Evaluation of Horizontal Recirculatory Air Curtain Efficiencies - Cooler to Conditioned Space.

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Abstract

Air curtains have long been used to reduce losses from high traffic doorways in cold storage facilities. Different varieties of air curtains have been used including vertical (down blast) non-recirculatory, horizontal non-recirculatory and horizontal recirculatory. The effectiveness of air curtains to provide thermal separation between rooms of differing temperatures has long been a matter of debate. Previous experiments have been done to determine the effectiveness using tracer gas decay methods. This paper will evaluate the thermal effectiveness of the horizontal recirculatory air curtain when applied in a typical cooler application using an environmental chamber. This allowed testing of the air curtain's ability to reduce infiltration on the actual thermal envelope. A vertical non-recirculatory air curtain was also evaluated for comparison purposes. The air curtains were tested in an environmental chamber that was divided into two rooms of differing temperature. The chamber was instrumented to measure all energy transfer into and out of each room. The cold room was maintained at a temperature of 4°C (39°F) and the warm room was maintained at a temperature of 24°C (75°F) and 60% RH throughout all tests. Three phases of testing were conducted for each air curtain to completely evaluate all energy losses – a completely closed doorway test for calibration, a completely open doorway test with no air curtains and an open doorway test with the air curtains adjusted and running. Completion of the testing revealed the horizontal recirculatory air curtain had a thermal efficiency of 71% while the vertical non-recirculatory air curtain had an efficiency of 38%. The results were slightly lower than theoretical calculations which may be attributable to chamber size. Additional tests were conducted using a smaller doorway for comparison. Applying the results from the smaller doorway yields an effectiveness of 76.2%. Using the theoretical open doorway energy loss yields an effectiveness of 80%

Introduction

Air curtains are generally used to reduce infiltration to both cold and warm spaces. They are used extensively in the cold storage industry to provide thermal separation and minimize refrigeration loads for coolers of differing temperature, freezers of differing temperature, freezers to coolers, coolers to ambient areas, and freezers to ambient areas. There have been several tests done in recent decades to determine the effectiveness or performance of different air curtain configurations at differing temperatures; however, most if not all of these tests were conducted using a tracer gas decay method. A summary of results from a few of these tests is summarized in Table 1. (Downing and Meffert 1993, Pham and Oliver 1983, Longdill and Wyborn 1979, Valkeapaa et al. 2005). For single air curtains in static cooler applications the results vary from as low as 42% to as high as 82%.

The current method for determining the effectiveness of doorway protective devices such as air curtains as recommended in the 1990 ASHRAE Handbook-Refrigeration is to take the ratio of the infiltration at the doorway with the protective device in place and the infiltration of the doorway with no protective device in place calculated from the Gosney and Olama model (ASHRAE 2002) for air infiltration and subtract from 1. The 2002 ASHRAE Handbook-Refrigeration states the maximum effectiveness of air curtains to be 0.7.

Table 1. Previously reported effectiveness values – single air curtains in cooler applications.

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

No previous data was found where air curtains were tested by trying to measure actual heat and moisture gains or losses at the doorway. This is understandable considering the difficulties of measuring infiltration velocities in three dimensions at an existing facility not to mention the temperature and humidity measurement grid required. A solution to this was to do a calorimeter test in an environmental test chamber large enough to be representative of typical industry applications. A calorimeter test allows the evaluation of all energy losses at the doorway with and without a doorway protective device in place. The effectiveness is then determined by the ratio of the measured energy loss at the doorway with the protective device in place to the measured energy loss at the doorway with no protective device in place and subtract from 1.

The cold storage industry today still does not recognize a particular number for air curtain efficiency. There is still a need for better information regarding these efficiencies especially now with a focus on sustainability and green technologies. The main objective is to begin providing those numbers starting with single air curtains in a static cooler application with a large enough opening and a significant temperature differential to provide meaningful results. A static condition for the purpose of these tests is defined as infiltration at a doorway due to temperature differences only. No pressure imbalance exists to create additional infiltration.

Experimental Method

According to ASHRAE the effectiveness of a protective device is defined by the following equation:

$$E = (1 - I_p/I_u) \times 100\% \qquad (1)$$

where

I_p = protected doorway air exchange rate (ft^3/hr)
I_u = unprotected doorway air exchange rate (ft^3/hr)

I_u is calculated from the Gosney and Olama model (ASHRAE 2002) to be

$$I_u = 795.6\ (A)\ (1-\rho_i/\rho_r)^{0.5}\ (gH)^{0.5}\ (Fm) \quad \text{in } ft^3/hr \qquad (2)$$

Where

795.6 = 3600 s/h x 0.221;
A = doorway area, ft^2;
ρ_i = density of infiltration air, lbm/ft^3;
ρ_r = density of refrigerated air, lbm/ft^3;
g = gravitational constant, 32.17 ft/s^2;
H = doorway height, ft; and
Fm = density factor.

The density factor is defined by Gosney and Olama (ASHRAE 2002) as

$$Fm = [2/(1+(\rho_i/\rho_r)^{1/3})]^{3/2} \tag{3}$$

This method of calculating the doorway protective device effectiveness is fine for a fixed barrier since the effectiveness is easily calculated using standard heat transfer equations and applying a leakage factor; however, it does not work as well for air curtains because they are more complex systems to evaluate. In proper determination of effectiveness of air curtains, all refrigeration losses should be accounted for and thus should include all sensible and latent heat gains or losses from the infiltration. The theoretical effectiveness calculation of the air curtain becomes a direct calculation of the ratio of the calculated doorway refrigeration loss with the air curtain in place and the calculated open doorway refrigeration loss.

$$E = (q_o-q_a/q_o) \times 100\% \quad \text{or} \quad (1-q_a/q_o) \times 100\%$$

q_a = energy loss with air curtain, Btu/hr or kW
q_o = energy loss of of open doorway, Btu/hr or kW

The open doorway energy loss is determined by using the Gosney and Olama model for fully established flow (ASHRAE 2002) which is the same as equation (2) above but multiplied by density of the refrigerated air and the enthalpy difference of the infiltrating air and the refrigerated air. Thus the equation for refrigeration loss of an open doorway becomes

$$q_o = 795.6\ (A)\ (h_i - h_r)\ \rho_r\ (1-\rho_i/\rho_r)^{0.5}\ (gH)^{0.5}\ (Fm)\ (D_f) \tag{4}$$

The theoretical refrigeration loss for the horizontal recirculatory air curtain is more complicated and cannot be shown here but the calculations and results used for theoretical effectiveness calculations is included in Appendix A. The theoretical effectiveness of a horizontal recirculatory air curtain placed on a 7-ft by 7-ft opening between a 39°F cooler at 90% RH and a 75°F area at 60% RH is calculated to be

q_a = 33,840 Btu/hr
q_o = 174,000 Btu/hr

$$E = (1 - 33{,}840/174{,}000) \times 100\% = 80.6\%$$

q_o is calculated using a doorway flow factor of 0.80 per Hendrix et al. (1989) (ASHRAE 2002).

The requirement was to determine the experimentally measured effectiveness of air curtains instead of the theoretical effectiveness. The same effectiveness equation applies but q_a and q_o become measured values using the calorimeter environmental chamber.

Test Equipment

A calorimeter capable of producing a wide range of environmental conditions was used to evaluate the performance of the horizontal recirculatory air curtain. An insulated wall with a 7-ft by 7-ft opening was centered between the air handling equipment on both sides of the chamber.
Figure 1 shows a schematic of the environmental chamber and the components contained inside.

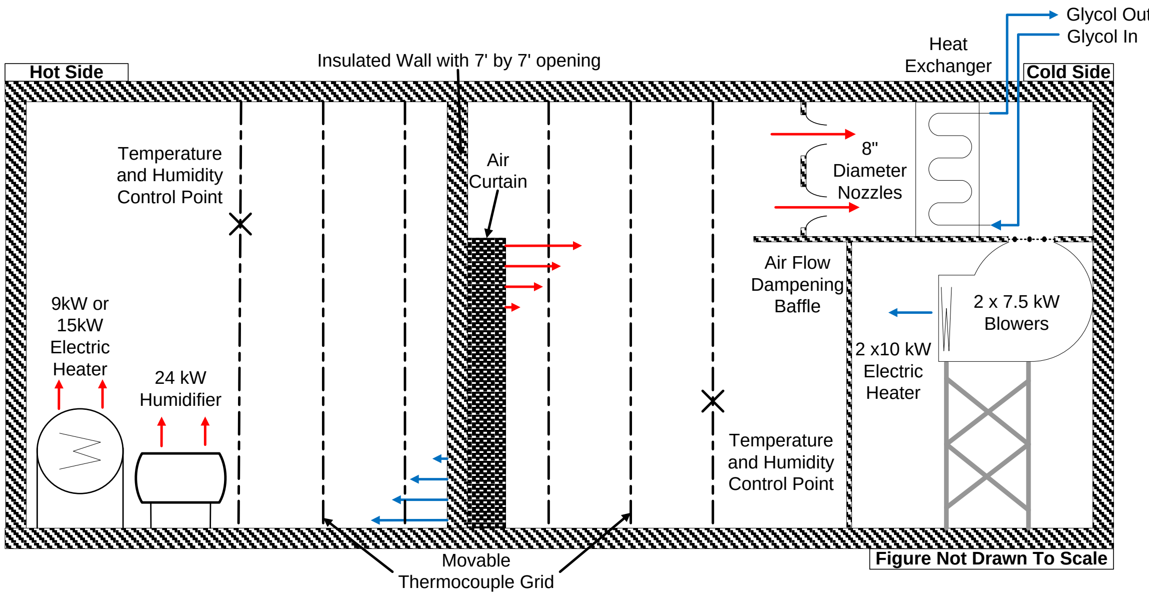

Figure 1. Environmental chamber schematic.

Cooling was supplied to the cold side heat exchanger by circulating ethylene glycol from two air cooled chillers each rated at 17 tons of cooling capacity. For the majority of tests, only one chiller was used and the cooling load on the chamber was varied as necessary through modification of flow rates, reduction in 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 air flow though the opening. The cooling load on the chamber can be 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 using the formula $Q=mc_p\Delta T$. It was necessary to derive the glycol concentration from the solution density and use this value along with the average solution temperature for a given test to derive its specific heat using interpolations of values from ASHRAE Standard 21.6. The power input to the cold side was also measured using watt transducers. This includes power going to the blowers, fans and VFDs. The power going to the air curtain was measured by a separate watt transducer. The chamber was calibrated and all heat coming through the walls was also measured using thermocouples on all inner and outer wall, floor, and ceiling surfaces.

Heating was provided to the warm side by a 9kW or a 15kW heater connected to a variable frequency drive which was controlled to maintain the chamber temperature using the Agilent VEE Pro data acquisition program. Humidification was achieved through the use of a steam generator which was also varied by a variable frequency drive and controlled using the data logger. Several box fans were placed near the air handling and temperature and humidity control equipment and were arranged to blow parallel to the opening to encourage air circulation on the cold and hot side of the chamber while not interfering with energy and air flow from the warm side to the cold side. The fans were positioned to help ensure a 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, lights, etc. All electrical heating inputs were accounted for.

Airflow measurements were taken at the 7-ft by 7-ft 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 40 ft/min limit commonly found in testing standards.

The acquisition of planar air temperature profile data across a 17-ft by 12-ft cross section perpendicular to the opening was accomplished using a type T thermocouple grid configured so as to move 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 1.5-ft vertical intervals from top to bottom for a total of 48 thermocouples. A spacing of 3-ft was maintained between each vertical column for every test. Three temperature plane locations were used for each test 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. All sensors types shown in Table 2 were connected to three HP data loggers controlled by this program. 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 hot 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.

Table 2. Instrumentation list

Measurement	Instrument	Brand	Range	Accuracy	Description/Location
Temperature	Type T Thermocouple Probe	Omega	-250 to 220ºC	1ºC or 0.75% above 0ºC 1.5% below 0ºC	Glycol temperature measurements

Temperature	Type T Special Limits of Error Welded Thermocouple	Omega	-250 to 350ºC	Greater of 0.5ºC or 0.4%	Air temperature measurements
Dew Point	Model D-2-SR 2 stage chilled mirror sensor	General Eastern	-25ºC to 85ºC (up to 100% RH)	±0.2ºC	Hot side (E6), Cold Side (South Blower), Cold Side Nozzle
Air-side Pressure Drop	Differential Pressure Transducer Model 26512R5WD1 1T1F	Setra	0 to 2.5" WC	±0.25% F.S.	Pressure drop across cold side heat exchanger
Glycol Flow Rate/Density	Mass Flow Sensor Model CMF200	Micro Motion	0 to 24.2kg/s	+/-0.05% of flow rate/ +/-0.0002 g/cc	Glycol (liquid) refrigerant loop mass flow rate/density
Watt Transducer	Model GW5-019D	Ohio Semitronics	0-2kW	±0.2% F.S.	Lights, fans, and other 120V devices
Watt Transducer	Model GW5-006E	Ohio Semitronics	0-120kW	±0.2% F.S.	Heaters, Blowers, Humidifiers on hot and cold sides
Watt Transducer	Model GW5-024E	Ohio Semitronics	0-16kW=4-20mA	±0.2% F.S.	Air Curtain

Testing Conditions and Procedure

Three phases of testing were performed to ascertain the performance of the air curtains. Table 3 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 hot and cold sides of the chamber based on measured temperature differences in future tests.

A closed door calibration test was also completed with both warm and cold rooms brought to steady state conditions. There was no measurable loss from the warm room due to similar temperatures inside the warm room and the surrounding building. The measured loss for the cold room was 2.2kW. This test allowed for the calculation of heat transfer through the central 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 open door, horizontal flow air curtain, and down blast air curtain were tested consisted of a cold side temperature of 39ºF and warm side temperature of 75ºF 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 8.2-ft from the opening. The cold room temperature control point was located 4.6-ft off the floor of the cold room while the warm room temperature/humidity control point was located 4.6-ft from the ceiling of the warm room. The testing conditions and iterations are shown in Table 3.

Table 3. Test conditions

Doorway Status	Doorway Size	Cold Side Set Point	Warm Side Set Point	Air Curtain Status
*Open	7-ft x 7-ft	Steady State	Steady State	Off
*Closed	7-ft x 7-ft	Steady State	Steady State	Off
Open	7-ft x 7-ft	4ºC	24ºC	Off
Open	7-ft x 7-ft	4ºC	24ºC	HCR On
Open	7-ft x 7-ft	4ºC	24ºC	Mars On

*Chamber Calibration Tests

The horizontal recirculatory air curtain was factory adjusted for optimal air flow velocity and air deflection angle and was tested at the standard operating condition. The unit was installed on the cold side to simulate a typical installation found in warehouse stores where the customer walks into a cold room to access perishable products.

A final test was completed to compare the horizontal, recirculatory air curtain performance to that of a down blast air curtain with no recirculation. A Mars high velocity industrial air curtain (Model HV84-2UH-BG) was tested. The down blast air curtain was installed by carefully following the manufacturer's instructions in the same way an end user would install the unit. The air curtain was mounted on the cold side of the chamber with the nozzle outlet raised one inch above the top of the doorway. The nozzle was configured using an angle template to blow air at a 15° angle outward from a vertical position so as to blow air slightly away from the doorway. The model provided was designed to have the greatest performance for doorways of a height between 10 and 12 feet. Because the doorway being tested had a height of 7 feet, it was necessary to significantly reduce the power supplied to the air curtain blowers. A VFD was installed and adjusted along with the air intake louvers to attain a condition where the air flow just reached the floor as specified in the operator's manual (Note: there is no VFD used in the operator's manual). A VFD setting of 30Hz was found adequate to meet the flow recommendations found in the installation manual. Tests were later run with the air curtain at VFD settings of 27.5Hz, and 25Hz to see the effect of air flow rate reduction although the end user would have limited means to accomplish this and determine the benefit in the field.

Data and Results

The effectiveness of the air curtain is defined by the difference of the heat transferred through the open door and the heat transferred through the air curtain divided by the heat transferred through the open door. The total heat input measured on the warm side minus the heat lost through the walls equals the heat transferred through the opening or air curtain if running. The cold side balance is used as a check for this hot side heat transfer calculation. The completely unprotected 7-ft x 7-ft 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.

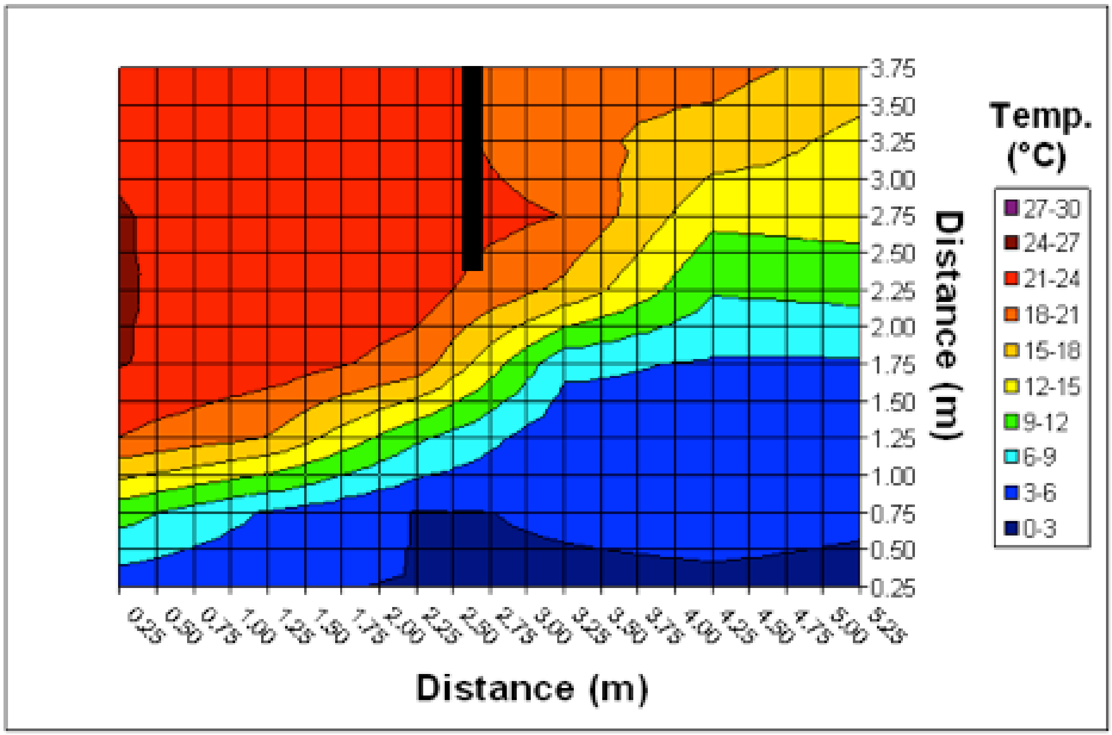

Figure 2. Temperature profile for open door test.

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 clearly visible in the 3D planar data taken during the open door test shown in Figure 3.

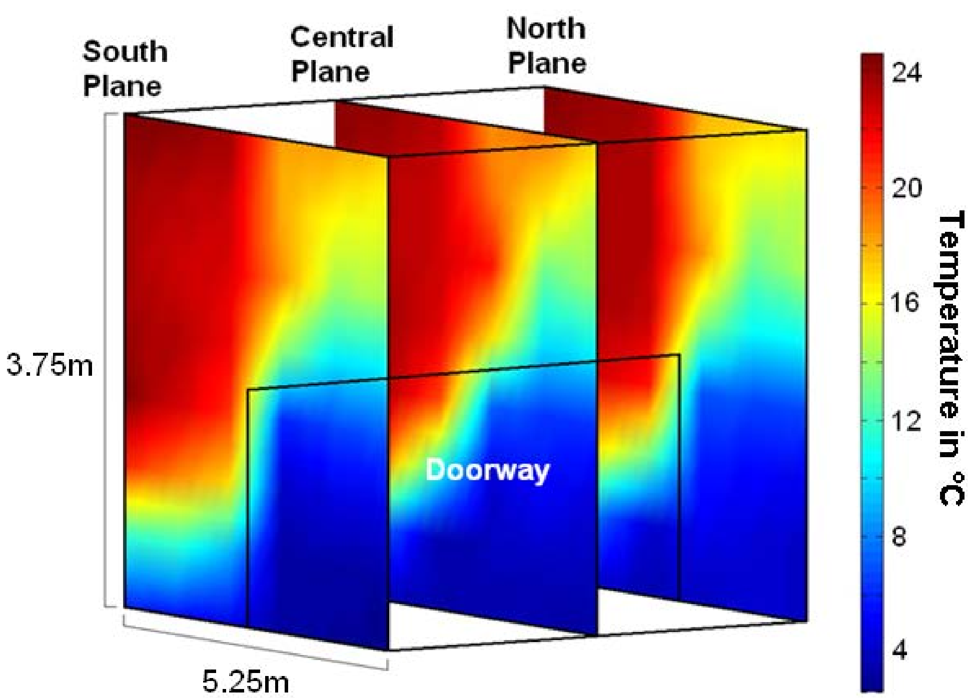

Figure 3. Planar temperature profiles with open doorway.

The horizontal recirculatory air curtain was adjusted for optimal air flow velocity and air deflection angle and was tested at the standard operating condition. With the air curtain running, the heat transferred through the opening was found to be 10.3kW. When compared to the open doorway, an effectiveness of 71% was obtained for the re-circulatory air curtain. Figure 4 and Figure 5 show this significant heat transfer reduction ith the vertical temperature gradients. It is clear that the warm air is largely kept on the warm side and that the cold air is largely kept on the cold side.

Figure 4. Temperature profile for horizontal flow, recirculatory air curtain.

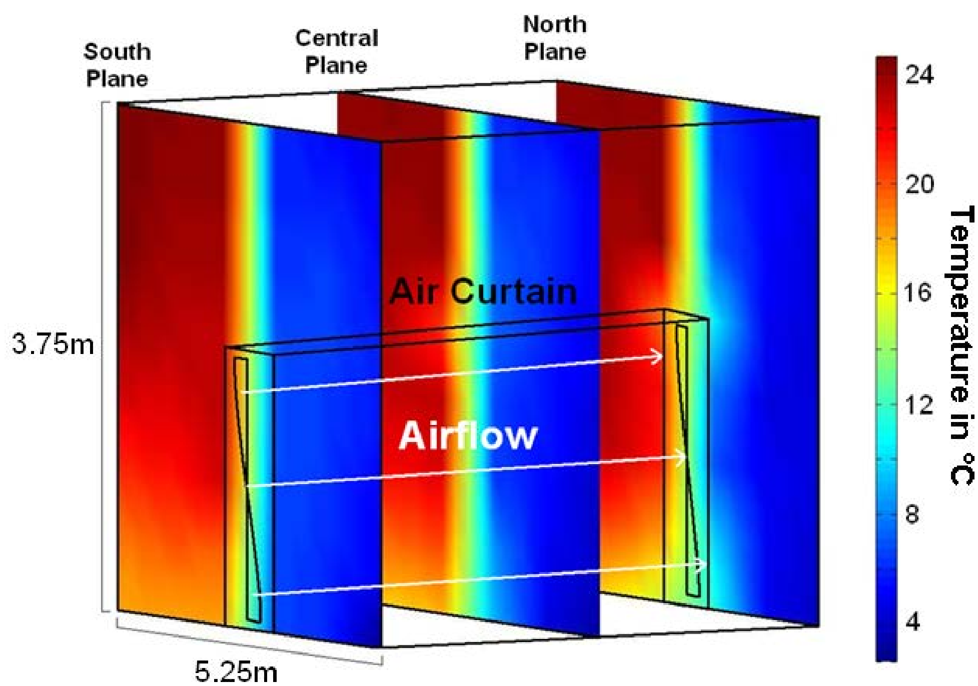

Figure 5. Planar temperature profiles for horizontal recirculatory air curtain.

It is worth noting that the measured open doorway heat loss was approximately 70% of the calculated theoretical open doorway heat loss using a doorway flow factor, Df of 0.8. Downing and Meffert (1993) as well as Pham and Oliver (1983) noted that a low ceiling height and obstructions near the door could effect measured open doorway infiltration air. The calorimeter rooms were significantly smaller than typical cold storage facilities and the ceiling heights much lower. The room height and size seem to have a great effect on heat loss due to recirculation of air back through the doorway. Upon completion of the air curtain testing, a smaller 3-ft by 3-ft opening was used between the warm and cold rooms to see if the size of the opening compared to room size or ceiling height had an impact on open doorway results. The calculated heat loss for the 3-ft by 3-ft opening yields 5.8kW. The measured value of open doorway heat loss was 4.9kW which is 85% of the calculated value. This is closer to theoretical than the 70% of calculated for the 7-ft by 7-ft opening. If the air curtain effectiveness is recalculated using 85% of the calculated value for the 7-ft by 7-ft opening, the air curtain effectiveness result is 76.2%. If we are applying air curtains to very large rooms with high ceilings we could expect the theoretical model for open doorway infiltration to match measured results based on prior testing and empirical data. With this being the case the final horizontal recirculatory air curtain effectiveness would be 80%.

The single vertical non-recirculatory air curtain was tested under the standard operating conditions with a VFD setting of 30Hz. The total heat transferred with the air curtain running was 22.1kW yielding a down blast air curtain effectiveness of 38% using the open doorway value of 35.7kW previously noted. Further optimization is possible although unlikely in the field.

Condensation was observed on the down blast air curtain housing and on the floor at the base of the opening directly below the air curtain. This was not observed at all with the horizontal recirculatory air curtain.

Discussion and Conclusion

The final results form the calorimeter testing indicates that the measured effectiveness of a horizontal recirculatory air curtain in a static cooler application with a temperature differential of 36°F is 71%. The vertical non-recirculatory air curtain had a much lower measured effectiveness applied in the same conditions at only 38%. There is clear visual evidence seen in the temperature profiles of the ability of a single horizontal recirculatory air curtain to provide an effective barrier between rooms of differing temperatures. In addition to the energy savings there is a significant benefit in preventing moisture migration, compared to the vertical unit, to the cold room. The room sizes and ceiling heights used seemed to have a negative effect on the measured open doorway infiltration results due to recirculation of air back through the opening. In typical cold storage applications where warm and cold room sizes may be as large as 200,000 square feet and ceiling heights at 30-ft the recirculation of air through the opening would not exist. In these applications we would conclude that horizontal recirculatory air curtains in this same temperature range would have effectiveness values as high as 80% and the vertical non-recirculatory air curtain would have effectiveness values as high as 56%. The recirculatory air curtain value compares very well to the theoretical calculated value of 80.6%.

At 80% effectiveness the horizontal recirculatory air curtain would be compared to a typical fast acting traffic door operating at 20% door open time (DOT). The issue with the traffic door is that it provides no barrier when the door is open and this can create significant energy issues in extremely high traffic applications. In high traffic applications the single horizontal recirculatory air curtain would save energy when standard traffic doors are operating above a 20% DOT. This very high traffic exists in many large grocery distribution centers. Many other cold storage facilities; however, do not have this extremely high traffic. In these applications the coupling of a horizontal recirculatory air curtains with a fast acting traffic door will provide the highest energy savings possible. If an 80% effective barrier is provided whenever the traffic door is open and the traffic door provides a 95% + effective barrier whenever it is closed. The net result is extremely low energy loss at a doorway.

Future Work

This experiment focused on application of a horizontal recirculatory air curtain as a thermal barrier in a static cooler application. Future work would be to test double air vestibules in static freezer applications and expand the single and double testing to non-static pressure imbalanced doorways. If larger calorimeter rooms become available it would also be a benefit to test the conclusion of open doorway infiltration numbers being effected by room size and ceiling height.

Acknowledgements

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Appendix A

Model AC
Energy Aanlysis

PROJECT

CUSTOMER	Creative Thermal Solutions	DATE	11/16/10
DOOR DESCRIPTION	7-ft by 7-ft Doorway	PROJ #	0

ENGINEERING DATA

DOORWAY SIZE			OPERATIONS		
WIDTH	84	inches	ANNUAL WAREHOUSING HRS	5840	hrs
HEIGHT	84	inches	UNIT COST OF ELECTRICITY	0.10	$/kWh
CONDITIONS					
FREEZER TEMPERATURE	39	°F	ANTEROOM TEMPERATURE	75	°F
RELATIVE HUMIDITY	80	%	RELATIVE HUMIDITY	60	%
AIR-CURTAIN PROPERTIES					
AC-1 TEMPERATURE	36.4	°F	AC-2 TEMPERATURE	N/A	°F
UNIT HEATER	N/A	kw	AC-3 TEMPERATURE	N/A	°F
CONTRIBUTION TO FREEZER LOAD			**BENEFIT TO ANTEROOM**		
SENSIBLE HEAT	1.50	ton	SENSIBLE	-1.5	ton
LATENT HEAT	1.32	ton	LATENT	-1.3	ton
EQUIPMENT RELATED LOAD			Exfiltration of the lower humidity air-curtain normally provides a latent benefit to anteroom conditions.		
SENSIBLE HEAT		ton			
LATENT HEAT		ton			
TOTAL	2.82	ton	TOTAL	-2.8	ton