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Title:
The use of CFD for predicting buoyant and forced convection flows occurring in food refrigeration

THE USE OF CFD FOR PREDICTING BUOYANT AND FORCED CONVECTION FLOWS OCCURRING IN FOOD REFRIGERATION

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A dissertation submitted to the University of Bristol in accordance with requirements of the degree of Doctor of Philosophy in the Faculty of Engineering.

Food, Refrigeration and Process Engineering Centre

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Abstract

Being able to maintain the temperature of food is of vital importance to ensure optimal food quality, safety and shelf life. Most unwrapped food and all types of wrapped foods are stored in large refrigerated rooms with forced air circulation. A reduction in the average storage temperature can significantly increase the length of retention and quality of many chilled foods. Air infiltration through entrances can account for more than half the total heat load for refrigerated stores.

Chilled foods are often transported to supermarkets where they are then displayed in open fronted refrigerated display cabinets. These cabinets are one of the weakest links in the chilled food chain with large temperature ranges between products in the same cabinet. This range in temperature causes problems for food manufacturers when defining shelf life and results in shelf lives that are either unduly cautious or potentially risky.

The aim of this thesis was to use CFD to solve problems encountered in the food/refrigeration industry. It describes the use of CFD for two separate problems (cold store door opening and open fronted refrigerated display cabinets) which are caused by the same physical phenomena (natural convection between air at different temperatures).

CFD predictions of unprotected entrances give a significant improvement in accuracy over the fundamental analytical equations. Factors such as the extent of the ambient domain that needs to be modelled are an important consideration in gaining accurate predictions. A 2D CFD model was able to predict the effectiveness of an air curtain; however, the predictions were higher than measured. A 3D CFD model was able to provide a better prediction of effectiveness and optimum jet velocity than the 2D model. This thesis highlights the importance of setting up a cold store air curtain properly, if it is to work at its optimum efficiency.

Dedication and acknowledgements

I would like to thank the government bodies (Department of Trade and Industry, Department of the Environment, Food and Rural Affairs) and industrial companies who have supported my work over the years.

I would also like to thank all current and former members of the Food Refrigeration and Process Engineering Research Centre who have encouraged me and shown an interest in my work.

Also with thanks to Sara who kindly proof read my thesis.

Declaration

I declare that the work in this dissertation was carried out in accordance with the regulations of the University of Bristol. The work is original and no part of the dissertation has been submitted for any other academic award. Any views expressed in the dissertation are those of the author.

SIGNED: ………………………………… DATE: ………………………………

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Papers submitted for the award of a PhD by published work

Academic Journal Papers (reproduced at the back of the thesis)

1. **Foster, AM** & Quarini, GL. Using advanced modelling techniques to reduce the cold spillage from retail display cabinets into supermarket stores to maintain customer comfort. Journal of Process Mechanical Engineering, Proceedings of the Institute of Mechanical Engineers, 215(E);29-38. 2001.

   Contribution of A. Foster: Sole CFD modeller, shared experimental measurements, shared author.

2. **Foster, AM**, Barrett, RV, James, SJ & Swain, MJ. Measurement and prediction of air movement through doorways in refrigerated rooms. International Journal of Refrigeration, 25(8);1102-1109. 2002.

   Contribution of A. Foster: Sole CFD modeller, shared experimental measurements, main author.

3. **Foster, AM**, Swain, MJ, Barrett, RV. & James, SJ. Experimental verification of analytical and CFD predictions of infiltration through cold store entrances. International Journal of Refrigeration, 26(8); 918-925. 2003.

   Contribution of A. Foster: Sole CFD modeller, shared experimental measurements, main author.

4. **Foster, AM**, Madge, M & Evans, JA. The use of CFD to improve the performance of a chilled multi-deck retail display cabinet. International Journal of Refrigeration, 28(5); 698-705. 2005.

   Contribution of A. Foster: Sole CFD modeller, main author.

5. **Foster, AM**, Swain, MJ, Barrett, R, D'Agaro, P & James, SJ. Effectiveness and optimum jet velocity for a plane jet air curtain used to restrict cold room infiltration. International Journal of Refrigeration, 29(5); 692-699. 2006.

   Contribution of A. Foster: Sole CFD modeller, main author.

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   Contribution of A. Foster: Sole CFD modeller, main author.

Conference contributions

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   Contribution of A. Foster: CFD modeller, shared author.

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10. **Foster, AM**. The benefits of computational fluid dynamics for modelling processes in the cold chain (Clarence Birdseye prize winner). Proc. of the 20th International Congress of Refrigeration, Sydney, Australia. pp 19-24 September 1999.

11. **Foster, AM** & Swain, M.J. Using computational fluid dynamics modelling to predict airflow in and through the entrances of cold storage rooms. Proceedings of the International Institute of Refrigeration Commission C2 Rapid Cooling of Food Conference. Bristol 28-30th March 2001. pp 287-293.

    Contribution of A. Foster: Sole CFD modeller, main author.

12. Evans, JA, Madge, M & **Foster, AM**. The use of CFD to improve the performance of a chilled multi-deck retail display cabinet. Commercial Refrigeration, IIR International Conference, Vicenza, Italy, 30-31 Aug 2005. pp 63- 70.

    Contribution of A. Foster: Sole CFD modeller, shared author.

Overview and context for the work submitted

THE USE OF CFD FOR PREDICTING BUOYANT AND FORCED CONVECTION FLOWS OCCURRING IN FOOD REFRIGERATION

Keywords

Computational fluid dynamics (CFD), refrigeration, air curtains, infiltration, display cabinets.

1. Nomenclature

A	cross sectional area of entrance, m2
b	thickness of door frame, m
g	acceleration due to gravity, 9.81 m s^{-2}
$K_{f,L}$	correction factor, dimensionless
h_n	neutral height, m
H	height of entrance, m
p_o	neutral pressure, Pa
t	time, s
T_o, T_i	temperature outside and inside colds store, ºC
u	velocity, m.s^{-1}
Greek letters	
ρ_i, ρ_o, ρ_{avg}	density inside and outside cold store and average, kg m-3

2. Introduction

In the UK, retail sales of frozen and prepared chilled food were worth £11.8 billion p.a. (source, British Food Federation 2006 and Chilled Food Association 2005). Being able to maintain the temperature of this food is of vital importance to ensure optimal food quality, safety and shelf life [13]. Most unwrapped meat, poultry, fruit and vegetables and all types of wrapped foods are stored in large refrigerated rooms with forced air circulation. Separate rooms are used for the storage of chilled or frozen foods.

A reduction in the average storage temperature can significantly increase the length of retention and quality of many chilled foods. For example, every 5ºC rise in storage temperature of beef above 0ºC will approximately half the storage time that can be achieved. However, this is not always the case, for example, for oranges the optimal storage temperature is 12ºC, with the storage times reducing at temperatures warmer and colder than this value [14].

Improvements such as the introduction of state of the art refrigeration systems with advanced controls can help reduce energy usage and temperature control [15]. However, in any refrigerated store, entrances are required for loading and unloading raw materials or finished product and these entrances are a major source of heat infiltration. Air infiltration through entrances can account for more than half the total heat load for refrigerated stores [16].

Chilled foods (meat, ready meals, dairy products) are often transported to supermarkets where they are then displayed in open fronted multi-deck refrigerated display cabinets. These are the primary means of displaying chilled food in a supermarket.

Open display cabinets are one of the weakest links in the chilled food chain [17] and it has been shown that mean food temperatures between chilled multi-deck cabinets can range from -1°C to 16°C [18]. This range in temperature causes problems for food manufacturers when defining shelf life and results in shelf lives that are either unduly cautious or potentially risky.

When there is no physical barrier between the warm environment and refrigerated space, natural convection or buoyant flow is the primary infiltration mechanism. This is the case when doors are open continuously during busy loading and unloading periods, in cold stores and for open fronted cabinets where there are no doors. Air curtains are used in multi-deck display cabinets and sometimes over cold store doors to restrict this infiltration whilst not impeding movement between the two environments.

In 1994 when the initial work presented in this thesis started, CFD was relatively commonplace in the aerospace, chemical and motor industries. It was, however, in its infancy in the food and refrigeration industry. The aim of the work presented in this thesis was to use CFD to solve problems encountered in these industries. This thesis describes the use of CFD for two separate problems which are caused by the same physical phenomena (natural convection between air at different temperatures).

This thesis is for a PhD by publication and the publications covered work started in 1994 and finished in 2005, although the publication dates spanned from 1996 to 2007. Sections 2, and 3 are an explanation of the physical phenomena of natural convection through openings and the use of air curtains to alleviate it. Section 4 is an explanation of the benefits of CFD. Section 5 presents a coherent summary of the findings presented within the submitted journal papers (included in the abstract). The conference publications are not summarised in Section 5 as they generally come from the journal papers.

The work in this thesis started during a project titled ‘Demonstration of CFD to the food industry’ funded by the Department of Trade and Industry (DTI). This project was aimed at learning how CFD can be used in the food/refrigeration industry and only produced conference papers, not journal papers.

The cold store application publications [2,3,5,6] were a progression of measurements and modelling on the same cold store. The project titled ‘Reducing the infiltration in refrigerated storage rooms using CFD’, was funded by the Ministry of Agriculture, Fisheries and Food (MAFF). This work used a variety of different methods to verify the CFD modelling; array of mini vane anemometers, laser Doppler anemometry (LDA) and tracer gas measurement.

The initial display cabinet work [1] funded by MAFF and titled ‘Ventilation and containment’ was chronologically the first work in this thesis. The second display cabinet paper [4] was carried out much later and was commercially funded.

Figure 1 shows a map of the projects and publications presented in this thesis.

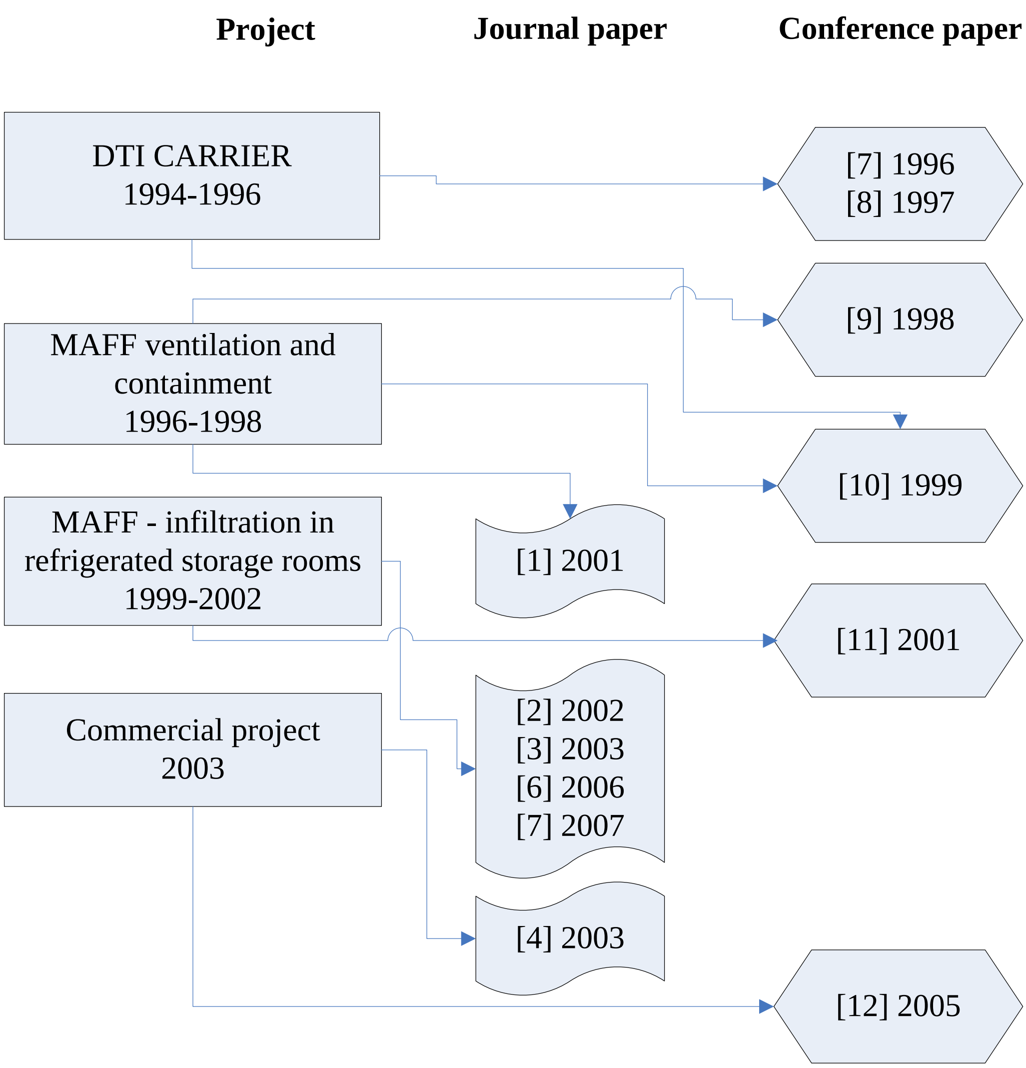


Figure 1. Projects and their respective outputs in terms of journal and conference papers.

3. Natural convection through openings

The theory of natural convection through an opening has been developed by a number of authors. Natural convection through an opening is caused by the temperature difference and thus density difference between the air inside and outside the room. This effect is commonly known as the stack effect.

Eighty years ago Emswiler [19] expressed the basic theory for natural convection through openings in a partition, separating fluids at different densities. His investigation concentrated on multiple openings. His theory was based on the Bernoulli equation for ideal flow and introduced the concept of the neutral level. This is the height at which the pressure is the same either side of the partition.

The following models are described in full in Table 1.

Brown and Solvasson [20] developed a theory of natural convection through single vertical rectangular openings in partitions. They showed that a pressure profile is developed in the opening caused by the difference in density due to temperature difference between the environments on either side of the partition. This velocity profile generated from this pressure distribution is shown in Figure 2. They assumed that the neutral height was half the height of the entrance.

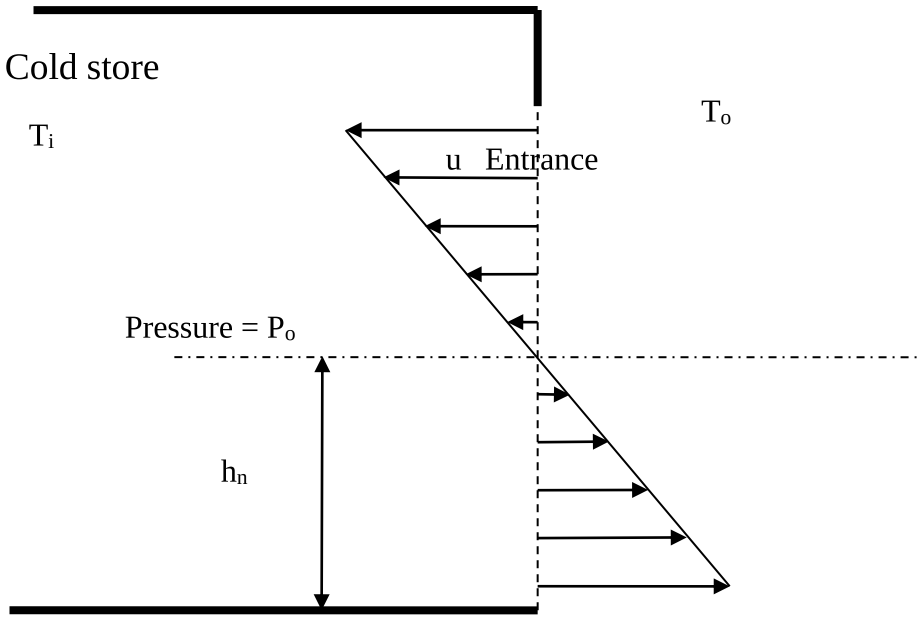


Figure 2. Schematic showing natural convection through a refrigerated room entrance.

Tamm [21] improved on this model, calculating the height of the neutral level and using inside and outside densities for inflow and outflow respectively, where appropriate instead of an average value. Fritzsche and Lilienblum [22], who conducted experiments using vane anemometers, added a correction factor to Tamm's equation to take into account the contraction of the flow, friction and thermal effects.

Fritzsche and Lilienblum's equation assumed that the volume flow rate into and out of the room were the same. This is only the case if the air entering the room does not cool. This may be a good approximation for a small room, where infiltration through the door causes the air in the refrigerated room to rise in temperature. If the room is large and the air entering the room cools to the refrigerated room air temperature, the mass flow rate into and out of the room will be the same but not the volume flow rate. Gosney and Olama [23] provided an equation for constant mass flow rate. By fitting measurements with their model they provided a different correction factor.

Pham and Oliver [24] conducted experiments on air flow through refrigerated room doors and produced a factor of 0.68 which should be applied to Tamm's equation to fit their experimental data. Jones and Becke [25] investigated moisture transfer through openings, and showed that the mass transfer of water vapour could be adequately described by the air convection currents at the opening, which have a much greater effect than diffusion.

Table 1. Analytical models of infiltration rate through an open door due to natural convection.

Brown and Solvason (1963) $I = 0.343\ A\ (gH)^{0.5}\left[\frac{(\rho_i - \rho_o)}{\rho_{avg}}\right]^{0.5}\left[1 - 0.498\left(\frac{b}{H}\right)\right]$

Tamm (1966) $I = 0.333\ A\ (gH)^{0.5}\left[\frac{(\rho_i - \rho_o)}{\rho_i}\right]^{0.5}\left(\frac{2}{1+\left(\rho_o/\rho_i\right)^{0.333}}\right)^{1.5}$

Fritzsche and Lilienblum (1968) $I = 0.333\ K_{f,L}\ A\ (gH)^{0.5}\left[\frac{(\rho_i - \rho_o)}{\rho_i}\right]^{0.5}\left(\frac{2}{1+\left(\rho_o/\rho_i\right)^{0.333}}\right)^{1.5}$

$K_{f,L} = 0.48 + 0.004(To-Ti)$

Gosney and Olama (1975) $I = 0.221\ A\ (gH)^{0.5}\left[\frac{(\rho_i - \rho_o)}{\rho_i}\right]^{0.5}\left(\frac{2}{1+\left(\rho_i/\rho_o\right)^{0.333}}\right)^{1.5}$

Pham and Oliver (1983) $I = 0.226\ A\ (gH)^{0.5}\left[\frac{(\rho_i - \rho_o)}{\rho_i}\right]^{0.5}\left(\frac{2}{1+\left(\rho_o/\rho_i\right)^{0.333}}\right)^{1.5}$

4. Air curtains

Air curtains can be used to form barriers to heat, moisture, dust, odours, insects etc. without impeding movement between two environments as a door would. The origin of air curtains dates back to a patent applied for by Van Kennel in 1904 and they have been popular for around 50 years.

The primary purpose of the air curtains investigated in this work is to provide a barrier to heat and moisture. These air curtains are effectively a plane turbulent jet of air forming a barrier between the refrigerated environment and the warm ambient air.

The ability of air curtains, as well as other devices (e.g. PVC strip curtains and air lock vestibules) to reduce infiltration is defined as the effectiveness and is derived by the following equation;

$$E = \frac{Q_b - Q_a}{Q_b}$$

where E is the effectiveness, Q_b is the infiltration without and Q_a is the infiltration with the infiltration reducing device.

An effectiveness of 1.0 means that infiltration is totally eliminated by the device, 0 means that there is no effect on the infiltration and a negative value indicates that the infiltration has been made worse.

4.1 Cold store application

In the case of cold store air curtains, the fan unit is usually above the door blowing a jet vertically downwards, although air curtains which blow sideways do exist. Some air curtains re-circulate refrigerated air via a return duct but it is simpler and more common not to do so. The simplest and most common type blows ambient air which is dissipated as it impinges on the floor. The effectiveness of air curtains has been shown to vary considerably. Both Micheal [26] and Foster et al [**5**] have shown that incorrect specification, installation, set-up and adjustment can have an adverse effect on their effectiveness.

Figure 3 shows a schematic of a simple non-recirculating air curtain. Close to the air curtain nozzle is the flow development region. Inside the potential core the velocity of the jet remains constant, outside of the potential core is the turbulent mixing region. The length of the potential core is approximately 8 to 12 times the jet thickness. In the fully developed region the velocity of the jet on the centre-line is decreasing. Close to the floor is the impinging or recompression zone, here the jet experiences a pressure from the floor and turns from a vertical 'free jet' to a horizontal 'wall jet'.

There are optimum conditions for the momentum of the air curtain jet; too low a velocity and the air curtain will not be able to counteract the stack forces, too high and the increased entrainment caused by the higher velocity, more turbulent jet will cause more infiltration.

4.2. Display cabinet application

For the sake of simplicity a refrigerated display cabinet can be considered as a small cold store with no door to provide a barrier between the refrigerated and ambient environments. Instead a recirculating refrigerated air curtain provides this barrier.

In the majority of multi-deck retail display cabinets the evaporator (cold heat exchanger) is in the base of the cabinet, (although it is becoming more common for them to be in the rear duct) and fans draw air through the evaporator and up a duct at the rear of the cabinet. Air exits the duct through holes or slots in the cabinet rear grille and also through a slot or honeycomb grille placed at the front of the cabinet canopy, termed the discharge grille (Figure 4). The purpose of the discharge grille is to create a vertical air curtain (jet) from the front of the cabinet canopy to a grille placed at the front of the cabinet well (termed the return grille). As the air from the curtain is colder than the surrounding air it will fall due to buoyancy.

As the air curtain falls from the cabinet discharge grille, the curtain entrains cold air from inside the cabinet (entered through the rear grille) and warm air from the environment. The entrainment of warm air from the environment causes the air curtain to become warmer as it passes down the cabinet. Entrainment causes the mass flow rate of the air curtain to increase as it moves down the front of the cabinet. Due to conservation of mass flow, not all of the air in the air curtain will be taken away by the return grille, a significant portion will overspill onto the floor. As this air will be colder and denser than the ambient air, it will sit on the floor, causing a customer comfort issue which is commonly termed the 'cold feet effect'.

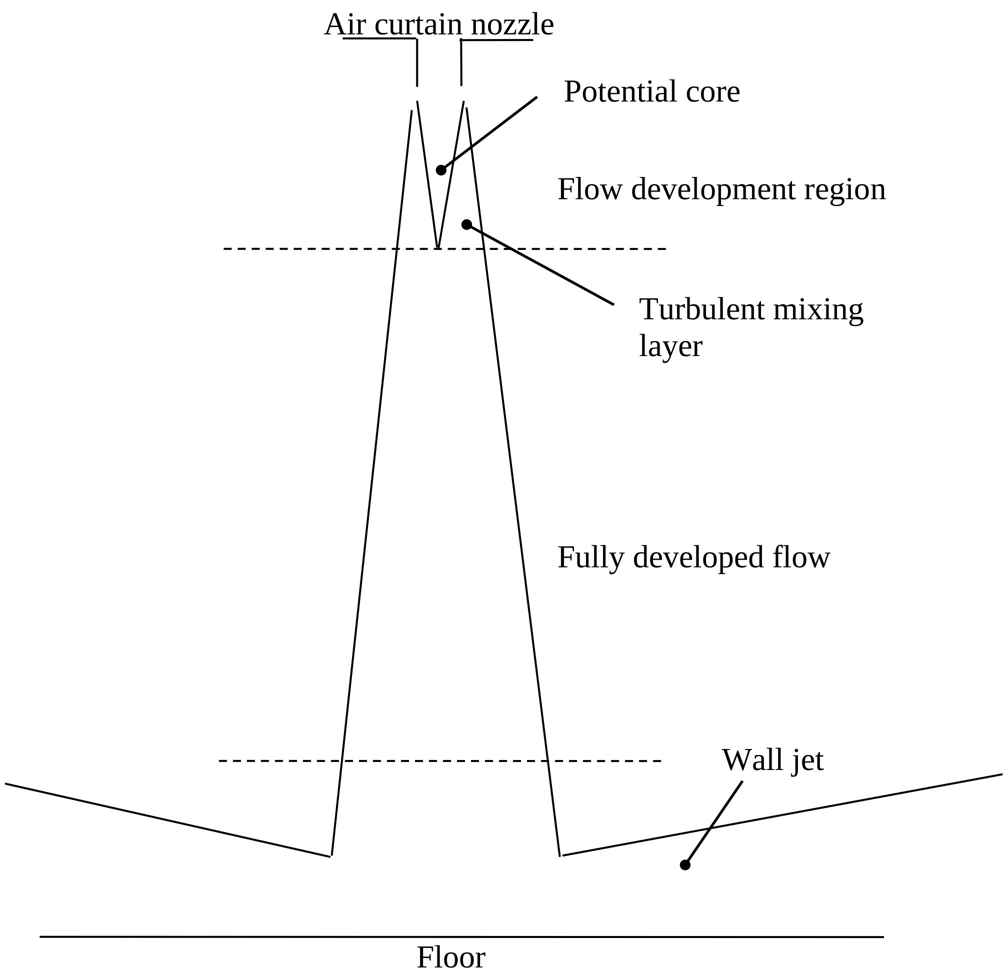


Figure 3. Schematic of non-recirculating plane air curtain.

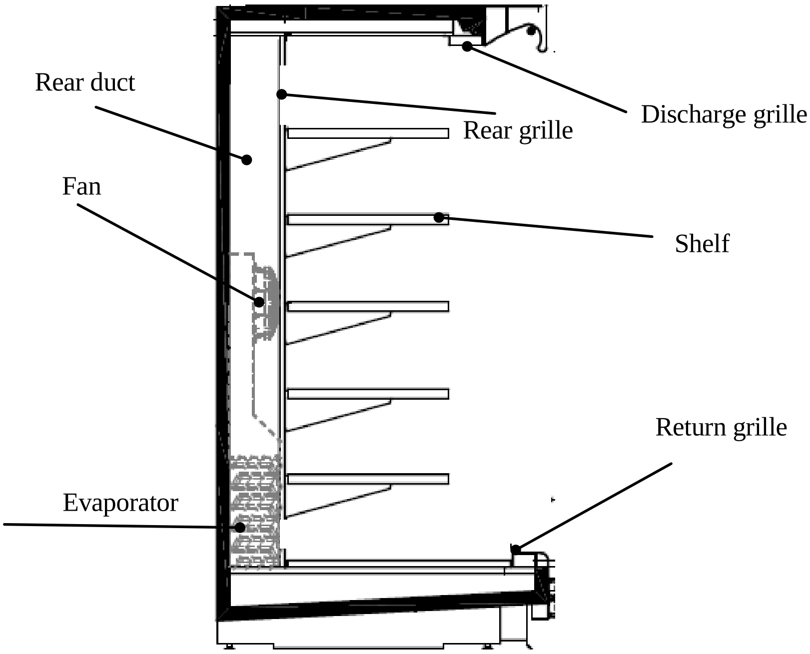


Figure 4. Vertical section of multi-deck display retail cabinet with the evaporator fan in the rear duct.

5. Computational fluid dynamics

Computational fluid dynamics (CFD) modelling has become commonplace in many engineering industries for example, the aerospace, chemical and motor industries. It is also becoming more widely used by the food and refrigeration industry.

As the type of flow differs between different industries, the CFD code may also differ. The aerospace industry is concerned with high Mach number flows which are likely to be isothermal. The food and refrigeration industry are interested in low Mach number flows, where heat transfer is the primary purpose and the convection is more likely to be buoyancy driven.

Due to the speed with which a variety of scenarios can be modelled, CFD provides the ideal means to both understand the natural and forced convective flow around refrigerated equipment and to determine which design options would provide improvements to this equipment. These improvements may be required to meet temperature control legislation and energy efficiency requirements.

There are a number of commercially available CFD codes that can be used; four popular codes are ANSYS CFX, FLUENT, STAR-CD and PHOENICS. Modern codes are easier to use than previous versions, features such as automatic mesh generation can reduce the time taken to create a mesh from days to seconds. Moreover, today's standard desktop PCs are easily capable of running these models.

In this thesis ANSYS CFX initially ran on UNIX based workstations and later Intel based PCs is used for all of the CFD modelling.

The equations which are solved by the CFD code are described by Foster et al [**6**].

6. Contributions to knowledge in the field

This section summarises my contribution to knowledge in the field of predicting buoyant and forced convection flows occurring in food refrigeration. The publications presented in this thesis represent an evolution of knowledge over 11 years. Later publications built on the knowledge gained previously and used more powerful hardware and better software, improving on the complexity and accuracy of the predictions. The findings of this work have been compared with work by other authors (references from this thesis are in bold to better allow this comparison).

The contribution to knowledge in this field has been divided into two discrete sections: cold store and display cabinet applications. Both sections summarise academic journal papers presented in the Appendix (4 papers for cold store and 2 papers for display cabinet). Six conference papers are also presented in the Appendix, however, these are not summarised in this section as they tend to be a summary of the journal papers.

A brief summary of the evolution of these journal papers is shown below.

1. CFD model of whole supermarket was too large to model accurately. Smaller models of chilled aisles were able to predict the effect of different ventilation systems to varying degrees of accuracy.

2. CFD model of air movement through unprotected doorway, verified against conventional and LDA. It predicted the shape of the velocity profile well, although it under-predicted the maximum velocity by 0.1 $m.s^{-1}$.

3. Basically the same CFD model from [2] verified against tracer gas infiltration measurements and compared to analytical models. The CFD model was more accurate than fundamental analytical models, but less accurate than those based on a semi-empirical approach.

4. CFD was used to model and suggest changes to improve a refrigerated display cabinet within commercial timescales. Changes were made which reduced power consumption and product temperatures within the cabinet

5. Air curtain fitted to doorway from papers [2] and [3]. Tracer gas measurements showed the importance of setting up the air curtain to give optimum effectiveness. The

2D CFD model showed how the effectiveness was related to shape of the jet, but the model over predicted the effectiveness of the air curtain.

6. Same measurement data from [5]. A 3D CFD model predicted effectiveness 0.1 to 0.15 lower than measured. Predictions showed that the flow cannot be considered as 2D.

6.1. Cold store applications

Many authors have used CFD to predict heat and mass transfer inside refrigerated rooms [27, 28, 29, 30] but it been used far less to predict natural convection through cold store entrances. Many of the investigations of natural convection through openings are not for cold storage rooms but for other types of building, e.g. warehouses, offices, retail outlets etc. The work reported in this thesis has provided the majority of published work on infiltration through cold store entrances.

All the work in this section (5.1. Cold store applications) was carried out on a purpose built refrigerated test room of internal dimensions 5 x 6 x 4 m high [**2**]. The room had a large single opening, 2.3 wide x 3.2 m high. The room was constructed within a large hall to avoid wind effects and to keep the ambient air temperature steady.

5.1.1. Velocity through unprotected doorway

A CFD model was created of the test room and a volume outside using CFX 5.4 (ANSYS) on a Pentium 3 PC running at 650 MHz with 256 Mb RAM.

The initial conditions of the air were zero velocity, -20°C inside the room and 20°C outside the room. All boundaries were modelled as non slip walls where the velocity of the fluid at the wall boundary was set to zero. A full buoyancy model was not used, instead the Boussinesq model was used. Turbulence of the air was modelled using the k-ε turbulence model. Air flowing out through the door was a transient effect and there was no steady state. The model was run for a period 30 seconds from the moment the door was opened and a solution obtained every second.

To simplify the model, a number of assumptions were made. These were:

- There was no heat flow through the walls of the test room.
- The test room had no thermal mass.
- Humidity had no effect on the flow rate through the door (it does, however, have an effect on heat transfer through the door).
- The evaporator had no effect on the air flow.

- How the door was opened did not effect the air flow through it.
- The simplification of outside room conditions had no effect.
- The room was leak proof i.e. air can only move through the entrance.
- The initial conditions of constant temperature and velocity inside and outside the room before the door is opened were true.

The geometry was discretised with a tetrahedral mesh. The mesh size was 21 626 grid nodes.

Verification measurements were conducted using vane and laser Doppler anemometry. An array of 16 mini vane anemometer heads (diameter 25 mm) (2 columns x 8 vanes) on movable frames were used to determine airflow through the entrance. The advantage of this system was that it was a cheap way of measuring transient air velocity in multiple positions at the same time. However, the vanes were not accurate at low velocity (below 0.3 $m.s^{-1}$ the vanes would intermittently stop spinning giving a reading of 0 $m.s^{-1}$).

LDA was used to measure the velocity closer to the walls than was practical using vane anemometry. The LDA used was a 3-component, fibre-optically coupled system using a 5 watt Argon - Ion laser, with precision three axis traversing gear.

Both vane anemometer measurements and CFD predictions showed that the air flow decreased with time. Velocities measured by the top and bottom vanes and predicted velocities in the same positions are shown in Figure 5 for a period between 10 and 30 s post door opening. The predicted velocity into the room at the top and its decay with time is well within the repeatability of the measurements. The predicted velocity out of the room at the bottom was 0.3 to 0.5 $m.s^{-1}$ lower than measured. Both methods showed that air flows out (positive) of the bottom of the entrance and in (negative) through the top. The highest velocity out of the room was near the floor and the highest velocity in was close to the top.

The velocity profile with height, a third of the width of the entrance from the left hand end after 20 s, is shown in Figure 6. The shape of the profile was well predicted by the CFD. However, the CFD under predicted the velocity of the air leaving the chamber at the bottom of the opening.

LDA measurements were made along a vertical and a horizontal line, these measurements were compared to CFD predictions and are shown in Figure 7. CFD predictions of the velocity profile along the vertical line was accurate, however predictions of the horizontal line were less so. The reason for this was that there were not enough grid cells around the entrance for the CFD to capture the vena contracta caused by the door sides. The vena contracta is caused at a sudden contraction where the flow contracts to a diameter smaller than the contraction. This vena contracta can be seen from the LDA measurements as a region of zero flow near the sidewall (Figure 7 bottom chart). However, due to the coarseness of the mesh in the entrance, the CFD model predicted higher velocities near the side of the door.

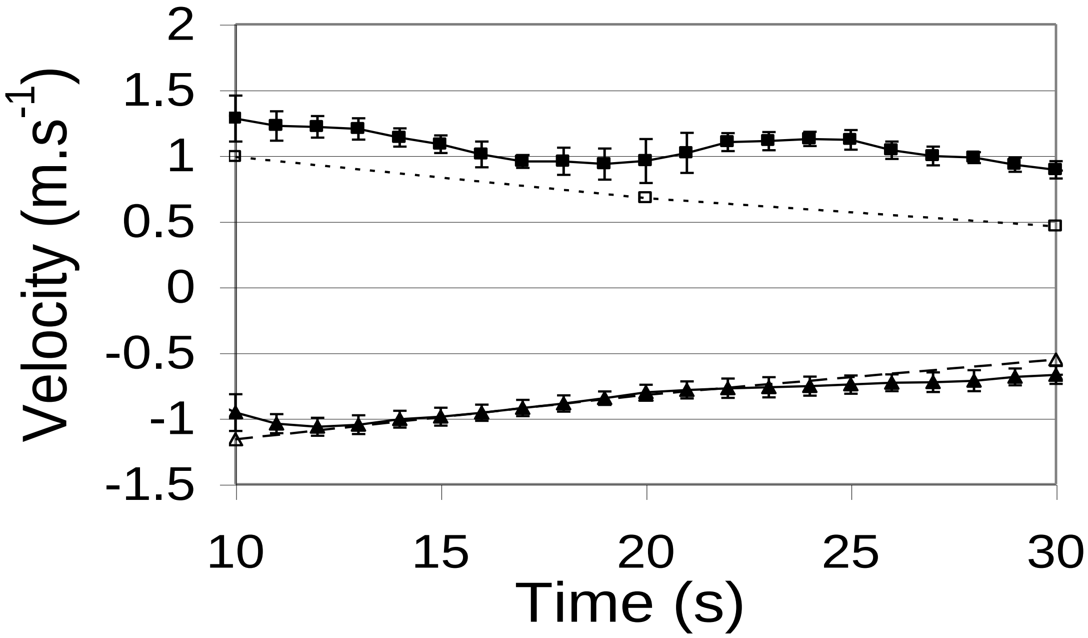


Figure 5. Velocities measured by both the top (triangular marker) and bottom (square marker) vane anemometer (mean of six replicates, error bars represent ±1 standard deviation) and predicted velocities (dashed lines) in the same positions (room temperature -20°C) during a period 10 to 30 s after opening. A positive velocity is out of the room

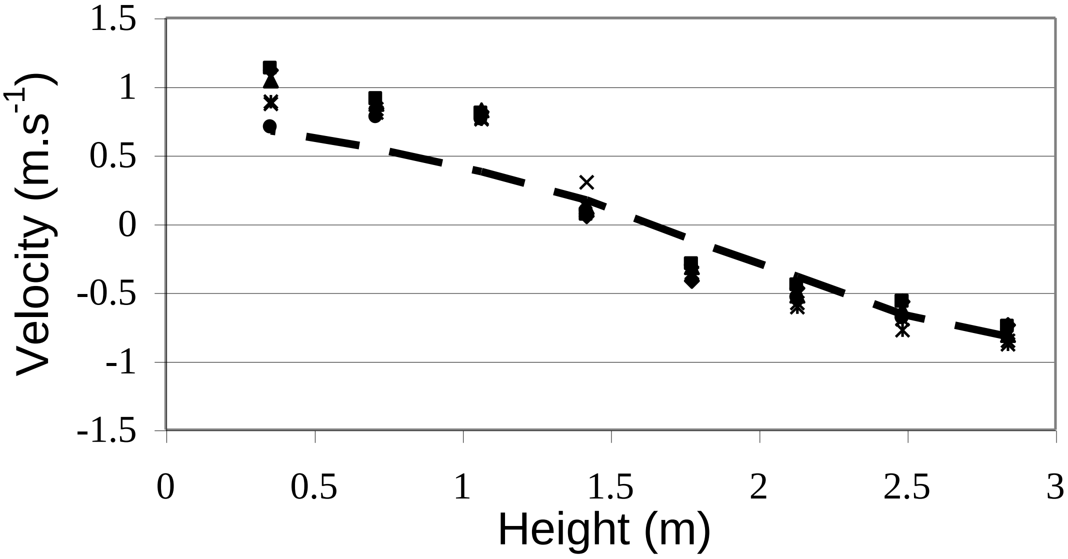
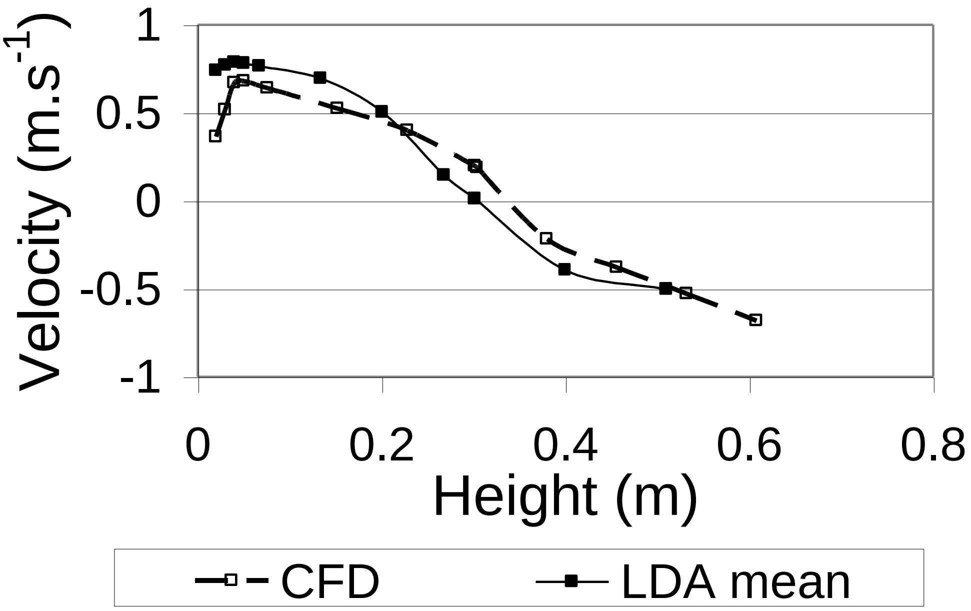


Figure 6. Velocities measured inside the large entrance 20 s post door opening at each anemometer position (6 replicates) plotted against height. The CFD predicted velocity profiles are represented by the dashed lines.



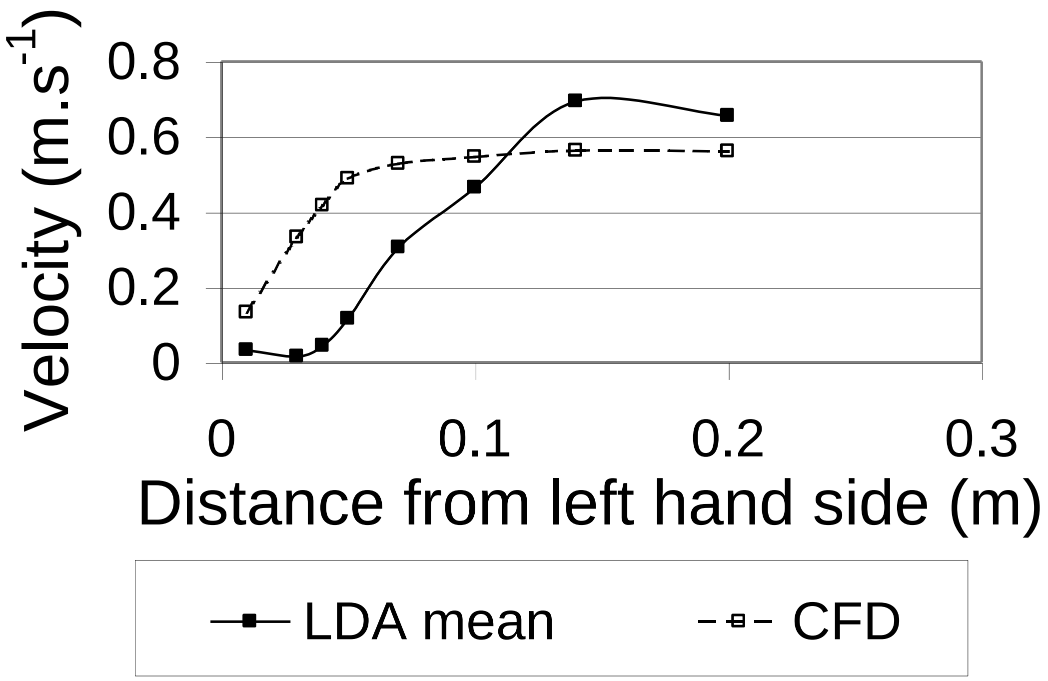


Figure 7. Velocities measured using LDA and predicted using CFD through the small entrance against height. Vertical line 0.143 m from the left hand edge of the entrance (top chart). Horizontal line at a height of 0.134 m (bottom chart)

6.1.2. Infiltration through unprotected doorway

The CFD model used in the preceding section was used to predict infiltration rates through the entrance **[3]**. CO_2 was used as a tracer gas to measure the rate of infiltration against the length of time the door was open and the resulting data were compared to the CFD predictions (Figure 8). To test whether the closeness of the outer boundary of the model to the cold store had an effect on the results, two CFD models were used, one which had the domain boundary 3 m from the outside of the walls of the cold room and one with boundaries 6 m away. Both CFD models predicted a similar infiltration rate to that measured. However, both models, especially the model without the extended boundary predicted a reduction of infiltration rate with time, which was not apparent in measurements. The reason for this was probably due to the region outside the cold store reducing in temperature faster (and therefore reducing the driving force) in the model than in reality, because the region outside was finite in the model and effectively infinite in reality.

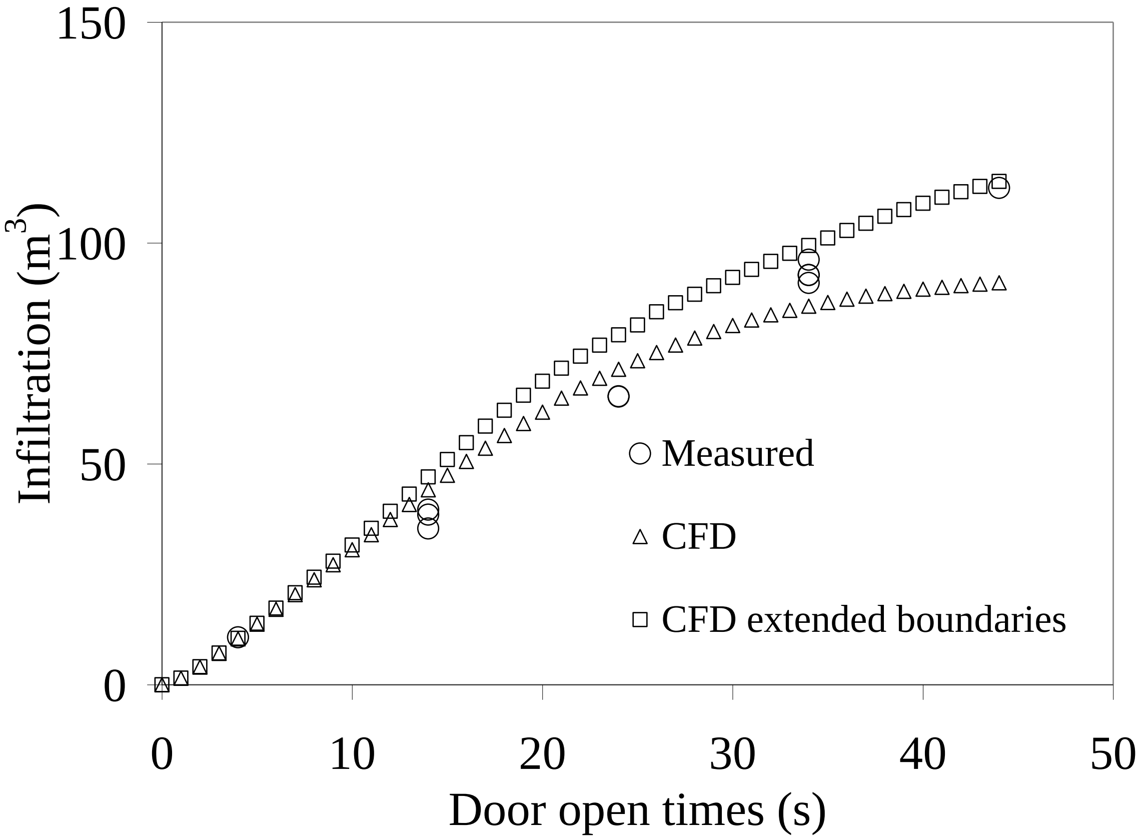


Figure 8. Measured and predicted infiltration through the 2.3 m wide door for different door opening times. The CFD predictions are for both the standard and extended boundary models.

The predictions also showed that infiltration varied with time and that the predictions could be split into three separate stages. The first stage was the lag stage covering the time required before the flow became fully developed. This was followed by a steady state stage where there was a constant flow rate through the entrance. The final stage was the tail off stage, where the temperature difference (driving force) between the cold store and the surroundings was reducing.

The lag stage was predicted to be between 0.3 and 1.6 s for the different models. This lag stage has been measured by Azzouz et al [31] to be of the order of 1.5 s. For long door opening times of 30 s, a 1.5 s lag time will reduce the infiltration by only 5% compared to no lag, while for shorter door opening periods of, for example 10 s, it will reduce the infiltration by 15%.

The author compared the analytical models described in Table 1 to measurements and concluded that the Brown and Tamm models substantially over predict the infiltration for all of the measurements (between 52% and 123% over prediction). Tamm's modified model predicted the measurements much more closely than the original, (it was within the experimental error for a cold store at 0°C). Taking all of the experiments into account, the

Gosney model performed best (maximum of 39% over prediction) followed by the Fritzsche model (maximum of 43% over prediction).

6.1.3. Two dimensional model of air curtain

There have been many academic studies using CFD to simulate the air curtains used for refrigerated display curtains. Fewer studies have been carried out on air curtains used on buildings.

A 1.0 m long air curtain with a 30 mm slot was fitted centrally above the door on the outside of the cold store [**5**]. For this study a 2D model was created to allow accurate prediction of the narrow (30 mm) air curtain jet whilst still allowing all of the cold store and some of its surroundings to be modelled. The 2D model was created using 3D finite volume code and by fixing only one numerical mesh cell over the width of the domain and applying symmetrical boundary conditions to the faces at either side. The domain of the model contained the volume inside the cold store for the width of the air curtain and a much larger volume outside the cold store, to provide a source of warm air for exchange. The volume outside the cold store was between 4 and 8 times larger than the volume inside the cold store. This gave a good compromise between accuracy and numerical speed.

The jet from the air curtain was modelled as an inlet into the domain with a constant velocity, temperature and turbulence intensity. The magnitude of the inlet velocity was varied from 0 to 18 $m.s^{-1}$ and the direction was taken to be normal to the boundary. The temperature of the jet was set at 20°C, which was the initial temperature of the ambient and the turbulence intensity was 10%. An opening boundary condition was set at the top side of the air curtain, removing an identical mass of air that entered from the inlet.

The numerical mesh was at its finest (8 mm) at the entrance of the door and around the air curtain nozzle. This was in order to accurately resolve the large shear created by the air jet at the nozzle exit. The total number of grid nodes for the problem was 42 000 which was a similar value to the previous 3D model. This was because of the small elements required to resolve the thin air curtain jet.

The hardware used to run the model was a Viglen Genie PC with an Intel Pentium 4 processor running at 1.6 GHz with 1 Gb RAM.

The CFD model was used to predict the effectiveness of the air curtain at differing jet velocities, this is shown against measured values in Figure 9. The maximum predicted effectiveness (0.84) was higher than that measured (0.72). The air curtains measured effectiveness was still increasing when it reached the maximum obtainable velocity (18 $m.s^{-1}$) of the air curtain. The optimum jet velocity (velocity at which maximum effectiveness is obtained) was predicted (10 $m.s^{-1}$) to be lower than was measured (> 18 $m.s^{-1}$). 2D models tend to predict a higher effectiveness than measured, as end effects allow air to leak around the edge of the air curtain, causing increased infiltration. Also a higher measured optimum velocity is required than predicted to counteract this effect.

Figure 10 shows velocity vectors and temperature contours predicted by the model. The example on the left shows an air curtain set up correctly (10 $m.s^{-1}$). The air curtain on the right hand example has too low a jet velocity (6 $m.s^{-1}$) and the momentum of the air curtain cannot counteract the transverse buoyant forces. The air curtain is deflected into the cold store and does not impinge on the floor, which leads to an unsealed entrance with very poor air curtain effectiveness.

Hayes and Stoecker [32, 33] developed an analytical model to predict heating and cooling loads across non-recirculatory air curtains. Their model allows the calculation of the 'deflection modulus', which is the ratio of air curtain momentum to transverse forces, caused by temperature difference either side of the curtain (stack effect). From this model it is possible to calculate the minimum air curtain velocity to provide an unbroken curtain. However, because this velocity is at the borderline of stability, a higher outlet velocity must be selected in order to provide a factor of safety. They showed that the heat transfer through the curtain is proportional to the jet velocity for velocities above the borderline of stability and it is not therefore beneficial to have too high a safety factor. The literature suggests a range of safety factors between 1.3 and 2.0 to use in this model [33].

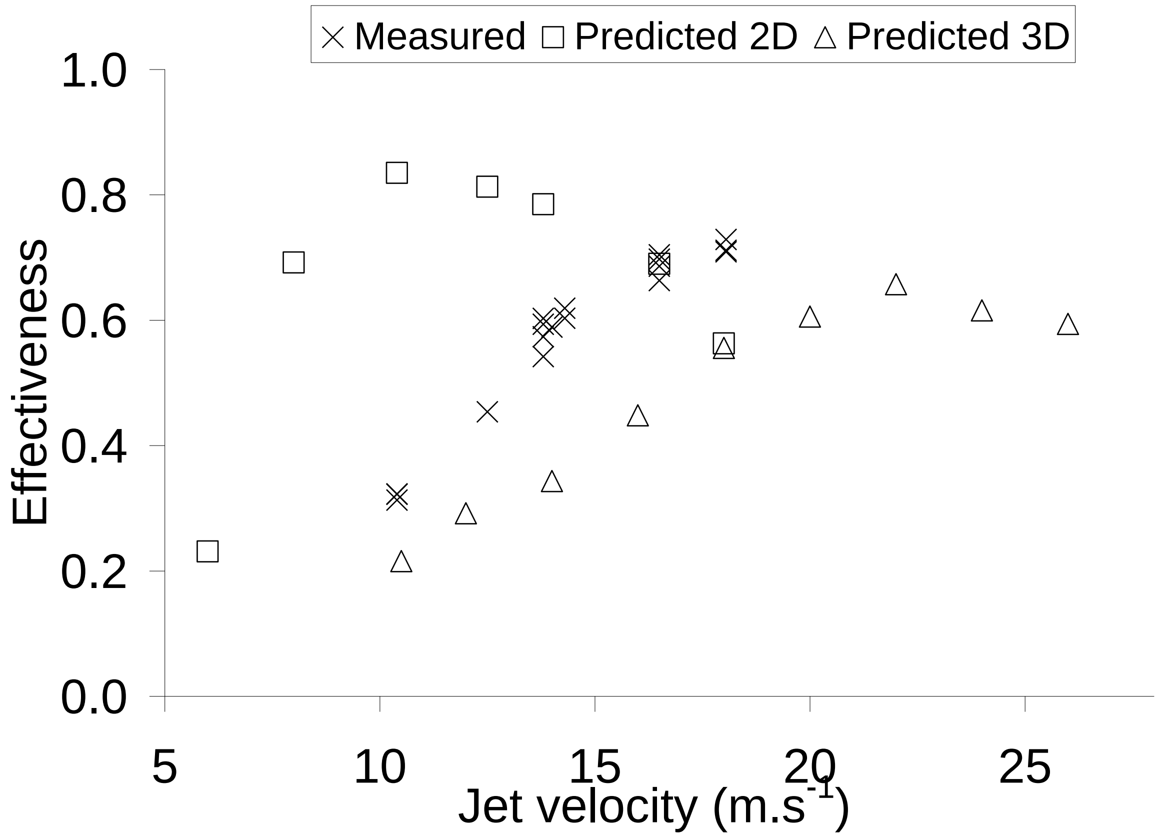


Figure 9. Measured and predicted effectiveness of the air curtain for different air jet velocities for both a 2D and 3D model.

The CFD model of the air curtain was compared to this analytical model and to measured data. Using the range of safety factors specified by Hayes and Stoecker, jet velocities from 11 to 17 m.s^{-1} could be chosen. Using these velocities resulted in an experimental effectiveness of the air curtain varying from 0.37 to 0.70. A higher safety factor of 2.2 yielded the best effectiveness in these tests.

6.1.4. Three dimensional model of air curtain

The 2D CFD model was extended to a 3D model [**6**]. The geometry of the modelled air curtain was an approximation of the real air curtain (Figure 11); the exact geometry would have required a finer mesh than was possible with the given computing resources when used in conjunction with such a large domain.

Important geometrical dimensions, such as the nozzle thickness and its relative position to the fan body were modelled as accurately as possible. The air return grille on the air curtain was modelled as being at the top of the air curtain. This was because the real return grille characteristics were complex and attempts at modelling it with the size of mesh used in the model caused unrealistic flows. The door rail obstructed entrainment to the room side of the outlet nozzle and so a simplified geometry was created to give the same effect (shown as the block labelled “simplified door rail” to the left of the outlet and return ducts in Figure 11).

The numerical mesh ranged from 30 mm at its finest point to 500 mm at its largest point. The width of the air jet boundary is the same as the minimum cell size, thus there is only one cell across the width of the air jet.

In the plane of the door entrance, and for a radius of 0.5 m from this plane, the mesh was 100 mm. The total number of grid nodes for the model was 383 945. This was almost 10 times the number of grid nodes used in the 2D model and was the maximum that could be

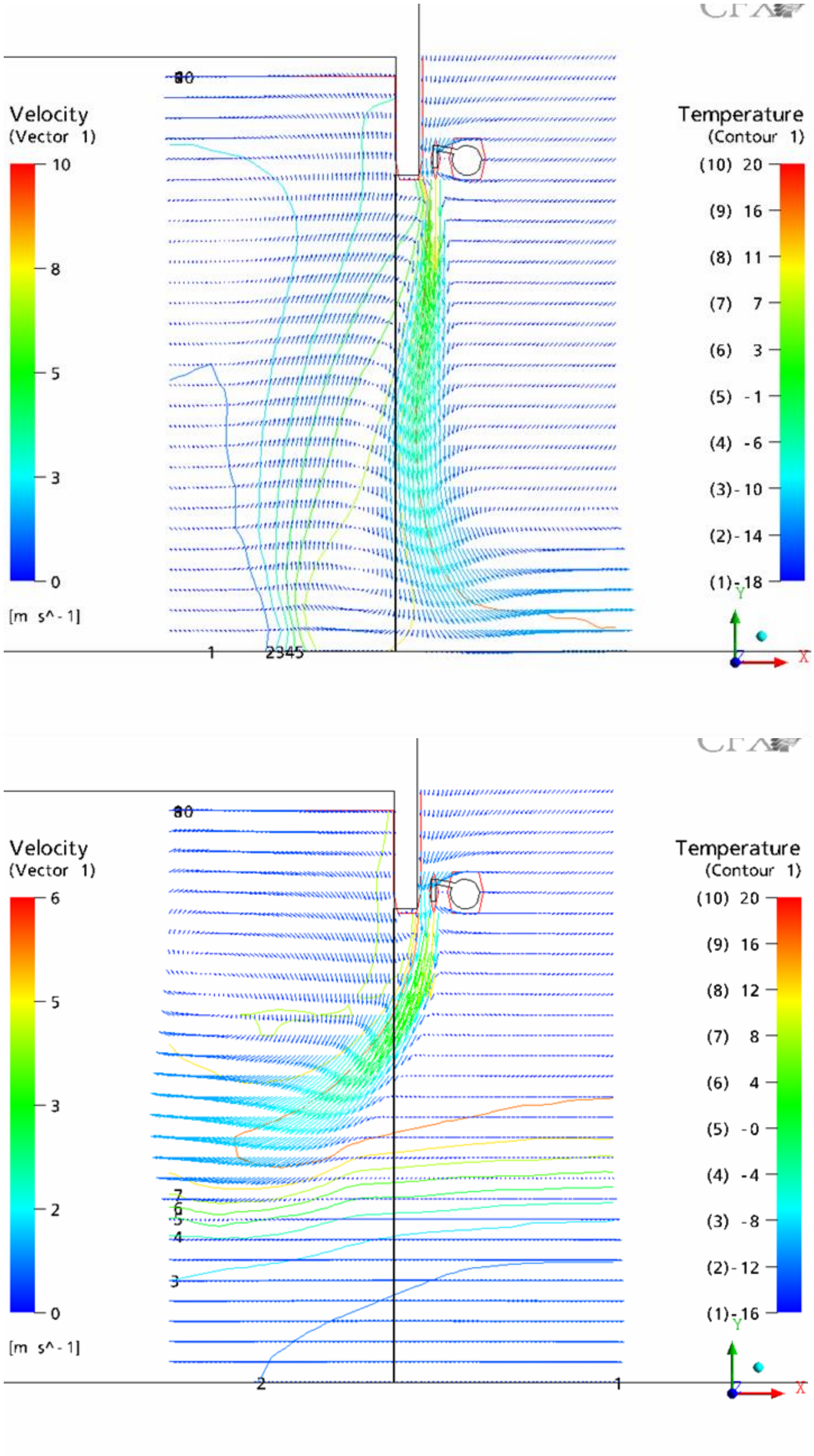


Figure 10. CFD visualisation of correctly set up air curtain (left) and an air curtain with too low a jet velocity (right).

obtained with the hardware available. To test the convergence of the mesh, finer meshes were used on a 'cut down' geometry. This 'cut down' geometry was essentially the same, except a symmetry plane was used to reduce the number of mesh points. Mesh sizes of 100, 70 and 40 mm in the plane of the door entrance were produced. Computer resources did not allow meshes smaller than 40 mm within the entrance in this model.

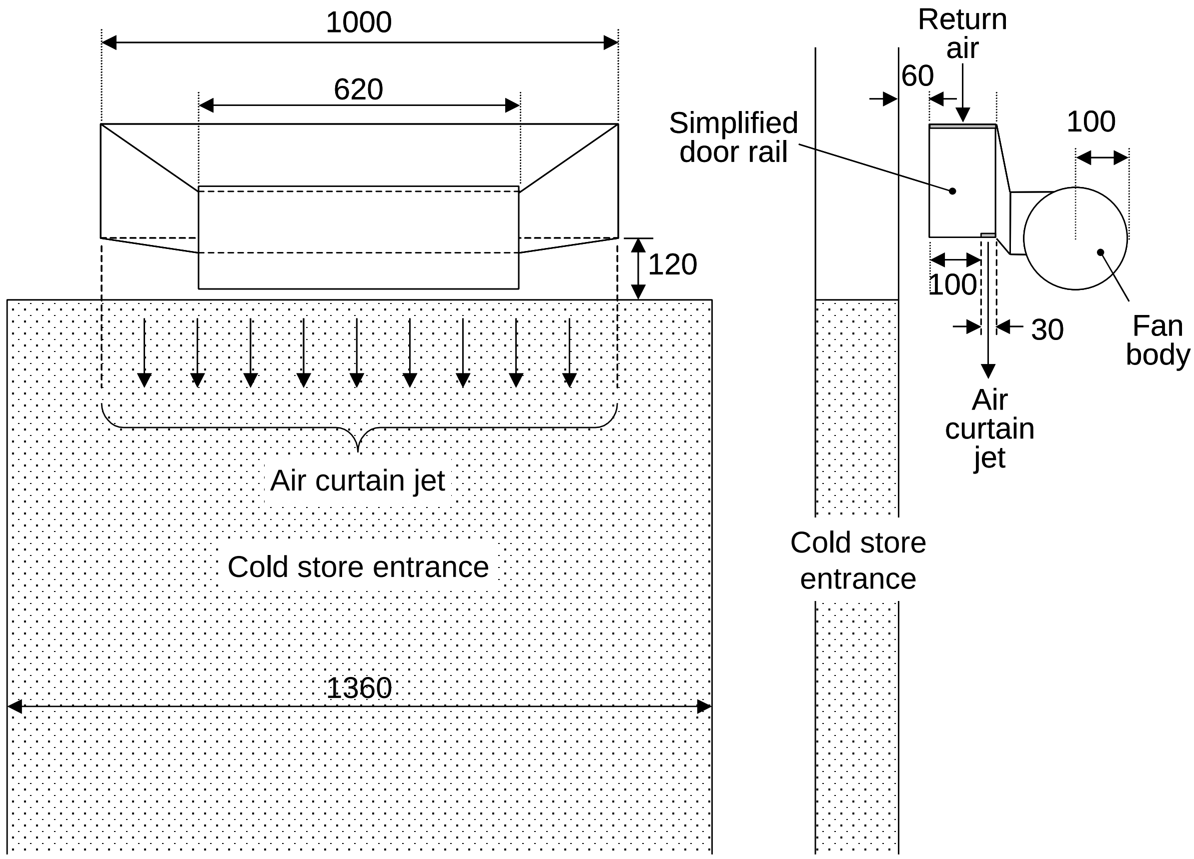


Figure 11. Front and side elevations of the modelled air curtain device and air curtain inlet and outlet boundaries. Dimensions in mm.

Computer simulation times were approximately 22 hours for a transient run of 30 s with a mesh size of 100 mm to a normalised residual of 1 x 10^{-4}.

Figure 9 shows that the predicted effectiveness was always lower than measured (0.10 lower at the minimum and 0.15 lower at the maximum measured velocity). This model was more accurate than the 2D model, which over predicted the effectiveness of the curtain to a greater degree.

The trends of the predicted and measured data are similar up to the maximum measured velocity, in that they had a similar, positive gradient that reduced with jet velocity. The predicted effectiveness reached a maximum of 0.66 at a jet velocity of 22 m.s^{-1}. It was not possible to increase the measured jet velocity above 18 m.s^{-1} in these experiments due to limitations in the installed system. However, the only data found in the literature [34] shows the shape of the predicted curve to be accurate, essentially a polynomial curve with one maximum effectiveness at a specific velocity. It is not possible to predict at what jet velocity the maximum effectiveness would be from the data. However, the (average) positive gradient between the highest two measured velocities indicates that the maximum effectiveness would be greater than 0.72, at a jet velocity above 18 m.s^{-1}.

It is difficult to see the 3D effects of this flow from a single 2D plane and therefore 3D iso-velocity plots for 3 different views are shown in Figure 12. The iso-velocity plots show the direction of air flow for all positions at 3 m.s.$^{-1}$. The front elevation shows that the jet narrows in the plane of the entrance. The side elevation highlights that the centre of the jet is bent out away from the room but the sides are folded back into the room. The plan view shows that the central part of the jet is deflected out of the room (upwards in the figure) and narrows, but that the sides of the jet are drawn into the room and across one another.

These predictions showed that the flow was far from two dimensional. The central portion of the air curtain was deflected away from the cold store by the Coanda effect (caused by the fan body). The sides of the curtain were deflected into the cold store by the stack pressures and were drawn into the void caused by the deflected central portion.

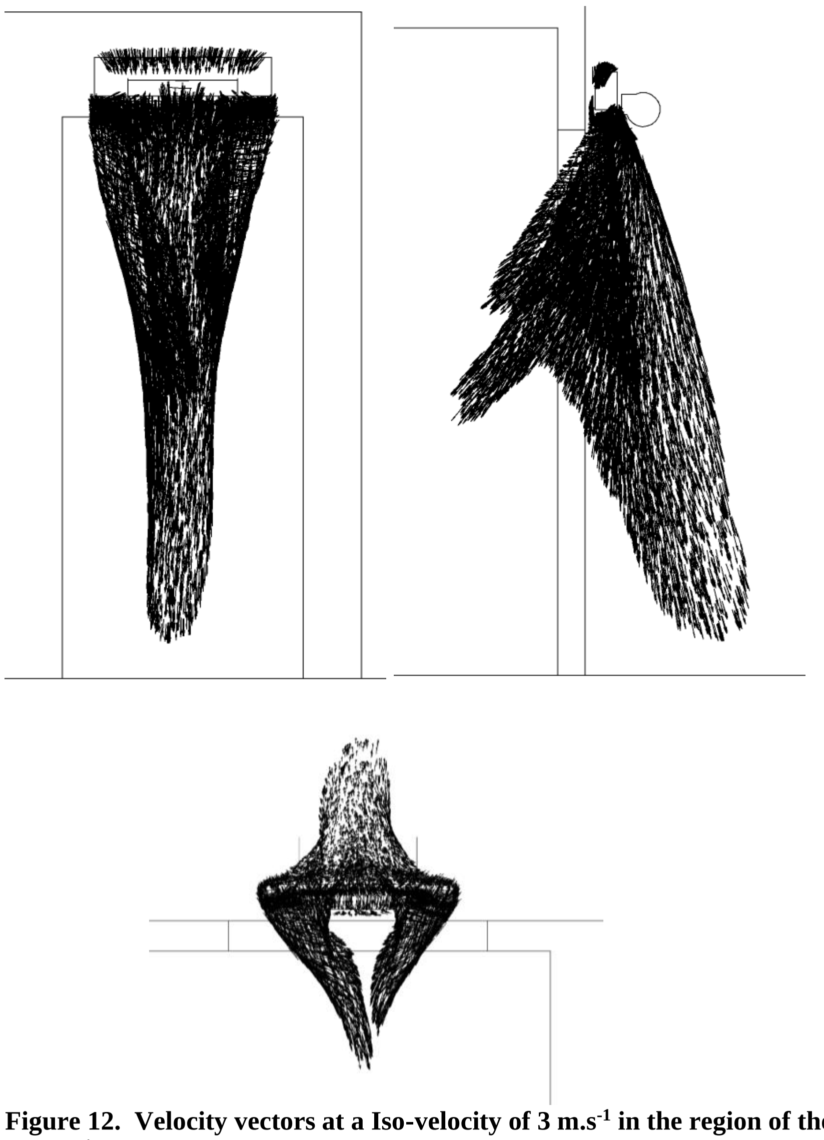


Figure 12. Velocity vectors at a Iso-velocity of 3 m.s^{-1} in the region of the entrance for a jet velocity of 18 m.s^{-1}, angle of 0^{o} and door-open duration of 30 s. Front elevation (top left), side elevation (top right) and plan view (bottom).

6.2. Display cabinet applications

Many authors [35, 36, 37, 38, 39, 40] have shown computational fluid dynamics (CFD) modelling to be a valuable tool to improve airflow within display cabinets. Foster and Quarini [**1**] presented the first comprehensive study, including CFD modelling, of the effect of the refrigerated display cabinet on customer comfort within a supermarket. Later work [**4**] investigated the feasibility of using CFD to improve a refrigerated display cabinet within commercial timescales.

6.2.1 Cold spillage

The cold feet effect is caused by mixing between the display cabinet air curtain and environment. The aim of this study was to quantify the effect in three supermarket chains and use CFD to investigate ways of alleviating it. Two different scales of CFD model were used. The CFD models were generated using CFX 4.1

The first model was of the whole store sales area, this was run on a Silicon Graphics R10000 workstation with 256 Mb RAM. The store was ventilated by inlet diffusers which forced air out in four directions (towards the front, back and both sides of the store). Because of this, each of the 13 diffusers had to be modelled as four separate inlets giving a total of 52 inlets in the model. The inlets were modelled as constant velocities of 4.2 $m.s^{-1}$ at an angle of 45° to the ceiling. The temperature of the air at the inlet was 26°C. These data were taken from measurements carried out at the store. The eight return diffusers were modelled as mass flow boundaries. By design, only half the air that entered the store, left through these diffusers. This was to keep the store at a slightly higher pressure than outside to avoid ‘dirty’ air entering the store. It was considered that most of this air would exit through doorways. Putting each of the doorways into the model would have made it too complex (in terms of the number of mesh points). Therefore a mass flow boundary was set along the bottom of the four side walls. The cabinets and shelving were modelled as solid blocks to restrict flow around the store. For the refrigerated cabinets (chilled and frozen) a heat sink was set in front of the blocks. This simulated the quantity of heat removed from the store. The values were chosen from ASHRAE [41] and varied for the different types of refrigerated cabinet. The store had a heated floor in the chilled and frozen aisles during the winter. The heated floor was simulated by setting a constant temperature boundary condition of 19°C and a heat transfer coefficient of 9.3 $W.m^2.K^{-1}$. The internal dimensions of the store were 64 x 37 x 24 m. The geometry was divided into a grid of 130 000 cells, with a resolution of between 30 and 600 mm.

Figure 13 shows predicted and measured temperatures of the supermarket store at two heights. Although the temperatures predicted were not the same as measured, the trends were very similar. The model clearly showed that the level of stratification reduced significantly between the chilled and ambient aisles. It showed that the temperature at floor level was dictated by the location of the chilled cabinets. At a height of 2 m the model showed that the cabinets had little effect and that the temperature range was much smaller. The model also showed that warm air from the inlets did not spread very far.

The computer model of the whole store was too large to model detail in the store and the time taken for convergence was too long to carry out many what-if scenarios in a limited time. To reduce the complexity of the model just a chilled aisle was modelled. Three symmetry planes were used to reduce the number of grid cells further. This reduced the number of grid cells by a factor of 40 to 3 500 cells and meant the model could be run on more easily available hardware (Silicon Graphics R5000 workstation with 64 Mb RAM).

The cabinet was modelled as a solid object with an inlet boundary at the top face blowing cold air vertically downwards. An outlet boundary was positioned at the bottom of the cabinet. The temperature and velocity at the inlet was -2°C and 1.0 $m.s^{-1}$ respectively. Warm ambient air was required to enter and leave the domain. Previous modelling and experiments showed that cold air escaped at floor level out of the aisle and warm air entered at high level. Because of the positions of the symmetry planes, the floor and ceiling of the model, there was only one boundary face that could exchange this air and it was at the end of the aisle. A constant pressure boundary was chosen to simulate the air entering and leaving the aisle. The temperature of air entering the aisle was 22°C and the temperature of the floor wall-boundary was 14°C (these were taken from measurements). Predicted and measured temperatures in chilled aisles in three different supermarkets are shown in Figure 14. The simulation was found to be a good approximation of the three different stores. The temperature near the floor, the gradient of the temperature stratification and the constant temperature region above the cabinet were all predicted accurately.

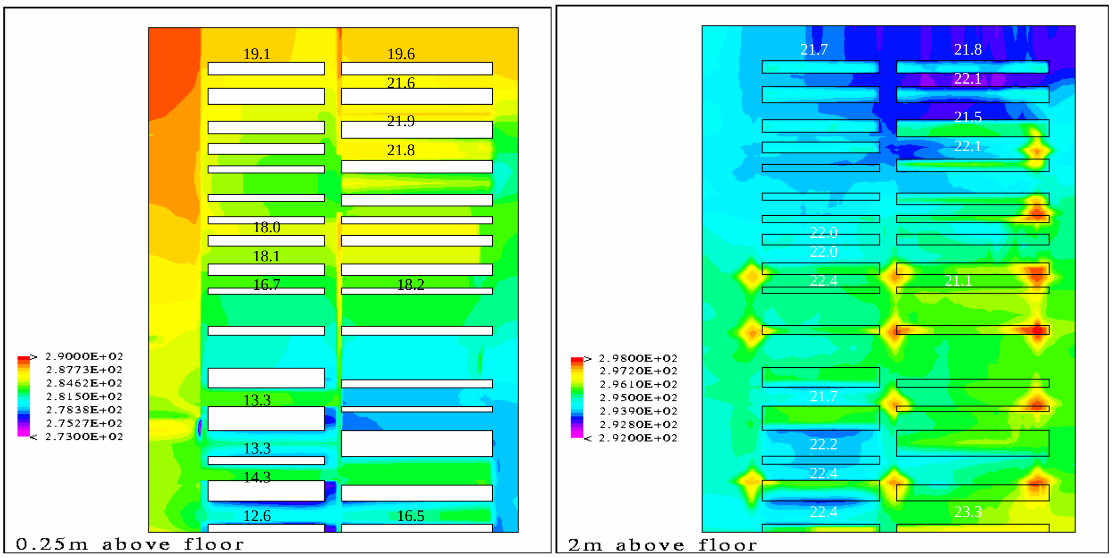


Figure 13. Plan view of store at a height of 0.25 and 2.0 m. Colors show predicted temperatures (K) and values of measured temperature (°C) are superimposed.

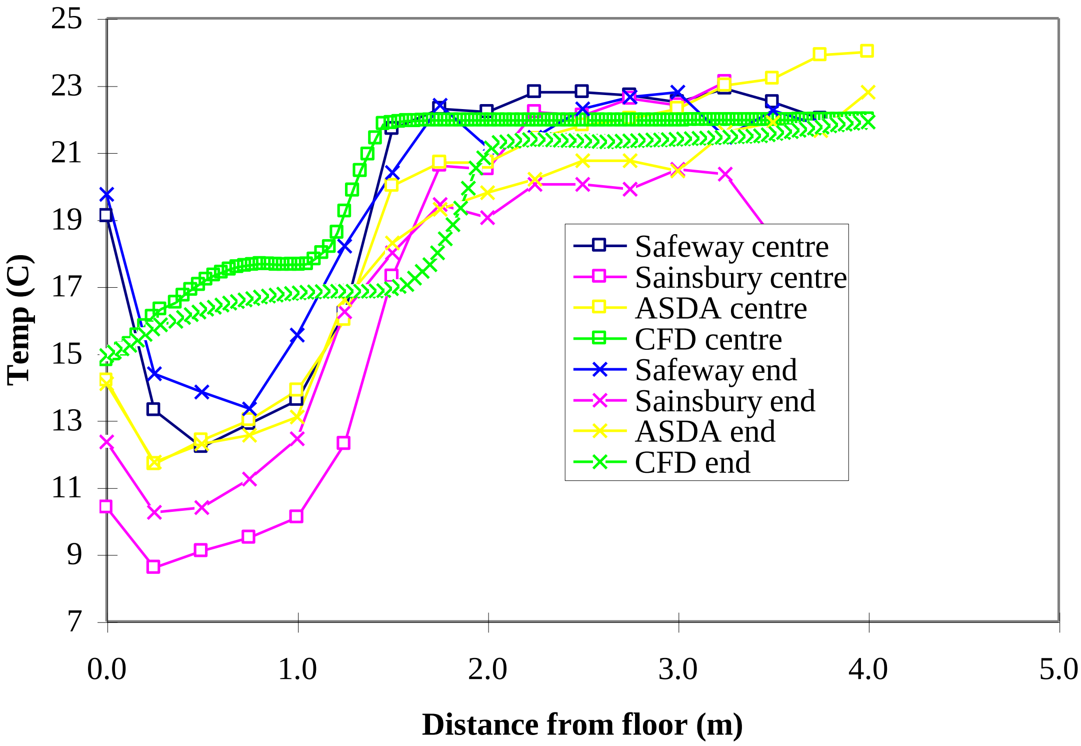


Figure 14. Temperatures measured at 2 positions in 3 different stores in the coldest aisles and temperatures predicted from the numerical model.

This model was then used to predict the effect of different heating and ventilation strategies. These were; heated floors, air mixing which drew cold air from under the cabinet and projected it upwards between the cabinets and finally high velocity vertically blowing fans in the centre of the aisle. The CFD model was able to predict the effect of these strategies and relay them back to the supermarket heating and ventilation engineers.

Tassou and Xiang [42] carried out a very similar study. They also found the complexity of modelling the entire store too great and chose to model an aisle using the same symmetry planes. However, with more modern hardware they were able to model the food within the cabinet. They showed that under-floor heating had little effect on air temperatures in the aisle (1 or 2°C in the centre of the aisle), this is in agreement with the work presented in this thesis. It is important to note that although the cabinet has an effect on the store environment, the store environment also has an effect on the cabinet. As Tassou and Xiang's model included the product they were able to simulate this. They showed that the heated floor causes a temperature rise on the surface of the products facing the aisle of around 2.2°C and a rise at the core of the product of approximately 0.8°C. This was due to heat transfer by radiation between the floor and the chilled products in the cabinet. Increasing the heat flux on the floor would reduce cold discomfort but would have a detrimental effect on the chilled products in the cabinet.

5.2.2 Improving performance of chilled multi-deck cabinet

CFD was used to rapidly identify the changes that would be required to an existing M2 (temperature range of food -1 to 7ºC) classification refrigerated display cabinet to achieve an M0 (temperature range of food -1 to 4ºC) rating [**4**Error! Bookmark not defined.]. This work was carried out using CFX 5.5.1 on an Intel Pentium 3 PC running at 650 MHz with 512 Mb of RAM.

Prior to modifications of the cabinet the temperature of all 'm' packs was above –1°C and below 7°C. However, only 42 'm' packs (out of 54) spent the entire test period between -1 and 4°C. A maximum temperature of 6.9°C was measured in the 'm' pack positioned at the top right front of the well. A minimum temperature of –1.0°C was recorded in the 'm' pack situated at the top rear left of the well. The area of most concern in terms of high temperatures was located at the centre and right front of shelf 1 and the front edges of the well.

The flow of air as it exited the discharge grille and travelled down the cabinet was simulated. To reduce complexity of the model and concentrate on the area of interest, the air curtain for only the top shelf was modelled. The existing air curtain exiting the discharge grille had a velocity of 1.5 $m.s^{-1}$ and a temperature of –1°C, with the jet pointing vertically downwards. The width of the air curtain was 15 mm. The problem was considered to be 2D as the cabinet and air curtain were very long; therefore, end effects could be ignored.

The CFD model predicted that the air curtain would be bent towards the product and hit the outer corner of the product (Figure 15). A vortex was then created which drew warmer air over the front of the product. The air curtain would also be disrupted bringing warmer air onto the shelves below.

CFD modelling was used to investigate the effect of a number of modifications including changing the width and angle of the curtain. Increasing the width of the air curtain to 60 mm, whilst keeping the volume flow rate through the air curtain constant, was predicted to remove the vortex. This was likely to keep the product cooler and also maintain a better air curtain for the next shelf down.

The CFD predictions also showed that there were benefits in angling the air curtain away from the shelves, and that the optimum angle depends on the width and velocity of the curtain.

The cabinet used in the previous section also had an uneven temperature left to right which could not be attributed to the air flow in the room. Asymmetries within the duct were examined.

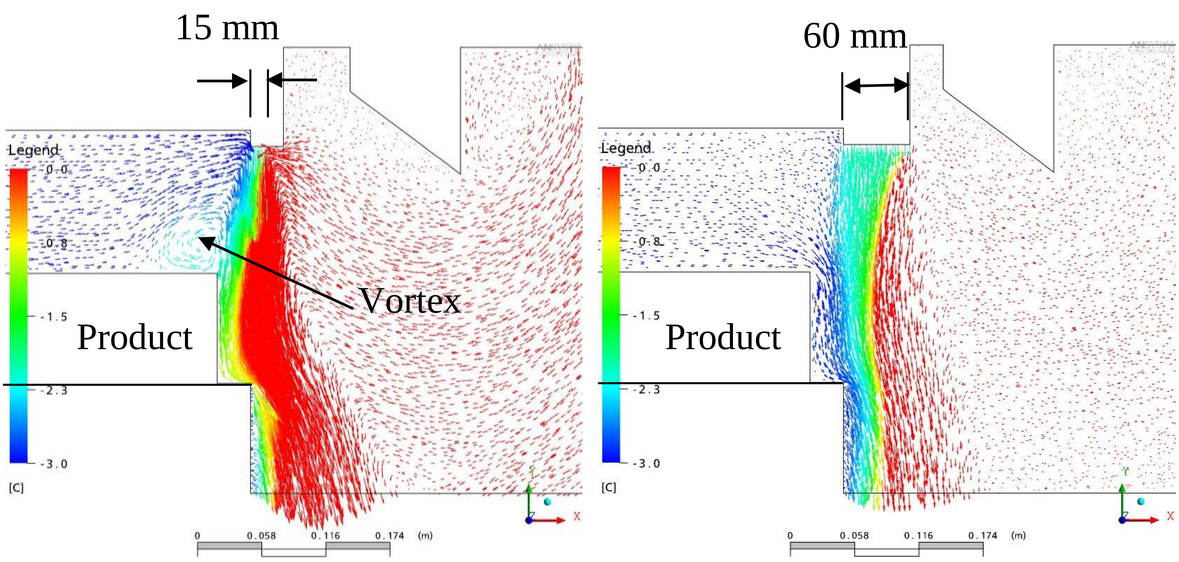


Figure 15. Velocity vectors predicted by CFD in a vertical plane at the top of a multi-deck refrigerated display cabinet. The colour of the vector represents the temperature (legend). The figure on the left shows an original thin high velocity air curtain, the figure on the right a thicker low velocity air curtain.

It is not possible for the evaporator to run the full width of the cabinet due to the turns at the ends of the evaporator; these are often greater on one side of the evaporator than the other. These problems can lead to both a 3D and unsymmetrical flow within the duct.

The cabinet had a 110 mm gap between each end of the duct and the evaporator. Air, which exited the evaporator, had therefore to change in direction by 90° to enter the edge of the duct. CFD was used to model the flow of air as it exited the evaporator and entered the rear duct. Only the bottom edge of the cabinet was considered, where air from the evaporator was required to fill the dead space previously identified at the edges of the cabinet.

Figure 16 shows streamlines predicted by CFD from the exit of the evaporator for the un-modified cabinet. The predictions show that as air leaves the evaporator it moves out to the side to fill the dead space and then travels up the rear duct. In doing so the air spirals up the rear duct at the edge. This type of airflow was likely to result in vortices as the air entered the cabinet rather than the uniform flow required. To address this problem the following design solutions were proposed:

1. Make the evaporator as wide as possible (reducing the length of the dead space)
2. Move the evaporator towards the front of the cabinet and create an angle such that the air would more easily move to the edge of the duct.
3. Insert an angled plate at the bottom of the duct to increase the pressure and even out the flow.

Figure 16 shows streamlines predicted by CFD from the exit of the evaporator for the modified cabinet (length of evaporator extended such that there was only a 50 mm dead space at each end of the cabinet, evaporator moved forward by 80 mm, angle to expand air to the edge of the duct and a turning vane). With these modifications the CFD predicted that no vortex would develop.

Many authors have proved CFD’s ability to predict air flow in refrigerated display cabinets. However, this has generally been carried out during research projects lasting a number of months or even years. The benefit to the display cabinet manufacturers is to get results quickly (within a couple of weeks). This work showed that this was possible and to the author’s knowledge is the only paper which shows this.

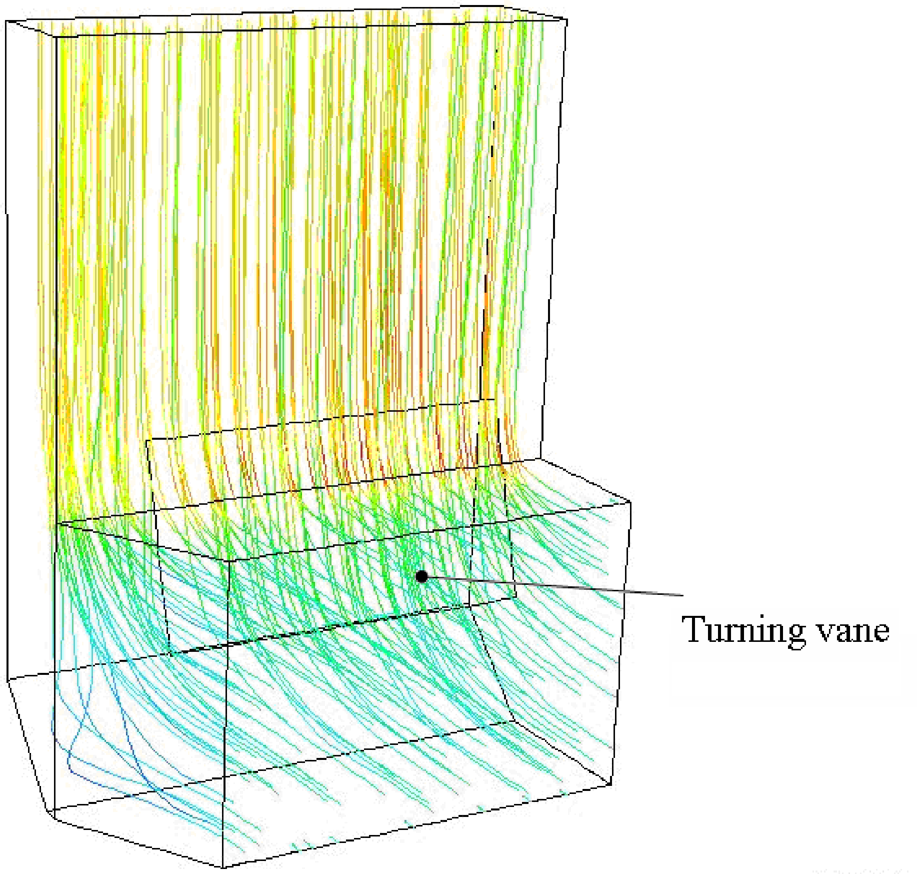


Figure 16. Streamlines predicted by CFD from the exit of the evaporator for the modified cabinet.

7. Conclusions

The author has shown using measurement and CFD, the importance of setting up a cold store air curtain properly, if it is to work at its optimum efficiency. Measurements have been used to investigate the accuracy of current empirical models. These measurements incorporated multi vane anemometry, laser Doppler anemometry (LDA) and tracer gas methods and were used to validate a CFD model developed using commercial code. The model was further developed to investigate air curtain behaviour and particularly its effectiveness. The 3D behaviour of air curtain flow has not previously been reported in the literature.

The extent of the ambient domain that needs to be modelled is an important consideration. If the ambient is small and the door is opened for a long time, the ambient temperature may rise, causing a drop off in the driving force which may not be apparent in reality. This effect was investigated in the infiltration study, but computer resources never allowed the ideal scenario whereby increasing the size of the ambient further has no effect on the results.

CFD predictions of unprotected entrances have been shown to give a significant improvement in accuracy over the fundamental analytical equations [20, 21]. However, the empirical coefficients added by Gosney [23] and Fritzsche [22] gave more accurate predictions than the CFD models. More detailed CFD models with more grid cells in the entrance may give better predictions. CFD models show a lag time which analytical predictions ignore, but unless the door opening time is very short this reduction in flow is quite insignificant.

The analytical model of Hayes and Stoecker predicts optimum jet velocity of an air curtain on a cold store; however it does not predict the air curtain's effectiveness. It only gives a guide to the optimum jet velocity and depending on which safety factor is chosen, a large range of air curtain effectivenesses can therefore be achieved. The 2D CFD model could predict the effectiveness of the air curtain, however, the predictions were higher than measured. This was because end effects allowed air to leak around the edge of the air curtain, causing increased infiltration, which could not be predicted in 2D. Higher jet velocities will to some extent negate this problem and this is a probable explanation for the increased optimum jet velocity of the real air curtain compared to the 2D CFD model.

The 3D CFD model was able to provide a better prediction of effectiveness and optimum jet velocity than the 2D model. The predicted effectiveness was lower and optimum jet velocity higher than measured, the opposite to the 2D model. The model showed that the flow from an air curtain cannot be considered as 2D. For the air curtain studied a complex 3D flow pattern was apparent.

This thesis has shown the importance of setting up air curtains to work at their optimum effectiveness. It is clear from the cold store air curtain manufacturer's viewpoint that air curtains are effective at reducing infiltration. Cold store operators often have a more negative viewpoint based on their own experiences. The difference between the operator's and manufacturer's perception can be put down to the air curtains not being effectively set up at many cold stores. This thesis has experimentally validated CFD models to predict the optimum condition of the air curtain. This work has shown that although 2D models can be used to good effect, it is only with 3D modelling techniques that the flow can be accurately predicted.

To the author's knowledge, he was the first to investigate the effect of the interaction of display cabinets with supermarket ventilation using CFD and the first to model an entire supermarket display area. This work incorporated comprehensive measurements of store temperatures. The work has informed the supermarkets of the most appropriate ventilation systems to use in chilled aisles. The author has also extended his work on CFD modelling of air curtains to air flow within the ducting of display cabinets. This work showed how geometrical constraints of the evaporator can cause spiralling flow in the rear duct which is unhelpful in maintaining safe product temperatures.

The author has shown that thought must be given to the level of detail or resolution required for the CFD model. A large amount of detail in a large volume may require more computer resources than are available. It is often more appropriate to split the model into large and small scale problems. With this in mind, it has been demonstrated that it is possible to use CFD to optimise the flow in a display cabinet within reasonable time-scales.

CFD modelling in the refrigeration industry is still in the hands of academics, whose project timescales are an order of magnitude longer than industrial project timescales, this is beginning to change. CFD codes have improved a lot during the course of the work presented in this thesis (1994 to 2005). The initial CFD code used for the cold spillage used only hexahedral grids which were positioned within the geometry by the user. These grids took many hours and often days to generate and were not particularly efficient, requiring small grids in areas where they were not needed. This meant that the 'whole store' model was very cumbersome and inefficient due to the small grid required for the inlet diffusers and large geometry of an entire supermarket store. This code was only able to be run on UNIX workstations which were expensive to maintain. The advent of automatically generated tetrahedral meshes (developed for finite element analysis) run on Intel based PCs made a step change in the speed and complexity of the models and the cost of the hardware to run them. Also the CFD code has become more user friendly and reasonable predictions can be obtained in days rather than months (total timescale of generating the model to final conclusions ready to test). However, it should be remembered that anybody can obtain a colourful picture that looks correct, it is still necessary to have a user with an understanding of the physics and to experimentally validate test results.

Many different experimental approaches have been used to provide data to verify these CFD models. These include arrays of mini vane anemometers, LDA and CO_2 tracer gas techniques. Each approach has its own advantages and disadvantages and therefore a complete validation may be better carried out using a range of these methods. To properly validate the 3D interaction between the air curtain and the natural convection in the entrance requires a tool such as digital particle image velocimetry (DPIV). This has been used in refrigerated display air curtains but not with cold store air curtains.

CFD is not yet at the stage where testing and validation can be eliminated. For this reason CFD should be considered as a tool to allow the reduction of testing not it's elimination.

Current computer power is still a limitation for accurate 3D predictions. Future increases in computer power, especially with the advent of 64-bit hardware and parallel processing should allow more accurate predictions.

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