded that the wall jet transition is initially triggered by the shear layer vortex and that the transition process strongly depends on the Reynolds number of the flow at the exit.
Abrahamsson (1997) used hotwire anemometry to study the wall jet in stagnant surroundings. The experiments involved the investigation of two- and three-dimensional turbulent wall jets. The measurements were focused on the fully developed region of the flow. The two-dimensional wall jet was found to be self-preserving, and the streamwise development of the maximum mean velocity and the half-width was found to be independent of the inlet Reynolds number. The lateral spreading of the three-dimensional wall jet was found to be five times larger than the wall-normal spreading rate. Transport of turbulent energy towards the wall region was observed due to advection and turbulent diffusion.

Gogineni and Shih (1997) studied the transitional process in unforced wall jets (parabolic velocity profile and laminar inflow conditions) and free rectangular jets in a range of Reynolds number from 330 to 3800 using flow visualization (laser-sheet/smoke). Growth of the wall jet (in y-direction) was found to be significantly slower than that of the corresponding free jet. For Reynolds numbers less than 2200, the flow was found to be initially laminar and eventually became turbulent following the transition process as shown in Figure 1.2. The streamwise location of vortex formation (for Re < 950) and the eventual vortex breakdown (for 950 < Re < 2200) was found to be closer to the jet exit with increasing Reynolds number. For a Reynolds number of 3800, the wall jet was found to be fully turbulent at the inlet. This is particularly significant to the present research since the air curtain is always turbulent at the exit, even at Reynolds numbers as low as 800 (as will be shown).
Gogineni and Shih also investigated the transition in an unforced wall jet at a Reynolds number of 145 using Particle Image Velocimetry (PIV). The flow development was related to the formation of the inner-region vortex and the subsequent interactions between the outer free-shear layer region and the inner boundary layer region. They showed that under the influence of the outer shear layer vortex, the local boundary layer at the wall becomes unstable causing detachment from the wall and a vortex formation in the inner region. The free shear layer vortex and the inner region vortex form a double-row vortical structure, which is convected downstream. The mutual interaction between the vortical structures dominates the transition process. The three-dimensionality is initiated in
the outer region and spreads to the inner region. However, the developmental stages observed by Bajura & Catalano (1975) were not observed at this Reynolds number.

Tong & Warhaft (1994) studied the effect of placing a fine circular ring close to the exit of an axisymmetrical jet on the flow transition. They found that spreading rate and turbulence intensity could be reduced by employing this method. The diameter of the wire did not have significant effect on the process. Schober et al. (1999) studied the effect observed by Tong & Warhaft (1994) on a two-dimensional plane wall jet. They used smoke flow visualization to investigate wall jet whose Reynolds number were varied from 2500 to 10000, based on the exit velocity and exit slot width. They observed that Kevin-Helmholtz instability leads to the formation of the shear layer vortices, which subsequently undergo one or more stages of vortex pairing. With increasing Reynolds number, the vortex pairing was observed closer to the wall jet exit (Figure 1.3). For Re = 5000, they observed the affect of placing a still wire at the edge of the shear layer. Placed directly behind the nozzle, this condition was found to prevent formation of shear layer vortices. Hence, vortex pairing was no longer observed. This was found to significantly reduce the size of turbulent structures, spreading of wall jet and mixing with the ambient. In the case of an oscillating wire, the vortex structures depended on the wire frequency. However, low oscillation frequencies were found to lead to the formation of large vortices, leading to a significant increase in spreading rate and mixing with the ambient (Figure 1.4).
Figure 1.3: Vortex breakdown in wall jets from Schober et al. (1997)
Karlsson et al. (1998) studied a wall jet in a water channel at Re = 9600. Laser Doppler Velocimetry (LDV) measurements were used to measure the velocities in the plane wall jet. Mean velocity profiles in outer scaling were similar for downstream locations ranging from $x/b \sim 20$ to 200. Further, the wall jet thickness based on $y_{1/2}$ was found to vary linearly with $x/b$. The results are shown in Figure 1.5. (Note that $b$ denotes the wall jet width at the inlet. In the present study, width at the inlet has been denoted by $H$).
1.2.2 Display Case Air Curtain Studies

Past studies of display cases have generally focused on the energy balance of the system. In this type of analysis, the display case as a whole is considered to be the system and heat input and heat dissipated due to various components such as fans, evaporator coil etc were measured to ascertain the energy efficiency. The display case is generally tested for 2-3 days in a carefully controlled laboratory atmosphere to estimate the energy statistics under different conditions. Most of these studies have concentrated on energy issues and defrost cycling, but did not focus on understanding the fluid physics of the air curtain itself relevant to the present study. Some exceptions are noted in the following.
Loerke and Nagib (1976) carried out a study into the effectiveness of reducing the initial turbulence of the air curtain by placing honeycomb sections at the inlet of the air curtain jet. They found that the resultant effect of the honeycomb was to reduce the turbulence transverse fluctuations in the velocity along the length of the honeycomb section. The result is a largely unidirectional velocity component at the exit of the honeycomb. The airflow exiting the honeycomb cells contains small-scale, high frequency turbulence due to the action of the honeycomb. However, the high frequency turbulence quickly dissipates resulting in a net reduction in turbulence intensity. Despite the positive advantages of honeycomb in reducing turbulence intensity, accumulation of dirt leads to increased pressure drop and also adds to the maintenance costs required by cleaning.

Stribling et al. (1997) studied the air curtain using a two-dimensional Computational Fluid Dynamics (CFD) model of a refrigerated display case. The objective was to study the effectiveness of computational studies in predicting display case performance. The CFD consisted of Reynolds-Averaged Navier-Stokes (RANS) equations coupled with a $\kappa$-$\epsilon$ turbulence model and a Reynolds stress model with multi discretization schemes. The honeycomb section was simulated by using a mathematical model for analyzing flow through porous media. The jet exiting the top was angled away from the vertical towards the ambient. However, the peak velocity in the mean profiles was found to follow a more vertical path, being deflected towards the shelves by the negative buoyancy effects. The computed results were dependent on the turbulence model chosen and compared with the experimentally obtained velocity profiles only in a qualitative sense.

Cortella et al. (2001) employed a finite element code based on the stream function vorticity formulation, which incorporates a Large Eddy Simulation (LES) turbulence model. The two-dimensional CFD simulation was used to analyze the air flow patterns and temperature distributions in open display cabinets. The air curtain considered had two different jets at the inlet exiting at different velocities and temperatures. The velocities of the inner and outer jets were varied to study the affect on air curtain energy efficiency. Higher velocity at the jet closer to the wall was found to be detrimental in terms of air curtain stability and energy efficiency. In current practice, some manufactures use an air curtain profile with higher velocity near the wall and find that as a favorable design. The discrepancy may be attributed to the effectiveness of turbulence model in simulating the high turbulence levels that are inherent to the display case air curtains.

Field and Loth (2001) investigated the air curtain behavior experimentally using cinematic PIV for flow visualization and velocity field measurements. Flow visualization studies in Reynolds number typical to display case air curtain operation ($3800 < Re < 8000$) showed that the air curtain was turbulent in nature. Clearly defined vortices were not found in the shear layer. The mixing in shear layer was found to be due to protrusions and indentations as shown in Figure 1.6. The fully stocked product shelves were replaced with a solid wall for wall-jet idealization to study the fundamental flow behavior. Isothermal and refrigerated air curtains with a uniform profile at inlet were studied to understand the velocity and buoyancy effects on ambient entrainment. Initial acceleration in the air curtain was observed until a streamwise distance equal to thrice the inlet width of the air curtain due to negative buoyancy effects and further down the air curtain spreading occurred due to momentum diffusion. In the Reynolds number range experimented ambient entrainment was found to increase with decrease in Reynolds number or an increase in Richardson number.
Bhattacharjee (2002) conducted a CFD study to understand the flow physics of air curtains and the effect on entrainment. The air curtain was modeled as a negatively buoyant wall-jet and the study concentrated on the air curtain rather than the display case itself. At Reynolds numbers of 2000 and lower, the primary solution technique was the Direct numerical Simulation (DNS) of the full Navier-Stokes equations with a laminar inflow condition whereas a RANS approach was adopted at higher Reynolds numbers (Re ~ 2000-10000) with turbulent inflow condition. In the laminar diffusion regime (100 < Re < 550), a larger spread in the flow was found as the viscous effects increase (Re decreases). Hence, thermal and momentum entrainment was higher at lower Reynolds numbers. The entrainment is lowest at about Re of 700 and further increase in Reynolds number yielded eddy production via flow transition and thus a gradual increase in entrainment. For fixed Reynolds number, the thermal & momentum entrainment were found to decrease with increasing Richardson number. The discrepancy of entrainment trends with the experiments of Field (2001) was primarily attributed to high turbulence levels in the air curtain during experiments, which was neglected in the simulations. However, the experimental and computational results for uniform profile at inlet were in good agreement in terms of energy loss as shown in Figure 1.7. The CFD study also
found that entrainment was to be significantly affected by velocity profile, wherein a constant gradient profile with the highest velocity towards the wall side gave least entrainment. This was qualitatively consistent with experiments of Field and Loth (2000).

D’Agaro and Cortella (2002) carried out a two dimensional CFD simulation of a vertical open display case for frozen food which has a three-layer air curtain at different velocities and temperatures at the inlet. The simulations were completed using two codes and compared with experimental results. The first code is a finite element code described in Cortella et al. (2000) and the second code is a commercially developed finite volume code with opportunity for steady and transient flow analysis with multiple order discretization schemes. The computational results from the two codes were not in close agreement with each other, which was attributed to the choice of turbulence model and interpolation schemes.

Navaz et al. (2002) combined a CFD simulation effort with an experimental investigation using digital PIV for analyzing the performance of air curtains. A commercial computer program, ROYA, was used with a Cebeci-Smith turbulence model and $\kappa$-$\varepsilon$ model for two-dimensional simulation. They studied the infiltration rate as a function of two variables: the inlet air velocity, and the inlet temperature. The computations were successful in predicting the velocity profiles and temperatures qualitatively but the computational results were found to overestimate the velocity profile. The experimental data was available only for two different velocities to make any certain conclusions. However, the entrainment trend was in agreement with Field (2001) as shown in Figure 1.8. The computations resulted in a conclusion that entrainment across the air curtain is predominantly momentum driven, which means that entrainment has a strong dependence on inlet air velocity and a very weak dependence on inlet temperature.

Figure 1.7: Comparison of computations of Bhattacharjee (2002) and experiments of Field (2001)
1.3 Outstanding Issues and Project Goals

The problem of reducing energy consumption of refrigerated display cases continues to be a challenge despite technology advancement in the last 20 years. The interaction of the air curtain with the warmer ambient is a complex phenomenon depending on various factors such as the inlet velocity, velocity profile, inlet temperature, turbulence levels in the air curtain, and the air curtain width and. However, design of air curtains has been primarily empirical in nature without completely understanding the behavior of air curtain under different conditions.

For the more fundamental flow of the wall jet, studies to date have primarily concentrated at understanding the transitional flow mechanism. In most cases the wall jet studied was well conditioned with very low turbulence levels at the jet exit, and the emphasis being on understanding the mutual interaction of boundary layer vortical structures with the shear layer vortical structures and how the interaction affects transition to turbulence and mixing with the ambient. In addition, most wall jet studies have exclusively concentrated on isothermal wall jets. However, the display case air curtain is a negatively-buoyant wall jet with high initial turbulence.

Therefore, the effect of Reynolds number, Richardson number and inlet velocity profile on the turbulence levels in the streamwise direction and mean velocity profile are important, but not well understood and will be studied herein. In addition, the thermal entrainment trends and energy loss due to entrainment will be quantified in a systematic manner as a function of different operating parameters. Finally, experiments aimed at investigating the effect of flow through back panel and the effect on product temperature will be undertaken in a fully stocked shelf environment. Designers may be able to use such results as guidelines for efficient design of air curtains.

Figure 1.8: Comparison of entrainment trends from Navaz et al. (2002) and Field (2002)
Chapter 2. Experimental Method

2.1 Air Curtain Facility

2.1.1 Refrigerated Display Case

The air curtain experiments for the current study were carried out on a Hussmann D6-4EU model commercial dairy display case used in supermarkets. The 1.2 meter long section of the display case (Figure 2.1) was provided with its own compressor and condenser. The cases manufactured in 1.2 m sections (4 feet) can be put end-to-end for installation in supermarkets as seen in Figure 2.2. In the supermarket, many cases attached together are typically provided with a single refrigerant line and the compressor and condenser are often mounted on the roof of the building.

Figure 2.1: An eight-foot version of the Hussmann D6-4EU dairy case used in this study
Figure 2.2: Typical end-to-end arrangement of the display cases in a supermarket

For the operation of the air curtain for the present experiments, steady-state conditions will result in the same amount of air being recirculated in the display case forming a closed loop (Figure 2.3). The amount of ambient air entrained is equal to the spillover at the capture area, thus satisfying the conservation of mass of the air being recirculated. The air curtain if formed by forcing refrigerated air at the top of the display case through a honeycomb. The honeycomb serves to straighten the flow and reduce irregularities in the flow. The honeycomb has a cell diameter of 4 mm and is 25 mm long. The curtain inlet has a uniform (base-line) width of 120 mm all along the length of the case. The length of the air curtain from the honeycomb to the capture area is 1.6 m. Two 20 cm diameter fans at the bottom of the display case draw the air through the capture area and propel it into the duct running up the back of the case behind the product area and through the evaporator coils mounted in the duct to refrigerate the air. Fans run by a variable-voltage DC motor were used to control the inlet jet velocity. The commercial name of the brushless DC tubeaxial fans is Caravel DC fans manufactured by Comairtron. They deliver a nominal airflow at 550 cfm at 1650 rpm. Note that this method allows greater control of flow rate than previous study where speed was controlled by changing fan blades of different pitch and/or removing fans (Field, 2001)
Figure 2.3: Refrigerated display case in the wall jet configuration

The duct turns at the top of the case and the air is blown forward where it is pushed into the deflector and emerges again as the air curtain (Figure 2.3). The shape of the deflector primarily determines the inlet velocity profile of the air curtain exiting the honeycomb. A deflector with uniform angle was used to generate a basic uniform velocity profile at the inlet, whereas, a deflector bent in multiple places was used to generate a ‘stepped’ velocity profile with higher velocity towards the product side and a lower velocity towards the ambient side.
2.1.2 Wall Jet Configuration
The air curtain for a refrigerated display case is a negatively-buoyant planar jet with interaction on the product side as well as the ambient side. When the shelves are stocked with product, the cold airflow from rear perforations provides support to the air curtain besides cooling the product. As a result an approximate wall jet exists along the face of the products, and a shear layer grows on the room side of the air curtain. Hence, modeling the air curtain as a wall jet is a reasonable initial approximation to understand the fundamental fluid dynamics of the air curtain. To allow the study of the fundamental flow features and avoid geometry specific issues, the wall jet like aspects of the air curtain were directly studied in some of the experiments. To realize the wall jet idealization, the product side of the display case has been replaced with a flat sheet metal wall (Figure 2.3). The x-axis is directed vertically downwards along the flow direction and the y-axis is directed away from the wall along the width of the honeycomb (Figure 2.4).

The capture areas for different display cases have different geometry. In the display case model used, fans pull the return air through a narrow return air grill near the front of the case. The capture area was modified to include all the area up to the solid wall to pull the air curtain straight into the capture area rather than the turn through the return grill. This resulted in the capture area being enlarged to 240 mm width as depicted in Figure 2.4.

2.1.3 Fully-Stocked Display Case Configuration
In supermarkets, the refrigerated display case air curtain is not a simple wall jet. The behavior of the air curtain is affected by the way in which the product shelves are stocked, the disturbance introduced by the consumer picking the product and the flow from rear perforations. The cold air from the rear perforations helps cool the rear and sides of the product. The interaction of the air curtain with product boxes, product shelves, and the cold flow from the rear perforations has been studied with the present display case as a fully stocked configuration. To simulate this configuration, the product shelves have been filled with cardboard boxes as seen in Figure 2.5, which approximate the dairy product usually stocked in this model display case.

Further, the study also involved understanding the effects of the rear perforations and the cold air flowing from these perforations on the air curtain behavior and interaction with the ambient. Different levels of rear perforations were left open to investigate the effect of the cold flow from the rear on the air curtain fluid dynamics.
Figure 2.4: Refrigerated display case dimensions

- H = 120 mm or 90 mm
- L = 1.6 meters
- 240 mm
2.2 Particle Image Velocimetry (PIV)

2.2.1 PIV System Setup

Particle Image Velocimetry (PIV) was used to determine the velocity of the air curtains being studied. PIV is a non-intrusive technique for measuring instantaneous fluid velocities and consists of a double-pulsed laser sheet illuminating a cross section of a seeded airflow. This technique employs a digital camera, which takes two pictures of the illuminated seed particles. A cross-correlation algorithm computes the 2-D velocity field for the illuminated field of view.

A fully digital, cinematic PIV system (Figure 2.6) was obtained from LaVision, Inc. The system consists of dual pulsing New Wave Gemini Nd:Yag lasers, a cross-correlation Kodak ES1.0 digital camera, and a Windows NT based computer system with LaVision's hardware board to synchronize the lasers, a digital camera, and the DaVis version 5.4.4 software to process the PIV images. The cross-correlation digital camera eliminates any directional ambiguity in the flow.
The fast shuttered PIV camera is set to double-frame/double-exposure mode to take two exposures shortly after one another, with the time interval between the two exposures being 1.2 milliseconds. The camera is capable of producing 1008 x 984 pixel images for cross correlation and taking 15 particle image pairs per second. Figure 2.7 shows the schematic of such a double exposure image pair divided into interrogation spots. The algorithm computes the cross correlation of all interrogation cells from the upper part (frame 1) with the corresponding cells in the lower part (frame 2). A complete description of double-frame cross-correlation PIV used in the processing is contained in Keane and Adrian, (1992). The analysis routines found in the DaVis version 5.4.4 software are also documented in the manufacturer's software manual (LaVision's PIV Flowmaster Manual, 2000).
The air curtain is assumed to be two-dimensional in nature and the particle images are taken from the middle cross section of the air curtain illuminated by the lasers. The lasers were operating at 75% of their 120mJ/pulse full power, and an f-stop of 1.8 for the 50mm Nikkor lens. The PIV images were analyzed using an Adaptive multi-pass vector analysis. The first pass uses a 64x64 pixel interrogation spot to produce a vector; the second pass refines the analysis with a 32x32 pixel interrogation spot. With a 50% vector overlap, this produces a vector field of 63x62 vectors per image frame. The field of view increased from approximately 240 mm square at the curtain inlet to approximately 350 mm square at the bottom of the curtain. Post-processing of the cross-correlated vector images is done to eliminate the spurious vectors and obtain good RMS values. The post-processing method is explained in detail in section 2.4.4. The number of double frame image pairs that can be obtained at a stretch is limited by the memory of the PC used for data acquisition. Hence, ten sets of fifty cross correlation image pairs were taken at each field of view.

2.2.2 Particle Seeding

The PIV technique is based on the assumption that the seed particles are dispersed uniformly and closely follow the flow. The seed particles have to be small enough to follow the mean velocity and the primary vortices. Also, the laser reflections from the particles have to be captured by the camera. Hence, the particles have to be large enough to be seen on the image. Therefore, the choice of seed particles forms an important factor in the PIV technique.

The ability of the particles to follow the flow is characterized by the Stokes number for the particles in the fluid. The Stokes number \( S_t \) is defined as the ratio of a particle characteristic response time to the characteristic time scale of the flow. The particle will respond to the instantaneous changes in the flow if the Stokes number is less than unity. For the experiment, 30 \( \mu \)m hollow glass spheres were used to seed the air curtain flow. They are
manufactured by 3M industries and the brand name is Microspheres. The particle diameter varies in the range of 15 – 65 \( \mu \text{m} \), with an average diameter of 30 \( \mu \text{m} \). The density of the particles is 600 kg/m\(^3\), yielding a response time of 5.0 milliseconds. The desired size of the particles was obtained using the methodology outlined in Adrian (1991). For the specifications of the laser and the optics used in the experiment as described in the previous section, the above method estimates a Stokes number for the particle in the flow of the order of \( St = 0.06 \) (Field, 2001). As the Stokes number is very less compared to unity, the particles follow the flow very closely.

The particles were introduced into the air curtain just upstream of the evaporator coil. For this, compressed air was forced into a 1 liter Erlenmeyer flask that contained particles and the particle were delivered into the air circuit by a narrow tube (inner diameter of about 5 mm). When the particles exited the honeycomb with flow, they were seen to be uniformly dispersed in the flow. For flow visualization of the air curtain dynamics, only the air curtain was seeded. However, to obtain a vector map for the full field of view it is necessary to seed the ambient as well. A manual dispersal method was employed to seed the ambient. For this, the seed particles were shaken in the ambient with two foam strips at a distance from the air curtain without significantly disturbing the air curtain itself.

### 2.3 Temperature Measurements

The entrainment of the ambient air is a function dependent on the inlet temperature, return air temperature of the air curtain and the temperature of the ambient air. Hence to estimate the entrainment, it is necessary to track the temperature changes. Three K-type chromium-aluminum thermocouples were used to record the temperature history. An Omega HH611PL4C data logger was used to record the three thermocouples simultaneously. The arrangement of the thermocouples is shown in Figure 2.8. The first thermocouple is located just upstream of the honeycomb and measures the inlet temperature if the air curtain, \( T_{\text{jet}} \). The second thermocouple records the ambient air temperature, \( T_{\text{amb}} \). The third thermocouple records the return air temperature of the air curtain. Near the capture area, the region of the air curtain closer to the wall is relatively colder than the region towards the ambient side, which has entrained considerable warm air from the ambient. Therefore, to measure the averaged temperature in the return air the third thermocouple is placed behind the fans. The fans mix the return air efficiently and the thermocouple located directly behind the fans records the averaged temperature of the return air, \( T_{\text{cape}} \).
The thermocouples log the temperature every one-minute since the compressor is switched on. The jet temperature and the return air temperature reach steady state approximately one-hour after the compressor is switched on. For evaluating the entrainment, the average steady state temperatures recorded by the three thermocouples are used. The data acquisition is started after the steady state temperature is established and finished before the defrost cycle is reached.

2.4 Data Analysis

2.4.1 Dimensionless Flow Parameters

The air curtain flow is characterized by two non-dimensional numbers, namely Reynolds number (Re) and Richardson number (Ri). Reynolds number, the ratio of the momentum forces to viscous forces in the flow is
defined as \( \text{Re} = \frac{V_{jet} \rho_{jet} H}{\mu_{jet}} \). A higher Reynolds number means that the momentum forces dominate the fluid dynamics of the air curtain. Richardson number, the ratio of the buoyancy forces to viscous forces in the flow is defined as \( \text{Ri} = \frac{(\rho_{amb} - \rho_{jet}) g H}{\rho_{amb} V_{jet}^2} \). Richardson number is a measure of the strength of negative-buoyancy forces in the refrigerated air curtain. For an isothermal air curtain, there is no negative-buoyancy driving the flow and the Richardson number is equal to zero. A high Richardson number indicates that the negative buoyancy forces significantly affect the fluid dynamics of the air curtain.

2.4.2 Dimensionless Entrainment Parameters

A wall jet (or air curtain) thickness parameter and a thermal entrainment parameter have been defined to measure the air curtain entrainment. In available literature on wall jets, the momentum parameter generally used was the air curtain half width (denoted by \( y_{1/2} \)), which is defined as the distance away from the wall at any cross section at which the mean streamwise velocity of the air curtain is half of the maximum streamwise velocity at that section of the air curtain (i.e. \( y = y_{1/2} \) at \( V_x(y) / V_{x,max} = 0.50 \)). The air curtain thickness (\( \delta \)) is another momentum entrainment parameter used by Field and Loth (2001). Air curtain thickness, \( \delta \) is defined as the \( y \)-distance away from the wall at which the mean streamwise velocity reduces to 25% of the peak streamwise velocity, i.e. \( \delta = y \) at \( V_x(y) / V_{x,max} = 0.25 \).

To characterize mixing, the thermal entrainment parameter (\( \alpha \)) is defined as the ratio of mass flow rate of ambient air to the mass flow rate of the recirculating air in the wall jet (or air curtain) closed loop. By energy balance at the inlet and the capture area (assuming constant \( C_p \)), thermal entrainment parameter can be expressed in terms of measured temperatures as, \( \alpha = \frac{T_{capture} - T_{jet}}{T_{amb} - T_{jet}} \), where \( T \) is the temperature and ‘jet’, ‘amb’ and ‘capture’ refer to the conditions at the inlet, ambient and the return air grill respectively. In general, thermal entrainment parameter varies in the range 0 to 1 and is defined only for refrigerated air curtains (\( T_{amb} \neq T_{jet} \)). The thermal entrainment parameter equals zero when there is no ambient air entrained at all, an ideal situation that is never realized in practice. By continuity, the mass flow of cold refrigerated air that is lost to the ambient is equal to the mass flow rate of ambient air entrained into the refrigeration loop. Thus \( \alpha \) reflects the loss of mass flow from the air curtain to the ambient.

The thermal entrainment parameter, \( \alpha \) as defined above assumes that \( C_p \) is constant in the wall jet and the ambient. However, with varying humidity levels the differences in \( C_p \) vary because of the higher heat content of the moist air. Therefore, \( \alpha \) was found to have some lack of repeatability over different times of the year. This however does not mean that the volume fraction of refrigerated air curtain lost to the surroundings is dependant on the humidity content of ambient air. The observed trend is due to higher energy loss to the surroundings when the humidity levels are high.
2.4.3 Energy Loss Due To Entrainment

The energy lost to the surroundings due to the entrainment of the ambient air is expressed in terms of the basic dimensionless parameters defined in the previous section. Thermal energy loss for cooling caused by entrainment losses alone (per unit width) is based on mass flow rate (per unit width) and equal to

\[ E = \dot{m} C_p \left( T_{\text{capt}} - T_{\text{jet}} \right) \]

Where \( \dot{m} \) is the mass flow rate per unit width, defined as \( \dot{m} = \rho_{\text{jet}} V_{\text{jet}} H \). Note that the convective energy loss due to entrainment does not include the energy loss due to radiation losses & pressure drop. The energy loss can be expressed in terms of Reynolds number (Re) and thermal entrainment parameter (\( \alpha \)) as

\[ E = \text{Re} \mu_{\text{amb}} C_p \alpha \left( T_{\text{amb}} - T_{\text{jet}} \right) \]

where factor \( \mu_{\text{amb}} C_p \left( T_{\text{amb}} - T_{\text{jet}} \right) \) is typically fixed for a display case. Therefore, dividing the energy with the constant term \( \mu_{\text{amb}} C_p \left( T_{\text{amb}} - T_{\text{jet}} \right) \) leads to an expression for the dimensionless energy loss due to entrainment (\( E' \)), i.e.

\[ E' = \text{Re} \alpha \]

Hence, one goal for the designer would be to minimize \( E' \), while maintaining the desired product temperature.

2.4.4 Velocity Profile Computation

The air curtain fluid dynamics is unsteady in nature. The mean velocity development of the air curtain was thus studied using the time-averaged flow field of the unsteady air curtain. The cross-correlation vector images taken over a short period of time were averaged to obtain the velocity statistics. At each point in the air curtain, the time averaged velocity vector is obtained by averaging 500 vectors obtained by cross correlation of the double exposure images taken at 500 different time instances. The five hundred vector images averaged together produced a time-averaged velocity flow field for the curtain.

Post processing of the 2D vectors field obtained by cross correlation was important to reduce the statistical noise. As the ambient was seeded by dispersing the particles manually, often there were regions of PIV data that were not sufficiently seeded. Hence the 2D cross correlation vector field contained regions of insufficient statistical points resulting in artificially high values RMS velocity values. The post processing removes the spurious vectors based on a median filter. The DaVis median filter computes the average vector from the 8 neighbor vectors and compares the center vector with this average. If the RMS component of a center vector differed from the RMS component of the median vector of its neighbors by more than 20%, then it was considered a spurious vector. Such a spurious vector is replaced with the average of the neighboring vectors. If one of the neighbor vectors is a zero vector, the vector at this point is omitted from the calculations. In order to make the computations of RMS values more accurate an additional condition stipulating that each vector to have at least 5 non-zero neighbor vectors was imposed. Interpolation and smoothing of vector filed was used to fill the gaps and reduce noise.

The mean velocity profiles at any x-location are obtained by time averaging as well as spatial averaging. The cross correlation algorithm gives a time-averaged 2-D velocity field. Velocity profiles were obtained from the time-averaged 2-D velocity field by averaging fifteen horizontal rows of velocity vector data, centered about the row of interest. This averaging was done to reduce statistical noise in each row of PIV vectors. For consistent evaluation of the Reynolds number and the Richardson number, the mean stream wise velocity at x = H is used as \( V_{\text{jet}} \), i.e.,
\[ V_{\text{jet}} = \left( \frac{1}{H} \right) \int V(x = H, y) \, dy \]

This location was chosen since the mean velocity at the inlet could not be obtained because of the noise in the velocity data due to the laser reflections from the honeycomb.

To calculate the root-mean-square (rms) values of velocity fluctuations for the air curtain, the stream wise velocity for each point was first averaged over a nine-point stencil, yielding \( V_{j,\text{nine}} \). The formula used for the calculation of RMS velocity for a point \((i, j)\) in row \(i\) column \(j\) was:

\[
V_{\text{RMS}} = \sqrt{\frac{1}{500} \sum_{j=1}^{500} (V_{j,\text{nine}} - \overline{V_{j,\text{avg}}})^2},
\]

where, \( V_{j,\text{nine}} = (V_{i,j} + V_{i-1,j} + V_{i+1,j} + V_{i,j+1} + V_{i+1,j+1} + V_{i-1,j+1} + V_{i,j-1} + V_{i-1,j-1} + V_{i+1,j-1})/9 \)

\[
\overline{V_{j,\text{avg}}} = \frac{1}{15} \sum_{k=1}^{15} V_{j,(i+k-8)}
\]

\( V_{j,\text{nine}} \) is the average stream wise velocity for the nine points about point \((i, j)\), and \( \overline{V_{j,\text{avg}}} \) is the average of the stream wise velocity component of fifteen rows of the time-averaged vector field centered about point \((i, j)\). Averaging over a 25-point stencil did not offer any significant improvement in the RMS velocity values. Therefore, averaging over a nine-point stencil was used for all data analysis.

The removal and replacing of spurious vectors as described in the post processing method was sufficient to eliminate noise from the RMS values while avoiding the use of spurious data. The RMS values obtained by this method showed significant difference from the RMS values computed without post processing. However, the post processing method did not introduce any significant changes (i.e. less than 0.02\%) to the computed mean velocity components (Figure 2.9), which is very desirable. The change in mean velocity due to processing was less than 0.02\%. The effect of post processing method on RMS values is shown in Figure 2.10.
Figure 2.9: Effect of post-processing on stream wise profiles at Re = 7000,
a) at x/H = 1, b) at x/H = 6
2.5 Test Conditions

The fluid dynamics of wall jet configuration as well as the refrigerated air curtain in the fully stocked configuration were both studied. The wall jet configuration was studied to understand the basic phenomenon occurring in the development of the wall jet as it grows and entrains air. For the wall jet configuration, a nominally uniform profile at the inlet was set as the baseline velocity profile. This profile was achieved by using a constant
gradient deflector to turn the flow into the honeycomb. In addition, a stepped velocity profile (typical of manufactured display cases) was used that has a higher velocity towards the product/wall side and a lower velocity towards the ambient side. The stepped velocity profile was obtained by using a twin-bend deflector, such that the flow underwent two turns instead of one. Figure 2.11 shows the uniform velocity profile compared against a stepped velocity profile at $x = H$, where one can note the region of $y/H < 0.7$ is nearly constant for the former condition. The baseline wall jet inlet width was 120 mm, but a 90 mm inlet width configuration was used for comparison.

Figure 2.11: Comparison of ‘Uniform’ profile and ‘Stepped’ profile at the location $x = H$

Further, the refrigerated air curtain was investigated in a fully stocked configuration. The shelves were stocked with cardboard boxes yielding interactions with product, shelves. In addition, the back perforations were considered in open, half-open and closed conditions to investigate the effect of cold flow from behind.
### Table 2.1: Test conditions

<table>
<thead>
<tr>
<th>Test Conditions</th>
<th>Reynolds number</th>
<th>Richardson number</th>
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</thead>
<tbody>
<tr>
<td>Uniform profile</td>
<td>4400</td>
<td>0</td>
</tr>
<tr>
<td>Isothermal wall jet</td>
<td>6600</td>
<td>0</td>
</tr>
<tr>
<td>H = 120 mm</td>
<td>7600</td>
<td>0</td>
</tr>
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<td>Uniform profile</td>
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<td>Refrigerated wall jet</td>
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<td>H = 120 mm</td>
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<td>Uniform Profile</td>
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<td>Refrigerated wall jet</td>
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<tr>
<td>H = 90 mm</td>
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<td>Refrigerated wall jet</td>
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<td>H = 120 mm</td>
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<td>Uniform profile</td>
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<td>Fully stocked refrigerated case</td>
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<td>All perforations closed</td>
<td>5900</td>
<td>0.21</td>
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<tr>
<td>H = 120 mm</td>
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<td>0.16</td>
</tr>
<tr>
<td>Uniform profile</td>
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</tr>
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<td>Fully stocked refrigerated case</td>
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<td>Half-perforations closed</td>
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3.1 Wall Jet Flow Visualization

3.1.1 Wall Jet Instabilities

The negatively-buoyant wall jet was visualized at a wide range of Reynolds numbers to study the flow structures and features (Figure 3.1). The interaction of a wall jet with the ambient is driven primarily by the interactions at the shear layer interface and the instabilities at the wall. The three different types of shear layer structures were observed for negatively-buoyant wall jets: 1) discrete free vortices (by the growth of the Kevin-Helmholtz instability), 2) mushroom type vortices, and 3) protrusions and indentations. These structures are the primary means of mixing with the ambient. In the following, instantaneous structures will be discussed followed by the time-evolving dynamics. Low Reynolds number (Re ~ 870 – 1500) resulted in the formation of simple, well-ordered and clearly defined discrete free vortices or mushroom type vortices with counter-rotating features. These are similar to the observations of Gogineni and Shih as seen in Figure 1.2.

Figure 3.1: Flow visualization of shear layer structures at different Reynolds numbers
Detachment from the wall was observed to occur at Re = 1500 for isothermal wall jets by Field (2001). In current experiments the negatively-buoyant wall jet was observed to be stable and attached to the wall for all Reynolds numbers tested (870 - 7700). This is attributed to the negative buoyancy effects, which cause the acceleration of flow along the wall and stabilization of the wall jet. At higher Reynolds numbers (Re > 2200), the flow was dominated with mixing structures that were more complex with random protrusions and indentations observed by Field (2001) and as shown in Figure 3.2 for the present study. The scale of these irregular features was observed to grow larger as the jet moves downstream. This was similar to the flow visualization of Gogineni & Shih at Re = 3800 (see Figure 1. 2) and the generally irregular protrusions and indentations observed by Field (see Figure 1. 5). These structures caused the engulfment and entrainment of ambient air into the wall jet. Further downstream, the protrusions and indentations developed into complicated shear layer features.

![Figure 3.2: Schematic of shear layer structures for isothermal and refrigerated wall jets in the Reynolds number range Re = 3500 - 7700](image)

However, complex structures can be seen at Reynolds numbers as low as 1270 with the turbulent inflow conditions (Figure 3.3). At the top of the field of view is the roll-up of a discrete vortex. Immediately downstream a mushroom types vortex is seen detaching from the wall jet into the ambient. Further downstream, a large discrete vortex can be observed at the bottom of the field of view. A number of swirls in the vortex core are clearly visible, compared to the single swirl in the vortex at the top of the field of view where the roll-up has just begun.
In the current experiments the vortex formation was generally observed much closer to the inlet (at downstream locations $x/H = 1.5$), while formation of mushroom type structures were also observed close to the inlet (at downstream locations $x/H = 2$). In previous studies, the formation of discrete free vortices in the low Reynolds number range ($Re \sim 870-1270$), was not observed until more downstream locations ($x/H > 6$) in the smoke flow visualizations of Gogineni & Shih (Figure 1.2). In the DNS studies of Bhattacharjee (2002), the initial discrete vortex formation was observed at $x/H > 8$. The initial formation of mushroom type vortices was not observed until $x/H > 20$. The variation is attributed to the differences in initial turbulence levels. As will be explained later, the initial jet turbulence levels in the current experiments were very high whereas Gogineni & Shih and Bhattacharjee employed laminar inflow condition. The high turbulence levels in the present wall jet at the inlet are expected to significantly alter the onset and growth of the Kevin-Helmholtz and wall instabilities in the flow. Due to the high initial turbulence levels, even at modest Reynolds numbers complex flow pattern was observed in the shear layer. At higher Reynolds numbers ($Re > 2200$), the interface instead indicated turbulent flow (large range of scales and three-dimensionality).

At low Reynolds numbers (and high Richardson numbers), the interface was clearly defined (Figures 3.4 & 3.5). A single instability in the shear layer rolls up into vortices. Figure 3.4 shows two different flow visualizations of Kevin-Helmholtz instability growth into discrete free vortices. The individual pictures in each sequence were taken spaced 0.3 seconds apart and both sequences are qualitatively similar. The development of these vortices is schematically shown in Figure 3.6.a. Sometimes the wall instabilities yield a counter-rotating vortex that combined...
with a forward rotating vortex, resulting in the development of a mushroom type structure (Figure 3.5). Such a development is schematically depicted in Figure 3.6.b. The detachment of a mushroom structure from shear layer was generally found to be quicker than the detachment of the discrete free vortex. The breakdown of mushroom structure in the ambient resulted in complex patterns as seen in Figure 3.5. At higher speeds the flow dynamics are more non-linear. Figure 3.7.a shows the development and roll-up of a discrete vortex at $Re = 1270$. The observed scale of the vortex was found to be larger than the scale of the vortex at $Re = 870$ and the convolutions were generally more complex. The formation of mushroom type structure in the shear layer at the same conditions is presented in Figure 3.7.b, which also indicates increasing size and irregularity as compared to lower speeds.

Figure 3.4: Two separate time sequences showing vortex roll-up for negatively-buoyant wall jet; $Re = 870$, $Ri = 2.69$
Figure 3.5: Flow visualization of vortex breakdown for negatively-buoyant wall jet; Re = 870, Ri = 2.69
Figure 3.6: Schematic of the vortex development in shear layer; a) Discrete free vortex, b) mushroom-type vortex with counter rotating arms

(a) Onset of instability
Initiation of discrete vortex roll-up
Vortex roll-up complete; further development leads to detachment

(b) Onset of instability
Initiation of secondary roll-up
Figure 3.7.a: Flow visualization of vortex roll-up for negatively-buoyant wall jet; Re = 1270, Ri = 1.50
3.2 PIV Results

3.2.1 Isothermal Wall Jet – Reynolds Number Effects.

Mean Velocity Profile Development

The velocity profiles at different locations were obtained by time averaging 500 images in the field of view and further spatial averaging of the cross correlation 2D velocity field. The typical Reynolds number range for the operation of a medium dairy display case is from 3800-8000. The wall jet dynamics, stream wise development and lateral growth were studied in this range of Reynolds numbers for both isothermal and refrigerated conditions.

Figure 3.8 shows the false color composites of the mean streamwise velocity and show the isothermal wall jet growth, for Reynolds numbers 4400, 6600 and 7600. There is a gradual velocity deceleration in the core of
the wall jet as the width increases. The growth and development of the wall jet is approximately independent of the Reynolds number.

Figure 3.8: Color composites for isothermal wall jets; a) Re = 7600, b) Re = 4400 (x location shown in mm)

Figure 3.9 shows the mean stream wise velocity profile development of the wall jet (normalized by the mean velocity at the inlet, $V_{jet}$) for Reynolds numbers 4400, 6600 and 7600. In general, isothermal wall jet development at all these Reynolds numbers is similar. As the wall jet exits the honeycomb, there is an initial settling region near the inlet where the velocity profile adjusts from a characteristic inlet profile to a diffused profile due to the interactions with the wall and the ambient. The settling region usually extends till $x/H = 3$ at which point the further differences in the velocity profile shape are no longer obvious and there is a gradual decrease in peak velocities and spreading of velocity profile due to momentum diffusion. The isothermal wall jet is primarily momentum-driven.
Figure 3.9: Stream wise velocity profiles for isothermal wall jet; a) Re = 4400, b) Re = 6600, c) Re = 7600
Velocity Profile Similarity

The study of the mean streamwise profiles suggests that the wall jet development downstream of $x/H > 3$ was not significantly affected by the changes in Reynolds number and initial velocity profile. Therefore the similarity of velocity profiles for this region at different Reynolds numbers was investigated. Based on past research on wall jets with laminar inflow conditions (Karlsson et al., 1998), the outer scaling (wall jet ‘half-width’) was used as a similarity parameter. The half-width is defined as the wall normal location where $V/V_{\text{max}} = 0.5$, where $V_{\text{max}}$ is the peak velocity at the given streamwise location. Figure 3.10 shows the similarity profiles for the isothermal wall jet at different Reynolds numbers at the streamwise location $x/H = 5$ and at $x/H = 8$. It is evident that the mean streamwise profiles collapsed onto a single curve when wall jet ‘half-width’, $y_{1/2}$ was used as the similarity parameter. Figure 3.11 shows the streamwise velocity profiles in outer scaling at different downstream locations in the fully developed region for $Re = 7600$ and $Re = 4400$. The mean profiles at different downstream locations approximately collapsed onto a single curve for both Reynolds numbers for $x/H > 3$ (though there is some additional widening of velocity profiles as we move downstream). This demonstrates that there is reasonable similarity in isothermal wall jet profiles based on wall jet half-width, $y_{1/2}$ and peak velocity even though the initial turbulence levels are very high ($\sim 5\%$) as will be shown in later sections.
Figure 3.10: Stream wise velocity profiles at different Reynolds numbers for the isothermal wall jet in outer scaling; a) at $x/H = 5$, b) at $x/H = 8$
Figure 3.11: Stream wise velocity profiles in the fully developed region of the isothermal wall jet in outer scaling; a) Re = 4400, b) Re = 7600

Wall Jet Thickness
The preceding discussions on wall jet velocity profile development and similarity analysis have shown that Reynolds number effects did not significantly affect the isothermal wall jet profile development for x/H > 3 and Re > 4400. To examine whether the same is true for jet width, jet thickness based on $\delta$ and also $y_{1/2}$ was studied. Figure 3.12 shows the wall jet thickness growth (based on $\delta$ and $y_{1/2}$) at different downstream locations at Re = 7700 and 4400. Both the wall jet thickness parameters were found to be independent of Reynolds number and were found to vary nearly linearly with downstream location away from the inlet. Karlsson et al (1998) observed a similar linear variation for the growth of wall jet (based on $y_{1/2}$) in their study.
The turbulence intensity profiles for isothermal wall jet at Re = 4400 and 7700 are shown in Figure 3.13. The turbulence intensity was uniform in the core of the isothermal wall jet (about 5%) and peaked at the shear layer (about 15%). The peak at the location y/H = 1 (the interface) is attributed to the high velocity gradients in this region (see Figure 3.9). The peak in the turbulence intensity profiles was found to widen at downstream locations, associated with an increase in turbulence levels in the core of the wall jet. This signifies the increased mixing with the ambient in the shear layer as the flow moves downstream from inlet. The turbulence levels in the wall jet core increased to 10-15% at x/H = 8, which is consistent with the flow visualization which indicated a distribution of the interface structures over a large width at far downstream locations.
Figure 3.13: Turbulence intensity profiles for isothermal wall jet; Re = 4400, b) Re = 7600

3.2.2 Negatively-Buoyant Wall Jet – Richardson Number Effects

*Stream Wise Velocity Profile Development*

Richardson number is the ratio of the buoyancy forces to the viscous forces. A higher Ri means a stronger effect of downward accelerating buoyancy forces for the jet flow (due to higher wall jet density). Therefore, the effects of negative-buoyancy on the wall jet development are herein referred to as the Richardson number effects. Figure 3.14 shows color composite of mean velocity profile intensity for Ri = 0.11, 0.16 and 0.38 respectively. The negatively-buoyant wall jet is found to accelerate as it moves downward particularly near the inlet and more so as Ri increases. This is a marked difference from the color composites for the isothermal wall jet, where it was seen that the momentum diffusion was effective throughout the flow field causing a gradual deceleration in the velocity profile as we move downstream. At downstream location away from the inlet (x/H > 3), the acceleration was less apparent, which is attributed to the momentum diffusion effects dominating the negative-buoyancy effects.
Figure 3.14: Color composites for negatively-buoyant wall jets; a) Re = 7700, Ri = 0.11, b) Re = 6700, Ri = 0.16, c) Re = 4000, Ri = 0.38

The mean stream wise velocity profile development is shown in Figure 3.15 for Ri = 0.11 at Re = 7700 and Ri = 0.38 at Re = 4400. The Richardson number effects are more prominent near the inlet in the region from x/H = 1 to 3 (rapid diffusion in velocity profile, associated with the acceleration due to negative-buoyancy). Further downstream (x/H > 4), momentum forces start to dominate the flow leading to a deceleration of the velocity profiles and a self-similarity distribution. The negative-buoyancy effects were stronger for higher Richardson number. Figure 3.16 shows the mean velocity profiles near the inlet (for low Reynolds numbers) at Re = 870 and Re = 1270.
Since the Richardson number in these cases is very high, very high accelerations were observed near the inlet. Also, the edge of the wall jet (shear layer interface) at $x/H = 1$ is located at $y/H \sim 0.6$ due to the high accelerations caused by negative-buoyancy. As a result, the shear layer instabilities are located much inside the wall jet and hence may cause increased ease of mixing of the ambient with the wall jet further downstream.

![Stream wise velocity profiles](image)

Figure 3.15: Stream wise velocity profiles for negatively-buoyant wall jet; $Ri = 0.11$, $Re = 7700$, b) $Ri = 0.38$, $Re = 4000$
Figure 3.16: Stream wise velocity profiles near the inlet; a) Re = 870, Ri = 0.27, b) Re = 1270, Ri = 1.23

Figure 3.17 illustrates the effect of Richardson number on the mean velocity profiles for isothermal and negatively-buoyant conditions. For Ri = 0.38, the acceleration due to negative-buoyancy caused the peak velocity profiles to increase by 110% from the inlet to x/H = 7. Whereas, for Ri = 0.11, the peak velocity profiles increased by 40% from the inlet to x/H = 7. The region closer to the wall is most affected by the Richardson number effects (since turbulent diffusion has not yet moved into this region) and therefore, the peak velocities in the velocity profiles are located near the wall. The increase in peak velocities near the wall may be of practical advantage in display cases because it enhances the effectiveness of convective heat transfer from the products.
Note on Similarity in Negatively-Buoyant Wall Jets

As the Richardson numbers affects the mean profiles significantly when compared to isothermal profiles, similarity in negatively-buoyant wall jets poses and intriguing possibility. Figure 3.18 shows mean profiles in the fully developed region ($x/H > 3$) normalized by the peak velocity, while the $y$-direction is normalized by $y^{1/2}$ for $Ri = 0.11$ and $Ri = 0.38$. It is observed that the mean profiles in fully developed region collapsed closely onto a single curve for different $x/H$, but one which is significantly different from the isothermal similarity profile (see Figure 3.11). However, the profiles show a gradual variation at different downstream locations as indicated in the figure. Therefore, at modest Richardson numbers, the outer scaling parameter (wall jet half-with, $y^{1/2}$) is a reasonable characteristic dimensional scale for similarity analysis.
Figure 3.18: Mean velocity profiles in similarity scaling; a) Re = 7700, Ri = 0.11, b) Re = 4400, Ri = 0.38

Turbulence Levels
The typically observed turbulence levels in the negatively-buoyant wall jet are shown in Figure 3.19. The initial turbulence levels at x/H = 1 were found to be about 5% and uniform over the core of the wall jet and peak to about 10-15% at y/H = 1 (similar to that seen in isothermal case). However, the turbulence intensity in the wall jet core is found to increase at downstream locations, reaching a peak of nearly 25% at x/H = 7. The peak in the turbulence intensity profiles also is seen to shift towards the wall at downstream locations. This is attributed to the acceleration in the mean velocity profiles causing increased velocity gradients and thus promoting instabilities. The shift is also consistent with the movement of the inflection point in the mean velocity profiles close towards the wall (Figure 3.15), and the fact that the Kevin-Helmholtz type shear layer instability is associated with the velocity
inflection point. Hence the Richardson number effects cause a strong increase and shift in the turbulence peak towards the wall.

Figure 3.19: Turbulence intensity profiles for negatively-buoyant wall jet; Re = 7700, Ri = 0.11

Vorticity Variation

To understand the increase and shift in turbulence peak, the mean vorticity profiles were examined at similar Reynolds numbers. Figure 3.20 shows the time-averaged vorticity variation in the flow field at different downstream locations for isothermal and negatively-buoyant wall jets. A sharp vorticity peak was observed at x/H = 1 in all cases, indicating steep gradients in the shear layer at this location. The peak in vorticity for the negatively-buoyant was located closer to the wall than the isothermal wall jet. This is due to the acceleration in the wall jet core due to negative buoyancy effects. At downstream locations, the peak vorticity was found to be lower indicating diffusion of vorticity due to turbulent mixing. Also, the peak profiles were observed to be wider at downstream location, which is an indication of the increase in scale of the shear layer structures as we move downstream. For locations x/H > 3, the vorticity peak is diffused over a large distance. This signifies that the protrusions and indentations at these locations are fairly large spanning a large area on either side of the interface (y/H = 1), extending into the core of the wall jet and the ambient.
Figure 3.20: Vorticity profiles for wall jet; a) $x/H = 1$, b) $x/H = 2$, c) $x/H = 3$, and d) next page
At downstream locations, the peak in vorticity profiles for isothermal wall jet (even though very small) is located at $y/H \sim 1$, but moves gradually towards the wall for negatively-buoyant wall jet. At $x/H = 7$, the peak in vorticity profile for the negatively-buoyant wall jet is located at $y/H \sim 0.5$, indicating that the location of the steep gradient has moved closer to the wall.

The shift in turbulence intensity and vorticity peaks imply that the Richardson number effects enhance the mixing with the ambient. Therefore, even as the Richardson number affects stabilize the air curtain and favorably affect the heat transfer from the wall (due to acceleration near the wall), they also enhance mixing with the ambient by pulling the instability into the wall jet core.

**Momentum Entrainment**

Figure 3.21 shows the growth of the negatively-buoyant wall jet based on ‘$\delta$’ for different Richardson numbers. The Richardson numbers have caused a reduction in the wall jet thickness due to increase in peak velocities. In addition, there is a significant deviation from the linear growth observed in isothermal wall jet (particularly near the inlet). The wall jet thickness variation was non-linear in the negatively-buoyant case compared to the linear variation in isothermal case. The deviation from linear variation was higher for higher Richardson number. Further downstream the growth rate was linear and similar to the isothermal case, and $\delta$ was approximately independent of Richardson number. Thus the negatively-buoyant wall jet width transitioned from a neck-in region (of narrow width) to a momentum diffusion region. The rapid growth in momentum diffusion region indicates an increased mixing (for $x/H > 5$) for the negatively-buoyant wall jets (which are associated with the shift in turbulence and vorticity peaks towards the wall).
Figure 3.21: Wall jet thickness width for different Richardson numbers

**Thermal Entrainment**

Figure 3.22 shows the variation of thermal entrainment parameter, $\alpha$, with Reynolds number and Richardson number. It should be noted that $Re$ and $Ri$ vary simultaneously in this study; a higher $Re$ implies a lower $Ri$. The results indicate that at lower Reynolds numbers (where the flow was dominated by well ordered vortex formation in shear layer) the thermal entrainment parameter increased with decreasing Reynolds number. For higher $Re$ typical to the operation of display cases ($Re \approx 3800 – 7700$), the thermal entrainment parameter did not vary significantly with Reynolds number since momentum diffusion is independent of $Re$ for this range of $Re$. The changes are attributed to Richardson number effects only. Any changes in this regime are thus attributed to increased mixing caused by Richardson number effects due to the location of the steep velocity gradient moving closer to the wall as established by the above study of velocity profiles, turbulence intensity profiles and vorticity profiles.
For 3800 < Re < 7700, the thermal entrainment parameter variation was similar to the variation observed by Field (2001). The velocity (or Re) was changed in the study of Field (2001) by replacing fan blades of different camber to drive the flow, which may introduce different levels of turbulence in the flow. However, in the present study speed control was obtained by using the same set of fan blades and changing the voltage. Therefore, the exiting turbulence in the flow at different Re was more uniform in the present study.

**Energy Loss Due To Entrainment**

Figure 3.23 shows the variation of non-dimensionalized energy loss, $E^\alpha$, with Reynolds number. This does not include radiation losses and pressure losses. It is seen that the energy loss due to entrainment is lower at lower Reynolds number which is consistent with the variation of $\alpha$ as a function of Re or Ri. Therefore in terms of energy
efficiency, it may advantageous to operate the negatively-buoyant wall jet at low speeds as long as the wall heat transfer requirements are met (i.e. the desired product temperature is maintained in display case applications).

Figure 3.23: Non-dimensionalized energy loss (due to entrainment), $E^*$ as a function of Reynolds number

### 3.2.3 Inlet Width Effects

**Stream Wise Development**

The inlet width effects were studied by comparing the flow of negatively-buoyant wall jet, with 90 mm and 120 mm exit widths. Figure 3.24 shows the color composite showing the mean velocity development for a 90 mm wide negatively-buoyant wall jet. The overall flow development is very similar to that of 120 mm wide negatively-buoyant wall jet i.e. the flow is found to accelerate near the inlet and further downstream, and the acceleration is concentrated in the near wall region. As expected, the higher Richardson number $Ri = 0.25$ results in a stronger acceleration near the inlet compared to $Ri = 0.1$. 
The development of mean velocity profiles at different downstream locations is shown in Figure 3.25. The mean velocity profiles accelerate rapidly (especially for the higher Ri) until \( x/H = 3 \), associated with a transformation of velocity profile from a uniform profile to a nearly linear profile (especially at the high Ri case). Further flow acceleration occurs until \( x/H = 6 \), beyond which the velocity profile becomes approximately self-similar as momentum diffusion forces dominate and lead to flow deceleration. The comparison of mean velocity profiles for different inlet widths and Reynolds numbers at the same Richardson number (Ri = 0.1) is shown in Figure 3.26. As the velocity profiles develop from \( x/H = 1 \) through \( x/H = 7 \), the profiles are not only similar but also show the same rate of acceleration near the wall.
Figure 3.25: Stream wise velocity profiles for negatively-buoyant wall jet (H = 90 mm); a) Re = 5800, Ri = 0.1, b) Re = 4000, Ri = 0.25
Figure 3.26: Stream wise velocity profiles for refrigerated air curtain at Ri = 0.1; a) at x/H = 1, b) at x/H = 2, c) at x/H = 5 & d) next page
Figure 3.26: Stream wise velocity profiles for refrigerated air curtain at Ri = 0.1, d) at x/H = 7

Figure 3.27 shows the mean velocity profiles for different inlet widths having different Richardson numbers at the same Reynolds number, Re = 4000. The non-dimensionalised profiles were found to be similar for location x/H = 1 through x/H = 7. The maximum velocity in the mean velocity profiles with Ri = 0.38 was found to increase by 110 %, against a 70 % increase in peak velocity for Ri = 0.25. However, when the Ri = 0.25 profile was very similar and showed the same acceleration when compared against a velocity profile of comparable Richardson number, \( \text{Ri}_{\text{avg}} = 0.24 \). This particular profile was obtained by averaging the velocity profiles at x/H = 7 for Ri = 0.38 and 0.11 (for H = 120 mm). As a practical note decreasing the inlet width while maintaining the Reynolds number (mass flow rate) results in a lower Richardson number.
Therefore, it can be concluded that the flow development in similarity coordinates in negatively-buoyant wall jets with $Re > 4000$ and $x/H = 1$ to 7 is independent of inlet width, and controlled by Richardson number effects alone. Further downstream momentum diffusion is the dominant factor causing the gradual deceleration in wall jet velocity.

Turbulence Levels

The turbulence levels observed in a negatively-buoyant wall jet with width, $H = 90$ mm are shown in Figure 3.28. The turbulence levels in the wall jet did not show any significant variation due to the reduction in wall jet thickness. As in the 120 mm results the initial turbulence levels were about 5% and showed a gradual increase downstream reaching levels were as high as 25% by $x/H = 8$. Also a shift in turbulence peak towards the wall as observed with $H = 120$ mm is seen and is stronger for higher Richardson number ($Ri = 0.25$). Thus, the turbulence levels and the variation observed here may be considered typical for negatively-buoyant wall jets.
Jet Thickness

The wall jet growth based on momentum parameter, $\delta$, is shown in Figure 3.29. The wall jet thickness development was identical when compared at the same Richardson number with different inlet widths. Therefore, the dimensionless wall jet thickness was not affected by changing the inlet width and was strictly affected by the Richardson number affects alone.
Figure 3.29: Wall jet thickness at different Richardson numbers

**Thermal Entrainment**

The variation of thermal entrainment parameter, \( \alpha \), is shown in Figure 3.30 with Richardson number. Since the velocity profile development above the capture area was similar, the reduction in thermal entrainment (for \( H = 90\text{mm} \)) is attributed to the fact that the dimensional capture area geometry was unaltered. Therefore, the ratio of capture area size to the wall jet thickness at the capture area is higher for \( H = 90 \text{ mm} \) than with \( H = 120 \text{ mm} \). Hence, less spillover may result with reduction in dimensional wall jet width. That explains the lower thermal entrainment values observed when the inlet width is reduced.
Reducing the wall jet inlet width seems to be an affective way of reducing entrainment in display case air curtains. However, it should noted that when the inlet width is reduced below a certain limit, the L/H may be prohibitively high. The growth in the air curtain after momentum diffusion starts dominating the flow (x/H > 8); which in turn may cause the air curtain to entrain a large amount of ambient air. Therefore, there may be a lower limit on reducing the air curtain thickness in display case air curtains.

3.2.4 Inlet Velocity Profile Effects

Stream Wise Velocity Development

The comparison of the uniform profile with the stepped profile at the inlet is shown in Figure 3.31. The color composite for the stepped velocity profile has two regions of different velocities. The inner region close to the wall with higher velocity and the outer region towards the ambient with lower velocity. Note that the Richardson number and Reynolds number for the wall jet with the stepped profile are calculated based on the average velocity at the inlet. The mean velocity development and wall jet growth for the negatively-buoyant wall jet with stepped profile at the inlet is shown in Figure 3.32 for Ri = 0.24 and Ri = 0.32. The velocity in the region close to the wall is seen to increase throughout, while no significant acceleration is observed in the region of low velocity close to the ambient. Significant growth of the wall jet is not observed for this region.
Figure 3.31: False color composite near the inlet; a) Uniform profile, b) Stepped profile
Figure 3.32: Color composites for negatively-buoyant wall jet, stepped profile at the inlet; a) Re = 6100, Ri = 0.24, b) Re = 5200, Ri = 0.32

Figure 3.33 shows the mean velocity development for a negatively-buoyant wall jet having a stepped profiles at inlet for Ri = 0.24 and Ri = 0.32. There are two shear regions in the velocity profile near the inlet. One is between the outer and inner regions of the wall jet core and the other between the outer core of the wall jet and the ambient. Due to the Richardson number effects, the mean velocity development in the shear layers is complete by the downstream location x/H = 3. Beyond this the velocity profiles in the wall jet are identical to the fully developed profiles for a wall jet having an uniform profile at the inlet. One marked difference is the high level of acceleration even at Ri = 0.25. The profile for Ri = 0.25 has acceleration identical to the profile for Ri = 0.32. This is because the inner core of the wall jet is shielded from the influence of the ambient by the interaction of the outer core with the ambient. Hence the inner core continues to stay cold and accelerates. Since the ambient sees a shear layer that has lower gradients compared to those found with the uniform profile of same Reynolds number, the shear layer structures are smaller. Therefore, wall jet mixing with the ambient is slower when compared to the uniform profile wall jet with same Reynolds number. This may explain why significant wall jet growth is not observed in Figure 3.32.
Figure 3.33: Stream wise velocity profile development with stepped profile at the inlet; a) Re = 6100, Ri = 0.24, b) Re = 5200, Ri = 0.32

Turbulence levels

The turbulence levels for the wall jet with stepped profile are shown in Figure 3.34 and are somewhat higher than that for the uniform profile, especially near the exit. The initial turbulence levels in the inner core of higher velocity are about 7% (against 5% in uniform profile) and the outer core has higher turbulence levels of about 20% (against 15%) because of the shear layer interactions on both sides. Further downstream, the turbulence levels in the wall jet increases rapidly because the outer layer has shear interactions on both sides. Turbulence levels at the location x/H = 5 were about 25%. Wall jet with uniform profile was found to have similar turbulence levels at this location.
Figure 3.34: Turbulence intensity profiles for the refrigerated air curtain with stepped profile; $Re = 6200$, $Ri = 0.24$

*Thermal Entrainment*

Figure 3.35 shows the variation of the thermal entrainment parameter with Richardson number. The data points for the different profiles at $Re = 4000$, were marked on both figures. It is seen that the even though stepped profile has a higher Ri, surprisingly lower values of thermal entrainment parameter were observed. This is attributed to the lower gradients in the shear layer where the wall jet interacts with the ambient. Therefore the shear layer structures may be smaller here, resulting in lower mixing of the ambient with the wall jet.

Figure 3.35: Thermal entrainment parameter variation with Richardson number

*Energy Loss Due to Entrainment*

The non-dimensionalized energy loss due to entrainment, $E^*$ is plotted as a function of Reynolds number in Figure 3.36. It is seen that the wall jet with lower width has a lower energy loss. Also, the wall jet with stepped profile at the inlet has a lower energy loss compared to the wall jet with uniform profile at the inlet. In either case,
the energy loss was found to be lower by about 10% when compared to the negatively-buoyant wall jet with an inlet width of 120 mm having a uniform profile at the inlet.

![Non-dimensionalized energy loss variation with Reynolds number](image)

**Figure 3.36**: Non-dimensionalized energy loss variation with Reynolds number

### 3.2.5 Fully Stocked Display Case Configuration – Back Perforation Effects

**Stream Wise Velocity Development**

The mean velocity profile development and air curtain growth in the fully stocked case configuration for the refrigerated dairy case is shown in color composites in Figure 3.37. Figure 3.37.a is for the fully stocked case when all the perforations were closed such that there was no air flow from the back perforations of the display case. Figure 3.37.b and Figure 3.37.c are for the configurations where half-of-the perforations were open and all the perforations were open, respectively. The Richardson number effects cause acceleration in the flow near the inlet. At downstream locations, the flow acceleration is a combination of Richardson number effects and increased mass flow due to the cold air added from the back perforations. After crossing the shelves, the flow accelerated along the product side, due to a combination of Richardson number effects and increased mass flow. A marked difference from the wall jet studies was that a large increase in air curtain thickness was observed especially as the back perforations were opened indicating an increased growth rate of air curtain.
Figure 3.37: Color composites for the fully stocked case; a) all perforations closed (Re = 5900), b) half-of-the perforations closed (Re = 4900), c) all perforations open (Re = 4200)

Figure 3.38 shows the velocity profile development near the inlet for the fully stocked configuration with perforations effects. The mean velocity development is affected by the Richardson number affects causing flow accelerations. Higher accelerations are seen for higher Richardson number, which is expected from the wall jet studies. A product shelf is located at x/H = 2. However, the velocity profile at x/H = 2.5 does not show any significant difference on the product side. Therefore, after passing the shelf the air curtain accelerated in normal fashion. It should be noted that between x/H = 1 to 2.5, there was no significant addition to the air curtain mass from back perforations. Therefore, the accelerations here are entirely due to negative-buoyancy effects.
Figure 3.38: Mean velocity profiles for fully stocked case; a) x/H = 1, b) x/H = 2.5

The mean velocity profiles for the full air curtain with back perforations open is shown in Figure 3.39. The peak velocities in the velocity profile are not at the edges of the boundary layer (i.e. very close to the wall), as seen in wall jet studies. The peak velocities are located away from the wall at y/H ~ 0.3. This is attributed to the wake created behind the shelves. However, in the configuration with all perforations open, the near-wall velocities are higher by 40% when compared to the average velocity at the inlet. Therefore, the air curtain may still be very effective in terms of convective heat transfer on the front face of the product. Also, in the region y/H > 1, the mean velocity profiles in the display case air curtain are stretched and wider than the wall jet configuration, indicating greater turbulent diffusion.
The effects of mass flow addition through back perforation on the air curtain development and growth can be studied by comparing the mean velocity profiles in the fully stocked configuration with the wall jet mean velocity profiles at comparable Reynolds numbers and Richardson numbers. Such a comparison is shown in Figure 3.40 indicating the difference in the wall jet and display case air curtain profiles at locations $x/H = 5$ and $x/H = 8$. It is seen that the velocity profiles are significantly different and the difference between the profiles increases with the downstream location. The difference between the curves accounts for the differences in entrainment and the effect of mass addition through back perforations. Therefore, the rapid air curtain growth observed in Figure 3.37 is not due to entrainment alone, but also due to the back perforation effects.

Figure 3.39: Mean velocity profiles for fully stocked case; a) all perforations closed ($Re = 5900$, $Ri = 0.21$), b) all perforations open ($Re = 4200$, $Ri = 0.46$)
Energy Loss Due to Entrainment

Estimation of energy loss due to entrainment in the fully stocked display case is based on the assumption that the temperature of air entering the back perforations and the temperature of the air curtain at the inlet is the same. This is a reasonable assumption because once the air passes through the evaporator coil there is no heat addition and negligible heat loss (because of insulation) in the circuit. Therefore, the air entering the back perforations and the air at the curtain inlet have nearly the same temperature.

Figure 3.41 shows the variation of thermal entrainment parameter and non-dimensional energy loss due to entrainment with the Reynolds number measured at the inlet. Note, lower Reynolds numbers are observed with perforations open for a given cfm because a part of the total mass flow rate is delivered through the back perforations. The perforations open condition yields generally lower thermal entrainment due to the continuous addition of cold air along the curtain length thus ensuring a colder low-mixing region just above the capture area i.e,
the effect of flow from perforations causes the air curtain thickness to grow in such a way that the mixing region with warm air is located at the edge of the return grille. Therefore, at higher Reynolds numbers the negative buoyancy effect of the back flow is reduced yielding increased horizontal momentum such that the cold flow more strongly fills the core of the air curtain that is ingested at the capture area. Thus the back perforation flow at high Reynolds numbers (typical operating conditions) was well designed for the display case capture area. Changing the capture area would thus probably require changes to the cfm or mass flow rate allowed through back perforations such that optimum amount of cold air is captured maintaining lower entrainment values.

Figure 3.41: Thermal entrainment variation in fully stocked case; a) thermal entrainment parameter, b) non-dimensional energy loss
It should be noted that the energy loss due to entrainment shown in Figure 3.41, \( E^*_{\text{inlet}} \) (equal to \( \alpha R_e^{\text{inlet}} \)) versus \( R_e^{\text{inlet}} \), is based on the mass flow rate (or Reynolds number) at the inlet and does not represent the actual energy loss due to entrainment in the refrigerated display case. The total energy loss variation can be depicted if the mass flow rate through the back perforations is known. In the present study, the mass flow rate through back perforations could not be obtained.

If the ratio of mass flow rate at the inlet to the mass flow rate through back perforations is denoted by \( \beta \), it is possible to include the back perforations effects in the Reynolds number by defining a ‘corrected Reynolds number’ as,

\[
R_{e_{\text{corr}}} = R_e (1+\beta)
\]

wherein \( R_e \) is the Reynolds number of the air curtain defined on inlet properties. Therefore, the actual energy loss due to entrainment in the refrigerated display case is:

\[
E^* = R_{e_{\text{corr}}} \alpha = E^*_{\text{inlet}} (1+\beta)
\]

Therefore the variation of total energy loss due to ambient with total mass flow rate in the display case circuit is depicted in the plot of \( E^* \) versus \( R_{e_{\text{corr}}} \), which is equivalent to plot of \( E^*_{\text{inlet}} (1+\beta) \) versus \( R_e (1+\beta) \). This can be obtained by scaling both axes in Figure 3.41 by a factor \( (1+\beta) \).
Chapter 4. Summary

4.1 Conclusions

In the present investigation, the display case in the wall jet configuration was studied in a wide range of Reynolds numbers and Richardson numbers. The effects of changing the inlet width, velocity profile were also studied. Further, the display case was studied in the fully stocked configuration to investigate the back perforation effects. In terms of measurements, mean and turbulent profiles were obtained along with flow visualization. The wall jet thickness was documented to study the effect of different parameters on the momentum entrainment. Further, the variation of thermal entrainment and dimensionless energy loss due to entrainment were studied to quantify the entrainment losses.

The flow visualization results indicated that the refrigerated wall jets were attached to the wall even at \( \text{Re} = 870 \). In the low Reynolds number range (\( \text{Re} \approx 870 \) to 1270), the shear layer mixing was either due to discrete free vortices or mushroom type shear layer features with counter-rotating arms. Due to high turbulence levels at the inlet, vortex formation was found to occur close to the inlet at \( x/H = 1.5 \) to 2. For \( \text{Re} > 3800 \), the shear layer features were not well defined and yielded irregular protrusions and indentations causing engulfment of the wall jet and hence mixing with the ambient.

For mean velocity results, isothermal wall jets were found to be driven by momentum forces alone leading to a reduction in peak velocities downstream and continuously diffusing velocity profile. The associated variation in wall jet thickness in terms of \( y_{1/2} \) as well as \( \delta \) was linear. Similarity of velocity profiles was observed (in outer scaling) not only at different downstream locations but also for different Reynolds numbers. For negatively-buoyant wall jets Richardson number effects were dominant and caused flow acceleration alone till \( x/H \) of about 7. The observed wall jet width was narrower, in particular near the inlet and narrower for higher Richardson number. As in the isothermal wall jet, the turbulence levels were uniform within the wall jet core and peaked at the edge of the shear layer. The turbulence levels increased downstream, while the peak in turbulence intensity moved closer to the wall due to Richardson number effects. This is due to the steep velocity gradients (and hence the instability) moving closer to the wall due to the Richardson number effects.

While the accelerations caused by Richardson number effects cause higher velocities towards the wall side, which could be useful for applications where heat transfer at the wall is the key factor. Also, negative buoyancy increases wall jet stability at lower Reynolds numbers in terms of stability. However, Richardson number effects also enhance mixing of the wall jet with the ambient and increase thermal entrainment. Therefore, Richardson number effects are a ‘necessary evil’.

The thermal entrainment was found to be high at low Reynolds numbers and reduced with increasing \( \text{Re} \) until \( \text{Re} \approx 2000 \). For higher \( \text{Re} \), especially in the range of operation of display cases (\( \text{Re} \approx 3500 \) to 7700) the thermal entrainment did not vary significantly with changes in Reynolds number. Within this range of \( \text{Re} \), thermal entrainment increases with increasing Richardson number due to the point of inflexion moving into the wall jet and closer to the wall. The energy loss due to entrainment increases approximately linearly with increasing Reynolds number (or the total mass flow rate). Therefore it may be advantageous to operate the wall jets at lower \( \text{Re} \) (or mass...
flow rate), however the lower limit on operating Re may be defined by the heat transfer requirements at the wall and in particular product temperature requirements in refrigerated display cases.

Changing the inlet width itself did not change the mean velocity development of the negatively-buoyant wall jets. The differences in flow development were due to the differences in Richardson number (when Reynolds number is maintained as constant) associated the change in width of the inlet. The wall jet growth in the case of a stepped profile was slower because of lower gradients on the ambient side. Otherwise, the flow development with the reduced width and the stepped profile was very similar to the base uniform profile case.

The energy loss is lower with the narrower inlet width due to less spillover at the capture area (as dimensional capture area is kept constant in this study when the inlet width was reduced). Since the negatively-buoyant wall jet mean velocity development, growth and stability is not affected by size of the inlet width, reducing the inlet width is a means of reducing energy loss. If the inlet width is reduced maintaining inlet velocity, the wall jet operates at a lower mass flow rate and hence energy loss is lower. Reducing the inlet width maintaining the Reynolds number implies higher velocities, which can be advantageous in applications where wall heat transfer is crucial.

Further, the dimensionless energy loss is lower for the stepped profile against the uniform profile because of the lower gradients at the ambient. This leads to smaller shear layer structures and a slower growth of shear layer structures and hence lower mixing of the wall jet with the ambient.

For the fully stocked display case with perforations open raid growth of the air curtain and high acceleration near the shelves and rapid growth of the air curtain were observed. This is due to the cold air addition to the air curtain from back perforations. Due to the effect of back perforations, at higher Reynolds numbers the flow filled the return grille favorably with colder air such that more cold air was captured while warm air in the mixing region was spilled over. In display case air curtains the cfm, fraction of mass flow through back perforations and the capture area configuration have to be optimized to capture more cold air. The energy loss due to entrainment in the fully stocked display case increased with increasing total mass flow rate. Therefore it may be advantageous to operate the display case air curtain at the low volume flow rates. However, the lowest possible volume flow rate (or Reynolds number based on inlet properties) is limited by the product temperature requirements. However, for any given flow reducing the air curtain inlet width optimally and/or using a stepped velocity profile at the inlet instead of a uniform profile can further reduce rate the energy loss.

4.2 Recommendations

The investigation has established the air curtain behavior, thermal entrainment variation at various flow conditions. In general, energy losses reduced with lower cfm. However, the limits on operating conditions are prescribed by the maximum product temperature requirements. Therefore, a study in which the product temperature is monitored in the fully stocked display case while the different conditions such as the inlet width, velocity profile, etc. are changed is suggested. Such a study may establish the optimum operating conditions for the air curtains in refrigerated display cases.

The total energy loss in the refrigerated display case in the fully stocked configuration may be obtained if $\beta$ (ratio of mass flow rate through inlet to the mass flow rate through back perforations) is known (but was not
investigated herein). Therefore understanding the variation of $\beta$ at different Reynolds numbers and with different levels of perforation is required to accurately estimate the total energy loss in the refrigerated display case when back perforations are open.

Flow control may be another avenue for further research. Currently, supermarket display cases are run at higher velocities throughout even when the entrainment is less at certain times of the day. Potentially, a closed loop control based on variations of cfm may be established to reduce energy levels for a given configuration and condition. The product temperature can be the actuator and fan speed as the variable. This would allow the air curtain to be operated at levels just above the product temperature requirements and hence lead to reduced energy losses over the long-term operation.
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