University of Southern Queensland
Faculty of Engineering and Surveying

Numerical Investigation of the Thermal efficiency of glass refrigeration doors

A dissertation submitted by

Michael Allan Eagle

in fulfilment of the requirements of

Courses ENG4111 and ENG4112 Research Project

towards the degree of

Bachelor of Engineering (Mechanical)

Submitted: January 2005

Abstract

Infiltration occurs when warm, moist air from the outside environment becomes trapped inside the refrigerated compartment. Glass doors on a refrigerated display case create a sealed unit keeping the majority of the warm, moist air out.

This paper presents a numerical investigation into the thermal efficiency of display refrigeration doors.

This is achieved by varying a number of features and evaluating their thermal performance. An introduction into the background and context of the work to be undertaken is presented. A literature review is provided, outlining the previous work completed in this field. In the final stages of this report an appropriate methodology, risk assessment, realistic timelines and resource requirements are specified.

University of Southern Queensland

Faculty of Engineering and Surveying

ENG4111 & ENG4112 ***Research Project***

Limitations of Use

The Council of the University of Southern Queensland, its Faculty of Engineering and Surveying, and the staff of the University of Southern Queensland, do not accept any responsibility for the truth, accuracy or completeness of material contained within or associated with this dissertation.

Persons using all or any part of this material do so at their own risk, and not at the risk of the Council of the University of Southern Queensland, its Faculty of Engineering and Surveying or the staff of the University of Southern Queensland.

This dissertation reports an educational exercise and has no purpose or validity beyond this exercise. The sole purpose of the course pair entitled 'Research Project' is to contribute to the overall education within the student's chosen degree program. This document, the associated hardware, software, drawings, and other material set out in the associated appendices should not be used for any other purpose: if they are so used, it is entirely at the risk of the user.

Prof G Baker
Dean
Faculty of Engineering and Surveying

Certification

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

Michael Allan Eagle

Student Number: Q1121597X

Signature

Date

Acknowledgements

The author wishes to recognise the support of the following people:

Dr Ruth Mossad, University of Southern Queensland, Toowoomba

For advice throughout the duration of the research project and preparation of the dissertation.

Mike Malli, Orford Refrigeration Pty Ltd, Toowoomba

For assistance with the practical knowledge on glass refrigeration doors.

The University of Southern Queensland, Toowoomba

For access to the library and engineering computer labs

Special thanks to:

Mum, Dad, Matthew, Shelley and Courtney for their auspicious altruism in constantly providing much appreciated inspiration and support. Thanks also to Beth, Raging Bull, Bradley P Moody, Macca, Spitz, Ryano, The Griff, Ali G and The Dog for being there for me.

Table of Contents

List of Figures

List of Tables

Glossary of Terms

Argon	An inert, non-toxic gas used in IG units to reduce heat transfer
CFD	Computational Fluid Dynamics
Conduction	Heat transfer through a solid material by contact of one molecule to the next. Heat flows from an area of higher temperature to a lower-temperature
Convection	A heat transfer process involving motion in a fluid (eg. air) caused by the difference in density of the fluid and the action of gravity. Convection affects heat transfer from the glass surface to the ambient air, and between the panes of glass
IG Unit	Insulated Glass Unit
Desiccant	An extremely porous crystalline substance used to absorb moisture from within the sealed air space of the IG unit
Edge effects	Two-dimensional heat transfer at the edge of the IG unit due to the thermal properties of spacer and sealant
Emittance	The ratio of the radiant flux emitted by a specimen to that emitted by a blackbody at the same

	temperature and under the same conditions
Gas fill	A gas other than air, usually argon or krypton, placed between the glazing panes to reduce the U-factor by suppressing conduction and convection
Glass	An inorganic transparent material composed of silica (sand), soda (sodium carbonate), and lime (calcium carbonate) with small quantities of alumina, boric, or magnesia oxides.
Krypton	An inert, nontoxic gas used in IG units to reduce heat transfer
Low-Emittance coating	(Low-E) Thin metal or metallic oxide layers deposited on a glazing surface primarily to reduce the U-factor by suppressing radiative heat flow
Radiation	The transfer of heat in the form of electromagnetic waves from one separate surface to another
Sealant	A compressible plastic material used to seal the glass panes and the spacer, commonly made of silicone, butyl tape, or polysulphide
U-factor (U-value)	A measure of the rate of non-solar heat loss or gain through a material or assembly
Warm-edge technology	The use of low-conductance spacers to reduce heat transfer near the edge of the IG unit

1 Introduction

This paper presents current methods employed in Insulated Glass design and discusses the results of a numerical investigation into the thermal efficiency of display refrigeration doors.

1.1 Background

The refrigerated display cabinet allows more products to be faced towards the customer, increasing product visibility and promoting impulse sales through increased visibility and better presentation. A high efficiency cooling system needs to keep the product at the right temperature all year round.

The major contributor to the total cooling load is infiltration from the outside, the flow of air into a void through openings in the outer surfaces of the void. It can occur through joints and gaps around the IG frame and glazing. It occurs when warm and moist air from the room becomes trapped inside the refrigerated compartment. The display doors reduce the amount of infiltration and reduce the cooling load considerably. Reducing the infiltration loading will decrease the power consumption.

The Insulated Glass unit (IG unit); Figure 1.1, consists of two glass panes, separated by a spacer between the glass panes and a desiccant to absorb moisture. These are all contained in an airtight seal of butylene rubber. The IG unit is positioned inside a door rail or frame, which is fixed to the front of the cabinet.

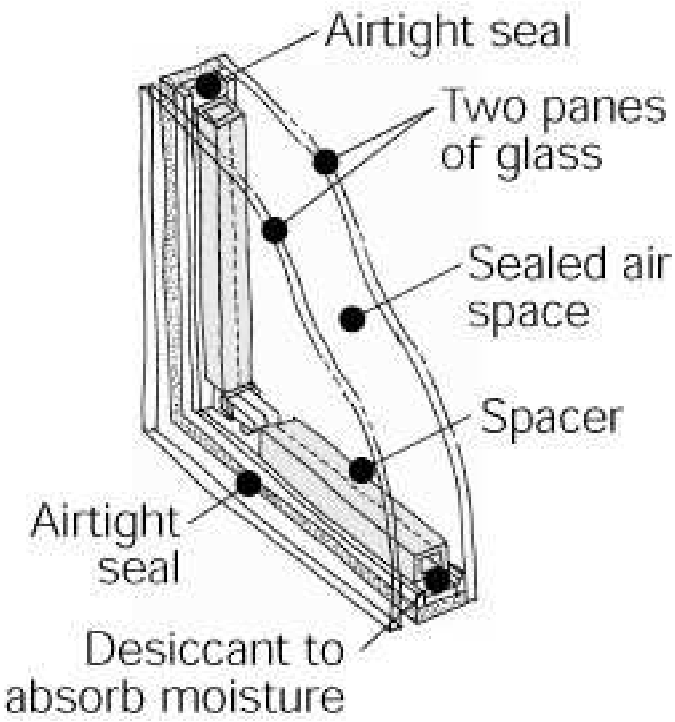


Figure 1.1 The IG Unit

(Source: Efficient Window Collaborative)

Heat transfer through the unit affects the cooling load. To create an IG unit that achieves optimum thermal efficiency the affects of gas type, spacing between the panes and spacer construction are to be analysed in this report.

1.2 Objectives

In agreement with the requirements of the course ENG4111/ENG4112 Engineering Research Project provided by the University of Southern Queensland the objectives of the investigation are to:

- Build background knowledge for door design of a display refrigeration unit
- Conduct literature reviews on this topic

- Study the effect of gap thickness in a display door to achieve optimum thermal efficiency
- Conduct computer simulation with CFD software 'Fluent' to investigate a variety of designs and evaluate their performance
- Analyse these results and recommend criterion for thermally efficient design

As time permits:

- Design an optimum thermally efficient display refrigerator door design

1.3 Organisation of Study

Chapters 2 and 3 discuss the current situation of IG Unit technology. Chapter 4 describes the theories for the numerical investigation whilst Chapter 5 discusses the numerical investigation. Chapters 6 and 7 illustrate the Results and a Discussion of the numerical investigation. Chapter 8 puts forward the conclusions from the research and provides recommendations for future work.

1.4 Scope and Limitations

This study examines a complete numerical investigation. No experimental investigation was conducted to validate the findings. The modelling and simulation was based on one and two-dimensional techniques.

1.5 Research Methodology

The first step has been to research the current practices in design and manufacture for the door in a display refrigeration unit to apply to our designs.

Orford Refrigeration has been very helpful in this area by providing their knowledge and expertise.

The second step has been to conduct literature reviews in related areas to find out what previous research has been done in this field. There is not an abundance of research in this area; however, a great deal has been conducted in the performance of household windows. This research is quite similar in nature and can be applied directly to our work.

The third step is to commence the theoretical investigation into door design. Analytical analysis is to be undertaken into the effect of gap thickness to achieve optimum thermal efficiency. Subsequent to the analytical investigation the numerical simulation will follow, the CFD software FLUENT was used to conduct the analysis of a variety of designs and evaluate their performance.

With this completed we can analyse the results and generate recommendations for thermally efficient design criterion. If time permits we will build the final design that achieves perfect thermal efficiency.

2 IG Unit Fundamentals

This chapter isolates the techniques being utilised at the present-day in IG Unit design and defines the goal of these techniques. To understand the importance of proper IG Unit design, current design techniques and common IG problems are discussed.

2.1 What is an IG Unit?

An insulated glass unit starts out as two or three individual panes of glass that are chemically bonded by a sealant to spacer bars.

Figure 2.1 Complete IG units

2.2 IG design concepts

Heat loss or heat gain through the IG unit occurs via three methods: radiant heat transfer, convection heat transfer and conductive heat transfer.

Radiant heat transfer is the transfer of heat in the form of electromagnetic waves from one separate surface to another. It is heat flow via absorption and then re-radiation.

Convection heat transfer is heat flow via air movement, the flow of heat through a circulating gas or liquid, such as the air in a room or gas between glass panes.

Conduction heat transfer is heat flow through materials. Heat conducts from the warmer side to the cooler side of the IG unit as each molecule excites its neighbour passing along the energy. Conduction occurs through the glass, the frame and in the gap between the panes.

An energy efficient IG unit will minimise heat losses by controlling all three sources of heat transfer.

Figure 2.2 illustrates the performance of a typical double-glazed unit with two layers of clear glass. The inner and outer layers of glass are both clear and separated by an air gap. Double-glazing; compared to single glazing, cuts heat loss in half due to the insulating air space between the glass layers. In addition to reducing the heat flow, a double-glazed unit with clear glass will still allow the transmission of high visible light, this is highly desirable so as to maintain full product visibility. The glass layered with a low-emittance coating reduces heat loss even further; however, the investigation of this method of enhancing door performance has been assigned to future research.

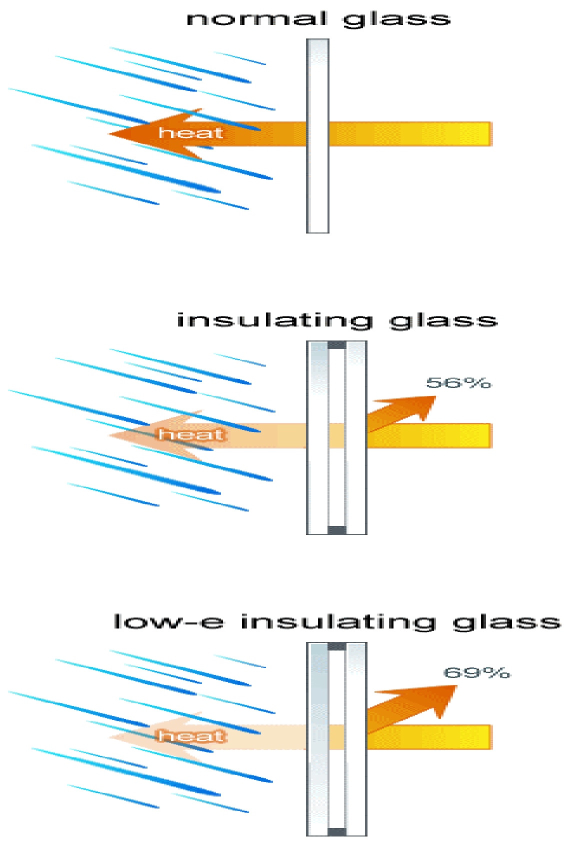


Figure 2.2 The effects of insulating and low-e glass

(Source: Efficient Window Collaborative)

There are a variety of glass related options that window manufacturers use to create energy efficient windows. These methods improve the thermal performance of the glass. In the window industry they use a term known as the U-factor to measure the rate of heat flow through a material or assembly. It is expressed in units of W/m^2.°C. The U-factor describes the rate of non-solar heat loss or gain through a window. This means the lower the U-Factor, the lower the amount of heat lost or gained creating a higher thermal performance. (Source: PPG)

The following five factors affect the U-value of the IG unit

- The type of glazing material (glass, plastic, treated glass)
- The number of layers of glazing
- The size of the air space between the panes
- The thermal resistance of the frame and spacer materials
- The tolerance of the entire assembly

2.3 Enhancing Door Performance

There are four main techniques utilised to enhance door performance:

Insulating glass

As mentioned earlier, insulating glass is comprised of two (or more) pieces of glass separated by a spacer material and sealed together to create an insulating glass unit or (IG unit). IG units reduce convection and conduction heat loss. An extension to this is to create a triple pane IG unit by placing a piece of stretched plastic film in between the two panes.

Low-E glass

Low-emittance (Low-E) coatings are thin, virtually invisible, metal or metallic oxide layers deposited on the glazing surface to reduce the U-factor by suppressing radiative heat flow. Coating a glass surface with a low-emittance material and facing that coating into the gap between the glass layers blocks a significant amount of this radiant heat transfer, thus lowering the total heat flow through the window.

Heavy inert gas in the IG unit

An improvement that can be made to the thermal performance of insulating glazing units is to reduce the conductance of the air space between the layers. Originally, the space is filled with air or pumped with dry nitrogen before sealing. In a sealed IG unit heat flows inside the gap to the top of the unit, while cold regions form in the bottom. Filling the space with a less conductive, heavy gas minimises the convection currents in the air gap, conduction in the gas is reduced, and the heat transfer of heat between the inner and outer panes is reduced.

Argon or krypton gas can be used in the IG unit as they provide a considerable improvement in thermal performance. Argon is cheap, non-toxic, non-reactive, clear, and odourless. Krypton has better thermal performance, but is expensive. A mixture of krypton and argon can be used as a trade off between thermal performance and cost. (Source: Truseal)

Figure 2.3 Argon Filled Unit

(Source: Truseal)

Warm edge spacer system

The layers of glazing in an IG unit must be held apart at the appropriate distance by spacers. Because of its good structural properties, aluminium spacers have been used for many years. Aluminium is a very good conductor of heat and the aluminium spacer used in the majority of edge spacers causes a thermal leak at the edge of the unit. Not only does it increase heat loss, this edge may develop condensation.

A variety of edge systems to alleviate these problems have been created. Replacing the aluminium spacer with a less conductive metal such as stainless steel and changing the shape of the spacer can reduce heat loss.

Another design idea is to use a better insulating material instead of metal. A common design uses a spacer, sealer, and desiccant in a thermoplastic compound. Another concept is an insulating silicone foam spacer that uses a desiccant and a strong adhesive to bond the edges to the glass. (Source: Edgetech)

The thermal boundary is implemented to stop the heat transfer at the edges and between the layers. Warm edge spacers have become beneficial as manufacturers make the move from standard double-glazing to high performance glazing. Thermally efficient spacers can increase performance by up to 25%. (Source: Good Residential Design Guide)

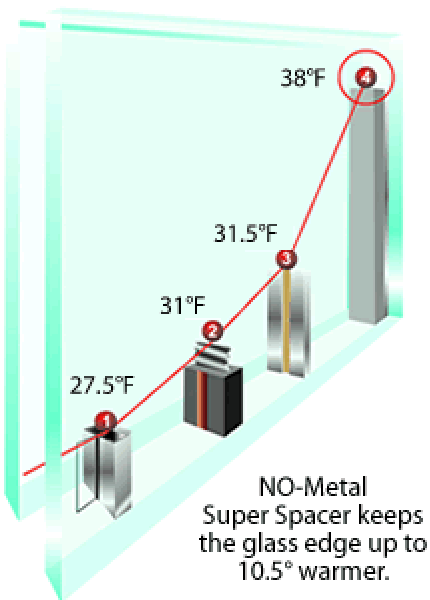


Figure 2.4 No metal super spacer

(Source: Edgetech)

Figure 2.4 shows that by constructing the spacer from materials of lower thermal conductivity the glass edge is kept warmer; reducing the heat transfer through the assembly, increasing the thermal performance of the unit.

Figure 2.5 depicts a solid model of a typical IG unit currently used in glass doors today.

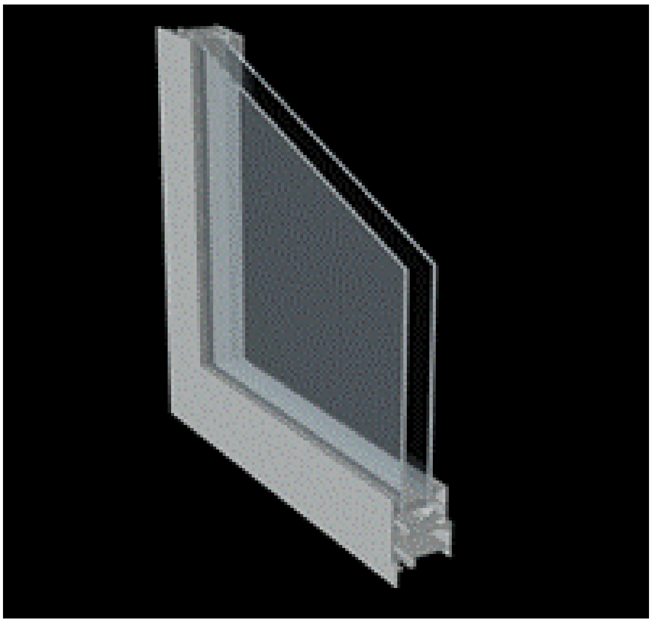


Figure 2.5 Solid model cross section of typical IG unit

(Source: Efficient Window Collaborative)

2.4 Common IG problems

2.4.1 Cold Edge

Condensation occurs around the IG unit's edge - where the glass insulates least effectively and where surface temperatures are the coldest. When a standard Cold Edge spacer exists in the unit the outside temperatures fall very low, causing condensation to form on the glass edge. Figure 2.6 shows that 20% of the pane area is affected by cold edge in an IG unit.

The answer to preventing condensation forming on the glass pane is to increase the thermal efficiency of the edge of the glass, the weak link in the unit's thermal circuit. When the cold edge spacer is substituted with a warm edge spacer the problem is virtually eliminated.

Figure 2.6 IG unit suffering from Cold Edge

(Source: Edgetech)

A Warm Edge Spacer is a spacer made from a material with a much lower thermal conductivity than the conventional aluminium used. The spacer increases the resistance of the thermal circuit as it does not allow heat to flow easily, reducing the amount of heat that is allowed to flow through the unit, increasing the thermal performance. Warm Edge Spacers diminish the likelihood of condensation and mould build up.

Figure 2.7 depicts the effects of the spacer. The full metal spacer has a massive build up of condensation, the less metal spacer contains steel or aluminium wrapped in butyl rubber, resulting in less condensation accumulating.

The No-metal spacer is constructed from polymer structural foam, effectively eliminating the problem.

Figure 2.7 Benefits of Warm Edge Spacer Systems

(Source: Edgetech)

3 Literature Review

Textbooks, engineers, journal articles, the World Wide Web and industry partners were employed to carry out a literature review on the current procedures in IG Unit design.

3.1 Heat Transfer Principles

The boundary conditions that govern the behaviour of our enclosed air space are of great importance; the principles governing the rate of heat transfer addressed by the authors below have been critical in our analysis.

Wang (2000) described the refrigeration load as the rate at which heat is extracted by the evaporated refrigerant at the evaporator. This report stated that heat entering a space is mainly in the form of convective heat and radiative heat transfer. While heat increase through window glass is mainly due to conduction from the temperature difference in the outdoor and indoor temperature. Infiltration was specified as the uncontrolled inward flow of unconditioned outdoor air through cracks and openings in a sealed cavity because of the pressure difference across the cavity. This pressure difference is caused by the outdoor-indoor temperature difference.

Wang (2001) suggests the thermal resistance of an enclosed airspace is related to the surface conditions, average temperature, temperature difference between surfaces at right angles to the heat flow, the air gap between the surfaces and the direction of airflow. A fenestration is also described as the term used for assemblies containing glass or plastic that allow the transmission of light such as framing, mullions and dividers.

3.2 Factors Affecting Heat Transfer

Smith & Adams (1993) conducted a numerical simulation model to study the thermal performance of windows. This model is directly related to our area of work, their preliminary findings can be applied to our research. Their model looks at two dimensional heat transfer by conduction, convection and radiation. The model also calculates the local temperatures, gas velocities, and heat fluxes. The model is capable of varying:

- Glass pane widths
- Heights and spacings
- Frame designs and materials
- Thermal breaks
- Vertical convective coefficients at the inner and outer surfaces
- Multiple gases
- Bars and partitions between the glass panes

Their research looks at the effects of these design parameters on the window thermal performance. The efforts of Smith & Adams (1993) has a direct correlation with our research.

Energy Saving for Windows (ESFW) states that double glazing decreases the heat loss of a single pane of glass but at the same time still allows the transmission of natural light, it also decreases the possibility for condensation to develop on the inside pane.

(ESFW) reports that Low-e glass can and improve performance by up to 38% over standard double glazing.

3.3 Door Construction

Smith & Adams (1993) stated that during the past two decades, window technology has seen the development of double pane windows, low-emittance coatings, between glass panes and gas-filled spaces.

Wang (2001) specifies that most window glasses, or glazing, are vitreous silicate consisting of silicon dioxide, sodium oxide, calcium oxide, and sodium carbonate. They can be classified as:

- Clear plate, sheet glass or plastic
- Tinted heat-absorbing glass
- Insulating glass
- Reflective coated glass
- Low-emissivity

The article also illustrated the exact construction of an IG unit. IG consists of two panes of glass, an outer pane and an inner pane, or three panes separated by metal, foam, or rubber spacers around the edges and hermetically sealed in a stainless steel or aluminium alloy structure. The dehydrated space between the glass panes usually has a thickness of 3.2 – 19 mm and is filled with air, argon, or another inert gas. An air or gas filled space, increases the thermal resistance of the fenestration.

(ESFW) gave their definition of an IG unit. They stated that a double glazed window consist of two panes of glass divided by a sealed air space between 6 mm and 20 mm wide.

Some of these methods are employed in our display doors, the gas fill will be investigated for its feasibility.

4 Analytical Analysis of IG Unit

This chapter looks at Heat Transfer due to the combination of different Transfer modes and presents a theoretical approach to analysing the current IG Unit design.

4.1 Introduction to Heat Transfer

Where a temperature difference exists within a system or two systems at different temperatures make contact with each other, energy is transferred. This energy transport process is known as ***Heat Transfer***. Heat Transfer has three distinct modes of transmission: ***conduction, convection*** and ***radiation***. In the natural world heat is transferred by a number of means working simultaneously. The following sections assess the effects of these transmission modes functioning together.

4.2 Heat Transfer Modes

4.2.1 Conduction

Where a temperature difference exists, heat will flow from the region of higher temperature to the region of lower temperature. The rate at which this heat is transferred by conduction is given as:

$$q_k = -kA\frac{dT}{dx}$$

where: $q_k =$ rate of heat transfer via conduction

$k =$ thermal conductivity of the medium

$A =$ area through which the heat is transferred

$\frac{dT}{dx} =$ temperature gradient

4.2.2 Convection

This heat transfer mode exists in two parts. It consists of a conductive element where the energy transfer is due to molecular motion. The second mode is the macroscopic motion of the surrounding fluid particles. The rate of heat transfer between a surface and its surrounding fluid is given as:

$$q_c = \overline{h}_c A \Delta T$$

where: $q_c =$ rate of heat transfer via convection

$\overline{h}_c$ = average convection heat transfer coefficient

$A =$ area through which the heat is transferred

$\Delta T =$ temperature difference between the surface temperature T_s and the fluid temperature T_∞

4.2.3 Radiation

The amount of heat dissipated from a surface as radiant heat energy can be found from the direct correlation below:

$$q_r = A.F_{12}\sigma(T_1^{\ 4} - T_2^{\ 4})$$

where: $q_r =$ rate of heat transfer via radiation

$A =$ area through which the heat is transferred

$F_{12} =$ shape factor, a dimensionless unit that accounts for the emittance of each body and the physical geometries

$\sigma =$ Stefan-Boltzmann constant of $5.67\times10^{-8}\ \dfrac{W}{m^2.K^2}$

The quantity of heat is not dependant on the surrounding conditions; however, the transfer does require a temperature difference between the surfaces in question.

4.2.4 Combined Heat Transmission

The three fundamental methods of heat transfer were analysed above. In reality heat is transferred by utilising a number of these transfer modes concurrently. Heat Transfer between the panes of the IG Unit occurs by convection and radiation functioning in parallel, whilst the transfer through the panes of glass occurs via conduction with a measurable amount of radiation passing directly through the entire IG system.

4.3 Heat Transfer Across a Double-Pane Window

The following section applies the above theories and analyses the heat losses through the IG unit.

In figure 4.1, the IG Unit is shown in a vertical cross section. Two panes of glass of height (L) and thickness (t) are separated by a distance (δ). The IG Unit is installed inside the door rail with inside air temperature T_i and outside temperature T_o. This analysis determines the equivalent thermal circuit and the rate of heat loss for the Unit.

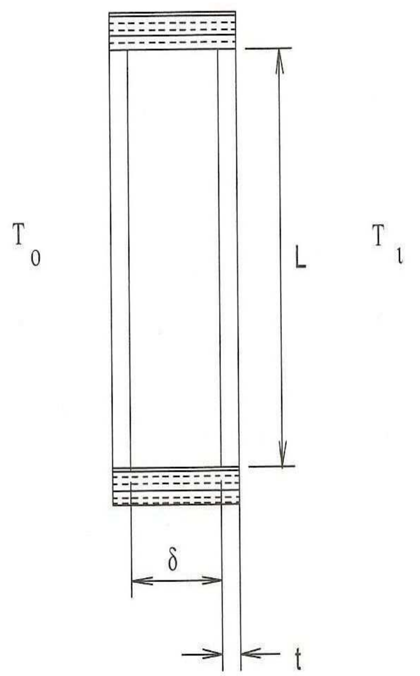


Figure 4.1 Vertical cross section of IG unit for analysis

(Source: Kreith, F & Bohn, M.S.)

4.3.1 Analysis Constraints

$L = 1.5m$ $\quad t = 0.003m$

$\delta = 0.019m$ $\quad T_o = 298K$

$T_i = 273K$ $\quad \sigma = 5.67\times10^{-8}\dfrac{W}{m^2.K^2}$ Stefan-Boltzmann constant

$M = 28.8\dfrac{gm}{gmmole}$ Molecular weight of air

$R = 0.08205\dfrac{L}{gmmole.K}$ Universal gas constant

$k_{glass} = 0.78 \dfrac{watt}{m.K}$ Thermal conductivity of glass at room temperature

$A_{total} = 0.45m^2$ Total surface area of unit

$V = 0.1\dfrac{m}{s}$ Side draft, wind velocity across outside pane

$W = 0.65m$ Width of the Unit in the direction of the draft

4.3.2 The Thermal Circuit

The thermal circuit is depicted in Figure 4.2. It illustrates the combined methods of heat transfer through the different mediums. Progressing from the outside pane to the inner pane, there are two resistors in parallel representing radiation and convection heat transfer from the outer surface of the outer pane to the outside temperature. The outer pane has a thermal resistance due to conduction. In the air gap between the panes a thermal resistance exists similar to that of the outer pane. Two resistors are in parallel due to radiation and convection in the centre from the two panes. Similar to that of the outer pane, the inner pane has a resistance due to conduction. The final two resistors in parallel represent the radiation and convection transfer from the inner surface of the inner pane to the inside temperature.

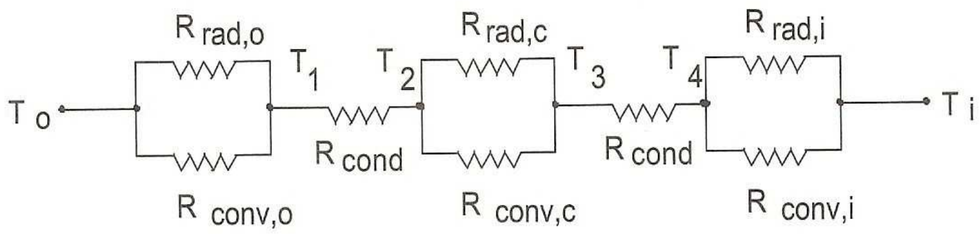


Figure 4.2 The Thermal Circuit

(Source: Kreith, F & Bohn, M.S.)

4.3.3 Assumptions

Outer air temperature and radiation from the outside environment is included in T_o.

Inner air temperature and radiation from the inside environment is included in T_i.

4.3.4 Conduction

The thermal resistance due to conduction is calculated from

$$R_{cond} = \frac{t}{k_{glass}} = 3.846\times10^{-3} . \frac{K.m^2}{watt}$$

4.3.5 Convection

4.3.5.1 Outer Convection

$R_{conv,o}$ is calculated from correlations for flow due to forced convection over a flat surface. The effects of natural convection are thought to be minimal so it has been left out. The Reynolds number for the flow is needed to select the correct correlation. The Reynolds number is calculated from the properties of the air at the average temperature of the outer pane and the outside air. To do this an estimate of T_1 is made based on its fractional position in the thermal circuit.

$$T_{1.estimate} = \frac{4}{5}.T_o + \frac{1}{5}.T_i = 293K \qquad T_{average} = \frac{T_o + T_i estimate}{2} = 295.5K$$

The properties of air can be interpolated at this average temperature, once the complete solution is reached; the temperature estimate can be validated. If it is not valid then the estimate will be adjusted and the solution repeated until convergence.

$$\mu = 1.835 \times 10^{-5} \frac{kg}{m.s}$$

$$\rho = \frac{M}{T_{average.}.R} = 1.188 \frac{kg}{m^3}$$

$$k = 0.0253 \frac{watt}{m.K}$$

$$\Pr = 0.71$$

$$\mathrm{Re} = \frac{\rho . V . W}{\mu} = 4208.17$$

This Reynolds number is less than the critical value of 5×10^{-5}. The boundary layer is laminar and the resulting heat transfer coefficient can be found.

$$Nu = 0.664 . \mathrm{Re}^{0.5} . \mathrm{Pr}^{0.5} = 38.43$$

$$h_c = \frac{Nu . k}{W} = 1.496 \frac{watt}{m^2 . K}$$

$$R_{conv,o} = \frac{1}{h_c} = 0.669 \frac{m^2 . K}{watt}$$

4.3.5.2 Inner Convection

$R_{conv,i}$ is calculated from the Grashof number and correlations, using the properties of air at about room temperature. In order to find these values the following process is followed:

- Estimate T_4
- Solve for the heat transfer
- Calculate T_4
- Reiterate

A weighted average of the inner and outer temperatures is used again to estimate T_4 based on its fractional position in the thermal circuit.

$$T_{4.estimate} = \frac{1}{5} . T_o + \frac{4}{5} . T_i = 278K$$

$$T_{average} = \frac{T_{4.estimate} + T_i}{2} = 275.5K$$

Before the Grashof number can be established, the coefficient of expansion (β), absolute viscosity (μ) and density (ρ) must be found.

For an ideal gas:

$$\beta = \frac{1}{T_{average}} = 3.63 \times 10^{-3}$$

$$\rho = \frac{M}{T_{average}.R} = 1.274 \frac{kg}{m^3}$$

Assuming 1 atm pressure,

$$\mu = 1.7554 \times 10^{-5} \frac{kg}{m.s}$$

$$\Pr = 0.71$$

Grashof Number:

$$Gr = \frac{g.\beta(T_i - T_{4.estimate}).L^3}{\left(\frac{\mu}{\rho}\right)^2} = -3{,}160{,}369{,}911$$

The natural convection flow is therefore turbulent.

$$Nu = 0.13(Gr.\Pr)^{1/3} = 170.19$$

$$k\ @\ T_{average} = 0.023875 \frac{watt}{m.K}$$

$$h_c = \frac{Nu.k}{L} = 2.71 \frac{watt}{m^2.K}$$

$$R_{conv,i} = \frac{1}{h_c} = 0.369 \frac{m^2.K}{watt}$$

4.3.5.3 Natural Convection between the panes

First an estimate of the surface temperatures of the panes is taken using a weighted average based again on their fractional position in the thermal circuit:

$$T_{3.estimate} = \frac{3}{5}.T_i + \frac{2}{5}.T_o = 283K$$

$$T_{2.estimate} = \frac{3}{5}.T_o + \frac{2}{5}.T_i = 288K$$

From these the Grashof Number can be determined using air properties at $T_{4.estimate}$.

$$Gr = \frac{g.\beta(T_{3_{estimate}} - T_{2_{estimate}}).\delta^3}{\left(\frac{\mu}{\rho}\right)^2} = -6{,}461.7$$

Since Gr < 8000, the heat transfer is due to conduction alone. (Source: Kreith, F & Bohn, M.S.)

$$T_{23_{average}} = \frac{T_{2_{estimate}} - T_{3_{estimate}}}{2} = 285.5K$$

$$k @ T_{23_{average}} = 0.0246 \frac{watt}{m.K}$$

$$R_{conv,c} = \frac{\delta}{k} = 0.773 \frac{m^2.K}{watt}$$

4.3.6 Radiation Heat Transfer

Radiation occurs in three locations; from the outside, the inside and the centre.

$$\varepsilon_{glass} = 0.86$$

$$R_{rad,o} = \frac{T_{1_{estimate}} - T_o}{\varepsilon.\sigma.(T_{1_{estimate}}{}^4 - T_o{}^4)} = 0.1987 \frac{m^2.K}{watt}$$

$$R_{rad,i} = \frac{T_{4_{estimate}} - T_i}{\varepsilon.\sigma.(T_{4_{estimate}}{}^4 - T_i{}^4)} = 0.245 \frac{m^2.K}{watt}$$

Now the radiation between the two panes.

4.3.6.1 Shape Factors

$$\frac{W}{\delta} = 34.21$$

$$\frac{L}{\delta} = 78.95$$

$$\therefore F_{1-2} = 1$$

$$F_{12} = \frac{\varepsilon(1+(1-\varepsilon))}{1+(1-\varepsilon)^2} = 0.96$$

4.3.6.2 Radiation heat flux

$$q_r = F_{12}.\sigma.(T_{3_{estimate}}{}^4 - T_{2_{estimate}}{}^4) = 25.34 watts$$

4.3.7 Resistance

$$R_{rad,c} = \frac{T_{3_{estimate}} - T_{2_{estimate}}}{q_r} = 0.197 \frac{m^2.K}{watt}$$

Now to reduce the thermal circuit to one equivalent thermal resistance.

The equivalent resistance for the parallel outside section,

$$R_{p,o} = \frac{R_{rad,o}.R_{conv,o}}{R_{rad,o} + R_{conv,o}} = 0.153 \frac{m^2.K}{watt}$$

The equivalent resistance for the parallel centre section,

$$R_{p,c} = \frac{R_{rad,c}.R_{conv,c}}{R_{rad,c} + R_{conv,c}} = 0.157 \frac{m^2.K}{watt}$$

The equivalent resistance for the parallel inside section,

$$R_{p,i} = \frac{R_{rad,i}.R_{conv,i}}{R_{rad,i} + R_{conv,i}} = 0.147\frac{m^2.K}{watt}$$

Now to add the resistances in series,

$$R_{eq} = R_{p,o} + R_{cond} + R_{p,c} + R_{cond} + R_{p,i} = 0.465\frac{m^2.K}{watt}$$

The value of the temperature at each node needs to be validated and improved if necessary.

Analysing T_1,

$$T_1 = T_o + \left(\frac{T_i - T_o}{R_{eq}}\right).R_{p,o} = 289.77K$$

$T_{1_{estimate}} = 293K$, this value is close enough not to introduce too much error.

Analysing T_4,

$$T_4 = T_i + \left(\frac{T_i - T_o}{R_{eq}}\right).R_{p,i} = 280.9K$$

$$T_{4_{estimate}} = 278K$$

Analysing T_3,

$$T_{3,corrected} = T_4 - \left(\frac{T_i - T_o}{R_{eq}}\right).R_{cond} = 281.1K$$

Analysing T_2,

$$T_{2,corrected} = T_{3,corrected} - \left(\frac{T_i - T_o}{R_{eq}}\right).R_{p,c} = 289.5K$$

Taking the corrected average,

$$T_{23average} = \frac{T_{2,corrected} + T_{3,corrected}}{2} = 285.3K$$

Originally $T_{23average} = 285.5K$, this value is close enough to the estimated value, hence requires no further iterations.

4.3.8 Rate of Heat Transfer

The U-value for the unit is,

$$\frac{1}{R_{eq}} = 2.15\frac{watts}{m^2.K}$$

Although the effects of the aluminium spacer were not fully considered in the analysis, this U-value is a valid one-dimensional estimate for the unit. The estimate of the heat transfer coefficient tends to coincide with values given by IG unit manufacturers. Smith and Adams (1993) reported U-values between $2-4\frac{watts}{m^2.K}$.

The heat transfer through the IG unit on a unit-area basis is,

$$q = \frac{T_i - T_o}{R_{eq}} = -53.8 \frac{watt}{m^2}$$

So what this means is that for every square metre of insulated glass there is on the front of the cabinet 53.8 watts is lost through the door.

The door has a surface area of 0.45m² and a temperature change of 25℃, so the heat transferred through the standard door is 24.21 watts.

5 Numerical Analysis of IG Unit

This chapter illustrates the governing equations, an overview of the CFD software, a description of the door geometry, the boundary conditions and the numerical investigation procedure.

5.1 Governing Equations

The conservation equations used to describe 2-Dimensional fluid flow and heat transfer for laminar flow, steady state and no heat generation are:

5.1.1 Continuity

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$$

5.1.2 Momentum

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial v}{\partial y} \right) = \mu\frac{\partial^2 u}{\partial y^2} - \frac{\partial p}{\partial x}$$

5.1.3 Energy

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)$$

5.2 CFD Software

For research purposes a mainframe set up is preferred with a multiple workstation structure. In CFD analysis there are a large number of computations to be performed, the software requires a computer system with a large amount of computing power. USQ has access to such a system in the engineering CatLab via the Sun Microsystems.

There are a variety of different CFD packages available; however, the engineering faculty already possesses an educational licence for FLUENT. So the obvious choice for the analysis was to use FLUENT.

The numerical analysis was conducted in the engineering CatLab at USQ. Via the CatLab the HPC server was opened providing access to the meshing software GAMBIT and the CFD software FLUENT.

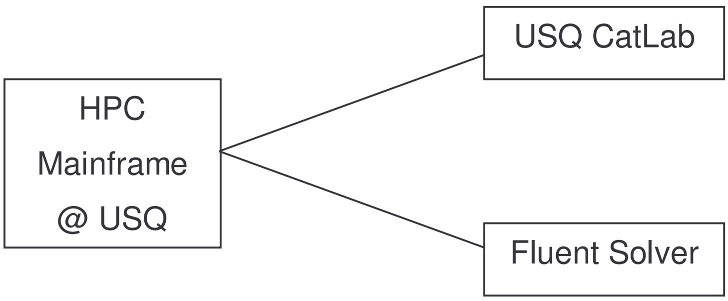


Figure 5.1 Hierarchy of software access

5.3 Fluent Overview

Fluent is at the cutting edge of technology when it comes to CFD software. This software package contains a large range of models to cover all fluid flow situations.

Fluent is proficient in modelling:

- Convection, conduction and radiant heat transfer
- Chemical mixing and reaction
- Discrete phase
- Phase change
- Transient flows
- Free surface flows
- Granular flows

For this analysis the standard heat transfer model has been adopted.

5.4 Boundary Conditions

For the model to accurately present the physical situation, the boundary conditions have to be set so as to simulate the real conditions. The conditions along the inner and outer glass panes and the spacer need to be set.

Along all solid boundaries the velocity conditions are zero in the u and v directions. The boundary layers along the top and bottom spacers of the unit are adiabatic. The table below explains the boundary conditions for each boundary.

Boundary Name	Boundary Type	Thermal Conditions	Free Stream Temperature	Heat Transfer	Wall Thickness
Inner Pane	Wall	Convection	0℃	1.5 W/m². K	0.003 m
Outer Pane	Wall	Convection	25℃	2.7 W/m². K	0.003 m
Spacer	Wall	Convection	25℃	2.5 W/m². K	0.025 m

Table 5.1 List of Boundary Conditions

5.5 Material Properties

The table below describes the properties of the materials utilised in the investigation. These materials are used to construct the glass panes and spacer assembly.

Material	Glass	Aluminium	PVC	Air	Argon
Density (kg/m^3)	2800	2719	1450	Incompressible Ideal Gas	Incompressible Ideal Gas
C_p (J/kg.K)	800	871	1046	1006.43	520.64
Thermal Conductivity (W/m.K)	0.81	202.4	0.15	0.0242	0.0158
Viscosity (kg/m.s)	-	-	-	1.7894×10^{-5}	2.125×10^{-5}

Table 5.2 Material List

5.6 Overall Heat Transfer Coefficient

The heat transfer coefficient or U Factor of the IG unit is defined as

$$q'' = h_f (T_w - T_f) + q''_{rad}$$

where,

$h_f =$ fluid side local heat transfer coefficient

$T_w =$ wall surface temperature

$T_f =$ local fluid temperature

$q'' =$ convective heat flux from the wall

$q''_{rad} =$ radiative heat flux

Calculation of the fluid side heat transfer coefficient is found from

$$h_f = k_f \frac{\partial T}{\partial n} |_{wall}$$

where,

n is the local coordinate normal to the wall

The heat transfer through the unit is due to the sum of the convective and radiative heat flow from the boundaries.

5.7 Numerical Procedure

The CFD program uses a control-volume approach to convert the governing equations or Navier-Stokes equations to be solved numerically. The control-volume approach integrates the governing equations about each control volume yielding discrete equations that conserve each quantity on a control-volume basis. The complete geometry is divided into small control volumes for which the discrete equations are solved.

In order for FLUENT to implement the heat transfer model it solves the energy equation in terms of enthalpy.

$$\frac{\partial}{\partial t}(\rho h)+\frac{\partial}{\partial x_i}(\rho u_i h)=\frac{\partial}{\partial x_i}(k+k_t)\frac{\partial T}{\partial x_i}-\frac{\partial}{\partial x_i}\sum_{j'} h_{j'} J_{j'}+\frac{Dp}{Dt}+(\tau_{ik})_{eff}\frac{\partial u_i}{\partial x_k}+S_h$$

where,

$k =$ molecular conductivity

$k_t =$ conductivity due to turbulent transport $(k_t = c_p \mu_t / \mathrm{Pr}_t)$

$J_{j'} =$ diffusion flux of j'

$S_h =$ source term containing heat of chemical reaction

Enthalpy is defined as

$$h = \sum_{j'} m_{j'} h_{j'}$$

where,

$m_{j'} =$ the mass fraction of j' and

$$h_{j'} = \int_{T_{ref}}^{T} c_{p,j'} dT$$

where,

$T_{ref} = 298.15K$

Convergence is determined by checking the scaled residuals and ensuring they are less than 10^{-5} for all variables, except the energy equation where the residuals must be less than 10^{-6}.

To evaluate the thermal performance, a number of variables have been varied. This project aimed to investigate a variety of IG unit designs. This was achieved by implementing the numerical methodology.

5.7.1 Numerical Methodology

The analysis to evaluate door performance utilised one reference model. The model was copied to separate cases and each variable analysed at a time. Figure 5.2 illustrates the door performance analysis.

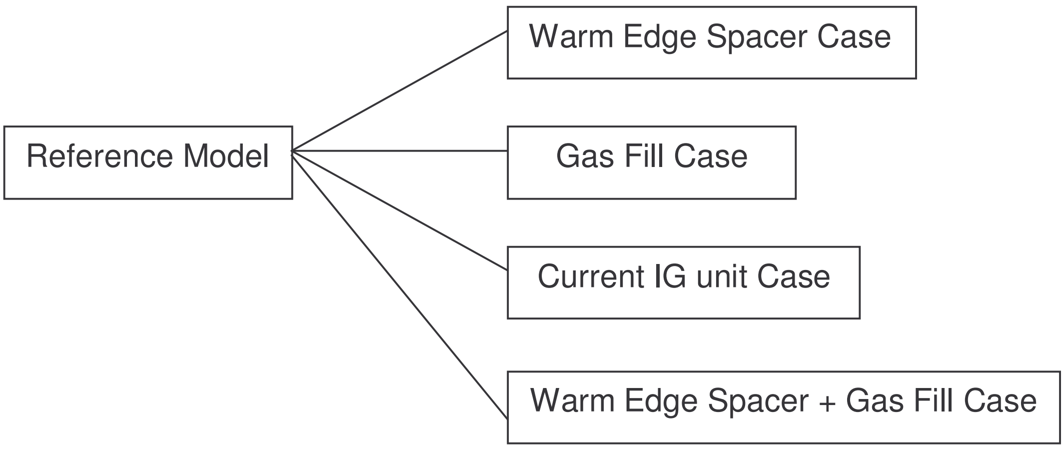


Figure 5.2 Investigation of Door Performance

The analysis to evaluate the effect of gap thickness utilised multiple reference models. Figure 5.3 demonstrates how each case required a separate reference model.

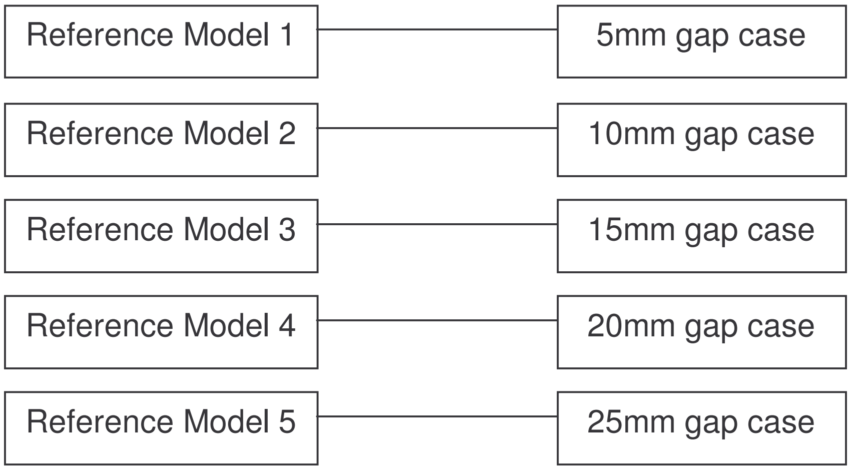


Figure 5.3 Investigation of gap thickness

After planning the CFD analysis the direction taken to solve the cases is shown below.

1. Create model geometry and mesh in GAMBIT
2. Import mesh into FLUENT
3. Check the mesh
4. Choose the heat transfer model to be solved
5. Specify material properties
6. Specify boundary conditions
7. Initialise solution
8. Iterate solution
9. Examine the results and if necessary modify the model
10. Save results

5.7.2 General Solution Variables

With any finite element package, the general solution variables of mesh density are universal. Below is a description of how these variables were addressed.

Mesh Density

A high mesh density produces a closer node spacing resulting in less estimation in the calculations ensuring a more accurate result. For the analysis a node spacing of around 2.5mm - 3mm was used.

6 Results

This chapter provides the results from the numerical investigation. The results are presented in the following sections:

- Geometry Model and Mesh
- Geometry Solution
- Door Performance Analysis
- Gap Thickness Analysis

To examine the various models the analysis is split up into two sections, the Door Performance Analysis and the Gap Thickness Analysis. The door performance investigates the type of gas within the IG unit and the material the spacer is constructed from. The gap thickness analysis investigates the effect the width of the air gap has on the heat loss.

Results of interest from the analysis include the temperature distribution, velocity distribution and the heat loss through the glass panes.

6.1 Geometry Model and Mesh

Before any analysis takes place, the model must be created. Fluent is coupled with a program called GAMBIT that allows complex geometries to be generated and meshed.

It was chosen to model the interior of the IG unit, the void created inside the unit enclosed by the two panes of glass and spacer on the ends. Figure 6.1 illustrates the geometry of the model and the meshing approach taken. As the geometry is quite simple mapped quad elements suited the geometry perfectly.

Depending on the analysis being completed, the mesh spacing varies from around 0.5mm to 4mm. The areas of main interest lie along the boundaries of the geometry so it was decided to increase the mesh gradient in these locations, creating a finer mesh to enhance the accuracy of the results in these regions.

After the geometry and mesh for each model were successfully created in GAMBIT and exported to FLUENT, the analysis of each one could be completed as set out in the numerical methodology.

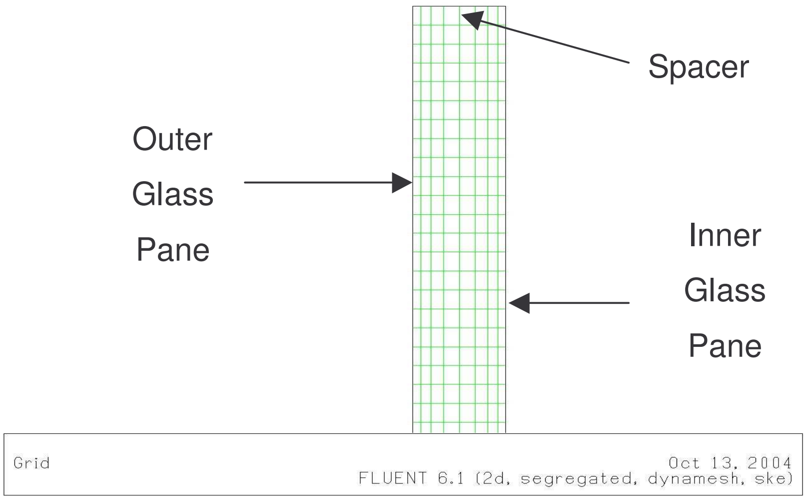


Figure 6.1 Typical Meshed Geometry

6.2 Geometry Solution

The IG unit modelled consists of two glass panes at a thickness of 3mm, separated by a distance of 19mm (except in the gap analysis where this distance is varied). In each model the height of the unit is 1500mm.

After the analysis of the model was iterated the following diagrams illustrate the results taken. The results shown here are for the existing door, comprised of two glass panes each 3 mm thick separated by an aluminium spacer at a distance of 19 mm. The temperature distribution, stream function, and velocity flow field were taken from FLUENT. As the profile of the door is quite thin, the cross sectional images have been zoomed in to illustrate the critical areas.

Appendix C contains the complete numerical analysis including temperature distribution and velocity vectors of the stream function. The complete numerical analysis contains both the Door Performance Analysis and the Gap Thickness Analysis. The plots have been magnified at the areas of interest; this has made some plots larger than others to demonstrate the areas of interest. Due to the plotting and duplicating restrictions, there is not an exact match between the dimensions of all the plots.

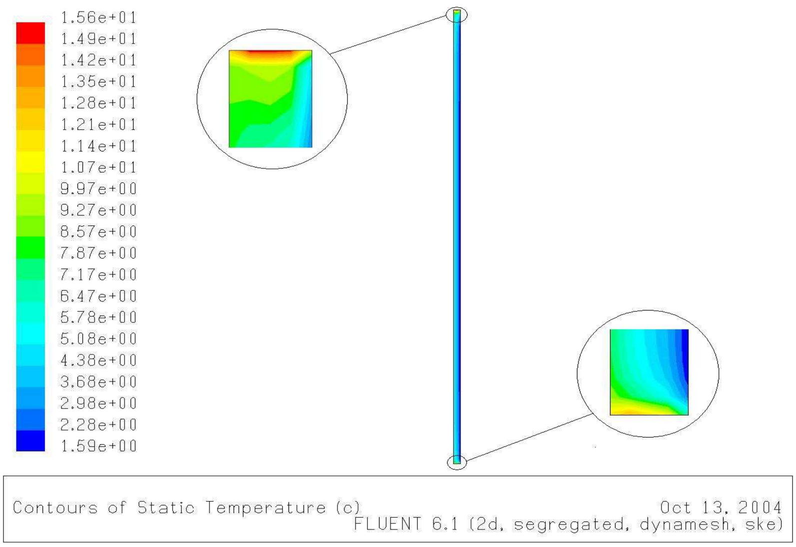


Figure 6.2 Temperature Distribution

The temperature distribution clearly shows the cold region formed on the inner pane closest to the cabinet and the warmer region formed on the outside pane exposed to the ambient air. The effects of the spacer can be seen at the top and bottom of the unit in the red and yellow zones. The increased temperature in these regions compared to the rest of the assembly demonstrates the dramatic heat loss. A major leak in the systems thermal circuit is produced in these regions, this is addressed further on in the spacer heat loss.

The stream function in Figure 6.3 illustrates the range of flow rates of the gas in the space.

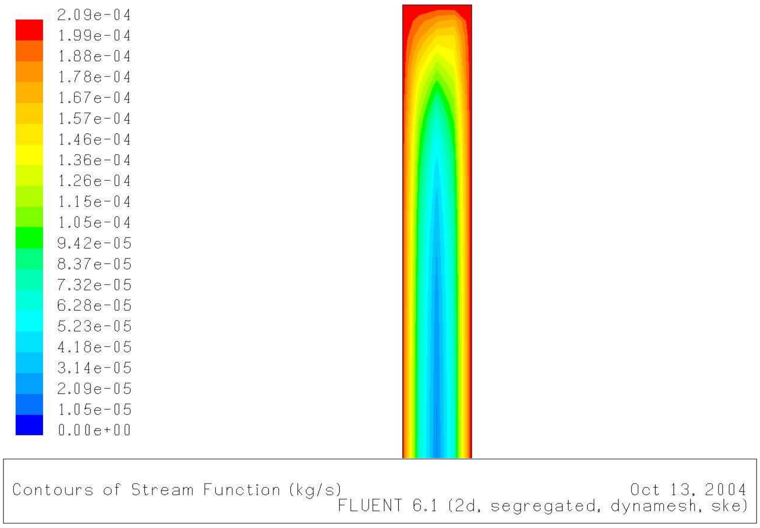


Figure 6.3 Stream Function of Airflow

The vector plot of the stream function in Figure 6.4 demonstrates the magnitude and direction of the flow currents within the unit. The airflow within the unit is in a clockwise direction, this is due to the warm air rising on the outer pane and the cool air falling on the inner pane. The temperature difference between the panes causes this magnitude and direction of flow. As the flow reaches the spacer and changes direction the magnitude of the velocity increases until it returns in the opposite direction in a uniform fashion. This velocity increase is very noticeable in the upper and lower regions of the IG unit.

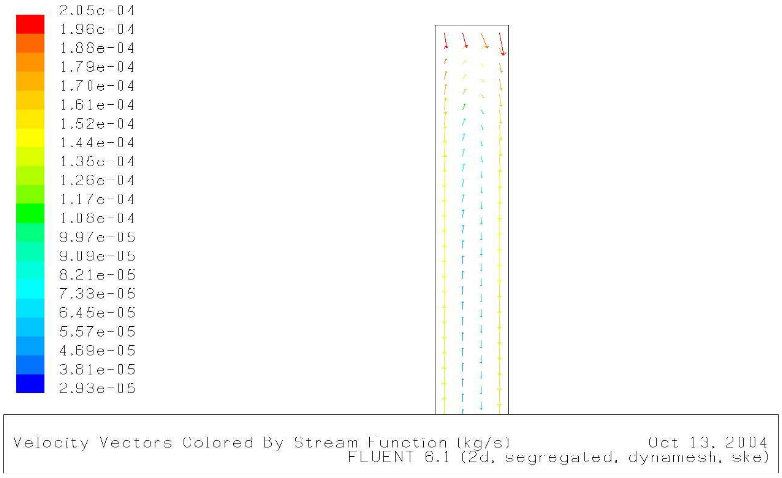


Figure 6.4 Velocity Vectors of Air Flow

6.3 Door Performance Analysis

The door performance investigates how the type of gas within the IG unit and the material the spacer is constructed from alter the heat loss through the unit.

A comparison of the door performance is given in table 6.1. The results of interest from the analysis are shown. The existing door uses an aluminium spacer.

Door Design	Heat Loss (W)
Existing Door	26.22
PVC Spacer	26.01
Argon Fill	23.79
PVC Spacer + Argon Fill	23.72

Table 6.1 Door Performance Analysis

The temperature distribution and velocities (see Appendix C) tend to be parallel in the vertical direction; changing in a linear fashion from the magnitude and direction of the velocity vectors, suggesting that the heat transfer in the gas filled space is due to conduction. The heat loss for each case was obtained from the flux report. The heat transfer rate in this report indicates the amount of heat transfer through each boundary. The heat loss through the inside pane was used to quantify the performance of each design. As the door construction is altered the heat loss decreases.

6.4 Gap Thickness Analysis

The gap thickness analysis investigates how varying the width of the air gap alters the heat loss through the unit.

A comparison of the gap thickness is given in table 6.2. The results of interest from the analysis are shown. All cases are of similar construction to the existing door utilising an aluminium spacer.

Door Design	Heat Loss (W)
5mm	31.18
10mm	27.92
15mm	26.92
20mm	26.11
25mm	25.73

Table 6.2 Gap Thickness Analysis

The results of interest decrease as the gap is increased; the heat loss to the ambient atmosphere tends to be lower at higher gap widths. The next chapter will examine where the optimum thickness lies.

7 Discussion

7.1 Factors Affecting Door Performance

The numerical model for the existing door design given in the door performance results presents a heat loss of 26.22 W, the heat loss obtained from the one dimensional analytical analysis yielded a loss of 24.21 W. It can be seen the analytical and numerical investigations are quite justified as both have produced results of a similar magnitude.

The existing design contains an aluminium spacer, replacing the aluminium with a PVC spacer reduces the heat loss to 26.01 W, although it is only a slight reduction in comparison to the existing door, the heat loss is still reduced. Replacing the air in the space with argon reduces the heat loss to 23.79 W, or a 9% reduction in comparison to the existing door. Combining the PVC spacer and the argon fill in the door reduces the heat loss to 23.72 W, or a 9.5% reduction in comparison to the existing door.

When the PVC spacer and the argon fill are combined it only increases the performance by 0.5% over the argon fill door. The task of evacuating the air from the chamber and replacing it with argon could be a very expensive process. The physical benefits of implementing these changes can be seen from the increase in performance.

From Figure 7.1 it is demonstrated that the heat loss decreases as changes are made to the door design. Making the simple change from aluminium to PVC for the spacer material reduces the heat loss in this region. Incorporating an argon fill reduces the heat loss even more and combining these two concepts reduces it even further. There are major benefits to the thermal efficiency warranting these design changes. From the results it is clear that an argon fill reduces the heat loss substantially and is well worth consideration.

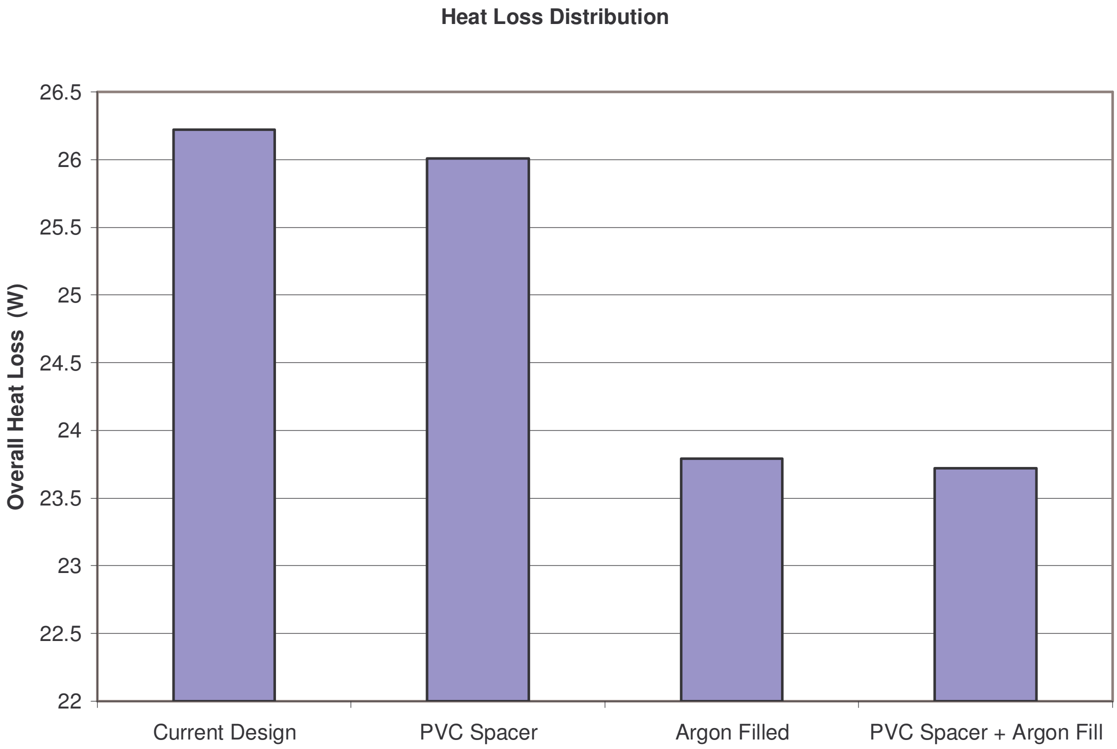


Figure 7.1 Heat Loss Distributions

An additional analysis was conducted to ascertain the effects of using a fine mesh in the investigation. The current design was utilised with a mesh spacing of 0.0015 mm and a spacing ratio of 1.005. With a mesh this fine the convergence time is very high, it took just under 1000 iterations to converge. This analysis yielded a heat loss of 35 W, which seems consistent with the previous results taking into consideration the differences between the meshes. The velocity vectors of the stream function tended to exhibit similar behaviour to the previous results. The temperature distribution; however, did not appear consistent with the previous results. The use of fine mesh analysis will need further investigation; these effects can be examined in future research work. From this it can be inferred that utilising a finer mesh produces results of a higher accuracy, but far more computing time and resources are required. The finer mesh requires around 2000% more convergence time.

7.2 Effect of Gap Thickness

Figure 7.2 displays the effect the width of the air gap has on the heat loss. In general as the width of the space increases the heat loss decreases. The losses decrease rapidly between 5 and 15 mm; however, tending to taper off for larger widths. From this analysis it can be seen that the minimum U-Factor occurs around the range of 15 to 25 mm. The ESFW specifies a minimum gap of 14mm for optimum performance. The current design uses a 19 mm width, which is well within the optimum range specified here.

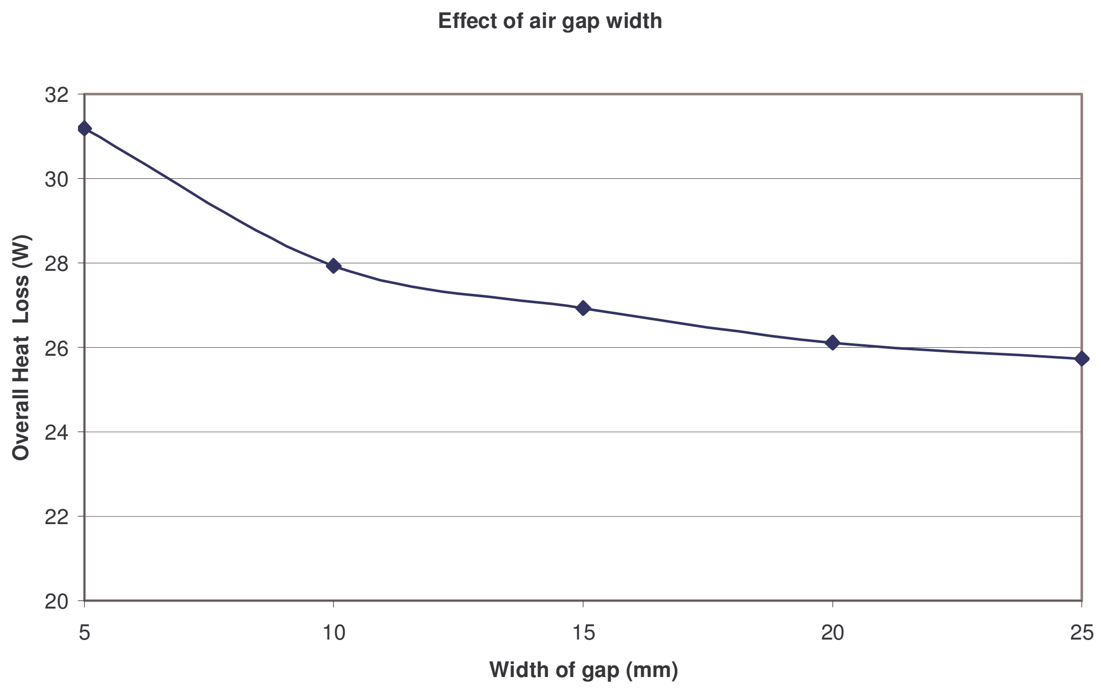


Figure 7.2 Effect of Air Gap Width

8 Conclusions

8.1 Investigation Conclusions

The thermal efficiency of different designs of display refrigeration doors and the major factors that influence thermal efficiency were examined.

With the help and advice of Orford Refrigeration excellent background knowledge on the design and construction of display doors has been achieved.

The computer simulation with the CFD software FLUENT was conducted to investigate the performance of a variety of designs and study the effect of the gap between the glass panes.

A graphical evaluation of the numerical investigation was achieved presenting considerable evidence to recommend an optimum gap thickness and improvements to the design to enhance door performance.

8.2 Summary of Contributions

The project has made substantial contributions to the area of IG unit design. It has been shown that PVC is much better suited as a spacer over aluminium. An argon filled unit increases the door performance considerably. Through the combination of the warm edge spacer system and gas fill the heat loss through the unit can be reduced significantly. The gap thickness plays a major role in determining the heat loss for the IG unit, for optimum performance a gap between 15 - 25 mm is recommended.

8.3 Recommendations for Future Work

A number of possibilities exist for future research in this field. The suggestions for further research are related to the analytical and numerical investigations. The analytical analysis could take into account the effects of the airflow within the unit and the sash design. The feasibility of a 2D analytical investigation could also be considered. The numerical investigation would involve analysing extended design cases to observe how the heat transfer is affected by factors, such as:

- Sash and frame designs
- 3D modelling of the IG unit
- Low-E coating on the inside face of the outer glass pane
- A vertical partition through the middle of the door creating two vertical air currents
- A horizontal bar in the centre of the door creating two separate air currents, in the top and bottom portions of the door
- Type and thickness of glass panes
- A vacuum sealed IG unit

The findings presented by this project form a solid base from which further research could be carried out in IG unit design and its application to display refrigeration doors.

9 References

American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1994, *'1994 ASHRAE HANDBOOK, REFRIGERATION Systems and Applications'*.

Australian Institute of Refrigeration, Air Conditioning and Heating (n.d.) [Online], Available: http://www.airah.org.au/, [Accessed 15 May 2004].

Australian Window Association (n.d.) [Online], Available: http://www.awa.org.au/, [Accessed 20 May 2004].

Edgetech (n.d.) [Online], Available: http://www.edgetechig.com/, [Accessed 5 August 2004].

Efficient Windows Collaborative Website (n.d.) [Online], Available: http://www.efficientwindows.org/index.cfm, [Accessed 19 May 2004].

Energy saving for windows (ESFW), (n.d.) [Online], Available: www.countryenergy.com.au, [Accessed 20 May 2004].

Good Residential Design Guide (n.d.) [Online], Available: http://www.greenhouse.gov.au/yourhome/technical/fs18a.htm, [Accessed 20 May 2004].

Kreith, F & Bohn, M.S., 2001, '*Principles of Heat Transfer'*, 6th Edition, Brooks/Cole, United States of America.

PPG Residential Glass (n.d.) [Online], Available: http://www.ppg.com/gls_residential/share/hom_hot.htm, [Accessed 17 May 2004].

Smith, T.F. & Adams, C.C., 1993, *'A Numerical simulation model for studying the thermal performance of windows'*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., vol. 99, pt 2.

Truseal, Flexible Spacer Technologies (n.d.) [Online], Available: http://www.swiggle.com/, [Accessed 25 August 2004].

Wang, S.K., 2000, *'Air Conditioning and Refrigeration Engineering'*, CRC Press, United States of America.

Wang, S.K., 2001, *'Handbook of Air Conditioning and Refrigeration',* 2nd Edition, McGraw-Hill, United States of America.

Windows and Daylighting Group, Lawrence Berkeley National Laboratory (n.d.) [Online], Available: http://windows.lbl.gov/, [Accessed 18 May 2004].

Windows Energy Rating (n.d.) [Online], Available: http://www.wers.net/mixed.htm, [Accessed 20 May 2004].

Appendix A

Project Specification

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING
ENG 4111/4112 Research Project
PROJECT SPECIFICATION

FOR:	**MICHAEL EAGLE**
TOPIC:	NUMERICAL INVESTIGATION OF THE THERMAL EFFICIENCY OF DIFFERENT DESIGNS OF DISPLAY REFRIGERATION DOORS
SUPERVISORS:	Dr. Ruth Mossad Dr. Harry Ku
ENROLMENT:	ENG 4111 – S1, D, 2004; ENG 4112 – S2, D, 2004
PROJECT AIM:	This project aims to examine the thermal efficiency of different designs of display refrigeration doors, investigate the major factors that influence thermal efficiency and recommend thermally efficient design criterion.
SPONSORSHIP:	Orford Refrigeration
PROGRAMME:	**Issue A, 22nd March 2004**

1. Build the background knowledge necessary for door design of a display refrigeration unit.

2. Study the effect of gap thickness in a display refrigerator door to achieve optimum thermal efficiency.

3. Conduct literature reviews for information on how to improve thermal efficiencies of refrigerator doors.

4. Conduct computer simulation with CFD software 'Fluent' to investigate a variety of designs and evaluate their performances.

5. Liase with Orford Refrigeration Company for their advice and manufacturing processes.

As time permits:

6. Design an optimum thermally efficient display refrigerator door design.

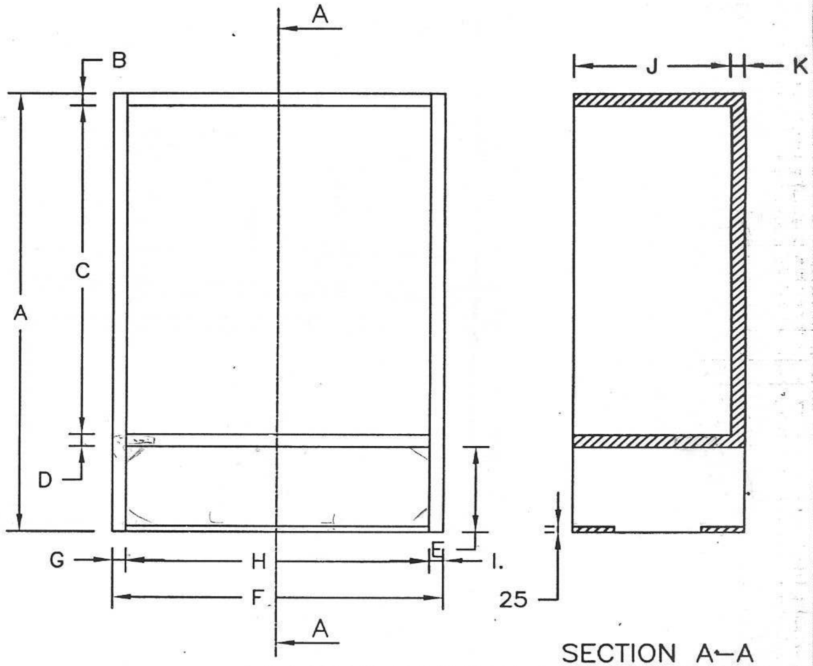
AGREED: __________________ (student) ______________________ (supervisor)

(dated) ………………..

Appendix B

IG Unit & Cabinet Drawings

CABINET	A	B	C	D	E	F	G	H	I	J	K
BM20B/C	1925	53	1447	53	372	756	53	650	53	607	53
BM21C	2025	53	1547	53	372	756	40	650	53	607	53
BM30C	1925	53	1447	53	372	1094	53	988	53	607	53
BM36C	1925	53	1447	53	372	1286	53	1180	53	607	53
CC36	1925	53	1447	53	372	1286	53	1180	53	607	53
ES10	1310	50	892	53	315	670	53	564	53	510	50
ES12	1565	50	1147	53	315	670	53	564	53	510	50
ES14	1815	50	1397	53	315	670	53	564	53	510	50
ES15	1905	50	1487	53	315	670	53	564	53	510	50
FLM272PT	1560	45	1158	53	304	1070	40	990	40	500	0
FM20	1925	53	1447	53	372	756	53	650	53	607	53
FM30	1925	53	1447	53	372	1094	53	988	53	607	53
FM30PT	1925	53	1447	53	372	1094	53	988	53	607	53
FM36	1925	53	1447	53	372	1286	53	1180	53	607	53
FM36PT	1925	53	1447	53	372	1286	53	1180	53	607	0
FM45	1925	53	1447	53	372	1640	53	1534	53	607	53
FM60	1925	53	1447	53	372	2170	53	2064	53	607	53

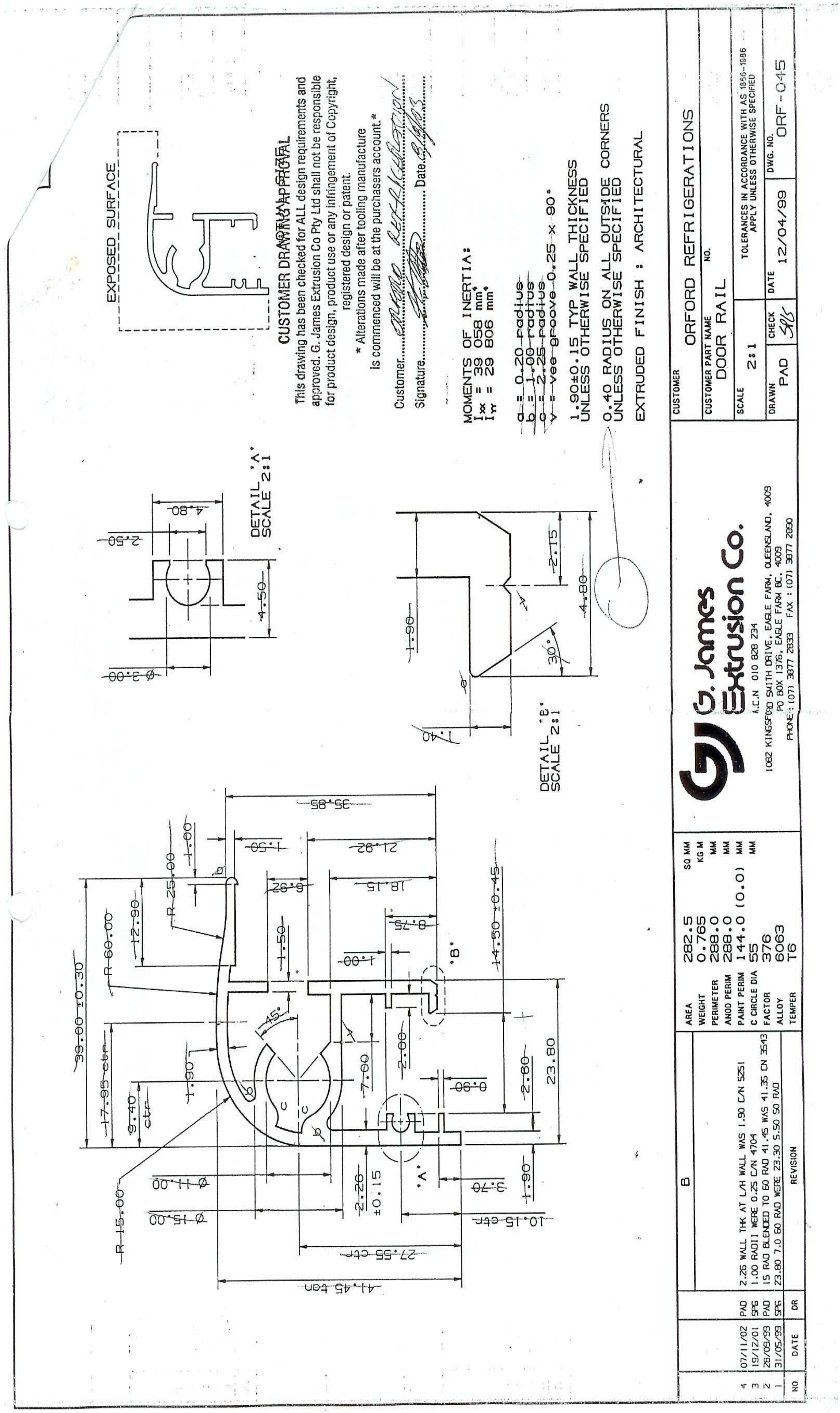


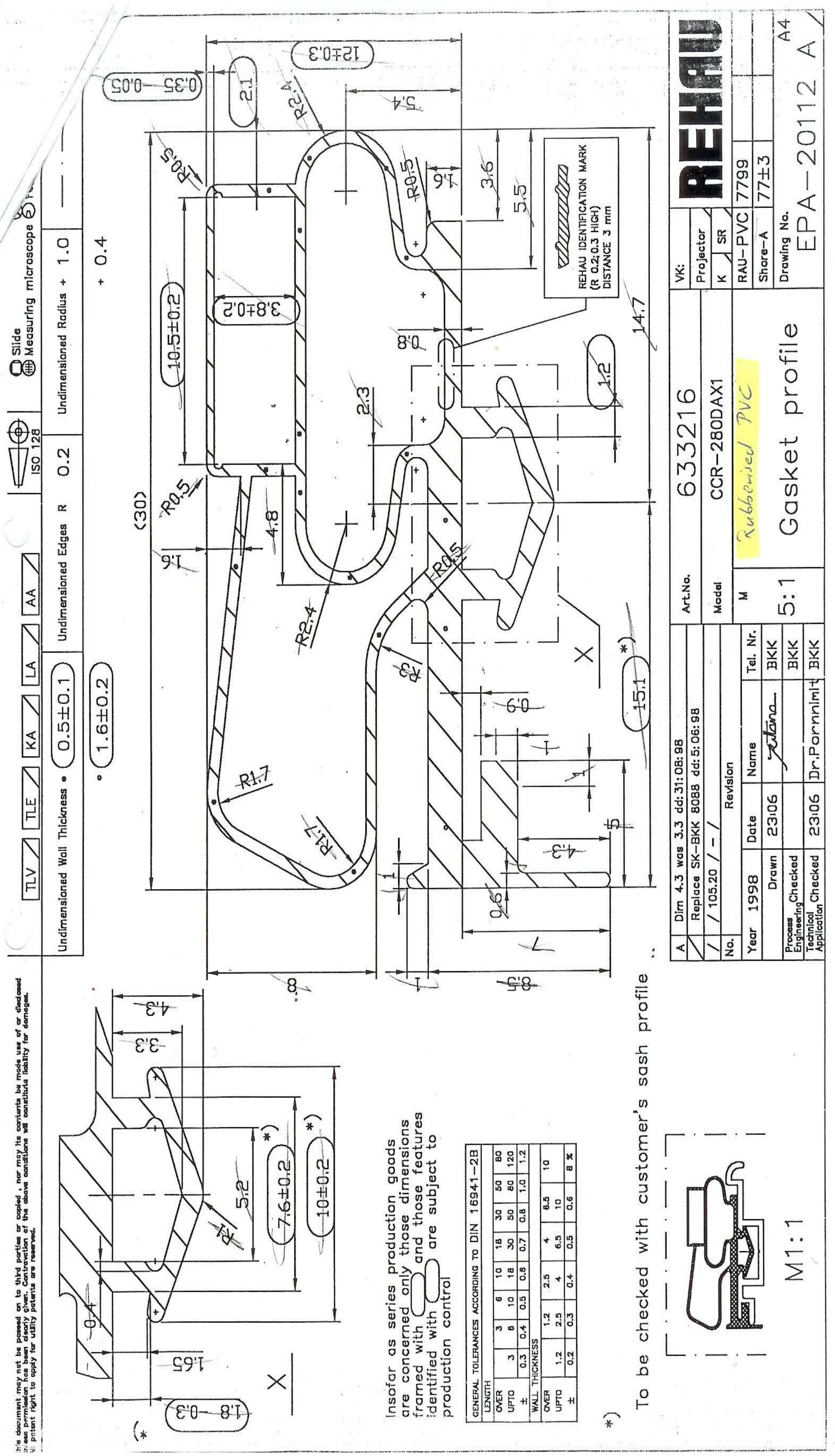
PART– CABINET DIMENSIONS
DETAIL–

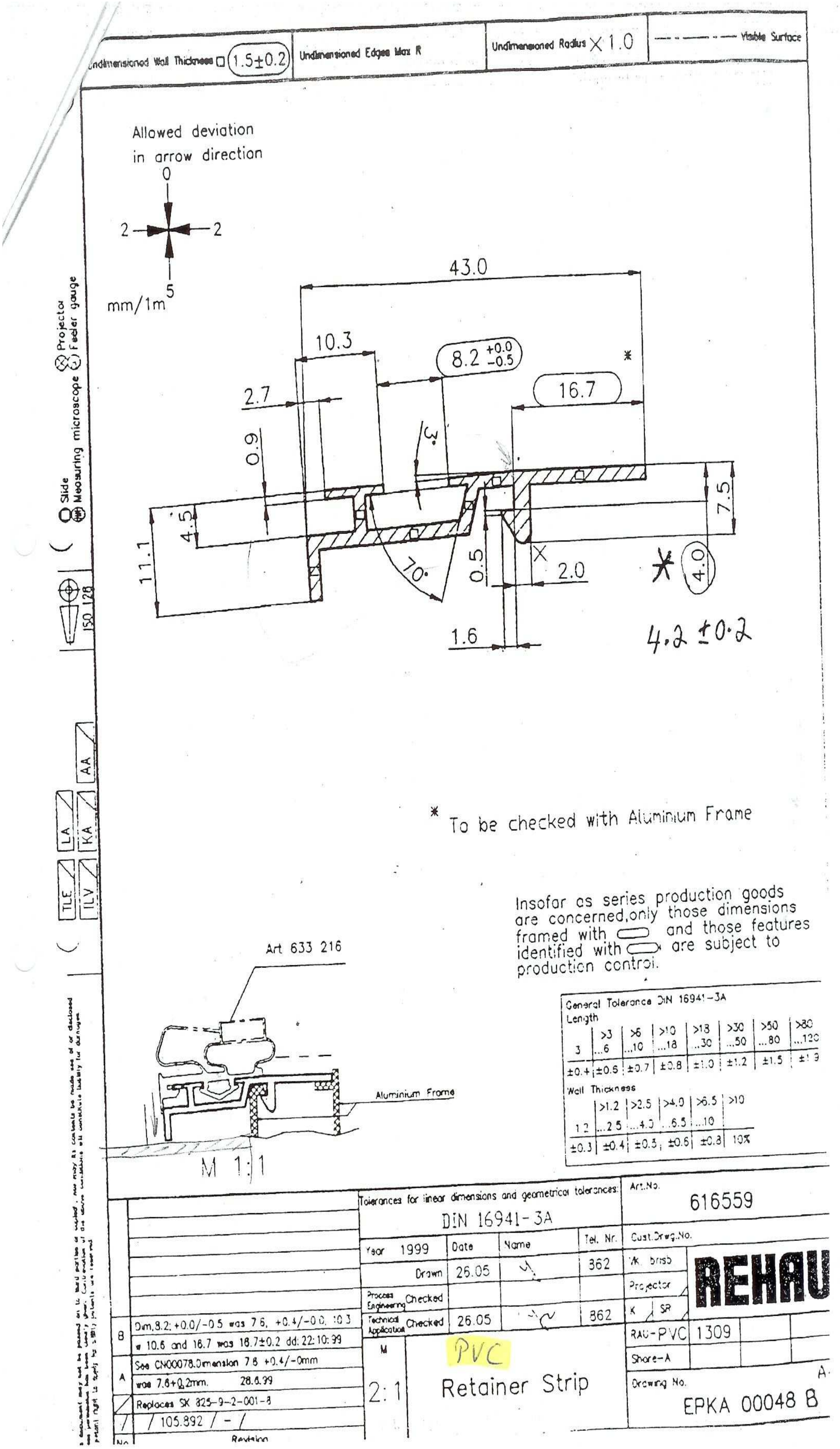
WGT	08/09/1999	Corrected Dimensions E.	VER.	1 14-9-98
WGT	19/10/1999	Removed extra usless column.	DRWN.	PRC
SJF	18/01/2000	Added IM25 & IM50	APPD.	
WGT	28/05/2003	Added ES10 & ES14	ISSUED	

DWG No.CAB_DIMS DIRECTORY: \CAB_DRAW

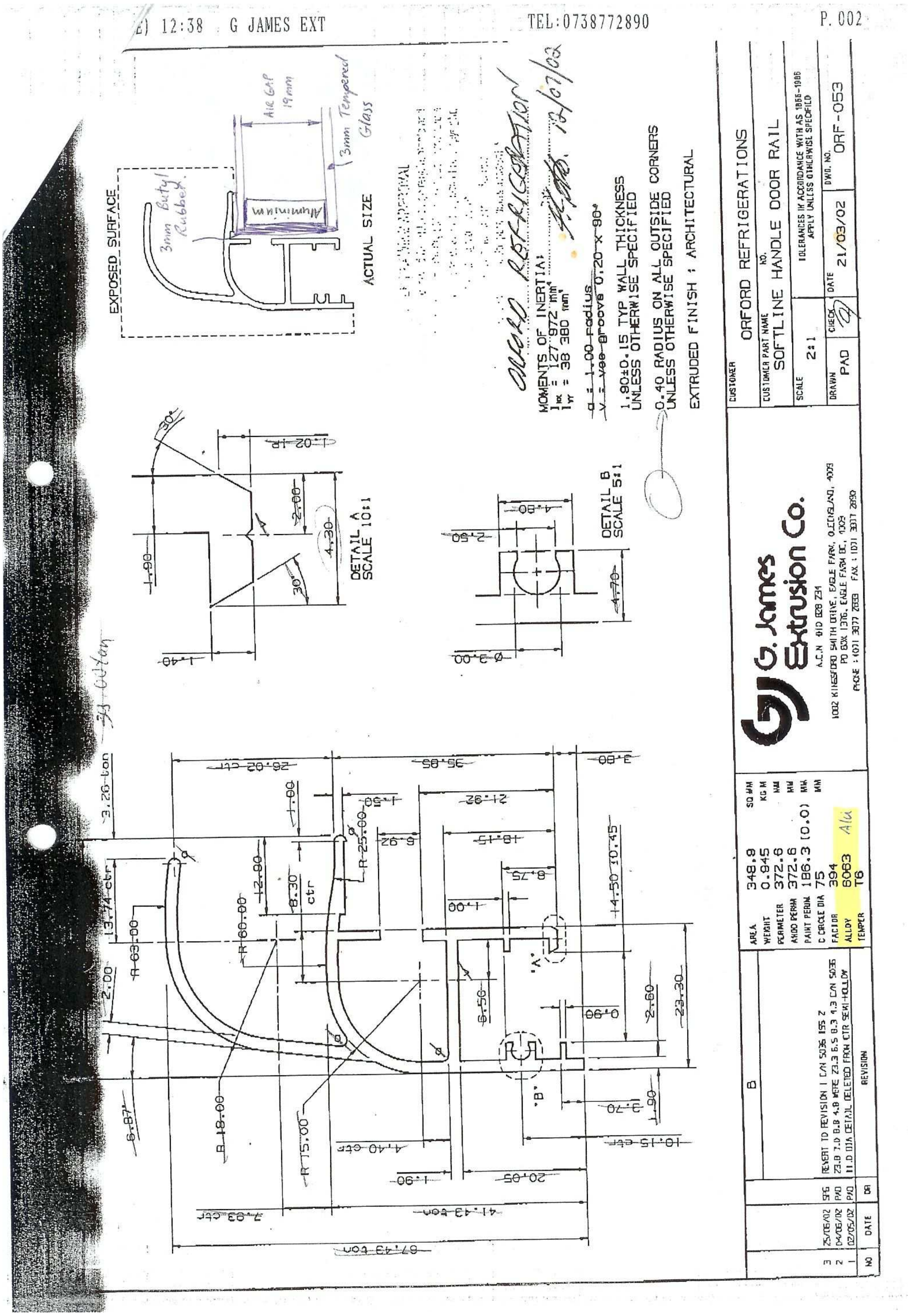
ORFORD REFRIGERATION. A4


EXPOSED SURFACE
CUSTOMER DRAWING APPROVAL
This drawing has been checked for ALL design requirements and approved. G. James Extrusion Co Pty Ltd shall not be responsible for product design, product use or any infringement of Copyright, registered design or patent.
* Alterations made after tooling manufacture is commenced will be at the purchasers account.*
Customer.
Signature.
Date.
MOMENTS OF INERTIA:
Ixx = 39 058 mm4
Iyy = 29 806 mm4
1.90±0.15 TYP WALL THICKNESS UNLESS OTHERWISE SPECIFIED
0.40 RADIUS ON ALL OUTSIDE CORNERS UNLESS OTHERWISE SPECIFIED
EXTRUDED FINISH : ARCHITECTURAL
DETAIL 'A' SCALE 2:1
DETAIL 'B' SCALE 2:1
CUSTOMER ORFORD REFRIGERATIONS
CUSTOMER PART NAME DOOR RAIL
SCALE 2:1
DRAWN PAD
DATE 12/04/99
DWG. NO. ORF-045
TOLERANCES IN ACCORDANCE WITH AS 1866-1986 APPLY UNLESS OTHERWISE SPECIFIED
G. James Extrusion Co.
AREA 282.5 SQ MM
WEIGHT 0.765 KG M
PERIMETER 288.0 MM
ANOD PERIM 288.0 MM
PAINT PERIM 144.0 (0.0) MM
C CIRCLE DIA 55 MM
FACTOR 376
ALLOY 6063
TEMPER T6
4 07/11/02 PAD 2.26 WALL THK AT L/H WALL WAS 1.90 C/N 5251
3 19/12/01 SPG 1.00 RADII WERE 0.25 C/N 4704
2 28/05/99 PAD 15 RAD BLENDED TO 60 RAD 41.45 WAS 41.35 CN 3543
1 31/05/99 SPG 23.80 7.0 60 RAD WERE 23.30 5.50 50 RAD
NO DATE DR REVISION


REHAU
Art.No. 633216
Model CCR-280DAX1
Rubberised PVC
Gasket profile
5:1
Drawing No. EPA-20112 A
A4
RAU-PVC 7799
Shore-A 77±3
REHAU IDENTIFICATION MARK (R 0.2;0.3 HIGH) DISTANCE 3 mm
Insofar as series production goods are concerned only those dimensions framed with and those features identified with are subject to production control
To be checked with customer's sash profile
M1:1
Undimensioned Wall Thickness
Undimensioned Edges R 0.2
Undimensioned Radius + 1.0
Slide
Measuring microscope
ISO 128
GENERAL TOLERANCES ACCORDING TO DIN 16941-2B



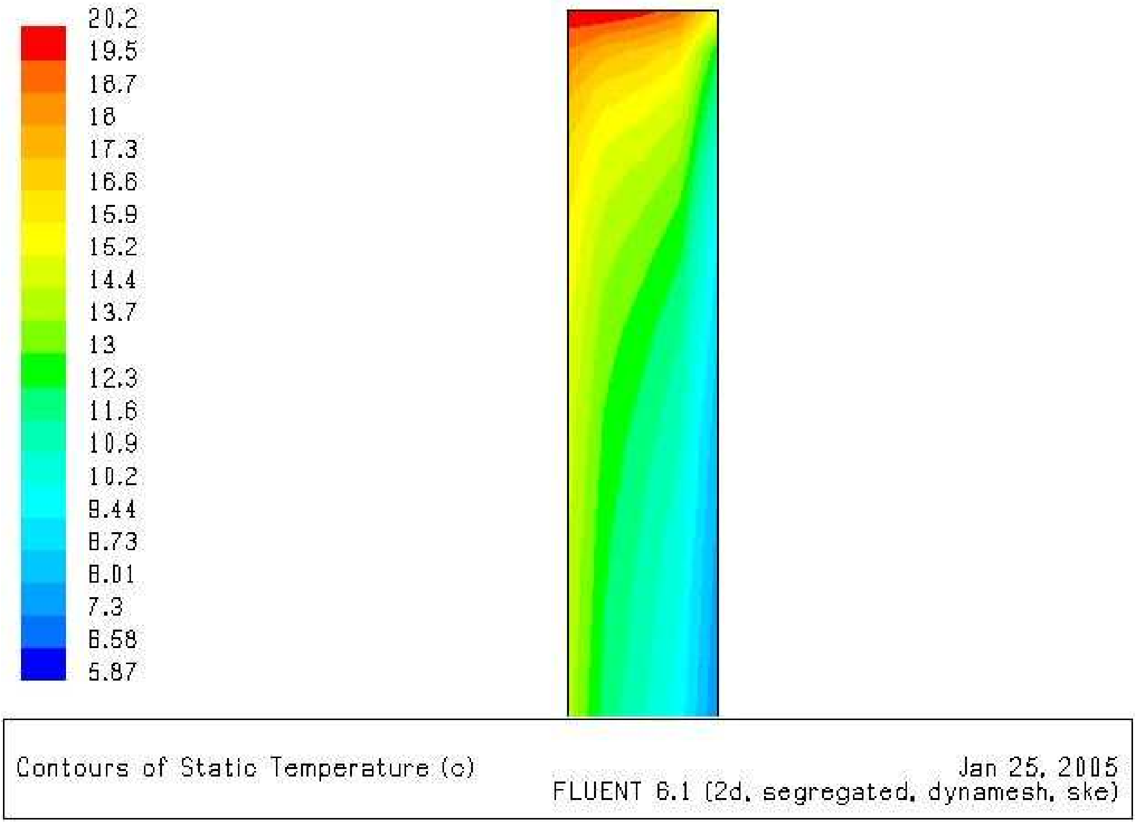
Undimensioned Wall Thickness 1.5±0.2
Undimensioned Edges Max R
Undimensioned Radius X 1.0
Visible Surface
Allowed deviation
in arrow direction
mm/1m
43.0
10.3
8.2 +0.0 -0.5
16.7
2.7
0.9
4.5
11.1
70°
0.5
2.0
1.6
7.5
4.0
4.2 ±0.2
* To be checked with Aluminium Frame
Insofar as series production goods are concerned, only those dimensions framed with ⊂⊃ and those features identified with ⊂⊃ are subject to production control.
Art 633 216
Aluminium Frame
M 1:1
General Tolerance DIN 16941-3A
Length
Wall Thickness
Tolerances for linear dimensions and geometrical tolerances:
DIN 16941-3A
Art.No.
616559
Year 1999
Date
Name
Tel. Nr.
Drawn 26.05
362
Checked
Checked 26.05
362
Cust.Drwg.No.
Projector
REHAU
RAU-PVC 1309
Shore-A
PVC
2:1
Retainer Strip
Drawing No.
EPKA 00048 B
Dim.8.2: +0.0/-0.5 was 7.6, +0.4/-0.0, 10.3
w 10.6 and 16.7 was 16.7±0.2 dd: 22.10.99
See CN00078.Dimension 7.6 +0.4/-0mm
was 7.6+0.2mm. 28.6.99
Replaces SK 825-9-2-001-8
105.892
Revision
Slide
Measuring microscope
Projector
Feeler gauge
ISO 128
TLE
ILV
LA
KA
AA

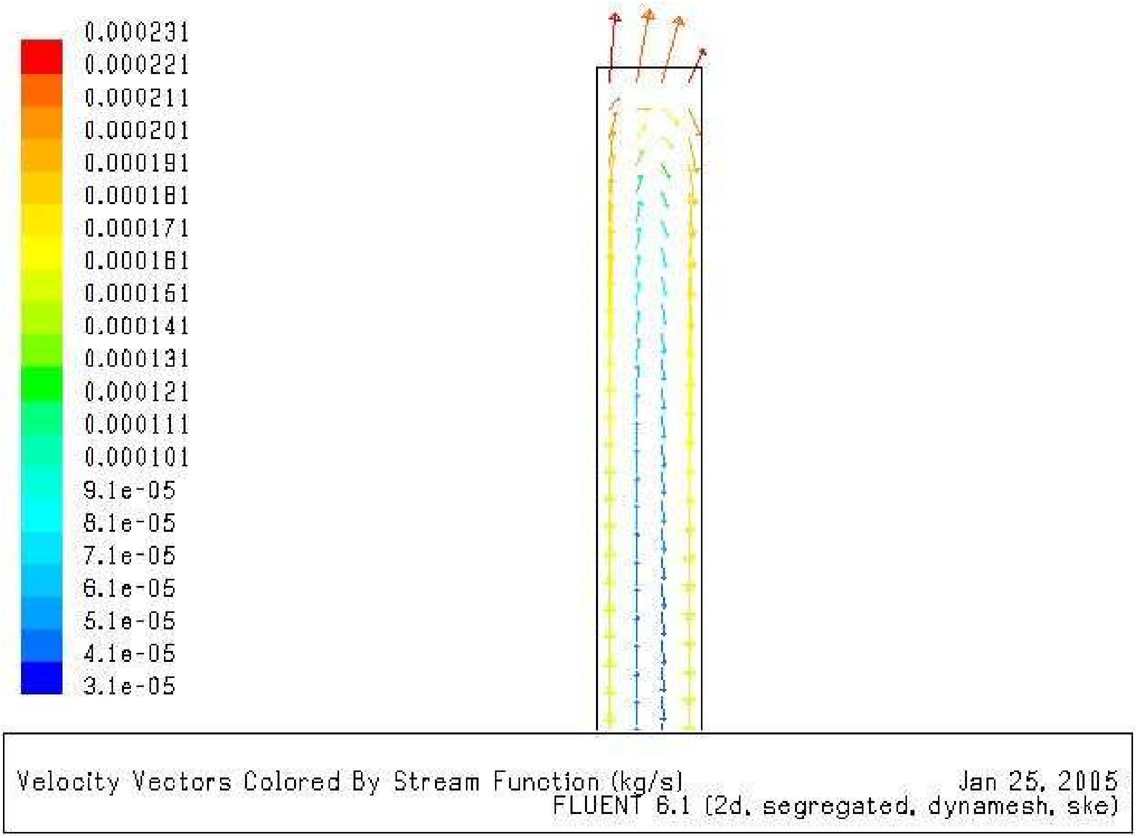

12:38 G JAMES EXT
TEL:0738772890
P.002
EXPOSED SURFACE
3mm Butyl Rubber
Aluminium
Air GAP 19mm
3mm Tempered Glass
ACTUAL SIZE
ORFORD REFRIGERATION
MOMENTS OF INERTIA:
Ixx = 127 972 mm⁴
Iyy = 38 380 mm⁴
1.90±0.15 TYP WALL THICKNESS
UNLESS OTHERWISE SPECIFIED
0.40 RADIUS ON ALL OUTSIDE CORNERS
UNLESS OTHERWISE SPECIFIED
EXTRUDED FINISH : ARCHITECTURAL
DETAIL A
SCALE 10:1
DETAIL B
SCALE 5:1
CUSTOMER
ORFORD REFRIGERATIONS
CUSTOMER PART NAME
SOFTLINE HANDLE DOOR RAIL
SCALE 2:1
TOLERANCES IN ACCORDANCE WITH AS 1866-1986 APPLY UNLESS OTHERWISE SPECIFIED
DRAWN PAD
DATE 21/03/02
DWG. NO. ORF-053
G. James Extrusion Co.
AREA 348.9 SQ MM
WEIGHT 0.945 KG M
PERIMETER 372.6 MM
ANOD PERIM 372.6 MM
PAINT PERIM 186.3 (0.0) MM
C CIRCLE DIA 75 MM
FACTOR 394
ALLOY 6063 Alu
TEMPER T6
REVISION

Appendix C

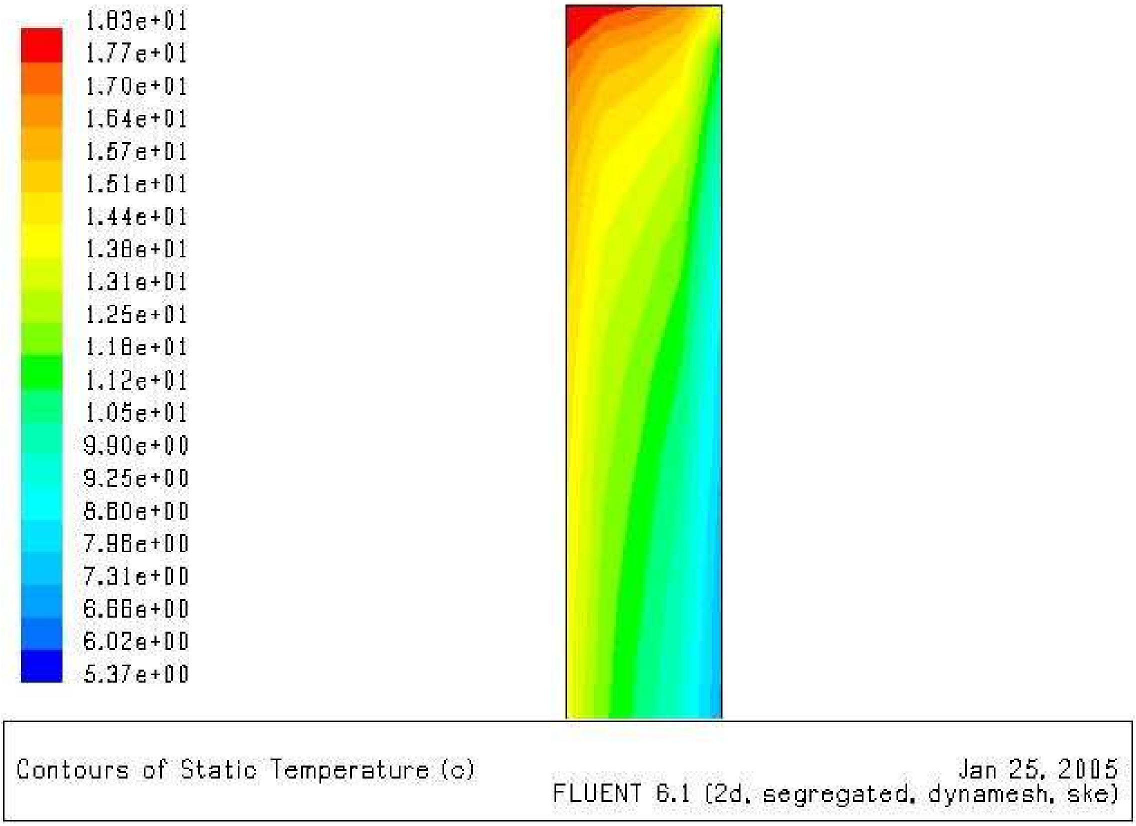
Numerical Modelling Results

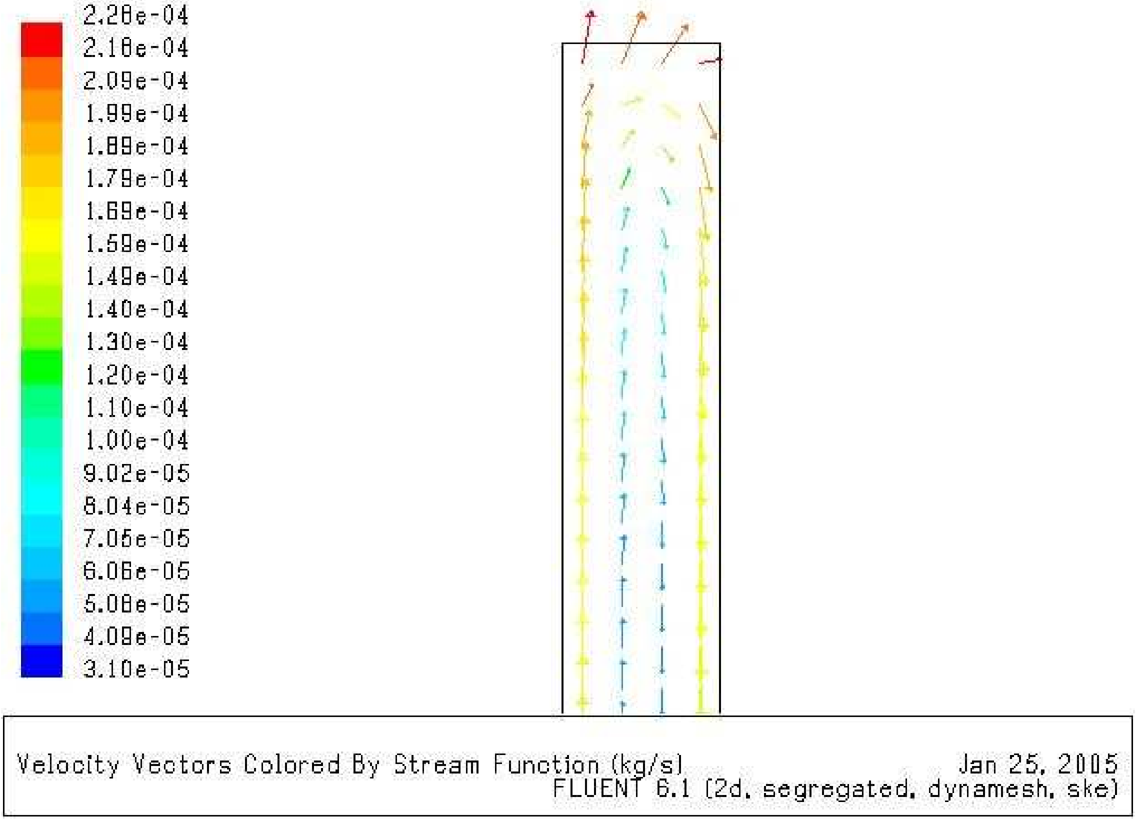
Analysis 1 - Current IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	4.75mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	Aluminium	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s²	
Temperature Distribution & Fluxes		
Heat Loss	26.22	W


20.2
19.5
18.7
18
17.3
16.6
15.9
15.2
14.4
13.7
13
12.3
11.6
10.9
10.2
9.44
8.73
8.01
7.3
6.58
5.87
Contours of Static Temperature (c)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

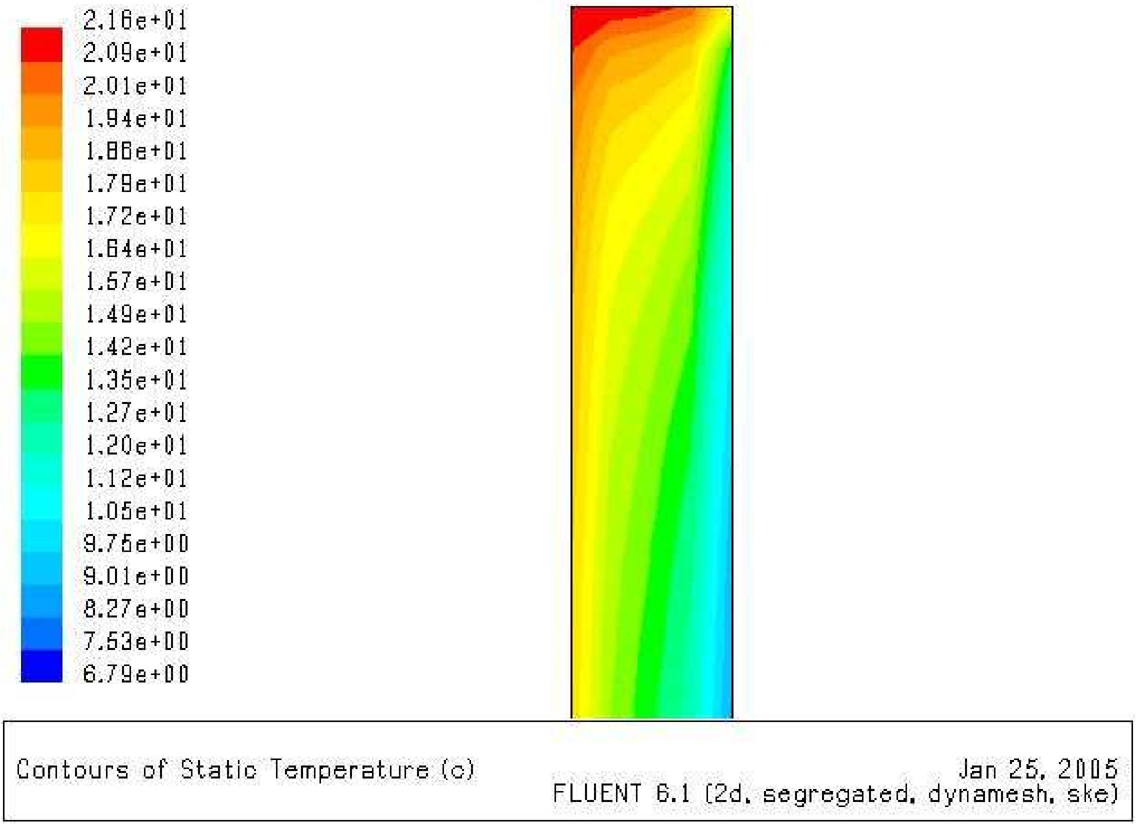

0.000231
0.000221
0.000211
0.000201
0.000191
0.000181
0.000171
0.000161
0.000151
0.000141
0.000131
0.000121
0.000111
0.000101
9.1e-05
8.1e-05
7.1e-05
6.1e-05
5.1e-05
4.1e-05
3.1e-05
Velocity Vectors Colored By Stream Function (kg/s)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

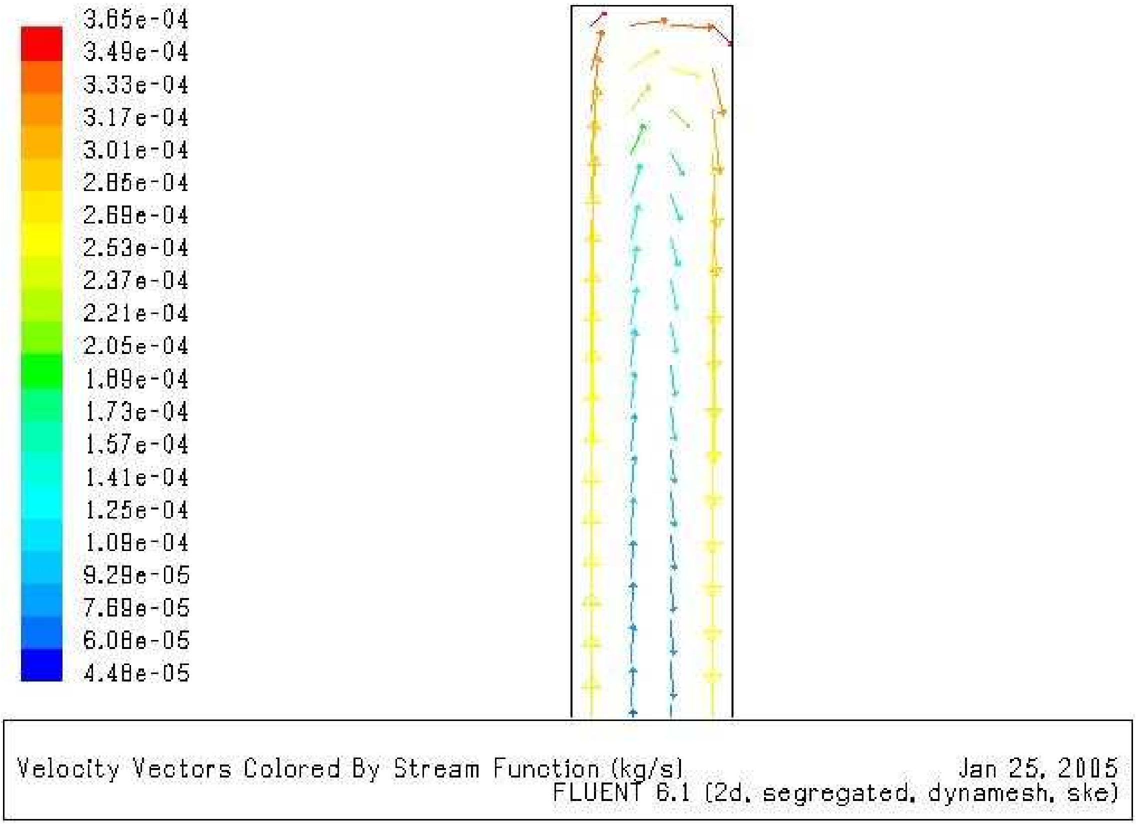
Analysis 2 – Argon Filled IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	4.75mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	Aluminium	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s²	
Temperature Distribution & Fluxes		
Heat Loss	26.01	W


1.83e+01
1.77e+01
1.70e+01
1.64e+01
1.57e+01
1.51e+01
1.44e+01
1.38e+01
1.31e+01
1.25e+01
1.18e+01
1.12e+01
1.05e+01
9.90e+00
9.25e+00
8.60e+00
7.96e+00
7.31e+00
6.66e+00
6.02e+00
5.37e+00
Contours of Static Temperature (c)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

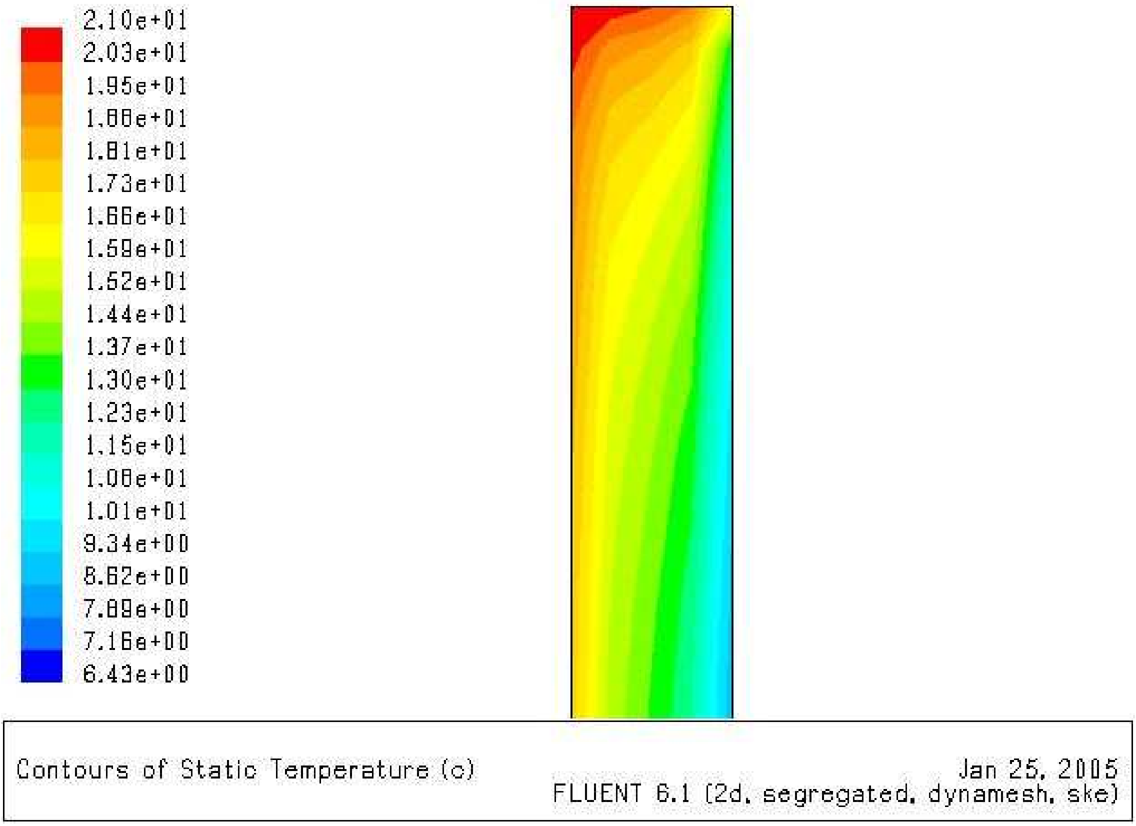

2.28e-04
2.18e-04
2.09e-04
1.99e-04
1.89e-04
1.79e-04
1.69e-04
1.59e-04
1.49e-04
1.40e-04
1.30e-04
1.20e-04
1.10e-04
1.00e-04
9.02e-05
8.04e-05
7.05e-05
6.06e-05
5.08e-05
4.09e-05
3.10e-05
Velocity Vectors Colored By Stream Function (kg/s)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

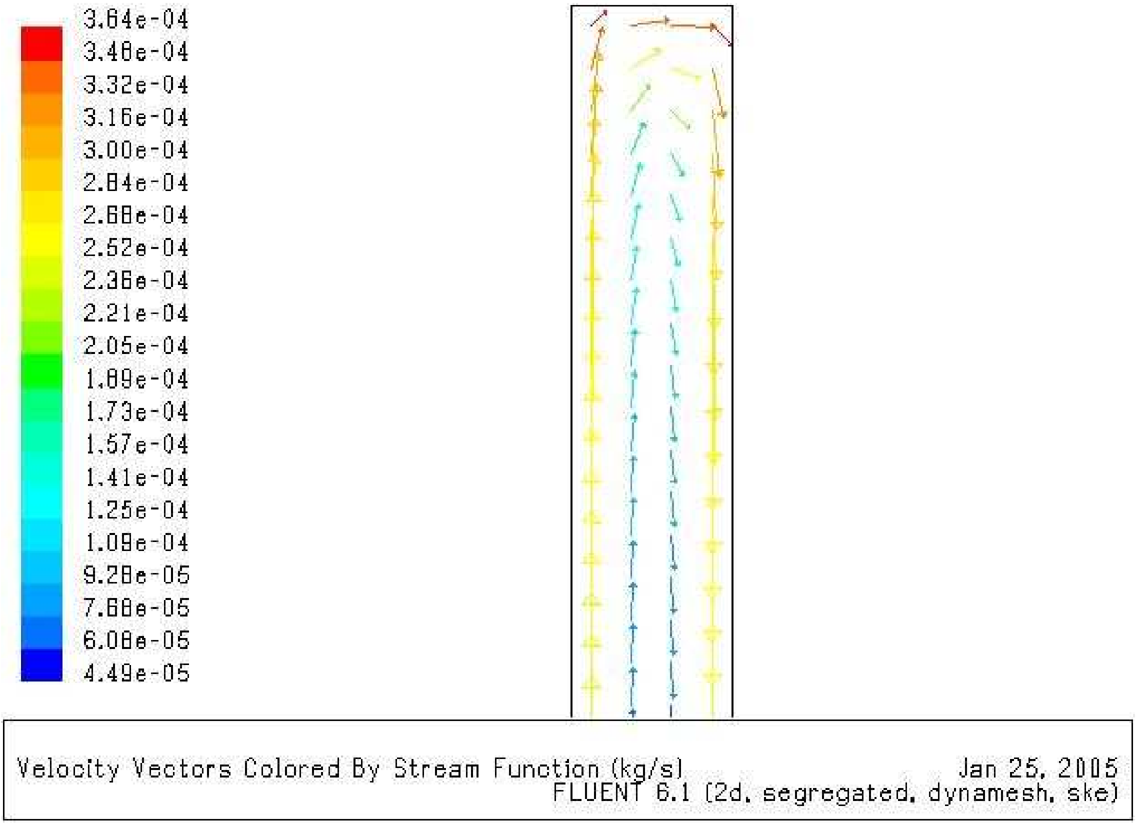
Analysis 3 – PVC Spacer IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	4.75mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	PVC	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s²	
Temperature Distribution & Fluxes		
Heat Loss	23.79	W


2.16e+01
2.09e+01
2.01e+01
1.94e+01
1.86e+01
1.79e+01
1.72e+01
1.64e+01
1.57e+01
1.49e+01
1.42e+01
1.35e+01
1.27e+01
1.20e+01
1.12e+01
1.05e+01
9.75e+00
9.01e+00
8.27e+00
7.53e+00
6.79e+00
Contours of Static Temperature (c)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

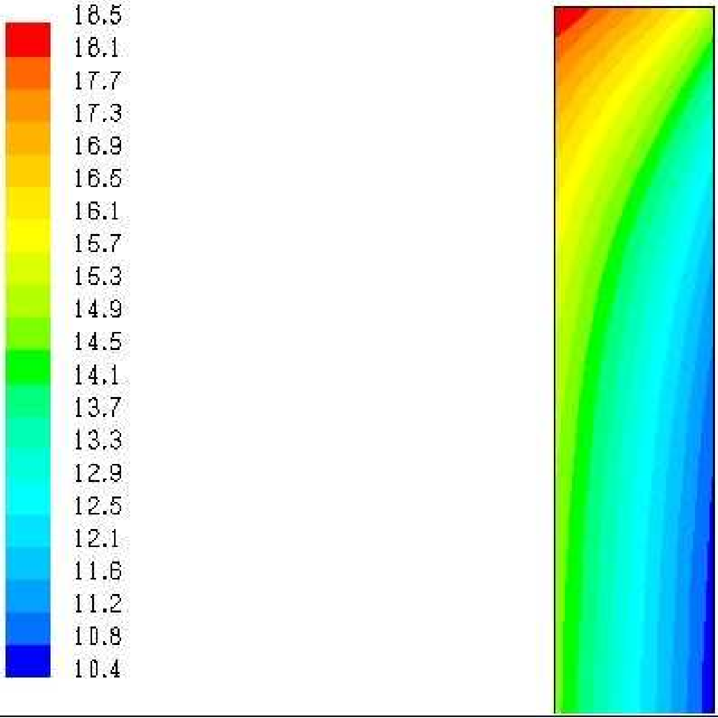

3.65e-04
3.49e-04
3.33e-04
3.17e-04
3.01e-04
2.85e-04
2.69e-04
2.53e-04
2.37e-04
2.21e-04
2.05e-04
1.89e-04
1.73e-04
1.57e-04
1.41e-04
1.25e-04
1.09e-04
9.29e-05
7.69e-05
6.08e-05
4.48e-05
Velocity Vectors Colored By Stream Function (kg/s)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

Analysis 4 – Argon Filled + PVC Spacer IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	4.75mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	PVC	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s²	
Temperature Distribution & Fluxes		
Heat Loss	23.72	W

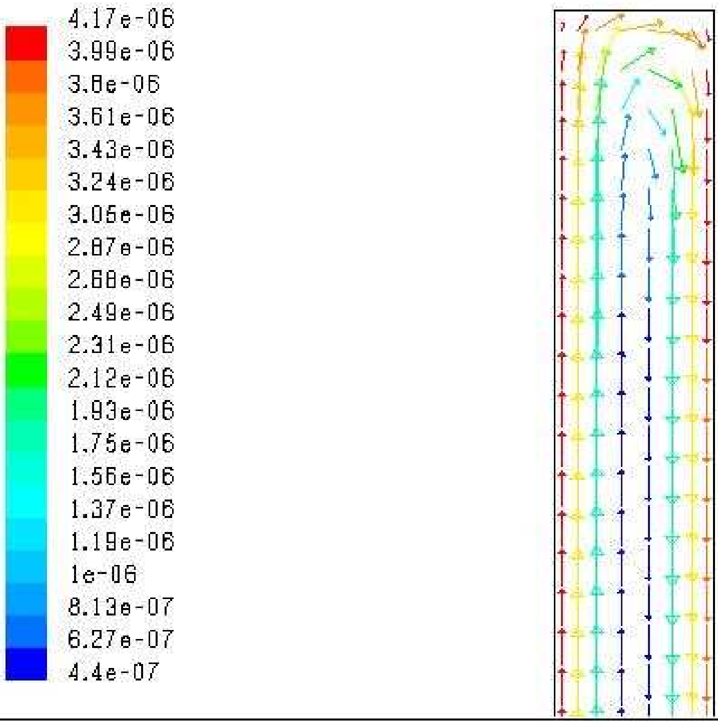

2.10e+01
2.03e+01
1.95e+01
1.88e+01
1.81e+01
1.73e+01
1.66e+01
1.59e+01
1.52e+01
1.44e+01
1.37e+01
1.30e+01
1.23e+01
1.15e+01
1.08e+01
1.01e+01
9.34e+00
8.62e+00
7.89e+00
7.16e+00
6.43e+00
Contours of Static Temperature (c)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)


3.64e-04
3.48e-04
3.32e-04
3.16e-04
3.00e-04
2.84e-04
2.68e-04
2.52e-04
2.36e-04
2.21e-04
2.05e-04
1.89e-04
1.73e-04
1.57e-04
1.41e-04
1.25e-04
1.09e-04
9.28e-05
7.68e-05
6.08e-05
4.49e-05
Velocity Vectors Colored By Stream Function (kg/s)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

Analysis 5 – 5mm IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1.25	
Mesh Spacing	0.625mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	Aluminium	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s²	
Temperature Distribution & Fluxes		
Heat Loss	31.18	W

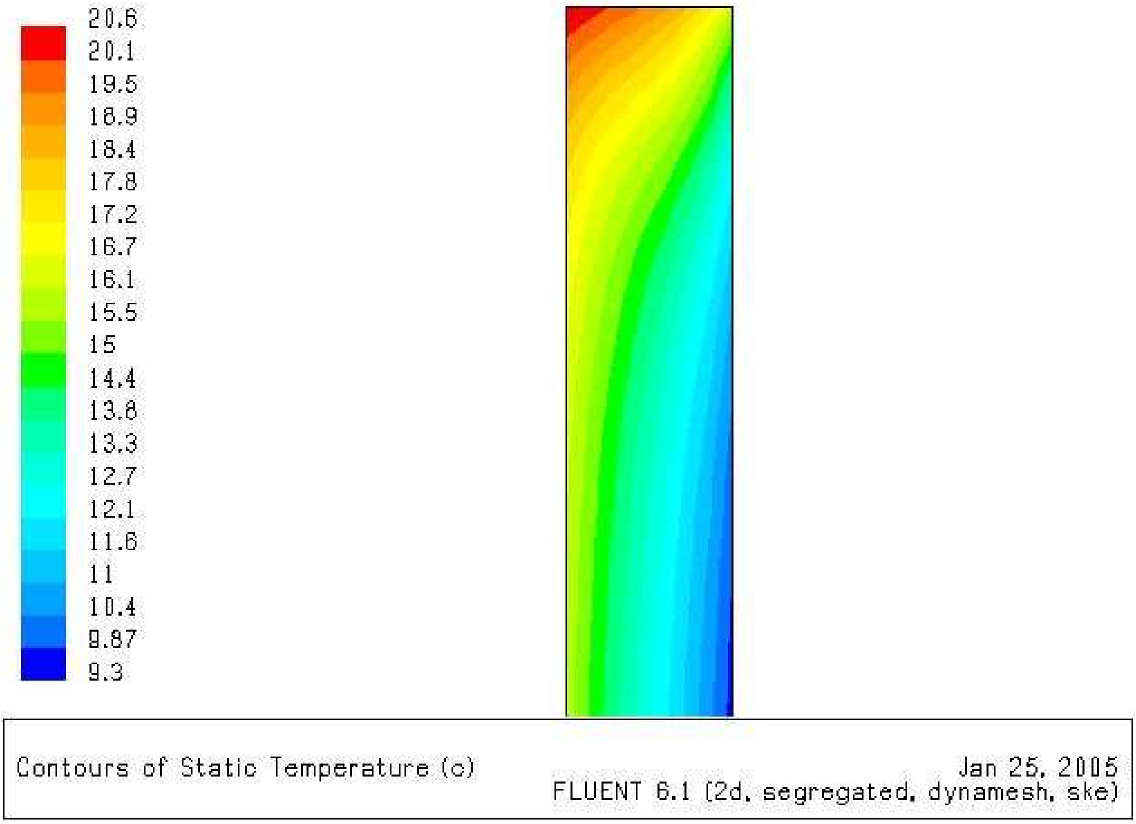


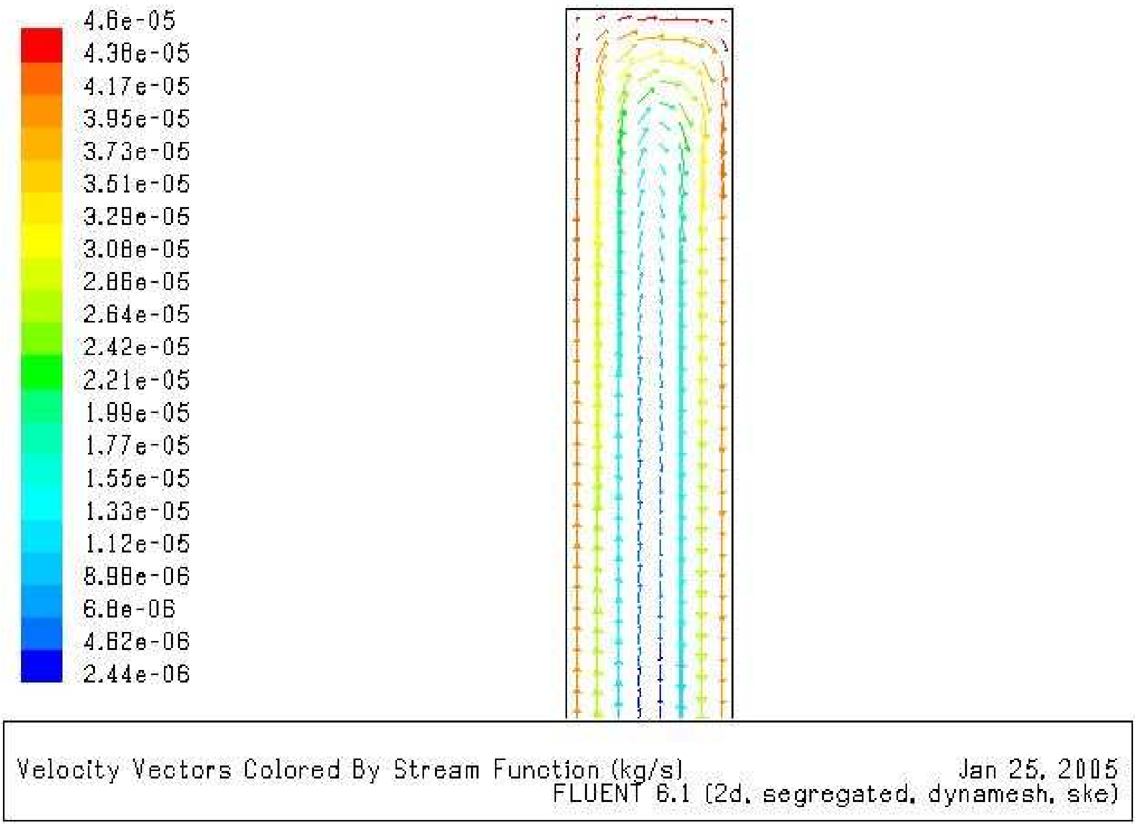
Contours of Static Temperature (c) Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)



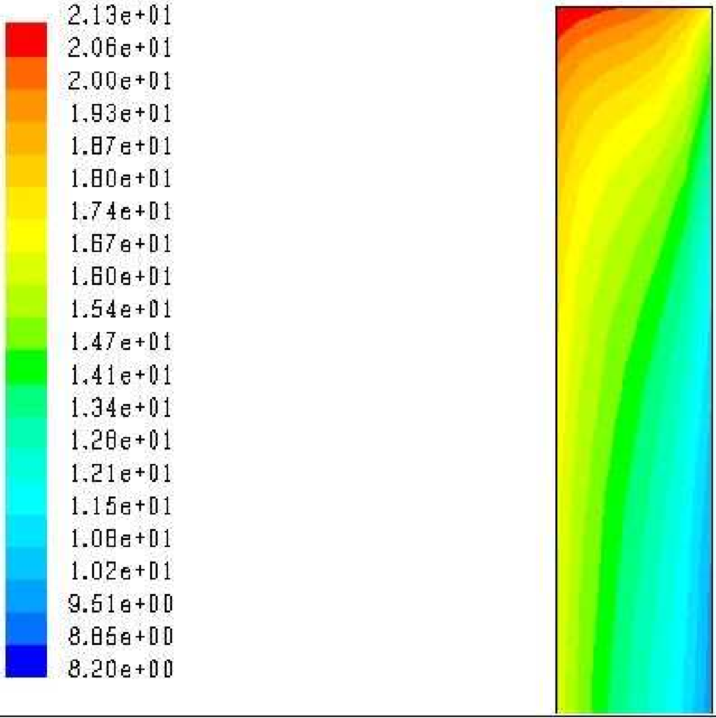
Velocity Vectors Colored By Stream Function (kg/s) Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

Analysis 6 – 10mm IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	1.25mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	Aluminium	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s^2	
Temperature Distribution & Fluxes		
Heat Loss	27.92	W

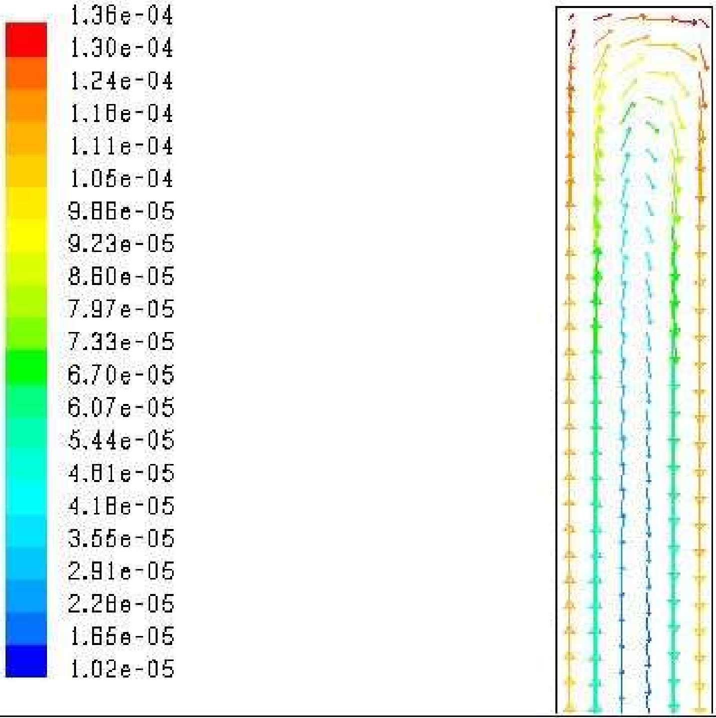

20.6
20.1
19.5
18.9
18.4
17.8
17.2
16.7
16.1
15.5
15
14.4
13.8
13.3
12.7
12.1
11.6
11
10.4
9.87
9.3
Contours of Static Temperature (c)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)


4.6e-05
4.38e-05
4.17e-05
3.95e-05
3.73e-05
3.51e-05
3.29e-05
3.08e-05
2.86e-05
2.64e-05
2.42e-05
2.21e-05
1.99e-05
1.77e-05
1.55e-05
1.33e-05
1.12e-05
8.98e-06
6.8e-06
4.62e-06
2.44e-06
Velocity Vectors Colored By Stream Function (kg/s)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

Analysis 7 – 15mm IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	2.5mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	Aluminium	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s²	
Temperature Distribution & Fluxes		
Heat Loss	26.92	W

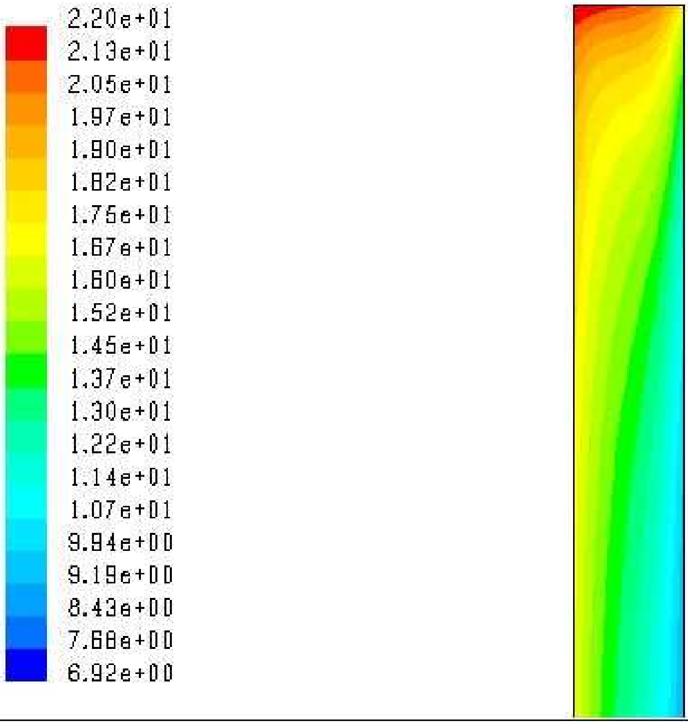


Contours of Static Temperature (c)   Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

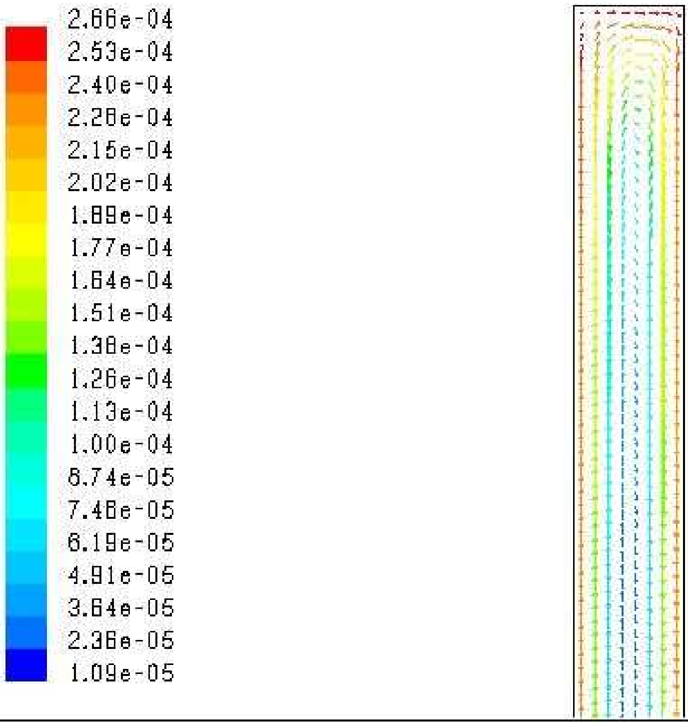


Velocity Vectors Colored By Stream Function (kg/s)   Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

Analysis 8 – 20mm IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	2.5mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	Aluminium	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s²	
Temperature Distribution & Fluxes		
Heat Loss	26.11	W

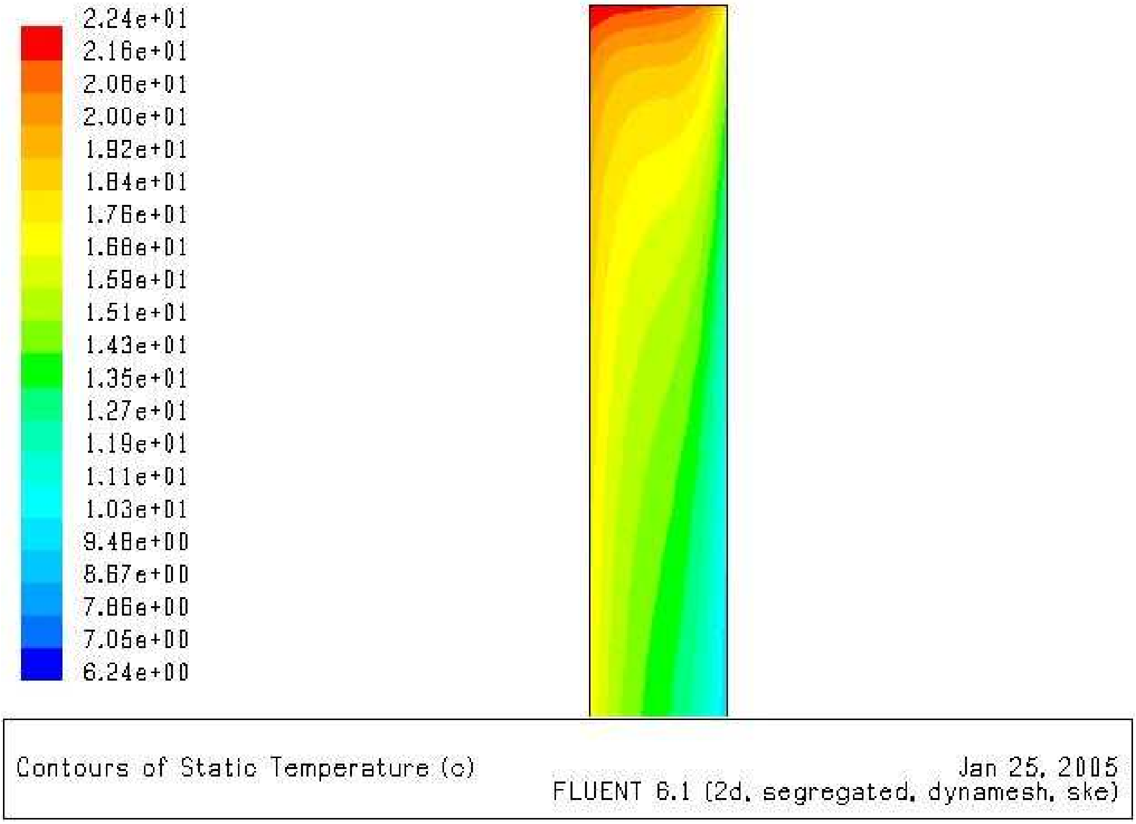


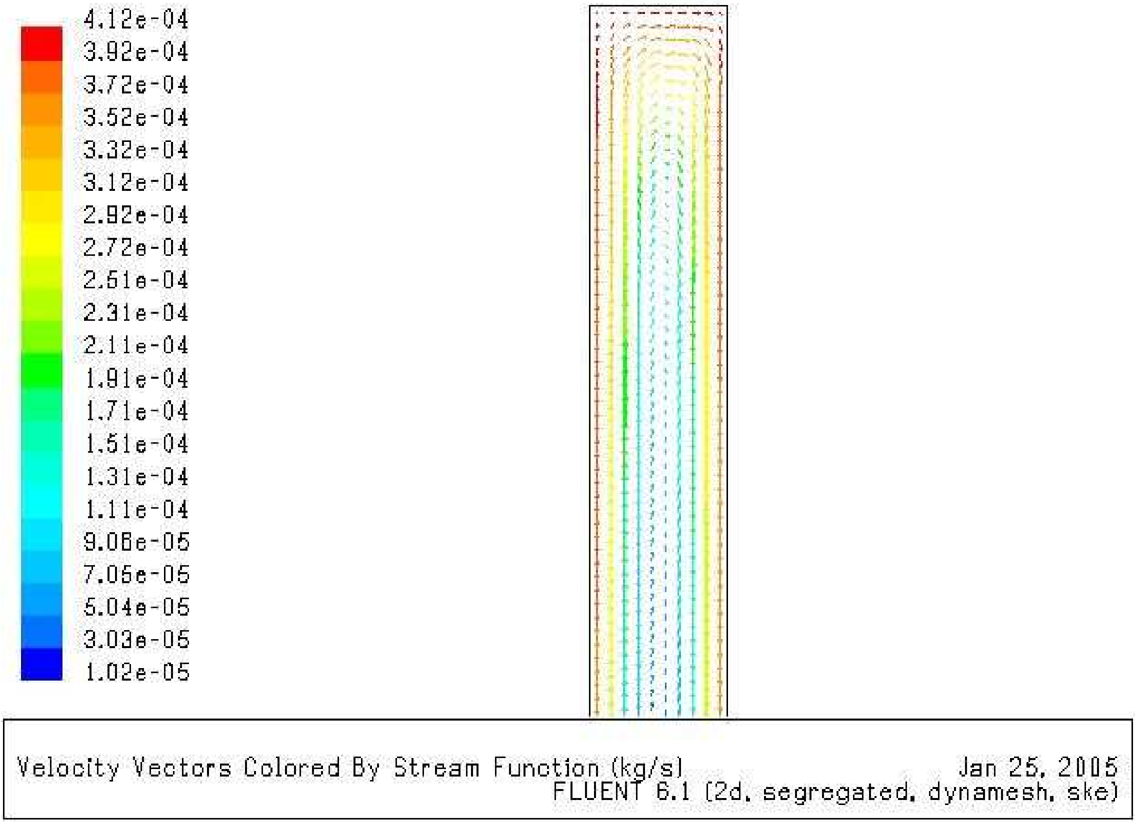
Contours of Static Temperature (c) Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)



Velocity Vectors Colored By Stream Function (kg/s) Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

Analysis 9 – 25mm IG Unit		
Model		
Solver	Segregated	
Formulation	Implicit	
Time	Steady	
Models	Energy	
	Viscous	
Element Type	Mapped Quad	
Spacing Ratio	1	
Mesh Spacing	2.5mm	
Solver Properties		
Normalisation	Scale	
Convergence Criteria	0.001	
Boundary Conditions		
Spacer	Aluminium	
Inside Pane	Glass @ 0℃	
Outside Pane	Glass @ 25℃	
Operating Conditions		
Operating Pressure	101 325 Pa	
Reference Location	(0,0)	
Gravitational Acceleration	-9.81 m/s^2	
Temperature Distribution & Fluxes		
Heat Loss	25.73	W


2.24e+01
2.16e+01
2.08e+01
2.00e+01
1.92e+01
1.84e+01
1.76e+01
1.68e+01
1.59e+01
1.51e+01
1.43e+01
1.35e+01
1.27e+01
1.19e+01
1.11e+01
1.03e+01
9.48e+00
8.67e+00
7.86e+00
7.05e+00
6.24e+00
Contours of Static Temperature (c)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)


4.12e-04
3.92e-04
3.72e-04
3.52e-04
3.32e-04
3.12e-04
2.92e-04
2.72e-04
2.51e-04
2.31e-04
2.11e-04
1.91e-04
1.71e-04
1.51e-04
1.31e-04
1.11e-04
9.08e-05
7.05e-05
5.04e-05
3.03e-05
1.02e-05
Velocity Vectors Colored By Stream Function (kg/s)
Jan 25, 2005
FLUENT 6.1 (2d, segregated, dynamesh, ske)

Appendix D

Project Resources Schedule

RESOURCE	SUPPLIER	APPLICATION	DATE REQD	COST	ACTION TAKEN
Refrigeration knowledge	Orford Refrigeration, Toowoomba	Access to factory, refrigerator background & expert knowledge	Complete	-	Site access granted to factory, allocated industry contact from Orford's
Literature Reviews	UQ Engineering Library and USQ Library	Journal articles, textbooks & relevant literature	Complete	$50	Obtained all relevant literature pieces
GAMBIT & FLUENT 5/6 CFD software access	Engineering Cat Lab, University of Southern Queensland	Numerical Investigation & Analysis	Complete	-	Obtained login, completed analysis
Compiling Dissertation	Own	Labour - word processing dissertation, formatting & editing	28 Oct 2004	$2000	In progress
Binding Dissertation	TBA	-	28 Oct 2004	$50	In progress

Appendix E

Project Timeline

Task	Completed	March				April				May				June				July				August				September				October			
		8	15	22	29	5	12	19	26	3	10	17	24	7	14	21	28	5	12	19	26	9	16	23	30	6	13	20	27	4	11	18	28
Begin initial research	✓																																
Make initial contacts with Orford Refrigeration	✓																																
Project Specification	✓																																
Start to become familiar with the software GAMBIT & FLUENT	✓																																
Begin preliminary software simulation	✓																																
Literature Review	✓																																
Safety Analysis	✓																																
Project Appreciation	✓																																
Conduct 1D Analytical Investigation	✓																																
Execute 2D Numerical Investigation with FLUENT	✓																																
Validate simulation	✓																																
Dissertation write up	✓																																
Complete Dissertation	✓																																♥
As time permits:																																	
Create efficient display door design	☒																																
Execute 3D Numerical Investigation with FLUENT	☒																																

Note: The given timeline is a guide only. Flexibility was needed in the time constraints to allow for study and assessment in other courses undertaken at the same time.

Appendix F

Risk Assessment

There are minimal risks associated with carrying out the simulation in this project; however, completing the write up for this dissertation comes with some risk. The major area for risk is when we conduct sight visits to Orford Refrigeration. Before the commencement of cooperation with Orford's we attended an OH & S site induction that highlighted a number of the major hazards in the workplace and how to minimise the risk. This evaluation will identify these risks and specify the procedures to be put in place in order to minimise the risk throughout the duration of this project.

Risk Management Plan for Factory floor

Plan 1

Description of Hazards

- Forklifts carrying loads in the walkways

Parts of body at risk

- All

Risk Level

- Moderate

Controls

- Forklifts are fitted with flashing lights and beepers
- Always look before stepping out into a walkway for potential hazards

Plan 2

Description of Hazards

- Foreign matter coming in contact with the eye

Parts of body at risk

- Eye

Risk Level

- Moderate

Controls

- All personnel and visitors must wear safety glasses at all times whilst on the factory floor
- Eye wash facilities are provided

Plan 3

Description of Hazards

- Objects falling from a height onto the feet

Parts of body at risk

- Feet

Risk Level

- Moderate

Controls

- All personnel and visitors must wear steel capped boots at all times whilst on the factory floor

Plan 4

Description of Hazards

- A fire breaks out in the factory

Parts of body at risk

- All

Risk Level

- Moderate

Controls

- All personnel must swipe in and out of the factory floor via the electronic security card, a personnel member at all times must accompany visitors
- Escape plans are clearly posted in key locations for all to see

Plan 5

Description of Hazards

- Exposure to excessive noise

Parts of body at risk

- Ears

Risk Level

- Moderate

Controls

- Earplugs are available at PPE stations throughout the factory.

Plan 6

Description of Hazards

- Strain caused from sitting in front of a computer for extended periods of time

Parts of body at risk

- Back
- Neck
- Eyes
- Fingers
- Wrists

Risk Level

- Moderate to Low

Controls

- Ensure correct posture is maintained at all times
- Limit exposure time to between 3 to 4 hours per computing session
- When feeling tired or posture begins to become poor take a walk around and stretch, grabbing a drink of water