PERFORMANCE OPTIMIZATION OF A SECONDARY REFRIGERANT DISPLAY CABINET USING TESTS AND CFD MODELLING

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ABSTRACT

In this paper a computational investigation to address some of the design parameters that have significant effects on the performance of a secondary refrigerant refrigerated display cabinet is presented. The investigated design parameters include air curtain velocity, width, discharge angle and positioning of the air curtain outlet from the front edge of the display cabinet's top shelf. In addition the effect of using honeycomb on the flow path at the air curtain outlet was also investigated. A 3D CFD model was constructed to assess the effect of the side flow on the performance of the vertical display cabinet. The suitability of using 2D CFD to carry out the display cabinet design optimization was investigated and it was concluded that for the length of the cabinet considered, the flow could be assumed to be two-dimensional for most of the cabinet length. The results also indicated that the optimum air curtain mass flow rate constituted approximately a third of the total air mass flow rate. For the geometry considered with respect to air curtain discharge and top shelf outer edge position it was found that shaping the honeycomb to produce more uniform flow and reduce recirculation at the inner edge of the air curtain outlet improved the air curtain performance and led to a 4% reduction in the cabinet load.

Keywords: air curtain, open refrigerated display cabinet, design optimization, CFD

1. INTRODUCTION

Vertical multi-deck display cabinets are large consumers of electrical energy because of their negative interaction with the ambient environment. Air curtains are widely used in refrigerated display cabinets to reduce the air, thermal and moisture transfer between the conditioned environment (inside the cabinet) and the surrounding ambient. Hence, the design of an energy efficient display cabinet is a much desired research topic. In a display cabinet, products are maintained at the desired temperature by air circulating through an evaporator where it is cooled before it passes in to a rear tunnel from where part of it flows through a perforated back panel to the shelves holding the product and the remainder exits from a discharge air grille at the front top of the cabinet forming an air curtain. The air curtain is designed to provide a barrier between the environment inside the cabinet and the surrounding air outside (Chan *et al.* 2005). Interaction between the chilled air curtain air with the warm air outside the cabinet and the cold air from the shelves cannot be avoided. A significant portion of this air is drawn back into the evaporator whereas the remainder spills out of the cabinet into the aisle of retail food stores. The air drawn from the ambient into the cabinet is called the infiltration air and the cooling load arising from the temperature and moisture content difference between the ambient and the air in the cabinet is the infiltration load. The infiltration load of a vertical chilled food cabinet can be significant and up to 80% of the total load (D'Agaro *et al.* 2006).

In recent years Computational Fluid Dynamics (CFD) modelling has been used for the investigation of different aspects of display cabinet design. Early studies used 2D modelling primarily to reduce computational time, neglecting the effect of external flows or air disturbances on the performance of the air curtain and the cabinet (Stribling *et.al.* 1997; Cortella *et.al.*, 2001). More recent studies used 3D CFD modelling and considered the influence of individual components on the performance of the display cabinet (Foster *et.al*. 2005; Hadawey 2006, D'Agaro *et al.* 2006).

Yu *et al.* (2007) considered the use of the mid section of the cabinet as satisfactory for the investigation of the performance of the air curtain whereas D' Agaro *et al.* (2006) reported that 3D simulations can produce more reliable results than 2D simulations. This paper employs both 2D and 3D CFD simulations for the investigation and optimisation of the performance of a secondary refrigerant vertical multi-deck cabinet. The simulation results were validated against data obtained from test results which are also reported in the paper.

2. EXPERIMENTAL WORK AND TEST STANDARDS

In order to validate the developed CFD models experimental investigations were carried out on a vertical multi-deck display cabinet. The cabinet as shown in Figure 1, was 2.45 m long, 2.07 m high and 0.7 m deep (internal depth 0.6 m) with a single air curtain. The opening height was 1.3m. The cabinet was designed to meet M1 class requirements (product temperature in the range between -1°C to +5°C (*ISO23953-2*, 2005)

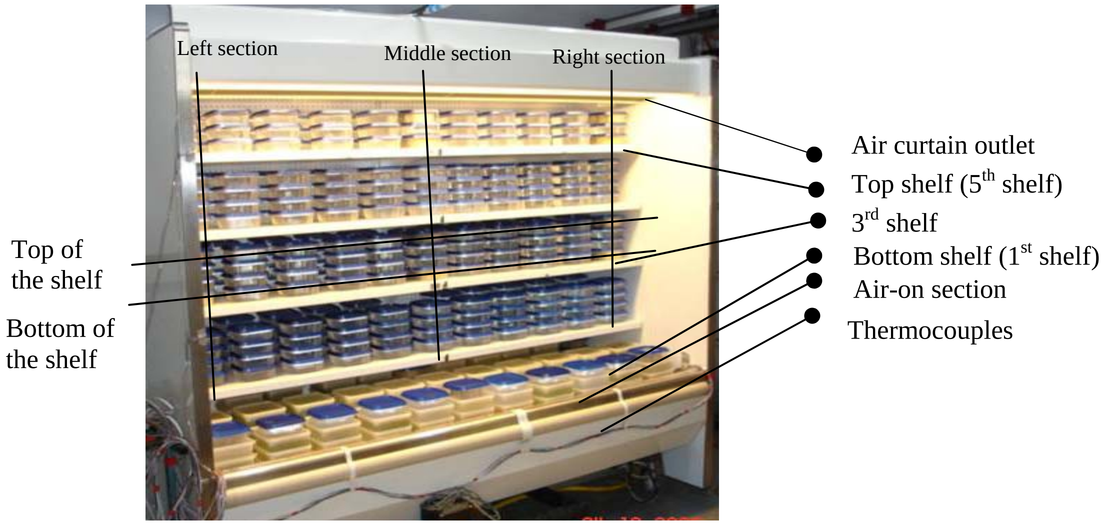


Figure 1. Vertical multi-deck display cabinet

The air curtain width and discharge angle were 110 mm and 12° respectively. Three evaporator fans were installed to provide the cabinet with the required air flow with an air mass flow rate of around 0.28 kg/s. The refrigeration to the display cabinet was provided by an absorption chiller with propylene glycol/water mixture (33% by volume with a temperature of -5°C) as the secondary heat transfer fluid. The mass flow rate of the secondary refrigerant was 0.8 m^3/h. The tests were carried out in an environmentally-controlled test chamber at *ISO23953-2* (2005) standards climate class III conditions. Air velocity measurements were taken at evaporator coil inlet (air-on) and outlet (air-off) on the right, left and middle sections of the cabinet. Right and left sections were 0.75 m from the respective side panels. 42 M-packs of 500g (100x100x50) mm, which had a calibrated T-type thermocouple inserted into the geometric centre of the pack, were used to measure product temperature at the positions where product temperature needed to be measured. The rest of the cabinet was loaded with 420 water containers of (160x160x80) mm to provide thermal mass to the display cabinet.

3. MODELLING THE DISPLAY CABINET AND DATA COMPARISONS

One of the most significant issues considered in the modelling of refrigerated display cabinets is whether the cabinet should be modelled in 2D or 3D. 2D modelling was considered in previous reported work by Stribling *et al.* (1997) and Cortella *et.al,* (2001) amongst others. In order to assess the suitability of a 2D model for cabinet design optimization, the cabinet was first modelled in 3D to investigate the effect of the side flow (cross flow in relation to air curtain flow direction) on the cabinet performance along its length using the FLUENT CFD package. The CFD results showed that with a side (cross) flow of 0.2 m/s the path lines of the air curtain were observed to be affected mostly on the right hand side, leading edge of the cabinet, close to the side panel, Figure 2**.**

The results indicate that the side flow created a recirculation area close to the side panel, which in turn formed an area of low pressure close to the air curtain. The low-pressure area "pulled" the air curtain outwards away from the cabinet, and this resulted in a reduction in the display cabinet's effectiveness close to the side panel.

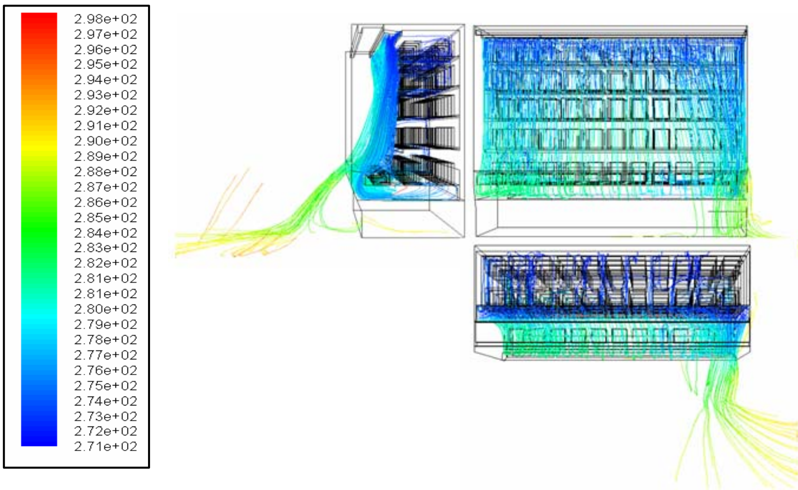


Figure 2. Path lines of air curtain coloured by static temperature (K) with 0.2 m/s cross flow

Figure 2 also shows that side flow influenced only a small section of the display cabinet. In retail food stores display cabinets are arranged in a line-ups, which can extend to anything up to 10 meters or more. In the case where a draught is along the face of the cabinet, only a small part of the cabinet line up will be affected by the side flow. In a real situation there will be many other disturbances to the flow such as shopping activity and non uniformity of product loading and air curtain velocity along the length of the cabinet amongst many others that make the flow 3-dimansional. At test conditions in the laboratory at *ISO23953-2*, however, the flow in the cabinet away from the side panels can be considered to be close to 2-dimensional. Unless the study specifically seeks to investigate and minimise side flow effects in a cabinet, the use of 2D modelling for the optimisation of specific design parameters of the cabinet can be justified. This approach was followed in this paper

The test results showed that front product temperature values were higher than product temperatures at the rear of the cabinet for all the shelves. Products at the rear complied with the *ISO23953-2* class M1 requirements whereas some products at the front of the cabinet exceeded the limit of + 5 ^{o}C. The test results also indicated that product temperatures at the mid section of the cabinet approximately represented the average value of temperature variation from right to left. Hence, the mid-section of the cabinet was used for the 2D CFD simulations.

Table 1 shows a comparison between measured and simulation results for the product temperatures on three shelves at the mid-section of the cabinet together with the calculated and predicted load on the cooling coil. It can be seen that the percentage average error of the 2D model in predicting the cooling load and product temperatures was approximately 9% and 10% respectively.

Table1. 2D CFD simulation and test results for product temperature and cooling load.

Central section	Shelf 1	Shelf 3	Shelf 5	Sensible load (W/m)	Latent Load (W/m)	Total Load (W/m)
	Front	Front	Front			
Experimental	5.7	5.3	5.2	930	688	1618
2D CFD	5.8	4.9	4	857	598	1455

4. ASSESSING THE FACTORS OF ENERGY-EFFICIENT CABINET DESIGN

The results showed the product temperatures on all the three shelves at which product temperatures were measured reaching values as high as 8.5°C. The failure of this cabinet to meet the performance criteria for class M1 could be attributed mainly to the characteristics of the air curtain and the poor air distribution system of the cabinet. The performance of the air curtain and the heat and mass transfer across it depends on several factors, which are investigated below with the aid of the validated CFD model.

4.1 Optimum Total Air Mass Flow Rate for the Display Cabinet

A temperature below 278 K was considered as the maximum product temperature, which the product can reach during simulation. A uniform mass flow distribution was assumed with regard to the air leaving the evaporator at the air-off section. The evaporator coil air off temperature was set at 271K and 93% RH and this temperature provides a good compromise between airflow requirement and evaporator fan and compressor power consumption, as was found by Tassou and Maki (2005). Table 2 shows product temperatures for different airflow rates. As it can be seen, for air flow rate of 0.13 kg/s per m cabinet length and above, total flow rate of 0.32 kg/s, the higher product temperatures were on shelf 1 (bottom) and shelf 5 (top). Increasing the air flow through the cabinet by 30% to 0.43 kg/s reduced product temperatures in all positions of the cabinet to below 5.0 ^{o}C.

Table 2. Product temperature and cooling load for different air mass flow rate

Mass flow rate (kg/s)	Sh1 (°C)	Sh2 (°C)	Sh3 (°C)	Sh4 (°C)	Sh5 (°C)	Cooling load (kW)	ACV (m/s)
0.32	5.9	4.2	4	4.1	4.8	4.27	0.32
0.40	5.2	3.8	3.6	3.8	4.1	4.61	0.39
0.43	4.2	3.5	3.4	3.4	3.9	4.73	0.41

4.2 Effect of Air Curtain Width on Cabinet Performance

To investigate the influence of the air curtain width on the performance of the cabinet, widths of 50, 70, 90 and 110 mm were considered in the simulations with a constant air flow rate through the cabinet of 0.43 kg/s Figure 3 shows the product and air temperature contours inside the display cabinet for the different air curtain widths. It can be seen that even though the product temperature variation for the cases considered is relatively small the lowest product temperatures at the front of the cabinet were achieved with air curtain slot widths of between 90 mm and 110 mm.

Table 3 summarises the maximum product temperature values and cooling load variation for the different air curtain widths together with the air curtain velocity and percentage flow through the air curtain. The maximum variation of product temperature with slot width on the bottom shelf (shelf 1) was found to be around 1 °C. Increasing the air curtain width resulted in a lower mass flow rate through the perforated back panel and a higher mass flow rate through the air curtain. The percentage air flow variation through the air curtain compared to the total flow rate in the cabinet was found to be between 28% and 36% with the 35% flow rate giving best results. Therefore it can be concluded that the air distribution in the display cabinet playing important role of controlling the performance of the display cabinet and depends on the design parameters of the display cabinet such as the opening height, depth of the cabinet (Gary *et al*. 2008).

The momentum of the vertical downward flowing air curtain was found to be consumed by the entrainment process and results in vortex development, which taking place at the outer edge of the air curtain. Moreover, the air emitting from the perforated back panel tended to push the air curtain away from the shelves. Hayes and Stoecker (1969) reported that heat transfer through the AC is nearly proportional to the air curtain velocity. Therefore a 90 mm air curtain width was found to achieve the best performance for the air curtain among the considered cases. This indicated that the momentum of the air curtain is high enough to protect the entire display height, but not too high to increase the rate of heat transfer across the air curtain.

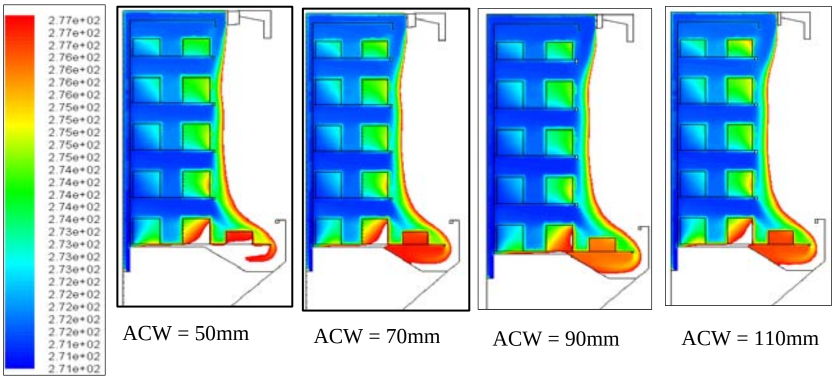


Figure 3. Product temperature contours (K) at different air curtain width (ACW)

Table 3. Product temperature and cooling load variation for different air curtain width

ACW (mm)	Sh1 (°C)	Sh2 (°C)	Sh3 (°C)	Sh4 (°C)	Sh5 (°C)	Cooling load (kW)	AC percentage flow rate (%)	ACV (m/s)	Minimum ACV Required (m/s) According to Hayes
50	4.9	3.7	3.7	3.7	3.7	5.16	28 %	0.67	0.6
70	4.8	3.6	3.5	3.5	3.7	4.80	34 %	0.55	0.5
90	3.8	3.4	3.4	3.4	3.9	4.60	35 %	0.46	0.45
110	3.9	3.5	3.4	3.4	4.1	4.73	36 %	0.41	0.4

4.3 Air Curtain Velocity (ACV)

The sealing ability of an air curtain depends on its initial momentum and the size of transverse forces, against which the air curtain is attempting to seal. The dimensionless ratio of these two forces is known as the deflection modulus (Hayes and Stoecker 1969):

$$D_m = \frac{\rho_{ac} b\, U}{9.81\ H^2\ (\rho_c - \rho_h)} \qquad (1)$$

Where:

- ρ_{ac} = density of air at air curtain outlet
- b = air curtain width
- U = initial air curtain velocity
- H = air curtain (opening) height
- ρ_c = density of air at cold side of air curtain
- ρ_h = density of air at warm side of air curtain

For application to refrigerated display cabinets, eq. (1) can be used to calculate the minimum air curtain velocity required, by replacing H (opening height) with the maximum shelf-height available in the display cabinet Xiang (2004). In order to achieve stable operation, the air curtain momentum should always be slightly above the minimum value determined by the deflection modulus. The CFD results showed that the wider air curtain widths of 90 mm and 110 mm resulted in air curtain velocities of 0.46 and 0.41 m/s respectively. The corresponding velocities from equ. 1 were 0.45 and 0.41, showing that the lower velocities still provided a good ceiling of the cabinet opening against the ambient air.

4.4 Effect of Air Curtain Discharge Angle on the Performance of the Cabinet

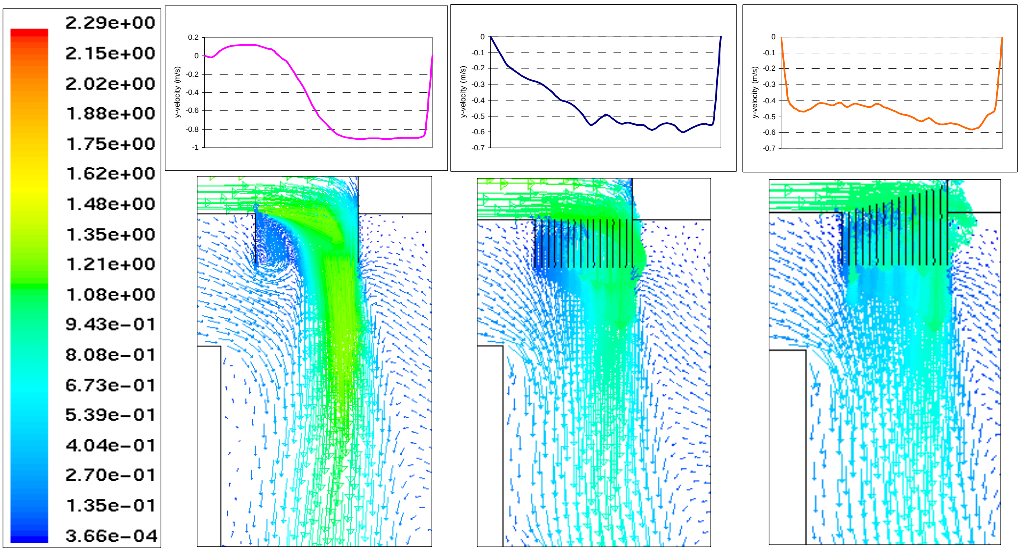
At an air curtain width of 90 mm, simulations were carried out to investigate the effect of the discharge angle of the air curtain on the performance of the cabinet. It was found that products on the top shelves were most affected by the discharge angle with a maximum temperature variation of

0.5°C for a discharge angle variation between 0° and 15°. For the cabinet geometry investigated a discharge angle between 5° and 10° was found to result in minimum cooling load, Table 4.

Table 4. Effect of the discharge angle on the cooling load

Discharge angle	Cooling load (kW)	Discharge angle	Cooling load (kW)
0°	4.57	12°	4.60
5°	4.54	15°	4.63
10°	4.56	-	-

4.5 Effect of Using Honeycomb in the Discharge Air Grille (DAG) on the Performance of the Cabinet
Velocity vectors for the cabinet with and without honeycomb are shown in Figure 4.



(a) Without Honeycomb (b) With Normal honeycomb (c) With Modified Honeycomb

Figure 4. Effect of using honeycomb on the air curtain velocity (m/s) distribution at the air curtain outlet

As it can be seen, the use of honeycomb results in a more uniform flow and lower turbulence and recirculation at the air curtain outlet. Changing the shape of the honeycomb can change the pressure drop and the velocity profile at the outlet of the DAG. In this case the modified honeycomb, Figure 4 (c) produced a more uniform discharge velocity and a thicker air curtain. This has lead to a 4% reduction in the total cabinet load compared to the discharge air grille without a honeycomb, Figure 4 (a). The impact of the discharge air velocity profile on ambient air entrainment depends on many factors and it is beyond the scope of this investigation. Ideally, though, for a single air curtain a low velocity at the outer edge of the air curtain in contact with the ambient air will lead to lower entrainment and a higher velocity at the inner edge will provide better sealing of the cabinet air from the environment.

4.6 Position of the Air Curtain Outlet relative to the Front Edge of the Top Shelf
The validated 2-D CFD model was used to investigate the effect of the DAG position in relation to the front edge of the top shelve at a constant evaporator coil air-off mass flow rate (0.18 $kg/s^{-1}m^{-1}$) and evaporator coil air-off temperature of 271 K. Figure 5 shows path lines coloured by static temperature values for three distances between the front edge of the top shelf and the inner edge of the DAG: 0, 50 and 100 mm. It can be seen that when the distance is increased, the length of the air curtain jet from the DAG to the RAG (return air grille) increases which in turn increases the area where entrainment of ambient air into the air curtain air can take place. This has lead to a longer air curtain jet from the air curtain outlet to the return air grill, which resulted in a larger area in which ambient air entrainment and mixing can take place.

The CFD results also showed that increasing the distance between the front edge of the top shelf and the air curtain outlet by 100 mm caused an increase in the cooling load by around 2% with slightly higher product temperature.

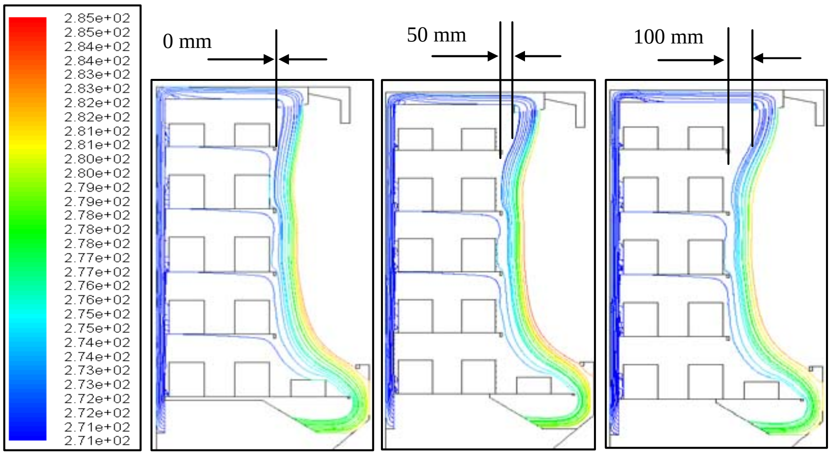


Figure 5. Path lines coloured by static temperature (K) for different positions of air curtain outlet

5. CABINET MODIFICATIONS

Based on the results from the CFD model, a new course of experiments was designed on the display cabinet. In this new course of experiments, the display cabinet was fitted with 6 evaporator fans to increase the air mass flow rate up to 0.43 kg/s. A glycol temperature of -7°C with a volume flow rate of 0.8 m³/h was found to be enough to maintain the cooling coil air-off temperature at around -2°C. The air curtain width was changed to 90 mm instead of 110 mm and the air curtain discharge angle was reduced to around 8° instead of 12°.

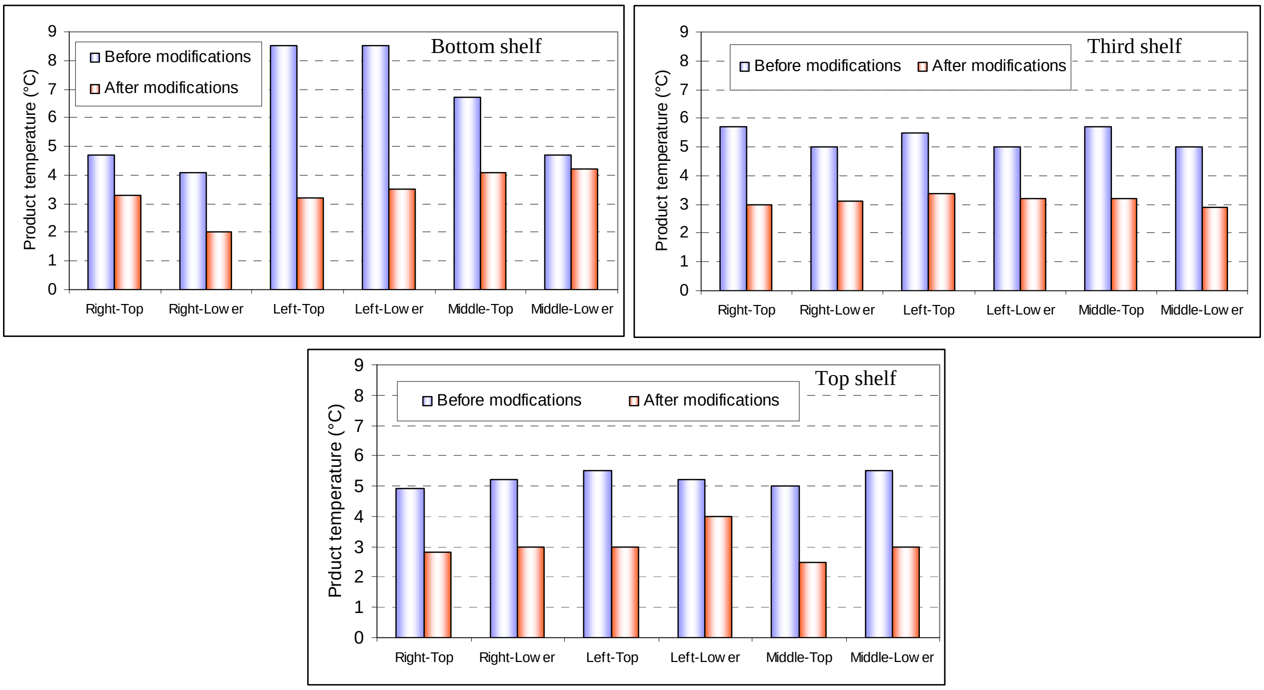


Figure 6. Product temperature variation before & after the cabinet's modifications

The display cabinet was already fitted with a honeycomb. The mass flow rate through the air curtain was controlled by controlling the perforation rate of the back-panel. Figure 6 shows the product temperature variation before and after the modifications. It is clear that the changes made to the display cabinet contributed to reducing the product temperature to within the M1 range at climate class III conditions.

6. CONCLUSIONS

The computational and experimental investigations showed that for determining important design parameters for display cabinets of sufficient length, in this case 2.5 m, the flow in the cabinet can be assumed to be 2-D. The optimum air curtain mass flow rate through the air curtain was found to be around a third of the total mass flow rate in the cabinet. The remainder of the flow is directly to the shelves from the perforated back panel. Minimising the distance between the inner edge of the discharge air grille and the front edge of the top shelve provided better protection to the air and products in the display cabinet. An air curtain slot width of 90 mm with the air curtain inclined outwards by 8° provided best performance. A honeycomb in the DAG shaped to provide better air velocity uniformity at the air curtain discharge, reduced the cabinet load by 4%.
Despite increased research activity on air curtains in recent years, there is still significant scope for further research and development to reduce air infiltration and the load of vertical multi-deck refrigerated display cabinets.

REFERENCES

1. Atkins WS. 1992, Chilled multi-deck cabinet airflow study. *Report for Safeway Stores* PLC, UK.
2. Chen Y, Chen G, Yuan X, Yuan L. 2005, Experimental study of the performance of single band air curtains for a multi-deck refrigerated display cabinet, *Journal of Food Engineering*, 69: 261-267.
3. Cortella, G., Manzan, M., Comini, G., 2001, CFD Simulation of Refrigerated Display Cabinets, *International Journal of Refrigeration*, 24: 3, 250-260.
4. D'Agoro, P., Cortella, G., Croce, G., 2006, Two- and three-dimensional CFD applied to vertical display cabinet simulation, *International Journal of Refrigeration*, 29: 178-190.
5. Foster AM, Madge M, Evans JA. 2005, The use of CFD to improve the performance of a chilled multi-deck retail display cabinet, *International Journal of Refrigeration*, 28: 698-705.
6. Gray I, Luscombe P, Mclean L, Sarathy CSP, Sheahen P, Srinivasan K. 2008, Improvement of air distribution in refrigerated vertical open front remote Supermarket display cases, *International Journal of Refrigeration*, 31: 902-910.
7. Hadawey A. 2006, *Design of chilled food display cabinets for better temperature integrity and longer product shelf life*, PhD Thesis. Brunel University.
8. Hayes FC and Stoecker WF. 1969, Heat transfer characteristics of air curtain, *ASHRAE Transactions*, 153-167.
9. *ISO* 23953-2, 2005, Refrigerated Display Cabinets-Part 2: Classification, requirements and test conditions.
10. Yu K, Ding G, Chen T., 2007, Simulation of air curtains for vertical display cases with a two-fluid model, *Applied Thermal Engineering*, 27: 2583-2591.
11. Stribling D, Tassou SA, Marriot D. 1997, A two-dimensional CFD model of a refrigerated display case, *ASHRAE Transactions*, 103: 88-94.
12. Maki A., Tassou SA. 2005, Design optimization evaluation methodology of vertical multi-deck display cabinets, *TTR International Conferences*, Vicenza.
13. Xiang W. 2004, *Investigation and optimization multi-deck display cabinet and retail store environment,* PhD Thesis. Brunel University.