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Past, Present, and Future Research Toward Air Curtain Performance Optimization

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ABSTRACT

This paper presents a comprehensive discussion on past, present, and future research focused on display case air curtain performance characterization and optimization. The past research mostly relies on simplified analytical solutions for jets. The present approach takes a more comprehensive step toward understanding and quantification of all major parameters that affect the air curtain flow field by utilization of modern analytical/computational and experimental techniques. The goal of future work is to optimize air curtain performance as a function of the major design parameters by adoption a systematic approach. This approach would be independent of any particular display case design specifics and should be useful to all display case manufacturers.

PAST RESEARCH

Air curtains for open vertical refrigerated display cases are initiated at a supply cold air grille called the discharge air grille (DAG) that is basically a slot jet. Professor Ronald H. Howell and his associates have pioneered numerous and significant studies on air curtains. Initially, they investigated the transfer of heat and moisture through the plane of an air curtain (Howell et al. 1976). One of their most important findings was the direct proportionality of heat transfer across an air curtain to the discharge air velocity (DAV). Later studies by Howell and Shiabata (1980) revealed that the ratio of the opening height (H) to the DAG width (w) and also the jet velocity (V) affect the "performance" of air curtains. This research was further extended to the turbulent flow formulation of a free jet. It examined the effects of the turbulence intensity at the delivery jet or DAG on the turbulence development process along the air curtain as it moves downward. Howell et al. (1983) showed that higher turbulence intensity (I_{jet}) at the DAG or jet accelerates the widening of the jet, causing a higher heat transfer across the air curtain. Their formulation was based on the incompressible boundary layer theory applied to shear layers. The analysis relied on the eddy viscosity model for turbulence flows. Howell and Adams (1991) extended their analysis to the field. They have shown that about 75% of the refrigeration load in an open vertical display case is a result of the warm air entrainment across the air curtain.

Although the research above used simplistic formulations for air curtains, its importance lies in identifying most parameters that impact "any" air curtain performance. For instance, turbulence intensity at the DAG (I_{jet}) as a boundary condition is a measure of mixing enhancement and the air curtain width. The more distance that the air curtain travels *(H)* also provides more opportunities for the air curtain to widen. The width of air curtains *(w)* provides the initial length for the flow to move laterally, which can enhance widening of the jet. The velocity at the jet *(V)* specifies how much kinetic energy is available at the boundary to be implemented toward the initiation and amplification of turbulence kinetic energy within the air curtain. These parameters are crucial to understanding air curtain performance. In terms of nondimensional quantities, these parameters can be grouped as (H/w), Reynolds number (Vw/ν), and (I_{jet}/V). However, a free jet model is not quite applicable to an air curtain because of the presence of a return air grille (RAG), the asymmetrical nature of display cases, non-aligned supply and return air passages, and usually complex geometry before the exit plane of the DAG that can affect the initial velocity profile at the DAG. Furthermore, the eddy viscosity model requires a mixing length model that is based on the definition of a boundary layer "edge." This edge

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is defined arbitrarily and its location significantly affects the turbulence viscosity and the extent of mixing. Therefore, it can be concluded that although the earlier works of Howell et al. (1976-1991) provided information regarding the major parameters impacting the air curtain performance, but a more sophisticated model is required to "quantify" the dependency of the air curtain performance on the aforementioned parameters.

Stribling et al. (1999) made an attempt to combine computational fluid dynamics (CFD) and experimental results to study the velocity and turbulence in a display case. In their CFD model they utilized a two-equation turbulence model that is better suited for free jet research. This model does not utilize the boundary layer theory and therefore does not require a definition for the boundary layer "edge." Their research indicated a good qualitative agreement but demonstrates some quantitative discrepancies between the experimental and computational results.

Further application of CFD codes to air curtains has been inconclusive due to nonmatching results between two CFD codes (Cortella and D'Agaro 2002). They also found discrepancies among turbulence models within the same computer program. They recommended further investigation to identify the source of the inconsistencies. One should realize that CFD provides a numerical solution to the conservation of mass, momentum, and energy equations, commonly known as the Navier-Stokes (NS) equations. It is mathematically known that there is no unique solution for these equations. So it is quite possible that a careless implementation of a boundary condition (from a user or programmer) could propagate and yield inconsistent results. Above research may have benefited from addressing a simpler problem and then gradually introducing complexities and comparing inconsistencies.

Combining experimental and analytical methodologies in understanding air curtains dates back to the 1960s. Early works of Hetsroni et al. (1963) and Hetsroni and Hall (1963) were based on the laminar formulation of the boundary layer equations with body forces to study buoyancy effects. The analytical approach provided a correlation among nondimensional groups, such as Reynolds, Nusselt, Grashoff, and Prandtl numbers. Then experimental methods were used to curve-fit data and quantify the amount of air curtain heat transfer. It is evident that although the amount of heat transfer could be estimated, no detailed information could have been obtained from this approach. A more modern analytical approach with the same basic goal, which took advantage of sophisticated tools such as CFD, was adopted by Axell and Fahlen (2002, 2003). Their research resulted in development of a correlation for evaluating the Nusselt number for an air curtain and evaluation of heat transfer and cooling load thereafter.

The effect of the Richardson and Reynolds numbers on the shape of the streamlines representing the entrained air at the DAG has been studied by Field et al. (2002). This work was quite valuable because it quantified the effects of the Richardson number, $Ri = Gr/Re^2$ (ratio of Grashoff to the square of Reynolds number), on the entrainment of ambient air into the cold air jet. The buoyancy effects that are represented by the Grashoff number demonstrate a controlling role on the entrainment. It was found that, for a Reynolds number based on the DAG width of about 100, the buoyancy forces become significant and must be taken into consideration. Creating an air curtain at this very low Reynolds number requires either a rather small opening or low DAG velocity and may bring about issues of practicality. Furthermore, variations in entrainment may not translate directly into infiltration of warm air into the display case.

PRESENT RESEARCH

The body of reviewed previous research work reviewed was focused on an attempt to understanding air curtain behavior and its controlling parameters. Most of the recent works intend to use modern techniques such as CFD and experimental methods to better understand and quantify the behavior of air curtains. The application of CFD methods by itself could not be totally relied upon for the reason of existence of multiple solutions for the same problem. On the other hand, modern experimental methods are too time-consuming and expensive and they require a great deal of know-how. The best solution methodology appears to depend on an effective and careful combination of both technologies. Navaz et al. (2002) have demonstrated that a marriage between the digital particle image velocimetry (DPIV) experimental technique and CFD simulation can be quite effective. The DPIV can calibrate the numerical technique after which the CFD code can be used for parametric studies. They have shown that this hybrid approach can effectively produce curve fits similar to previous works that can be useful for engineering calculations for heat transfer and entrainment rate.

Furthermore, the wheel should not stop at just "engineering calculations." There is a need to identify, quantify, and optimize all the variables that can affect the air curtain performance. Recent works of Navaz et al. (2003, 2004) take a more modern perspective of those issues that have previously been pointed out by Howell on entrainment rate as a function of Reynolds number and turbulence intensity at the DAG.

It was found that the Reynolds number based on the DAG width, the shape of the velocity profile, and turbulence intensity at the DAG, the length of the opening (vertical distance between DAG and RAG), and angle of throw will affect the entrainment rate. Based on simulation results, it is concluded that the turbulence level observed at the back panel flow inlet (if any) does not contribute much to the overall entrainment. To demonstrate the importance of the DAG design, the original DAG geometry in a specific display case at the DAG was varied. The original DAG geometry of this case generated a two-peak velocity profile with relatively high turbulence intensity. To eliminate this double-peak velocity profile, the vertical surface in the original design was initially replaced by a 20°—and later 57°—slanted surfaces, postulating that a

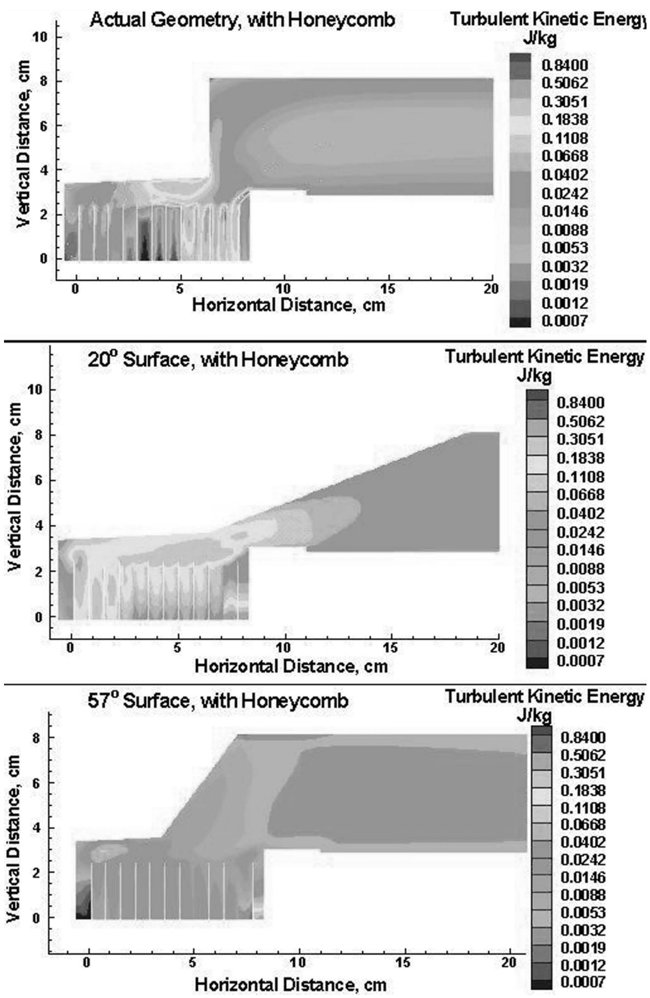

Figure 1 *Turbulent kinetic energy contours for a variety of geometries at the DAG region.*

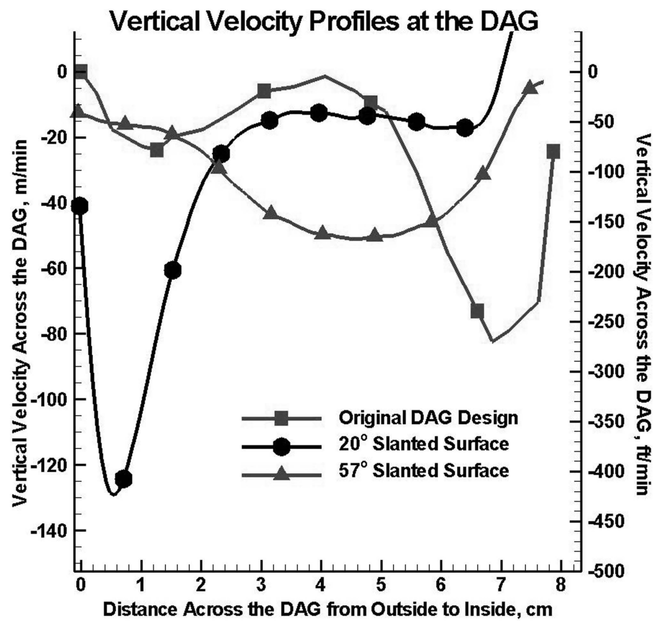

Figure 2 *Vertical velocity profile at the DAG for all three scenarios.*

more gradual change in the direction of flow would lessen turbulence at the DAG. The first angle represents the original design, and the second angle was suggested by CFD results. Turbulent kinetic energy contours for the original DAG geometry (actual geometry), the 20° slanted surface design, and a 57° slanted surface design are shown in Figure 1. After many simulations with different angles for this surface, it became clear that the 57° with a wider throat provides the least turbulence intensity at the DAG for this particular case.

In Figure 2, the velocity profile at the DAG exit for each geometry is shown and the two-peak profile of the original case is clearly seen. These two peaks cause a shear between two layers of fluid that can trigger mixing. The 20° slanted surface profile seems to have a pronounced peak toward the outside of the case with another small peak to the right. It appears that this case may be less effective than the original design. However, as the angle is changed to 57° and the flow passage area at the throat is widened, significant improvement with respect to the shape of the velocity profile is observed. Figure 2 shows a skewed parabolic profile with the peak shifted toward the inside of the display case. When the three velocity profiles were imposed as a boundary condition on a display case with a total CFM of 750 (Reynolds number based on DAG width = 3400), it is observed that the 57° scenario yields the minimum entrainment for every turbulence intensity imposed at the DAG. Figure 3 depicts the entrainment rate for all three cases as a function of turbulence intensity at the DAG. Also, the entrainment worsens for the 20° slanted surface design due to the shape of the velocity profile at the DAG.

Furthermore, when the field turbulent kinetic energy contours are examined in Figure 4, it becomes clear that for the 57° scenario, less turbulent kinetic energy develops in the outside field, therefore reducing the amount of entrainment. So, we may conclude that the shape of the velocity profile at the DAG is of great importance and can only be altered by changing the DAG duct geometry.

Another conclusion in the most recent work also indicates that there had been a misconception regarding the entrainment rate in the sense that it has always been associated with the infiltration rate. Parametric studies (Navaz et al. 2003, 2004) indicated that one should always make a distinction between the two. Increased entrainment does not necessarily mean increased infiltration because only a portion of the

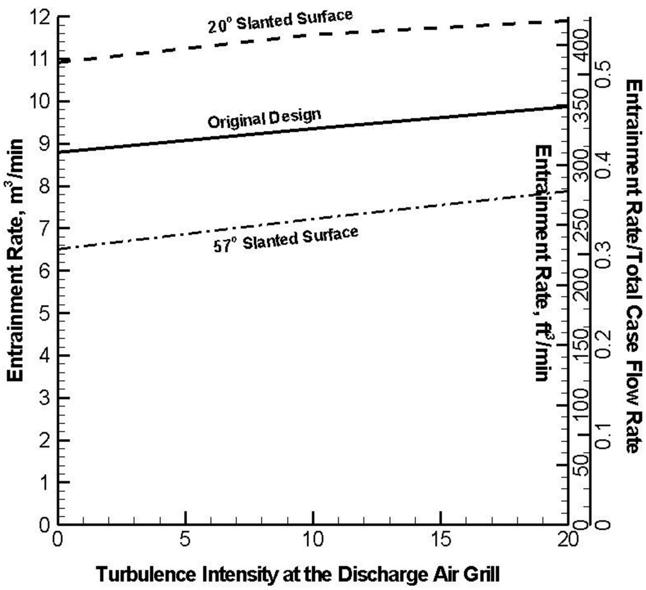

Figure 3 Entrainment rate as a function of turbulence intensity at the DAG for different DAG design scenarios at the optimized Re = 3400.

entrained air is infiltrated into the display case through the RAG, depending on the extent of mixing. A 100% ideal air curtain may entrain air but will not allow any entrained air to infiltrate into the display case. Therefore, a nondimensional quantity that defines the ratio of the infiltrated air to the entrained air (or total volumetric flow rate of the case) is identified. It is this ratio that needs to be minimized for a high-performance air curtain.

FUTURE RESEARCH

We intend to develop a general description of the amount of infiltrated warm air into the display case as a function of flow parameters. By using similitude and nondimensional analysis, it can be concluded that

$$R = f\left(\mathrm{Re}_w, H/w, L/w, \alpha, \beta, \frac{T}{T_{surr}}, I\right), \quad (1)$$

where

R = $\dot{m}_{Infiltrated}/\dot{m}_{Total\ Case}$,

$\dot{m}$ = mass flow rate,

$\mathrm{Re}_w = \overline{V}w/v$ = Reynolds number based on the DAG width and with being the average velocity at the DAG,

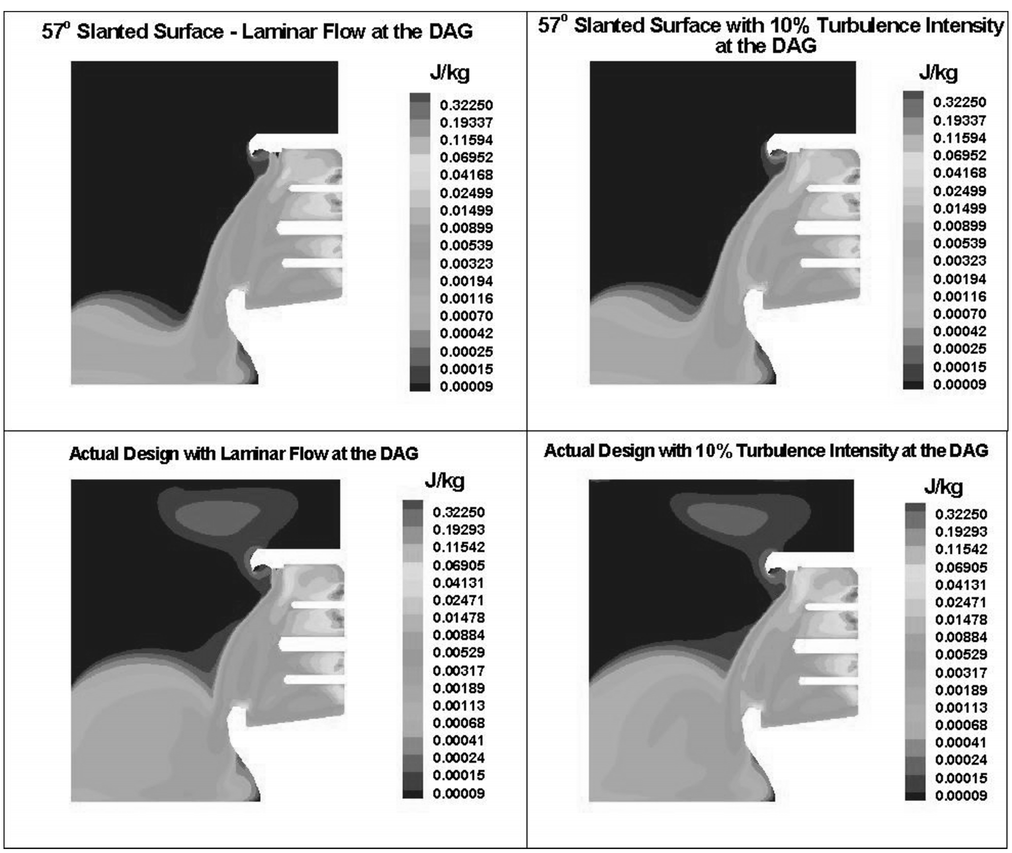

Figure 4 Turbulence intensity contours for the proposed and actual cases with laminar and 10% turbulence intensity being imposed at the DAG.

H = normal vertical distance from the DAG to the RAG,

w = DAG and RAG widths,

L = DAG and RAG lengths,

T = absolute average temperature at the DAG,

T_{surr} = absolute room temperature,

I = turbulence intensity at the DAG

α = angle between the line that connects the centers of the DAG and RAG and the vertical direction, offset angle when the RAG is shifted laterally, and

β = throw angle, the angle between the surface normal to the DAG and RAG.

Previous research indicated that for Reynolds numbers (based on DAG width) above 100 (Field et al. 2002), the temperature difference does not affect the entrainment and infiltration rates significantly. So, Equation 1 can be rewritten as

$$R = f(\mathrm{Re}_w, H/w, L/w, \alpha, \beta, I)\ . \qquad (2)$$

It is evident that the design of a modular display case is necessary to perform all required parametric studies concerning Equation 2.

A schematic of this test air curtain is shown in Figure 5. A combination of DPIV and CFD experimental techniques will be used for the two following scenarios:

1. A modular display case composed of only a DAG and RAG. The position of the RAG can be varied with respect to the DAG. In this geometry, the room air is allowed to mix with the incoming air from the air curtain along the length of the DAG (H). The domain is bounded by two surfaces on the width of the DAG and RAG. This simple yet revealing setup would allow us to understand the behavior of the flow in relation to the parameters discussed above without the added complexity of an enclosure that is characteristic of a display case.
2. The modular air curtain system will be modified to represent an enclosure similar to a display case so that the air can be entrained from one side only. This configuration will allow us to extend our fundamental understanding acquired in step 1 (above) toward the more complex flow configurations found in real display cases.

The quantity R in Equation 1 will be obtained by calculating the entrainment rate from DPIV and CFD results. The infiltration rate will be directly measured by injecting a known amount of a tracer gas (such as carbon dioxide, CO_2) into the incoming flow and measuring its infiltrated amount in the RAG. This method is expected to be far more accurate than the enthalpy method used previously (Navaz et al. 2004). The RAG is exhausted to the outside and fresh air will be brought into the DAG continuously. A gas analyzer will measure the amount of the tracer gas in each stream.

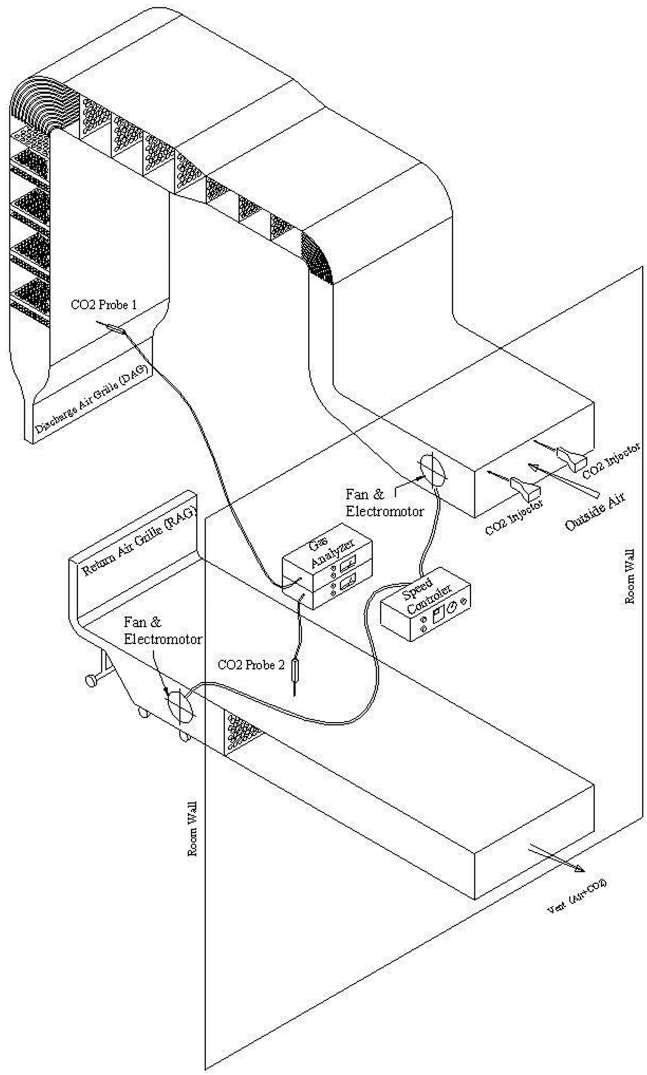

Figure 5 *Schematic of the experimental air curtain setup.*

CONCLUSION

A systematic approach will be developed to map the optimum performance of air curtains under different design criteria. The method also will provide a means of modifying the existing display cases for better performance with minimum possible changes. When the coordinate of operating conditions of a display case is located on these graphs, one would know how an "optimum" condition, i.e., minimum R, can be achieved.

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NOMENCLATURE

CFD	=	computational fluid dynamics
CFM	=	cubic feet per minute (ft^3/min)
DAG	=	discharge air grille
DAV	=	discharge air velocity
DPIV	=	digital particle image velocimetry
H	=	vertical distance between the DAG and RAG planes
I	=	turbulence intensity
L	=	length of the DAG or RAG
$\dot{m}$	=	mass flow rate
R	=	ratio of infiltrated to entrained mass flow rates
RAG	=	return air grille
Re	=	Reynolds number
T	=	absolute temperature
V or $\overline{V}$	=	mean velocity
w	=	DAG or RAG width

Greek Symbols

α	=	angle between the line that connects the centers of the DAG and RAG and the vertical direction, offset angle when the RAG is shifted laterally
β	=	throw angle, the angle between the surface normals to the DAG and RAG
ν	=	viscosity

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