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Energy Transfer Based Test Method Development and Evaluation of Horizontal Air Flow Recirculatory Air Curtain Efficiencies

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ABSTRACT

Air curtains are often used to reduce the energy transfer across high traffic doorways of cold storage facilities and cold food sections of warehouse type supermarkets. A test method and facility were developed to quantify and visualize the performance of air curtains covering a doorway between a simulated cold storage building and the simulated surrounding ambient conditions. This method was used to determine the effectiveness of horizontal air flow recirculatory air curtains compared to an open doorway. Tracer gas methods have been used in the past to determine losses but this method is designed to measure the actual energy flow through the doorway. Two environmental calorimeters were configured side-by-side with a 2.1m by 2.1m doorway separating high temperature and low temperature spaces. The higher temperature calorimeter contained PID controlled heaters and steam generators to maintain temperature and humidity conditions. The power input to these and all other electrical devices as well as the power loss through the calorimeter walls were measured. The difference at steady state between these values yields the energy transfer through the doorway. The lower temperature side contained a cooling coil located in a wind tunnel and PID controlled trim heaters to finely control the temperature. The coolant flow rate and temperature difference across the coil were measured along with the power consumption of all electrical devices and heat transfer through the walls. The difference between these values yields the heat transfer through the doorway which was used as a check for the calculation from the higher temperature calorimeter. The heat transfer was first measured through the open doorway with no air curtain with the warm side controlled to 24ºC and 60% relative humidity and the cold side controlled to 4ºC. The horizontal air flow, recirculatory air curtain was then installed, optimized, and tested at the same ambient conditions. The air curtain reduced the heat transfer between the calorimeters from 35.7kW to 10.3kW yielding an effectiveness of 71%. There is clear visual evidence that was measured by air thermocouple grids and shown in temperature gradient plots which exemplifies the effectiveness of the air curtain at creating a barrier between the warm and cold sides. The flow of warm air through the top of the doorway and the return flow of cold air through the bottom of the doorway with no air curtain was clearly visible and the temperature gradients became diagonal to horizontal. Upon air curtain activation, the temperature gradients became vertical showing that there was little energy transfer from side to side. In addition to energy savings, there is a comparable benefit in the reduction of humidity transfer which helps to prevent icing and condensation on products and on the floor. The reduction in humidity transfer also helps to prevent frosting of the cooling coils which results in fewer defrost cycles being necessary. Reduced frosting of cooling coils improves performance and further reduces energy consumption.

1. INTRODUCTION

Air curtains are used in many applications to reduce the infiltration of heat and humidity between warm and cold spaces. A test method and facility were developed to directly measure the sensible and latent heat transfer through an opening with and without an air curtain running to determine the air curtain effectiveness. There have been several tests done in recent decades to determine the effectiveness of different air curtain configurations at various temperatures, however, most if not all have used a tracer gas decay method. Table 1 shows a summary of previously reported air curtain effectiveness values for in cooler applications using the tracer gas decay method for various air curtain types.

Table 1: Previously reported effectiveness values

Air Curtain Type	Effectiveness	Test Method	Source
Vertical non-recirculatory	79% +/-3 42% +/- 15 68-83%	Tracer Gas Decay	Pham and Oliver (1983) Downing and Meffert (1993) Longdill and Wyborn (1979)
Horizontal non-recirculatory	42%	Tracer Gas Decay	Valkeapaa et al. (2005)
Horizontal recirculatory	76% +/- 3 82%	Tracer Gas Decay	Pham and Oliver (1983) Longdill and Wyborn (1979)

No previous data was found where it was attempted to measure the actual energy and moisture transfer with and without an air curtain in operation. In order to accomplish this, a horizontal air flow recirculatory air curtain was placed between two calorimeters where all energy sources and sinks were measured. The calorimeters were chosen to be large enough to be representative of typical industry applications. Tests were run to measure the energy transfer through an opening with and without the air curtain in operation. The effectiveness is then determined by subtracting from 1 the ratio of the measured energy loss at the doorway with the air curtain active to the measured energy loss at the doorway with no air curtain in place.

2. EXPERIMENTAL SETUP

A calorimeter capable of producing a wide range of environmental conditions was used to evaluate the performance of the horizontal air flow recirculatory air curtain. An insulated wall with a 2.1m by 2.1m opening was centered between the air handling equipment in each calorimeter. Figure 1 shows a schematic of the environmental chambers and the components contained inside.

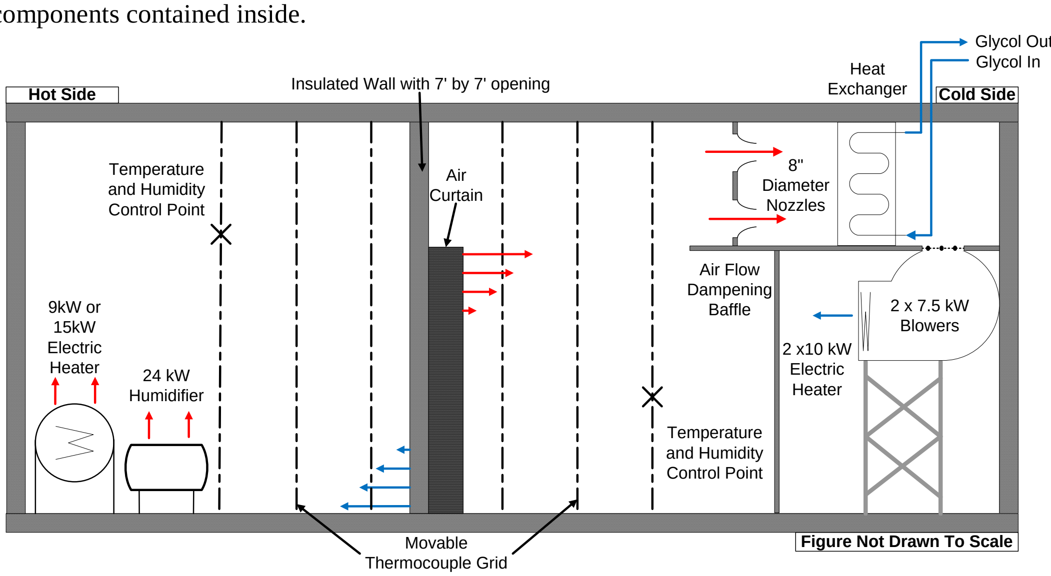

Figure 1: Environmental chamber schematic

Cooling was supplied to the cold side heat exchanger by circulating ethylene glycol from two air cooled chillers each rated at 60kW of cooling capacity. For the majority of the tests only one chiller was used and the cooling load on the chamber was varied as necessary through modification of flow rates, reduction in the number of chiller compressors in use, and the addition of heat to the chiller loop. Airflow dampening baffles were placed in front of the blowers on the cold side to prevent air flow in the direction of the natural air flow though the opening. The cooling load on the chamber was calculated based on the measurement of refrigerant loop mass flow rate, the temperature change between the cold side heat exchanger inlet and outlet, and the specific heat of glycol. The power input to the cold side was measured using watt transducers. This includes power going to the blowers, fans, heaters, lights, and VFDs. The power going to the air curtain was measured by a separate watt transducer. The chamber was calibrated and all heat transfer through the walls, ceiling, and floor was measured using thermocouples on all inner and outer wall, floor, and ceiling surfaces.

Heating was provided to the warm side by a 9kW and/or a 15kW heater controlled by a variable frequency drive which was controlled to maintain the chamber temperature using the data acquisition and control program Agilent VEE Pro. Humidification was achieved through the use of a steam generator which was also controlled by a variable frequency drive and data logger. Several fans were placed near the air handling, temperature, and humidity control equipment and were arranged to blow parallel to the opening to encourage air circulation on the cold and warm side of the chamber while not interfering with energy and air flow from the warm side to the cold side through the opening. The fans were positioned to help ensure the effect of an “infinite” and relatively uniform thermal reservoir on each side. All heating loads on the warm side were measured using watt transducers. These measured loads included the heaters, steam generator, VFDs, fans, and lights.

Airflow measurements were taken at the 2.1m x 2.1m opening with all air handling equipment and fans running but with no temperature difference between the sides to ensure that the air flow created by these devices had a negligible effect on the air flow from side to side. All air flow measurements were well below the 0.25m/s limit commonly found in testing standards.

The acquisition of planar air temperature profile data across a 5.25m long by 3.75m high cross section perpendicular to the opening was accomplished using a type T thermocouple grid configured to be mobile along the length and width of the chamber while being able to maintain planar form. The grid was composed of 6 vertical columns each having 8 thermocouples attached at 0.46m vertical intervals from top to bottom for a total of 48 thermocouples. A spacing of 0.9m was maintained between each vertical column for every test. Three temperature plane locations were used for most tests to give a 3D representation of the temperature profile.

A data logging program was developed in Agilent VEE Pro to monitor test conditions in real time. The output of the program provides sufficient information to formulate a comprehensive energy balance on the chamber including all heating and cooling loads. It also includes the planar temperature distribution, cold and warm side dew point readings, and differential pressure readings across the cold side heat exchanger and air flow measuring nozzles. All data was exported to Microsoft Excel for further analysis and visualization. Contour plots were developed for planar temperature profile visualization with data playback capability to enable the user to quickly visualize how any planar temperature profile data set changes over time. A data sampling rate of approximately one data set per nine seconds was achieved with the primary data logging program. A secondary program was developed for monitoring only the planar temperature profile which had a data sampling rate of approximately one data set per three seconds. The faster sampling program was used to develop time lapse videos of the temperature distribution when the air curtain was turned on and off.

3. TESTING CONDITIONS AND PROCEDURE

Three phases of testing were performed to ascertain the performance of the Air Curtain. Table 2 gives a basic description of the chamber setup for each test. A full chamber open door calibration test was run to determine the heat loss through the chamber walls, ceiling, and floor not including the central wall. This test consisted of using only box fans and blowers to heat the entire chamber to steady state conditions. Using inner and outer wall, floor, and ceiling temperature sensor readings combined with watt transducer readings for the total heat input into the chamber and the dimensions for the chamber, a relationship was developed to calculate the heat loss or gain through the walls of the warm and cold sides of the chamber as a function of measured temperature differences in future tests. A closed door calibration test was also completed which heated only the warm side of the chamber to steady state conditions with box fans while leaving the cold side open to the surrounding building. This test allowed for the calculation of heat transfer through the central divider wall in the chamber through proportional comparison of heat loss, temperature differential, and its associated surface area relative to the full chamber test.

The test conditions at which the horizontal air flow recirculatory air curtain was tested consisted of a cold side temperature of 4ºC and a warm side temperature of 24ºC with 60% relative humidity. The warm side and cold side temperature and humidity control points were located on the outer edge of the measurement region relative to the opening with each being 2.5m from the opening. The cold room temperature control point was located 1.4m off the floor of the cold room while the warm room temperature/humidity control point was located 1.4m from the ceiling of the warm room. The testing conditions and iterations are shown in Table 2.

Table 2: Test conditions

Doorway Status	**Doorway Size**	**Cold Side Set Point**	**Warm Side Set Point**	**Air Curtain Status**
*Open	2.1m x 2.1m	Steady State	Steady State	Off
*Closed	2.1m x 2.1m	Steady State	Steady State	Off
Open	0.91m x 0.91m	4ºC	24ºC	Off
Open	1.5m x 1.5m	4ºC	24ºC	Off
Open	2.1m x 2.1m	4ºC	24ºC	Off
Open	2.1m x 2.1m	4ºC	24ºC	On

*Chamber Calibration Tests

Additional open door tests were performed to test smaller openings of 0.91m x 0.91m and 1.5m x 1.5m in order to determine if the chamber size relative to the opening affected the energy transfer through the opening.

4. DATA AND RESULTS

The open door calibration was first run by heating the interior of both the warm and cold sides of the chamber with a measured and evenly distributed heat load created by operating fans and blowers adding up to 1.54kW. This produced an average interior surface temperature (walls, ceiling, and floors) of 46.4°C. At this condition the average exterior surface temperature measured was 25.7°C yielding an average ΔT across the chamber walls, floor, and ceiling of 20.7°C. The surface area of the outer walls, ceiling and floor on the warm side is $73.7m^2$ while the surface area on the cold side is $131.6m^2$. Assuming consistent chamber construction, the ratio of these areas can be used to divide the heat loss of 1.54kW between the warm side outer walls, ceiling, and floor and the cold side outer walls, ceiling, and floor to be 0.55kW and 0.99kW respectively for a ΔT of 20.7°C. This ΔT is very similar to that seen in the cold chamber during a standard test of the air curtain and relationships were developed to relate the heat loss to the measured ΔT across the walls, ceiling, and floors for the warm and cold sides of the chamber by fitting curves to the data above. The relationship for the warm side was calculated to be:

$$q_{warm\_loss} = 0.0269\left(T_{warm_{o,s}} - T_{warm_{i,s}}\right) \quad (1)$$

where:

q_{warm_loss} = heat lost/gained through the warm side chamber walls, ceiling, and floor, kW
$T_{warm_{o,s}}$ = temperature of warm side outer surfaces, °C
$T_{warm_{i,s}}$ = temperature of warm side inner surfaces, °C

The relationship for the cold side was calculated to be:

$$q_{cold\_loss} = 0.0475\left(T_{cold_{o,s}} - T_{cold_{i,s}}\right) \quad (2)$$

where:

q_{cold_loss} = heat lost/gained through the cold side chamber walls, ceiling, and floor, kW
$T_{cold_{o,s}}$ = temperature of cold side outer surfaces, °C
$T_{cold_{i,s}}$ = temperature of cold side inner surfaces, °C

Using this relationship the typical heat loss through the cold chamber walls during the standard test condition with the air curtain running is 0.810kW.

The closed door calibration was performed by closing the opening with an insulated sheet similar in thickness to the chamber wall construction and the divider wall construction. The warm side was then heated with fans and the power was measured similar to the previous calibration. The cold side was maintained at the ambient temperature of the surrounding building. In this case 0.437kW of heat input was measured and the average temperature difference between the inner and outer wall, floor, and ceiling surfaces was measured to be 13.2°C. Using this temperature difference with the relationship developed for the warm side outer wall loss yields 0.355kW lost through the outer walls, ceiling, and floor. The difference between the 0.437kW heat input and 0.355kW outer wall loss yields a loss through the center divider wall and closed door of 0.082kW for a measured wall ΔT of 13.0°C. This loss can then be scaled using the total divider wall area with closed door of 15.7m^2 compared to the divider wall area with open door of 11.1m^2. This yields a loss through the divider wall with open door of 0.058kW for a measured wall ΔT of 13.0°C. In comparison to the 35.7kW going through the opening in a standard open door test shown below, the order of magnitude of the loss through the insulated portion of the wall is extremely small. A relationship was developed using the above data yielding

$$q_{divider\_loss} = 0.0045\left(T_{divider_o} - T_{divider_i}\right) \quad (3)$$

where:

$q_{divider\_loss}$ = heat lost from warm room to cold room through divider wall, kW
$T_{divider_o}$ = temperature of warm room side of divider wall, °C
$T_{divider_i}$ = temperature of cold room side of divider wall, °C

The effectiveness if the air curtain is determined by subtracting from 1 the ratio of the measured energy loss at the doorway with the air curtain running to the measured energy loss at the doorway with no air curtain in place. The total heat input measured on the warm side minus the heat lost through the walls equals the heat transferred through the opening or the activated air curtain. The cold side balance is used as a check for this warm side heat transfer calculation. The 2.1m x 2.1m opening was first tested at the standard operating conditions and the total heat transferred through the opening was found to be 35.7kW. The temperature profile shows the considerable flow of air and heat between the sides with the horizontal temperature gradients that are visible in Figure 2.
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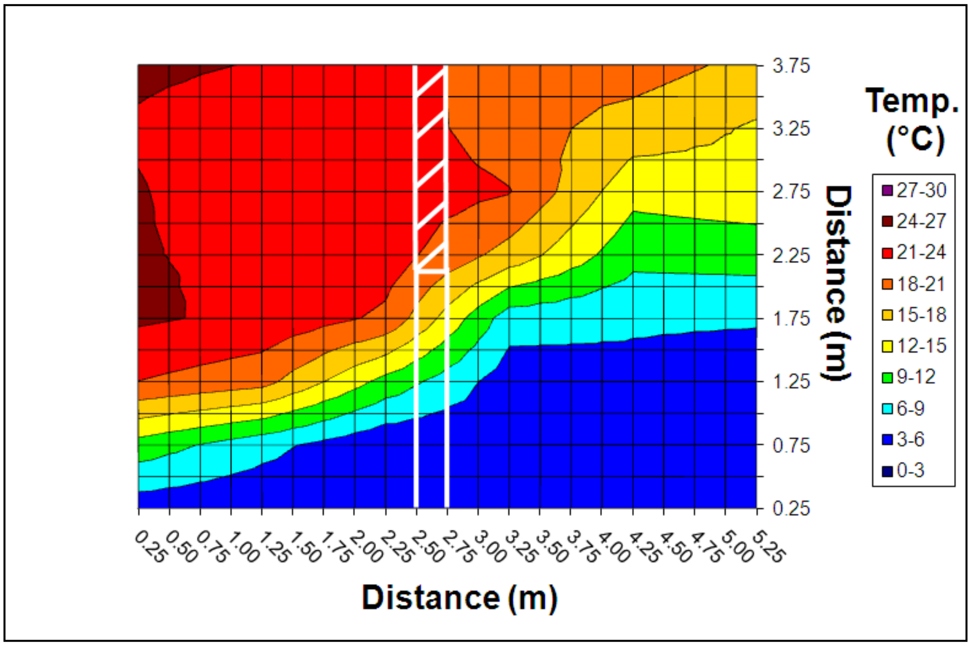

Figure 2: Temperature profile for 2.1m x 2.1m open door test

The flow of the warm air from the warm side through the top half of the opening and the flow of the cold air back from the cold side through the lower half of the opening are also clearly visible in the 3D planar data taken during the open door test shown in Figure 3.

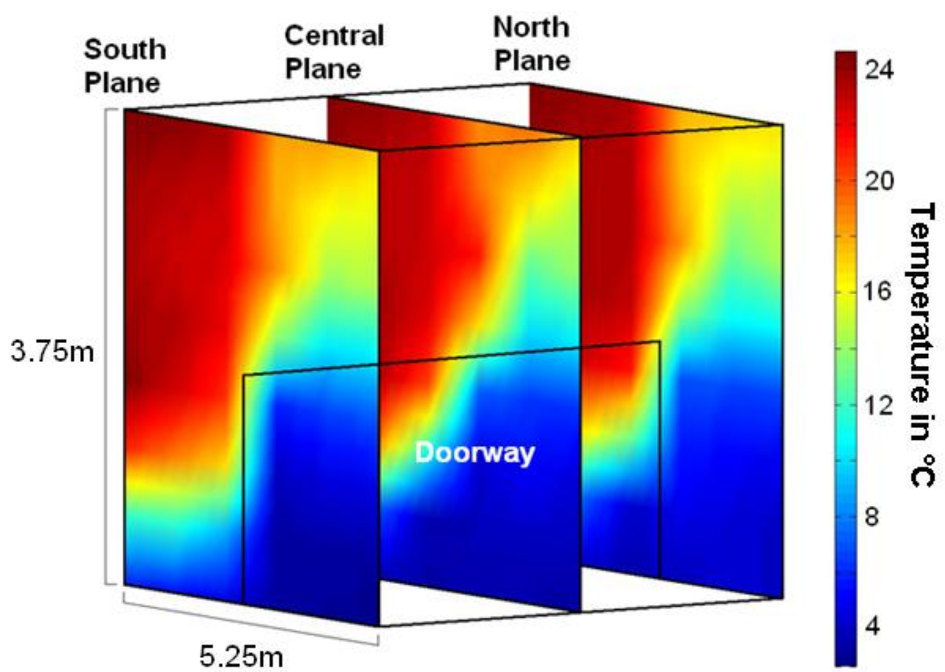

Figure 3: Planar temperature profiles with 2.1m x 2.1m open doorway

Theoretical calculations for the open door heat loss were made using a method for quantifying infiltration through open doorways described in the Industrial Refrigeration Handbook (Stoecker, 1998). Infiltration can be calculated using the following formula (Gosney and Olama, 1975):

$$Q = C_{inf} A\sqrt{H}\ \left(\frac{\rho_i - \rho_o}{\rho_i}\right)^{1/2} \left[\frac{2}{1 + \frac{\rho_i}{\rho_o}^{1/3}}\right]^{3/2} \qquad (4)$$

where:

Q	=	volume rate of flow, m^3/s
C_{inf}	=	infiltration coefficient = 0.692 $m^{1/2}/s$
A	=	area of doorway, m^2
H	=	height of doorway, m
ρ_i and ρ_o	=	air densities, kg/m^3, of the cold and warm air, respectively

The mass flow rate of air is the product of the volume flow rate and the mean density of air:

$$\dot{m} = Q\ \left(\frac{\rho_i + \rho_o}{2}\right) \qquad (5)$$

where:

$\dot{m}$	=	mass flow rate, kg/s

The heat transfer through the opening is the mass flow rate multiplied by the difference in enthalpies:

$$q = \dot{m}\left(h_{a,o} - h_{a,i}\right) \qquad (6)$$

where:

q	=	heat transfer through the opening, kW
$h_{a,i}$ and $h_{a,o}$	=	air enthalpies, kJ/kg, of the cold and warm air, respectively

The model results predicted a heat loss through the 2.1m by 2.1m opening of 54.5kW compared to the measured value of 35.7kW. Stoecker also references later experiments which showed the Gosney and Olama model to over predict the heat transfer through an opening which is observed in this case. Test points were added to test smaller openings of 0.91m x 0.91m and 1.5m x 1.5m in order to determine if the chamber size relative to the opening affected the energy transfer through the opening. The measured heat loss through the opening vs. the predicted heat loss is shown in Table 3. The percentage difference between measured and predicted decreases as the opening size decreases.

Table 3: Open door heat loss

Opening Size (m)	Measured Q (kW)	Predicted Q (kW)
0.91 x 0.91	4.84	6.73
1.5 x 1.5	15.7	23.5
2.1 x 2.1	35.7	54.5

It was expected that a more pronounced “stack effect” would be seen with a column of higher density cold air building above the opening. The high column of cold air relative to the opening is especially visible in Figure 4 which is in accordance with the data showing better agreement between measured and predicted capacities for the smaller openings with the column of high density cold air causing a relatively higher air flow through the smaller opening.

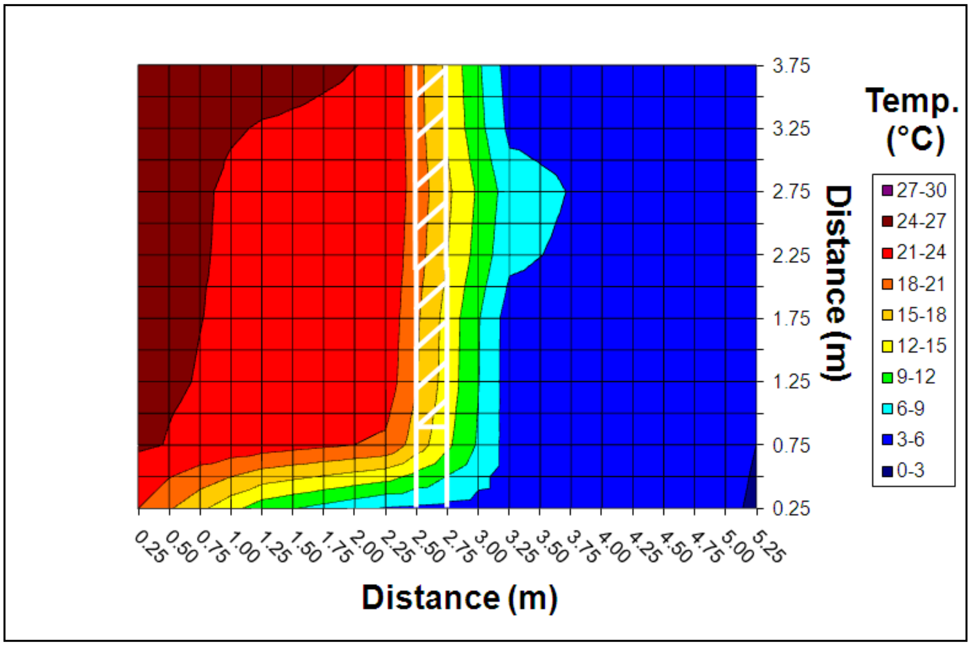

Figure 4: Temperature profile for 0.91m x 0.91m open door test

The horizontal air flow recirculatory air curtain was installed and adjusted for optimal air flow velocity and air deflection angle while running at the standard operating condition. With the air curtain running, the heat transferred through the opening was found to be 10.3kW. It was assumed that in the worst case the entire power consumption of the air curtain of 330W was transferred into the cold room and this value was included in the heat transfer calculation with the air curtain running. When compared to the open doorway heat transfer of 35.7kW, an effectiveness of 71% was obtained for the horizontal air flow recirculatory air curtain. Figure 5 and Figure 6 show this significant heat transfer reduction as exemplified by the vertical temperature gradients between the cold and warm side. It is clear that the warm air is largely kept on the warm side and the cold air is largely kept on the cold side. Higher air curtain efficiencies may be achievable in the field as the "stack effect" would be more prominent for an open door of this size with the ceiling height being much higher in warehouse facilities when compared to the environmental test chamber. As larger calorimetric chambers are now available, it would be informative to run the tests with and without the air curtain with increased ceiling height relative to the opening.

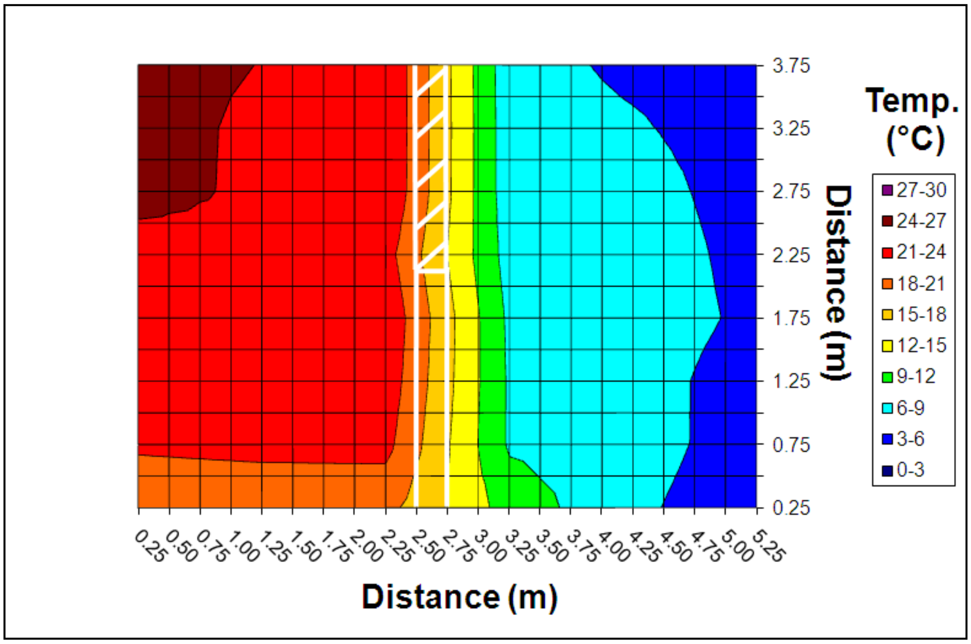

Figure 5: Temperature profile with the horizontal air flow recirculatory air curtain in operation

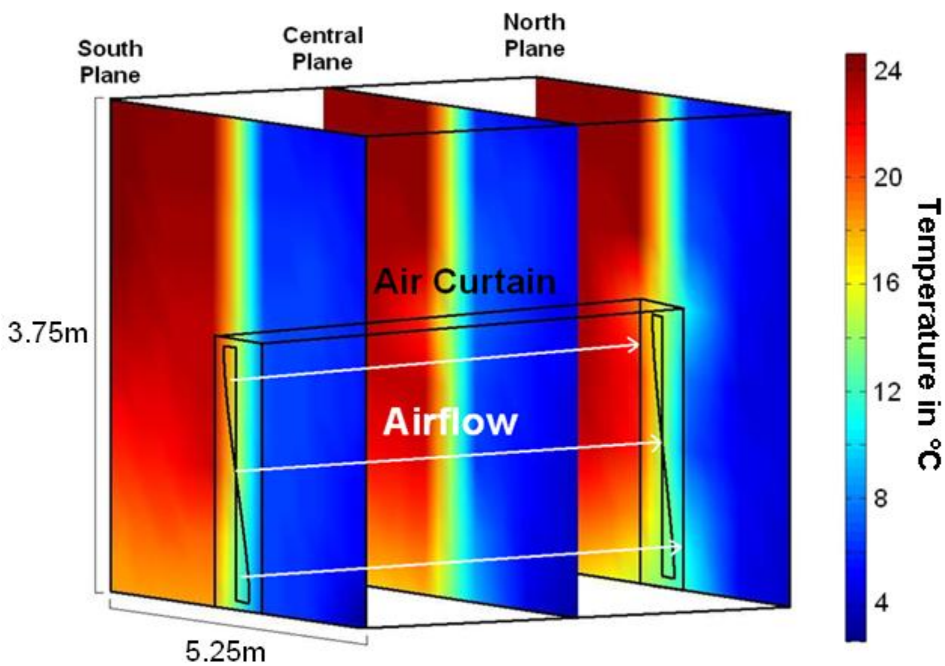

Figure 6: Planar temperature profiles with the horizontal recirculatory air curtain in operation

5. CONCLUSIONS

A testing method and facility were developed to measure the effectiveness of a horizontal air flow recirculatory air curtain by measuring the energy flow through an opening with and without the air curtain in operation. Previously tracer gas decay was commonly used to estimate the effectiveness of air curtains. In order to measure the actual energy flow, two calorimeters were separated by a divider wall with an opening outfitted with a horizontal air flow recirculatory air curtain. The heat transfer was measured through the opening with and without the air curtain in operation with the warm side controlled to 24ºC and 60% relative humidity and the cold side controlled to 4ºC. All energy inputs and sinks were measured on the warm side with the difference being the energy transfer through the opening. The air curtain reduced the heat transfer between the calorimeters from 35.7kW to 10.3kW yielding an effectiveness of 71%. Additional studies were performed with smaller openings showing that a high ceiling relative to the opening height can increase the heat transfer due to the “stack effect” of the dense cold air when compared to theoretical calculations. This may cause air curtain efficiencies in the field to have greater effectiveness ratings than what was measured due to higher ceilings and adds to the interest in further experimentation now that larger calorimetric chambers are available. Clear visual evidence of the air curtain effectiveness was also obtained from plots of thermocouple grid data showing vertical temperature gradients indicating little energy flow through the opening. In addition to the energy savings achieved by using the horizontal air flow recirculatory air curtain, moisture transfer is significantly reduced to the cold space. This prevents ice from forming on floors and products as well as on the cooling coils which reduces the amount of defrost cycles necessary and improves coil efficiencies.

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