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This thesis has been
accepted for the award
of the degree in the
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HEAT TRANSFER PROCESSES AND STRESSES IN VACUUM GLAZING

by

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Abstract

Vacuum glazing consists of two sheets of glass separated by an evacuated space which is made possible by an array of support pillars and a seal of fused solder glass around the edges of the sheets. Due to the vacuum between the sheets, there is a negligible interpane heat transfer by internal gas conduction and convection. The main heat flows across the glazing are therefore: the radiative heat transfer between the internal glass surfaces, the conduction through the support pillars, and the conduction along the glass sheets and through the edge seal (the edge conduction). Each of these heat flows is studied separately by modelling and experimental methods. An exact method for calculating the interpane radiative heat transfer for both uncoated and doped tin oxide coated glass surfaces is presented; the calculations for doped tin oxide coated glass surfaces are based on optical properties determined by a novel approach. A simple analytic model and a more exact finite element model for analyzing the pillar conduction are presented and validated by experimental results. Finite difference and finite element modelling approaches are used to study the edge conduction, and it is found that the theoretical results are in agreement with data obtained from infrared thermographic measurements of a vacuum glazing. Methods for reducing the edge conduction are investigated.

Two types of stresses in vacuum glazing are considered: the stresses due to the presence of a temperature difference across the glazing, and the stresses due to atmospheric pressure. The first type of stress is related to the heat transfer processes across the glazing and thus the calculation of these stresses is based on the understanding of these processes. These temperature induced stresses are calculated with a finite element model which is validated on the basis of the measurements of the surface strains and deflections of a glazing which is exposed to well defined environmental conditions. The atmospheric pressure induced stresses are calculated with a finite element model which simulates an appropriate portion of an entire pillar array. Results from this model are also compared with experimental results.

The temperature and pressure induced stresses are then combined for a calculation of the total probability of glazing failure. This calculation is done on an elemental basis with a modified glass failure prediction model. The probabilities of failure are calculated for different pillar separations, and for different amounts of edge insulation on the glazing (the insulation influences the probability of failure via its effect on the temperature induced stresses). The heat transfer, stress and probability of failure results are brought together for a study of the design of vacuum glazing. Design constraints are specified, and the factors which influence these constraints are considered. This design study shows how the theoretical concepts of this thesis can be applied to the practical design of commercially acceptable vacuum glazing products.

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I would like to thank the following technicians for their crucial help with the design and construction of the thermal conductance apparatus and the deflection and strain measurement apparatus: Hector Haldane, Graham Mannes and Ken Weigert. All of the vacuum glazings tested for this thesis were manufactured by Jack Tang. I would also like to thank Ruth Lambert and Christopher Dey for their generous help with the final preparation of this thesis.

The accurate analysis of the interpane radiative heat transfer would not have been possible without the novel method of calculation developed by Qi-Chu Zhang. Equally crucial to this analysis are the measurements made by Dr. Zhang and Christopher Dey and by Martin Brunotte and Andreas Gombert (both of the Fraunhofer Institute for Solar Energy Systems). The results from the infrared thermographic work performed by Fred Beck, Brent Griffith, Daniel Türler and Dariush Arasteh of Lawrence Berkeley National Laboratory provided an experimental validation of the finite element model for edge conduction. Since this model serves as a basis for most of the results of Chapters 5 to 9, the thermographic results for this validation are some of the most important data presented in this thesis.

The collaboration with the researchers at the Fraunhofer Institute and Lawrence Berkeley National Laboratory came about from my involvement with the International Energy Agency Task 18 (Advanced Glazing Materials). Several participants of this Task were an invaluable resource of information for my research.

Most importantly, I would like to thank my parents, Michael and Anna Simko, for their support of my education. I have benefited greatly from their generosity and from the importance that they placed on education. Thus, a good deal of the credit for this thesis is due to them.

Authour's Contributions

Except where otherwise noted, all of the work reported in this thesis is the original work of the Authour.

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1. Introduction

1.1 The Development of Vacuum Glazing

The use of highly insulating glazings is very important in the design of energy efficient buildings. This is because glazings are generally less insulating than the surrounding building structures and thus they can represent a pathway for significant heat exchange between the inside and outside environments. For example, the percentage of the total national energy consumption that is required to offset the heat loss through poorly insulating windows (i.e. glazings and the supporting frames) is approximately 3 % in the United States [1], and 7 % in Sweden [2].

Various methods have been developed to improve the thermal performance of glazings. The simplest method is to use two or more panes of glass separated by air filled cavities which present insulating layers. In some multiple layer glazings, suspended transparent thin films are used instead of internal glass panes in order to reduce the weight of the structure. The glass-to-glass heat flows across a multi-pane structure are: radiative heat transfer between the internal glass surfaces, conductive and convective heat transfer within the internal cavities, and conductive heat transfer along the glass sheets and through the edge seal (edge conduction). The radiative heat transfer can be greatly reduced with low emittance coatings on the internal glass surfaces. The internal conductive heat transfer can be reduced in two ways: by filling the cavities with low thermal conductivity gases (e.g. argon and/or krypton), and by widening the cavities. However, widening the cavities can also increase the internal convection. The edge conduction is commonly reduced with well designed edge seals and frames which usually incorporate thermal breaks and materials with low thermal conductivities. Obviously, the internal conduction and convection can be virtually eliminated if there is no air or gas between the glass panes. This is the key thermally insulating feature of vacuum glazing. Since it is also the principle behind the operation of the Dewar flask [3] (which is more commonly known by the modern trade name of "thermos flask"), it is not surprising that the concept of vacuum glazing was first proposed only twenty years after the announcement of Dewar's invention. The proposal was in the form of a patent application by Zoller, in 1913 [4].

Zoller's patent, which was granted in 1924, noted that vacuum glazing would require a multiplicity of "ribs, grooves, prisms or projections" in order to maintain the separation of the glass sheets under the influence of atmospheric pressure [4]. Since 1913, many different designs of vacuum glazing have been proposed, and most of them have been

recorded in the patent literature. Until recently, none of these proposals led to a report in the refereed scientific literature of the successful manufacture of a highly insulating flat vacuum glazing. An extensive review of these early designs is reported by Fischer-Cripps [5]. A few of the designs are worth noting in the present work because they highlight several challenges associated with the manufacture of vacuum glazing, and because they give some insight into the origins of the current solutions to these challenges.

Several of the early proposals dealt with the issue raised by Zoller concerning the requirement for a method of maintaining the interpane separation. For example, in 1921, Kirilin suggested that the glass sheets be separated by an array of projections [6] and, in 1976, Chalons proposed the use of "marbles" to separate the glass sheets [7]. The need to seal the edges of the glazing was addressed by Whattan and Myers who, in 1947, proposed forming a seal around the edges by fusing the glass sheets together with rods which have a lower melting point than the glass [8]. These rods would be located in the gap between the glass sheets and the fusing process would be performed with the entire glazing in a vacuum. In 1987, Benson and Tracy described a method of forming edge seals in a vacuum with a laser welding process [9]. However, they were unable to overcome certain practical difficulties associated with this procedure. An alternative to the manufacture of the glazing under vacuum involves first forming the edge seal and then evacuating the internal space between the glass sheets. This can be done via a pump out tube, such as that proposed by Falbel in 1976 [10].

The first report in the refereed scientific literature of the successful production of a flat vacuum glazing with a high degree of thermal insulation was made in 1989 by Robinson and Collins of the University of Sydney [11]. The University of Sydney design, which is dealt with in this thesis, is described below.

1.2 The Design and Manufacture of Vacuum Glazing

Figure 1.1 shows a schematic diagram of the vacuum glazing developed at the University of Sydney. Two sheets of soda-lime glass are separated by an array of support pillars, and they are sealed together around their edges by fused solder glass. In order to reduce the glass-to-glass radiative heat transfer, there are usually low emittance coatings on the internal surfaces of one or both sheets of glass. The edge seal is formed under atmospheric pressure and the interpane space is later evacuated

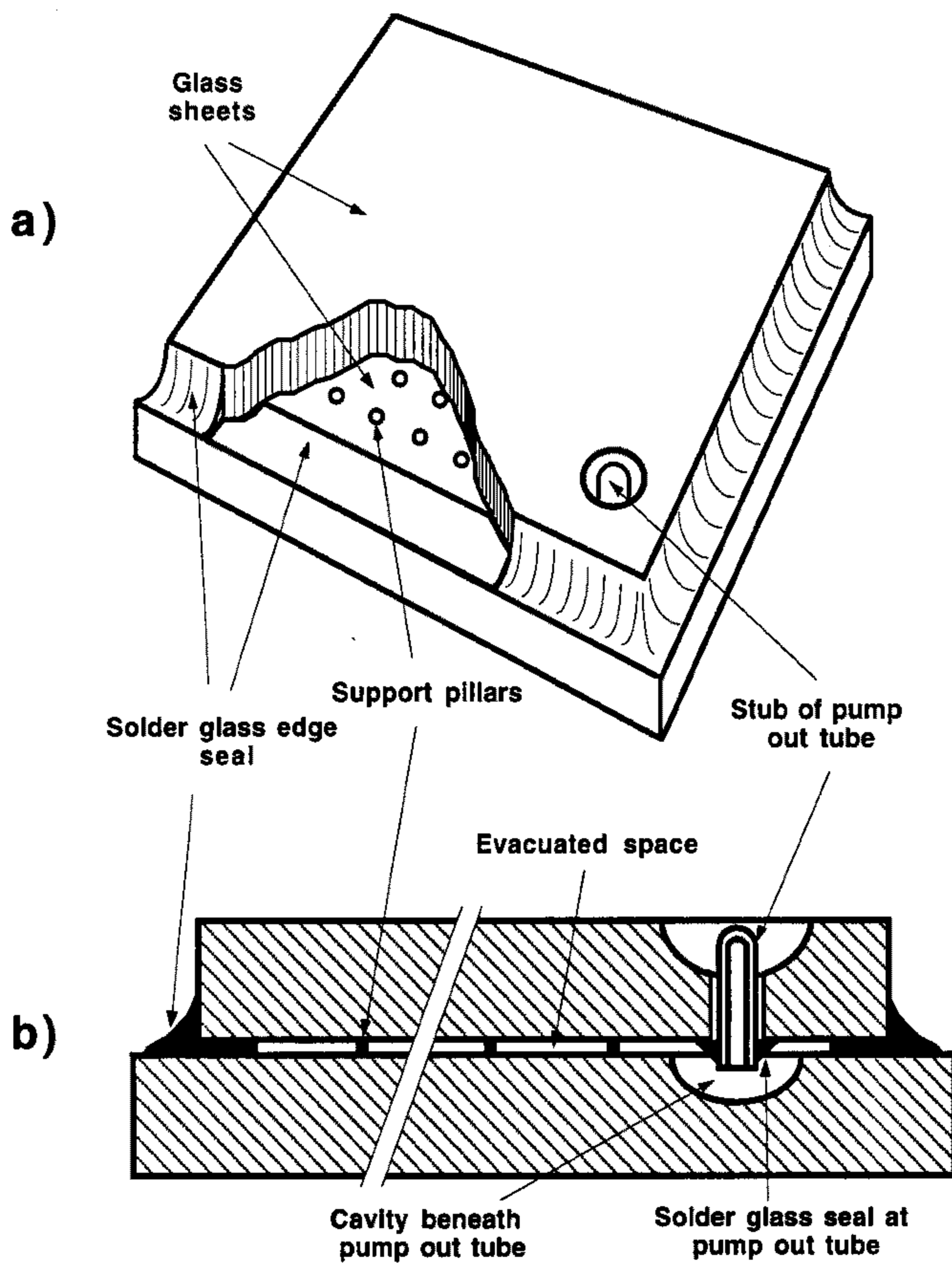


Fig. 1.1: Schematic diagram of the vacuum glazing currently manufactured at the University of Sydney (from Collins et al. [12]): (a) perspective view and, (b) cross-sectional view.

via a pump out tube which is sealed to one of the glass sheets with solder glass. The laboratory manufacturing process described below for this design of vacuum glazing employs techniques which involve a large amount of labour, and require a high level of technical skill and patience. However, wherever possible, the techniques that are used have been developed with the intention that they could be readily adapted to a large scale and automated commercial manufacturing process. A detailed description of the current manufacturing methods is given by Collins et al. [12], and ways in which these methods could be adapted to commercial manufacture (and the estimated cost of that manufacture) are discussed by Garrison and Collins [13].

The first step in the manufacturing process is to machine two hemispherical cavities and one hole in the glass sheets (shown in Fig. 1.1) in order to permit the incorporation of the pump out tube. The sheets are then cleaned and dried. The pump out tube is placed in the hole which is in one of one of the sheets, and a regular square array of small metal or ceramic support pillars of uniform height is positioned on the surface of the other sheet. At present, the pillars are placed by hand. For a large volume manufacturing process, this step must be automated. Solder glass in a water slurry is applied around the pump out tube on the unmachined surface of the sheet in which it rests. This sheet is then placed on top of the pillar array which has been laid out on the other sheet of glass. Thus, the two sheets of glass are separated by the support pillars. A fillet of solder glass is applied around the periphery of the structure on the L-shaped ledge that is formed by virtue of the upper sheet being slightly smaller in size than the lower sheet. The entire assembly is then heated to a temperature of around 500°C, at which point the solder glass melts, wets the surfaces of the glass sheets, and flows by capillary action over a short distance into the space between sheets, thereby forming an hermetic seal around the edges of the glazing. A seal is simultaneously formed around the pump out tube with the solder glass at this location.

After the formation of the seals around the edges and the pump out tube, the assembly is cooled to room temperature in order to solidify the solder glass. A connection is made to a vacuum system via an O-ring seal at the external glass surface around the pump out tube, and then the internal space is evacuated to below 0.1 Pa. During the evacuation procedure, the assembly is heated to between 100 and 250 °C in order to remove most of the small amount of gas that is present at and just below the internal surfaces of the glass sheets.

After the evacuation process, the assembly is cooled to room temperature and the external end of the pump out tube is melted and fused shut. This melting process is

accomplished by positioning a heated tungsten coil close to the end of the pump out tube within the evacuated external hemispherical cavity surrounding the tube. The connection to the vacuum system is removed and a small thin glass slide is glued to the glazing to cover the external cavity and protect the stub of the pump out tube from damage.

Over 600 samples of vacuum glazing with dimensions of up to 1 m by 1 m have been produced at the University of Sydney using constructions which are the same as, or similar to, that which is shown in Fig. 1.1. Glass sheets ranging in thickness from 2.6 to 6 mm have been used. The edge sealing process is currently quite slow; typically the temperature of the sample is increased at 200 °C per hour, and it is held at a constant value for about an hour. The sample is then allowed to cool over several hours. The evacuation of samples typically takes 2 to 4 hours, with the time being limited by the rate at which the sample is heated to the outgassing temperature and subsequently cooled [14].

Many materials have been used for the support pillars, including solder glass, metals and ceramics. So far, however, only two materials - alumina, and Inconel 718 - have been found which can be manufactured to acceptable tolerances, and which are sufficiently strong for use in highly insulating designs of the pillar array. Alumina (Al_2O_3) is a ceramic, and Inconel 718 is a nickel based alloy. Arrays are routinely made with pillars as small as 0.25 mm in diameter, 0.15 to 0.20 mm in height, and separated by a regular distance of 25 mm. These pillars are very difficult to see with the unaided eye. The optical clarity of a vacuum glazing with such pillars is therefore virtually identical to that of a double glazing made from the same type of glass.

Most of the highly insulating vacuum glazings made so far incorporate glass coated with pyrolitically deposited doped tin oxide. This is the only economical transparent low emittance coating that has been found to withstand the high temperature glazing manufacture process without significant degradation of its optical and thermal properties.

Obviously, atmospheric pressure leads to large stresses in vacuum glazing (these stresses are noted below). It might therefore seem appropriate to use high strength tempered glass in vacuum glazing. However, tempered glass is unsuitable for this application because the high temperature required to melt the solder glass for the formation of the glazing edge seal causes a relaxation of the internal and surface stresses that are responsible for the strength of the glass. Furthermore, as a result of

the glass manufacturing process, tempered glass is not very flat. Since the variation in the glass surface profile can be greater than the width of the evacuated space, tempered glass sheets can come in contact with each other across this space, thereby degrading the thermal performance of the glazing.

1.3 Prior Work

The success of the above design of vacuum glazing can be evaluated on the basis of the thermal performance of the glazing and on the total probability of glazing failure associated with the stresses in the structure. An analysis of the thermal performance must involve investigations into the following three glass-to-glass heat flows across the glazing: the radiative heat transfer between the internal glass surfaces, the conductive heat transfer through the support pillars and the edge conduction. In order to perform a probability of failure analysis, it is first necessary to determine the stresses in the glazing from the two main sources of stress: the presence of a temperature difference across the glazing, and atmospheric pressure.

Previous work on the radiative heat transfer between the internal glass surfaces of vacuum glazing has been both experimental and theoretical in nature. Radiative heat flows which were measured with a guarded hot plate apparatus were found to be a few percent greater than those which were predicted according to a simple and commonly used equation [15]. After accounting for the possible errors associated with the measurement technique, it was concluded that the equation must be in error [15].

There is a very subtle issue related to the glass-to-glass radiative heat transfer that should be noted. Enhanced radiative heat transfer can occur between two very closely spaced surfaces via the interference of multiply reflected thermal radiation waves, and via the energy transfer that occurs if the evanescent electromagnetic field from each surface bridges the gap and reaches the other surface [16]. However, these effects only occur if the surfaces are separated by a gap which is of the order of, or smaller than, the dominant thermal radiation wavelengths (i.e. for separations of about a few micrometers) [16]. Since the internal surfaces of vacuum glazing are separated by distances of about 0.15 to 0.20 mm (i.e. two orders of magnitude greater than the dominant thermal wavelengths), these effects need not be considered in the thermal analysis of vacuum glazing.

The heat conduction through the support pillars results from the fact that the pillars present localized thermal short circuits between the glass sheets which are otherwise, except at the edges, thermally well isolated. An analytic model for calculating the thermal resistance of a single pillar has been developed by Collins et al. [17]. This model has been theoretically validated by results from a finite element model. However, the accuracy of the finite element model is uncertain. The experimental results of Collins et al. apparently confirmed the analytic model [17], but subsequent concerns about the accuracy of the measurement technique have led to a need for new and more accurate experimental data with which to check the analytic expression.

The edge conduction in vacuum glazing has been *indirectly* studied by using a finite element model to predict the glass surface temperature non-uniformities which are associated with this heat transfer [18]. However, no comparable measurements were made and thus the modelling approach was not experimentally validated. The influence of edge insulation on these non-uniformities was briefly considered.

The long term thermal performance of a vacuum glazing is of course dependent on the stability of the internal vacuum. Degradation of the vacuum can occur by leaks at the edge seal or the pump out tube, or by the outgassing of the residual gases that remain just below the internal glass surfaces after evacuation. Outgassing is enhanced by exposing the glazing to high temperatures. Long term thermal measurements of glazing samples which have been stored at room temperature show that leaks are not a concern; for most samples of vacuum glazing, there is a negligible degradation of the vacuum over at least several years [12, 19]. Measurements of glazing samples which have been stored at elevated temperatures demonstrate that, although outgassing does degrade the vacuum, this is mostly a temporary effect if the temperature is returned to more realistic levels; this is possibly due to a gas re-absorption process [19]. Outgassing is the subject of a continuing study and the details of this process are not considered in this thesis.

The presence of a temperature difference across a vacuum glazing leads to a differential thermal expansion and contraction of the two glass sheets. This can cause the sheets to bend, thereby leading to stresses within the glass. Fischer-Cripps analyzed these stresses with a finite element model [18]. This model for the stresses was *indirectly* validated by measuring the surface deflections due to the temperature induced bending of the glazing. However, the conditions under which the measurements were made were not well known. It is noted that a more direct validation could be accomplished by measuring the strains associated with the stresses

being studied. The effect of edge insulation on the temperature induced stresses was briefly considered by Fischer-Cripps [18].

Atmospheric pressure leads to the following stresses in vacuum glazing: compressive stresses in the pillars, stresses associated with the indentation of the pillars into the internal glass surfaces, and localized tensile stresses at the external glass surfaces above the pillars and near the edge seal. The types of support pillars that are currently used in vacuum glazing are of sufficient strength to withstand the compressive stresses within the pillars without experiencing significant deformation. Fischer-Cripps conducted a finite element modelling study of the stresses at the internal and external glass surfaces around a single pillar [18]. Fischer-Cripps also studied the atmospheric pressure induced stresses near the edge seal, but only in combination with an approximate simulation of the effects of the nearby pillars [18].

Results from the previous work on the stresses in vacuum glazing were used in a study of the total probability of glazing failure [18]. This study was based on a glass failure prediction model which was modified to account for the probability of delayed failure from long term stresses [20]. Modelling results for the stresses due to a temperature difference across the glazing and the stresses due to atmospheric pressure were combined for this study. However, since the non-uniform and biaxial nature of these stresses was not properly accounted for, this failure study provided only gross estimates of the actual probabilities of failure. Results from the above thermal, structural and probability of failure studies have been combined to present a set of approximate guidelines for the design of vacuum glazing [18].

1.4 Rationale for, and Overview of Thesis

The purpose of the present work is to develop a thorough understanding of the heat transfer processes and the stresses in vacuum glazing. The study of how these issues are affected by certain parameters ultimately leads to the development of a set of constraints which can be used to design practical and commercially acceptable vacuum glazing products. Some of the studies in this thesis build upon some of the previous work mentioned above. For example, a more accurate heat flow measurement apparatus is designed and constructed, previously developed modelling techniques are applied to create better models and some of these techniques are experimentally validated either for the first time or in a more direct manner. This results in more

comprehensive and exact experimental and modelling methods which, together with the other work in this thesis, are used to provide new understandings of the thermal and stress issues in vacuum glazing

Several experimental and theoretical methods are developed for the accurate quantification of the three glass-to-glass heat flows mentioned above: internal radiative heat transfer, pillar conduction, and edge conduction. Most of the measurements are made with the guarded hot plate apparatus which is described in Chapter 2. This apparatus is based on the design of the apparatus referred to above that was used in the previous work on vacuum glazing [15], but it incorporates several improvements which permit more accurate measurements of the small localized heat flows in the glazing. A method for properly calibrating the apparatus is presented in this chapter.

In Chapter 3, new methods are used to calculate the radiative heat transfer between the internal glass surfaces of vacuum glazing. Both uncoated glass surfaces and glass surfaces which are coated with a low emittance film of doped tin oxide are considered. The theoretical approaches are validated by accurate measurements made with the above mentioned improved guarded hot plate apparatus.

Chapter 4 deals with the heat conduction through the support pillars. Results for the thermal resistance of a single pillar - as determined from an analytic model, a finite element model, and heat flows measured with the improved guarded hot plate apparatus - are compared for the purpose of cross validation. A proper convergence test is performed on the finite element model. Also presented in this chapter is an analysis of the suppression of the interpane radiative heat transfer around the pillars that results from the temperature non-uniformities associated with the pillar conduction.

The issue of edge conduction is addressed in Chapter 5. Three models are developed for the analysis of edge conduction and its effects: a one-dimensional analytic model, a one-dimensional finite difference model, and a three-dimensional finite element model. For these models, the glass-to-glass center-of-glass U-value for the glazing (explained below) is based on models presented in Chapters 3 and 4. In order to validate the edge conduction models, a comparison is made between theoretical and experimental results for the glass temperature non-uniformities due to edge conduction. The experimental results for this comparison are from infrared thermographic measurements of a vacuum glazing. Also in this chapter, U-values are calculated on the basis of glass temperatures determined from the one-dimensional finite difference model for edge conduction.

Some U-value results are compared with those obtained from a commonly used fenestration thermal analysis program. Different methods of reducing the edge conduction and the window U-values are investigated in this chapter.

After the above thermal issues have been covered, the focus of the thesis shifts to a study of the stresses in vacuum glazing. Only glazings with unconstrained edges are considered. In Chapter 6, the stresses due to temperature induced bending are calculated with a finite element model on the basis of temperature results from the finite element model which is used for the study of edge conduction. The stress analysis model is validated by comparisons between modelling and experimental results for both the deflections and strains due to the temperature induced bending of a vacuum glazing. The latter quantity provides a more direct indication of the stresses. The influence of edge insulation on these stresses is investigated in more detail than in the previous work.

The stresses due to atmospheric pressure are studied in Chapter 7. A finite element model is developed to simulate the external surface stresses around the pillars of an appropriate portion of a full array together with the external surface stresses near the edge seal. As will be explained, the internal surface stresses are not significant in the context of this work and thus they are not considered. The model is approximately validated by a comparison between theoretical and experimental results for the atmospheric pressure induced deflections of a glazing. The influence of the pillar separation on these stresses is studied over a wide range of realistic separations.

In Chapter 8, results for the stresses due to temperature induced bending, and the stresses due to atmospheric pressure (as obtained from the models of the previous two chapters) are combined and are used to calculate the total probability of glazing failure. This is done according to the modified glass failure prediction model referred to above in Section 1.3. Unlike the previous work, this failure analysis is done on an elemental basis so that the non-uniform and biaxial nature of the stresses can be properly accounted for. Furthermore, since the stress results are based on appropriate values for the thermal and material properties of soda-lime and solder glasses, this analysis provides more exact total probability of failure results than the previous work. The effect of edge insulation on the total probability of failure (via the temperature induced bending stresses) is also studied.

In Chapter 9, results from the thermal, stress and probability of failure analyses of the previous chapters are combined in a study of the design constraints for vacuum

glazing. This study ties together the work presented in this thesis to demonstrate how the theoretical concepts of the thesis can be applied to the practical design of actual vacuum glazing products.

Several issues related to the work presented in this thesis that would be worthwhile to investigate in future studies are noted in the concluding chapter, Chapter 10.

Most of the modelling work of this thesis is performed for the environmental conditions that are specified by the ASTM E 1423-91 standard fenestration test conditions [21]. Some U-values are calculated for the American Society of Heating Refrigerating and Air-Conditioning Engineers (ASHRAE) winter test conditions which are given in Chapter 5. The simulation of such commonly used conditions facilitates comparisons with standard results for other types of glazings. Most of the modelling is performed for a typical vacuum glazing. Such a glazing consists of two sheets of doped tin oxide coated glass, and pillars of radius, height and separation of 0.125, 0.20 and 25 mm, respectively. Only the typical glass sheet thickness of 4 mm is simulated.

All of the finite element work of this thesis is performed with STRAND6, which is a commercially available finite element analysis package [22]. For each model, all of the elements are four-node square elements, except along the boundaries of different node densities, where three-node triangular elements are used.

The thermal performance of a glazing is commonly described in terms of the U-value (the preferred ASHRAE term is the U-factor). The U-value has units of $\text{Wm}^{-2}\text{K}^{-1}$ and thus it is equivalent to the thermal conductance, or the inverse of the product of the thermal resistance and the area. In this thesis, references are made to both glass-to-glass and air-to-air U-values. The air-to-air U-value represents the series combination of the glass-to-glass U-value, U_{g-g} , and the warm and cold side ambient heat transfer coefficients, h_{warm} and h_{cold} :

$$U = \frac{1}{\frac{1}{U_{g-g}} + \frac{1}{h_{warm}} + \frac{1}{h_{cold}}} \quad (1.1)$$

The method for calculating the air-to-air U-value from modelling results is given in Chapter 5. As explained in that chapter, there are different U-values for the different regions of a window. The most commonly noted U-value in this thesis is the center-of-glass U-value because it is mostly independent of the effects related to the edge conduction. In this work, the center-of-glass U-value is usually given as a glass-to-

glass quantity because this gives a more direct reflection of the influence of the design of the pillar array on the thermal performance of the glazing.

A thorough thermal analysis of vacuum glazing should consider the effects relating to solar radiation. This is because these effects influence the amount of solar energy which passes through the glazing and into the inside ambient environment. Also, one of these effects, the effect of solar absorptance, can increase the temperature of the glass sheets and thus influence the edge conduction and the temperature induced stresses. However, solar radiation effects are not considered in this thesis because the simulation of these effects would significantly complicate the analyses, and because testing facilities were not readily available to validate experimentally the results from any such simulations. As noted in the last chapter, these effects could be considered in future studies, possibly by appropriately modifying commercially available fenestration analysis programs to simulate vacuum glazing under solar irradiation.

Stresses in vacuum glazing can also be caused by the forces due to wind. The simulation of wind loading can be very complex due to the transient and non-uniform nature of the forces. Therefore, wind loading stresses are not considered in this thesis. For the same reason, the transient stresses due to impacts or due to time-dependent illumination are not studied.

It is important to note that the main purpose of this thesis is to demonstrate the methods for calculating the heat transfer processes, stresses and probability of failure for the glazing and, in so doing, to obtain typical results for a limited range of design options. The same methods of calculation can be used, as part of any future work, to obtain a more comprehensive set of data for a wide variety of options.

Another purpose of this thesis is to show how to optimize the thermal performance of vacuum glazing. A direct comparison with the thermal performance of conventional glazings is not made throughout this work. It is known that, for a standard double glazing unit with an argon filled cavity and with a low emittance coating on one internal surface, it is possible to achieve an air-to-air center-of-glass U-value of $1.36 \text{ Wm}^{-2}\text{K}^{-1}$, and a corresponding total window U-value of $1.59 \text{ Wm}^{-2}\text{K}^{-1}$ [23]. As will be shown, lower U-values can be obtained with vacuum glazing. However, it is noted that even lower U-values can be achieved with advanced designs of multiple pane gas filled glazings (e.g. air-to-air center-of-glass U-values of 0.63 to $0.74 \text{ Wm}^{-2}\text{K}^{-1}$ for some triple pane designs [1]).

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2. Thermal Conductance Apparatus

2.1 Introduction

The total heat flow through a vacuum glazing can be measured with conventional glazing testing devices such as guarded hot boxes and calorimeters. Localized heat flow measurements such as at positions close to or far from the support pillars are also of interest and require a different approach.

The development of a sensitive, accurate and reproducible method for measuring localized heat flows through vacuum glazing presents considerable challenges. Firstly, the insulating part of the structure (the evacuated space) is located between two sheets of glass which, relative to the vacuum, have a low thermal resistance. This can give rise to large heat flows along the glass sheets unless great care is taken to reduce the lateral temperature gradients to very low levels. Secondly, away from the edges of the glazing, the measured heat flow at any location is a combination of the radiative heat transfer across the evacuated space, conduction through the support pillars, and conduction through the residual internal gas. In high accuracy measurements, it is extremely difficult to reduce the heat flow from the pillars to negligible levels at any point over the surface of the glazing unless there is a large pillar separation. However, as discussed in later chapters, there are limitations as to how far apart the pillars can be separated without incurring unacceptably large stresses in the glass sheets. Therefore, any measurement of the separate contributions from the different heat flows requires a method of accounting for the conduction through the pillars. Finally, since the absolute amount of heat which flows through a small area of a vacuum glazing is very small, the measurement apparatus must have a high resolution.

A thermal conductance apparatus (TCA) has been designed and constructed to measure localized heat flows in vacuum glazings. These heat flows are used to calculate the thermal conductance (i.e. the U-value) across glazing samples. The measurement method is based on the principle of the standard guarded thermal hot plate. The apparatus is similar to other guarded devices [1, 2], but it incorporates several adaptations in order to meet the challenges described above.

For each chapter of this thesis - including this one - the symbols used in the chapter are listed after the conclusion section.

2.2 Design and Construction of The TCA

2.2.1 Overview of The Design and The Principle of Operation

As shown in Fig. 2.1, the apparatus has a central metering piece which is surrounded by an isothermal guard. The thermal resistance between these two parts must be as high as possible. However, this is difficult to achieve because the metering piece and the guard must be in contact with a thin faceplate of glass which presents a low thermal resistance between the parts. This assembly is referred to as the hot plate, and one side of the vacuum glazing sample is placed in contact with this plate. The other side of the sample is placed in contact with an isothermal cold plate. The hot guard and the cold plate are kept at the desired temperatures by water which flows from temperature controlled baths and circulates through water jackets in the plates. These water jackets are sealed with O-rings. A very thin layer of gel is used to ensure that there is good thermal contact between the sample and the two plates.

Heat flows from the hot plate, across the sample, and to the cold plate. The source of the heat lost from the hot guard is the circulating water. Since the central metering piece is thermally well isolated from the guard and the water jacket, this metering piece does not receive a similar heat input. Consequently, the temperature of the metering piece, T_m , is lower than that of the guard, T_g . A measurement of $T_m - T_g$ can be used to estimate of the magnitude of the heat flow from the metering piece through the sample. This estimate can be made absolute by dissipating a known amount of thermal power in the metering piece via a resistor in order to increase T_m . When $T_m = T_g$, no heat flows between the metering piece and the guard, and all of the thermal power supplied to the metering piece flows through the sample. This is defined as the null condition and the corresponding power dissipated in the resistor is referred to as the null power.

The temperature difference between the metering piece and the guard, $T_m - T_g$, is detected with a thermistor in each part. A thermistor is an intrinsic semiconductor device which has a very large negative temperature coefficient of resistance. In order to achieve an adequate sensitivity (~ 1 mK), the thermistors are connected into a Wheatstone bridge circuit, the output of which is fed to a high stability operational amplifier. The amplified bridge voltage, V_b , is proportional to $T_m - T_g$. The bridge voltage corresponding to $T_m - T_g = 0$ (the null condition) can be measured under controlled conditions and it is referred to as the zero bridge voltage, V_{bo} . In an actual measurement, the application of power to the resistor changes T_m - and thus the bridge voltage. When enough power is applied so that $V_b = V_{bo}$, the null condition is established and this applied power corresponds to the null power. As will be shown, a linear

relationship exists between $T_m - T_g$ and power. Therefore, the null power can be easily determined by interpolation between or extrapolation from the bridge voltages at two different powers.

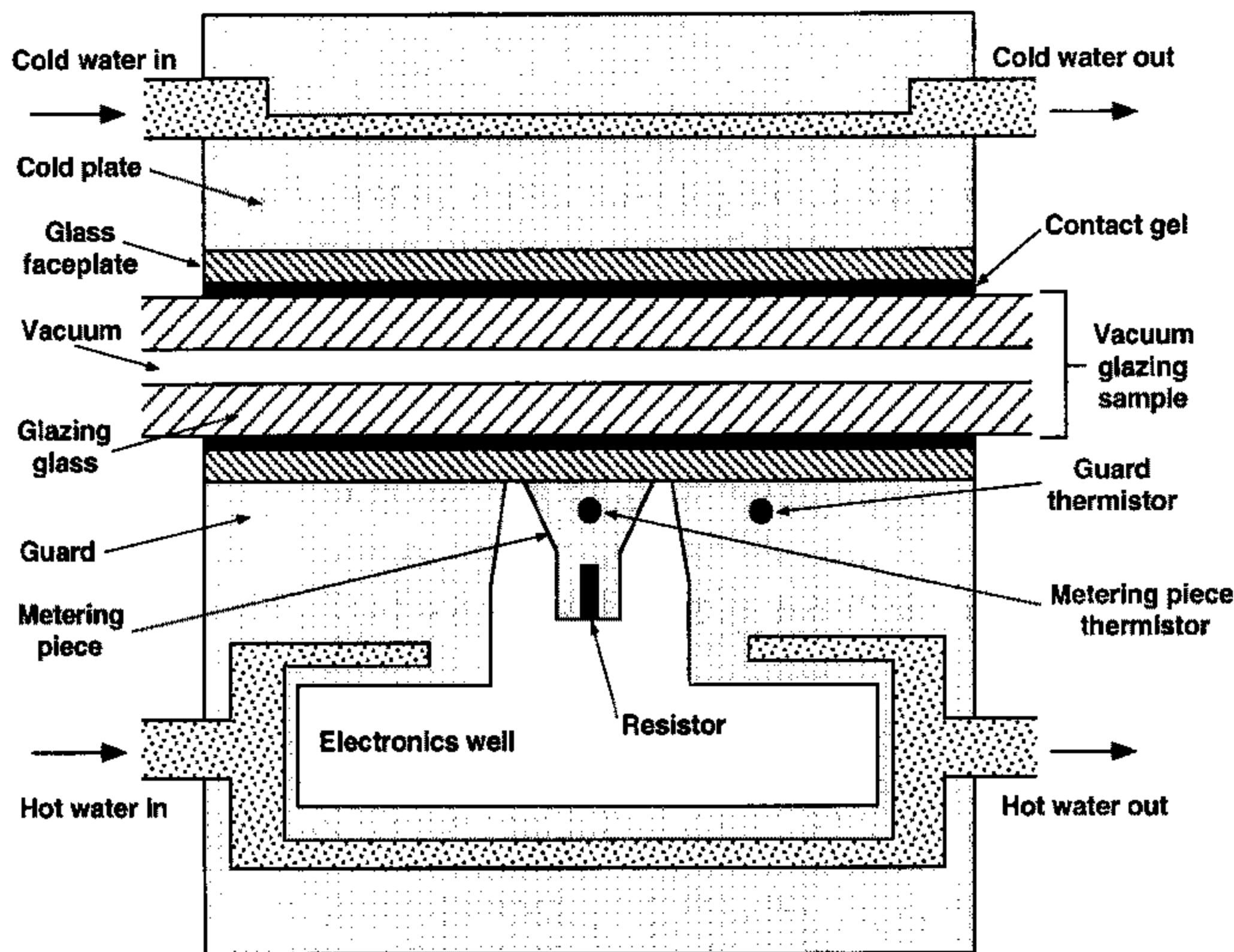


Fig. 2.1: Schematic diagram of the thermal conductance apparatus. The guard thermistor is actually placed along a radius that is perpendicular to a line between the ports for the inlet and outlet hot water flows.

Two alternatives to the use of thermistors as temperature sensors were considered: individual thermocouples, and a heat flux meter. Thermocouples would be unsuitable for the TCA because their sensitivity is about an order of magnitude less than that which is required (~ 1 mK). In the devices described by Hollands [1] and Symons and Peck [2], there is a heat flux meter between a heater plate and a surrounding plate (these plates are respectively analogous to the metering piece and the guard of the TCA). The heat flux meter is located at the base of the heater plate. Electrical power is added to the heater plate until the voltage of the heat flux meter is zero, at which point the temperature of the heater plate is approximately equal to that of the surrounding plate. Hollands notes that this arrangement for determining the temperature equality gives a better sensitivity than a thermocouple imbedded in each plate. However, a heat flux meter would be unsuitable for the present apparatus because of the very tight

geometrical constraints associated with the small area of the metering piece. Also, the many wires from a heat flux meter would significantly decrease the thermal resistance between the metering piece and the guard. It is noted that a possible disadvantage of using thermistors in the present apparatus is that the power dissipated in these devices can affect the local temperatures. However, as will be shown, this does not affect the TCA measurements.

In order to reduce thermally induced variations in the bridge voltage, the bridge voltage amplifier circuit and the bridge resistors are enclosed in an electronics well in the hot plate, where the temperature is well defined by the temperature of the hot water flow. Since the internal electronics and the thermistors are temperature sensitive, the hot plate is operated at the same temperature for all of the measurements. In order to minimize the heat transfer between the plate and ambient, and thus the effects of fluctuations in the air temperature on the guard temperature, the temperature for the hot plate is approximately at room temperature (23 °C). The ambient in the TCA measurement room is maintained at this temperature with an air conditioner system.

The terms "hot plate" and "cold plate" are used because, for most of the measurements, the former plate is indeed at a higher temperature than the latter plate. However, if one wishes to perform a measurement with the average sample temperature above room temperature, then the cold plate must be at the higher temperature. In this reversed mode of operation, heat flows across the sample *to* the metering piece and the guard. Since, for this case, the metering piece is hotter than the guard, the addition of heat to the metering piece via the resistor will not balance the bridge. However, since the linear relationship between $T_m - T_g$ and power also exists for this situation, it is possible to determine the heat flow by extrapolation from two power readings.

The nature of the samples to be measured - vacuum glazings - results in complications that differentiate this guarded hot plate apparatus from most other similar devices. The insulating part of the sample - the evacuated space - is located between two relatively highly thermally conductive glass sheets. It is therefore unavoidable that the measurement faces of the metering piece and the guard will be in good thermal contact with a glass sheet. In most guarded hot plate designs this does not occur; the metering piece and the guard are usually in contact with the insulating layer under test. The contact with the glass complicates the design of the TCA in two ways. Firstly, the glass sheet reduces the thermal resistance between the metering piece and the guard, making it necessary to detect very small temperature differences between the components in

high sensitivity measurements. Secondly, the presence of the glass sheet results in larger non-uniformities in the heat flow, and significant temperature gradients, within the measurement region. This necessitates a careful selection of the locations for the thermistors in the metering piece and the guard since these sensors must detect very small temperature differences during the normal operation of the apparatus.

2.2.2 Design of The Plates, Metering Piece and Gap

The outside diameter of the guard must be large enough to ensure that the lateral parasitic heat flow from the metering piece along the glass in contact with the hot plate is negligible. This glass includes both the faceplate affixed to the plate, and the sheet of glass on the hot side of the sample. The parasitic heat can flow along the glass to the surroundings, and to the glass on the cold side through the edge seal of the vacuum glazing being tested. Collins et al. have conducted a finite element modelling analysis of such lateral heat flow in an apparatus which is similar to the TCA [3]. The results show that, for a guard diameter of 100 mm and a total hot side glass thickness of 6 mm (2 mm for the faceplate and 4 mm for the sample), the lateral heat flow associated with the edge conduction is negligible - provided that the edge seal of the glazing is not located within the guard diameter. The modelling results also show that, for the same dimensions, the operation of the apparatus at temperatures slightly above ambient does not lead to significant lateral heat flow. Therefore, a guard diameter of 100 mm was selected for the TCA. In order to ensure that there is a fairly uniform heat flow across the sample, the diameter of the cold plate is the same as that of the hot guard.

A 2 mm thick glass faceplate is bonded with epoxy resin to the sample side of each plate. The guard, metering piece and cold plate are made of copper, and the water jackets are made of brass. Engineering drawings of the hot and cold plates are presented in Figs. 2.2 and 2.3. Photographs of the apparatus are shown in Figs. 2.4 and 2.5.

Ideally, the area of the metering piece should be as large as possible in order to maximize the sensitivity and accuracy of the apparatus. However, the area is limited by the requirement to be able to distinguish among the different heat flows in vacuum glazing - those due to radiative heat transfer, pillar conduction and gas conduction. The diameter of the metering piece is therefore constrained by the dimensions of the samples to be measured - particularly by the thickness of the glass sheets and by the regular pillar separation. A metering piece diameter of 13 mm was selected because it provides a reasonable spatial resolution for a typical total measurement side (i.e. hot

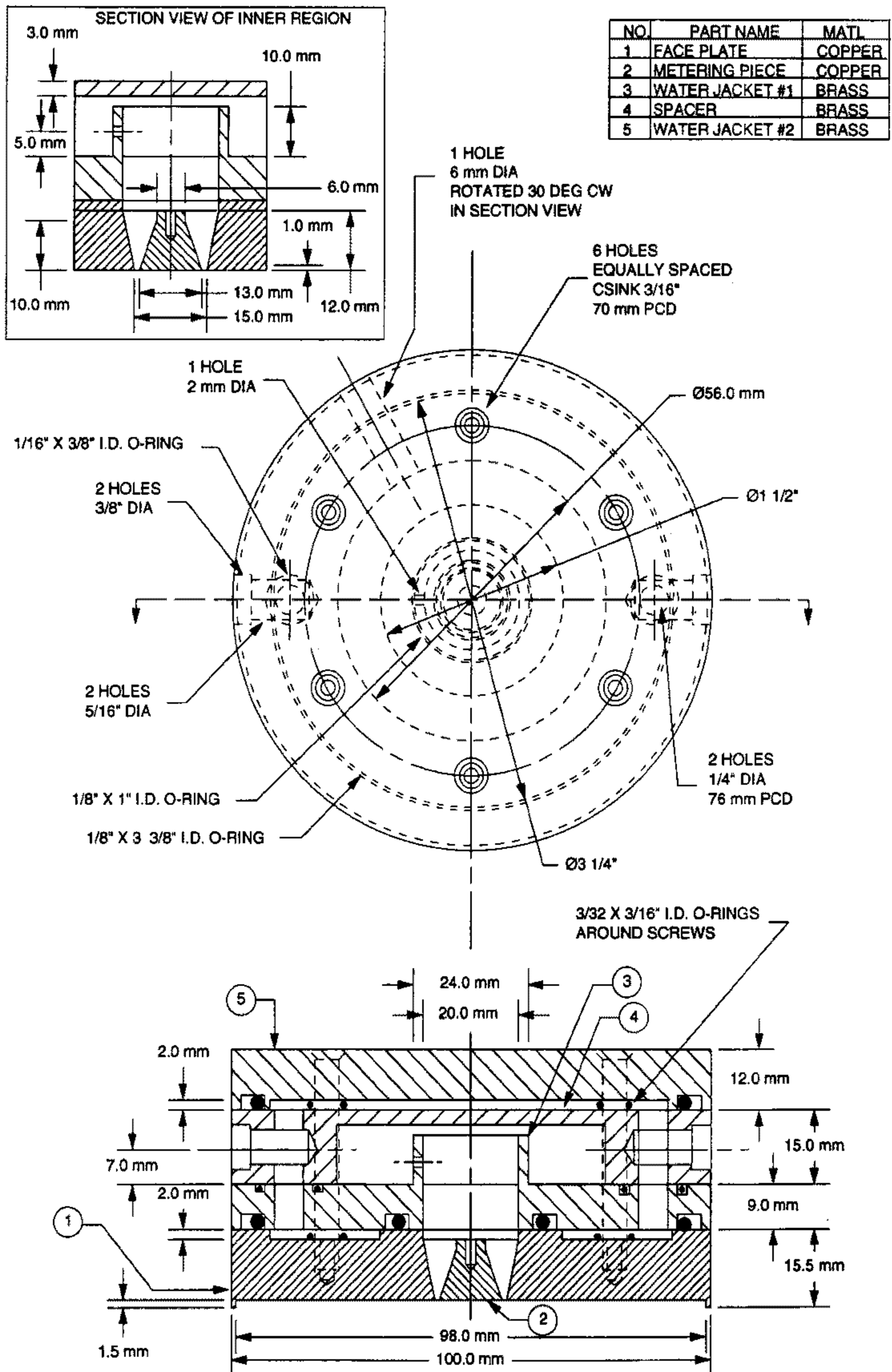


Fig. 2.2: Engineering drawing of the TCA hot plate. Holes for the thermistors are not shown. O-Rings are represented in the lower section view as black circles. O-Ring grooves are not shown in the top view. Scale 1:1.25

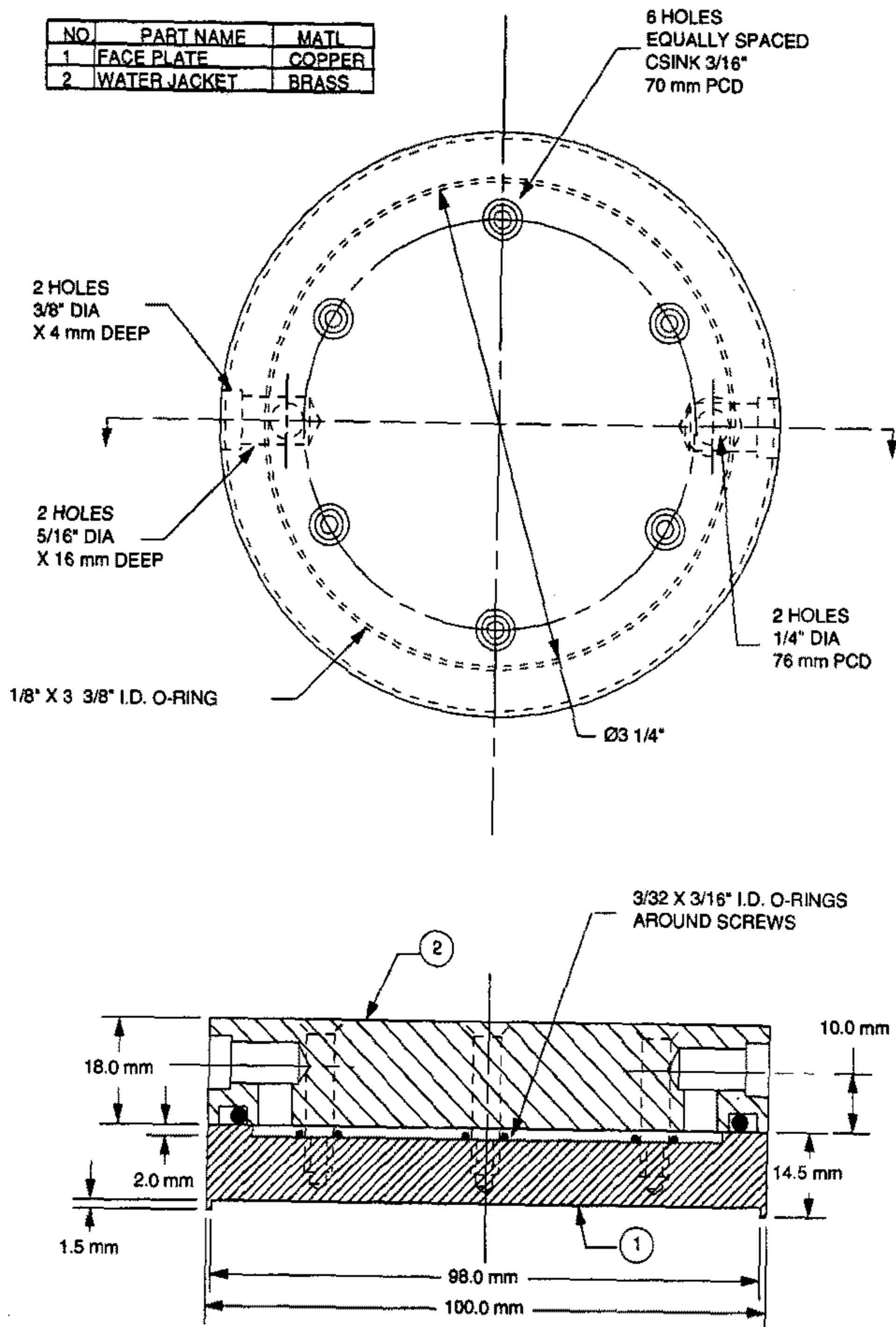


Fig. 2.3: Detailed drawing of the TCA cold plate. O-Rings are represented in the section view as black circles. O-Ring grooves are not shown in the top view. Scale 1:1.25

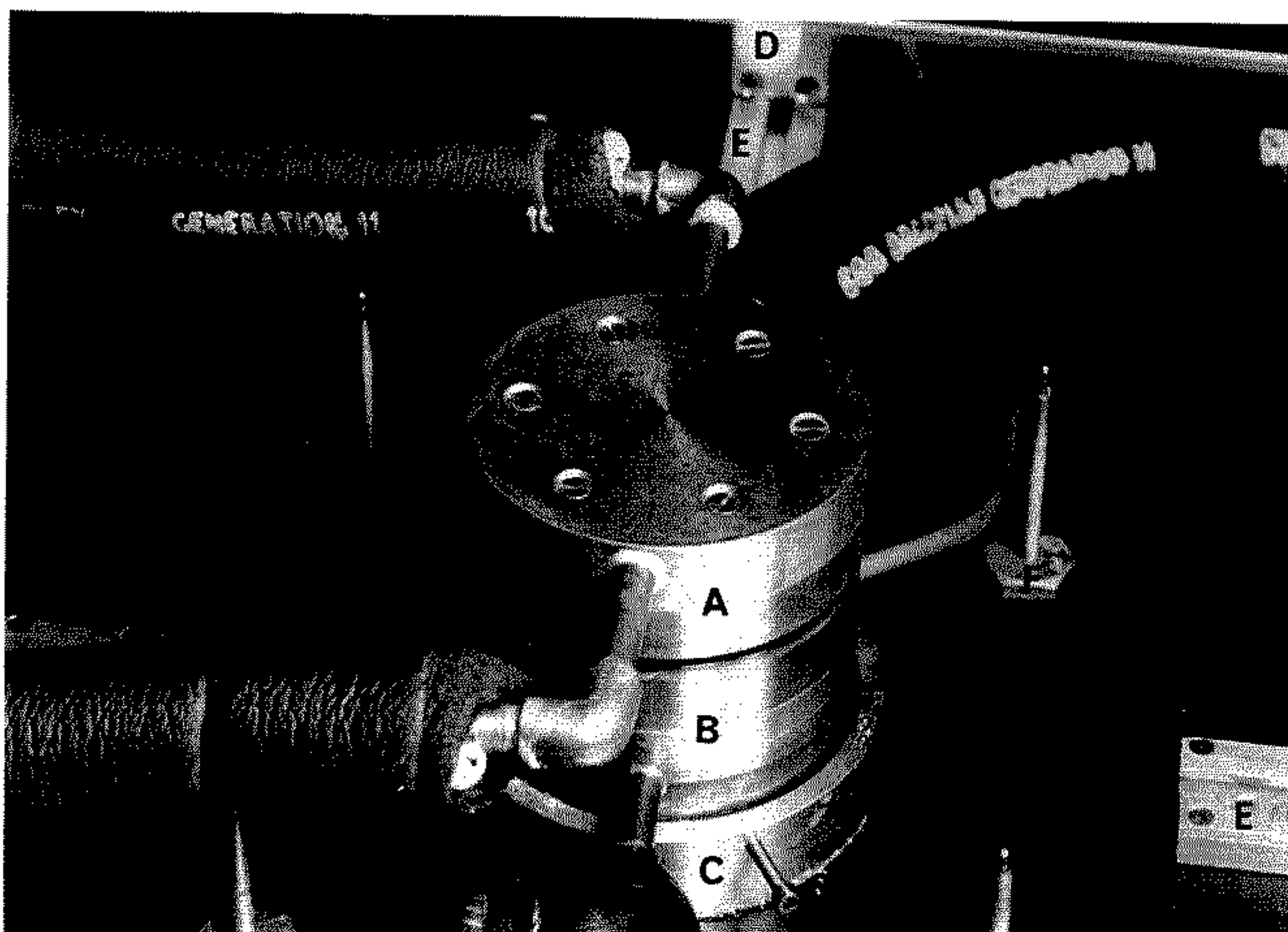


Fig. 2.4: A close-up photograph of the TCA showing a vacuum glazing sample between the cold plate (A) and the hot plate (B) of the apparatus. The hot plate rests in a mounting cup (C). The glazing is held in place by four holders - one of which (D) is shown in this view - which are slid into position along tracks (E) and locked in place. Large glazing samples is supported from below by adjustable ball bearing tipped screws (F). The faint white dots in a regular pattern are the support pillars of the glazing. Water flows to and from the plates through the insulated pipes.

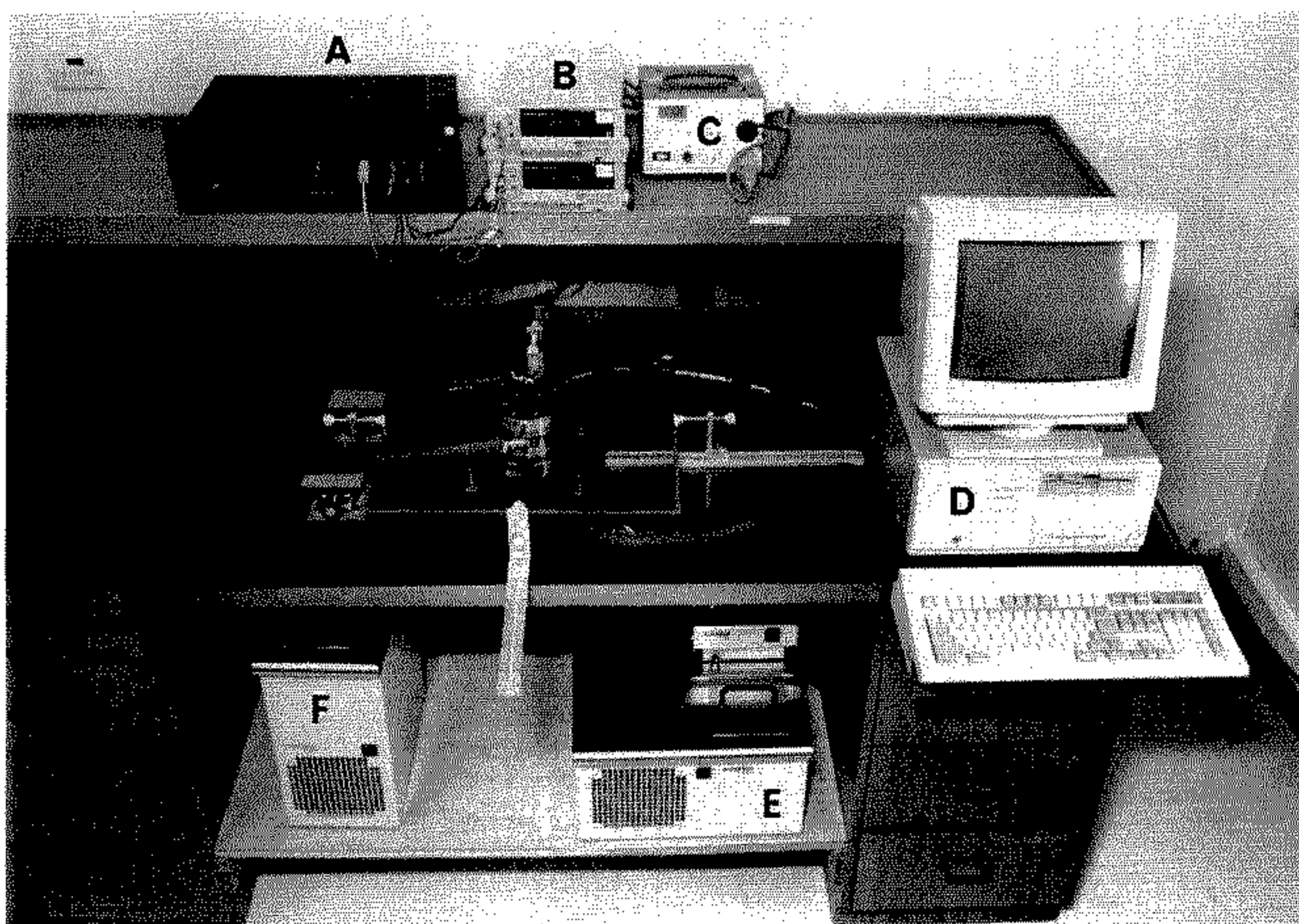


Fig. 2.5: A photograph of the entire apparatus. The TCA is in the center of the view, and there is a vacuum glazing sample between the plates. The power supply and control box (A), digital multimeters (B), and the mains voltage regulator (C) are above the TCA. To the right is the computer for logging and processing the data (D). The hot and cold water baths (E and F) are below the TCA.

side) glass thickness of 6 mm and for a typical pillar separation of 25 mm. As shown below, for all three of these dimensions, the pillar conduction makes only a small contribution to the heat flow measured in the middle of a unit cell of pillars. If the pillar separation is such that the metering piece is about 30 mm from the pillars, as is the case at one location in most vacuum glazing samples for the purpose of thermal testing, then this contribution is negligible.

The copper metering piece is tapered from the measurement diameter at the base of the piece to the much smaller diameter required to hold the power input resistor at the top of the piece. This tapering reduces the thermal mass of the metering piece. The lower thermal mass decreases the response time of the bridge voltage to a change in the resistor power, thereby permitting faster measurements.

In order to achieve a high thermal resistance between the metering piece and the guard, these parts are separated by a narrow annular gap. Within the glass below this gap, the heat flow is not parallel, as shown in Fig. 2.6a. The heat flow from the metering piece can be thought of as spreading out into the hot side glass of the unguarded gap region. This spreading of the heat flow is almost entirely within this layer of glass because the glass thickness is much larger than the width of the annular gap. Thus, the heat flow from the metering piece spreads out from the geometric area of the metering piece to a certain area at the evacuated space, which is referred to as the effective area, A_{eff} .

Since the TCA is to be used to calculate the thermal conductance across the evacuated space of glazing samples, one must know the temperature difference and the heat flow across the space, and the area through which this heat flows across the space. As explained below, the temperature difference can be determined from the temperatures of the water flows in the apparatus. For the TCA measurements, the heat flow across the evacuated space is equal to the null power applied to the metering piece, and the area through which this heat flows across the space is the above described effective area. Therefore, for the purpose of calculating the thermal conductance of samples, A_{eff} can be thought of as the effective measurement area for the metering piece. Clearly this area must be well known. A precise determination of the effective area requires a measurement on a sample of a known thermal conductance, as discussed below.

Similar guarded hot plate devices made at the University of Sydney have shown that the effective area is approximately equal to the area defined by the mid-gap radius. This is consistent with an approximation by Pratt [4]. If the gap is very large, significant non-uniformities in the heat flow can persist even on the cold side of the

apparatus - as shown in Fig. 2.6b. In this case, the effective area at the evacuated space will be poorly defined and dependent on the thermal conductance across the evacuated space. As will be shown, the thermal resistance between the metering piece and the guard is not strongly dependent on the gap width. Therefore, the choice of the gap width is mostly determined by the desirability of having a well defined effective area for the heat flow, and by the ease of construction. With these considerations in mind, the TCA was designed with a 1 mm radial gap between the measurement side surfaces of the metering piece and the guard. Both the metering piece and the surrounding surface of the guard are tapered so that the gap width is 7 mm at the top of the metering piece. This tapering slightly increases the thermal resistance of the air gap between these parts.

The thicknesses of the epoxy resin below the metering piece and the guard can affect the effective area. As shown in Fig. 2.6c, if there is more epoxy resin below the metering piece than the guard, then the isothermal front surface of the metering piece will be displaced further away from the sample than the front surface of the guard. This will result in the heat flow from the metering piece spreading out to less than the expected mid-gap radius. Therefore, the effective area will be less than the area defined by the mid-gap radius. The opposite is true for less epoxy resin below the metering piece. Since the ratio of the thermal conductivities of the glass and the epoxy resin is about 5 to 1, just 0.2 mm of extra epoxy resin below the metering piece can be thought of as being roughly equivalent to an extra 1 mm of glass. Clearly this will have a significant effect on the effective area. Therefore, during the design and construction of the apparatus, it was necessary to consider the possibility of non-uniform epoxy resin thicknesses across the hot plate.

As noted below, it was decided to design the apparatus so that it would have an effective area defined by about the mid-gap radius. Therefore, in constructing the TCA, care was taken to ensure that the thicknesses of the epoxy resin layers below the metering piece and the guard were as nearly equal as possible. The measurement side surfaces of the metering piece and the guard were fixed at the same level with a specially designed temporary holder (which was screwed into the guard), and then these surfaces were bonded to the faceplate at the same time with epoxy resin; a uniform force was applied to the faceplate, and excess epoxy resin was allowed to flow out at the edge of the faceplate in order to leave a thin and even layer of adhesive. A removable teflon plug kept the gap between the metering piece and the guard free of epoxy resin during this procedure. The epoxy resin thicknesses determined from micrometer measurements before and after assembly are: 0.110 ± 0.005 mm below the

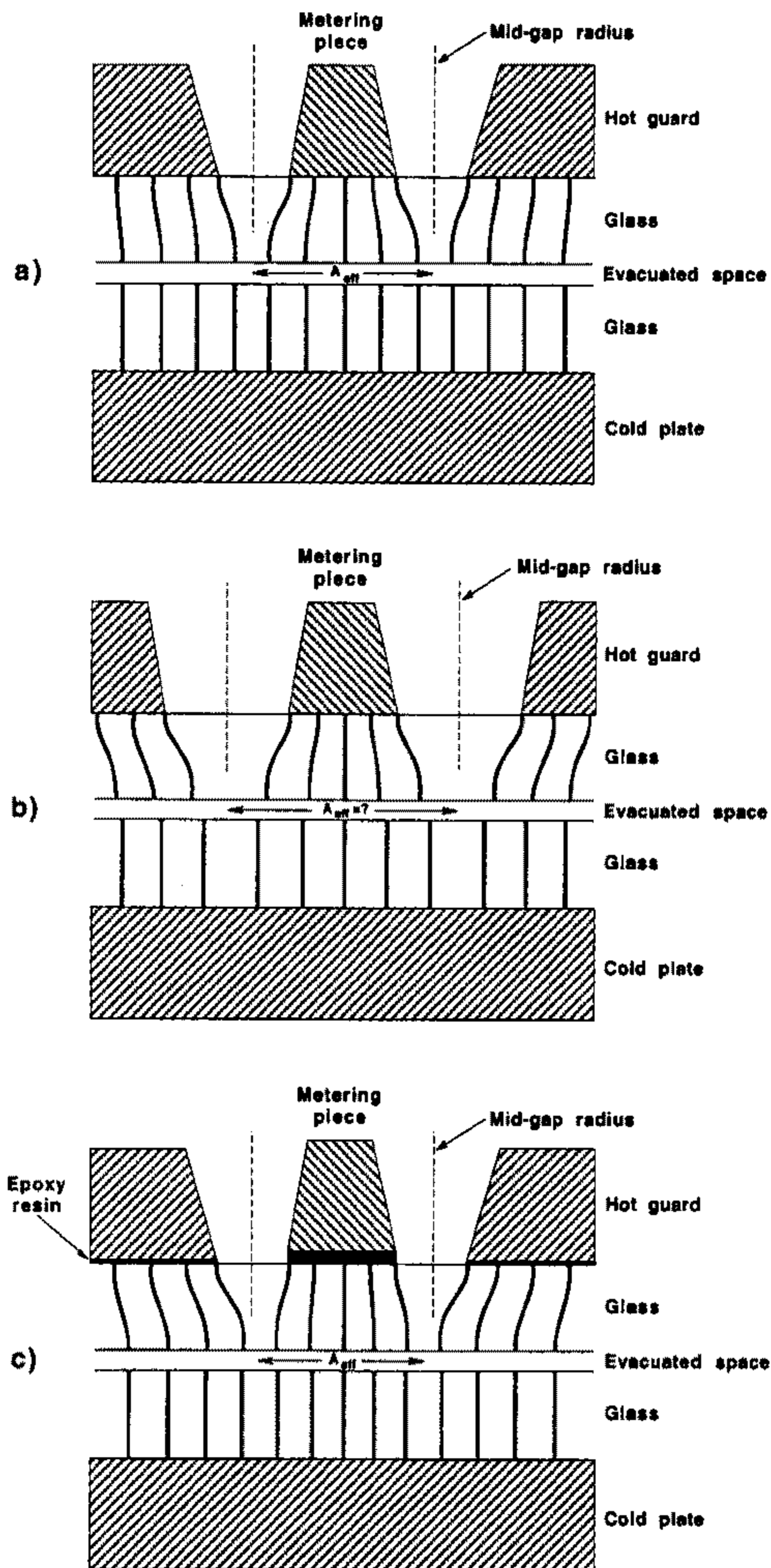


Fig. 2.6:

- (a) The spreading out of the heat flow into the glass below the gap leads to an effective area, A_{eff} for the measurement of the thermal conductance across the evacuated space. For a narrow gap, this area is approximately defined by the mid-gap radius.
- (b) If the gap is very large, significant non-uniformities in the heat flow can persist on the cold side of the apparatus. In this case, the effective area will be poorly defined.
- (c) If there is more epoxy resin below the metering piece than below the surrounding guard, the heat flow from the metering piece will spread out to an effective area which is less than the area defined by the mid-gap radius. All of the above three figures show simplified representations of the actual heat flow.

metering piece and 0.100 ± 0.005 mm below the guard. The thickness of epoxy resin between the cold plate and the cold side faceplate (which does not affect A_{eff}) was measured to be 0.090 ± 0.005 mm.

In order to prevent air bubbles from being trapped in the epoxy resin layers (i.e. from causing non-uniformities in these layers) air was removed from the mixed epoxy resin by placing it in a vacuum immediately prior to the application to the plates. The type of epoxy resin used was "Araldite Super-Strength", for which the manufacturer, CIBA-GEIGY Ltd., estimates the thermal conductivity to be about $0.20 \text{ Wm}^{-1}\text{K}^{-1}$ [5].

2.2.3 Metering Piece Thermal Resistances

The sensitivity of the TCA is strongly dependent on the thermal isolation of the metering piece from the surrounding guard. It is therefore necessary to design the apparatus so that the total thermal resistance between the metering piece and the guard, R_{m-g} , is maximized. As shown in Figs. 2.7a and b, this total resistance is composed of three parallel thermal resistances: the resistance across the air gap $R_{m-g}(air)$, the resistance of the wires that bridge the gap $R_{m-g}(wires)$, and the series combination of the resistances of the glass and epoxy resin below and between the metering piece and the guard, $R_{m-g}(glass+epoxy)$.

The metering piece and the guard are bonded with epoxy resin to a glass faceplate which is put in good thermal contact (via a gel) with one of the glass sheets of the sample. This layered assembly of epoxy resin and glass presents a thermal resistance between the metering piece and the guard. Clearly this resistance can be increased with a thicker layer of epoxy resin or a thicker faceplate. However, as discussed below, increases to the epoxy resin or the faceplate thicknesses will result in greater temperature differences between the hot plate and the hot side internal glass surface; this will result in less well defined temperature differences across the evacuated space and thus less accurate thermal conductance results. Furthermore, for thicker layers of epoxy resin, there can be larger non-uniformities in the thicknesses below the metering piece and guard and thus, as explained above, greater uncertainties in the effective measurement area.

The thermal resistance $R_{m-g}(glass+epoxy)$ was determined by using a finite element model which was constructed with the STRAND6 modelling package. Figure 2.8 is a diagram of the geometry of the cylindrically axisymmetric model that was used for this analysis. All of the dimensions, including the thicknesses of the epoxy resin layers, are

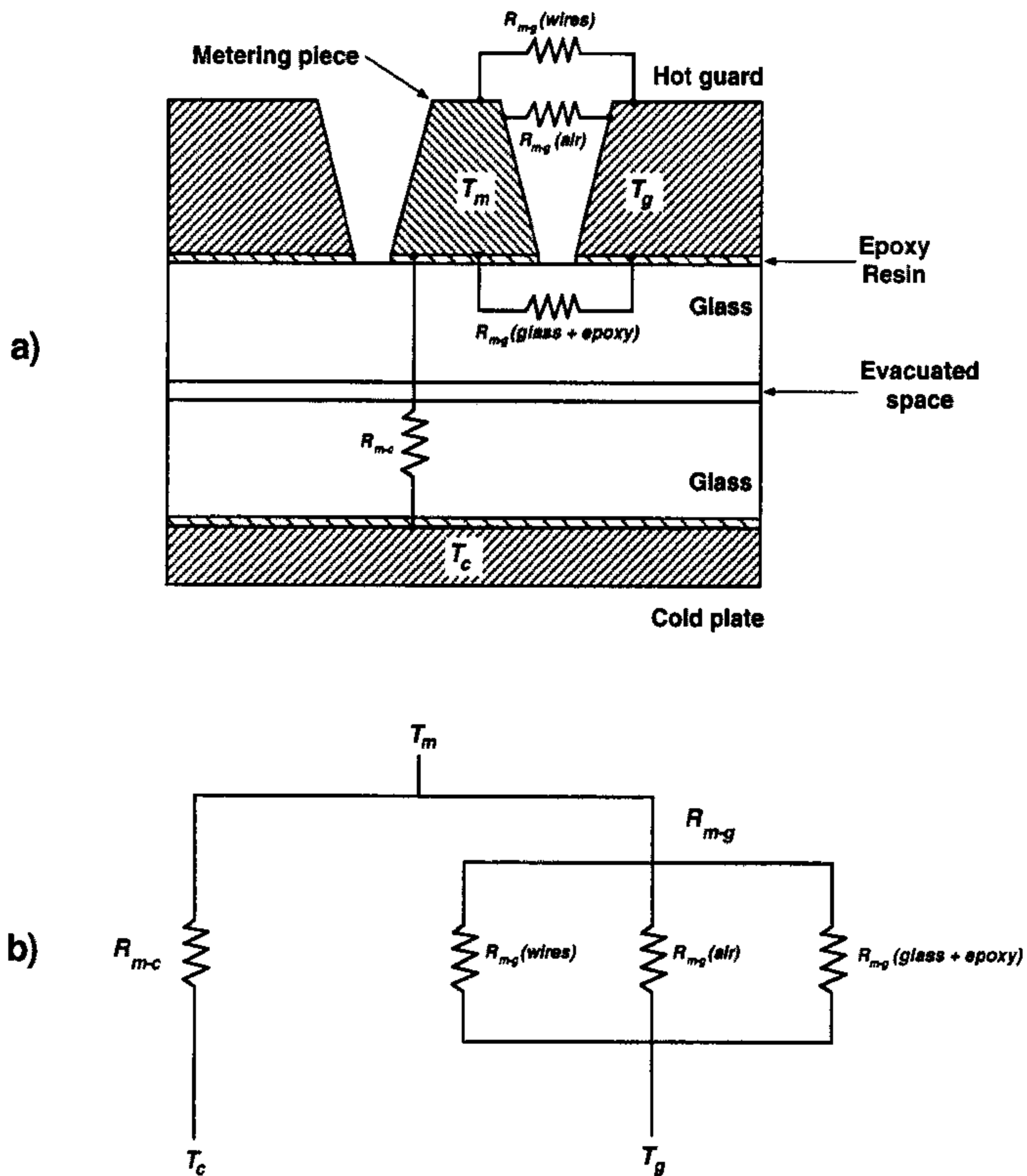


Fig. 2.7: Thermal resistance circuit diagrams of the metering piece resistances:

(a) The thermal resistance between the metering piece and the guard, R_{m-g} , is composed of three resistances in parallel - those associated with the air gap, the wires that bridge this gap, and the series combination of the resistances of the glass and epoxy resin below and between the metering piece and the guard. In parallel with R_{m-g} is the thermal resistance between the metering piece and the cold plate.

(b) A simplified representation of the thermal resistance circuit shown in (a).

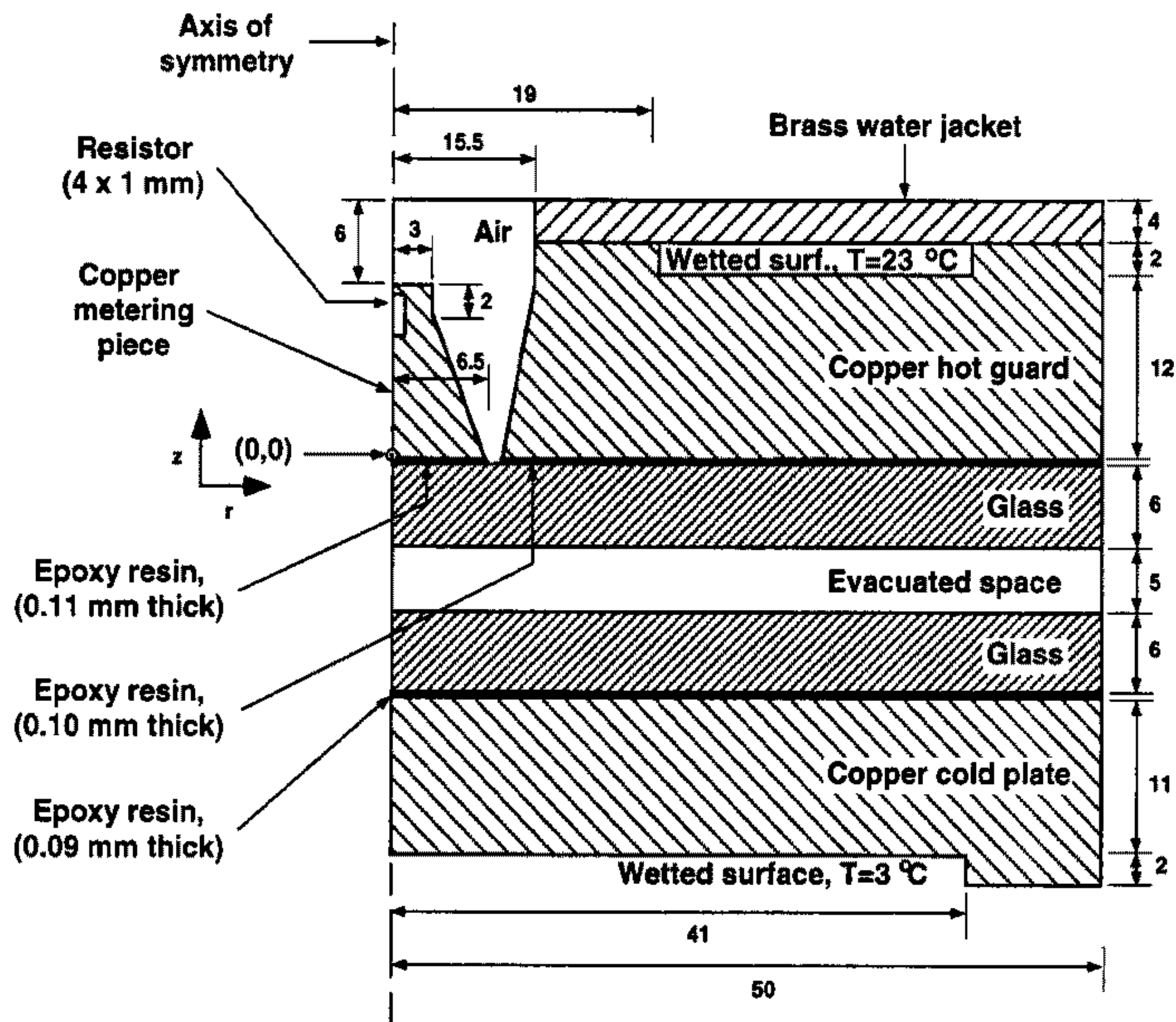


Fig. 2.8: The two-dimensional cylindrically axisymmetric finite element model of the TCA that was used to determine the thermal resistances and the temperature profiles in the apparatus. The gap between the metering piece and the guard is 1 mm wide. The wetted surfaces of the sample side hot water jacket and the cold copper plate are indicated. All dimensions are in millimeters.

the same as those of the actual apparatus and glazing - except for the thickness of the simulated evacuated space which, for the sake of convenience, was chosen to be 5 mm. Only the region between the sample side hot water jacket and the wetted surface of the cold copper plate is modelled. The simulated water temperatures are those which apply to most of the actual tests: 23 and 3 °C. As noted below, the heat transfer coefficients at the wetted surfaces of the water jackets are very large (about 5000 Wm⁻²K⁻¹), and the typical per unit area measured heat flows are of the order of 10 Wm⁻². Therefore, there is only a small temperature difference (a few millikelvin) between the water flows and the wetted surfaces, and it is a sufficient approximation to assign the above water temperatures to the respective wetted surfaces of the model. The issue of the temperature non-uniformities across the wetted surfaces due to the water flows is addressed later. Between each metal plate and the evacuated space is 6 mm of soda-lime glass (2 mm for the faceplate and 4 mm for the glass sheet of the glazing which is in contact with this faceplate). The thermal conductivity of the glass is equal to that given in Appendix 1 for soda-lime glass: 1.0 Wm⁻¹K⁻¹. As shown in later chapters, the heat transfer across the evacuated space of a vacuum glazing can be simulated as conduction through a slab with an appropriate uniform thermal conductance. For this model, the evacuated space is modelled by a 5 mm thick slab which has a thermal conductivity of 0.005 Wm⁻¹K⁻¹; this slab simulates a thermal conductance of 1.0 Wm⁻²K⁻¹ - which is close to the typical sample conductances for the TCA measurements. Heat is dissipated in the resistor elements, with the elemental power density set to correspond to the total desired power. The thermal conductivities for the copper, brass, and epoxy resin components of the model are, respectively, 386 Wm⁻¹K⁻¹ [6], 111 Wm⁻¹K⁻¹ [6], and 0.20 Wm⁻¹K⁻¹ [5]. The thermal conductivity for the air around the metering piece corresponds to the approximate hot plate temperature of 23 °C: 0.03 Wm⁻¹K⁻¹ [7]. The external surfaces of the model are defined to be adiabatic.

Using this finite element model, the following was determined for each of two resistor powers: the total radial heat flow in the glass at the mid-gap radius, and the difference between the average temperatures of the base of the metering piece and the base of the guard around the metering piece. The series resistance $R_{m-g}(glass+epoxy)$ is approximately equal to the change in this temperature difference divided by the change in this radial heat flow for the two resistor powers and it was calculated to be 34.9 KW⁻¹. A simple analytic model that is presented in Appendix 2 was used to estimate this resistance. The analytic estimation of 35.5 KW⁻¹ is approximately consistent with the above more exact finite element modelling result. The analytic model shows that the largest contribution to $R_{m-g}(glass+epoxy)$ comes from the glass

component of this total thermal resistance, $R_{m-g}(glass)$. The analytic model reveals the parameters which influence this glass resistance:

$$R_{m-g}(glass) = \frac{1}{2k_{glass}r_{gap} \ln(\frac{t}{g/2})}, \quad (2.1)$$

where r_{gap} is the mid-gap radius, t is the thickness of the glass and g is the width of the gap. This equation shows that the thermal resistance of the glass between the metering piece and the guard is not strongly dependent on the width of the gap, as long as the gap width is much less than the total glass thickness (i.e. if $g \ll t$). This is confirmed by Collins et al. whose finite element modelling results show that, for a cylindrically shaped metering piece of diameter 12 mm, a total glass thickness of 6 mm (on each side of the apparatus), and an annular gap of width 2 mm, the total thermal resistance between the metering piece and the guard (of which the thermal resistance of the intervening glass is the most significant component - as noted below) changes by only 3.7 % for a 10 % change in the gap width [3].

The thermal resistance across the air gap, $R_{m-g}(air)$, is approximately equal to the change in the difference between the average surface temperatures of the metering piece and the guard on each side of the gap divided by the change in the radial heat flow across the gap (with these changes being between results at two different resistor powers). The same finite element model described above was used to determine the temperature differences and heat flows for this analysis, from which $R_{m-g}(air)$ was calculated to be 236 KW⁻¹.

The above modelling result for $R_{m-g}(air)$ can be checked with an simple analytic estimation which involves treating the air gap between the central metering piece and the encircling guard as a layer of insulation between a long isothermal cylinder and its surrounds. The equation for the thermal resistance presented by the air gap to the radial heat flow from this insulated cylinder is:

$$R_{m-g}(air) = \frac{\ln(r_o/r_i)}{2\pi k_{ins} L_c}. \quad (2.2)$$

where r_o and r_i are the outer radii of the insulation and the cylinder (which, in this case, are the outer and inner gap radii respectively), k_{ins} is the thermal conductivity of the insulation (the air), and L_c is the length of the insulated cylinder (the metering piece). Since the gap is tapered, the above radii vary with the lineal dimension and thus an

exact solution of the equation requires an integration with respect to this dimension. An approximate solution can be found by using the average radii of $r_o=8.75$ mm and $r_i=4.75$ mm. Substituting these values into equation 2.2 along with $k_{ins}=0.03$ Wm⁻¹K⁻¹ for air at 23 °C [7], and $L_c=12$ mm gives a value for $R_{m-g}(air)$ of 270 KW⁻¹. The modelling result of 236 KW⁻¹, which was given above, is approximately consistent with this analytic estimation.

There are six copper wires from the metering piece to the surrounding guard - four which are connected to the resistor and two which are connected to the thermistor. The wires are about 20 mm long and they have a copper core of diameter 0.1 mm. The thermal resistances of these wires are in parallel. Neglecting the thermal resistance of the thin teflon coating around the cores, the wires present a combined thermal resistance between the metering piece and the guard, $R_{m-g}(wires)$, of about 1100 KW⁻¹.

The total thermal resistance between the metering piece and the guard is therefore:

$$\begin{aligned}
 R_{m-g} &= \left(\frac{1}{R_{m-g}(glass+epoxy)} + \frac{1}{R_{m-g}(air)} + \frac{1}{R_{m-g}(wires)} \right)^{-1} \\
 &= \left(\frac{1}{34.9 \text{ KW}^{-1}} + \frac{1}{236 \text{ KW}^{-1}} + \frac{1}{1100 \text{ KW}^{-1}} \right)^{-1} \quad (2.3) \\
 &= 29.6 \text{ KW}^{-1} .
 \end{aligned}$$

It is noted that this total thermal resistance is dominated by the smallest of the three contributing thermal resistances: $R_{m-g}(glass+epoxy)$. Since, as shown by the analytic model referred to above, almost all of $R_{m-g}(glass+epoxy)$ is due to the glass below and between the metering piece and the guard, the thermal resistance of this glass makes the most significant contribution to R_{m-g} .

There is a fourth thermal resistance associated with the metering piece - the resistance between the metering piece and the cold plate, R_{m-c} , which is in parallel with R_{m-g} - as shown in Figs. 2.7a and b. The resistance between the metering piece and the cold plate (which is at temperature T_c) is dominated by the resistance of the sample under test so that $R_{m-c} \approx 1/(C_{sample}A_{eff})$, where C_{sample} is the thermal conductance across the evacuated space of the sample and A_{eff} is the effective measurement area. For the present analysis, the effective area can be assumed to be defined by the mid-gap radius of 7 mm - i.e. $A_{eff}=1.54 \times 10^{-4}$ m². At locations remote from the edges of glazing

samples, the measured local sample conductance depends on the emittance of the internal glass surfaces and on the proximity of pillars to the metering piece, and it is expected to be between about 0.6 and 5 Wm⁻²K⁻¹. For the above assumed effective area, the corresponding values for R_{m-c} are 10 800 and 1300 KW⁻¹. It is noted that these thermal resistances are much greater than the above modelling result for R_{m-g} (i.e. $R_{m-c} \gg R_{m-g}$)

Heat which is added to the metering piece from the power resistor flows through the effective thermal resistance of the metering piece, R_{eff} . Figure 2.9 shows such a heat input, Q , to a thermal resistance circuit which is equivalent to the circuit of Fig. 2.7b. At point A (which is at the metering piece), there are two other heat flows: Q_1 through R_{m-g} (across which there is a temperature difference of $T_g - T_m$), and Q_2 through R_{m-c} (across which there is a temperature difference of $T_m - T_c$). Since these thermal resistances, heat flows and temperature differences are respectively analogous to electrical resistances, currents and voltages, the thermal circuit of Fig. 2.9 can be solved by basic electric circuit analysis techniques. A calculation of the temperature difference $T_g - T_c$ and a summation of the heat flows at point A, gives the following expression for the temperature difference $T_m - T_g$:

$$T_m - T_g = -\frac{R_{m-g}(T_g - T_c)}{R_{m-g} + R_{m-c}} + \frac{QR_{m-g}R_{m-c}}{R_{m-g} + R_{m-c}}. \quad (2.4)$$

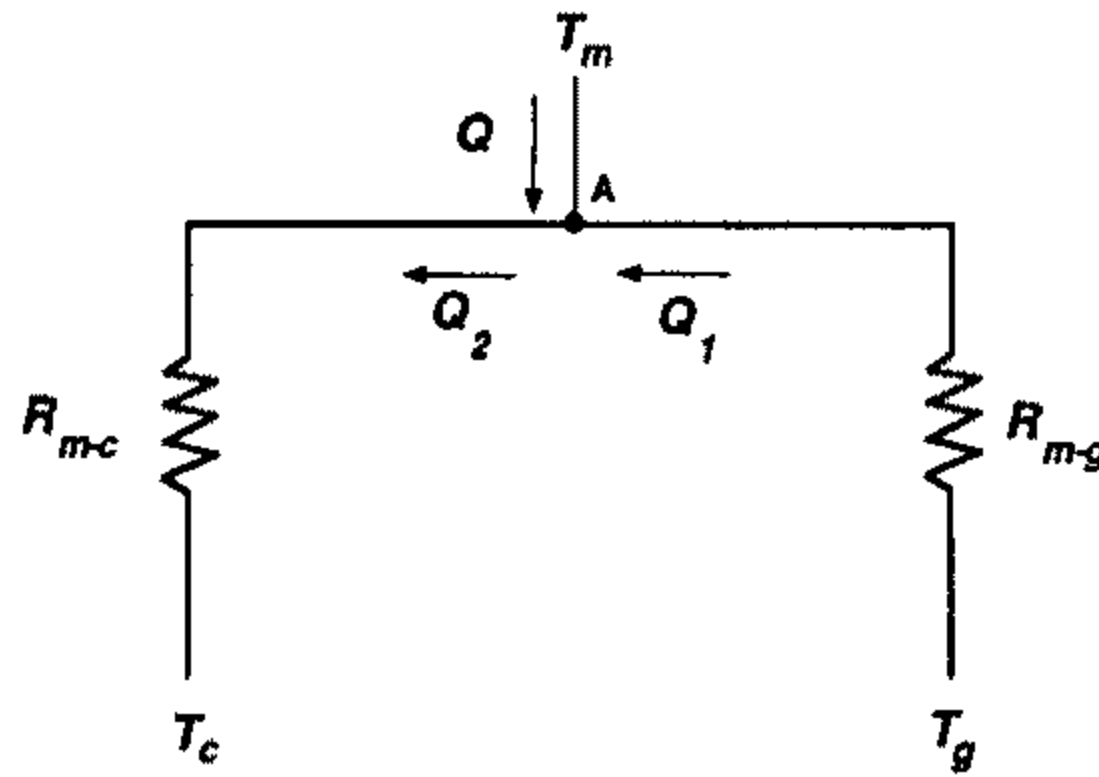


Fig. 2.9: The temperature difference between the metering piece and the guard, $T_m - T_g$, can be determined by considering the heat flows and temperatures of this thermal resistance circuit (which is equivalent to the circuit presented in Fig. 2.7b). The heat flows are: Q added to the metering piece via the resistor, Q_1 from the guard to the metering piece, and Q_2 from the metering piece across the sample and to the cold plate. Point A is at the metering piece.

Equation 2.4 gives the temperature difference between the metering piece and the guard. The second term of this equation represents the change in the temperature difference caused by the heat, Q , which is added to the metering piece via the resistor. This term shows that the thermal resistance through which this heat flows, R_{eff} , is equal to the parallel combination of R_{m-g} and R_{m-c} . Since, as shown above, $R_{m-c} \gg R_{m-g}$, it can be assumed that $R_{eff} \approx R_{m-g}$. Thus:

$$\begin{aligned} T_m - T_g &\approx -R_{m-g} \frac{(T_g - T_c)}{R_{m-c}} + QR_{m-g} \\ &\approx -R_{m-g} (Q_{sample} - Q), \end{aligned} \quad (2.5)$$

where Q_{sample} is the heat that flows from the metering piece and across the sample to the cold plate. As expected, $T_m - T_g = 0$ for the null condition of $Q = Q_{sample}$.

The temperature difference between the metering piece and the guard is proportional to R_{m-g} , as per equation 2.5. Since the bridge voltage is proportional to this temperature difference (as shown in the next section), V_b is also proportional to R_{m-g} . Since the heat input Q to the metering piece flows through the effective thermal resistance of the piece, Q is proportional to R_{eff} . Furthermore, since $R_{eff} \approx R_{m-g}$ (as noted above), Q is approximately proportional to R_{m-g} . Due to these dependencies of V_b and Q on R_{m-g} , a plot of the bridge voltage versus the power input to the metering piece resistor should be approximately linear and have a slope which is approximately equal to R_{m-g} times a constant of proportionality in units of VK^{-1} . This can be expressed as:

$$\frac{V_b}{Q} \approx G_{T-V} R_{m-g}, \quad (2.6)$$

where the constant of proportionality, G_{T-V} , is the temperature-to-voltage gain for the apparatus. This gain is derived in the next section.

2.2.4 Electronics

Figure 2.10 is a schematic diagram of the electronics of the apparatus. The four arms of the bridge circuit consist of two thermistors (each with a resistance of $R_{th} = 200 \text{ k}\Omega$ measured at room temperature), and two resistors (each with a resistance of $R = 10 \text{ k}\Omega$). The bridge voltage input of 6 V is from a highly stable power supply. The bridge voltage output is amplified by a Maxim 430CPA chopper-stabilised operational

amplifier (op-amp). This type of op-amp is used because it has a highly stable output signal. The feedback resistor has a resistance of $R_f=997\text{ k}\Omega$. The op-amp circuit and the two bridge resistors are mounted on a printed circuit board in the electronics well of the hot plate. All external connections are via a grounded and shielded cable. The hot plate is also grounded.

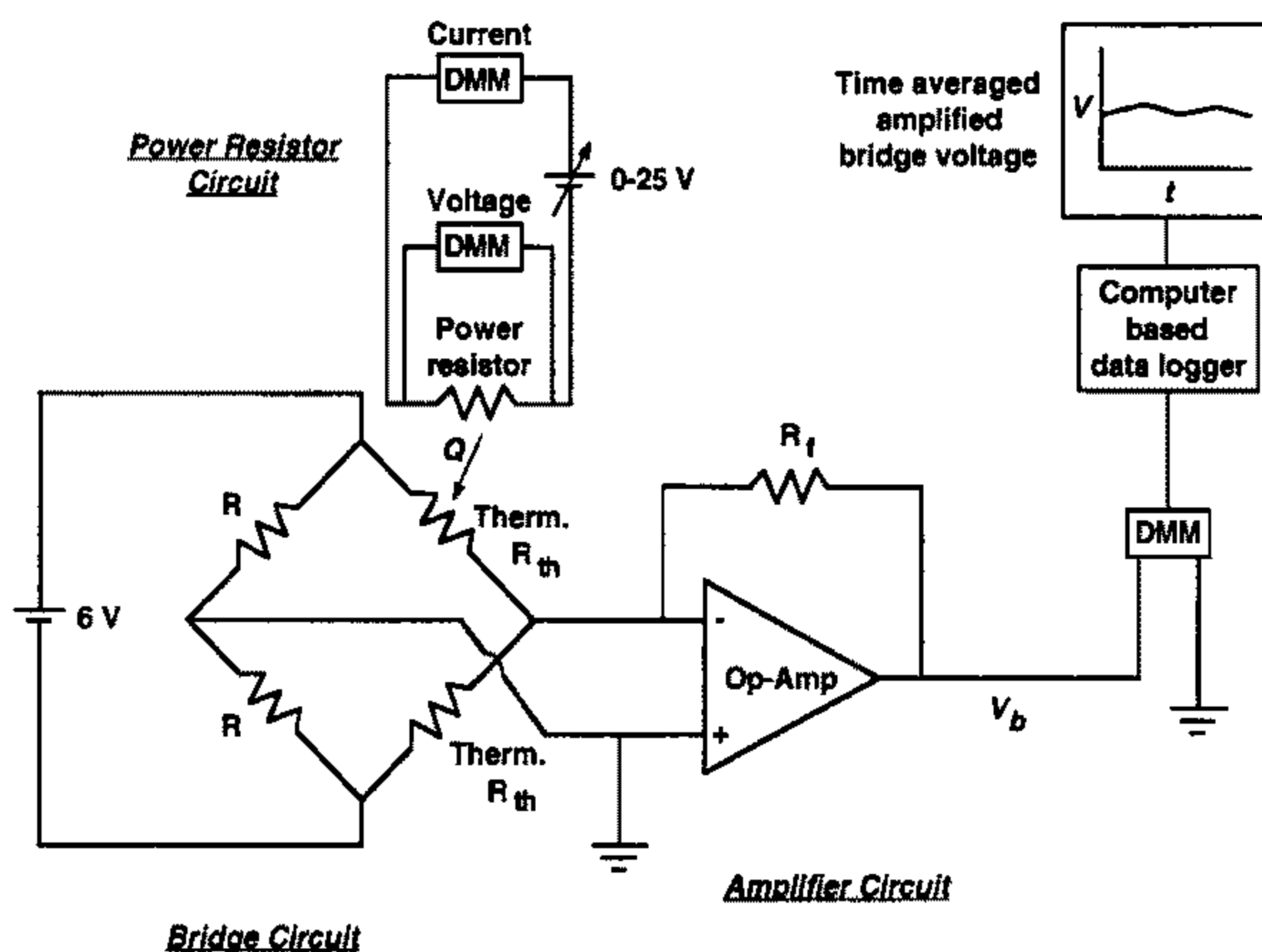


Fig. 2.10: Schematic diagram of the apparatus electronics.

The output from the op-amp, V_b , is connected to a digital multimeter for instantaneous readings, and also to a computer where it is recorded and time averaged over 15 seconds in order to filter out short term fluctuations in the signal. A running average of the time averaged bridge voltage results is also calculated by the computer. The power supply for the power resistor has a voltage range of 0 to 25 V. Since the resistance of the power resistor is $10\text{ k}\Omega$, the maximum power that can be dissipated in the metering piece is about 60 mW - which is more than enough for the measurements to be made with the TCA. The resistor has a ceramic core and a plastic coating and it is set in epoxy resin in a vertical hole at the top of the metering piece.

The most accurate method of measuring the power applied to the resistor is by a four-terminal connection to the resistor leads: two wires for the current supply and measurement, and two wires for the voltage measurement. This makes it possible to calculate the power on the basis of both the current and the voltage without having to

correct for wire resistances. Power calculations based on the resistance and either the current or the voltage could be inaccurate due to possible long term changes in the resistance or due to temperature dependencies of the resistance. The resistor current and voltage are measured with Fluke 8842A digital multimeters. The current readings are accurate to about $\pm 1\mu\text{A}$ and the voltage readings are accurate to about $\pm 0.5\text{ mV}$. All wires to the power resistor and to the thermistors are teflon coated copper wire, for which the core diameter is 0.1 mm and the coating thickness is 0.05 mm.

It is recalled that the amplified bridge voltage provides a relative indication of the temperature difference between the thermistors in the metering piece and the guard, i.e. $T_m - T_g$. It is useful to know the actual correlation between this voltage and the absolute temperature difference between the thermistors since this gives an indication of the temperature sensitivity of the apparatus. This correlation is the temperature-to-voltage gain - i.e. the constant $G_{T,V}$ of equation 2.6.

In order to calculate $G_{T,V}$, one can consider the bridge and amplifier circuits as shown in Fig. 2.11. The resistance of a thermistor changes with temperature. Let thermistor 1 be the thermistor in the metering piece (where the temperature changes) and let thermistor 2 be the thermistor in the guard (where the temperature, and thus the thermistor resistance, are assumed to be constant). Both thermistors have a nominal resistance of R_{th} . It is assumed that the bridge is initially balanced so that, for equal thermistor temperatures, the amplified bridge voltage, V_b , is zero. In response to a small change of temperature in the metering piece, ΔT , the resistance of thermistor 1 changes by ΔR and thus has a new resistance of $R_{th} + \Delta R$. This results in an unbalanced bridge and thus a new value for V_b .

Consider point A between the thermistors. Since the op-amp input voltages must remain equal, and since the resistances of the resistors in the other arms of the bridge are about twenty times less than the thermistor resistances, the voltage at this point, V_A , remains essentially constant despite the change in the resistance of thermistor 1. As a result of this, and also due to the fact that the thermistor resistances are still approximately the same (since, as will be shown, $\Delta R \ll R_{th}$), the voltage across each resistor also remains constant and equal to one half of the bridge voltage ($V/2$). There are three currents at point A: the current from thermistor 1 (i_1), the current to thermistor 2 (i_2), and the current flowing towards the op-amp feedback loop (i_3). Due to the high input impedance of the operational amplifier, the current to the inverting terminal, i_- , and the current from the non-inverting terminal, i_+ , are extremely small (≈ 0). The return current to the bridge, i_4 (which is equal to i_3) flows from the ground

connection into the other arms of the bridge. Since $R_{th} \gg R$, this current does not significantly affect the potentials in the bridge.

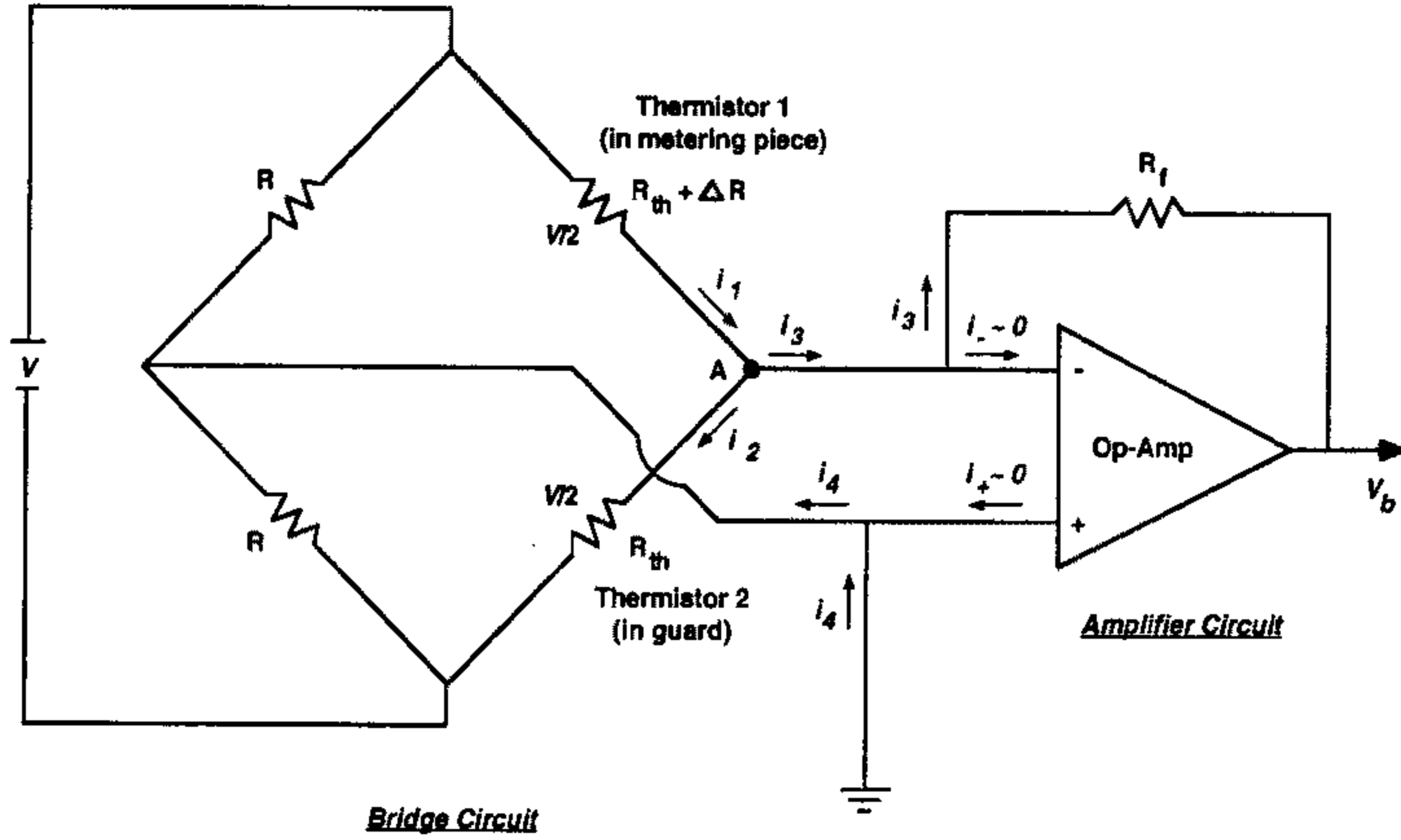


Fig. 2.11: Schematic diagram of bridge and amplifier circuits. The temperature-to-voltage gain can be determined by a summation of the currents at point A, and by recognizing that the voltages at the op-amp inputs remain constant despite a temperature induced change of ΔR in the resistance of thermistor 2. The currents at the op-amp terminals are assumed to be zero.

The current from thermistor 1 is:

$$i_1 = \frac{V}{2(R_{th} + \Delta R)} . \quad (2.7)$$

The current to thermistor 2 is:

$$i_2 = -\frac{V}{2R_{th}} . \quad (2.8)$$

Thus, the net current flowing towards the feedback loop is:

$$\begin{aligned} i_3 &= i_1 + i_2 \\ &= \frac{V}{2R_{th}} \left(-\frac{1}{1 + \frac{\Delta R}{R_{th}}} - 1 \right) . \end{aligned} \quad (2.9)$$

Since, as shown below, $\Delta R \ll R_{th}$, equation 2.9 can be approximately expressed as:

$$i_3 = -\frac{V}{2R_{th}}\left(\frac{\Delta R}{R_{th}}\right). \quad (2.10)$$

As almost no current enters the inverting terminal of the op-amp ($i_- \approx 0$), all of i_3 goes to the feedback resistor of resistance R_f . The absolute value of the amplified output voltage, V_o , is thus:

$$\begin{aligned} V_o &= i_3 R_f \\ &= \frac{VR_f}{2R_{th}}\left(\frac{\Delta R}{R_{th}}\right). \end{aligned} \quad (2.11)$$

The temperature-to-voltage gain for the apparatus is defined as the change in the amplified bridge voltage in response to a change in the difference of the thermistor temperatures:

$$G_{T-V} = \frac{\Delta V_b}{\Delta(T_m - T_g)}. \quad (2.12)$$

For small changes in temperature, the term $\Delta R/R_{th}$ of equation 2.11 can be expressed as a linear function of temperature in units of K^{-1} . If this term is expressed in this manner, equation 2.11 gives the bridge voltage as a function of the original change in the temperature of thermistor 1 - and thus as a function of the difference in the thermistor temperatures (since the temperature of thermistor 2 is assumed to be constant). In this form, equation 2.11 is equivalent to equation 2.12. The thermistor supplier quotes a value for $\Delta R/R_{th}$ of $0.04 K^{-1}$ for temperatures near room temperature (and thus this value applies to the operating temperature of the hot plate) [8]. Substitution into equation 2.11 of $\Delta R/R_{th}=0.04 K^{-1}$, along with $V=6 V$, $R_f=997 k\Omega$ and $R_{th}=200 k\Omega$ gives a temperature-to-voltage gain of $G_{T-V}=0.60 VK^{-1}$.

The maximum expected change in the temperature of the metering piece thermistor is about $0.4 K$. Based on $\Delta R/R_{th}=0.04 K^{-1}$ and $R_{th}=200 k\Omega$, this corresponds to a maximum expected change in the thermistor resistance of about $\Delta R=3 k\Omega$. Thus, as stated above, ΔR is indeed much less than R_{th} .

It was decided to design the TCA so that it would be possible to measure small heat flows to an accuracy of $\pm 1\%$. The amount of heat flow to be measured most

commonly with the TCA is about 3 mW. Therefore, the required absolute accuracy for the measurement of this amount of heat flow is ± 0.03 mW. For the metering piece-to-guard thermal resistance of $R_{m-g}=29.6 \text{ KW}^{-1}$ (as predicted by equation 2.3), this absolute accuracy in the heat flow corresponds to a temperature difference of about 0.9 mK between the metering piece and the guard - or, more specifically, between the thermistors in these parts. This is the maximum allowable temperature difference between the two thermistors for the required accuracy of $\pm 1 \%$ in the measurement of the 3 mW heat flow. In order to ensure that this temperature difference is not exceeded, the apparatus must be designed so that the thermistors are properly located - as discussed in the next section. It is of interest to note that, for $G_{T-V}=0.60 \text{ VK}^{-1}$, the ± 0.9 mK difference in the thermistor temperatures corresponds to about a ± 0.5 mV accuracy in the amplified bridge voltage.

It is important to realize that the above temperature difference - and the corresponding accuracy in the bridge voltage measurement - is proportional to the heat flow across the sample (and thus to the sample conductance if the temperature difference across the evacuated space is the same). This is due to the approximately linear dependence of the heat flow on the glass-to-glass temperature difference, and due to the fact that the thermal conductance of the sample is much less than the other conductances in the system. Therefore, the maximum allowable temperature difference will be greater for a larger heat flow, but it will still correspond to a $\pm 1 \%$ accuracy in the heat flow measurement. As discussed below, the dominant factor in determining the accuracy of the apparatus is the uniformity in the thicknesses of the layers of epoxy resin which bond the metering piece and the guard to the hot side glass faceplate.

2.2.5 Location of The Thermistors

In the design of this guarded hot plate apparatus, the value of the effective area is to some extent a matter of choice. This is because, in the operation of the TCA, it is not possible to set the temperatures of the measurement side surfaces of the metering piece and the guard to be precisely equal. It is instead only possible to ensure that the temperatures of these two parts are equal at the points at which the relevant temperature sensors are located. In the design of the apparatus, these sensors must be located at positions in the metering piece and the guard where, for a specific heat flow, the temperatures are equal or agree to within the required accuracy. The actual heat flow for this calculation was chosen to be equal to that which would intersect an effective area which is defined by the radius at the middle of the gap between the metering piece and the guard. Under this condition, the null power will correspond to

the heat flow across the sample as a whole. A finite element analysis of the TCA was performed to select thermistor locations that meet this requirement. This analysis is based on a null power which is approximately equal to the 3 mW heat flow considered above. It is recalled that, in order to achieve a $\pm 1\%$ accuracy in the measurement of a 3 mW heat flow, the thermistor temperatures must differ by less than 0.9 mK. As will be shown below, the thermistor locations selected from the results of this analysis are appropriate for all other expected null conditions.

The model used for this analysis is the same as the one that was described in Section 2.2.3 except that no epoxy resin layers were simulated. This is because the thermistor locations had to be selected, and the corresponding holes to these locations had to be drilled, prior to the assembly of the apparatus - i.e. before the epoxy resin thicknesses were known. The effect of the epoxy resin layers is discussed later. For this analysis, the thermal conductance across the evacuated space of the model is equal to the previously considered approximately typical sample conductance of $1.0 \text{ Wm}^{-2}\text{K}^{-1}$. As mentioned above, the analysis is based on an effective area of the metering piece which is defined by the mid-gap radius. Since this radius is 7 mm, $A_{\text{eff}} = 1.54 \times 10^{-4} \text{ m}^2$. The hot and cold plate wetted surface temperatures are 23 and 3 °C. According to the procedure presented in the next section, the corresponding temperature difference between the internal glass surfaces is 19.7 °C (although the procedure does consider the effect of the epoxy resin layers, this effect is negligibly small - about 0.02 °C). The null power which corresponds to this temperature difference and to the above sample conductance and effective area is $Q = 3.03 \text{ mW}$. The power density in the resistor was set to correspond to this value in order to simulate the null condition.

Figure 2.12 presents the modelling results for the temperature within the hot plate as a function of the radial distance from the axial centerline of the metering piece. Data are presented for three distances, z , from the measurement side surfaces of the metering piece and the guard: 2, 3 and 4 mm. The shape of each profile is as expected. The maximum temperature in the metering piece is immediately below the hot resistor at $r = 0 \text{ mm}$. The temperature decreases from this location and then varies rapidly across the high thermal resistance of the air gap. Due to the tapered shape of this gap, the discontinuities in the temperature gradients at the gap edges are at different locations in the three profiles. On the other side of the gap, in the guard, the temperature rises to a maximum value close to the mid-radius of the guard (due to the greater proximity to the entire hot side wetted surface at this location) and then it decreases again. As expected, the temperature increases with z - i.e. with increasing proximity to the resistor and the water jacket. These plots show that the two thermistors could be

located anywhere between $z=2$ and $z=4$ mm and still be well within the maximum allowable temperature difference of 0.9 mK for the simulated null power.

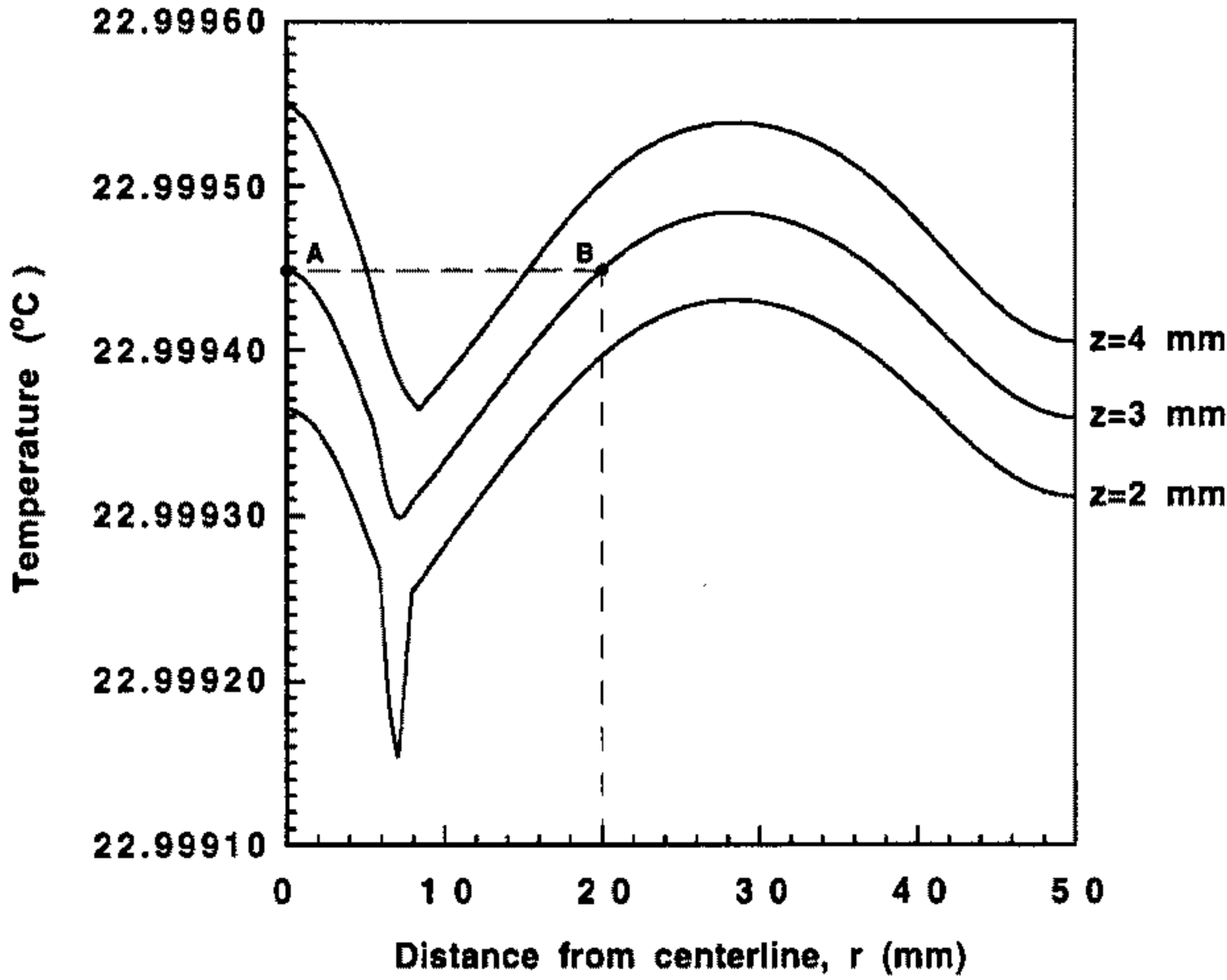


Fig. 2.12: Finite element modelling results for the temperature profiles within the hot plate for three different distances, z , from the measurement side surfaces of the metering piece and the guard: 2, 3 and 4 mm. The radial distance, r , is measured from the axial centerline of the metering piece. The sample conductance is $1 \text{ Wm}^{-2}\text{K}^{-1}$ and the temperatures of the wetted surfaces of the plates are 23 and 3 °C. The power which is dissipated in resistor, 3.03 mW, satisfies the null condition for an effective area that is defined by the mid-gap radius, $A_{\text{eff}} = 1.54 \times 10^{-4} \text{ m}^2$, and for a temperature difference between the internal glass surfaces of 19.7 °C. It was decided to locate the thermistors at points A and B which have (r, z) coordinates of (0, 3) and (20, 3), respectively.

It is recalled that the results of Fig. 2.12 are from a model without epoxy resin layers. Since the epoxy resin thicknesses were unknown before the thermistor locations had to be selected, the selection was done on the basis of the data in this figure. In order to best accommodate the inevitable effects from the epoxy resin on the measurements, it was decided to place the thermistors at locations which, according to Fig. 2.12, are approximately isothermal locations. This figure shows two such locations, points A and B, both of which are at $z=3$ mm from the measurement side surfaces of the metering piece and the guard. The metering piece thermistor is located at radius $r=0$ mm (point A) and the guard thermistor is located at radius $r=20$ mm (point B). The exact temperature difference between these two locations, $T_m - T_g$, is $0.6 \text{ } \mu\text{K}$ - which is

well below the required limit of 0.9 mK. It is important to note that the model shows that, even for the largest expected thermal conductance across the evacuated space ($5 \text{ Wm}^{-2}\text{K}^{-1}$) and for the same temperature difference of 20°C between the water flows (which is the expected maximum difference), $T_m - T_g$ is only 0.003 mK for the corresponding null power; this five-fold increase in the temperature difference with the same increase in the sample conductance (and thus in the heat flow, since the temperature difference across the evacuated space approximately the same) is consistent with the statement at the end of the last section above about $T_m - T_g$ being proportional to the heat flow across the sample. Therefore, the thermistor locations selected from the above analysis are appropriate for the range of the expected null conditions.

After the construction of the apparatus, the thicknesses of the epoxy resin layers at the metering piece and the guard were measured to be $0.110 \pm 0.005 \text{ mm}$, and $0.100 \pm 0.005 \text{ mm}$, respectively. As explained above, these unequal epoxy resin thicknesses will influence the effective area for the null measurement - and thus this area will not be defined by the mid-gap radius. It is recalled that the above determination of the approximately isothermal thermistor locations is based on the effective area being defined by the mid-gap radius. Therefore, by virtue of its influence on the effective area, the inequality in the epoxy resin thicknesses will result in a temperature difference between the thermistors. This difference was quantified by simulating the mean thicknesses of the epoxy resin layers in the finite element model (as shown in Fig. 2.8). The same sample conductance, wetted surface temperatures and null power considered in the above analysis were also simulated. For the above mean epoxy resin thicknesses below the metering piece and the guard of 0.100 and 0.110 mm, the temperature difference between the previously selected thermistor locations was determined to be $T_m - T_g = 2.3 \text{ mK}$. This is more than twice the desired maximum temperature difference of 0.9 mK. This is why it was so important to ensure that the epoxy resin thicknesses were as consistent as possible. The use of an epoxy resin with a higher thermal conductivity would have reduced the effect of the unequal thicknesses, but a suitable product could not be found. A higher thermal conductivity could have been obtained by loading the epoxy resin with fine metal particles (e.g. copper powder). However, the concentration of metal that would be required to increase significantly the thermal conductivity of the epoxy resin would prevent the formation of an effective adhesive bond.

It is important to note that the above temperature difference of $T_m - T_g = 2.3 \text{ mK}$ does not mean that the temperature measurements will actually be in error by this amount.

Rather, the null condition will correspond to an effective area which differs slightly from the area defined by the mid-gap radius. Therefore, this temperature difference between the thermistors is reflected in the actual effective area of the apparatus. The amount by which this area differs from the mid-gap area is properly quantified on the basis of the inequality (instead of the nominal values) of the epoxy resin thicknesses.

As noted above, despite the considerable care that was taken in the assembly of the apparatus, the thickness of the layer of epoxy resin at the metering piece is greater than the thickness of the layer of epoxy resin at the guard by 0.01 ± 0.01 mm - i.e. the non-uniformity in the epoxy resin thicknesses is between 0 and 0.02 mm. Thus, at the nominal null power, the temperature of the metering piece thermistor will be, respectively, equal to or greater than that of the guard thermistor. Consequently, as explained above in Section 2.2.3 with reference to Figs. 2.6a and c, the effective area will be equal to or less than the area defined by the mid-gap radius. Modelling results show that an extra 0.02 mm of epoxy resin at the base of the metering piece results in an effective area that is 5 % less than that bounded by the mid-gap radius. Therefore, it is expected that, for the 0.01 ± 0.01 mm non-uniformity in the epoxy resin thickness, the effective area will differ from the mid-gap area by between 0 and -5 %.

The significance of the inequality of the hot side epoxy resin thicknesses is summarized as follows. The optimum thermistor locations were determined from modelling results on the basis of an ideal effective area defined by the mid-gap radius, and without simulating the epoxy resin layers. These layers were not considered because, at the time of this analysis, the thicknesses of the layers were unknown. It was later determined that the thickness of the epoxy resin below the metering piece was 0.01 ± 0.01 mm greater than of the epoxy resin below the guard. As previously explained, this leads to an effective area which is less than or equal to the mid-gap area. This uncertainty in the effective area need not result in a corresponding error in the measurements. In order to avoid such an error, an appropriate procedure is required to calibrate the effective area of the apparatus. The above analysis shows that a calibration of this area on the basis of the geometry of the device (i.e. by assuming that it is equal to the mid-gap area) will result in quite a large error: 0 to -5 %. It is therefore preferable to perform the calibration on the basis of accurate measurements made on a sample which has a well known thermal conductance. Such a calibration procedure is described below, and it gives a significantly more accurate estimate of A_{eff} . This calibration procedure essentially provides a correction for the effect of the unequal epoxy resin thicknesses.

Another issue which can influence the thermistor temperatures is the non-uniformity of the temperature in the hot plate. As water passes through the water jacket of the hot plate of the apparatus, there is a temperature non-uniformity associated with the heat loss from the water flow. Therefore, neither the water nor the wetted surface of the plate is actually at a uniform and well defined temperature. In fact, under the normal operation of the apparatus, the difference in temperature between the water inlet and outlet flows can be as high as a few hundredths of a kelvin. This temperature difference is about an order of magnitude greater than the desired maximum temperature difference between the thermistor locations. Of course the temperature non-uniformities at locations remote from the water (such as the thermistor positions) can be expected to be less than the non-uniformities at and near the wetted surface. However, in order to minimize the effect of any such non-uniformities on the measurements, it is important to ensure that the thermistors are located at positions where the temperatures are well known - or where the temperatures can be most easily estimated. Since the temperature of the thermistor in the thermally well isolated metering piece is relatively independent of the local water temperature, the exact location of this sensor with respect to the water flow is not crucial. However, this is not the case for the guard thermistor because the guard is in direct contact with the water flow. It was decided to locate this thermistor along a radius which is perpendicular to the diameter along which the hot water inlet and outlet ports are located (i.e. along the mid-flow diameter). Since the guard thermistor is essentially below the mid-flow, the temperature at this location is assumed to be equal to the average of the measured water inlet and outlet temperatures. The exact temperature could of course be determined from a model which simulates the heat transfer from the water to the plate. However, the above approximate approach is sufficiently accurate.

It is possible that heat dissipation associated with the viscous flow of the water in the hot plate could affect the local temperatures of the thermistors. If the TCA is operated with both the hot and cold plates at room temperature, and with a low thermal conductance sample between these plates, then there should be no external heat flow to or from the hot plate water. Under these conditions, the heat generated as a result of the viscous flow can be expected to cause the hot plate water temperature to be greater at the outlet than at the inlet to the plate. However, careful measurements for these conditions did not reveal any such increase in the temperature of the water flow (to within the 0.02 °C resolution of the temperature measurement). Therefore, the effect of viscous flow heat dissipation on the thermistor temperatures is assumed to be negligible.

2.2.6 Temperature Differences Across The Apparatus

In order to calculate accurately the thermal conductance across the evacuated space, the temperature difference between the internal glass surfaces, ΔT_{int} , must be well known. In the TCA, the only temperature difference *across* the apparatus that is known is that between the two circulating water flows, ΔT_{total} . This temperature difference is based on the average of the measured inlet and outlet water temperatures for each plate. Due to the temperature differences across the thermal resistances between the water flows and the internal glass surfaces, ΔT_{int} is slightly less than ΔT_{total} . If ΔT_{int} is to be well defined, then the apparatus must be designed and constructed to minimize these intervening thermal resistances.

The thermal resistances between the water flow and the internal glass surface for each side of the apparatus are given in Table 2.1. These resistances correspond to the sections shown in Fig. 2.1, and also to the epoxy resin layers which are not shown in this figure. The thickness for the hot side copper plate corresponds to the height of the metering piece, and the thickness for the hot side epoxy resin corresponds to the epoxy resin layer below the metering piece. The resistances are based on the indicated thicknesses, thermal conductivities, and the *approximate* area of each wetted surface ($7.85 \times 10^{-3} \text{ m}^2$ for the plate diameter of 100 mm). The exact choice for the area is not important since it is factored out in the final result of this analysis (it is only used in the analysis for the sake of convenience). The thicknesses for the copper hot and cold plates are measured between the measurement and wetted surfaces of the plates; the thickness for the hot plate corresponds to that of the guard. The heat transfer coefficient for the wetted surfaces of a similar apparatus was experimentally determined to be about $5000 \text{ Wm}^{-2}\text{K}^{-1}$ [3]. For the above wetted surface area, this corresponds to a thermal resistance of $2.55 \times 10^{-2} \text{ KW}^{-1}$. This is the value that is used for the water-to-plate thermal resistance in the following analysis. It is possible to obtain a gross estimate for the above heat transfer coefficient by treating the flow over each wetted surface as a flow over an isothermal square plate of equal area. For this case, the heat transfer coefficient can be estimated from the appropriate equations given by Holman for the flow over an isothermal plate [9]. Based on a flow rate of 20 liters per minute and on a mean water temperature of 23°C , the estimated heat transfer coefficient is $3600 \text{ Wm}^{-2}\text{K}^{-1}$. This estimate provides an order of magnitude confirmation of the more reliable experimental result given above.

Since all of the thermal resistances of Table 2.1 are added in series, the three largest ones (those for the glass and the two epoxy resin layers) have the most significant effect on the total thermal resistance - and thus on the difference between ΔT_{int} and

ΔT_{total} . This is another reason (although a less important one) for minimizing the epoxy resin thicknesses. Copper was used for the plates because, relative to the other thermal resistances of the system, it presents a small thermal resistance. The thickness of the contact gel is estimated from the volume of gel applied at the measurement area ($\sim 0.9 \text{ cm}^3$). This gel is water based and thus it is assumed to have the same thermal conductivity as water for the expected range of plate temperatures: $0.6 \text{ Wm}^{-1}\text{K}^{-1}$ for 3 to 23 °C [10]. It is noted that, since the thermal resistance of the gel is much less than the three largest (and thus the most significant) thermal resistances of the system, the total system resistance is not strongly dependent on the exact value which is chosen for the gel thermal conductivity.

Thermal resistance section	Thickness (mm)	Thermal conductivity ($\text{Wm}^{-1}\text{K}^{-1}$)	Thermal resistance (KW^{-1})
Hot side water-to-plate			0.026
Hot side copper plate	12.0	386	0.004
Hot side epoxy resin	0.110	0.20	0.070
Hot side contact gel	0.115	0.6	0.024
Hot side glass	6.00	1.00	0.764
Cold side glass	6.00	1.00	0.764
Cold side contact gel	0.115	0.6	0.024
Cold side epoxy resin	0.090	0.20	0.057
Cold side copper plate	11.0	386	0.004
Cold side water-to-plate			0.026

Table 2.1: The thermal resistances between the water flow and the internal glass surface for each side of the apparatus. The thermal resistances are based on the indicated thicknesses and thermal conductivities, and on the approximate area for each wetted surface ($7.85 \times 10^{-3} \text{ m}^2$). The water-to-plate thermal resistances are based on experimentally determined heat transfer coefficients of $5000 \text{ Wm}^{-2}\text{K}^{-1}$ [3].

The sum of the thermal resistances between the water flows and the internal glass surfaces is 1.763 KW^{-1} which, based on the above total wetted area, corresponds to a thermal conductance of $C_{w-i} = 72.26 \text{ Wm}^{-2}\text{K}^{-1}$. The thermal conductance of the entire apparatus is equal to C_{w-i} in series with C_{sample} . Since conductances in series are added as inverses, and since C_{sample} is typically much less than C_{w-i} , the total thermal conductance is approximately equal to C_{sample} . The heat flow through the total conductance (across which the temperature difference is ΔT_{total}) is equal to the heat flow through C_{w-i} (across which the temperature difference is ΔT_{w-i}). Thus:

$$\Delta T_{w-i} = \frac{C_{sample} \Delta T_{total}}{C_{w-i}} \quad (2.13)$$

$$= 0.0138 C_{sample} \Delta T_{total} .$$

It is noted that this is an area independent result - i.e. it does not depend on the wetted surface area that was assumed in the above analysis.

The temperature difference between the two internal glass surfaces is equal to the total temperature difference across the apparatus minus the temperature differences between the water flows and the internal surfaces, i.e. $\Delta T_{int} = \Delta T_{total} - \Delta T_{w-i}$. Therefore, based on equation 2.13:

$$\Delta T_{int} = \Delta T_{total} (1 - 0.0138 C_{sample}) . \quad (2.14)$$

It is noted that this equation is based on the height of the metering piece and on the thickness of the of the epoxy resin layer below the metering piece. Therefore, this equation gives the temperature difference between the internal glass surfaces at the measurement area below the metering piece. Since, at the null condition, ΔT_{int} should be approximately the same below the metering piece and the guard, equation 2.14 is assumed to apply to the latter location as well. This analytic result is approximately consistent with modelling results. The finite element model described above in Section 2.2.3 was used to determine the difference in the average temperatures of the internal glass surfaces below the metering piece as a function of ΔT_{total} and C_{sample} . It is recalled that the epoxy resin layers are simulated in this model. It was found that:

$$\Delta T_{int} = \Delta T_{total} (1 - 0.0129 C_{sample}) . \quad (2.15)$$

The discrepancy between equations 2.14 and 2.15 is probably due to the fact that the finite element model (on which the latter equation is based) simulates the complex heat flows - and thus the non-uniform temperatures - in the glass below the metering piece due to the presence of the gap. It therefore gives a more appropriate result for the temperatures across the evacuated space at the measurement area. The fact that the finite element modelling approach does not simulate the water-to-plate thermal resistances does not fully account for the discrepancy. For the accuracy to which the internal temperature difference must be known, the discrepancy between these equations is insignificant. It was therefore decided to use equation 2.15 to calculate ΔT_{int} .

2.2.7 Thermal Termination of The Wires From The Power Resistor

When power is applied to the resistor in the metering piece, the ceramic core of the resistor is at a higher temperature than the metering piece. This is because of the thermal insulating effect of the plastic and the epoxy resin between the core and the metering piece (these structures are shown in Fig. 2.13a). Since, at the null condition, the temperatures of the metering piece and the guard are equal ($T_m=T_g$), the resistor core is also warmer than the guard. This temperature difference can result in heat flows along the four wires from the core to the guard (the heat flows actually leave the core via the resistor leads to which the wires are connected). In this case, not all of the power dissipated in the resistor will actually flow into the metering piece and across the sample and thus the null power will be overestimated. The following analysis shows that the heat loss along the wires can be significant unless the wires are thermally terminated to the metering piece.

There are two thermal resistances associated with the resistor core: that between the resistor and the surrounding metering piece, R_{r-m} , and that between the resistor and the guard via the wires, R_{r-g} . The equivalent thermal resistance circuit is shown in Fig. 2.13b. At the null condition, $T_m=T_g$ and thus these thermal resistances are in parallel. Therefore, the ratio of the heat dissipated in the resistor core that flows to the guard along the wires, to that which flows to the metering piece is:

$$\frac{Q_{guard}}{Q_{metering\ piece}} = \frac{R_{r-m}}{R_{r-m} + R_{r-g}} \quad (2.16)$$

In estimating the thermal resistance between the resistor and the metering piece, R_{r-m} , the resistor core can be considered to be a long insulated isothermal cylinder if it is assumed that there is negligible heat flow along the length of the resistor; the cross-section for such a cylinder is shown in Fig. 2.13c. For this situation, the plastic coating and epoxy resin around the core constitute the insulation. The form of the equation for the thermal resistance presented by a layer of insulation around an isothermal cylinder was previously given in equation 2.2. For the present analysis, L_c of this equation is the length of the resistor core (4 mm) and r_i and r_o are the outer radii of the core and the insulation layer (0.05 and 0.10 mm, respectively). The exact value for the thermal conductivity of the plastic is not crucial to this analysis and thus it assumed to be equal to the conductivity which was previously given for the epoxy resin: $0.20 \text{ Wm}^{-1}\text{K}^{-1}$. Therefore, the thermal conductivity for the entire insulation layer is $k_{ins}=0.20 \text{ Wm}^{-1}\text{K}^{-1}$. Substitution of these values into equation 2.2 for the appropriate corresponding quantities gives a thermal resistance of $R_{r-m}=138 \text{ KW}^{-1}$.

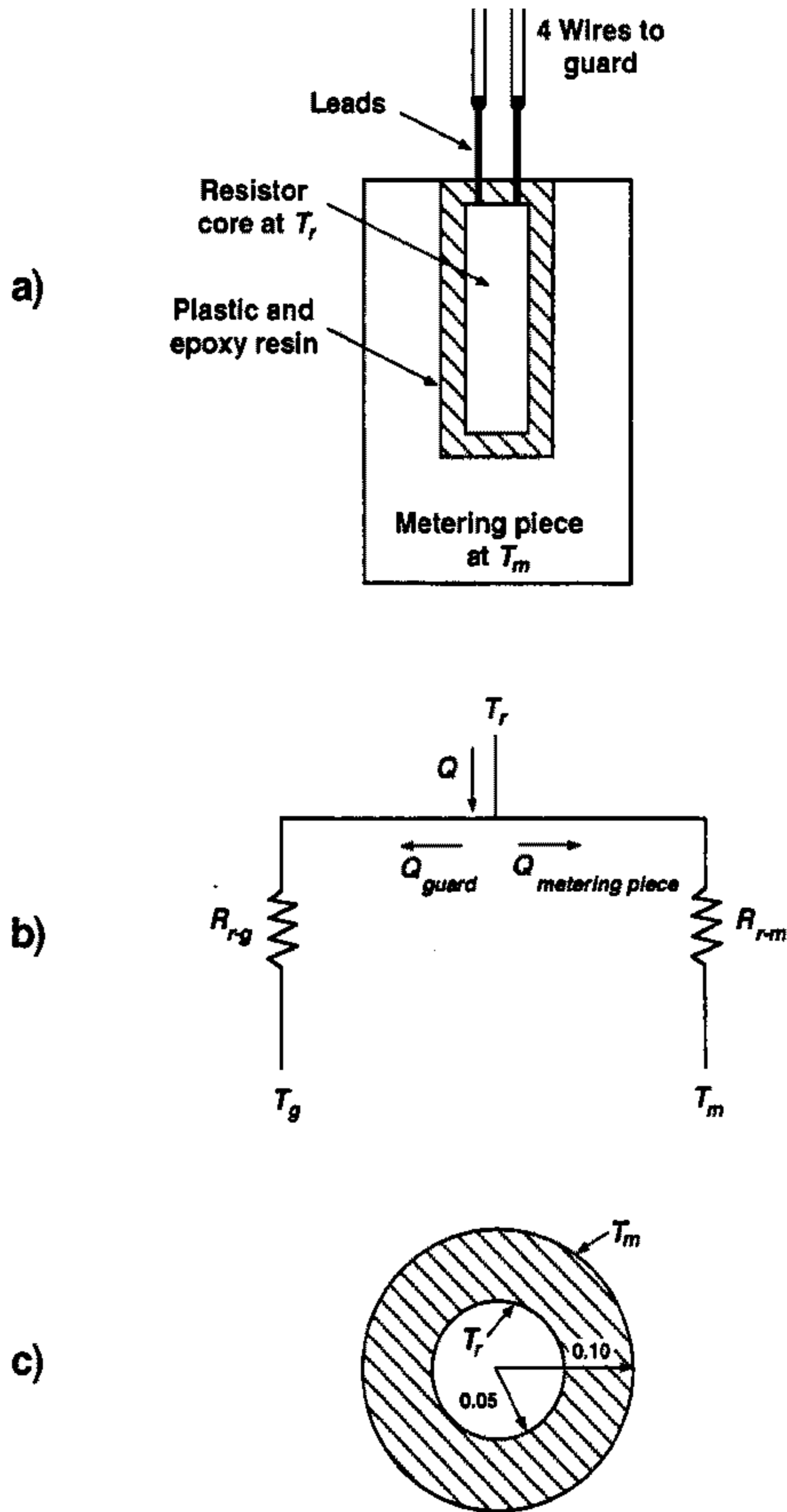


Fig. 2.13:

(a) At the null condition, when the temperatures of the metering piece and the guard are equal, the resistor core is warmer than the metering piece. This is because of the insulating effect of the plastic and epoxy resin which surrounds the core. Therefore, at the null condition, heat can flow along the four wires from the core to the guard (the heat flow actually leaves the core via the resistor leads to which the wires are connected).

(b) The thermal resistance circuit for the heat flow from the resistor core. The ratio of the heat flow from the resistor core that passes to the guard via the resistor wires to that which passes to the metering piece through the intervening plastic and epoxy resin can be calculated from the corresponding thermal resistances of R_{r-g} and R_{r-m} . The heat added to the core is represented as Q .

(c) The thermal resistance between the resistor core and the metering piece can be estimated by modelling the encapsulated resistor as a long insulated cylinder. The plastic and the epoxy resin are considered to be the insulation. The radial dimensions for the resistor core and the surrounding insulation are in millimeters. The surface temperatures are as indicated.

The total thermal resistance for all of the six wires between the metering piece and the guard (including the two wires which are connected to the thermistor) was estimated in Section 2.2.3 to be $R_{m-g}(\text{wires})=1100 \text{ KW}^{-1}$. The wire thermal resistance of interest in the present analysis, R_{r-g} , is the equivalent resistance for just the four wires which are connected to the resistor, and it is equal to 1650 KW^{-1} .

Substitution of the above values for R_{r-m} and R_{r-g} into equation 2.16 shows that the heat lost from the resistor to the guard via the wires is about 8 % of the heat which flows from the resistor to the metering piece - and across the sample. Since this heat loss will result in a significant overestimation of the null power, the apparatus must be designed to minimize the heat flow along the wires. This can be easily accomplished by putting the wires into good thermal contact with the metering piece by wrapping them around the piece; this contact must occur between the separate connections to the resistor and the guard. It is necessary to determine what length of wire in contact with the metering piece will sufficiently minimize the above heat loss.

Consider a length of wire from the resistor which is wrapped around the metering piece before being brought out to the guard. The heat flow along the portion of this wire which is in contact with the metering piece is analogous to that along a long thin fin in which the temperature of one end is fixed (at T_r in this case) and the temperature of the other end is approximately equal to that of the surroundings (T_m in this case). For the wire, this last temperature condition will be satisfied if there is a sufficient contact length between the wire and the metering piece. Whereas a conventional fin loses heat to ambient, the wire loses most of its heat to the metering piece through the teflon coating, and through the epoxy resin with which it is affixed to the piece. There is also some heat loss from the wire to ambient but, since this is much less than the conduction to the metering piece, it is neglected. The *general* solution for the temperature profile, $T(x)$, along a fin which has a temperature of T_o at one end (which is at $x=0$) is given by:

$$T(x) - T_o = Ae^{-x/\ell} + Be^{x/\ell}, \quad (2.17)$$

where ℓ is the characteristic distance. For the case of the wire "fin" described above, $T(x)$ is the temperature of the wire core at a distance x from the resistor core, and T_o is the temperature of the metering piece (i.e. $T_o=T_m$). Since, at large x , $T(x) \approx T_m$, the coefficient B is zero. The coefficient A is of no interest in the present analysis. What is of interest is the characteristic distance. For the wire, this distance is:

$$l = \sqrt{\frac{k_{wire} A_{wire}}{C_{lin}}}, \quad (2.18)$$

where k_{wire} is the thermal conductivity of the wire, A_{wire} is the cross-sectional area of the wire core, and C_{lin} is the thermal conductance per unit length between the wire core and the metering piece. The value for C_{lin} was determined from a two-dimensional finite element model which was constructed with the STRAND6 modelling package. As shown in Fig. 2.14, the model consists of surfaces representing the outer surfaces of the metering piece and the 0.10 mm diameter core of the wire. Between these surfaces are the teflon insulation of the wire (of thickness 0.05 mm) and a layer of the epoxy resin (which, in the actual apparatus, bonds the wire to the metering piece). The epoxy resin layer has an idealized shape and a minimum thickness of 0.05 mm. The thermal conductivity of the teflon is assumed to be equal to that of the epoxy resin: $0.20 \text{ Wm}^{-1}\text{K}^{-1}$. It is assumed that there are no non-uniformities in the surface temperatures of the wire core and the metering piece. The temperature of the wire core surface is set to 1°C , and the temperature of the metering piece surface is set to 0°C . Based on the calculated total heat flow at the interface between the epoxy resin and the surface of the metering piece, the lineal conductance for the heat flow between the wire core and the metering piece was determined to be $C_{lin}=0.84 \text{ Wm}^{-1}\text{K}^{-1}$.

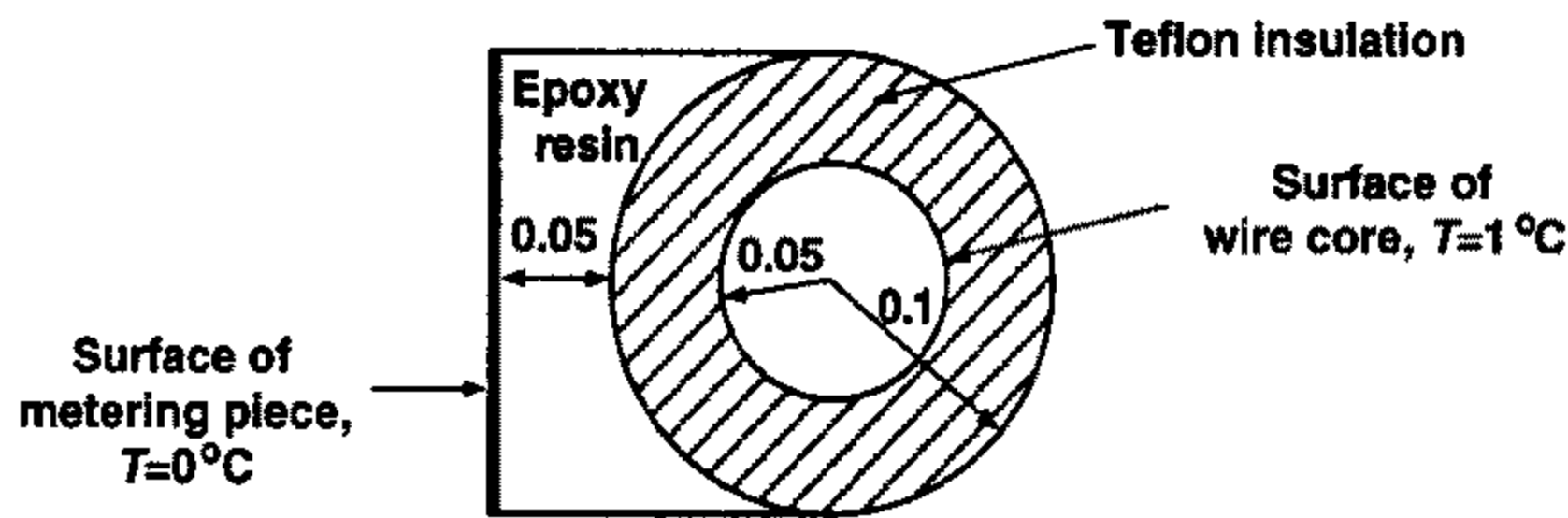


Fig. 2.14: The finite element model used to determine the lineal conductance for the heat transfer between the core of an insulated copper wire and the metering piece. The surface of the core is at 1°C , and the surface of the metering piece is at 0°C . The lineal conductance is based on the total heat flow calculated at the interface between the epoxy resin and the surface of the metering piece. The dimensions are in millimeters.

Substituting into equation 2.18 the above value for C_{lin} , along with $k_{wire}=386 \text{ Wm}^{-1}\text{K}^{-1}$ for the copper wire and $A_{wire}=7.85 \times 10^{-9} \text{ m}^2$ (based on the core radius of 0.05 mm) gives a characteristic distance of 1.9 mm. If the wire is wrapped once around the cylindrical surface at the top of the metering piece, where the diameter is 6 mm, then the contact length is 19 mm. This means that in one loop of wire around the metering piece, there are 10 characteristic distances for the temperature of the wire core to

approach that of the metering piece. Therefore, at $x=19$ mm, the difference between the wire and metering piece temperatures is e^{-10} , or 0.005 %, of the difference at the resistor core. Clearly, a single loop of wire around the top of the metering piece will ensure that there is a negligible heat flow along the wire from the resistor core to the guard. Thus, during the construction of the apparatus, all four resistor wires were wrapped once around the top of the metering piece and set in place with epoxy resin before being connected to the resistor and the guard.

At the guard, all of the wires from the metering piece were wrapped around a ridge of metal which is 24 mm in diameter before being connected to the electronic circuitry. This ensures that the guard ends of the wires are indeed at the guard temperature, and that there are no significant heat flows to the metering piece from the circuitry (which, due to ohmic heating, might be at a larger temperature than T_m).

2.2.8 Power Dissipated in The Metering Piece Thermistor

The power dissipated in the metering piece thermistor is approximately equal to the square of one half the bridge voltage divided by the nominal resistance of the thermistor: $(3\text{ V})^2/200\text{ k}\Omega$, or $45\text{ }\mu\text{W}$. It is assumed that the percentage of this power that is lost along the thermistor wires to the guard is *approximately* the same as the above result for the similar heat flow along the wires from the resistor core to the guard: 8 %. Thus, about $4\text{ }\mu\text{W}$ of the thermistor power can be expected to be lost as a heat flow to the guard via the thermistor wires. Since this heat loss is less than the smallest expected absolute error in the power measurements ($\sim \pm 10\text{ mW}$), there is no need to prevent this loss by thermally terminating the thermistor wires to the metering piece. It is important to determine if the rest of the thermistor power, which flows into the metering piece and is larger than the above error, will affect the measurements.

Since most of the $45\text{ }\mu\text{W}$ which is dissipated in the thermistor flows into the metering piece, it might appear that this power should be added to the null power when estimating the sample conductance. However, since this power is present when both the zero bridge voltage and the null power are determined, and since the null power is referenced to this voltage, the thermistor power simply represents a constant offset that need not be considered - provided that the following two conditions are satisfied.

The first condition is that the thermistor power must be constant. The maximum expected temperature change for the metering piece thermistor is about 0.4 K. Based on the previously given value for $\Delta R/R_{th}$ of 0.04 K^{-1} , this results in a 2 % change in the

thermistor resistance. Since it is only of interest to determine the null condition when the bridge voltages are identical to those which existed when the zero of the bridge was determined, the voltage across the thermistor can be assumed to be essentially unchanged. Therefore, the thermistor power will only be affected by the 2 % change in the thermistor resistance - for which the corresponding change in the thermistor power is about 1 μ W. Since this is less than the power resolution of the apparatus (which is given below as $\pm 10 \mu$ W) it can be assumed that the thermistor power is constant.

The second condition is that any effect of the thermistor power on the temperature difference between the metering piece and the guard, $T_m - T_g$, must be independent of changes in the thermal resistance of the sample. The second term of equation 2.4 gives the change in $T_m - T_g$ as a result of an amount of power, Q , which is dissipated in the metering piece. For the present analysis, Q is set equal to the total power dissipated in the thermistor minus the amount of this power which is lost along the wires, i.e. $Q = 41 \mu$ W. The second term of equation 2.4 shows that the contribution of this power to $T_m - T_g$ can change due to changes in R_{m-g} or R_{m-c} . The former thermal resistance is effectively constant, but the latter one is not. This is because R_{m-c} is approximately equal to the thermal resistance of the sample and, for samples in which the heat transfer is predominantly radiative, this resistance changes during a test if the temperature difference across the sample is varied. However, for the expected range of temperatures for the TCA measurements, the sample resistance is expected to change by no more than 25 % for each test. It is recalled from Section 2.2.3 that R_{m-g} is predicted to be 29.6 KW^{-1} , and that the smallest expected value for R_{m-c} is about 1300 KW^{-1} . The second term of equation 2.4 shows that, for the above thermal resistances, and for $Q = 41 \mu$ W, a 200 % increase in the sample resistance (and thus in R_{m-c}), will change the thermistor power contribution to $T_m - T_g$ by only 13 μ K. Therefore, it can be assumed that any effect of the thermistor power on $T_m - T_g$ will be essentially independent of the expected changes in the sample thermal resistance.

2.2.9 Parasitic Heat Flows

In theory, if a glazing sample has a zero thermal conductance, the null power must be zero - unless there are parasitic heat flows from the metering piece. The most likely potential parasitic heat flows are a heat flow from the metering piece along the wires to the guard or to ambient (which should be negligible due to the proper thermal termination of the wires), and a lateral heat flow along the hot side glass sheet (which, for the dimensions of the TCA, should also be negligible - as discussed above in Section 2.2.2). One can check for the presence of parasitic heat flows by determining if

the null power is in fact zero for a sample with a zero thermal conductance. Of course it is not possible for a sample to have an exactly zero conductance. However, a near-zero conductance is possible if the pillars are remote from the measurement area, and if the internal glass surfaces are coated with a material which has a very low hemispherical total emittance - e.g. gold. A parasitic heat flow test was performed on an apparatus which is similar to the TCA in dimensions and design [3]. For this test, measurements were made of the radiative heat flow across the evacuated space between two sheets of gold coated glass that were mounted in the standard sample apparatus that is described below. The pillars between the sheets were far enough away from the measurement area that the conduction through the pillars made a negligible contribution to the measured heat flows. Measurements of the small amount of heat flow across this sample were, to within 30 μW , in agreement with the predicted radiative heat transfer based on the optically measured surface emittances. It was therefore concluded that, for the particular apparatus used for this test, the parasitic heat flows were negligible. The gold coated sheets of glass were unavailable for a similar test with the TCA. However, since parasitic heat flows were shown to be negligible in the similar apparatus referred to above, and since modelling results indicate that these heat flows should be negligible in the present apparatus, it is assumed that there are no significant parasitic heat flows in the TCA.

2.3 Measurement Procedure

In order to measure the thermal conductance of a sample, the TCA and the sample are assembled as follows. The hot plate is firmly seated faceplate upwards in a cup which is attached to a tabletop. About 0.9 cm³ of a water based gel is dispensed from a syringe onto the center of the faceplate (this is all that is required). The sample is then lowered onto the hot plate, and the gel is allowed to spread out evenly. The sample area to be measured is located over the metering piece and then the sample is fixed in position by four holders which are slid into position along tracks and locked in place. Large samples are supported from below by long adjustable screws. Another 0.9 cm³ of gel is applied to the top surface of the glazing above the measurement area. The cold plate is lowered onto the sample at this location, once again allowing for an even distribution of the gel.

Water is circulated through each plate at the desired temperature. For each plate, the inlet and outlet water flow temperatures are measured with a Leeds and Northrup

8931 general purpose platinum resistance thermometer. This is done by lowering the thermometer probe into water-filled cavities located in the water flow pipes near the inlet and outlet ports of the apparatus. The outlet temperatures are measured before the inlet temperatures. As previously noted, the hot water is maintained at the desired hot plate operating temperature of 23 °C. The water flow temperature for each plate is taken to be equal to the average of the respective inlet and outlet temperatures.

In order to determine the null power, it is necessary to know the amplified bridge voltage at the null condition - i.e. the zero bridge voltage, or V_{bo} . At the null condition, no heat flows between the metering piece and the guard. This can be established by supplying water of equal temperature to both plates in order to ensure that there is no temperature difference across the sample. With no such temperature difference, there is no inter-plate heat flow, and thus no heat flow - or temperature difference - between the metering piece and the guard. The bridge voltage for this situation therefore corresponds to the zero bridge voltage. During the assembly of the apparatus, the bridge resistances were adjusted so that V_{bo} would be close to zero volts. This was done in order to minimize the effects due to any variations in the voltage supplied to the bridge, and also for the sake of convenience.

It is of course difficult to obtain precisely equal water flow temperatures in the hot and cold plates - usually, these temperatures can only be adjusted to within a few hundredths of a degree of each other. Therefore, V_{bo} is determined by interpolating between bridge voltages for small positive and negative temperature differences between the two water flows - as shown in Fig. 2.15a. Due to the small temperature differences across the system, the temperature difference between the flows in the hot and cold plates can be assumed to be equal to the temperature difference between the guard and the cold plate, $T_g - T_c$. The hot water temperature is maintained at 23 °C during this procedure. For a vacuum glazing sample, the zero bridge voltage is determined at a location which is remote from the edges and pillars of the sample. Once V_{bo} has been established, power measurements are performed.

A power measurement is made as follows. Power is added to the metering piece resistor, and this causes a rapid change in the bridge voltage. Once the computer processed bridge voltage has stabilized to within the usual variation (less than a millivolt), the data logger is reset. When the running average of the above bridge voltage has stabilized, it is recorded along with the resistor voltage and current which are read from the two digital multimeters. The power dissipated in the resistor is the product of the current and the voltage. The water flow temperatures are also recorded.

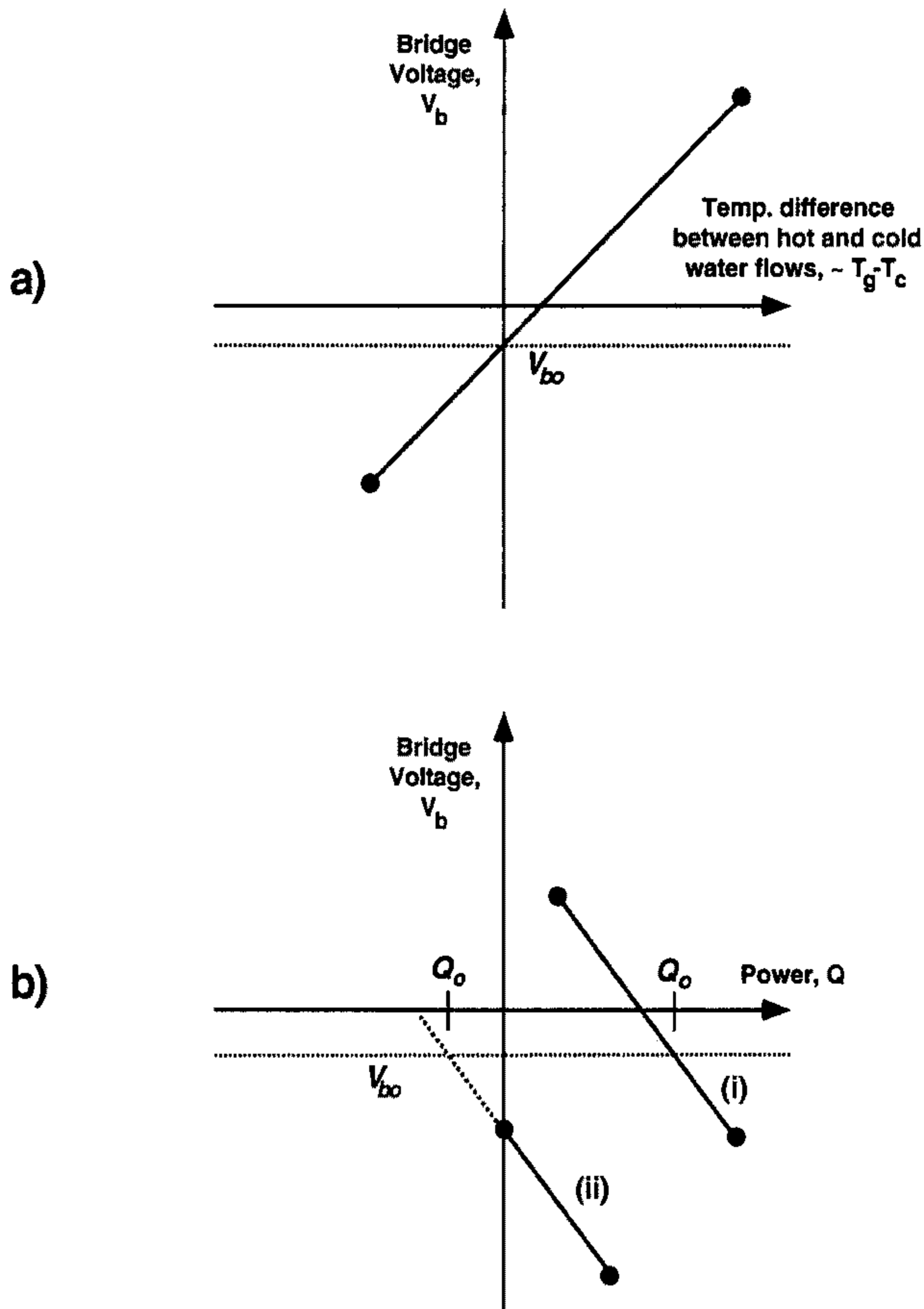


Fig. 2.15:

(a) The amplified zero bridge voltage, V_{bo} , is determined by interpolating between the bridge voltages measured at small positive and negative temperature differences between the water flows in the hot and cold plates. Due to the small temperature differences across the system, the temperature difference between these flows can be assumed to be equal to the temperature difference between the guard and the cold plate, i.e. $T_g - T_c$.

(b) The null power, Q_o , is usually determined as with case (i) by interpolating between two bridge voltages (which are measured at two different resistor powers) to the zero bridge voltage, V_{bo} . If the apparatus is operated in the reverse mode, with the hot plate colder than the cold plate, then an extrapolation is performed as with case (ii). The x-axis of this figure represents the difference between the temperature of the hot side internal glass surface, T_1 , and the temperature of the cold side internal glass surface, T_2 (i.e. $T_1 - T_2$).

If the hot plate is actually hotter than the cold plate, then the null power can be found by an interpolation between the bridge voltages for two such power measurements on either side of the zero bridge voltage, V_{bo} . However, if the hot plate is colder than the cold plate (in what was previously referred to as the reverse mode of operation), then the null power is determined by an extrapolation from the bridge voltages for two power measurements below V_{bo} . Both of these methods are shown in Fig. 2.15b. The x-axis of this figure represents the difference between the temperature of the hot side internal glass surface, T_1 , and the temperature of the cold side internal glass surface, T_2 (i.e. $T_1 - T_2$).

2.4 Calibration and Accuracy of The Apparatus

In order to calculate accurately the thermal conductance of a sample, the effective area must be well known. As explained above, it is preferable to calibrate this area on the basis of a TCA measurement of the heat flow through a reference standard which has a well known thermal conductance - and which is subjected to a well defined temperature difference. The standard selected for this calibration procedure is two plane parallel uncoated glass surfaces separated by an evacuated space. This was selected for the calibration because the heat transfer between the glass surfaces, which is just radiative due to the vacuum, can be accurately predicted by the exact method presented in Chapter 3. Although vacuum glazing essentially has the same structure as this standard, an actual glazing sample would be unsuitable for an accurate calibration because it is not possible to measure directly the internal pressure of the sample. Such a measurement is required for a check of the quality of the internal vacuum in order to ensure that the heat transfer between the glass sheets is entirely - or almost entirely - radiative. Therefore, a simulated glazing sample for which the internal pressure can be measured was constructed with a continuously pumped standard sample apparatus.

Figure 2.16 shows the configuration of the standard sample apparatus. Two sheets of 4 mm thick uncoated soda-lime glass are separated by a 110 mm diameter ring of twelve pillars and clamped between two stainless steel collars (the clamps are not shown in this figure). The pillars are approximately 0.2 mm in height. The space between the glass sheets and the collars is evacuated via a port to an oil diffusion vacuum pump, and seals are maintained with O-rings. The internal vacuum is monitored with an ion gauge which is located at a port which is remote from the connection to the vacuum system. This assembly is positioned between the hot and

cold plates of the apparatus, with the measurement area located below the center of the ring of pillars.

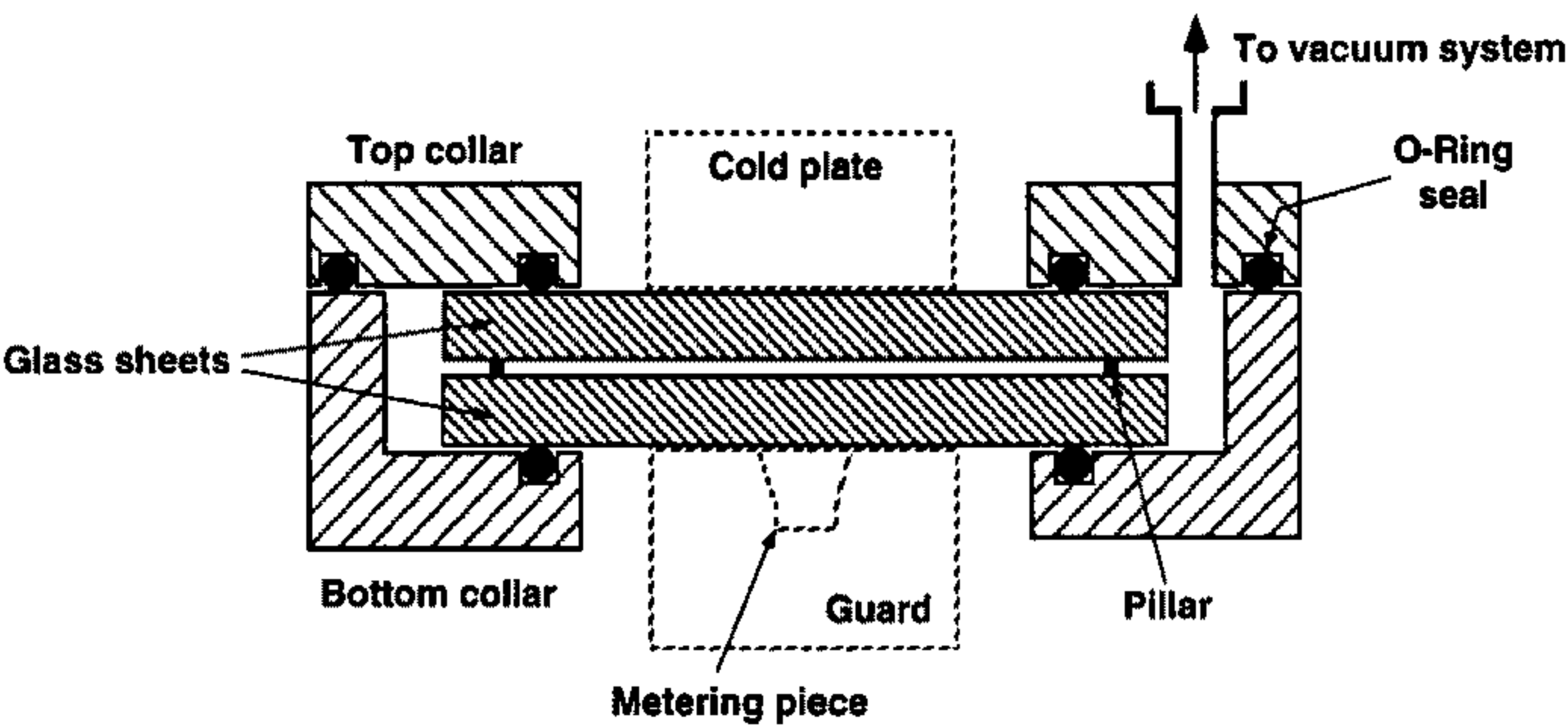


Fig. 2.16: The standard sample apparatus. Two sheets of glass are clamped between two stainless steel collars and separated by a ring of pillars situated outside the diameter of the guard. The entire space between the collars is evacuated, with seals being maintained by O-rings. The apparatus is mounted between the plates of the TCA, which are represented by the dashed lines. The clamps for the collars are not shown.

Since the edges of the glass sheets are situated in a vacuum, there is no need for an edge seal and thus there are no direct edge losses with this apparatus. There is of course some indirect heat flow between the glass sheets via the collars. However, for two reasons, it is assumed that the effect of this heat flow on the measurements is negligible. Firstly, since the collars are actually separated from the sheets - and from each other - by the relatively high thermal resistances of the rubber O-rings, this indirect heat flow is small. Secondly since the pathway for this small heat flow is located beyond the diameter of the guard, the heat flow will not lead to a significant lateral heat flow from the metering piece - as explained at the beginning of Section 2.2.2. Losses through the support pillars are similarly minimized by locating the pillars outside the diameter of the guard; results presented below show that the conduction through pillars at this distance from the central metering piece makes a negligible contribution to the total measured heat flow. Therefore, it is assumed that the heat flow measured at the metering piece is due to just the radiative heat transfer between the two internal glass surfaces - plus a much smaller conductive heat transfer through the residual gas in the vacuum (which is explained below).

The radiative heat transfer between two plane parallel surfaces (e.g. the two internal glass surfaces considered for this calibration procedure) of area A and at temperatures T_1 and T_2 can be written as:

$$Q_{rad} = \epsilon_{eff} \sigma A (T_1^4 - T_2^4) , \quad (2.19)$$

where ϵ_{eff} is the effective emittance of the two surfaces (explained in Chapter 3) and σ is the Stefan-Boltzmann constant ($5.67 \times 10^{-8} \text{ Wm}^{-2}\text{K}^4$). If T_1 and T_2 are not too different, this relationship can be linearized by a Taylor series expansion as:

$$Q_{rad} = C_{rad} A \Delta T , \quad (2.20)$$

where the radiative conductance is:

$$C_{rad} = 4\epsilon_{eff} \sigma T_{ave}^3 , \quad (2.21)$$

and where T_{ave} is the average of T_1 and T_2 . It is recalled that the situation of interest is the radiative heat transfer between the surfaces of the glass sheets which are separated by a vacuum. Although glass is quite transparent in the visible region of the spectrum, it is essentially opaque in the infrared where radiative heat transfer occurs. It is therefore assumed that there is a negligible heat transfer across the evacuated space from any processes except radiative heat transfer between the internal surfaces of the glass sheets. Thus, the per unit area heat transfer across the space is effectively only dependent on the temperatures of the inner surfaces, and on the effective emittance (which, as explained in Chapter 3, is not significantly influenced by the surface temperatures for the conditions considered in this thesis).

A standard sample of two sheets of uncoated glass was mounted between the plates of the TCA and it was evacuated to a pressure of $3 \times 10^{-3} \text{ Pa}$. For water temperatures of 23.19 and 3.26 °C ($\Delta T_{total} = 19.93 \text{ °C}$) the measured null power, Q_o , was 10.91 mW. The effective area for the apparatus can be calculated according to the following expression:

$$A_{eff} = \frac{Q_o}{C_{sample} \Delta T_{int}} . \quad (2.22)$$

There are two possible methods for determining A_{eff} from equation 2.22. The first method involves solving for the temperature difference between the internal glass

surfaces, ΔT_{int} , by equation 2.15. However, since this equation requires a knowledge of C_{sample} , which itself is a function of ΔT_{int} , an exact solution for ΔT_{int} is somewhat complicated. The second method involves a simple iterative approach which gives an equivalent result for the effective area of the apparatus. This second method is presented below.

The first step in solving for the effective area is to determine ΔT_{int} . The heat flow through the apparatus is known to be 10.91 mW, and the thermal conductance between the water flows and internal surfaces was calculated in Section 2.2.6 to be $C_{w-i}=72.26 \text{ Wm}^{-2}\text{K}^{-1}$. In order to determine the temperature difference across this conductance, ΔT_{w-i} , the effective area is at first assumed to be equal to that defined by the mid-gap radius ($A_{eff}=1.539 \times 10^{-4} \text{ m}^2$). Thus:

$$\Delta T_{w-i} = \frac{Q_o}{C_{w-i} A_{eff}}, \quad (2.23)$$

which, for $Q_o=10.91 \text{ mW}$, gives $\Delta T_{int}=0.98 \text{ }^\circ\text{C}$. Assuming that the temperature difference between the water flow and the internal surface is equal on each side of the apparatus, the internal surface temperatures will differ from the water temperatures by $\Delta T_{w-i}/2$. Thus, the internal surface temperature on the hot side is estimated to be $23.19 \text{ }^\circ\text{C} - 0.49 \text{ }^\circ\text{C}$, or $T_1=22.70 \text{ }^\circ\text{C}$, and that on the cold side is estimated to be $3.26 \text{ }^\circ\text{C} + 0.49 \text{ }^\circ\text{C}$, or $T_2=3.75 \text{ }^\circ\text{C}$. Thus, $\Delta T_{int}=18.95 \text{ }^\circ\text{C}$, and the average of these internal surface temperatures is $T_{ave}=286.4 \text{ K}$.

The next step is to calculate the total thermal conductance of the sample. The predicted per unit area heat flow between two uncoated sheets of glass at the surface temperatures given above is $75.2 \pm 0.7 \text{ Wm}^{-2}$. This was determined from the exact method presented in Chapter 3 on the basis of the optical constants for glass that are reported by Rubin [11]. As explained in Chapter 3, the error in the predicted heat flow is due to the uncertainty in the reflectance measurements from which the optical constants were calculated [12]. Substituting the mean predicted heat flow and the above estimated surface temperatures into equation 2.19 gives an effective emittance, ϵ_{eff} , of 0.744. Substituting this emittance, and $T_{ave}=286.4 \text{ K}$ into equation 2.21 yields a radiative conductance, C_{rad} , of $3.964 \text{ Wm}^{-2}\text{K}^{-1}$.

It is noted that the error in ΔT_{int} - i.e. the error in the internal surface temperatures - has not been considered. As explained below, this error is accounted for separately as an uncertainty associated with the heat flow measurements that are to be made with

the TCA. This is justified on the basis that ΔT_{int} is very stable and does not change significantly between the calibration of the apparatus and the heat flow measurements. It is possible that this error in ΔT_{int} could affect the present determination of the effective area by virtue of the temperature dependence of C_{rad} . However, a sensitivity analysis shows that C_{rad} is not strongly dependent on the estimated values for the internal surface temperatures; the predicted per unit area heat flow for the slightly different internal surface temperatures of 23.0 and 3.0 °C yields the same radiative conductance. Thus, C_{rad} is taken to be equal to the above value of 3.964 Wm⁻²K⁻¹.

The total sample thermal conductance is equal to the parallel combination of the radiative conductance and the conductance of the small amount of gas in the evacuated space, C_{gas} , which at pressures below about 70 Pa can be approximated by:

$$C_{gas} \approx 0.4P \quad , \quad (2.24)$$

where P is the gas pressure in pascals and C_{gas} is in units of Wm⁻²K⁻¹ [13]. This expression is based on the assumption that the combined accommodation coefficient for the molecule-surface interaction is 0.5 [13]. In typical vacuum glazings, the internal gas pressure is much less than 0.1 Pa. At such low pressures, $C_{gas} \ll C_{rad}$ and thus C_{gas} can usually be ignored. However, for the sake of completeness, it is included in this analysis. For the measured pressure of 3×10^{-3} Pa, C_{gas} is 1.2×10^{-3} Wm⁻²K⁻¹. Thus the total sample conductance, C_{sample} , is calculated to be 3.965 Wm⁻²K⁻¹.

The following are now known or estimated: Q_o , ΔT_{int} and C_{sample} . These are substituted into equation 2.22 to give a second estimation for the effective area of 1.452×10^{-4} m². The above procedure is re-iterated on the basis of this new effective area. Since the change in ΔT_{int} is very small, there is no need to re-estimate C_{rad} - as previously explained. After a few iterations, it is clear that the solution converges to the same result as that which is given above: $A_{eff} = 1.452 \times 10^{-4}$ m². It is recalled that this result is based on a predicted per unit area heat flow of 75.2 Wm⁻² which is accurate to about ± 1 % (the exact error is ± 0.9 %). Therefore, the error quoted for the effective area is ± 1 %, or $\pm 0.01 \times 10^{-4}$ m².

The effective measurement area of the TCA is therefore $(1.45 \pm 0.01) \times 10^{-4}$ m². This corresponds to an effective radius of 6.80 mm which is close to the ideal result - the mid-gap radius of 7.00 mm. It is noted that the effective area is 6 % less than the area defined by the mid-gap radius (1.54×10^{-4} m²). This is approximately consistent with

the prediction made in Section 2.2.5 that, as a result of the difference in the epoxy resin thicknesses, the effective area should be at most 5 % less than the mid-gap area.

The measurement of each water inlet and water outlet flow temperature is accurate to ± 0.02 °C. Thus, the average water flow temperature for each plate is accurate to ± 0.04 °C. This means that the difference between the two average water flow temperatures (ΔT_{total}) is accurate to ± 0.08 °C. Since most TCA measurements are for a ΔT_{total} of about 20 °C, and since, as shown by equation 15, $\Delta T_{int} \approx \Delta T_{total}$, the error in ΔT_{int} is taken to be ± 0.4 %. This error is associated with the measured heat flow due to the dependence of the null power on ΔT_{int} - as per equation 2.22. Therefore, as indicated by equation 2.22, this ± 0.4 % error in ΔT_{int} and the ± 1 % error in A_{eff} result in a ± 1.4 % total error in the null power for the calibrated effective area. Thus, the per unit area heat flows measured with the TCA are accurate to ± 1.4 %.

2.5 Results

2.5.1 Linearity Between The Bridge Voltage and The Power

Figure 2.17 is a plot of the amplified bridge voltage, V_b , as a function of the power input to the resistor for a standard sample of two sheets of low emittance glass (K glass). The temperatures of both plates - and thus both glass sheets - are about 23 °C. This figure demonstrates that, as stated above, there is a linear relationship between the bridge voltage and the power applied to the resistor. It is expected that the maximum null power (i.e. heat flow) for the TCA measurements to will be about 15 mW; this is for a measurement immediately over a pillar in a glazing sample which consists of two sheets of uncoated glass and which is exposed to a glass-to-glass temperature difference of about 20 °C. Figure 2.17 shows that the linear relationship between the bridge voltage and power extends to this expected maximum power. Therefore, the null power can indeed be correctly determined by interpolating between or extrapolating from two bridge voltages measured at different powers which are less than 15 mW. The running average of the computer processed amplified bridge voltage is accurate to ± 0.1 mV, and the reproducibility (i.e. the resolution) for the power measurements is ± 10 μ W.

Also shown in Fig. 2.17, on the second y-axis, is the temperature difference between the guard and metering piece thermistors, $T_g - T_m$, that corresponds to the bridge voltage. This temperature difference was calculated by dividing the bridge voltage by

the temperature-to-voltage gain, G_{TV} , of 0.60 VK^{-1} . This figure shows that the above maximum expected null power of 15 mW corresponds to a maximum expected temperature difference between the thermistors of about 0.4 K .

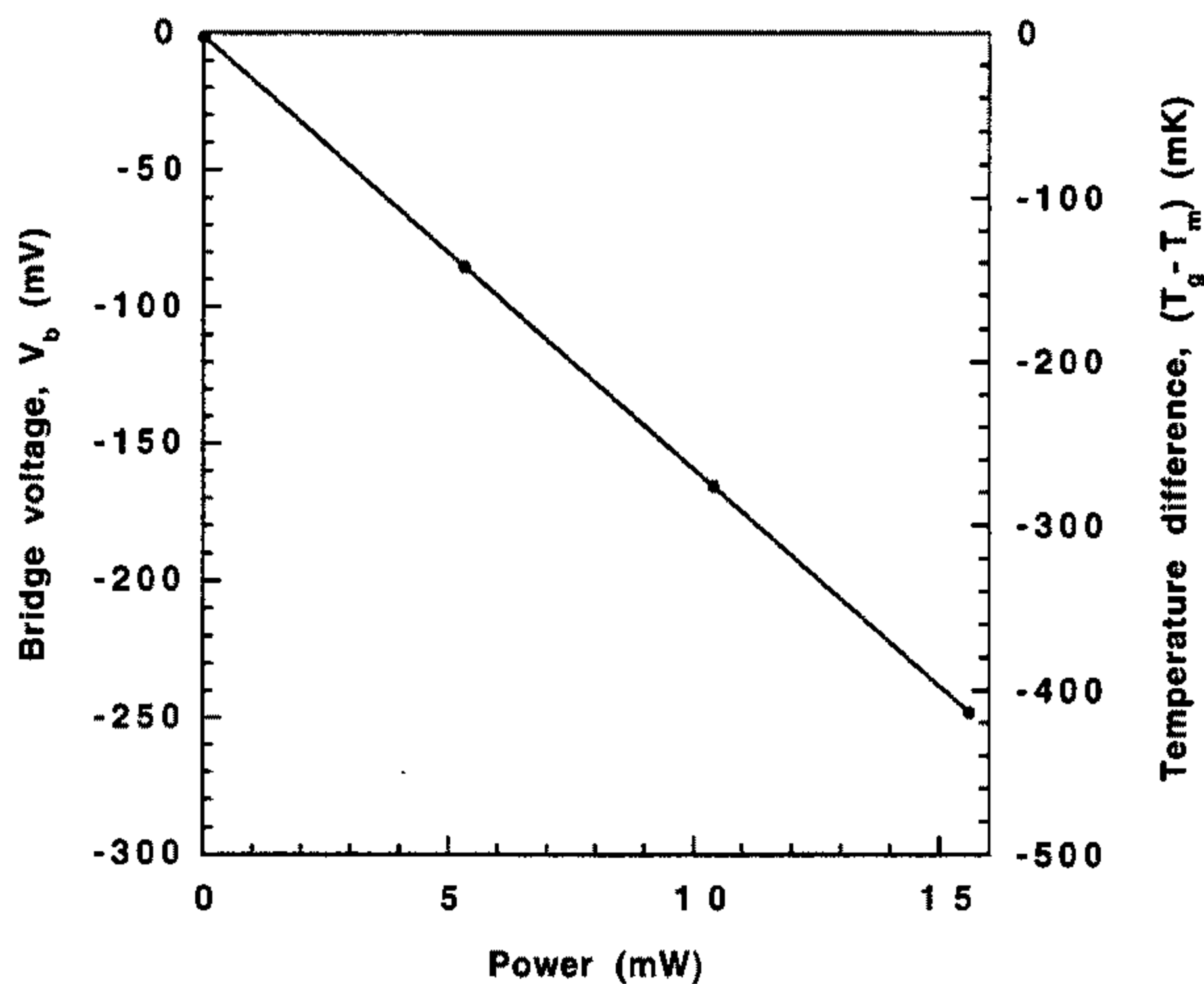


Fig. 2.17: A plot of the amplified bridge voltage, V_b , and the corresponding difference between the guard and metering piece thermistors, $T_g - T_m$, as a function of the power applied to the metering piece resistor. The thermistor temperature difference was calculated by dividing the bridge voltage by the temperature-to-voltage gain of the apparatus, G_{TV} . These data were taken with both plates of the TCA at about 23°C .

The linearity between the bridge voltage and power demonstrates that, as previously stated, there is also a linear relationship between the thermistor temperature and the ratio of the change in the thermistor resistance to the reference value of the thermistor resistance ($\Delta R/R_{th}$). Of course this only applies to small changes in the thermistor temperature (at least less than about 0.4 K according to Fig. 2.17).

The slope of the plot in Fig. 2.17 is 15.8 VW^{-1} , or 26.3 KW^{-1} . According to equation 2.6, the latter slope should be approximately equal to the thermal resistance between the metering piece and the guard, R_{m-g} . This experimentally based result for R_{m-g} is close to the estimated resistance of 29.6 KW^{-1} that was given in Section 2.2.3.

2.5.2 Calculation of The Heat Flow Through The Pillars

For most TCA measurements on vacuum glazing samples, the null power is a measure of the heat flow contributions from both the radiative heat transfer and the pillar conduction. There is also a contribution to the null power from the internal gas conduction but, as noted below, this is usually negligible. Thus, in order to calculate the radiative conductance, the total amount of pillar conduction detected by the metering piece must be subtracted from the null power.

Consider a measurement over a single pillar as shown in Fig. 2.18. Due to the thermal resistance of the glass below the hot plate, the large amount of heat flow from the hot plate to the pillar spreads out from an area which is much larger than the area of the metering piece. Therefore, not all the pillar conduction, $Q_{pillar}(total)$, is detected or "intercepted" by the metering piece. This spreading out of the heat flow is governed by the total thickness, t , of the faceplate and sample glass on the hot side - i.e. the measurement side - of the evacuated space. The portion of the pillar conduction that is detected by the metering piece, $Q_{pillar}(detected)$, depends on the metering piece diameter, D , and on the distance, L , of the pillar from the center of the metering piece. Christopher Dey of the University of Sydney performed a finite element analysis to calculate the ratio of $Q_{pillar}(detected)/Q_{pillar}(total)$ for the dimensions t , D and L [3]. Dey's results are presented in Fig. 2.19. These results were obtained from a model which is similar to the finite element model described in Chapter 4.

For the typical pillar array spacing of 25 mm, each of the four pillars of a square unit cell are about 18 mm from the middle of the cell. Thus, if the center of the metering piece is located in the middle of such a unit cell, $L=18$ mm. Since the metering piece diameter is 13 mm and the typical total hot side glass thickness is 6 mm, $D/t=2.2$ and $L/t=3$. The results of Fig. 2.19 show that, for these values of D/t and L/t , the percentage of the heat flow from a single pillar that is detected by the metering piece at this location is about 1 %. For a typical pillar radius of 0.125 mm, and for a typical temperature difference across the glass sheets of 20 °C, the individual pillar conduction is calculated on the basis of the analytic model presented in Chapter 4 to be 5 mW. Therefore, the conduction through all of the four pillars that are the closest to the metering piece contributes about 0.20 mW to measured heat flow (the contribution from the conduction through the other pillars of the array, which are even further from the metering piece, is negligible). This is about 5 to 10 % of the typical TCA measurements of the radiative heat flow through vacuum glazing (2 to 4 mW, depending on the emittances of the internal surfaces).

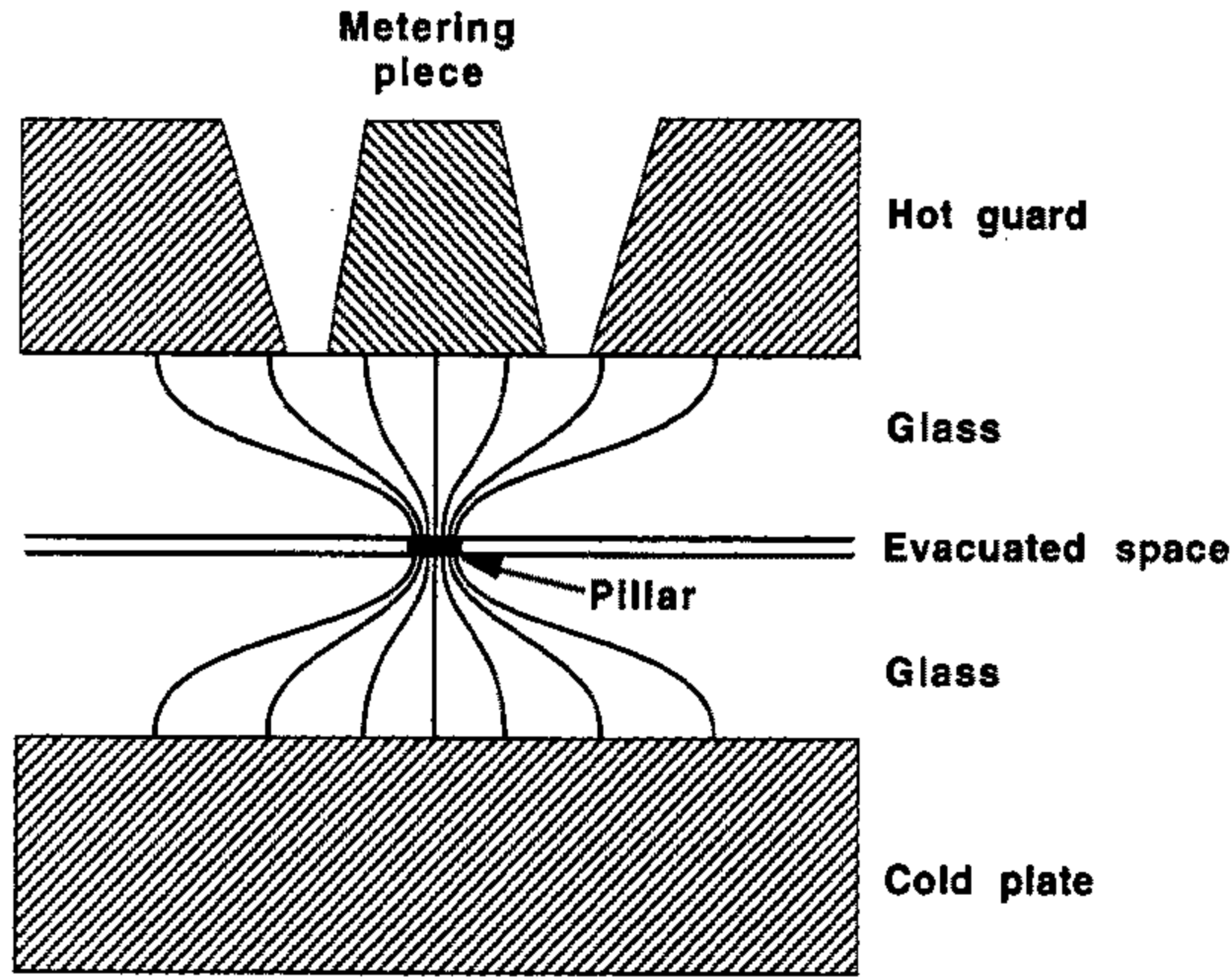


Fig. 2.18: As a result of the thermal resistance of the glass below the hot plate, the large amount of heat flow from the hot plate to the pillar spreads out from an area which is much larger than the area of the metering piece. Thus, if the TCA is used to measure the heat flow to a single pillar, only a fraction of this flow will be detected or "intercepted" by the finite sized metering piece. This figure shows a simplified representation of the heat flow.

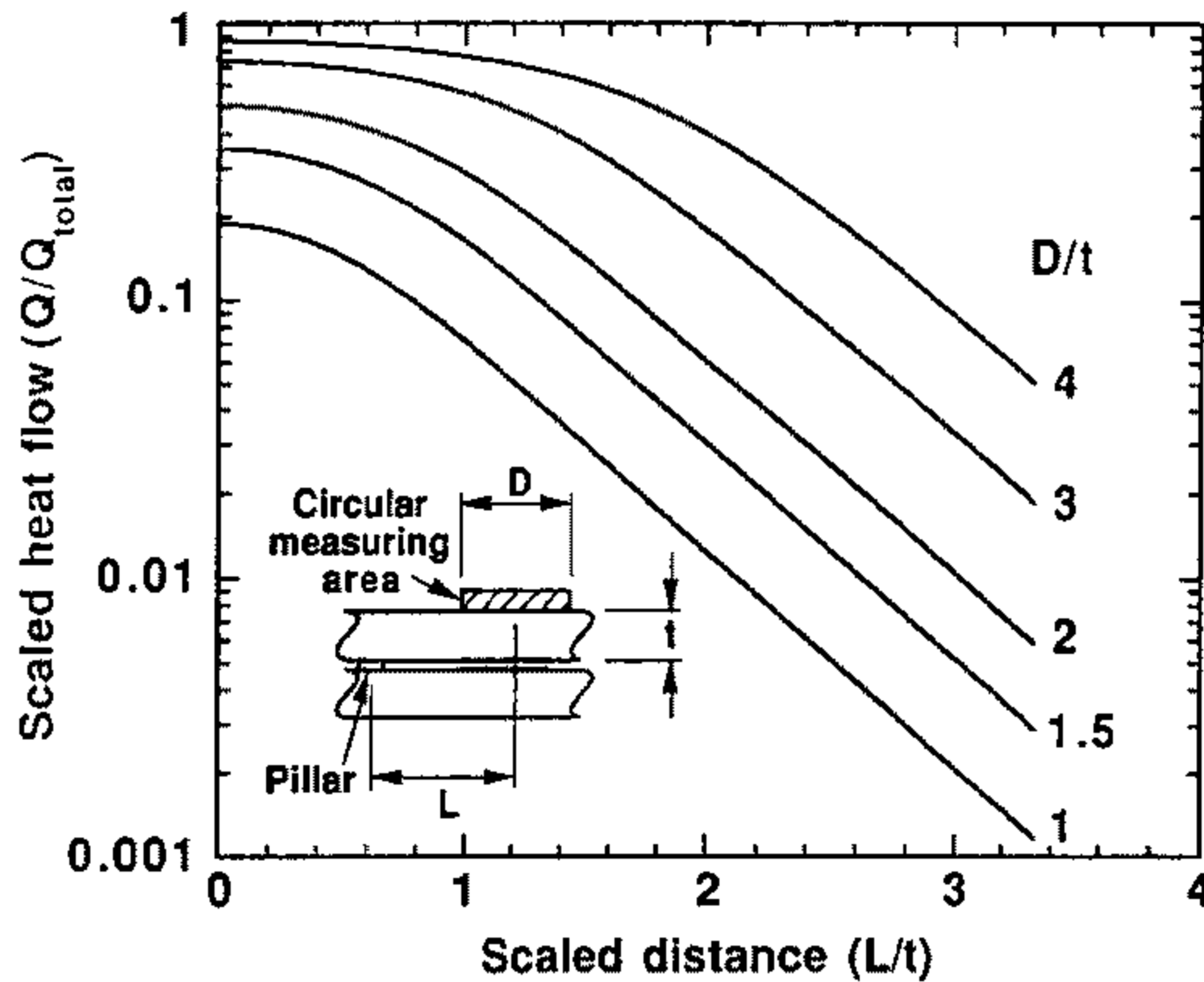


Fig. 2.19: Dey's results for the ratio of the pillar conduction detected by the metering piece to the total pillar conduction, $Q_{pill}(detected)/Q_{pill}(total)$, as a function of the dimensionless parameters L/t and D/t [3]. The variable L is the distance of the pillar from the center of the metering piece, t is the total thickness of the glass on the hot side of the evacuated space and D is the diameter of the metering piece.

In a typical vacuum glazing sample, there is a region near the center of the sample where the pillars of a unit cell of the array have been displaced outwards to about 30 mm from a central location where the metering piece can be positioned. Thus, for this situation, $L=30$ mm. The results of Fig. 2.19 show that, for the above values of D and t , and for a typical total pillar conduction of 5 mW, the contribution to the null power, Q_o , from such pillars at $L=30$ mm is of the order of microwatts. Since this contribution is much less than the typical measured radiative heat flows (2 to 4 mW), it can be neglected. As noted below, it is usually the case that gas conduction is also negligible. Therefore, Q_o measured in the middle of such a unit cell of displaced pillars can be assumed to be equal to Q_{rad} .

For a measurement over a single pillar, $Q_{pill}(detected)=Q_o-Q_{rad}$, where Q_{rad} is equal to the radiative heat flow determined in the manner described above. It is noted that, as shown in Chapter 4, there is some suppression of the radiative heat flow around a pillar due to the temperature non-uniformities associated with the pillar conduction. However, this has a negligible effect on Q_{rad} . The corresponding total pillar conduction, $Q_{pill}(total)$, can be calculated on the basis of the results presented in Fig. 2.19. For a measurement over an individual pillar (for which $D/t=2.2$ and $L/t=0$), Fig. 2.19 shows that $Q_{pill}(detected)/Q_{pill}(total)=0.57$. Therefore, the total pillar conduction is equal to the detected - i.e. the measured - pillar conduction divided by 0.57. This approach to quantifying $Q_{pill}(total)$ is validated in Chapter 4 by a comparison between measured and predicted results for the pillar thermal resistance.

Once $Q_{pill}(total)$ is known, Fig. 2.19 can be used to calculate the separate contributions from individual pillars to Q_o measured anywhere on the sample. As illustrated above, for $D=13$ mm and $t=6$ mm, pillars at distances greater than about $L=30$ mm make a negligible contribution to Q_o and thus they need not be considered for the guarded hot plate measurements. This is why it was valid to assume that, in calibrating the apparatus with the standard sample, the heat flow through the twelve pillars located beyond the radius of the TCA (50 mm) make a negligible contribution to Q_o .

If there is no central location with displaced pillars for a direct measurement of Q_{rad} , this heat flow can be determined simultaneously with $Q_{pill}(total)$ by measurements in the middle of a unit cell and over each of the nearby pillars. If the pillars are of uniform size and are in equally good contact with the glass, and if there are no shear cracks around the pillars, then a measurement over only one pillar required.

2.5.3 Calculation of The Radiative Conductance

The general equation for the radiative conductance is:

$$C_{rad} = \frac{Q_o - \sum Q_{pill}(\text{detected})}{A_{eff} \Delta T_{int}}, \quad (2.25)$$

where A_{eff} is $1.45 \times 10^{-4} \text{ m}^2$, and $\sum Q_{pill}(\text{detected})$ is the sum of any significant pillar heat flows that are detected by the metering piece. The numerator of this equation gives the total measured radiative heat flow.

2.5.4 Temperature Dependence of The Radiative Heat Transfer

Equation 2.19 implies that there is a nearly linear relationship between Q_{rad} and $T_1^4 - T_2^4$. However, as will be explained in Chapter 3, there are slight departures from this linearity for the radiative heat transfer between two glass surfaces due to the wavelength and angular dependencies of the surface emittances. These departures are best seen by the following residual analysis. A quadratic function passing through the origin is fitted to the data in a plot of Q_{rad} versus $T_1^4 - T_2^4$. The deviation of each data point from the linear portion of this quadratic fit is then calculated and plotted as a function of $T_1^4 - T_2^4$. The latter plot shows the departures from linearity - i.e. the residuals; for example, a residual of -0.02 mW indicates that the data point lies 20 mW below the linear portion of the quadratic fit to all of the data points. As shown in Figs. 3.6 of Section 3.5, the experimental and calculated residuals for Q_{rad} as a function of $T_1^4 - T_2^4$ for two K glass surfaces are in very good agreement - to within $\pm 10 \text{ } \mu\text{W}$. Such close agreement between the experimental and predicted residuals has also been observed for two uncoated glass surfaces. These residual analyses demonstrate the consistency between the experimental data and modelling results. They also demonstrate the linearity between the measured null power and the heat flow across the sample over a wide range of temperatures.

2.5.5 Calculation of The Gas Conduction

As previously stated, the gas conductance in vacuum glazing for pressures below about 70 Pa can be approximated by equation 2.24: $C_{gas} \sim 0.4P$, where P is the internal gas pressure in pascals and C_{gas} is in units of $\text{Wm}^{-2}\text{K}^{-1}$. Typically, the pressure in vacuum glazing is much less than 0.1 Pa and thus the gas heat conduction, Q_{gas} , is negligibly small in comparison with the other heat flows. However, in some samples, the internal gas pressure is larger than normal and this can lead to greater levels of gas conduction.

Such higher pressures can be caused by improper manufacture. They can also result from the outgassing from the internal surfaces that can occur very slowly over the service life of the glazing, or rapidly during high temperature ageing tests [13]. Usually, the consequent gas conduction is still negligible. However, it is useful to measure the magnitude of this heat flow in order to estimate the internal pressure of the vacuum glazings for quality control purposes, and to study the processes related to outgassing.

If gas conduction is significant, measurements of the radiative heat flow across a vacuum glazing will be greater than expected. The additional heat flow - i.e. the gas conduction - is usually so small that it is best revealed by a residual analysis like the one described in the above section. The separate radiative and gaseous heat flows can be respectively calculated with the exact method presented in Chapter 3 on the basis of the appropriate optical constants referred to in the same chapter, and with the gaseous conductance given by equation 2.24. These heat flows were calculated for different temperatures between the internal surfaces of a vacuum glazing made of two sheets of K glass, and for different internal pressures. The optical properties for the K glass corresponded to post-glazing manufacture (i.e. heat treated) K glass. For each pressure, the combined results for these heat flows ($Q_{rad}+Q_{gas}$) were plotted as a function of $T_1^4 - T_2^4$ and a quadratic curve which passes through the origin was fitted to the results. Figure 2.20a shows such quadratic fits to the combined heat flows. As expected, the absolute magnitude of the heat flow increases with the pressure. Figure 2.20b shows the deviations of the combined heat flows from the linear portions of the quadratic fits presented in Fig. 2.20a. Also presented in Fig. 2.20b are the deviations for two pressures that are not considered in Fig. 2.20a. At low pressures, the effect of gas conduction is small and the deviations for the null powers are negative - which is the usual trend for radiative heat transfer alone, as shown by Figs. 3.6 of Section 3.5. The deviations become less negative and then positive as the pressure increases to values for which the gas conduction is more significant. These calculated results for a sample made of two sheets of K glass show that, at pressures greater than 0.1 Pa, the effect of gas conduction is clearly obvious by the positive deviations. For a sample made with two sheets of uncoated glass, for which the radiative conductance is approximately five times greater, the same effect is clearly obvious at pressures greater than about 0.5 Pa.

The above effect of the gas conduction on the deviations from the linear portions of quadratic fits passing through the origin has been observed in vacuum glazing samples for which the internal vacuum has been degraded by outgassing at high temperatures

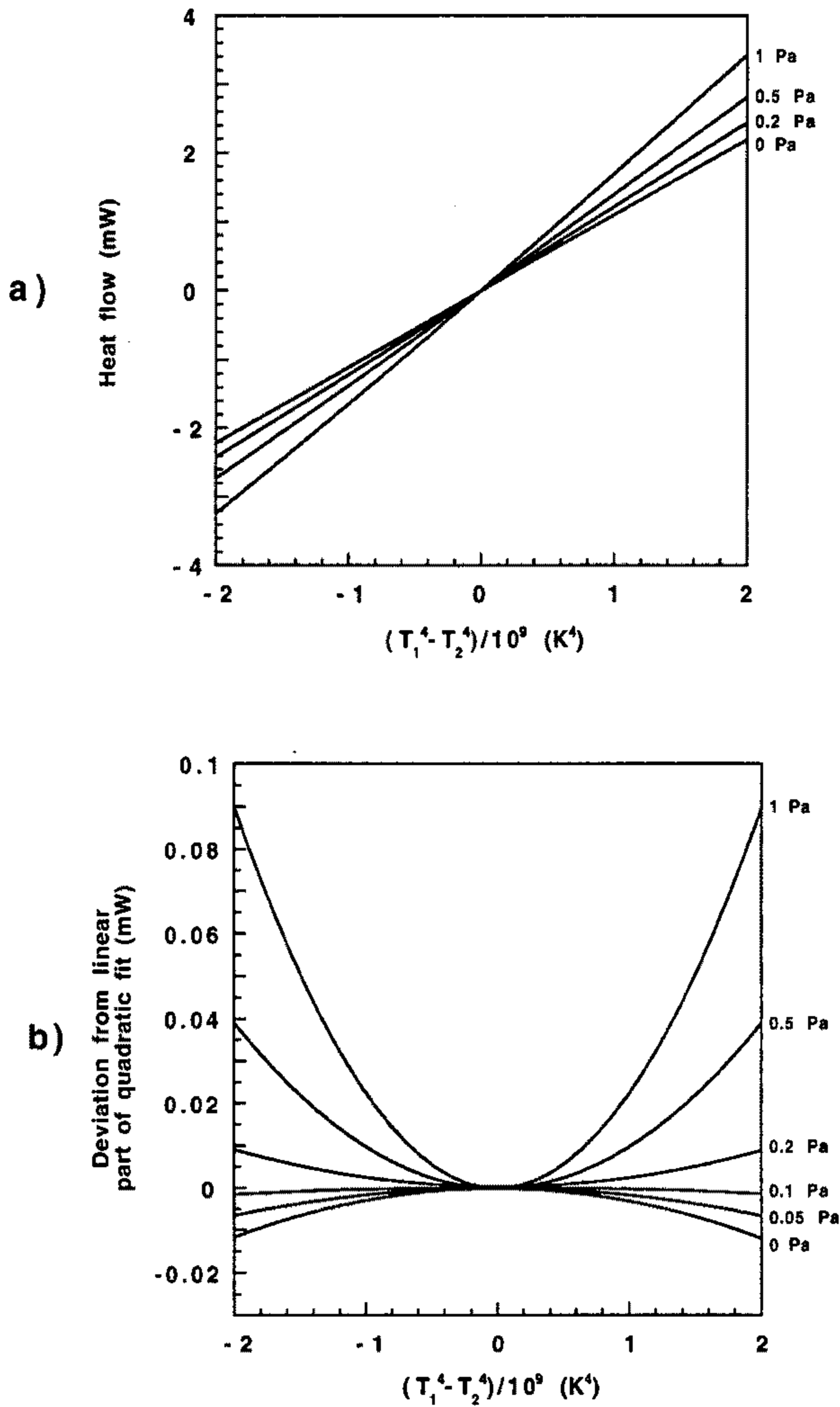


Fig. 2.20: (a) The temperature dependence of the combined calculated radiative and gaseous heat flows for several gas pressures inside a vacuum glazing. For each pressure, a quadratic curve passing through the origin was fitted to a plot of the combined calculated heat flow as a function of the internal surface temperatures, $T_1^4 - T_2^4$. This figure shows such quadratic fits. These results are for a glazing with two sheets of K glass.
(b) Deviations of the combined heat flows from the linear portions of the quadratic fits presented in (a). Also presented are the deviations for two pressures that are not considered in (a): 0.05 Pa and 0.1 Pa.

[13]. Figures 2.21a and b present results obtained with the TCA for a gassy (i.e. degraded) sample of vacuum glazing with two sheets of K glass. The internal pressure of the sample can be found by determining what pressure will give gaseous conductive heat flows which, when combined with the expected radiative heat flows, yield results which best agree with the experimental deviations (i.e. residuals) of Fig. 2.21b. The separate radiative and gaseous heat flows can be calculated as described above (with the radiative flows calculated on the basis of the optical constants for heat treated K glass). The best fit to the experimental deviations of Fig. 2.21b was obtained from this analysis with a pressure of 0.66 ± 0.09 Pa (the error is due to the uncertainties in the radiative heat flow and the internal surface temperatures). The curve of Fig. 2.21b shows the expected results for the mean pressure of 0.66 Pa.

In order to verify the above estimated internal pressure, the residual gas was removed from the sample for a comparative measurement of the null power. With the glazing mounted in the TCA, a vacuum system was connected to the sample around the location of the stub of the pump out tube; the seal for this connection was made at the external glass surface with an O-ring. The stub of the tube was then broken under vacuum with a custom made tool so that the gas in the glazing could be removed. The pressure just above the connection between the glazing and the vacuum system was measured with a vacuum ion gauge to be about 1×10^{-3} Pa. The circulating water temperatures were set to give a ΔT_{int} of 19.73 ± 0.08 °C. The measured null power was 0.47 ± 0.05 mW lower than the result obtained before the re-evacuation procedure for a consistent value of ΔT_{int} (19.69 ± 0.08 °C). This difference in the null powers represents the gas conduction, Q_{gas} , that was present prior to the removal of the residual gas. The internal pressure for the gassy sample can be calculated from this result on the basis of the gas conductance given by equation 2.24:

$$\begin{aligned}
 P &= \frac{Q_{gas}}{0.4 A_{eff} \Delta T_{int}} \\
 &= \frac{0.47 \text{mW} \pm 11\%}{0.4 (1.45 \times 10^{-4} \text{ m}^2 \pm 1 \%) (19.73 \text{ °C} \pm 0.4 \%)} \quad (2.26) \\
 &= 0.41 \pm 0.05 \text{ Pa} .
 \end{aligned}$$

Therefore, the above direct - and destructive - method of determining the internal gas pressure reveals that this pressure is 0.41 ± 0.05 Pa. This result is not consistent with the result of 0.66 ± 0.09 Pa obtained from the above non-destructive approach of

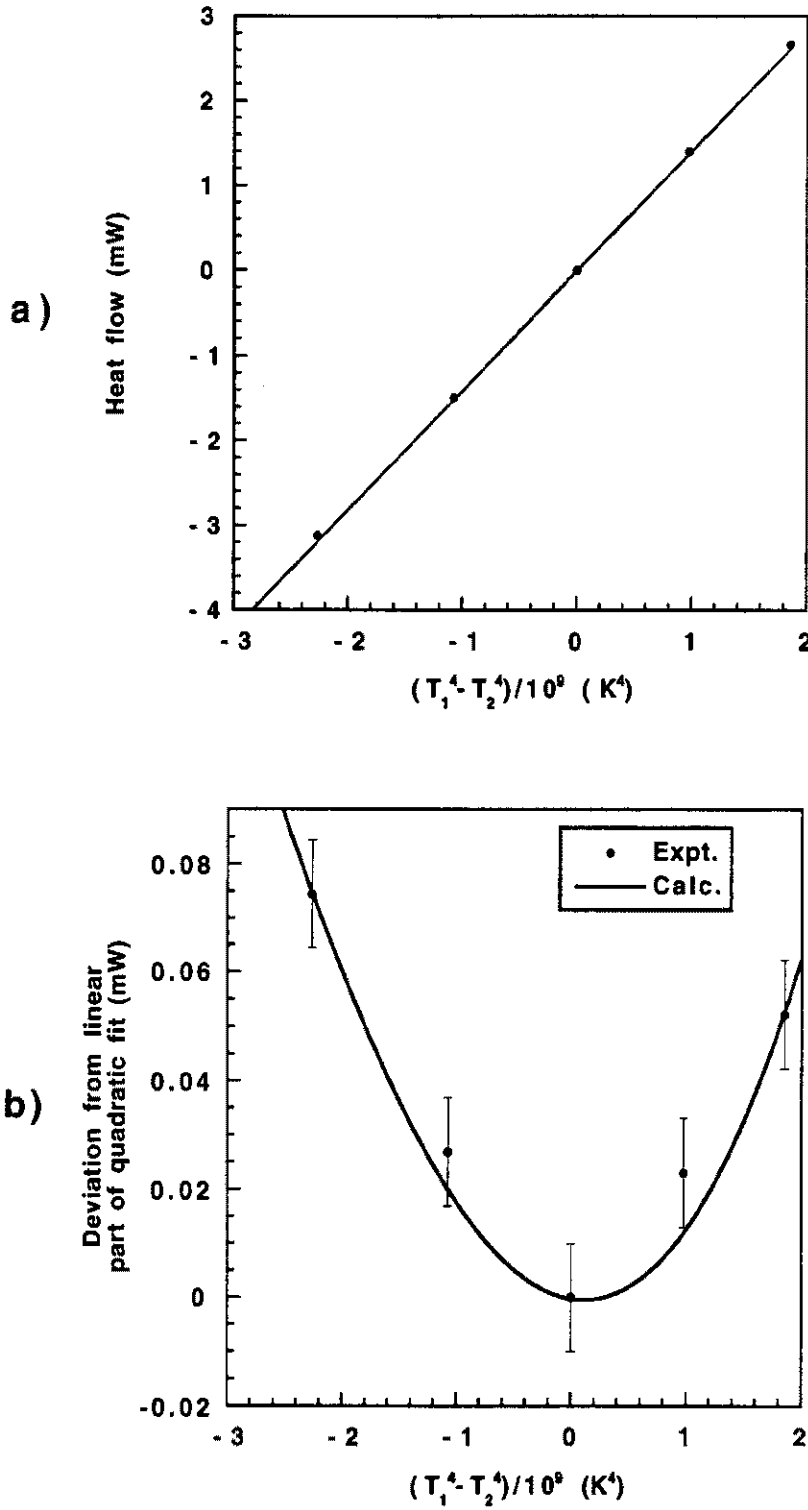


Fig. 2.21: (a) A plot of the measured null power as a function of $T_1^4 - T_2^4$ for a degraded vacuum glazing sample made with two sheets of K glass. Data points are shown with the linear portion of the quadratic fit to the data which passes through the origin. (b) Deviations of the measured null powers of (a) from the linear portion of the quadratic fit which passes through the origin. These deviations are indicated as the "Expt." data points. It is possible to estimate the internal gas pressure by determining at what pressure the best fit to the experimental data can be obtained with the calculated deviations for the combined radiative heat transfer and gas conduction processes. The curve indicated as "Calc." shows the calculated results for the internal pressure estimated in this way: 0.66 Pa.

obtaining a fit to the data of Fig. 2.21b. The difference between these results could be related to the location of the vacuum ion gage. Ideally, this gage should have been at a location that was remote from the connection to the vacuum system. However, it was actually located just above the connection and thus it might not have given a correct indication of the internal pressure; although the pressure at this location was stable after several hours of evacuation, it is possible that not all of the internal residual gas far from the connection had been removed from the glazing and thus that the gage slightly underestimated the internal gas pressure. The gage was not re-located because this would have required that a hole be drilled in the sample for the gauge - at the risk of breaking the sample, or of shifting the pillars (which could lead to glass-to-glass contacts upon re-evacuation). Given the uncertainty in the vacuum ion gage result, the agreement between the estimates of pressure from the above two different approaches is reasonably good. It is concluded that the non-destructive technique for estimating the internal gas pressure of vacuum glazing by fitting calculated results to the measured deviations in the temperature dependence of the null power is accurate to at least the *maximum* difference between the two above results: 0.4 Pa.

Therefore, it can be seen that an examination of TCA null power as a function of $T_1^4 - T_2^4$ provides a non-destructive method of determining whether a glazing sample contains a significant amount of gas. Of course a gassy sample has a larger heat transfer across the evacuated space, but so does a sample with higher internal surface emittances. As mentioned in Section 3.4, there are point to point variations in the emittance of the low emittance coating used in vacuum glazing. However, the non-destructive method presented above can, to within the stated limits of the resolution, distinguish between the thermal effects due to local variations in the surface emittances and the obvious thermal effects due to gas.

2.6 Conclusion

An apparatus has been designed and constructed to measure local heat flows in samples of vacuum glazing. The design of this thermal conductance apparatus (TCA) is similar to that of standard guarded hot plate devices, but it incorporates several modifications for the intended application. The TCA consists of a hot plate and an isothermal cold plate on either side of a sample. The hot plate is composed of an isothermal guard which surrounds a metering piece. The heat flow across the evacuated space of the sample is determined by measuring the amount of power - the

null power - that must be dissipated in a resistor in the metering piece in order to obtain equal temperatures at specific locations in the metering piece and the guard. The temperatures are measured with thermistors which are part of a Wheatstone bridge circuit.

The TCA was designed to achieve the spatial resolution required to distinguish among the three heat flows through vacuum glazing at locations remote from the edges: radiative heat transfer, pillar conduction and gas conduction. The size of the metering piece (13 mm in diameter) was selected to give a spatial resolution of a comparable size for the measurements; this spatial resolution is defined by the effective area, which is described below. In the design and construction of the apparatus, care was taken to locate properly the thermistors, to ensure that the epoxy resin layers were as thin and consistent as possible, to terminate thermally the wires from the resistor core to the metering piece, and to ensure that there is a high thermal resistance between the metering piece and the guard. This thermal resistance was determined by finite element modelling and experimental techniques, and these techniques give results which are in close agreement.

In order to determine the thermal conductance of a sample, the temperature difference across the evacuated space, and the effective measurement area for the heat flow from the metering piece across this space must be well known. The only temperature difference that is measured across the apparatus is that between the two water flows. Due to intervening thermal resistances, the temperature difference between the two internal glass surfaces is slightly less than this measured result but it can be determined from the measurement on the basis of analytic or finite element modelling techniques. The effective measurement area differs from the geometric area of the metering piece due to the spreading of the heat flow in the glass between the piece and the evacuated space. The effective measurement area was determined to be $(1.45 \pm 0.01) \times 10^{-4} \text{ m}^2$ from a null power measurement on a sample which has a well known thermal conductance: two sheets of uncoated glass separated by a vacuum. Based on the error in the effective area, and on the approximately $\pm 0.4 \%$ uncertainty in the temperature difference across the evacuated space, the accuracy for the per unit area heat flow measurements is estimated to be $\pm 1.4 \%$.

Measurements made with the TCA indicate that there is a linear relationship between the voltage output from the thermistor bridge circuit and the power input to the metering piece resistor. It is therefore possible to determine the null power by interpolation between or extrapolation from voltage readings at different powers.

Modelling results and null power measurements on a similar apparatus for a sample with a very low radiative conductance (two sheets of glass coated with a gold film and separated by a high vacuum) demonstrate that parasitic heat flows from the metering piece of the TCA can be expected to be negligible.

Due to the thermal resistance of the glass below the hot plate, the large amount of heat flow from the hot plate to a pillar spreads out from an area which is much larger than the metering piece. The fraction of the total pillar conduction that is detected by the metering piece of the apparatus is a function of the diameter of the metering piece, the total glass thickness on the hot side of the evacuated space, and the distance of the pillar from the center of the metering piece. The relationships among these parameters and the detected and total pillar heat flows have been determined from a finite element modelling analysis [3]. For the metering piece diameter of 13 mm and the typical total hot side glass thickness of 6 mm, only a negligible amount of heat flow is detected from pillars of the array which are located more than 30 mm from the center of the metering piece. In measurements on vacuum glazing samples, the heat flow due to the radiative heat transfer is equal to the null power minus the sum of any significant heat flows detected from nearby pillars.

If the amount of gas within the evacuated space is significant, it will alter the relationship between the null power and the difference in internal surface temperatures that would be expected for radiative heat transfer alone. This alteration can best be seen in a residual analysis of the results. The internal pressure can be estimated from the experimentally determined gas conduction. In most cases, gas conduction is negligibly small and need not be considered in the overall thermal performance of vacuum glazing. However, the experimental determination of the gas conduction is a useful quality control procedure for ensuring the integrity of the internal vacuum of the glazing. It is also useful for studying the processes associated with the outgassing from the internal glass surfaces.

The TCA can be used to determine the overall glass-to-glass center-of-glass thermal conductance, or U-value, for a vacuum glazing sample. This U-value is the sum of the conductance due to the radiative heat transfer, and the conductance due to the heat conduction through the pillar array. For the rest of this thesis, these conductances are respectively referred to as the radiative U-value, and the pillar array U-value. Both of these quantities can be determined experimentally from heat flows measured with the TCA.

The radiative U-value is usually determined from a measurement at a central location where the pillars have been displaced outwards; at this location, the pillars are far enough away from the metering piece (about 30 mm) that pillar conduction can be ignored. Alternatively, the measurement can be made in the middle of a regular unit cell of pillars, with the appropriate correction being made for the total detected pillar conduction. The pillar array U-value can be calculated from the experimentally determined thermal resistance of a single pillar (the total pillar conduction for one pillar divided by the temperature difference between the glass surfaces) and, as explained in Chapter 4, from the regular pillar separation for the array. As also explained in Chapter 4, the thermal resistance of a pillar is affected by the uniformity of the contact between the pillar and the glass sheets, and by the presence of any cracks around the pillar. Since these conditions can vary across a vacuum glazing, it is therefore usually necessary to base the calculation of the pillar array U-value on measurements over several typical pillars.

2.7 List of Symbols

A	area through which heat flows
A_{eff}	effective area of the metering piece
A_{wire}	cross-sectional area of the core of a wire from the power resistor
C_{gas}	gaseous thermal conductance across the evacuated space
C_{lin}	lineal thermal conductance between a wire core and the metering piece
C_{rad}	thermal radiative conductance across the evacuated space
C_{sample}	thermal conductance between the internal surfaces of a glazing sample
C_{w-i}	thermal conductance between the water flows and internal glass surfaces
D	diameter of the metering piece
g	width of the gap between the metering piece and the guard
G_{T-V}	temperature-to-voltage gain
i	electrical current
k	thermal conductivity
k_{epoxy}	thermal conductivity of epoxy resin
k_{ins}	thermal conductivity of the insulation around a cylinder
k_{wire}	thermal conductivity of the core of a wire from the power resistor
ℓ	characteristic distance
L	distance of a pillar from the center of the metering piece
L_c	length of an insulated cylinder
P	gas pressure inside a vacuum glazing
Q	heat flow, or ohmic power dissipated in the metering piece resistor
Q_{gas}	heat conduction through the residual gas inside vacuum glazing
Q_{guard}	heat flow from the resistor to the guard via the resistor wires
$Q_{metering\ piece}$	heat flow from the resistor to the metering piece
Q_o	null power
$Q_{pillar}(detected)$	pillar conduction detected by the metering piece
$Q_{pillar}(total)$	total pillar conduction
Q_{rad}	radiative heat transfer
Q_{sample}	heat flow from the metering piece across the sample to the cold plate
r	radial distance from the axial centerline of the metering piece
r_g	outer radius of insulation around a cylinder
r_{gap}	mid-gap radius
r_i	radius of an insulated cylinder
R	electrical resistance of bridge circuit resistor
R_f	electrical resistance of op-amp feedback resistor
R_{th}	reference electrical resistance of thermistor (at room temperature)
R_{eff}	effective thermal resistance of the metering piece
R_{m-c}	thermal resistance between the metering piece and the cold plate
R_{m-g}	total thermal resistance between the metering piece and the guard
$R_{m-g}(air)$	thermal resistance of the air gap between the metering piece and guard
$R_{m-g}(glass)$	thermal resistance of the glass between the metering piece and guard
$R_{m-g}(glass+epoxy)$	thermal resistance of the glass and epoxy resin below and between the metering piece and guard
$R_{m-g}(wires)$	thermal resistance of the wires between the metering piece and guard
R_{r-g}	thermal resistance between the resistor core and the guard

R_{r-m}	thermal resistance between the resistor core and the metering piece
t	total thickness of glass on the measurement side of the evacuated space
T_{ave}	average of the internal glass surface temperatures
T_c	temperature of the cold plate
T_g	temperature of the guard (or of the thermistor located there)
T_m	temperature of the metering piece (or of the thermistor located there)
T_o	temperature of the end of a fin
T_r	temperature of the resistor core
$T(x)$	temperature along a wire core from the resistor core, or the temperature along an analogous fin
T_1	hot side internal glass surface temperature
T_2	cold side internal glass surface temperature
V	voltage
V_A	voltage at the connection between the thermistors
V_b	amplified bridge voltage
V_{bo}	zero amplified bridge voltage (corresponds to the null condition)
x	distance along a wire from the resistor core
z	distance from the measurement side surfaces of metering piece and guard

Greek Symbols

ϵ_{eff}	effective emittance of the internal glass surfaces
σ	Stefan-Boltzmann constant ($5.67 \times 10^{-8} \text{ Wm}^{-2}\text{K}^{-4}$)
ΔR	temperature induced change in thermistor resistance
ΔT_{int}	temperature difference between the internal glass surfaces
ΔT_{total}	temperature difference between the hot and cold plate water flows based on the average of the inlet and outlet water temperatures for each plate
ΔT_{w-i}	total temperature difference between the water flows and the internal glass surfaces

2.8 References

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3. The Measurement and Calculation of Radiative Heat Transfer Between Uncoated and Doped Tin Oxide Coated Glass Surfaces

3.1 Introduction

The radiative heat flow across the evacuated space of vacuum glazing occurs between the parallel internal glass surfaces. Thus, this heat flow can be considered to be radiative heat transfer between plane parallel surfaces. It is known that the radiative heat exchange between surfaces is dependent on the spectral, angular and temperature dependencies of the surface emittances. The angular and wavelength dependencies are ignored in the elementary text book treatment of heat transfer [1]. A more sophisticated analysis involves an integration over wavelength of the surface emittance at each angle in order to give the directional total emittance for each surface. These angular dependent emittances are then combined conventionally and integrated over angle [2]. In this chapter, it is shown that this method leads to errors which can be as large as those inherent in the elementary analysis. An exact method is presented for the calculation of the radiative heat transfer between plane parallel surfaces in which both the wavelength and angular dependencies of the surface emittances are combined directly in one integration step. This method gives an exact result for specularly reflecting surfaces, provided that the optical properties of the surfaces are known.

The calculation of the interpane radiative heat transfer across the evacuated space according to the exact method referred to above requires a knowledge of the emittances for the surfaces of interest: uncoated glass and doped tin oxide coated glass. The emittances of these surfaces can be calculated from the corresponding optical properties. Such properties for glass can be easily determined from reflectance measurements and they can be found in the literature. However, the determination of the optical properties for doped tin oxide coated glass requires a significantly more sophisticated approach, and a novel method for calculating these optical properties is presented in this chapter. This method is validated by comparisons with independent measurements of the wavelength and angular dependencies of reflectance, and with thermal measurements of the heat flows with this type of coated glass. The glass coating is a pyrolitically deposited transparent conducting film that is commonly used in the glazing industry. The coated glass is known by a variety of trade names including 'K glass' [3]; this is the brand which is commonly used in vacuum glazing - and which is therefore considered in this study.

3.2 The Calculation of The Radiative Heat Transfer Between Plane Parallel Surfaces

The radiative heat transfer per unit area between two plane parallel grey surfaces of equal area at temperatures T_1 and T_2 can be written as:

$$q = \epsilon_{eff} \sigma (T_1^4 - T_2^4) . \quad (3.1)$$

Only plane parallel surfaces of infinite extent are considered in this paper. For other geometries, the heat transfer must be modified by a radiation shape factor. The quantity ϵ_{eff} is the effective emittance for the two surfaces and it is often written as:

$$\epsilon_{eff} = \left(\frac{1}{\epsilon_{h,1}(T_1)} + \frac{1}{\epsilon_{h,2}(T_2)} - 1 \right)^{-1} , \quad (3.2)$$

where $\epsilon_{h,1}(T_1)$ and $\epsilon_{h,2}(T_2)$ are the hemispherical total emittances of the surfaces at the specified temperatures. This relationship is valid only for surfaces with no wavelength and directional dependence for the emittances. The derivation of the form of this equation is based on a consideration of the net exchange of energy between the two surfaces, and it can be found in introductory text books on heat transfer [1].

According to the ISO 10292 standard [4], the radiative heat transfer between plane parallel glass surfaces should be calculated with a radiative conductance (referred to in the present work as the radiative U-value), which can be derived from a Taylor series expansion of equation 3.1 and is:

$$U_{rad} = 4\epsilon_{eff} \sigma T_{ave}^3 , \quad (3.3a)$$

where T_{ave} is the average of the surface temperatures, and the effective emittance is determined by equation 3.2. In this standard, the hemispherical emittances of equation 3.2 are referred to as "corrected emissivities." The corrected emissivity for uncoated glass is the hemispherical emittance (taken to be 0.837), and that for coated glass surfaces is determined by multiplying the normal emittance by the appropriate coefficient given in the standard. The radiative heat transfer per unit area is calculated from the radiative U-value according to:

$$q = U_{rad}(T_1 - T_2) . \quad (3.3b)$$

For the temperature differences considered in this chapter, the heat flows calculated by using a radiative U-value (equations 3.3a and b) differ by less than 0.1 % from those obtained by using equation 3.1. Thus, the two approaches are equivalent to within this error.

It is instructive to calculate the hemispherical total emittance of a single surface in two steps [5]. Firstly, the directional total emittance, $\epsilon_{\theta,T}(\theta,T)$, is determined by evaluating an integral over wavelength of the directional spectral emittance, $\epsilon_{\lambda,\theta}(\lambda,\theta,T)$, multiplied by the spectral blackbody emissive power at the given temperature, $E_\lambda(\lambda,T)$:

$$\epsilon_{\theta,T}(\theta,T) = \frac{\int_0^\infty \epsilon_{\lambda,\theta}(\lambda,\theta,T) E_\lambda(\lambda,T) d\lambda}{\int_0^\infty E_\lambda(\lambda,T) d\lambda} . \quad (3.4)$$

In the second step, this directional total emittance is integrated over the hemisphere, assuming that the emittance is a function of only the cone angle, θ , (measured from the normal to the surface) of the solid angle of view:

$$\epsilon_h(T) = \int_0^{\pi/2} \epsilon_{\theta,T}(\theta,T) \sin(2\theta) d\theta . \quad (3.5)$$

A commonly used method for calculating the effective emittance for plane parallel glass surfaces follows a similar procedure: the hemispherical total emittances in equation 3.2 are replaced by directional total emittances which are obtained by experimental measurements or by an integral as in equation 3.4. The effective emittance is then evaluated by integrating over angle these directional total emittances:

$$\epsilon_{eff} = \int_0^{\pi/2} \left(\frac{1}{\epsilon_{\theta,T,1}(\theta)} + \frac{1}{\epsilon_{\theta,T,2}(\theta)} - 1 \right)^{-1} \sin(2\theta) d\theta , \quad (3.6)$$

where the temperature dependence of the directional total emittance is assumed to be negligibly small. This equation gives the correct effective emittance for grey surfaces. The effective emittance is then substituted into equation 3.1 in order to obtain the radiative heat transfer. This is the common approach that was used in studies of the emittance properties of glass surfaces by van Nijnatten [2], Silva and Jones [6] and Geotti-Bianchini and Lohrengel [7]. This two step procedure to obtain the effective emittance is apparently justified by the assumption that, in general, the emittances of the surfaces vary only weakly with wavelength. In principle, this seems reasonable since it is known that the emittances of most surfaces depend much more strongly on the angle θ - particularly at the near-grazing incidence ($\theta = \pi / 2$).

Equation 3.2 can be derived from a consideration of the net energy absorbed and reflected by each surface. This equation is correct only in the case where the optical properties of each surface are independent of wavelength and angle. The derivation of this equation suggests an intuitively obvious way of including the spectral and angular dependencies of the emittances for the case of specular surfaces. This is done by defining an effective directional spectral emittance of the form of equation 3.2 at a specific wavelength, λ , and angle, θ , by replacing $\epsilon_h(T)$ with $\epsilon_{\lambda,\theta}(\lambda,\theta,T)$. The overall heat flow from one surface is obtained by an integral over wavelength and over all angles, using the blackbody spectrum at the temperature of the surface. The net heat transfer between the two surfaces is the difference between two such quantities at the respective surface temperatures:

$$q = \int_0^{\pi/2} \int_0^{\infty} \frac{[E_{\lambda}(\lambda, T_1) - E_{\lambda}(\lambda, T_2)]}{\frac{1}{\epsilon_{\lambda,\theta,1}(\lambda,\theta,T_1)} + \frac{1}{\epsilon_{\lambda,\theta,2}(\lambda,\theta,T_2)} - 1} d\lambda \sin(2\theta) d\theta \quad (3.7)$$

Berdahl et al. employ an integration over both wavelength and angle in order to derive an equation for the net heat loss from a covered radiator [8].

In the evaluation of equation 3.7, the range of wavelengths for the integration is determined by the blackbody spectrum. For this work, a range of 1 to 50 μm is appropriate for the temperatures of interest. According to Rubin [9], the hemispherical total emittance of glass is only weakly dependent on temperature ($d\epsilon_h(T)/dT \sim 10^{-4} K^{-1}$), which is also true of most dielectrics. In this study, the directional spectral emittances are calculated on the basis of optical properties

measured at room temperature since the temperature dependence of the directional spectral emittance is so small over the range of temperatures considered (3 to 102 °C).

The two simpler approaches to this problem (equations 3.2 or 3.6 in conjunction with equation 3.1) are presumably more commonly used because data for the emittance of common glass coatings as a function of both wavelength and angle are unavailable or difficult to obtain. The techniques presented below permit an evaluation of the required directional spectral emittances for both uncoated and doped tin oxide coated glass.

3.3 The Calculation of The Directional Spectral Emittance of Uncoated and Coated Glass

Figure 3.1a shows that for radiation incident upon a glass surface at an angle θ , some of the radiation is reflected from, absorbed by, or transmitted through the glass. The directional spectral reflectance, absorptance and transmittance, $\rho(\lambda, \theta)$, $\alpha(\lambda, \theta)$ and $\tau(\lambda, \theta)$, are related by:

$$\rho(\lambda, \theta) + \alpha(\lambda, \theta) + \tau(\lambda, \theta) = 1 . \quad (3.8)$$

Since by Kirchoff's Law, the absorptance equals the emittance, the directional spectral emittance is:

$$\epsilon_{\lambda, \theta}(\lambda, \theta) = 1 - [\rho(\lambda, \theta) + \tau(\lambda, \theta)] . \quad (3.9)$$

The directional spectral reflectance and transmittance can be calculated from the complex refractive index ($\tilde{n} = n + ik$) and from the thickness of the glass [10, 11]. Although not written as such, the complex refractive index is a function of wavelength. The indices n and k for clear float glass, evaluated from reflectance measurements of a sample at 20 °C, are tabulated by Rubin for wavelengths in the infrared [9].

For a film on a thick substrate, the analysis is much more complicated because there is reflectance at each interface, and absorptance and transmittance for each layer. Figure 3.1b shows such a bi-layer system, where the film is doped tin oxide of thickness d_f and the substrate is glass of thickness d_s . The calculation of the directional spectral emittance of the system requires a knowledge of the thickness and complex refractive index for each layer. While the complex refractive index for glass is known from

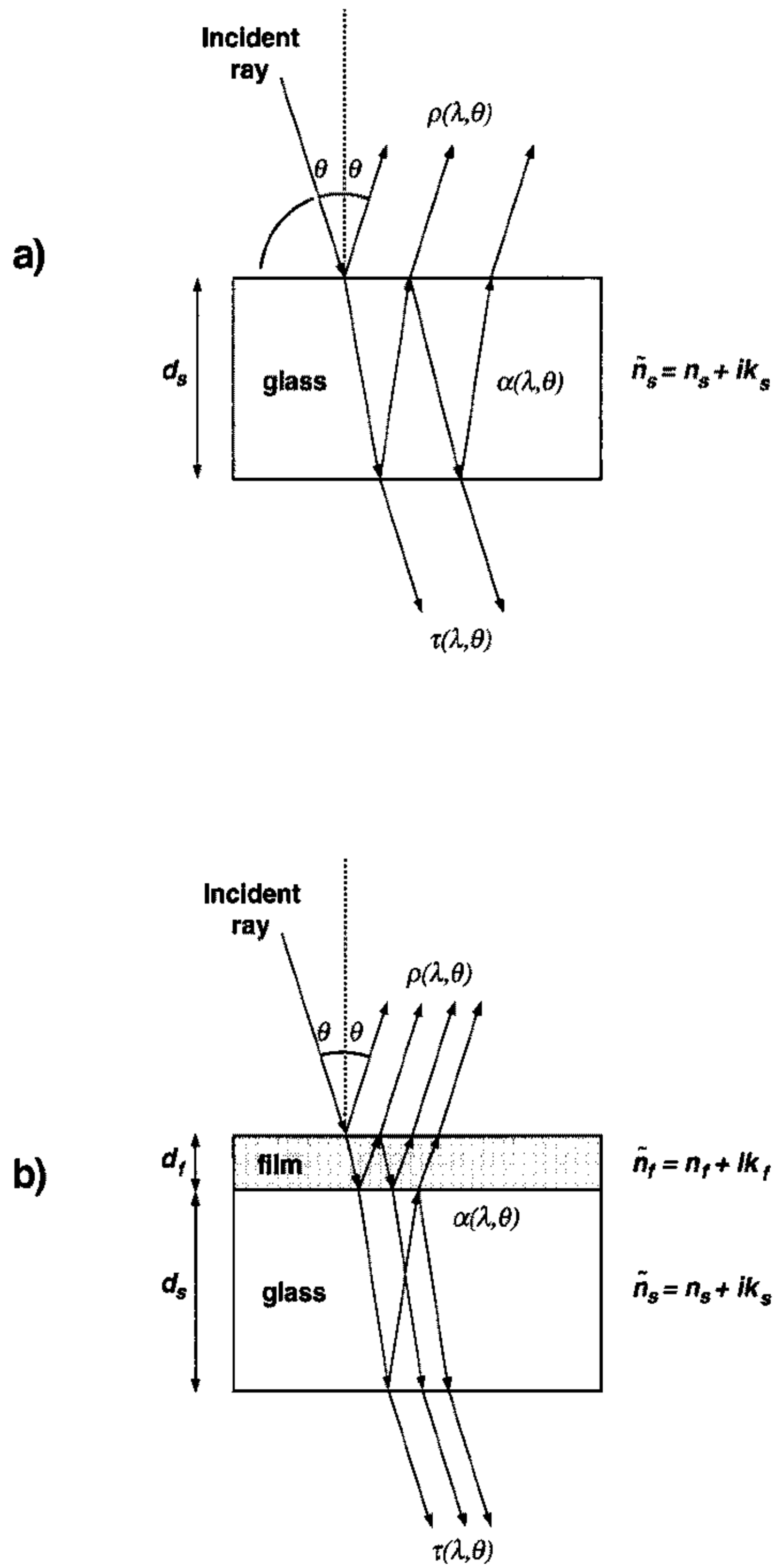


Fig. 3.1: Radiation incident upon a surface: (a) for an uncoated glass surface, and (b) for a thin film on a glass substrate. Only a few ray tracings are shown to demonstrate the multiple specular reflection at each boundary.

Rubin's data, there is little information about that for the particular doped tin oxide film considered in this study.

The complex refractive index for the doped tin oxide film on glass could be determined by applying the Kramers-Krönig formula to reflectance measurements of the film. However, this method is only useful for films for which the thickness is sufficient to ensure that there is a negligible contribution from the substrate to the total reflectance. For the present study of doped tin oxide coated glass, a coating thickness of a few micrometers would be required. However, since the coating is approximately 0.3 μm thick this method can not be used. The refractive index could also be determined from measurements of reflectance and transmittance for the bi-layer system, but since glass has a low transmittance for $\lambda > 5 \mu\text{m}$ [9], this method can not be used over most of the infrared region. Van Nijnatten and Simonis [12] assumed that n and k for doped tin oxide are independent of wavelength in the infrared; they selected values which, when substituted into the Fresnel equations, gave total directional emittances which best agreed with experimental values. This assumption has no true physical basis and it provides a poor fit to experimental emittance data at high angles. A novel method for calculating the complex refractive index for doped tin oxide [13], based on the Drude model [14, 15], is presented below. This method was developed by Qi-Chu Zhang of the University of Sydney, and it is presented in this chapter because the heat flow calculations for K glass are based on results obtained with this method.

The Drude model gives the relationship between the optical and electrical properties of conductors. It can also give the relationship between these properties for semiconductors, such as doped tin oxide, in the infrared. The complex dielectric function for semiconductors, $\tilde{\zeta}(\omega)$, can be described by the Drude model for free carriers as:

$$\tilde{\zeta}(\omega) = \zeta_{\infty} - \frac{\omega_p^2}{\omega^2 + i\omega\omega_{\beta}} , \quad (3.10)$$

where ω is the frequency, ζ_{∞} is the low frequency contribution from the high frequency interband transitions, ω_p is the plasma frequency and ω_{β} is the damping frequency [14]. The complex refractive index is equal to the square root of the complex dielectric function [16]:

$$\tilde{n}_f = n_f + ik_f = \sqrt{\tilde{\zeta}} . \quad (3.11)$$

The directional spectral reflectance and transmittance for doped tin oxide coated glass can be calculated from the thicknesses and complex refractive indices of both the coating and the substrate by a matrix formulation approach derived from Maxwell's equations [10, 11]. The directional spectral emittance is then calculated by equation 3.9. The method for first determining the complex refractive index of the doped tin oxide coating, $\tilde{n}_f$, is presented below.

The spectral reflectance for the doped tin oxide coated glass is measured at an angle of incidence of 10° over 2.5 to 25 μm with an infrared spectrophotometer as described in the next section. The reflectance spectrum for the same angle of incidence is calculated as follows. Initial values are set for ζ_∞ , ω_p and ω_β which are substituted into equation 3.10 to evaluate the complex dielectric function which is then substituted into equation 3.11 to give the complex refractive index. This refractive index, together with that for glass and the measured film and substrate thicknesses, are then used to calculate $\rho(\lambda, 10^\circ)$ and $\tau(\lambda, 10^\circ)$ for the system according to the matrix formulation approach referred to above. The calculated and measured reflectance values are compared by an analysis of the deviations between the results at certain wavelengths:

$$DR = \sqrt{\sum_{j=1}^N \{[\rho_{calc}(\lambda_j, 10^\circ) - \rho_{expt}(\lambda_j, 10^\circ)] / \rho_{expt}(\lambda_j, 10^\circ)\}^2 / N}, \quad (3.12)$$

where DR is the average deviation, j refers to the different wavelengths, N is the total number of wavelengths compared, and ρ_{calc} and ρ_{expt} are the calculated and experimental reflectance values at each specified wavelength. Typically, 10 to 15 wavelengths are selected between 3 and 17 μm . This entire procedure is re-iterated until DR is minimized. For each iteration, the Drude formula parameters ζ_∞ , ω_p and ω_β are re-estimated by a computer subroutine of the downhill simplex method in multidimensions [17]. The final results for the three parameters of the Drude formula are then substituted into equation 3.10. The resulting complex dielectric function is substituted into equation 3.11, from which the complex refractive index for the doped tin oxide coating is evaluated over wavelengths between about 1 and 50 μm . The above procedure can be performed on measurements taken at other angles of incidence, and it was also used to obtain the Drude formula parameters from spectral reflectance measurements at an angle of incidence of 70° . The reflectance spectrum for only one angle of incidence need be considered in order to determine these parameters for a K glass sample.

Since the doped tin oxide coating is highly inhomogeneous through its thickness, an accurate determination of the optical constants requires the use of a complex gradient index model [18]. However, the physical model presented above is based on the assumption that the doping concentration does not vary significantly through the thickness of the glass coating. Therefore, the optical properties for K glass that are determined with this model should be thought of as effective properties. The justification for this simplistic approach rests on the agreement obtained between the calculated and experimental reflectance data over a wide range of wavelengths and angles - as shown below.

3.4 Experimental

Reflectance spectra at near normal incidence (10°) were measured with a Shimadzu IR-470 infrared spectrophotometer over the wavelength range of 2.5 to 25 μm . The calibration standard for this measurement is a freshly evaporated copper film on a glass substrate, the infrared near-normal reflectance of which is taken to be 0.990. This estimation is based on a calibration with a Cary 5 spectrophotometer in the region of 0.25 to 3 μm , and on independent comparative measurements of a polished copper surface [19]. The infrared spectrophotometer measurements were made by Qi-Chu Zhang of the University of Sydney.

High angle reflectance measurements were made with a Fourier transform spectrophotometer (Bruker IFS 66) over the wavelength range of 0.3 to 14 μm . Attached to this spectrophotometer are a polarizer, analyzer, and goniometer [20]. These measurements were performed for s- and p-polarization - i.e. with the polarizer and the analyzer in the s- and p-directions. Results for the two polarizations were averaged, and it is these averaged data that are presented in this paper. Corresponding near normal measurements (at an angle of incidence of 8°) were made with an integrating sphere. The Fourier transform spectrophotometer and integrating sphere measurements were made by Martin Brunotte and Andreas Gombert of the Fraunhofer Institute for Solar Energy Systems at Freiburg, Germany.

The directional total emittance was measured with an emittance goniometer. A schematic diagram of this apparatus is shown in Fig. 3.2. The sample is placed on a hot plate, the temperature of which is measured with a platinum resistance thermometer. The sample is viewed through a rotating chopper aperture by a pyroelectric detector.

This detector can be positioned from 0° to 70° from the normal to the surface of the sample. The signal is processed with a lock-in amplifier and the emittance is calculated on the basis of the intensity of the detected radiation and the solid angle of view. In order to obtain an adequate signal-to-noise ratio, measurements were made at a sample temperature of about 100°C . This apparatus was calibrated by a comparison between measurements made on an uncoated glass sample, and the corresponding calculated results determined by equation 3.4 on the basis of Rubin's optical constants [9]. The accuracy for the emittance measurements is $\pm 0.5\%$. The emittance goniometer measurements were made by the authour, and by Martin Brunotte at the Fraunhofer Institute for Solar Energy Systems at Freiburg, Germany.

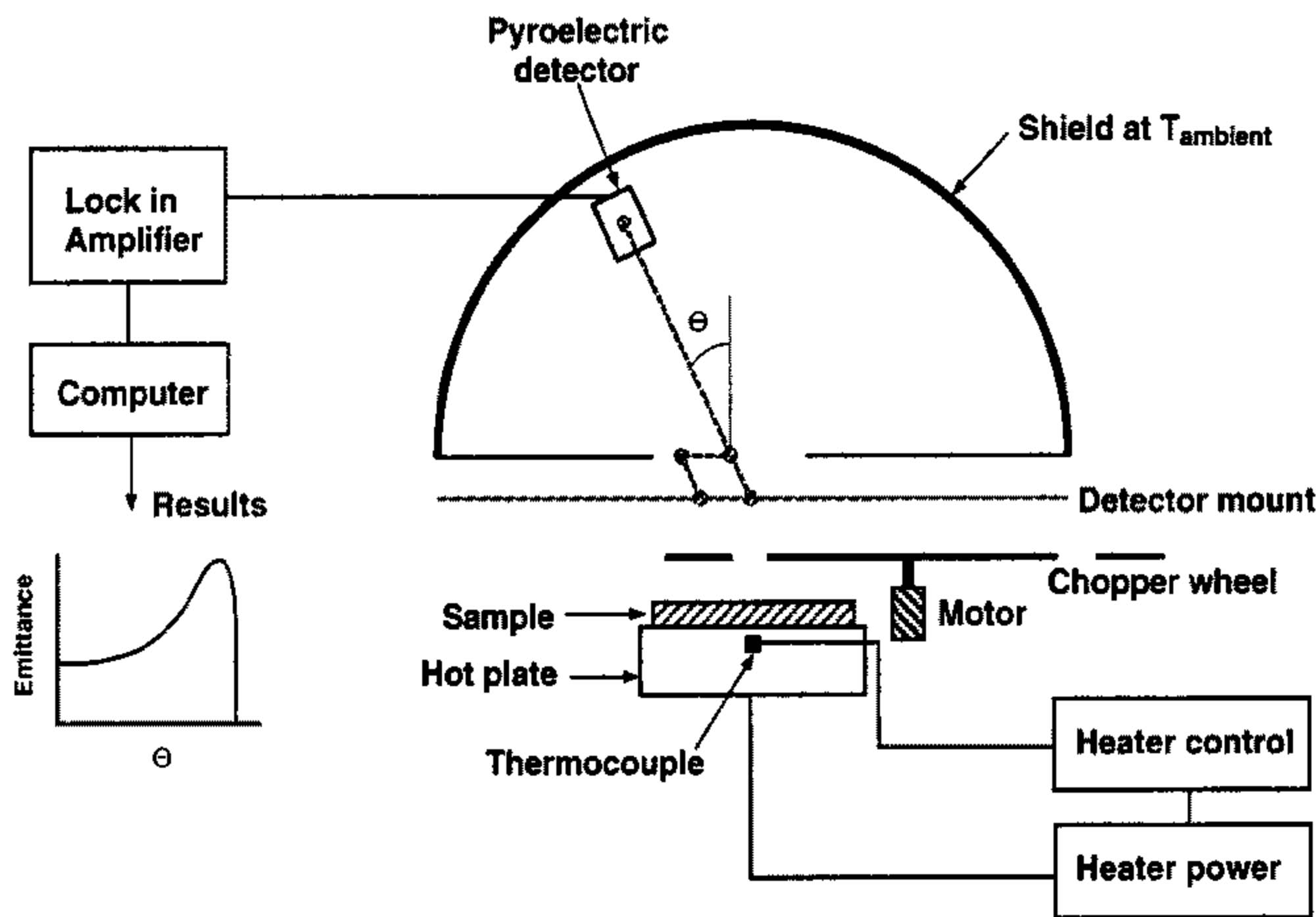


Fig. 3.2: Schematic diagram of the emittance goniometer apparatus. The detector is rotated on pivotable beams (shown as dashed lines) which are located off-center to the view of the sample.

Infrared measurements with the Shimadzu spectrophotometer indicated that there are significant variations ($\sim 10\%$) in the point-to-point reflectance at $2.5\ \mu\text{m}$ of K glass samples. Due to this spatial dependence of the reflectance (and thus in the emittance) of K glass, optical and thermal measurements for determining the calculated and experimental heat flows, respectively, should be made at the same location of the K glass samples. Since the measurement area of the Shimadzu spectrophotometer is about $10^{-4}\ \text{m}^2$, the TCA described in the previous chapter, which has an affected area of $1.45 \times 10^{-4}\ \text{m}^2$, is ideal for obtaining comparative thermal measurements. Thus, all

thermal measurements of the radiative heat transfer across samples with one or two sheets of K glass were made with the TCA described in the previous chapter. A comparative measurement was also made with the TCA of the radiative heat transfer between two sheets of uncoated glass. It is noted that since the reflectance of uncoated glass is highly uniform, variations in the measured radiative heat flow for two sheets of uncoated glass are negligible. All TCA measurements were made with 4 mm thick glass sheets mounted in the standard sample apparatus described in Section 2.4. The TCA measurements were made by the authour.

In order to investigate the relative accuracy of the different methods for calculating the interpane radiative heat transfer, very precise measurements are required. Therefore, a guarded hot plate apparatus, which is similar to the TCA but which has a larger effective measurement area, was employed. Due to the relative size of this area ($1.70 \times 10^{-3} \text{ m}^2$ versus $1.45 \times 10^{-4} \text{ m}^2$ for the TCA), the narrow gap between the metering piece and the guard, and the uniformity and thinness of the epoxy layers between the metal plates and the glass faceplates, the error in the effective measurement area for this apparatus area is less than the error in the effective measurement area for the TCA [13]. Therefore, the per unit area heat flows measured with this apparatus are more accurate than those measured with the TCA ($\pm 0.5 \%$ for this apparatus versus $\pm 1.4 \%$ for the TCA). The measurements with this apparatus were performed with the glass sheets separated by a 1 mm wide evacuated space maintained by a ring of support pillars, and by an air-tight rubber gasket between and around the edges of the sheets. Since the interpane supports must be beyond the large measurement area, the sheets bend somewhat and therefore experience large stresses. Thus, sheets of 6 mm thick tempered glass were used for these measurements. The large area guarded hot plate measurements were made by Christopher Dey of the University of Sydney.

The thickness of the glass was measured with a micrometer, and the thickness of the film was measured with a Tencor P-10 surface profiler. The calculated results for comparison with the TCA measurements are based on the mean dimensions of the samples which were measured with this apparatus: a glass thickness of 4 mm, and a film thickness of $0.33 \mu\text{m}$. The calculated results for comparison with the large area guarded hot plate measurements are for a glass thickness of 6 mm. Due to the opacity of glass at wavelengths greater than about $5 \mu\text{m}$, the hemispherical emittance of uncoated glass thicker than 4 mm is virtually independent of the thickness; modelling results show that the hemispherical total emittances of 4 and 20 mm thick glass differ by only 0.1 %. Therefore, it is valid to compare results for 4 and 6 mm thick glass sheets.

3.5 Results

The high temperature required to form the vacuum glazing edge seal causes a slight increase in the surface emittance of K glass used in this type of glazing. The process by which this occurs is not known. However, it is thought to be related to the significant increase in the surface roughness which is observed in scanning electron microscopy images of heat treated samples of K glass. The high temperature manufacturing process does not affect the surface emittance of uncoated glass. In this section, results are presented for uncoated glass and for the general case of pre-glazing manufacture (i.e. as-received) K glass. There is a similar agreement between the experimental and calculated results for heat treated (i.e. baked) K glass. Some of the results for the specific case of heat treated K glass are presented at the end of this section. Unless otherwise specified, all K glass results of this section refer to as-received samples.

In this section, the error associated with the method of calculating the directional spectral emittance of K glass is discussed. Results are then presented that validate this method of calculation. Estimations of the heat flows between various combinations of uncoated glass and K glass surfaces are given, as calculated by three different methods. A comparison between the calculated heat flows and the experimental data enables conclusions to be drawn about the accuracy of the methods of calculation.

The greatest source of error associated with the calculated emittances for K glass is the error in the reflectance measurements on which they are based. The reference standard for the Shimadzu spectrophotometer is a copper film, the spectral reflectance of which is assumed to be constant in the infrared and is taken to be 0.990. The error for the reference measurement, including the error associated with the assumption of a constant reflectance in the infrared, is about ± 0.005 . For the measurements on K glass, this is only a 0.6 % error in the average reflectance of 0.8, but it represents a 2.5 % error in the corresponding calculated average emittance of 0.2. Thus, for K glass surfaces, the error associated with the calculated emittances, and therefore with the calculated heat flows, is taken to be ± 2.5 %.

Figure 3.3 shows experimental measurements of the spectral reflectance of K glass at an angle of incidence of 10° that were obtained with the Shimadzu infrared spectrophotometer. The spectral reflectance calculated by the Drude formula described above is shown in this figure as a dashed line. The Drude parameters for the calculated results are: $\omega_p = 1.51 \times 10^4 \text{ cm}^{-1}$, $\omega_b = 1.66 \times 10^3 \text{ cm}^{-1}$, and $\zeta_\infty = 7.5$. The measurements obtained with the Shimadzu infrared spectrophotometer at this incidence were

independently confirmed with data from the above mentioned integrating sphere at an angle of incidence of 8°.

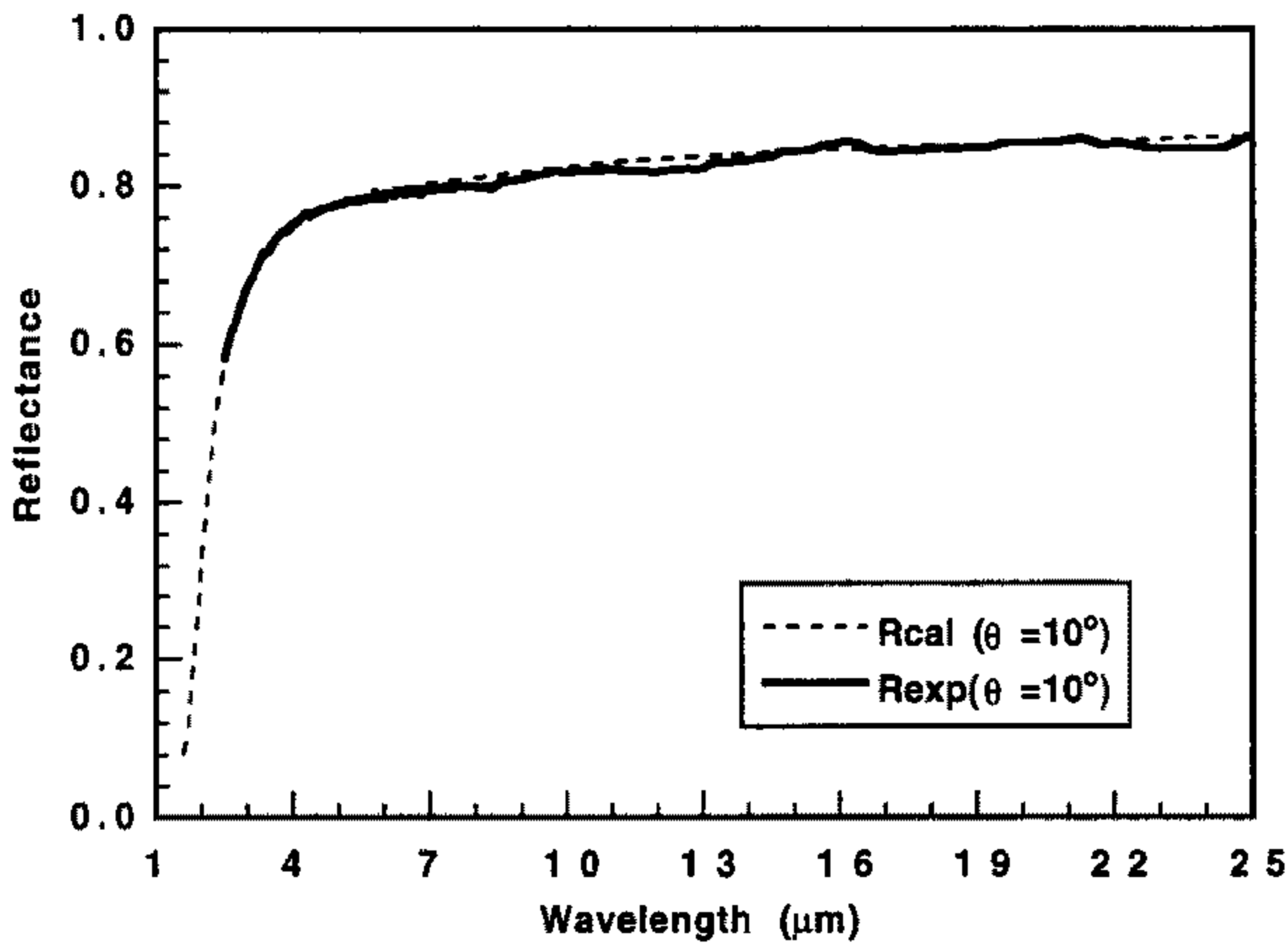


Fig. 3.3: The spectral reflectance for K glass at a 10° angle of incidence. Experimental results measured with the Shimadzu infrared spectrophotometer (full line) are compared with calculated results (dashed line).

Figure 3.4 presents the spectral reflectance for another sample of K glass measured at different angles of incidence using the Fourier transform spectrophotometer. Also shown in this figure, as dashed lines, are results calculated from the Drude formula parameters derived from the measured reflectance spectrum for an angle of incidence of 70° in the manner described above. This comparison is done at high angles of incidence because, as will be shown, the directional dependence of the emittance of K glass is much greater at higher angles. The Drude parameters for the calculated results are: $\omega_p=9.74 \times 10^3 \text{ cm}^{-1}$, $\omega_\beta=7.03 \times 10^2 \text{ cm}^{-1}$, and $\zeta_\infty=3.3$. The parameters for this sample are somewhat different from those for the sample considered for Fig. 3.3. This is probably due to variations in the composition of the film, or in the film manufacturing process; as previously mentioned, the point-to-point emittance of K glass has been observed to vary by as much as 10 %, which indicates that there are significant changes in the nature of the film.

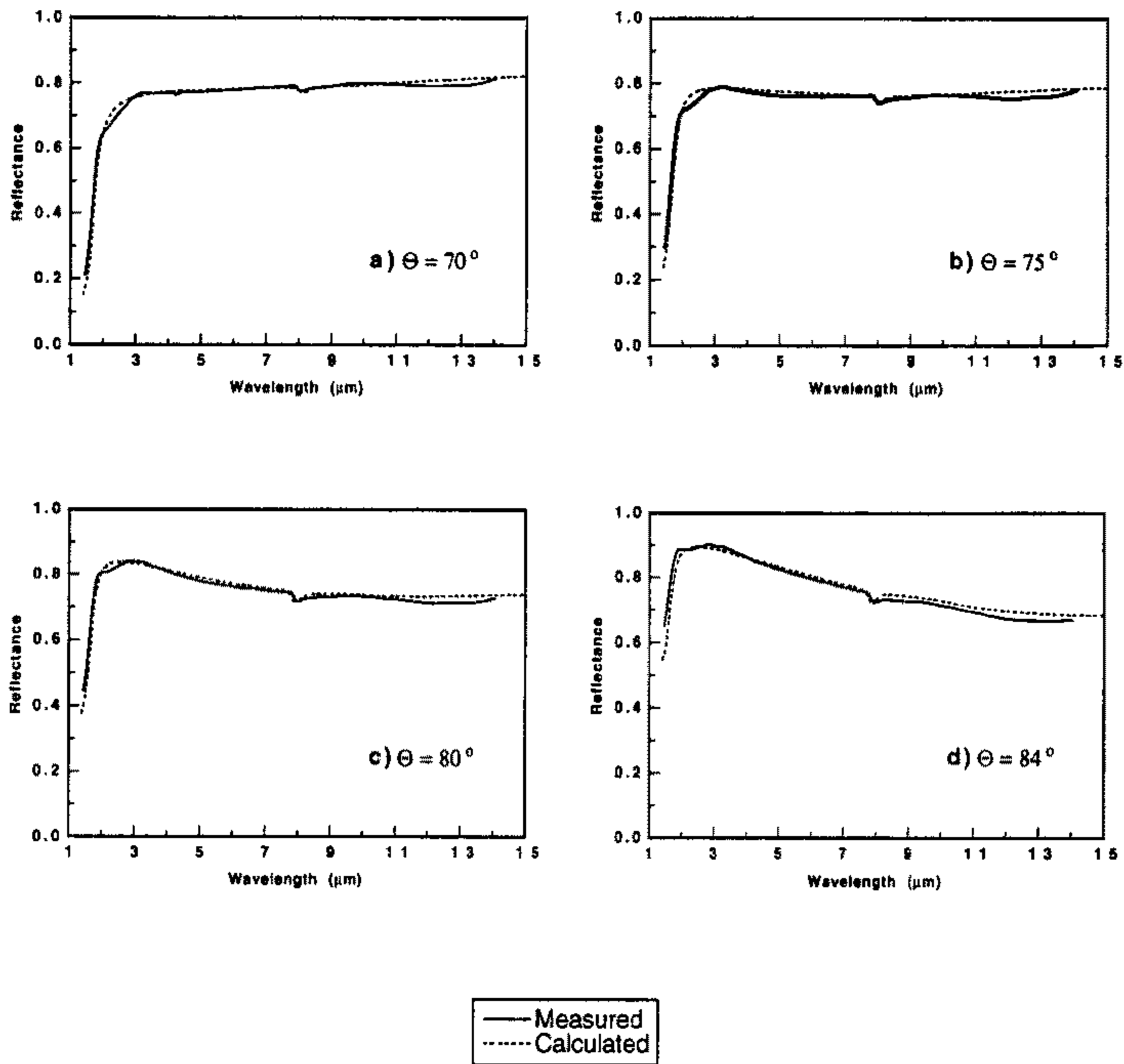


Fig. 3.4: The spectral reflectance for K glass at different angles of incidence, θ . Experimental results measured with the Fourier spectrophotometer (full lines) are compared with calculated results (dashed lines).

In Figs. 3.3 and 3.4, there is a dip in the reflectance at 8 μm which is associated with the change in reflectance of the glass substrate near this wavelength [9]. There are small differences between the two sets of results in Fig. 3.4 at about 2 μm , the origin of which is unknown. However, in both figures, there is generally a very good agreement between the experimental and calculated results. This validates the method of determining the directional spectral reflectance (and thus the emittance) of doped tin oxide coated glass. In particular, the assumption that there is no significant change in the doping concentration through the thickness of the film leads to an adequate representation of the optical properties in the thermal infrared region.

Figure 3.5 presents results for the directional total emittance for a sample of K glass. Experimental data measured with the emittance goniometer are compared with the calculated results for the same location on the same sample of K glass. The calculated results show the existence of a peak at high angles - as predicted by van Nijnatten and Simonis from an extrapolation of their experimental data [12]. Both sets of results in this figure are for a surface temperature of 102 $^{\circ}\text{C}$. The results agree to within the respective errors: $\pm 0.5\%$ for the experimental data, and about $\pm 2.5\%$ for the calculated data. This provides a further validation of the physical model.

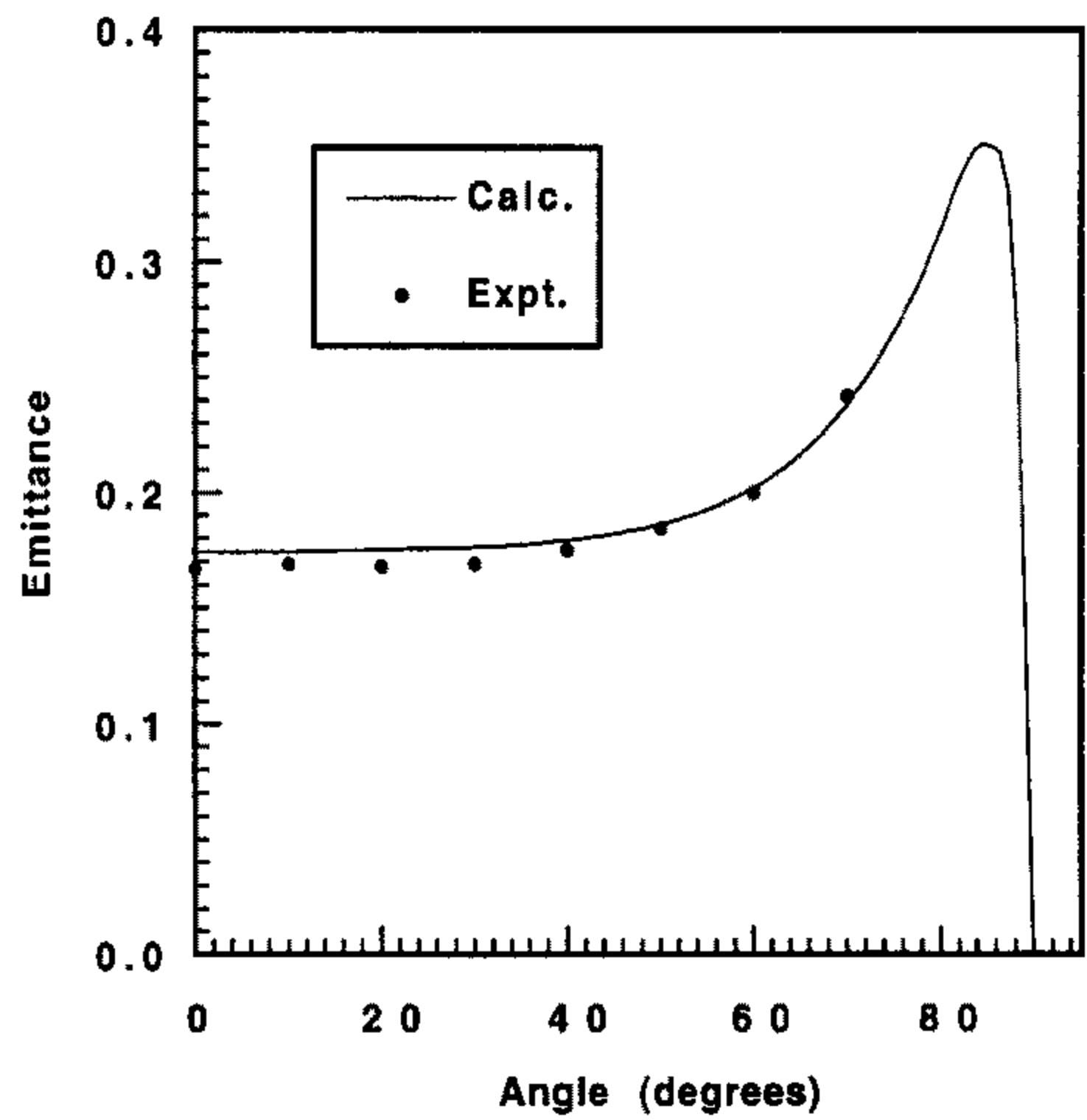


Fig. 3.5: The directional total emittance for as-received K glass. Emittance goniometer measurements (Expt.) are compared with calculated results (Calc.) for a surface temperature of 102 $^{\circ}\text{C}$.

Having established the validity of, and the error associated with the technique for predicting the emittance of K glass, heat transfer results are now considered. Table 3.1 lists the measured and calculated per unit area radiative heat flows between uncoated glass surfaces at the given temperatures. The measurements were made on tempered glass with the large area guarded hot plate apparatus. The calculated heat flows were determined by three different methods. For method 1, the effective emittance is calculated by equation 3.2, using a hemispherical total emittance of 0.838 (which is determined by equations 3.4 and 3.5). This hemispherical total emittance is only 0.1 % greater than that quoted in the ISO standard: 0.837. The effective emittance is then substituted into equation 3.1 to calculate the heat flow. As noted above, this approach gives results which are essentially identical to the ISO method which uses equation 3.2 in conjunction with a radiative conductance. Thus, method 1 is equivalent to the ISO standard. For method 2, the effective emittance is determined by an integration over angle as given by equation 3.6, and this emittance is then substituted into equation 3.1 to calculate the heat flow. For the third method, the heat flow is calculated in one step by equation 3.7, with the integration performed over both wavelength and angle.

The calculated results of Table 3.1 are based on Rubin's optical constants [9]. The error associated with these constants is related to the accuracy of the reference standard used for the reflectance measurements from which the constants were determined. Individual emittances based on these constants are estimated to be accurate to ± 0.5 % [18]. For each of the three methods, the calculation of the heat flow is based on a combination of emittances in an equation which is of a form that is similar to that of equation 3.2. Taking the average emittance for these combinations to be equal to the calculated hemispherical total emittance of 0.84 (± 0.5 %) leads to an error in the calculated heat flows of approximately ± 0.9 %. The error associated with heat flow measurements with the large area guarded hot plate apparatus is estimated to be ± 0.5 %. It is noted that the calculated results are based on Rubin's optical constants for untempered glass, whereas the large area guarded hot plate measurements are made on tempered glass. According to Rubin, all window glass has the same emittance to the third significant figure regardless of tempering, tinting or most other treatments [18]. Thus, it is valid to compare the calculated and measured heat flows of Table 3.1. For the surface temperatures of 4.01/22.81 °C and 34.27/52.68 °C, the experimental results differ significantly from those obtained by methods 1 and 2. However, for both temperature ranges, the experimental results are in good agreement with the results obtained from method 3 (i.e. equation 3.7). Thus it is concluded that, to within the respective errors, equation 3.7 provides results that are more consistent with the measured heat flows than the results obtained by the other two methods. This provides

		Heat flow, q (Wm ⁻²)	
		Surface temperatures 4.01/22.81 °C	Surface temperatures 34.27/52.68 °C
Calculated	Method 1	72.3 ± 0.6	96.2 ± 0.9
	Method 2	73.0 ± 0.7	97.1 ± 0.9
	Method 3	75.4 ± 0.7	101.1 ± 0.9
Measured		75.2 ± 0.4	100.5 ± 0.8

Table 3.1: The radiative heat transfer per unit area between uncoated glass surfaces, as calculated by three different methods. For method 1, the heat flows are calculated on the basis of a simple combination of the hemispherical total emittances (equations 3.2 and 3.1); these emittances are calculated by equations 3.4 and 3.5. For method 2, the heat flows are based on an integration over angle of the directional total emittances (equations 3.6 and 3.1). For method 3, the heat flows are calculated by an integration over wavelength and angle of the directional spectral emittances and the spectral blackbody emissive powers (equation 3.7). The experimental measurements were made with the large area guarded hot plate. The results are for inner glass surface temperatures of 4.01/22.81 °C and 34.27/52.68 °C.

a validation of the use of this equation for calculating the radiative heat transfer between glass surfaces.

Measurements were also made of the radiative heat transfer between surfaces of K glass. As discussed above, in order to reduce the effect of the point-to-point variations in the emittance of K glass, these measurements were made with the TCA. An accurate estimation of the effective measurement area for this apparatus can be obtained by a comparison of the measured and calculated absolute heat flows for a reference standard of a well known thermal conductance. As noted in Section 2.4, the reference standard selected for this calibration procedure is two sheets of uncoated (and untempered) glass separated by an evacuated space - i.e. the radiative heat transfer between two uncoated glass surfaces. This was chosen as the reference because the glass-to-glass radiative heat transfer can be accurately calculated on the basis of Rubin's data [9]. This calculation was done with equation 3.7 because, as shown by the results of Table 3.1, heat flows determined with this equation are in consistent agreement with the experimental results. This calibration of the effective area has the effect of defining the measured radiative heat flow between two uncoated glass surfaces to be precisely equal to that predicted by equation 3.7.

It should be noted that the experimental and calculated heat flows used for the above calibration of the TCA correspond to different types of clear float glass which are, respectively, that made by Pilkington (Australia) Limited, and that of a different manufacture studied by Rubin. A comparison of the near-normal reflectance spectra of these two glasses shows that the hemispherical total emittances differ by, at most, 0.5 %. This is equal to the error associated with the emittance of Rubin's glass, and it is less than the error associated with the emittance of the Australian glass (± 0.6 %). It is therefore appropriate to use Rubin's data to calculate heat flows for comparison with the heat flow measurements on the Pilkington (Australia) Limited product.

Table 3.2 lists the measured and calculated per unit area heat flows for all possible combinations of uncoated glass and K glass. The experimental results were obtained with the TCA, and the calculated results were obtained by the same three methods described above. For method 1, the effective emittance was calculated from equation 3.2. The hemispherical total emittances for this equation were calculated from a combination of equations 3.4 and 3.5, based on the appropriate optical parameters; as explained above the optical parameters for uncoated glass are from Rubin [9], and the optical parameters for K glass are determined from the Drude model on the basis of

Sample		Surface Temperatures (°C)		Heat flows (Wm ⁻²)				Emittance from $q = \epsilon_{\text{eff}} \sigma (T_1^4 - T_2^4)$ $\epsilon_{\text{eff}} = (\epsilon_1^{-1} + \epsilon_2^{-1})^{-1}$		
				Calculated			Measured			
				Method 1	Method 2	Method 3		ϵ_{eff}	ϵ_1	ϵ_2
Side 1	Side 2	T ₁	T ₂							
Glass	Glass	22.72	3.75	72.1 ± 0.6	72.8 ± 0.6	75.2 ± 0.7	75.2 ± 1.1	0.746	0.855	0.855
K glass	K glass	23.12	3.33	12.6 ± 0.3	11.5 ± 0.3	12.1 ± 0.3	11.9 ± 0.2	0.113	<u>0.203</u>	0.203
K glass	Glass	23.08	3.37	20.0 ± 0.5	19.4 ± 0.5	20.1 ± 0.5	19.5 ± 0.3	0.186	<u>0.192</u>	0.855

Table 3.2: The radiative heat transfer per unit area between various combinations of uncoated glass and as-received K glass surfaces, for the indicated glass surface temperatures. The calculated results were determined by the same three methods described for Table 3.1. The experimental measurements were made with the TCA. The effective and hemispherical emittances were determined by equations 3.1 and 3.2, respectively.

optical measurements of the glass samples. All thermal and optical measurements were made at the same locations on the respective samples of glass.

As discussed above, the heat flows shown in Table 3.2 for two uncoated glass surfaces as predicted by methods 1 and 2 are clearly inconsistent with the experimental result. The agreement between this experimental heat flow and that obtained by method 3 is due to the fact that the latter was used to calibrate the area with which the former was determined; as previously mentioned, as a result of the calibration procedure for the apparatus, these two heat flows are defined to be equal.

The results of Table 3.2 show that, for two K glass surfaces, only the heat flows calculated by methods 2 and 3 are in agreement with the experimental result, and that the heat flow predicted by method 1 and the experimental result differ by at least 2%. For this combination of surfaces, the effective emittance is significantly different from the hemispherical total emittance of either individual surface. For the heat flow between K glass and glass surfaces, the predictions by all of the methods are in agreement with each other, and they are consistent with the experimental result. This is as expected since, for this combination of surfaces, the effective emittance is quite close to the hemispherical total emittance of K glass. Relative to the K glass, the glass is a reasonably black surface. For the heat flow between a low emittance surface and a perfectly black surface (for which the hemispherical total emittance is 1), all methods give essentially identical results.

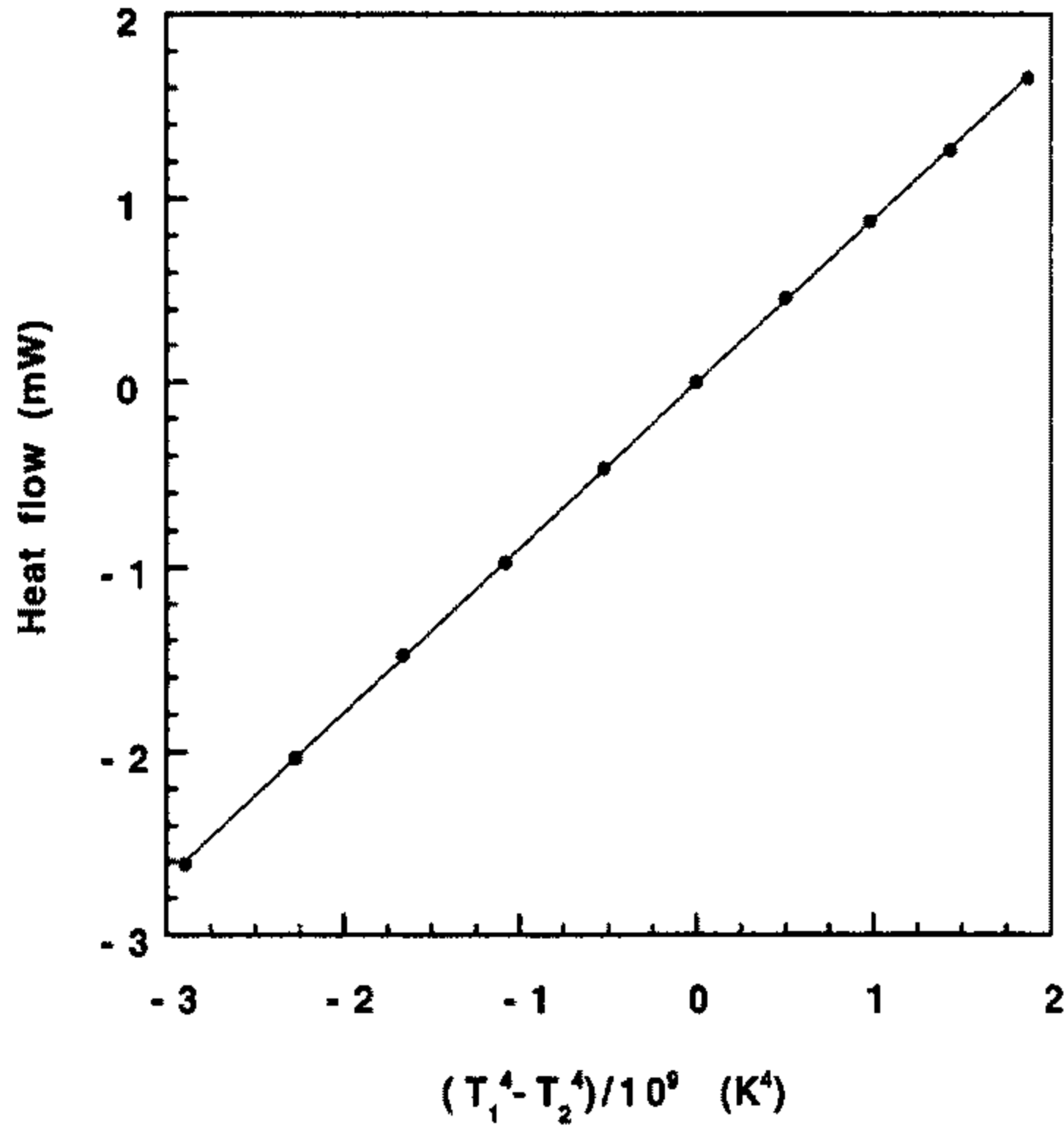
Table 3.2 also presents emittances for the same three combinations of surfaces considered above. The effective emittances are calculated from the given heat flows and temperatures (which are listed in Table 3.2) by equation 3.1. The hemispherical total emittances are then calculated from these effective emittances by using equation 3.2. The glass and K glass hemispherical total emittances are calculated from the results for glass/glass and K glass/K glass samples and they are found to be 0.855 and 0.203, respectively. For the K glass/glass arrangement, the hemispherical total emittance for K glass is determined from equations 3.1 and 3.2 using that for glass of 0.855. The result of 0.192 is 5 % lower than that obtained from equations 3.1 and 3.2 for the sample of two sheets of K glass. Since this discrepancy is much greater than the experimental error, it is apparent that equations 3.1 and 3.2 (and thus the basis for the ISO method) do not provide a reliable approach for calculating the radiative heat transfer between these surfaces. The value of 0.855 given above for the hemispherical total emittance of uncoated glass differs significantly from the value of 0.837 given in

the ISO standard, and by Rubin [9]. This is simply a reflection of the error that is intrinsic in the use of equation 3.1 to calculate the radiative heat transfer.

The conclusions drawn from the results of Table 3.2 about the validity of the different methods of calculation depend on the linearity with respect to temperature of the heat flows measured with the TCA. This linearity can be demonstrated by comparing the measured and calculated heat flows for different temperatures across the sample. As first noted in Section 2.5.4, this comparison is best made with a residual analysis. A quadratic function passing through the origin is fitted to the data in a plot of the radiative heat transfer versus $T_1^4 - T_2^4$. The deviation of each data point from the linear portion of this quadratic fit is then calculated and plotted as a function of $T_1^4 - T_2^4$ in order to show the departures from linearity. Figure 3.6a is a plot of the null power measured with the TCA (i.e. the absolute radiative heat transfer) between two as-received K glass samples measured with the TCA as a function of $T_1^4 - T_2^4$. The data points can be approximated by a quadratic fit which passes through the origin. The linear portion of such a fit is shown in this figure, and it can be seen that the data points exhibit a downward trend from this reference line. The same analysis was performed for heat flows calculated by equation 3.7 on the basis of the Drude model and optical measurements. The deviations from the linear portion of a quadratic fit to the calculated data (i.e. the predicted deviations) are also presented in Fig. 3.6b. The experimental and calculated results of Fig. 3.6b agree to within the resolution of the measured heat flows: $\pm 10 \mu\text{W}$. Comparable agreement was obtained for the same analysis of the radiative heat flows between two uncoated glass surfaces, and between K glass and uncoated glass surfaces. Similar agreements have also been obtained for the same combinations of surfaces with heat treated K glass. It should be noted that the measured deviations are near the limit of the resolution of the apparatus. However, there is generally a good agreement between the measured and calculated results. This demonstrates that the null power of the TCA is indeed proportional to the heat flow through the sample over a wide range of temperatures.

Some results for heat treated K glass are now considered. The heat treated samples were exposed to a temperature of 500 °C for one hour in order to approximately simulate the high temperature conditions during the manufacture of vacuum glazing. Figure 3.7 presents data for the directional total emittance of K glass for heat treated K glass samples. The experimental measurements were made with the emittance goniometer and the calculated results were determined from the Drude model and optical measurements according to the method given above. A comparison with the

a)



b)

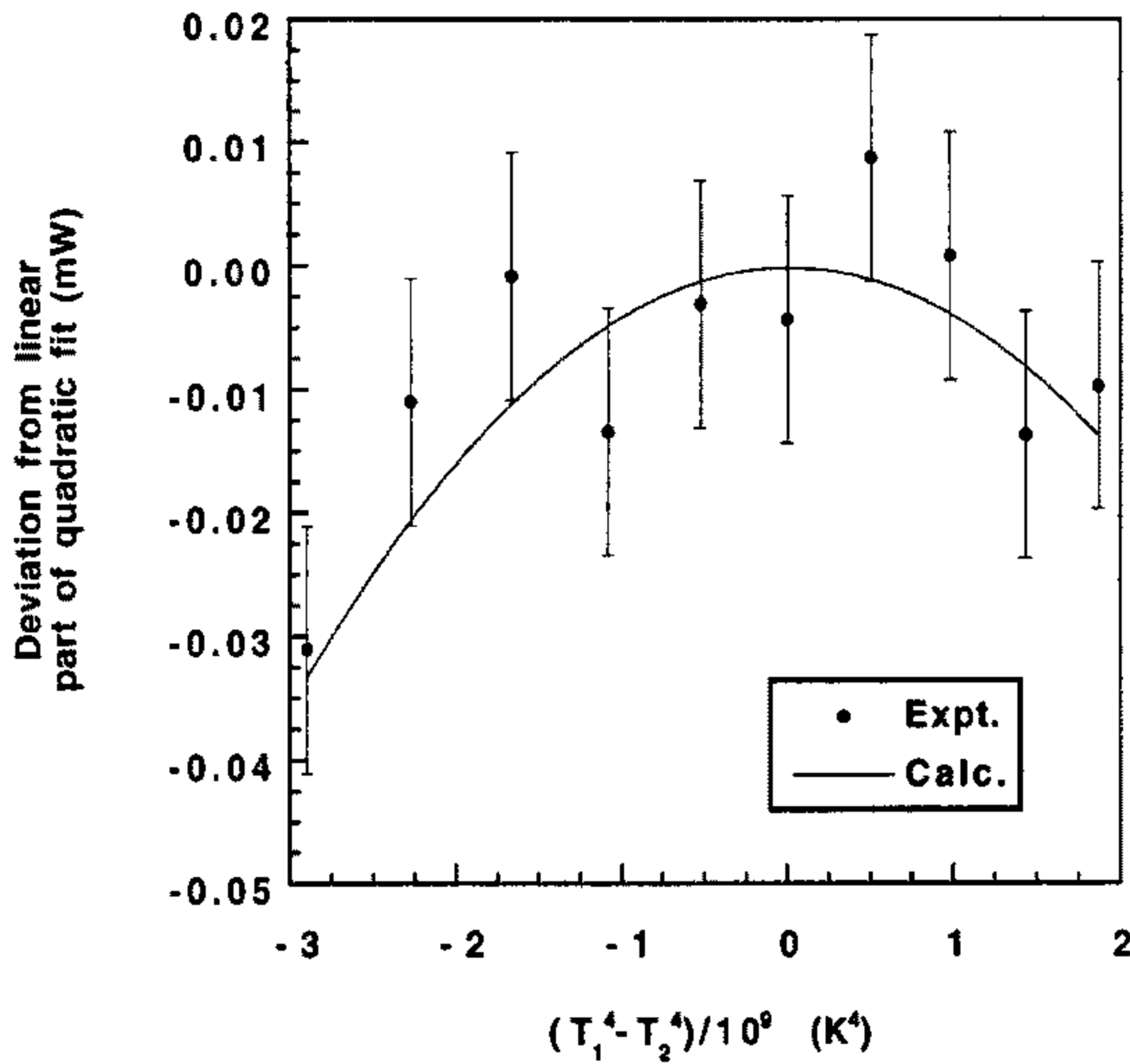


Fig. 3.6: (a) A plot of the measured TCA null power (i.e. the radiative heat transfer) as a function of $T_1^4 - T_2^4$ for two sheets of as-received K glass mounted in the standard sample apparatus. Data points are shown with the linear portion of the quadratic fit to the data which passes through the origin.
 (b) Deviations of the measured null powers of (a) from the linear portion of the quadratic fit which passes through the origin (Expt.). Also shown are the predicted deviations (Calc.).

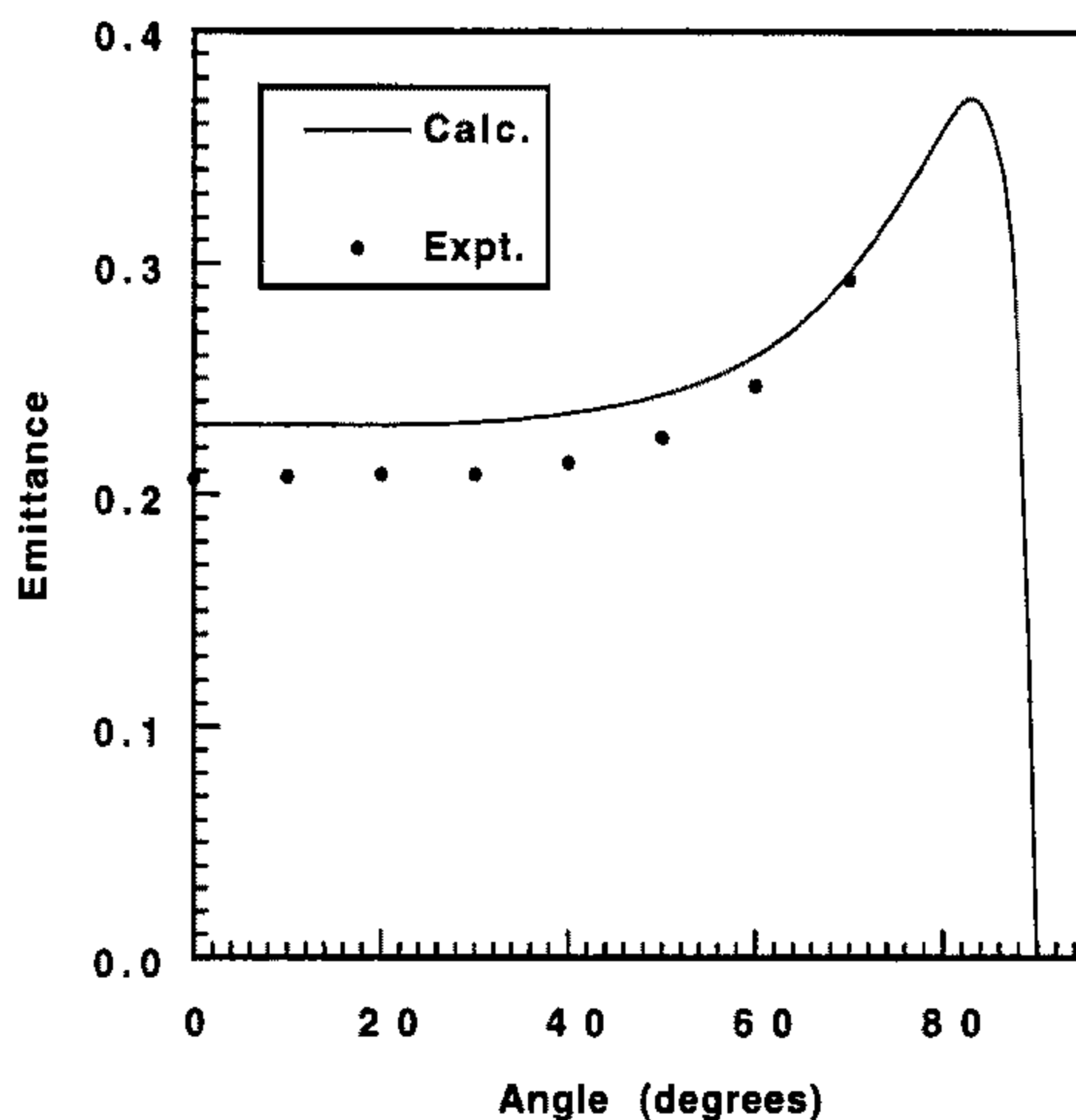


Fig. 3.7: The directional total emittance for heat treated K glass. Emittance goniometer measurements (Expt.) are compared with calculated results (Calc.) for a surface temperature of 102 °C.

results of Fig. 3.5 shows that the heat treatment of K glass results in a larger directional total emittance.

Table 3.3 compares some of the results of Table 3.2, which are for as-received K glass, with the corresponding results for heat treated K glass. This table shows that exposure to the high temperature glazing manufacturing process can increase the hemispherical total emittance of K glass by as much as 17 % of the value for the as-received product.

Sample		Measured heat flow q (Wm ⁻²) (+/- 1.4 %)	Surface Temperatures (°C)		Emittance from q = ε _{eff} σ(T ₁ ⁴ -T ₂ ⁴) ε _{eff} = (ε ₁ ⁻¹ + ε ₂ ⁻¹ -1) ⁻¹		
Side 1	Side 2		T ₁	T ₂	ε _{eff}	ε ₁	ε ₂
Glass	Glass	75.2	22.72	3.75	0.746	0.855	0.855
K glass	K glass	11.9	23.12	3.33	0.113	<u>0.203</u>	0.203
K glass	Glass	19.5	23.08	3.37	0.186	<u>0.192</u>	0.855
Baked K glass	Baked K glass	14.1	23.16	3.34	0.133	<u>0.235</u>	0.235
Baked K glass	Glass	22.5	23.11	3.40	0.214	<u>0.222</u>	0.855

Table 3.3: Some of the results of Table 3.2, which are for as-received K glass, compared with the corresponding results for heat treated (i.e. baked) K glass.

3.6 Conclusion

It has been shown that the Drude model, which gives the relationship between the optical and electrical properties of conductors, provides an accurate description of the infrared reflectance of a pyrolytically deposited transparent conducting layer of doped tin oxide, such as that on Pilkington K glass. Measurements of the near-normal reflectance of K glass for wavelengths of between 3 and 17 μm are used to obtain a best fit for the three parameters of the Drude model. This model then provides an estimation of the optical properties in the infrared which can be used to determine the directional spectral emittance. The calculated spectral reflectance data are in excellent agreement with experimental results measured to 84° from the normal to the glass surface. There is also a good agreement between calculated results for the directional total emittance of K glass, and measurements made up to an angle of incidence of 70° .

The radiative heat transfer between plane parallel surfaces of uncoated glass has been calculated using estimations of the effective emittance determined by two well known methods: a simple combination of the hemispherical total emittances (the ISO standard method), and an integration over angle of the directional total emittances. When compared with accurate experimental measurements, these methods result in heat flows which underestimate the heat flow by as much as 5 % and 4 % respectively. An exact method has been presented in which the wavelength and angular dependencies of the surface emittances are combined in a double integration over wavelength and angle; the blackbody spectra at the defined surface temperatures are also included in this integration. To within the range of the respective errors (± 1.4 %), the results of this exact method agree with accurate large area guarded hot plate measurements of the radiative heat flow between plane parallel uncoated glass surfaces. Smaller area measurements made with the TCA of heat flows between combinations of uncoated and coated glass surfaces are, to within the estimated uncertainties, in agreement with results calculated by this exact approach and by the second of the two approaches described above - the one involving an integration over angle. These small area measurements indicate that, while the ISO standard method for calculating the radiative heat transfer between uncoated glass surfaces is in error, it gives quite reasonable results for other combinations of surfaces (to within 2% of the limits of the uncertainties in the calculated and experimental results). The principal source of error in this comparison with coated glass surfaces is the reflectance of the copper film used as the reference for the infrared reflectance measurements from which the parameters of the Drude model are determined.

It is concluded that the most accurate method for calculating the radiative heat transfer between the internal glass surfaces of vacuum glazing involves an integration over both wavelength and angle - i.e. the above exact method. This calculation requires a knowledge of the directional spectral emittances of the surfaces, which can be determined from the appropriate optical constants. For uncoated glass, optical constants determined from reflectance measurements can be found in the literature. For K glass, the optical constants can be determined on the basis of the Drude model, as described above. For most calculations in this thesis, it is sufficient to calculate the interpane radiative heat transfer on the basis of the radiative U-value (i.e. the radiative conductance) which is given by equation 3.3a. However, for crucial calculations such as the prediction of the expected radiative heat transfer for the calibration of the TCA, the above exact approach must be employed - as was done in the previous chapter.

3.7 List of Symbols

A	surface area
d_f	film thickness
d_s	substrate thickness
DR	average deviation of reflectance
$E_\lambda(\lambda, T)$	spectral blackbody emissive power
k	extinction coefficient
N	total number of wavelengths
n	refractive index
$\tilde{n}_f$	complex refractive index of film
$\tilde{n}_s$	complex refractive index of substrate
q	radiative heat transfer per unit area
T	surface temperature
T_{ave}	average temperature of two emitting surfaces
U_{rad}	radiative U-value (or radiative conductance)

Greek symbols

$\alpha(\lambda, \theta)$	directional spectral absorptance
$\epsilon_h(T)$	hemispherical total emittance
ϵ_{eff}	effective emittance
$\epsilon_{\theta, T}(\theta, T)$	directional total emittance
$\epsilon_{\lambda, \theta}(\lambda, \theta, T)$	directional spectral emittance
λ	wavelength
θ	angle of incidence from the normal to the surface

$\rho(\lambda, \theta)$	directional spectral reflectance
σ	Stefan-Boltzmann constant
$\tau(\lambda, \theta)$	directional spectral transmittance
ω	frequency
ω_p	plasma frequency
ω_β	damping frequency
$\tilde{\zeta}(\omega)$	complex dielectric function
ζ_∞	contribution from high frequency band transitions

Common Subscripts

<i>ave</i>	average
<i>eff</i>	effective
<i>f</i>	pertains to film
<i>h</i>	hemispherical
<i>p</i>	pertains to plasma
<i>j</i>	wavelength number
<i>rad</i>	radiation
<i>s</i>	pertains to substrate
<i>1,2</i>	denote surfaces for heat transfer
θ, t	directional total quantity
λ, θ	directional spectral quantity

Superscript

indicates complex quantity

3.8 References

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4. Heat Conduction Through The Support Pillars

4.1 Introduction

The support pillars in a vacuum glazing represent thermal bridges between the glass sheets which are otherwise, except at the edges, thermally well isolated. Therefore, there is a conductive heat transfer between the glass sheets via the pillars. This heat transfer represents a significant fraction of the total center-of-glass heat flow across the glazing and thus it is important to be able to quantify it, and to know how to reduce it. Since the conduction associated with each pillar can be expected to be somewhat localized and non-uniform, it might seem that such an analysis would require the consideration of all of the pillars in the pillar array. However, sufficiently accurate results can be obtained with a much simpler analytic approach based on an equivalent uniform U-value for the entire array which is calculated from the predicted thermal resistance for a single pillar. This method is validated by comparisons with experimental and finite-element modelling results for the pillar thermal resistance and the effects of pillar conduction on the external glass surface temperature.

The pillars have another, and more subtle, effect on the total heat transfer across the glazing. The heat flow through each pillar alters the temperatures of the internal glass surfaces in the immediate vicinity of the pillar, thereby reducing the difference between the temperatures of these surfaces. Since the inter-pane radiative heat transfer is dependent on this temperature difference, the pillar conduction also reduces the local radiative heat flow between the glass sheets. The influence of this effect on the overall thermal performance of the glazing is investigated in order to determine if a significant correction should be applied to total center-of-glass U-value.

4.2 The Pillar Thermal Resistance

4.2.1 Analytic Estimation

The geometry to be considered in studying the heat flow through a pillar is shown in Fig. 4.1: a single cylindrical pillar between two sheets of soda-lime glass. At this point, internal radiative heat transfer and heat transfer with ambient are not considered. Thus, the internal glass surfaces not in contact with the pillar are defined to be adiabatic, and the external glass surfaces are at the specified temperatures of T_{warm} and T_{cold} . The

emittance of the pillar material is assumed to be so small that the pillar surface which is exposed to the vacuum can also be defined to be adiabatic.

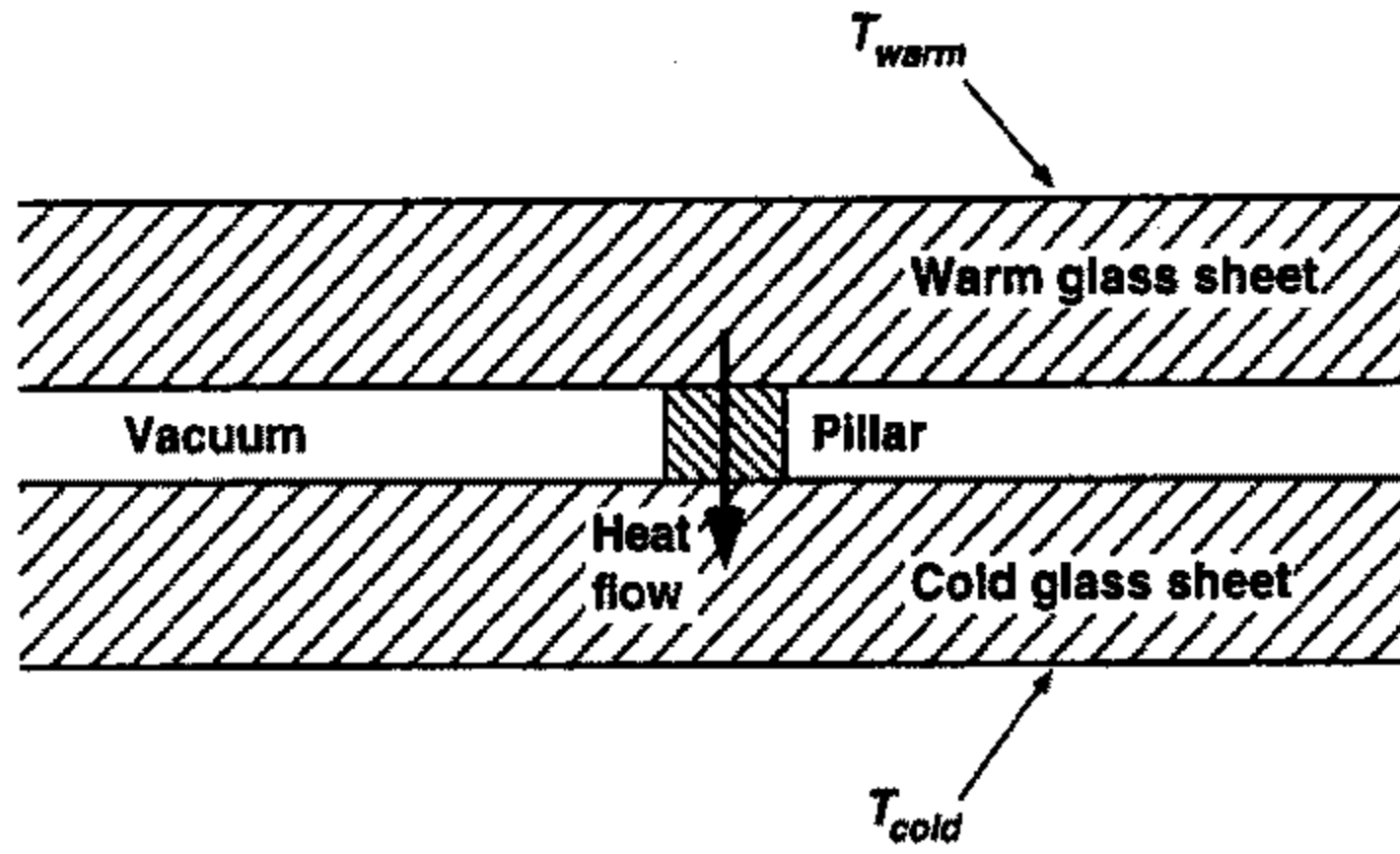


Fig. 4.1: A support pillar acts as a thermal bridge between the two glass sheets which are otherwise, except at the bonded edges (not shown), thermally well isolated.

The heat transfer through a single pillar is a function of the temperature difference between the two external glass surfaces, and of the thermal resistances associated with the nearly parallel heat flow through the pillar and the non-uniform (i.e. spreading) heat flow through the glass sheets close to the points of contact with the pillar. The non-uniform heat flow arises from the fact that the pillar presents a constriction of a very small size to the heat flow between the glass sheets. It has been shown that if the glass sheets are assumed to be of semi-infinite thickness and extent, then the total thermal resistance can be approximated by the series combination of the resistance associated with the spreading heat flow in the glass, $1/2k_{glass}a$, and of the resistance associated with the plane parallel heat flow in the pillar itself, $h/k_{pillar}\pi a^2$ [1]:

$$R_{pillar} = \frac{1 + \frac{2k_{glass}h}{k_{pillar}\pi a}}{2k_{glass}a}, \quad (4.1)$$

where k_{glass} and k_{pillar} are the thermal conductivities of the glass and pillar, and a and h are the radius and height of the pillar. The pillars used in vacuum glazing are made of metallic or ceramic materials for which the thermal conductivities are much greater than the thermal conductivity of glass (i.e. $k_{pillar} \gg k_{glass}$). Thus, the pillar thermal resistance can be approximated on the basis of just the resistance for the spreading heat flow in the glass sheets [1]:

$$R_{pillar} = 1/2k_{glass}a . \quad (4.2)$$

This is equivalent to assuming that the thermal resistance for the heat flow in the pillar is much smaller than the resistance for the spreading heat flow in the glass. Strictly speaking, the thermal resistance given by equation 4.2 refers to the pillar-glass system because it governs the heat flow through both the pillar and the glass. However, for the sake of simplicity, it is referred to as the pillar thermal resistance. As shown in Appendix 3, this relationship can be derived from the heat conduction shape factor given by Holman for a situation which is similar to the one presented in Fig. 4.1 [2]. This is also the experimentally validated equation obtained by Holm for the electrical analogue to this problem: the flow of electrical current between two conductive sheets via a small resistive constriction with a circular cross-section [3]. Both the thermal and electrical derivations of equation 4.2 are also based on the assumption that the two conducting sheets are of semi-infinite thickness and extent.

As will be shown, for the typical pillar separations in vacuum glazing, it may be assumed that the heat conduction through each pillar is independent of that through the others. It is therefore possible to determine the overall U-value of the entire regular pillar array from the U-value for a single pillar. A square unit cell of the array has a pillar at each corner. Since these corner pillars are shared among adjacent cells, each cell contains quarter-sections of the four corner pillars - or the equivalent of one pillar. Alternatively, a unit cell can be defined to be centered around a single pillar. Regardless of the definition, there is one pillar per unit cell and the area of each cell is equal to the square of the pillar separation, λ . Therefore, since the pillar heat flows are assumed to be independent, the heat flow associated with a single pillar passes through a section of glass of area λ^2 . Since the U-value is equal to the inverse of the product of the thermal resistance and the area over which the heat transfer occurs, the U-value for this heat flow is $2k_{glass}a / \lambda^2$. Since the pillars are uniformly distributed within the pillar array, the overall U-value for the array, U_{array} , is the same. Thus the glass-to-glass center-of-glass pillar array U-value is:

$$\begin{aligned} U_{array} &= 1 / R_{pillar} \lambda^2 \\ &= 2k_{glass}a / \lambda^2 . \end{aligned} \quad (4.3)$$

Alternatively, this expression can be derived by considering the pillar array U-value to be equal to the product of the inverse of the thermal resistance for one pillar, and the number of pillars per unit area. For the typical pillar radius and separation of

0.125 and 25 mm, and for a glass thermal conductivity of $1.0 \text{ Wm}^{-1}\text{K}^{-1}$ (as given in Appendix 1 for soda-lime glass), $U_{array}=0.40 \text{ Wm}^{-2}\text{K}^{-1}$. The typical glass-to-glass radiative U-value for a glazing with two sheets of baked K glass, and for the American Society for Testing and Materials (ASTM) standard glazing test conditions (given below) is about $0.70 \text{ Wm}^{-2}\text{K}^{-1}$. Thus, for this case, the pillar array U-value represents 36 % of the total glass-to-glass, center-of-glass U-value of $1.10 \text{ Wm}^{-2}\text{K}^{-1}$. Equation 4.3 shows that the pillar array component of the total U-value can be reduced by decreasing the pillar radius, and/or by increasing the pillar separation.

4.2.2 Experimental Results

In order to validate equation 4.2, the thermal resistances of pillars of different cross-sectional areas were determined from measurements made on vacuum glazing samples with the thermal conductance apparatus described in Chapter 2. The procedures given in Sections 2.5.2 and 2.5.3 were used to discern between the heat flows due to radiative heat transfer and pillar conduction, and the pillar thermal resistance was calculated on the basis of the latter heat flow. The widths of the pillars were measured under magnification to an accuracy of about $\pm 0.02 \text{ mm}$ with a scale incorporated into a microscope eyepiece. Pillars made of Inconel 718, nickel, and alumina (a ceramic) were studied, with each type of pillar corresponding to a separate glazing sample. The respective glazing sample numbers are 569, 322 and 469. The first two samples consist of two sheets of 4 mm thick uncoated glass, and the third sample consists of two sheets of 4 mm thick K glass. The metallic pillars are approximately circular in cross-section and vary in radius, while the alumina pillars have a square cross-section of only one dimension: 0.20 mm. The heights of the Inconel, nickel and alumina pillars are, respectively, about 0.11, 0.15 and 0.13 mm.

Since each thermal resistance is based on two measured heat flows - the radiative flow alone, and the radiative flow in combination with the pillar conduction - the percentage error associated with the resistance is greater than that for either measurement. The error for the alumina pillars reflects the variation in the point-to-point emittance of K glass ($\sim 10 \%$ as mentioned in Section 3.4). As there is no such variation for uncoated glass, it need not be considered for the other sets of results. However, the percentage errors in the data for the Inconel and nickel pillars are comparable to the percentage errors in the results for the alumina pillars. This is because the heat flows - and thus the absolute errors associated with these flows - are larger for the uncoated glass samples. These absolute errors represent larger percentage errors of the small differences in the heat flows that are calculated on the basis of equation 2.25.

It is important to note that the measurements correspond to a total glass thickness of 6 mm on each side of the glazing (4 mm for the glass sheet and 2 mm for the glass faceplate of the apparatus which is in contact with the sheet). This is the total thickness which is used for the calculation of the pillar heat flows according to the procedure given in Section 2.5.2. The thermal resistances are based on the difference in the temperatures of the outer surfaces of the 6 mm assemblies of glass - i.e. the surfaces of the faceplates which are in contact with the copper plates of the apparatus. Due to the low thermal resistances between these surfaces and the two water flows, this temperature difference can be assumed to be equal to the difference in temperature between the water flows. For the conditions considered in this study, this leads to less than a 1 % error in the glass-to-glass temperature difference.

Figure 4.2 presents experimental and analytic results for the *inverse* of the thermal resistance as a function of the pillar diameter. The analytic results are calculated according to equation 4.2 and they are based on a glass thermal conductivity of $k_{glass}=1.0 \text{ Wm}^{-1}\text{K}^{-1}$. As explained in Appendix 3, the thermal resistance for a square pillar is approximately equal to that for a circular pillar of the same area. Thus, the results for the square alumina pillars are presented for the equivalent diameter. It is noted that equation 4.2 is approximately correct if $k_{pillar} \gg k_{glass}$. This is certainly the case for pillars made of nickel, Inconel 718 or alumina since these materials have the respective thermal conductivities of $90 \text{ Wm}^{-1}\text{K}^{-1}$ [4], $45 \text{ Wm}^{-1}\text{K}^{-1}$ [5] and $30 \text{ Wm}^{-1}\text{K}^{-1}$ [6]. More exact analytic results for the latter two pillars can be obtained by equation 4.1. However, such results for the Inconel and alumina pillars are, respectively, only 4 % and 2 % less than those calculated by equation 4.2. This would not affect the conclusions drawn below about the agreement - or the lack of agreement - between the experimental and analytic results. The conclusions would also be unaffected by accounting for the effect of a finite glass thickness on the pillar thermal resistance; the analytic results are based on the assumption of a semi-infinite glass thickness, whereas the measurements correspond to a finite glass thickness which, as shown below, leads to a lower thermal resistance than that predicted by the analytic expression.

In Fig. 4.2, there are actually three coincident data points for the alumina pillars. To within the experimental errors, the alumina and nickel pillar data are consistent with the analytic predictions. However, the thermal resistances for the Inconel pillars are greater than the predicted values (it is recalled that this figure presents the *inverse* of the thermal resistance). This is probably due to two reasons. Firstly, the Inconel pillars do not appear to be in uniform contact with the glass sheets. Secondly, there are small crescent-shaped cracks at the internal glass surfaces in the vicinity of these pillars.

Such cracks can be caused by the shear stresses in the glass near the pillars that result from the non-uniform heating of one or both glass sheets [7]. This non-uniform heating occurs under normal service conditions. It also occurs during the tests with the thermal conductance apparatus because the glass in contact with the cold plate is at a different temperature than the surrounding glass of the same sheet which approaches room temperature with distance from the apparatus (in contrast, all of the warm side glass is close to room temperature). The resulting cracks restrict the heat flow in the glass, thereby increasing the thermal resistance of the pillar-glass system. For the nickel and ceramic pillars, there is neither poor surface contact nor cracks in the nearby glass. The absence of shear cracks for nickel pillars is not fully understood but it is thought to be related to the fact that the strength of nickel is less than that of Inconel. Although shear cracks are sometimes observed around ceramic pillars, there are no such cracks around the ceramic pillars considered in this study. The above analysis demonstrates how a comparison between the measured and predicted pillar thermal resistances can be used as a means of quality control for determining the conditions at the pillar-glass interface.

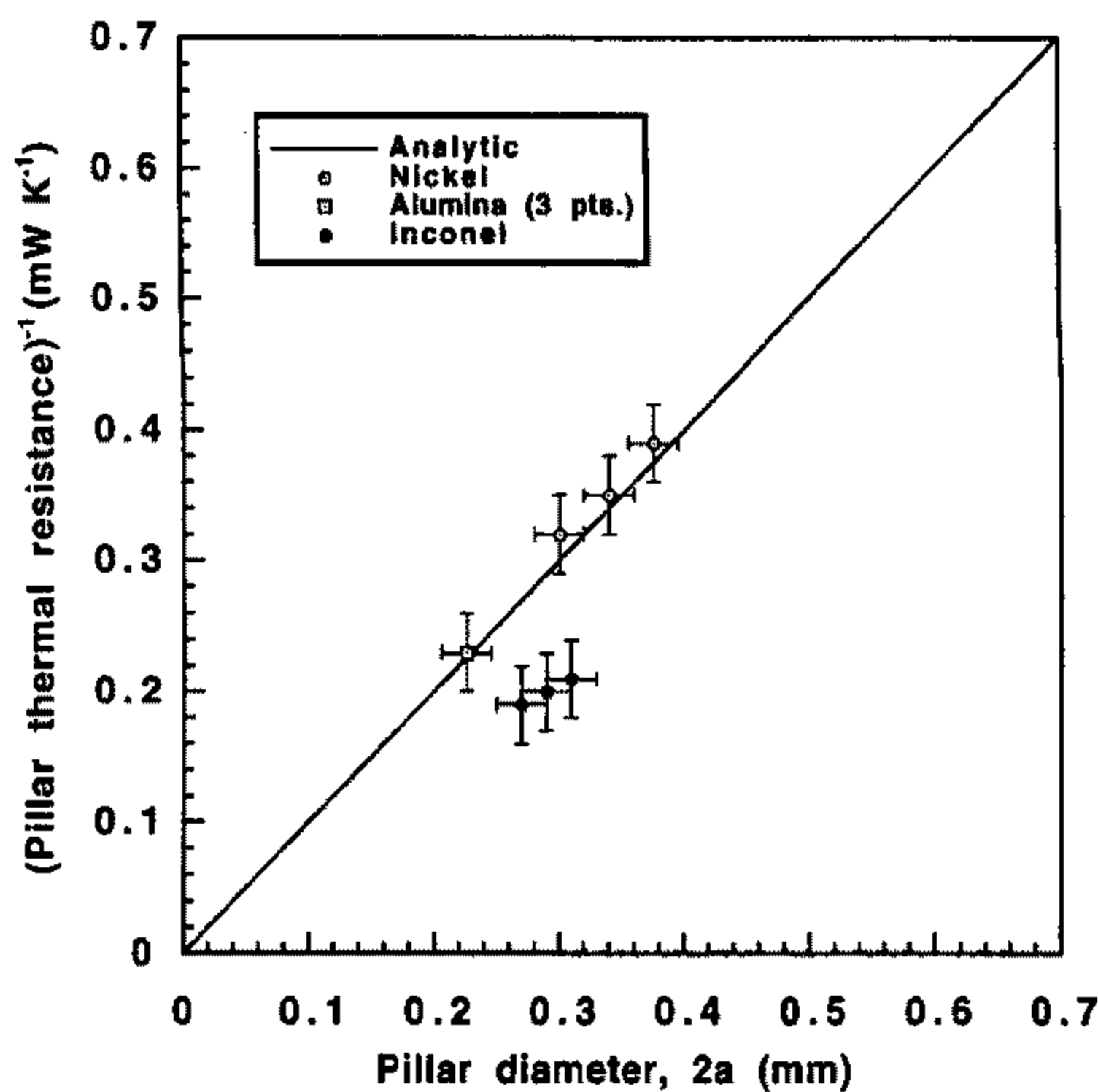


Fig. 4.2: The inverse of the pillar thermal resistance as a function of the pillar diameter. Experimental and analytic results are shown for three different types of pillars. The analytic results are obtained by equation 4.2 with $k_{glass}=1.0 \text{ Wm}^{-1}\text{K}^{-1}$. For the alumina pillars, there are three coincident data points.

Due to the difficulty in quantifying the effects of the poor pillar-glass contact and the cracks in the glass, the results for the Inconel pillars can not be used to validate the analytic relationship (equation 4.2). However, on the basis of the experimental results for the alumina and nickel pillars, it is concluded that this analytic relationship can be used to obtain a close estimation of the pillar thermal resistance if there is a good pillar-glass contact, and if shear cracks are absent.

4.3 Finite Element Model

A finite element model was developed for the analysis of the effects of the heat conduction through the pillars. The model, which is shown in Fig. 4.3, is the cylindrically axisymmetric equivalent of the geometry presented in Fig. 4.1. It consists of two sheets of soda-lime glass, each of thickness 4 mm, which are separated by an evacuated space and a pillar. The pillar is centered along the axis of symmetry of the model and it is of radius, a , and height, h . The height of the evacuated space is the same as that of the pillar. The thermal conductivity for the glass is that which is given in Appendix 1 for soda-lime glass: $1.0 \text{ Wm}^{-1}\text{K}^{-1}$. Except where otherwise specified, the thermal conductivity for the pillar is $20 \text{ Wm}^{-1}\text{K}^{-1}$, which is of the same order as the thermal conductivities for the two most commonly used pillar materials: Inconel 718 ($45 \text{ Wm}^{-1}\text{K}^{-1}$ [5]) and alumina ($30 \text{ Wm}^{-1}\text{K}^{-1}$ [6]). Equation 4.1 indicates that, for these typical thermal conductivities, and for the typical pillar dimensions of $a \sim 0.125 \text{ mm}$ and $h \sim 0.2 \text{ mm}$, the pillar thermal resistance should not be strongly dependent on the pillar height or on the exact value for k_{pillar} - as was found to be the case. The external glass surfaces are at specified uniform temperatures. All other glass surfaces, except those in contact with the pillar, are defined to be adiabatic. As mentioned above, the pillar surface which is exposed to the vacuum is also defined to be adiabatic.

Due to the symmetry on both sides of the pillar, only a half-thickness of the model (one sheet of glass and a half-height pillar) need be simulated. This can be accomplished by setting the temperature at the mid-plane of the pillar equal to the expected temperature at this location: the average of the two external glass surface temperatures. However, since the model is also used as the basis for another model which includes asymmetric external conditions (for which a half-thickness simulation would be inappropriate), it was decided to simulate the whole structure of Fig. 4.3.

The axial heat flow is defined to be the heat flow which is parallel to the axis of symmetry for the model (which is coincident with the centerline of the pillar). The total three-dimensional axial heat flow through the pillar, and thus between the glass sheets, is calculated as follows. The axial heat flow through each square element in a row of elements adjacent to either external glass surface is determined on the basis of the nodal temperatures, and the elemental dimensions and thermal conductivity. The elemental heat flows are then radially integrated over 2π times the radial distance between the axis of symmetry and the center of each element of the row to give the total heat flow. This calculation is based on the results closest to the one of the external glass surfaces because, near either surface, the temperature gradients - and thus the small errors inherent in this calculation - are the least. The pillar thermal resistance is calculated by dividing the temperature difference between the external glass surfaces by the total heat flow. The total heat flow can also be calculated, although less accurately, on the basis of the radial temperature profiles near the mid-plane of the pillar.

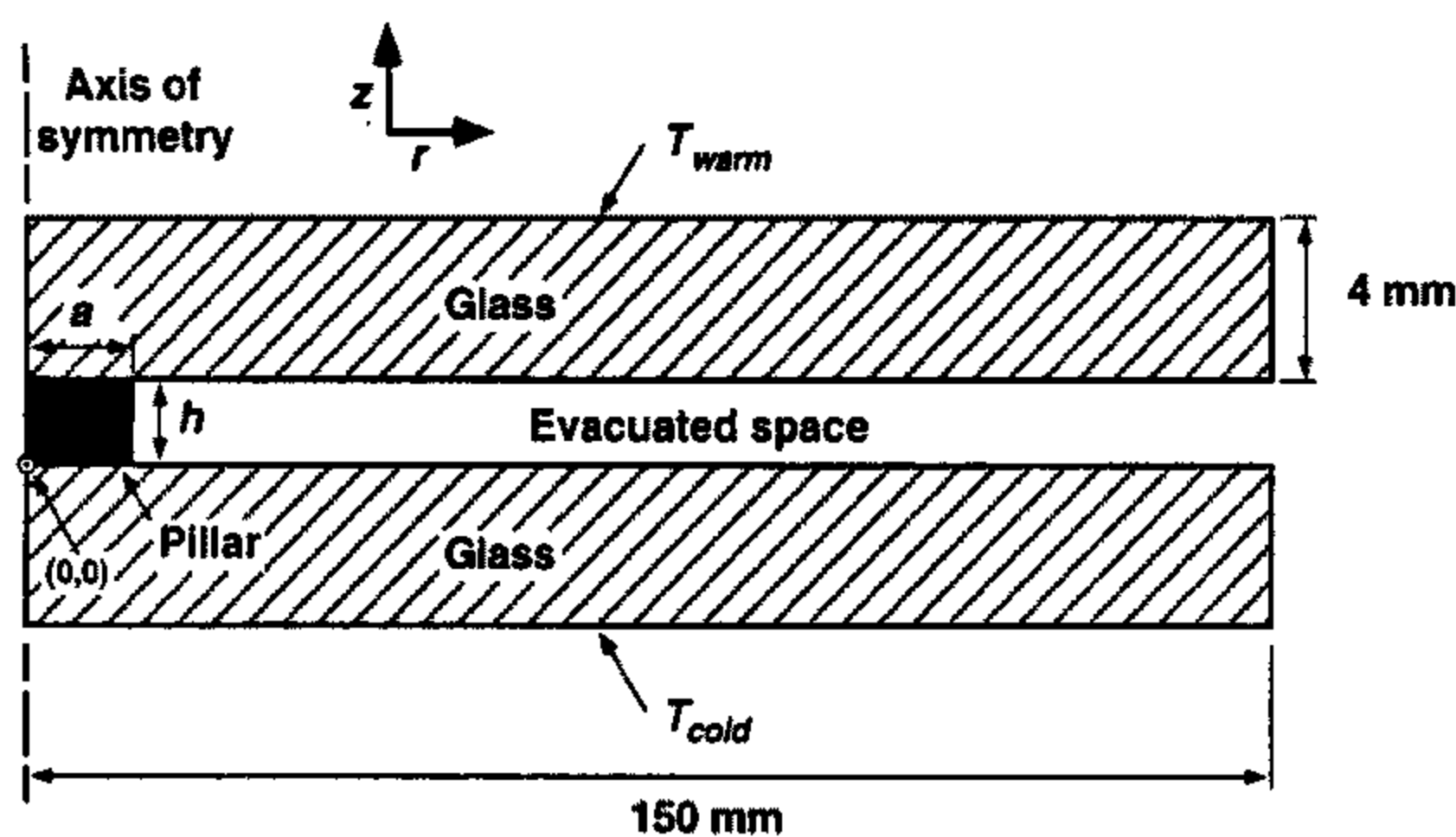


Fig. 4.3: The geometry of the cylindrically axisymmetric finite element model for the analysis of the effects of the heat conduction through a single pillar of radius a and height h . The glass sheets are 4 mm thick and extend to a radius of 150 mm from the axis of symmetry. The temperatures of the external glass surfaces are defined. All other glass surfaces, except those in contact with the pillar, are adiabatic. Not to scale.

The optimum node density for the mesh was determined with a convergence test based on the total number of nodes (N) in the pillar, and in the glass within a radius of 0.5 mm from the axial centerline of the pillar and within an axial distance of 0.5 mm from the pillar. This region in and around the pillar was selected for the convergence test because it is where the largest temperature gradients occur, and it is thus where

the greatest node density is required; about 90 % of the axial temperature variation through the thickness of each sheet occurs within 0.5 mm of the pillar (as shown below for conditions which are different from those which are used in this test). The node density elsewhere in the model was appropriately graded with the density decreasing with both the radial and axial distance from the pillar. The test was done with a pillar of radius 0.125 mm and of height 0.2 mm. The external glass surface temperatures were 1 and 0 °C. Equation 4.2 was used to predict a benchmark thermal resistance for the convergence test because, as demonstrated by the above experimental results, the analytic relationship can be used to provide a close estimation of the expected thermal resistance. Since this equation is based on the assumption that $k_{pillar} \gg k_{glass}$, and since it is important to simulate the same conditions, the pillar thermal conductivity for this test is set much higher than the glass thermal conductivity: $k_{pillar} = 10^4 \text{ Wm}^{-1}\text{K}^{-1}$. For the above pillar radius, and for a glass thermal conductivity of $1.0 \text{ Wm}^{-1}\text{K}^{-1}$, the pillar thermal resistance predicted by equation 4.2 is $4.00 \times 10^3 \text{ KW}^{-1}$. The results of the convergence test, which are presented in Fig. 4.4, indicate that modelled thermal resistance converges to a value which is about 5 % less than the benchmark predicted by the analytic expression. This is because the analytic expression is based on the assumption of a semi-infinite glass thickness, whereas the model simulates a finite glass thickness; as explained in Appendix 3, a finite glass thickness results in a lower thermal resistance. It was decided to use $N=1136$ nodes within the above specified region in and around the pillar. It is important to note that such a high density of nodes in this region is only required for the calculation of heat flows and thermal resistances. For the calculation of glass temperatures near the external surfaces, a much lower density is sufficient. For example, a change in N from 358 to 2866 nodes increases the glass temperature at 0.25 mm from the external surface of each sheet by less than 0.006 % of the glass-to-glass temperature difference across the system. Although the node density has only a small effect on the individual temperatures, it does significantly affect the total heat flows and thermal resistances - as shown above. This is because the small effect on the temperatures is compounded by the radial integration calculations.

Aside from having an appropriate node density, another requirement for ensuring the accuracy of the results is that the glass sheets must be of a sufficient radial extent. It was found that a sheet radius of 150 mm was more than adequate for this purpose, and thus this is what was used for the model. The model contains a total of 7048 nodes graded from a node spacing of 0.0125 mm within the pillar to 1 mm in the glass beyond 50 mm from the pillar.

For realistic dimensions and thermal conductivities, the modelled thermal resistances are different from the analytic results predicted by equation 4.2. This is because the model simulates two effects that are not considered by the analytic expression: the effect of the finite glass thickness, and the effect of the thermal resistance associated with the plane parallel heat flow through the pillar itself. These effects are investigated separately.

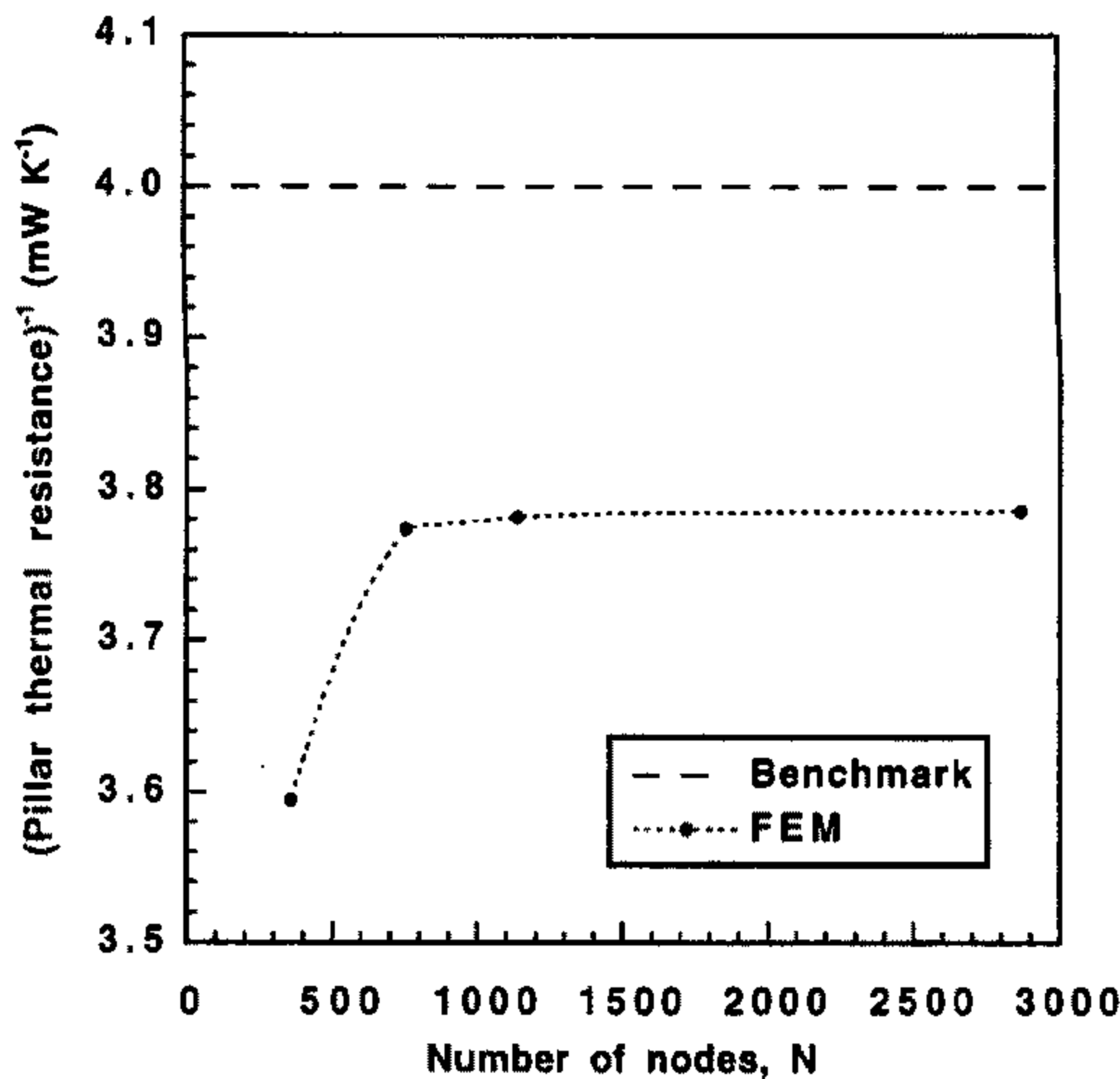


Fig. 4.4: Convergence test results. The pillar thermal resistance calculated from the finite element model (FEM) is presented as a function of the total number of nodes, N , in the pillar, and in the glass within a radius of 0.5 mm from the axial centerline of the pillar and within an axial distance of 0.5 mm from the pillar. The pillar radius, height, and thermal conductivity are 0.125 mm, 0.2 mm, and $10^4 \text{ Wm}^{-1}\text{K}^{-1}$. Also shown is the benchmark result obtained from equation 4.2 with the same pillar radius, and $k_{\text{glass}}=1.0 \text{ Wm}^{-1}\text{K}^{-1}$.

Figure 4.5 presents modelling and analytic results for the *inverse* of the pillar thermal resistance as a function of the pillar diameter for a highly conducting pillar ($k_{\text{pillar}}=10^4 \text{ Wm}^{-1}\text{K}^{-1}$). The linear fit to the modelling results of this figure is constrained to pass through the origin because the inverse of the thermal resistance must be zero for a zero diameter. As mentioned above, the modelled thermal resistances are less than the analytic predictions due to the fact that the model simulates a finite glass thickness. The divergence between the modelling and analytic results with pillar

diameter demonstrates that, as explained in Appendix 3, the effect of a finite glass thickness varies with the pillar radius if the glass thickness is fixed.

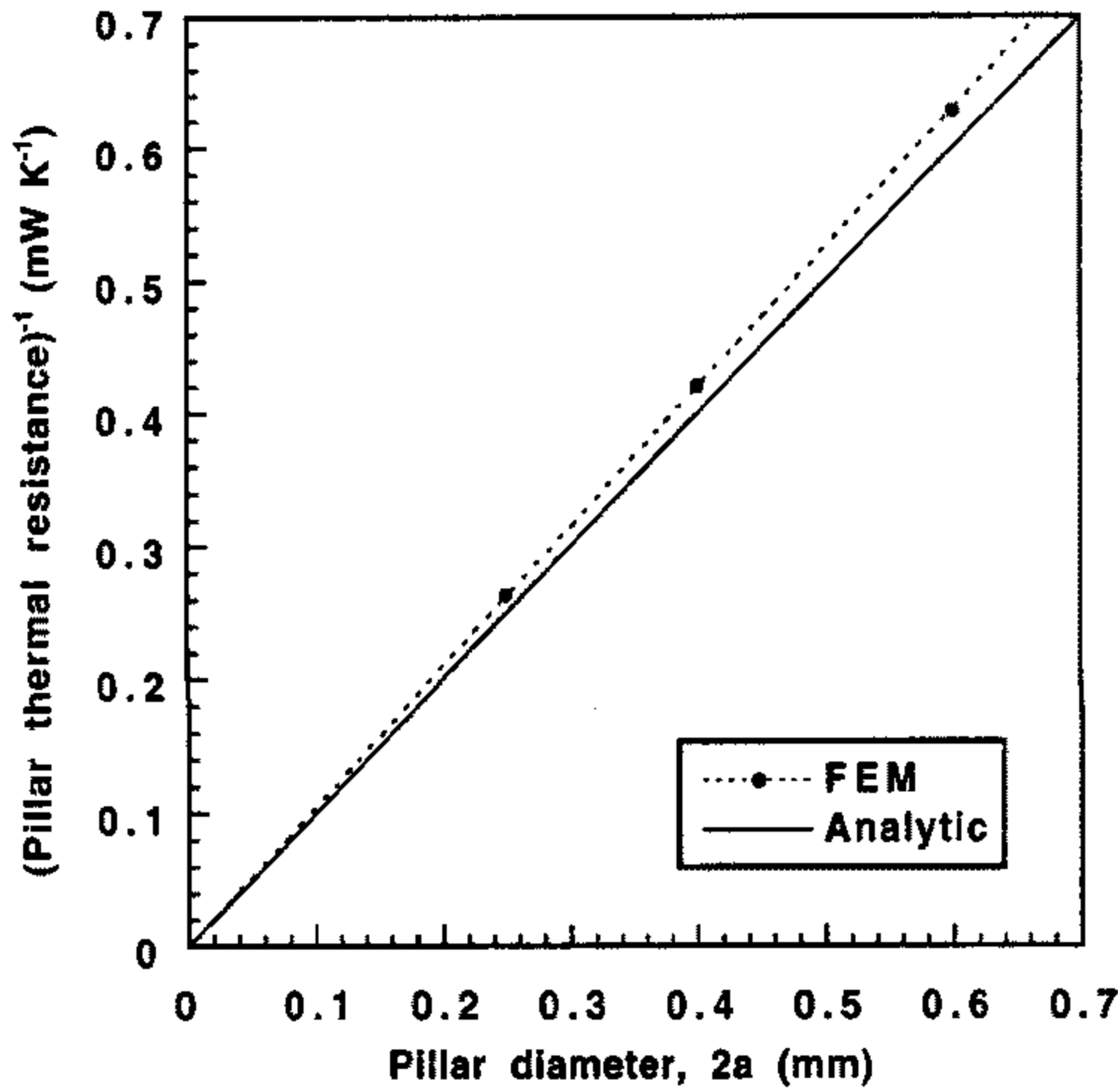


Fig. 4.5: The inverse of the pillar thermal resistance as a function of the pillar diameter for a highly conductive pillar. Results from the finite element model (FEM), and the analytic results (Analytic) are shown. Also shown is a linear fit to the FEM results (dashed line). This fit is constrained to pass through the origin. The modelled pillar height and thermal conductivity are 0.2 mm, and $10^4 \text{ Wm}^{-1}\text{K}^{-1}$. The analytic results are obtained from equation 4.2 with $k_{\text{glass}}=1.0 \text{ Wm}^{-1}\text{K}^{-1}$.

It was previously mentioned that the thermal resistance for the pillar-glass system, R_{pillar} , can be approximated by the series combination of the resistance of the spreading heat flow in the glass, and of the resistance associated with the plane parallel heat flow in the pillar:

$$\begin{aligned}
 R_{\text{pillar}} &= R_{\text{spreading}} + R_{\text{parallel flow}} \\
 &= R_{\text{spreading}} + \frac{h}{k_{\text{pillar}} \pi a^2} .
 \end{aligned}
 \tag{4.4}$$

It is recalled that equation 4.2 is based on the assumptions that $R_{\text{spreading}}$ is equal to that for a semi-infinite glass thickness ($1/2k_{\text{glass}}a$), and that $R_{\text{parallel flow}}$ is much less than $R_{\text{spreading}}$ (i.e. that $R_{\text{parallel flow}}$ is negligible). For realistic conditions, the spreading resistance is lower than that for a semi-infinite glass thickness. Since the modelling

results of Fig. 4.5 are for a highly conductive pillar (for which $R_{parallel\ flow}$ is certainly negligible), they represent the spreading resistance for a finite glass thickness of 4 mm. In order to ensure that the model provides results which are consistent with equation 4.4, a less conductive pillar was simulated - i.e. a small, yet significant, $R_{parallel\ flow}$ was considered: the pillar height was as before (0.2 mm), but the pillar thermal conductivity was $k_{pillar}=20\text{ Wm}^{-1}\text{K}^{-1}$. The external glass surface temperatures were 1 and 0 °C. Figure 4.6 presents modelled, expected and analytic results for the *inverse* of the thermal resistance as a function of the pillar diameter. The expected results were obtained according to equation 4.4, with $R_{spreading}$ determined from the above mentioned linear fit to the modelling results of Fig. 4.5. Near the origin, the functional dependence of the expected results is dominated by the $1/a^2$ dependence of the $R_{parallel\ flow}$ component of R_{pillar} . There is a close agreement (to within about 1 %) between the modelled and expected thermal resistances. For example, for $2a=0.25\text{ mm}$, the respective modelled and expected results are 4.02 and $3.99 \times 10^3\text{ KW}^{-1}$. It is fortuitous that, for the typical pillar dimensions and thermal conductivity considered for this figure (and throughout this thesis), equation 4.2 provides thermal resistances which are close to the more realistic modelling results. This is despite the fact that, unlike the finite element model, this analytic relationship does not account for the effects of the finite glass thickness or of the thermal resistance associated with the parallel heat flow in the pillar. Since both of these effects are small, and since they have opposite and approximately equal influences on the pillar thermal resistance, there is a close agreement between the analytic and modelling results.

The modelled thermal resistances of Fig. 4.6 are consistent with the experimental results presented in Fig. 4.2 for nickel and alumina pillars. It is noted that the experimental results correspond a total glass thickness of 6 mm (as mentioned above), whereas the modelling results are for a glass thickness is 4 mm. However, as explained in Appendix 3, since most of the temperature drop through each glass sheet is near the pillar, such a difference in the glass thickness has a negligible effect on the pillar thermal resistance.

The finite element model can be modified to include radiative heat transfer in the gap and/or heat transfer between the external glass surfaces and ambient. As shown in Fig. 4.7, these processes can be modelled by placing thermally conductive slabs within the gap, and at the external surfaces of the glass sheets. The thermal conductivity and thickness of the internal slab correspond to the required internal radiative U-value. For the external slabs, the thermal conductivities and thicknesses correspond to the specified heat transfer coefficients, and the outer surface temperatures correspond to

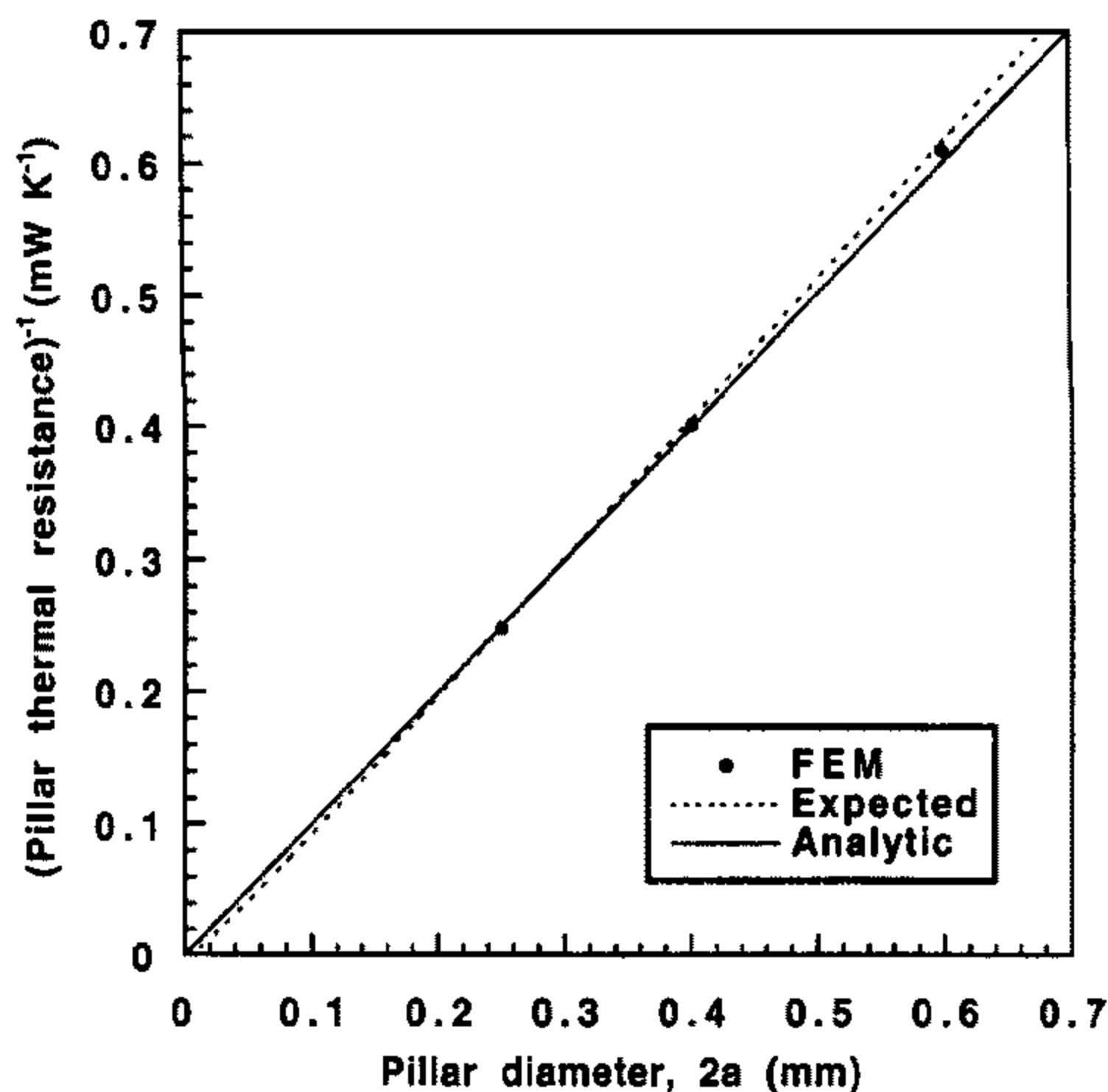


Fig. 4.6: The inverse of the pillar thermal resistance as a function of the pillar diameter. Results from the finite element model (FEM), the expected results (Expected), and the analytic results (Analytic) are shown. The modelled pillar height and thermal conductivity are 0.2 mm, and 20 Wm⁻¹K⁻¹. The expected results are obtained from equation 4.4, with $R_{spreading}$ determined from the linear fit to the FEM results of Fig. 4.5. The analytic results are obtained from equation 4.2 with $k_{glass}=1.0$ Wm⁻¹K⁻¹.

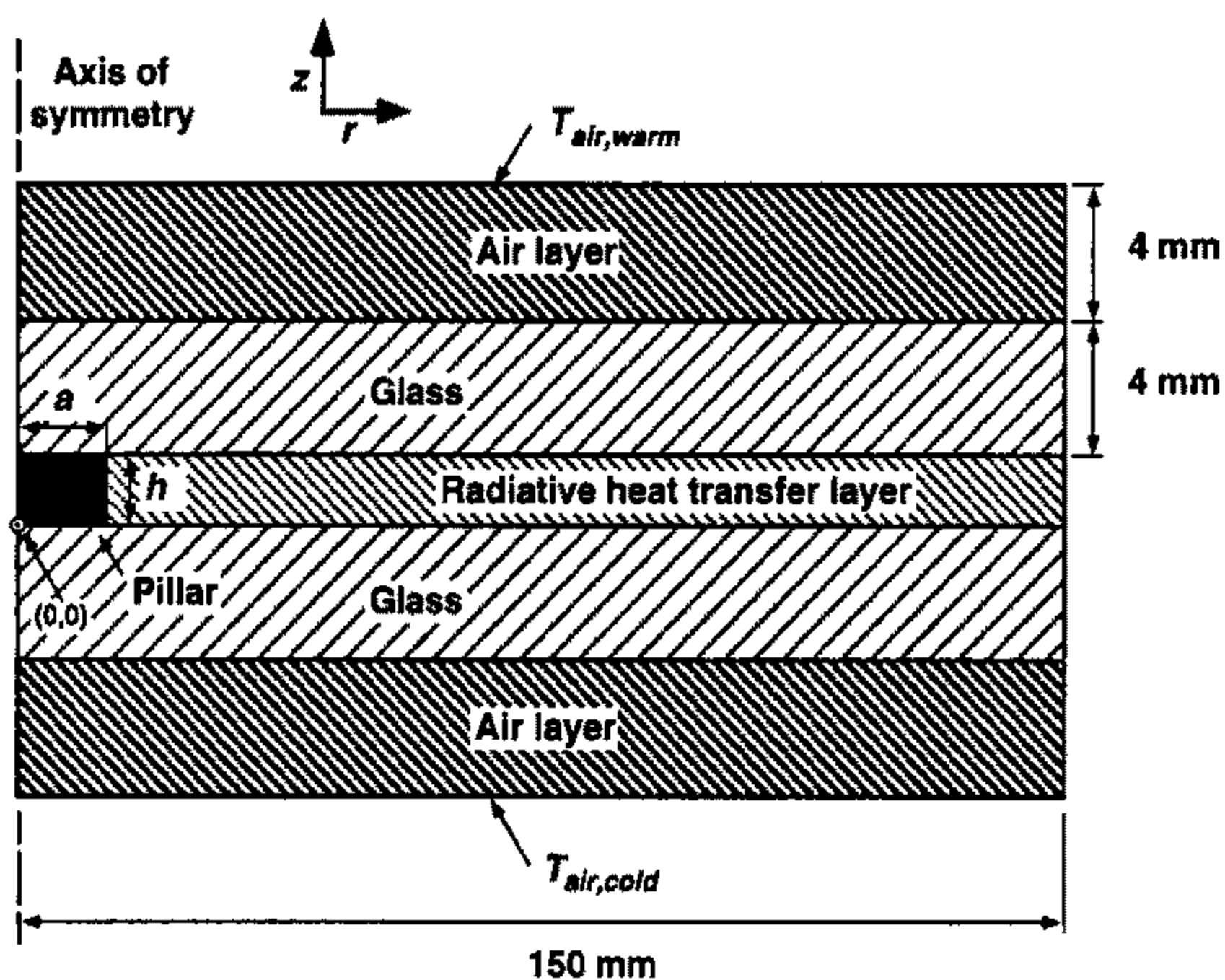


Fig. 4.7: The finite element model shown in Fig. 4.3 modified to simulate internal radiative heat transfer, and heat transfer between the external glass surfaces and ambient. The air temperatures are defined at the external surfaces of the air layers. Not to scale.

the specified air temperatures. The use of external slabs does not simulate actual convective flows but, as demonstrated in the next chapter, it does provide an adequate method for approximating the heat transfer between the external glass surfaces and ambient.

With both radiative heat transfer and air layers added to the model, the total number of nodes was greater than that which could be solved with the available computing resources. It was therefore necessary to economize on the nodes for this situation by using a smaller number of nodes in the region in and around the pillar that was specified for the above convergence test: $N=358$ nodes. This is justified on the basis that the model with both radiative heat transfer and air layers is only used to obtain external glass surface temperatures and, as previously mentioned, the temperatures near the external surfaces are not strongly dependent on N .

4.4 Results

4.4.1 External Glass Surface Temperatures

The finite element model for the heat flow through a single pillar of radius 0.125 mm and height 0.2 mm was modified to simulate heat transfer with ambient. The external conditions correspond to those of the ASTM E 1423-91 standard for determining the steady state thermal transmittance of fenestration systems: air temperatures of 21.1 and -17.8 °C, and corresponding heat transfer coefficients of 8.3 and 30.0 $\text{Wm}^{-2}\text{K}^{-1}$ [8] (these temperatures are based on exact conversions to metric units of the imperial unit values given in the standard). For this case, no internal radiative heat transfer was simulated.

Figure 4.8 shows results obtained with this model for the temperature in the pillar and the glass near the pillar. The region which is presented extends to a radius of 2.5 mm from the centerline of the pillar (the axis of symmetry for the model), and for a distance of 2.5 mm in the axial direction. The temperature interval between each coloured region is about 1.3 °C. The slight asymmetry in the locations of the isotherms in the glass on opposite sides of the pillar is merely a function of the temperature scale. What is important to note is that the shape of these isotherms is approximately the same; this indicates that, near the pillar, the heat flow in each glass sheet is approximately symmetric in shape with the heat flow in the other sheet. This is because the heat flows near the pillar spread out very little relative to the glass thickness. Further from the

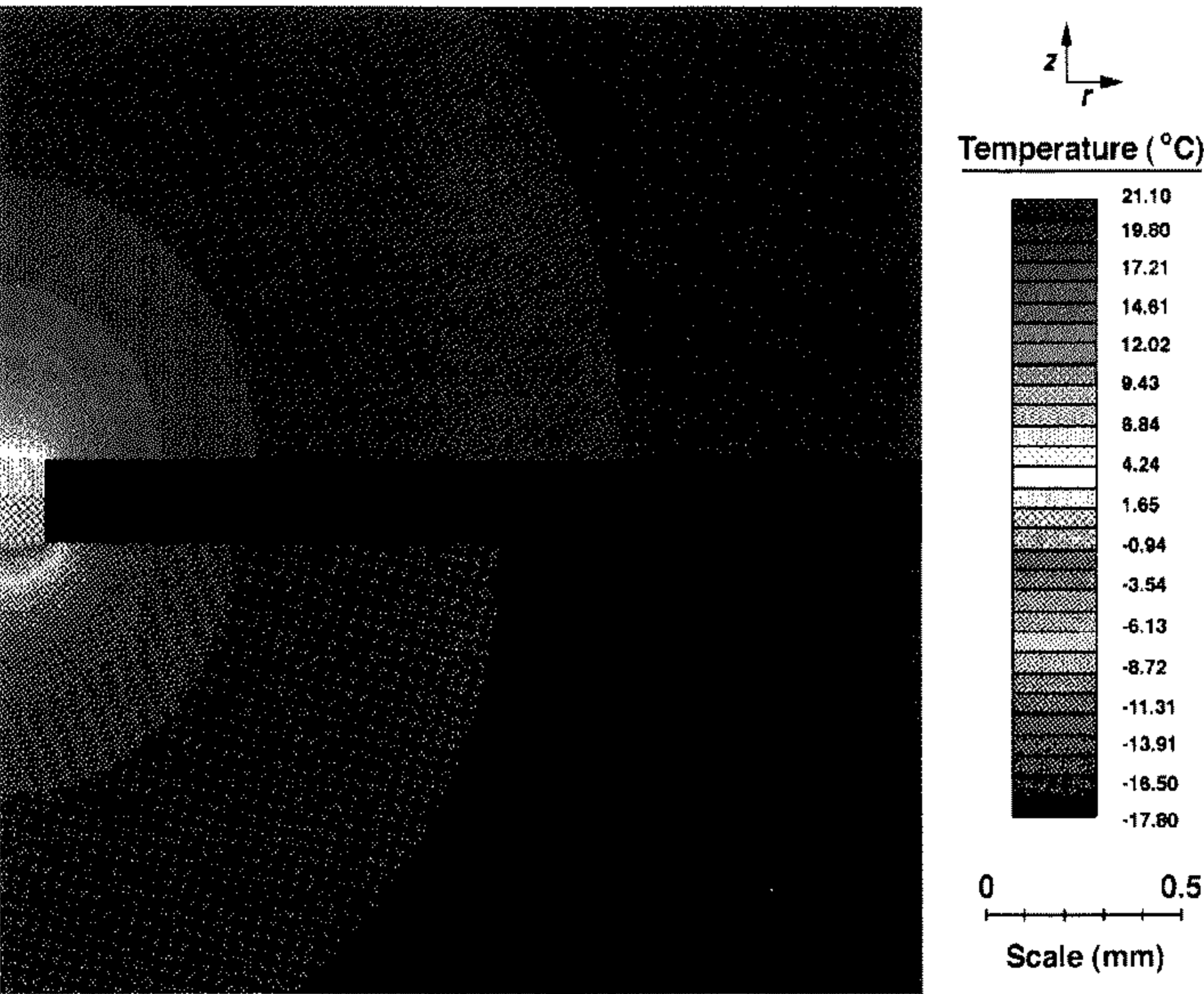


Fig. 4.8: Finite element modelling results for the temperature within a 2.5 mm by 2.5 mm region around a pillar with a radius and height of 0.125 mm and 0.2 mm, and for the ASTM standard fenestration test conditions. The evacuated space is represented by the black region. The pillar is to the left of, and is the same height as, this space. Portions of the two glass sheets are above and below the evacuated space-pillar section. No internal radiative heat transfer was simulated. The temperature interval between each coloured region is about 1.3 °C.

pillar, at distances greater than those shown in this figure, the isotherms - and thus the heat flows - are significantly asymmetric. This is a reflection of the inequality of the external heat transfer coefficients, and it is demonstrated by the results of the next figure.

Figure 4.9a shows two plots for the temperature of the warm side external glass surface as a function of the radial distance from the axial centerline of the pillar. Both sets of results were obtained with the modified model described above except that, for one of the plots, the warm side heat transfer coefficient is twice as large as the $8.3 \text{ Wm}^{-2}\text{K}^{-1}$ coefficient specified by the ASTM standard. For both of these warm side coefficients, the corresponding cold side external glass surface temperatures are within a few thousandths of a degree of each other. Thus, the cold side results presented in Fig. 4.9b are for both of the cases considered in Fig. 4.9a. As previously mentioned, no internal radiative heat transfer was simulated. Neglecting the very small effect of the radiative heat transfer suppression (described below), the inclusion of this mode of heat transfer for each case would not significantly change the shapes of the temperature profiles because the total temperature non-uniformity at each external surface is only a small fraction of the overall air-to-air temperature difference; the simulation of radiative heat transfer would essentially only change the magnitude of the difference between the warm and cold side temperature non-uniformities by an amount corresponding to the overall temperature difference between the external surfaces of the glass sheets at large distances from the pillar. For these conditions, the heat transfer coefficient on the warm side has little effect on the external surface temperature profile of the opposite side. This is due to two reasons. Firstly, the amount of heat flow through the pillar (and to the cold side glass sheet) is only slightly affected by the change in the warm side heat transfer coefficient because the warm side air-to-glass thermal resistance is such a small fraction of the total thermal resistance of the air-to-air system. Secondly, the distribution of the heat flow in the cold side glass is only influenced by the heat transfer coefficient to which this sheet is exposed, and not by that on the warm side.

The results of Figs. 4.9a and b demonstrate that, over a wide range of warm side heat transfer coefficients, the pillar conduction has only a small effect on the warm and cold side external surface temperatures beyond a radial distance of about 25 mm from the pillar. This is the basis for the above statement that, for the typical pillar separations in vacuum glazing ($\geq 25 \text{ mm}$), it can be assumed that the heat conduction through each pillar is essentially independent of that through the others. This also justifies one of the assumptions on which equations 4.1 and 4.2 are based - that the glass sheets are of

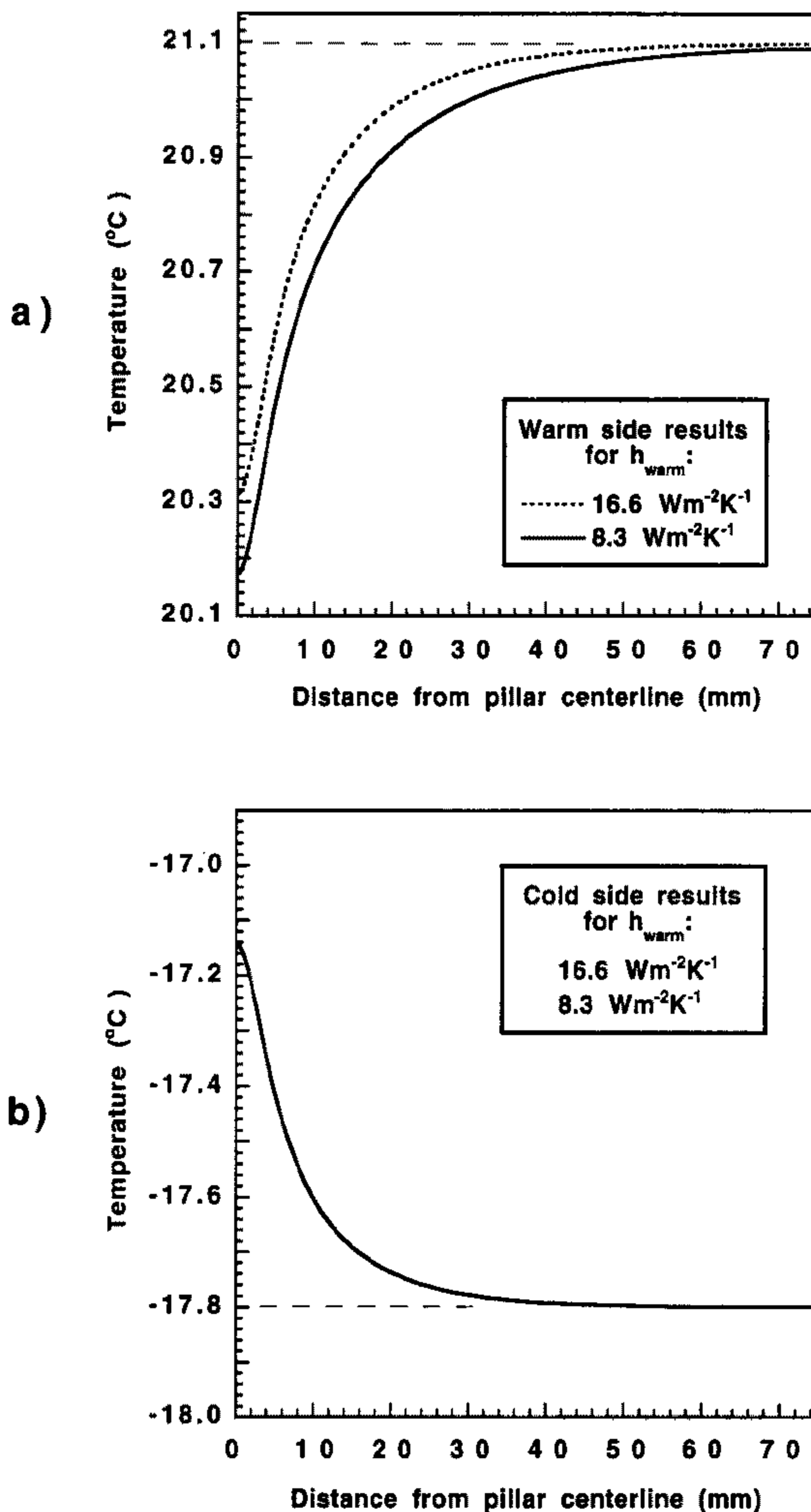


Fig. 4.9: Finite element modelling results for the external glass surface temperature as a function of the radial distance from the axial centerline of a pillar for (a) the warm side, and (b) the cold side. The results are for air temperatures of 21.1 and -17.8 °C, and for a cold side heat transfer coefficient of 30 $Wm^{-2}K^{-1}$. Two warm side heat transfer coefficients, h_{warm} , are considered: 8.3 and 16.6 $Wm^{-2}K^{-1}$. The cold side results correspond to both of these warm side coefficients. The conditions with $h_{warm}=8.3 Wm^{-2}K^{-1}$ are the ASTM standard test conditions. No internal radiative heat transfer was simulated.

semi-infinite extent. The fact that most of the effects of the pillar conduction occur within a short distance from the pillar is as expected because the heat flow per unit area in the glass sheets is very non-uniform, and it is greatest in and around the pillar. The temperature gradient associated with the heat flow through the pillar is therefore large near the pillar and very small elsewhere, where the heat flow spreads out laterally along the glass sheets.

It is important to note that the results of Figs. 4.9 might appear to indicate that most of the pillar conduction passes through each external glass surface within a radial distance of a few millimeters from the pillar. However, although the magnitude of the temperature variation - and thus the local heat flow per unit area - is greatest within this region, most of the total radially integrated pillar conduction which flows through each external glass surface comes from the smaller but more extensive temperature variation over the larger radii from the pillar.

The results of Figs. 4.9 indicate that, for both sets of conditions, the non-uniformity in the external glass surface temperature is greater in magnitude and extent on the warm side than that on the cold side. This is because there is a larger heat transfer coefficient on the cold side. Therefore, in the rest of this section, the analysis of the effect of pillar conduction on the external surface temperature (for the ASTM conditions) focuses on the side of the glazing where this effect is more significant: the warm side. The effect on the temperature of this side is also of interest because, under atypical conditions, condensation could form on the warm side external surface at the regions of reduced temperature around the pillars.

The data of Fig. 4.9a for the ASTM standard conditions (the plot for the warm side heat transfer coefficient of $8.3 \text{ Wm}^{-2}\text{K}^{-1}$) were used to calculate the warm side surface temperature across a 10 pillar by 10 pillar array for a regular pillar separation of 25 mm. This calculation was done with a computer program as follows. For each point across the array, the temperature variation (suppression) due to each pillar of the array was set equal to the result from Fig. 4.9 for the distance equal to that between the point and the pillar. The total temperature variation at this point was then determined by adding (i.e. superimposing) the corresponding variations from all of the pillars of the array. Due to the finite size of the array, the total temperature variations near the edges of the array are somewhat less than those near the middle of the array. This is because there are fewer pillars in the immediate vicinity of points near the edges than there are for points near the middle. This means that the most accurate results are near the middle of the array. Therefore only the results within the central 4 pillar by 4 pillar

section of the full array are considered in this analysis. Figure 4.10 shows the temperature distribution over this central section. The slight asymmetry near the edges is due to the residual effect of the finite size of the array. Figure 4.11 presents temperature profiles along three transits across this section. The transits are located at distances of $y = 0, 6$ and 12 mm from the second horizontal row of pillars in the 4 pillar by 4 pillar section (i.e. the row at 100 mm from the edge of the full array). Thus these profiles correspond to the temperatures across a row of pillars, and along lines close to the quarter-way and half-way positions between pillar rows. These results show that, for the ASTM standard test conditions, the maximum and minimum warm side external glass surface temperatures are about 19.41 and 18.84 °C. Therefore, the total variation in the temperature across the warm side external glass surface is approximately 0.57 °C, or 1.5 % of the air-to-air temperature difference across the glazing (38.9 °C). The results of Figs. 4.10 and 4.11 demonstrate that most of the non-uniformity in the external glass surface temperature due to pillar conduction occurs close to the pillar - i.e. that it is highly localized. The results of Figs. 4.9a and b indicate that, for these conditions, this non-uniformity should be even more localized on the cold side of the array. As mentioned above, this is because the heat transfer coefficient on the cold side is greater than that on the warm side. The localized nature of the effect of pillar conduction on the external glass surface temperature is further demonstrated by the infrared thermographic measurements presented in the next chapter.

For the results of Fig. 4.11, the average temperature across the profiles at $y = 0, 6$ and 12 mm are, respectively, $19.13, 19.27$ and 19.37 °C. The average temperatures at all other distances, y , across a unit cell of pillars (i.e. between two rows of pillars) are plotted in Fig. 4.12. Also shown in this figure is the average of these average temperatures (19.26 °C). It is recalled that these temperatures are based on modelling results obtained for a pillar of radius 0.125 mm, and on a superposition of temperatures throughout a regular array with a pillar spacing of 25 mm (i.e. $a=0.125$ mm, and $\lambda=25$ mm). Substitution of these dimensions and $k_{glass}=1.0$ Wm⁻¹K⁻¹ into equation 4.3 gives a pillar array U-value of 0.40 Wm⁻²K⁻¹. The uniform warm side surface temperature predicted with this U-value is 19.23 °C. This analytic result (which is also shown in Fig. 4.12) is only 0.03 °C less than the average temperature of 19.26 °C based on the above exact consideration of the individual pillar heat flows (and only 0.1 % of the air-to-air temperature difference of 38.9 °C). Therefore, the average surface temperature can be accurately estimated with a uniform pillar array U-value calculated by the analytic relationship of equation 4.3. This is also demonstrated in the next chapter by a comparison between results obtained from this analytic approach and results from infrared thermographic measurements.

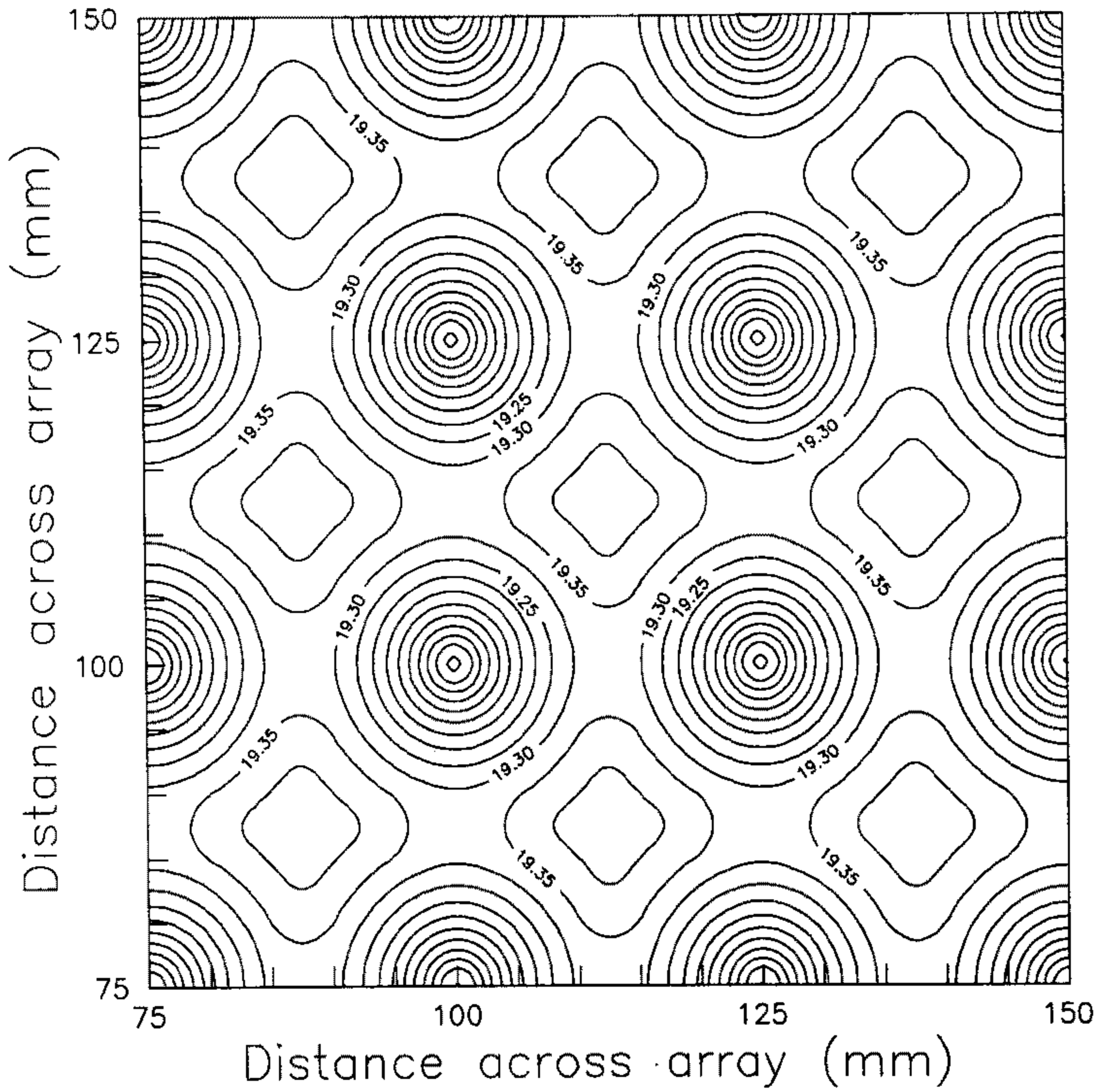


Fig. 4.10: Modelling results for the warm side external glass surface temperature across the middle 4 pillar by 4 pillar section of a 10 pillar by 10 pillar array. The pillar radius, height and separation are 0.125 mm, 0.2 mm and 25 mm, respectively. The results are for the ASTM standard test conditions. The pillars are located at the centers of the concentric circles. The slight asymmetry in the results near the edges of the plot is due to the finite size of the pillar array from which the results are obtained. The temperature contour interval is 0.05 °C.

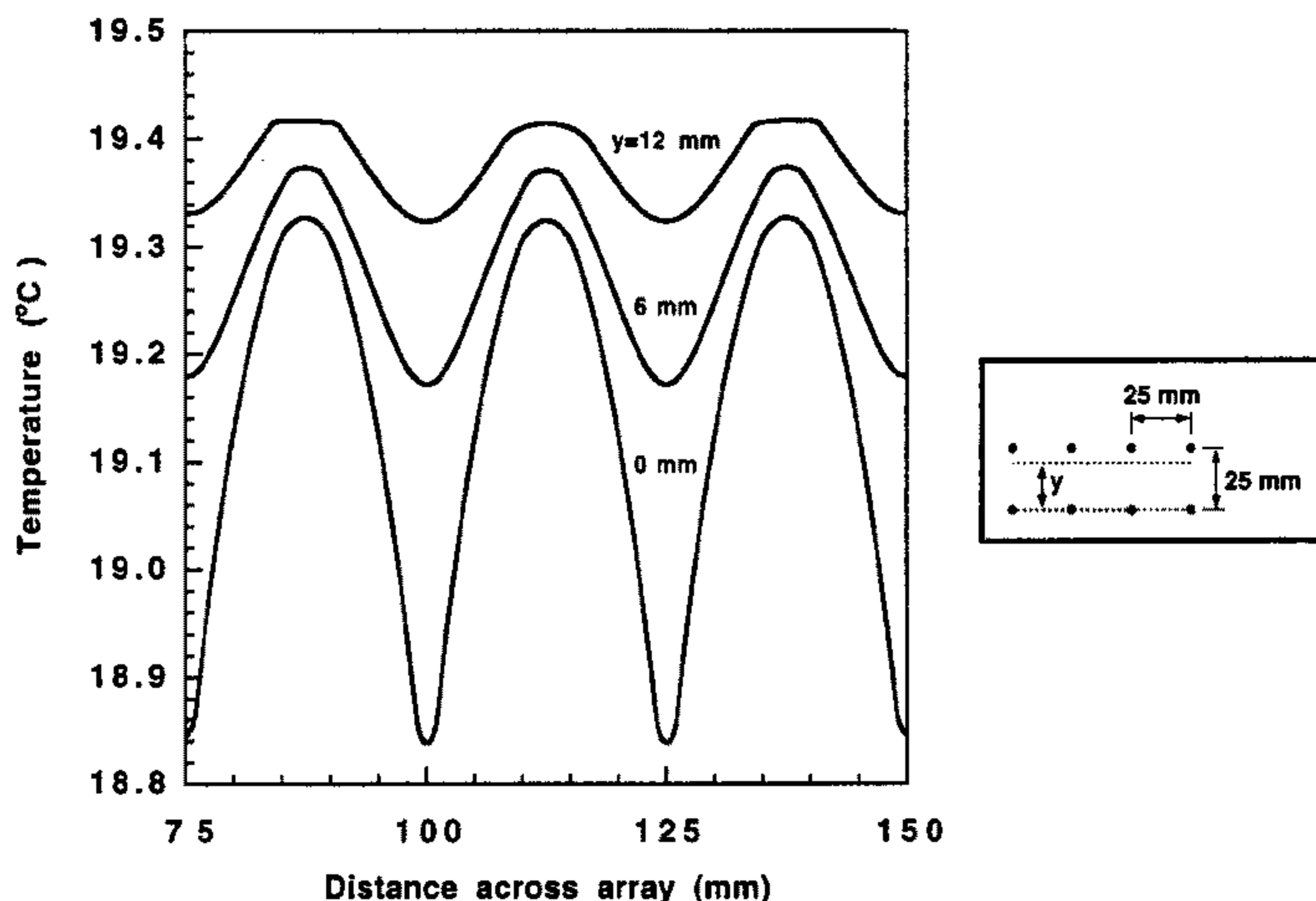


Fig. 4.11: Temperature profiles taken along three transits across the plot of Fig. 4.10. These transits are located at distances of $y = 0, 6$ and 12 mm from the second horizontal row of pillars in the 4 pillar by 4 pillar section (i.e. the row at 100 mm from the edge of the full array). The slight asymmetry in the results near the edges of the plot is due to the finite size of the pillar array from which the results are obtained.

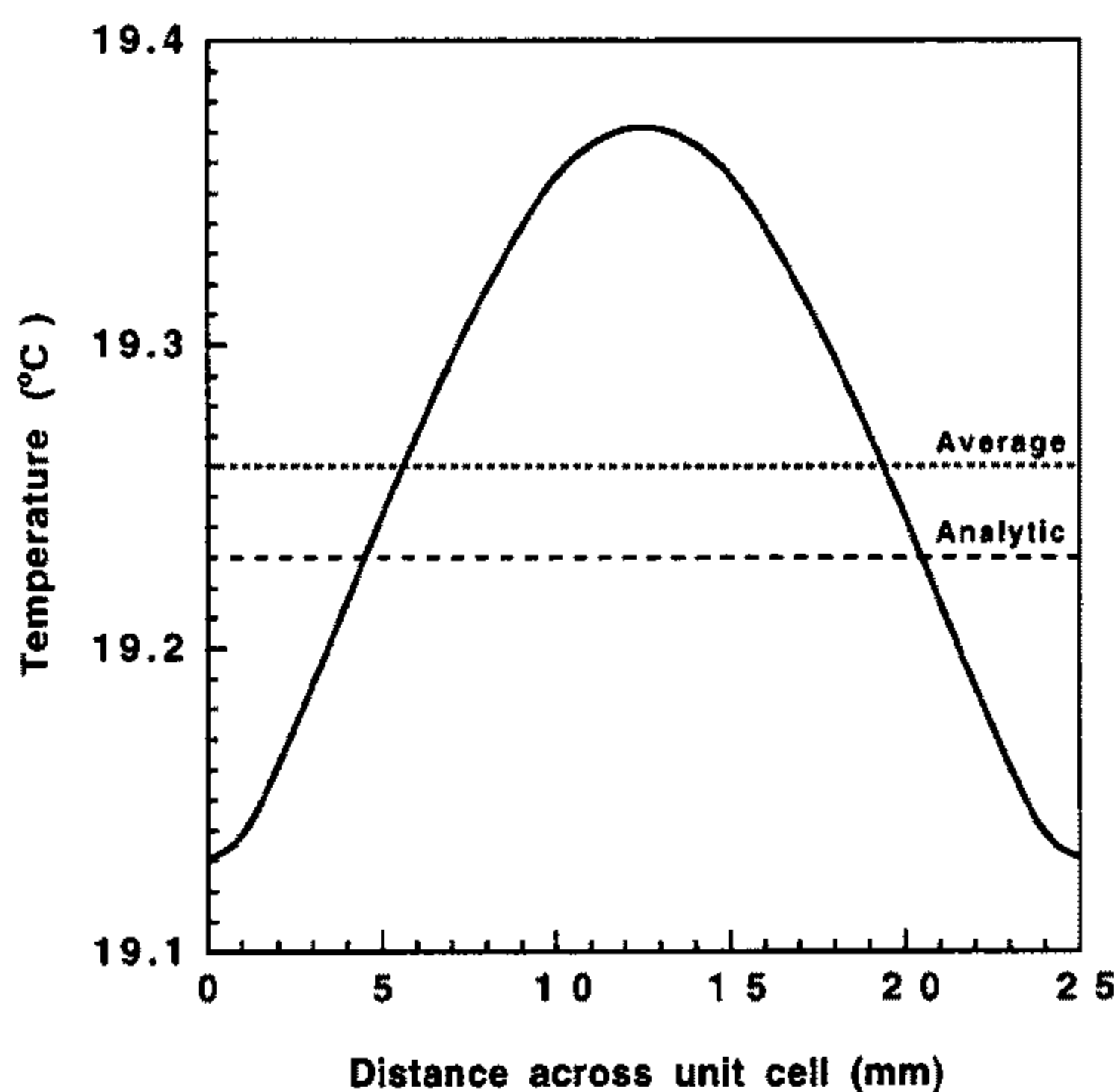


Fig. 4.12: The average external glass surface temperature as a function of the distance, y , across a unit cell of pillars (i.e. between two rows of pillars) for the results of Fig. 4.10. The average of these average temperatures is 19.26 °C. The uniform surface temperature based on the analytic estimation of U_{array} is 19.23 °C.

4.4.2 Suppression of The Radiative Heat Transfer

Most of the discussion in the previous section concerns the effect of the pillar conduction on the external glass surface temperatures. Pillar conduction also affects the temperatures of the internal glass surfaces, as shown by the finite element modelling results of Fig. 4.13. These results were obtained with the modified model described in the last section, for the ASTM conditions and a pillar radius and height of 0.125 mm and 0.2 mm. Thus, these results can be compared with the external surface temperatures shown in Figs. 4.9 for the same conditions. For each glass sheet, the internal glass surface temperature approaches the external glass surface temperature and, at distances larger than those shown in this figure, the internal and external temperatures are almost equal. Most of the temperature non-uniformities at the internal surfaces extend over a much shorter distance from the pillar (a few millimeters) than those at the external surfaces (~ 25 mm). This is because the spreading heat flow in the glass is more extensive with increasing distance through the sheets - as shown on a small scale in Fig. 4.8. According to equations 3.3a and b, the radiative heat transfer between the two internal glass surfaces is a function of the difference between the temperatures of these surfaces. Therefore, by virtue of its influence on the internal surface temperatures, the heat conduction through each pillar reduces - or suppresses - the radiative heat transfer close to the pillar. Although the largest variation in the temperature of each internal surface occurs within a few millimeters of the pillar, most of the contribution to the suppression of the total radially integrated radiative heat flow around the pillar actually comes from the smaller but more extensive variation over larger radii (e.g. about 1 °C on the warm side from 5 to 25 mm from the pillar, for the conditions considered in Fig. 4.13).

In order to investigate the effect of the pillar conduction on the radiative heat transfer, the following heat flows between the glass sheets were modelled: pillar conduction alone (Q_{pill}), pillar conduction combined with radiative heat transfer ($Q_{pill+rad}$), and radiative heat transfer alone (Q_{rad}). These were respectively simulated with the model presented in Fig. 4.3, the model presented in Fig. 4.7 without the external air layers, and the model presented in Fig. 4.7 without the air layers and with the pillar replaced by an extension of the radiative heat transfer layer. Heat transfer between air layers and the external glass surfaces was not modelled for this analysis. For all cases, the external glass surface temperatures were 1 and 0 °C, and the pillar radius, height and separation were 0.125 mm, 0.2 mm and 25 mm, respectively. The total three-dimensional axial heat flow between the glass sheets of each system was calculated in the manner described above.

For the above two systems with a pillar, and for the conditions considered in this analysis, there is a negligible contribution to the total heat flow through the pillar from elements beyond about 25 mm from the axis of symmetry of the model. However, in order to ensure the accuracy of the results, the total heat flow was calculated for an area defined by a larger radius from this axis (43 mm). Although Q_{rad} could be easily predicted analytically, it was calculated from the model in order to provide a result that would be consistent with the other heat flows in terms of the small errors associated with the elemental heat flow and radial integration calculations.

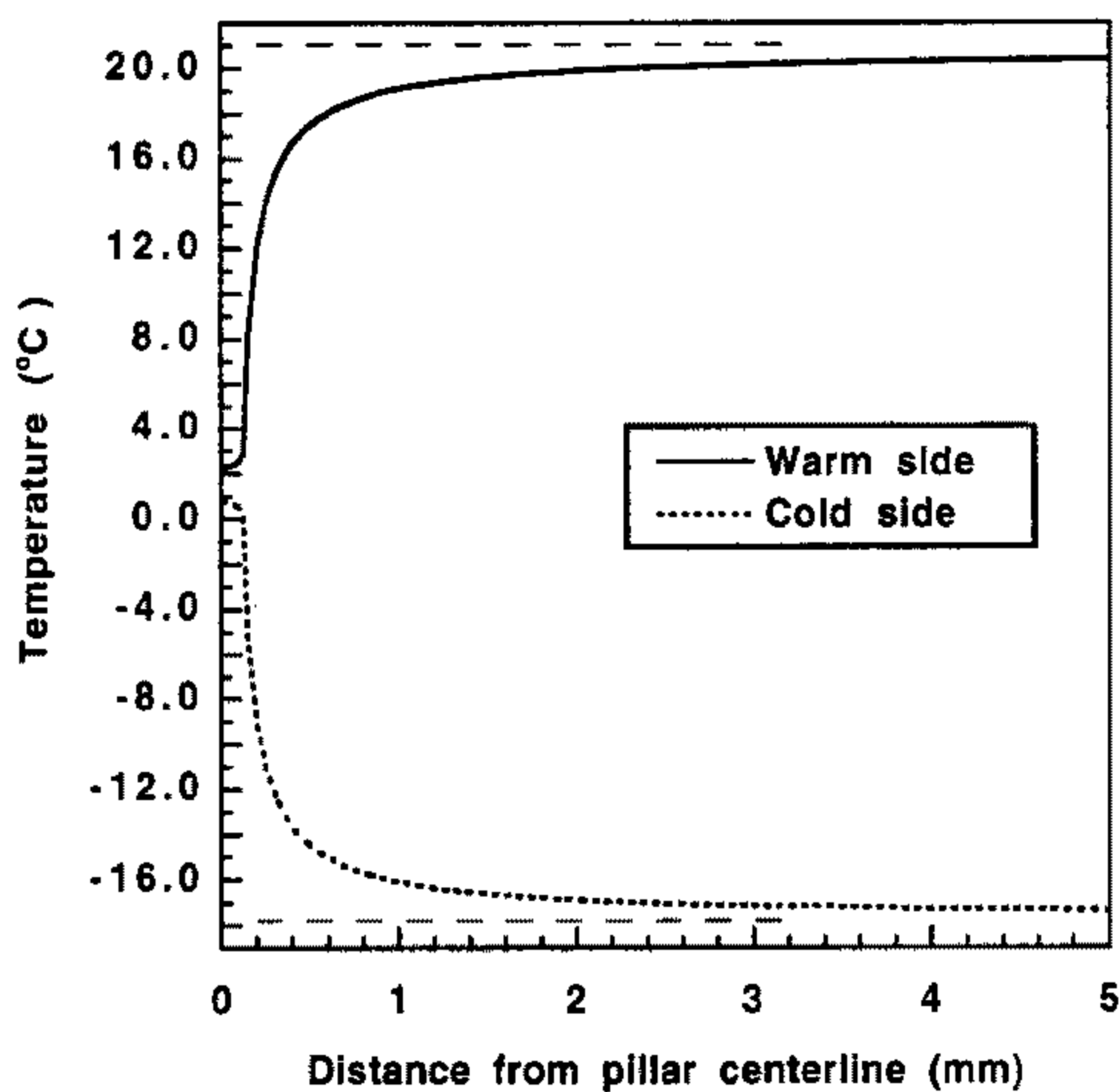


Fig. 4.13: Finite element modelling results for the temperatures at the internal glass surfaces as functions of the radial distance from the axial centerline of a pillar. The pillar radius, height and separation are 0.125 mm, 0.2 mm and 25 mm, respectively. The results are for the ASTM standard test conditions. The surface temperatures approach the temperatures indicated by the long dashed lines. No internal radiative heat transfer was simulated.

The results from this analysis for three approximately typical vacuum glazing glass-to-glass center-of-glass radiative U-values are summarized in Table 4.1. The U-values of 4.00, 1.15 and 0.75 Wm⁻²K⁻¹ respectively correspond to glazings with two sheets of uncoated glass, one sheet of uncoated and one sheet of baked K glass, and two sheets of baked K glass. These U-values are based on the results presented in Table 3.3 which are for internal surface temperatures of about 23 and 3 °C. The corresponding U-values for the temperatures considered below (and for the more realistic ASTM

standard test conditions) are a few percent lower but the ultimate conclusion from this analysis would be the same if they were used instead. For each U-value, the sum of the independently modelled heat flows for just the pillar heat conduction, Q_{pill} , and just the radiative heat transfer, Q_{rad} , is greater than the result obtained with both types of heat transfer modelled together, $Q_{pill+rad}$. This difference in the heat flows is due to the radiation suppression around the pillar. It is not due to the fact that, for the combined case, the presence of the pillar reduces the total area over which the radiative heat transfer occurs; this reduction in the total area is only 0.0008 % which, for the radiative U-values that are considered, is too small to account for the discrepancies in the heat flows. Collins et al. show that the suppression of the radiative heat transfer due to one pillar, Q_{sup} , is proportional to the heat conduction through the pillar, Q_{pill} [9]. The suppression of the radiative heat transfer can therefore be usefully expressed as a fraction of the heat conduction through the pillar:

$$\frac{Q_{sup}}{Q_{pill}} = \frac{(Q_{pill} + Q_{rad}) - Q_{pill+rad}}{Q_{pill}} \tag{4.5}$$

The modelled heat flows, and the corresponding radiation suppression results determined from this equation (which are expressed as percentages) are presented in Table 4.1.

Radiative U-value (Wm ⁻² K ⁻¹)	Result for pillar conduction alone Q_{pill} (mW)	Result for radiation alone Q_{rad} (mW)	Sum of independent heat flows $Q_{pill} + Q_{rad}$ (mW)	Result for pillar and radiation together $Q_{pill+rad}$ (mW)	Radiation suppression Q_{sup}/Q_{pill} (%)
0.75	0.260	4.434	4.694	4.690	1.5
1.15	0.260	6.777	7.037	7.031	2.3
4.00	0.260	23.051	23.311	23.294	6.5

Table 4.1: Modelling results for the three indicated heat flows between the glass sheets. The corresponding radiation suppression results, as defined by equation 4.5, are expressed as percentages. Three glass-to-glass center-of-glass radiative U-values were considered for the radiative heat flows.

It is important to note that, for typical conditions, the radiative heat flow in vacuum glazing is *approximately* linear with the temperature difference across the glazing - as per equations 3.3a and b. Since the heat flow through the pillar is a conductive flow, it is also linear with same temperature difference. Therefore, the radiation suppression

results given above, which are based on a simulated glass-to-glass temperature of 1 °C, are approximately independent of the temperature difference across the glazing.

The effect of the radiation suppression on the total center-of-glass U-value of a vacuum glazing can be accounted for by correcting either the radiative or pillar array components of this U-value. The correction to the radiative U-value involves a calculation of the total radiation suppression on the basis of the total number of pillars. However, since this requires the specification of both the pillar separation and the glazing area, it is preferable to employ the following more general correction which can be applied to the pillar array U-value. Since both the radiation suppression (as defined by equation 4.5) and the pillar array U-value (as calculated by equation 4.3) are based on the heat flow through a single pillar, the corrected pillar array U-value, U'_{array} , is:

$$U'_{array} = (1 - \frac{Q_{sup}}{Q_{pill}}) U_{array} , \quad (4.6)$$

where Q_{sup}/Q_{pill} is the radiation suppression. Therefore, the total glass-to-glass, center-of-glass U-value which is corrected for the effect of radiation suppression is:

$$U'_{total} = U_{rad} + U'_{array} . \quad (4.7)$$

Since changes to the total U-value result in changes to the internal surface temperatures which in turn affect U_{rad} , an exact solution for the corrected total U-value requires an iterative solution. This involves solving equations 4.6 and 4.7 in conjunction with equations 3.3 (for adjusting U_{rad}) for specified conditions, as well as re-calculating the radiation suppression. However, a sufficiently approximate result can be obtained by a straightforward substitution of the initial estimations for U_{rad} , Q_{sup}/Q_{pill} and U_{array} into these equations. For the radiative U-values and the modelled radiation suppressions of Table 4.1, and for the typical pillar array U-value of 0.40 Wm²K⁻¹ (which corresponds to a pillar radius and spacing of 0.125 mm and 25 mm), this approach shows that the radiative heat transfer suppression leads to only a 0.5 to 0.6 % correction to the total U-values considered above (a similar correction can be expected for the slightly different more typical U-values). The maximum uncertainty in the estimation of the U-value for vacuum glazing is about ± 2 % (this uncertainty is due to the error associated with the TCA measurements from which the U-value is determined). Since the correction for radiation suppression is less than this uncertainty, it can be neglected.

It was found that the radiative heat transfer suppression is not strongly dependent on the pillar radius. For a radiative U-value of $4 \text{ Wm}^{-2}\text{K}^{-1}$, and a pillar radius of $a = 0.125 \text{ mm}$, the radiation suppression changes by only 6 % of the initial fractional value over a 150 % increase in the pillar radius. This is as expected. An increase in the pillar radius results in an approximately proportional increase in the heat conduction through the pillar ($Q_{pill} \sim 2k_{glass}a$ by equation 4.2) but it does not affect the spreading (i.e. the distribution) of this heat flow beyond a short distance from the pillar. It is recalled from the beginning of this section that most of the contribution to the radiation suppression comes from the variation in the temperature of each internal surface beyond a few millimeters from the pillar. Therefore, since a change in the pillar radius merely scales this variation and thus the pillar conduction in this region and since, as mentioned above, the radiation suppression is proportional to Q_{pill} , the radiation suppression should indeed be essentially independent of the pillar radius.

The conclusion that most of the contribution to the radiation suppression comes from the small but extensive variation in the internal surface temperatures beyond a small radius from the pillar is supported by the fact that a good estimation of the radiation suppression can be obtained simply by replacing the pillar array with a uniform slab of an appropriate U-value. This U-value is equal to U_{array} for the calculation of Q_{pill} and the parallel combination of U_{array} and U_{rad} for the calculation of $Q_{pill+rad}$. As before, Q_{rad} is calculated with a slab of a uniform U-value of U_{rad} . The heat flows are determined on the basis of the corresponding U-values in series with the equivalent U-values for the two glass sheets (for each sheet, $U_{glass} = k_{glass}/t$). For example, with $U_{array} = 0.40 \text{ Wm}^{-2}\text{K}^{-1}$ and $U_{glass} = 250 \text{ Wm}^{-2}\text{K}^{-1}$, the radiation suppressions for the radiative U-values of 0.75, 1.15 and $4.00 \text{ Wm}^{-2}\text{K}^{-1}$ are, respectively, 1.2, 1.8 and 6.2 % - all of which are close to the corresponding results of Table 4.1 which were determined from the more exact modelling approach.

4.5 Conclusion

The conductive heat transfer between the glass sheets via the support pillars represents a significant fraction of the total center-of-glass heat flow across a vacuum glazing. It is therefore important to be able to quantify this conductive heat transfer, and to know how to reduce it. Since the pillars of vacuum glazings are made from materials with thermal conductivities that are greater than the thermal conductivities of glass, the thermal resistance associated with the plane parallel heat flow in each pillar is much

smaller than that associated with the spreading heat flow in the glass sheets. Thus, the thermal resistance for the pillar (or more correctly, for the pillar-glass system) is approximately equal to that for the spreading heat flow for a semi-infinite glass thickness: $1/2k_{glass}a$. In this expression, k_{glass} is the thermal conductivity of glass, and a is the pillar radius. It is important to note that this expression is based on the above assumption that the thermal resistance for the heat flow in the pillar is much smaller than that for the spreading heat flow in the glass, and the assumption that the glass sheets are of a semi-infinite thickness. Experimental and finite and element modelling results which reflect the realistic effects of both a thermal resistance for the heat flow in the pillar, and of a finite glass thickness, show that the analytic solution provides a close estimation of the actual thermal resistance.

The U-value for a pillar array is inversely proportional to the thermal resistance for an individual pillar, and also to the square of the pillar separation - i.e. $U_{array} = 2k_{glass}a/\lambda^2$. Thus, the pillar array U-value can be reduced by decreasing the pillar radius, and/or by increasing the pillar spacing. Obviously, changes to these dimensions - especially to the pillar separation - affect the stresses in the glass sheets and thus the probability of glazing failure. Therefore, the selection of the pillar radius and separation for a vacuum glazing involves trade-offs between two conflicting design goals: the minimization of the pillar array U-value, and maintenance of the structural integrity of the glazing. These issues are further discussed in Chapters 7, 8 and 9.

A finite element model for the analysis of the effects of the heat conduction through a pillar has been developed. Pillar thermal resistances obtained with this model are consistent with experimental results and they are close to thermal resistances obtained with the less exact above analytic expression. This model for an individual pillar is particularly useful for calculating the temperature non-uniformities at the external and internal glass surfaces due to the pillar conduction. These results can be used to determine the glass surface temperature across an array of pillars. Modelling results show that for typical pillar dimensions and spacings, and for the ASTM standard fenestration test conditions, the significant glass surface temperature non-uniformities due to the heat conduction through individual pillars mostly occur close to the pillars - especially at the internal surfaces. The average external glass surface temperature determined by this exact modelling approach is in very good agreement with that based on the pillar array U-value calculated by the analytic method.

The temperature non-uniformities at the internal surfaces which are caused by pillar conduction slightly reduce the local radiative heat transfer around the pillars. However,

this suppression of the radiative heat transfer has a negligible effect on the total center-of-glass U-value for typical glazings.

Infrared thermographic measurements of a vacuum glazing sample are consistent with results obtained with the finite element and analytic modelling methods of this chapter. Since the thermographic results are more relevant to the study of edge conduction, which is presented in the next chapter, a comparison between these measurements and the theoretical results for the effects of pillar conduction is given in that chapter.

4.6 List of Symbols

a	pillar radius
h	pillar height
h_{warm}	warm side heat transfer coefficient
k_{glass}	thermal conductivity of glass
k_{pillar}	thermal conductivity of pillar
N	number of nodes in the pillar, and in the glass within a radius of 0.5 mm from the axial centerline of the pillar and within an axial distance of 0.5 mm from the pillar
Q_{pill}	heat conduction through a single pillar
$Q_{pill+rad}$	heat transfer with both pillar conduction and radiative heat transfer modelled together
Q_{rad}	radiative heat transfer
Q_{sup}	radiation suppression due to a single pillar
$R_{parallel\ flow}$	thermal resistance associated with the plane parallel heat flow in the pillar
R_{pillar}	thermal resistance of a single pillar
$R_{spreading}$	thermal resistance for the spreading heat flow in the glass
t	glass sheet thickness
$T_{air,cold}$	cold side air temperature
$T_{air,warm}$	warm side air temperature
T_{cold}	external surface temperature of cold side glass sheet
T_{warm}	external surface temperature of warm side glass sheet
U_{array}	pillar array U-value
U_{glass}	equivalent U-value of a glass sheet
U_{rad}	radiative U-value

Greek symbols

λ	pillar separation

Superscript

corrected for the effect of radiation suppression

4.7 References

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5. Edge Conduction

5.1 Introduction

Thus far, only two heat flows across the glazing have been considered: the radiative heat transfer between the internal glass surfaces, and the heat conduction between the glass sheets via the support pillars. For a vacuum glazing with two sheets of K glass, a typical pillar radius of 0.125 mm and a typical pillar separation of 25 mm, and for the ASTM standard test conditions, these flows respectively constitute about two thirds and one third of the total center-of-glass heat flow across the glazing. There is another significant heat flow which must be considered in quantifying the total thermal performance of a vacuum glazing - the heat conduction along the glass sheets and through the edge seal, i.e. the edge conduction.

The edge seal acts as a thermal short circuit between the glass sheets. Edge conduction arises from the heat flow along the warm side sheet, through the edge seal and along the cold side sheet. The significance of edge conduction goes beyond the fact that it decreases the overall thermal performance and the condensation resistance of the glazing. The temperature non-uniformities associated with edge conduction (the edge effect) cause stresses in the edge region which can result in the structural failure of the glazing. This is further discussed in Chapters 6 and 8.

In this chapter, several modelling techniques are used to study edge conduction and its effects. A simple one-dimensional analytic model is developed, expressed in finite-difference form and solved numerically. The finite difference solution to this analytic model is theoretically validated by results from a three-dimensional finite element model. U-values calculated with the finite difference model are shown to be in close agreement with those determined with a commonly used commercial fenestration thermal analysis model.

Infrared thermographic measurements of a vacuum glazing sample are compared with results from the above mentioned finite element model, as well as with results from the finite element and analytic models presented in the previous chapter, in order to validate these simulation techniques. Infrared thermography and modelling approaches are used to study the effectiveness of reducing edge conduction with edge insulation, recessed edges, and the addition of a third pane of glass.

5.2 Modelling

5.2.1 Analytic Model for Edge Conduction

The complete analysis of the heat flow near the edges of a vacuum glazing requires the solution of a set of coupled three-dimensional equations. As a simpler alternative to such an analysis, a one-dimensional analytic model has been developed with only one spatial variable - the distance x along the glass sheets. This analysis involves the calculation of the heat flows at the internal and external surfaces, and along the lengths of the glass sheets. The basic assumption in the model is that the temperature of the glass is uniform throughout the thickness of each sheet, and that temperature gradients exist only along the planes of the sheets.

The specific situation to be considered is shown in Fig. 5.1a. Two sheets of soda-lime glass, each of thickness t , width L , and thermal conductivity k , are joined at their edges by a solder glass seal. Between the two sheets is an evacuated space. The dominant modes of heat transfer across this space are the radiative heat transfer between the internal glass surfaces and the conduction through the support pillars. The bottom sheet is exposed to a higher ambient temperature, T_i , than the top sheet, where the ambient temperature is T_o (these temperatures respectively correspond to the inside and outside ambient conditions). The warm and cold side external glass surface temperatures are, respectively, $T_1(x)$ and $T_2(x)$. The total convective, conductive and radiative heat transfer between each external glass surface and ambient is governed by the corresponding heat transfer coefficient: h_1 on the warm side, and h_2 on the cold side. In the context of quantifying the heat flow across glazings, the heat transfer coefficient is also referred to in the literature as the surface heat transfer coefficient, surface conductance, film conductance or film coefficient. The external glass surfaces near the edge seal are insulated over widths w_1 and w_2 along the warm and cold sides of the glazing. These widths, which are referred to as insulation recess depths, are measured from the origin at the inner edge of the seal.

As shown in Fig. 5.1b, the glass temperature at the origin is T' . For $x > 0$, the temperature of the warm side glass increases and approaches a uniform center-of-glass temperature at large values of x . For $x < 0$, the temperature of the cold side glass decreases and, at large values of x , also approaches a uniform center-of-glass temperature. The width and thickness of the edge seal are not considered in this one-dimensional analytic model. Therefore, no temperatures are presented for $x < 0$ and the temperatures of the glass sheets are equal at $x = 0$.

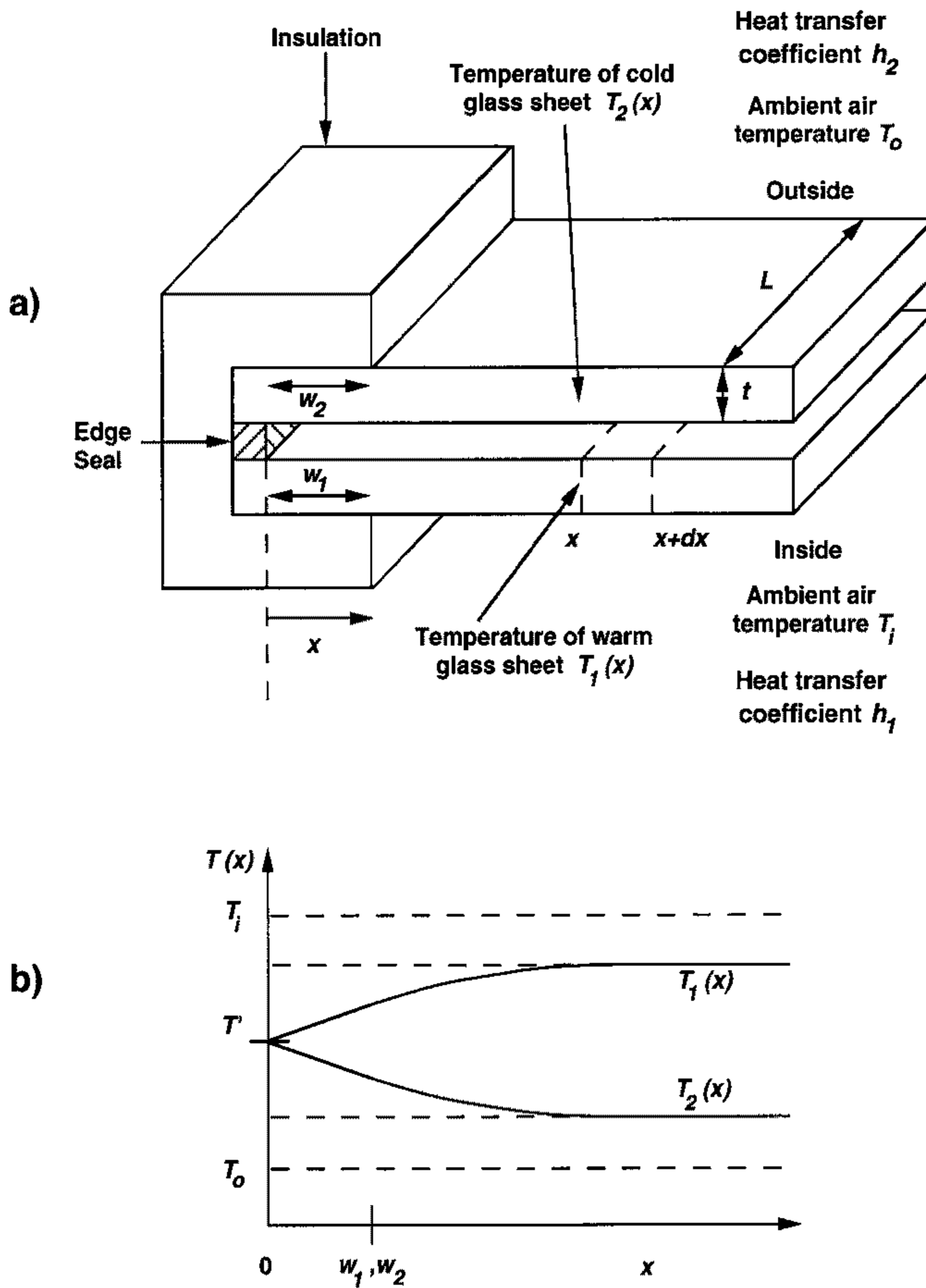


Fig. 5.1: (a) Schematic diagram of the geometry and conditions considered for the analytic model. Two sheets of glass are joined at their edges by a solder glass seal and are separated by an evacuated space. Each sheet is of thickness t , and width L . The sheets are exposed to warm and cold side air temperatures of T_i and T_o , and to corresponding heat transfer coefficients of h_i and h_2 . The glass surfaces near the edge seal are insulated over widths w_1 and w_2 along the warm and cold sides (these widths are measured from the inner edge of the seal, at $x=0$). (b) The conduction along the glass sheets and through the edge seal leads to variations in the temperatures along the glass sheets.

As mentioned in Chapter 3, if the temperature difference between the two glass sheets is not too large, the U-value for the radiative heat transfer across the evacuated space can be approximated with a radiative U-value:

$$U_{rad} = 4\epsilon_{eff}\sigma T_{ave}^3, \quad (5.1)$$

where ϵ_{eff} is the effective emittance of the two internal glass surfaces, σ is the Stefan-Boltzmann constant, and T_{ave} is the average of the internal glass surface temperatures (or, in the present case, the average of the two uniform center-of-glass temperatures).

As explained in Chapter 4, the pillar array U-value for the heat conduction between the glass sheets via the pillars can be approximated by equation 4.3 as:

$$U_{array} = 2ka/\lambda^2, \quad (5.2)$$

where k is the thermal conductivity of the glass, a is the pillar radius and λ is the pillar separation. Since the radiative and pillar array U-values are in parallel, the total glass-to-glass center-of-glass U-value, U_{g-g} , is equal to the sum of U_{rad} and U_{array} (which are defined by equations 5.1 and 5.2):

$$U_{g-g} = U_{rad} + U_{array}. \quad (5.3)$$

Figure 5.2 shows a cross-sectional element of the warm side glass sheet from x to $x+dx$, in the non-insulated region. The heat flows into and out of this element are:

1. Conductive heat flow along the glass sheet out of the element at x : $Q_{cond}(x)$
2. Conductive heat flow along the glass sheet into the element at $x+dx$: $Q_{cond}(x+dx)$
3. Heat flow out of the element by internal processes (radiative heat transfer, and conduction through the support pillars) between the glass sheets: dQ_{int}
4. Heat flow into the element by external processes (convective, conductive and radiative heat transfer) with the warm side ambient at temperature T_i : dQ_{ext}

Defining the heat flow into the element as positive, these heat flows are governed by the following equations:

$$Q_{cond}(x) = -k t L \left. \frac{dT_1(x)}{dx} \right|_x \quad (5.4)$$

$$Q_{cond}(x+dx) = k t L \left. \frac{dT_1(x)}{dx} \right|_{x+dx} \quad (5.5)$$

$$dQ_{ext} = h_1 L dx [T_i - T_1(x)] \quad (5.6)$$

$$dQ_{int} = -U_{g-g} L dx [T_1(x) - T_2(x)] \quad (5.7)$$

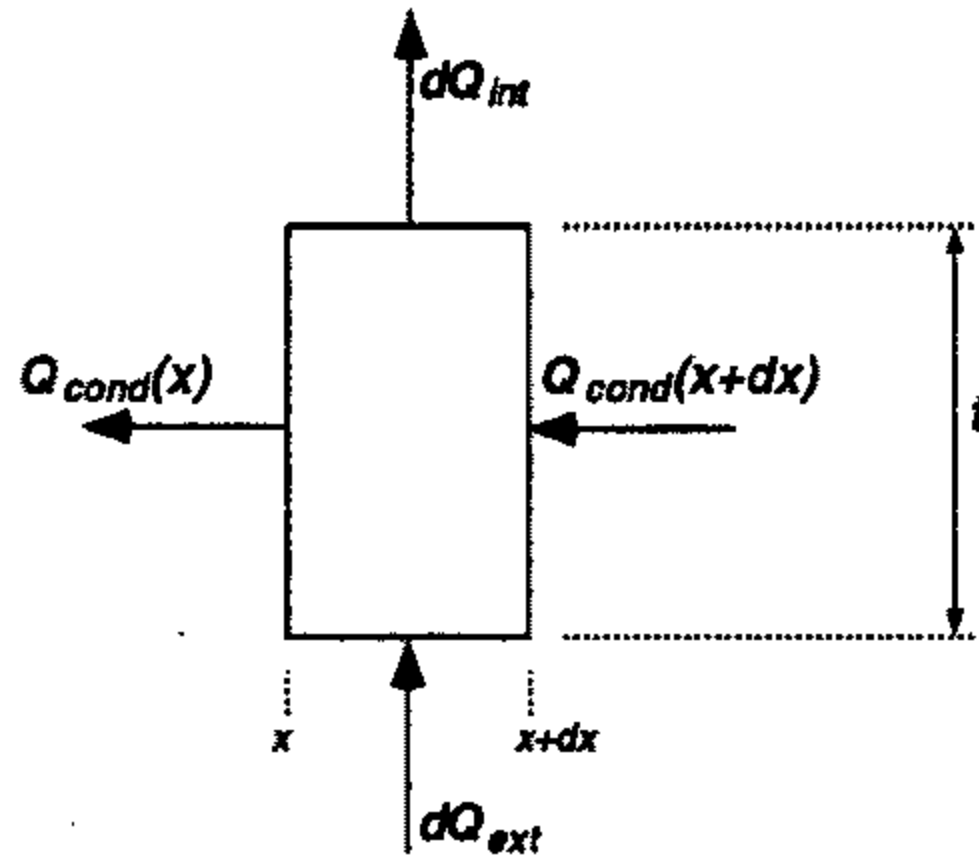


Fig. 5.2: The heat flows into and out of a cross-sectional element of the warm side glass sheet of Fig. 5.1a, from $(x$ to $x+dx)$.

The net heat flow through the element is zero. Balancing the heat flows into and out of the element yields the following differential equation:

$$\left(\frac{k t}{h_1 + U_{g-g}} \right) \frac{d^2 T_1(x)}{dx^2} - T_1(x) + \left(\frac{h_1 T_i + U_{g-g} T_2(x)}{h_1 + U_{g-g}} \right) = 0 \quad (5.8)$$

A similar analysis for an element in the non-insulated region on the cold side sheet yields the following differential equation:

$$\left(\frac{k t}{h_2 + U_{g-g}} \right) \frac{d^2 T_2(x)}{dx^2} - T_2(x) + \left(\frac{h_2 T_o + U_{g-g} T_1(x)}{h_2 + U_{g-g}} \right) = 0 \quad (5.9)$$

Equations similar to equations 5.8 and 5.9 apply to the warm and cold side insulated regions; for each side, the appropriate heat transfer coefficient (h_1 or h_2) is replaced by a thermal conductance which characterizes the heat flow between the glass and

ambient via the insulation. This conductance, h'_1 or h'_2 , is taken to be equal to the series combination of the insulation conductance and the ambient heat transfer coefficient to which the insulation is exposed - i.e. the inverse of the sum of the inverses of the insulation conductance and the corresponding heat transfer coefficient. This approximation ignores the fact that the ambient heat transfer coefficients include components for the radiative heat transfer with glass surfaces and not with other surfaces, such as insulation, which have different emittance properties.

The two coupled differential equations, 5.8 and 5.9 (and the analogous equations for the insulated regions), can be solved numerically, as described below in Section 5.2.2. The boundary conditions for this solution, and for the simple cases considered below, come from applying continuities in the temperature and the heat flow at the edge seal: $T_1(0)=T_2(0)$; $dT_1(0)/dx=-dT_2(0)/dx$. These boundary conditions are justified simply by the fact that this one-dimensional solution does not simulate the finite thickness or width of an actual edge seal. However, it is important to note they do in fact closely approximate the conditions for the situation with an actual seal. As shown by modelling results presented below, the continuity in temperature is approximately correct for an actual edge seal because the seal presents only a small thermal resistance to the edge conduction. The continuity in heat transfer is physically realistic because there should be no significant heat flow into or out of the seal through the edges along its small thickness.

In certain simple cases, equations 5.8 and 5.9 can be solved exactly. It is instructive to consider such solutions briefly since they provide useful insights into the physical processes which determine the heat transfer near the edges of the glazing.

The simplest case to consider is that for which the glazing is perfectly insulating, i.e. $U_{g-g}=0$, and for which there is no edge insulation. Equations 5.8 and 5.9 then decouple and the temperature on each side of the glazing is given by:

$$T_i - T_1(x) = (T_i - T') \exp(-x / \sqrt{kt / h_1}) \quad (5.10)$$

$$T_2(x) - T_o = (T' - T_o) \exp(-x / \sqrt{kt / h_2}) , \quad (5.11)$$

where T' , the temperature at the origin (the inner edge of the seal), is derived from applying a continuity in the heat transfer at $x=0$ and is:

$$T' = \frac{\sqrt{h_1}T_i + \sqrt{h_2}T_o}{\sqrt{h_1} + \sqrt{h_2}}. \quad (5.12)$$

In this simple case, the temperature of each glass sheet approaches the centre-of-glass value exponentially with a characteristic distance of:

$$l = \sqrt{kt/h}, \quad (5.13)$$

where h is the heat transfer coefficient for the side being considered (i.e. h_1 or h_2). For this case, the additional heat flow per unit length of the edge due to the edge conduction is:

$$Q_{edge} = \frac{kt(T_i - T_o)}{\sqrt{kt/h_1} + \sqrt{kt/h_2}}. \quad (5.14)$$

It is of interest to note that this is equivalent to the heat flow through the thermal resistance of a section of glass with a length equal to the sums of the characteristic distances for the two sides of the glazing, as defined by equation 5.13. This provides a useful physical insight for the consideration of the next simple case.

A second simple case which can be solved exactly is that for a perfectly insulating sample, i.e. $U_{g-g}=0$, as above but with adiabatically insulated surfaces extending over recess depths of w_1 and w_2 from the edge. In this case, the glass temperature varies linearly in the insulated region, and exponentially in the uninsulated region with the same characteristic distance given by equation 5.13. The thermal resistance circuit for such a glazing is shown in Fig. 5.3. Since $U_{g-g}=0$, the only thermal resistance between the two air temperatures is that associated with the heat flow along the glass sheets and through the edge seal. The thermal resistance on each side of the glazing is equal to the resistance of a piece of glass with a length equal to one characteristic distance, as determined by equation 5.13, in series with the resistance of the insulated length of glass. Therefore, the edge conduction per unit length of edge is proportional to the air-to-air temperature difference divided by the sum of the glass thermal resistances on both sides of the glazing:

$$Q_{edge} = \frac{kt(T_i - T_o)}{w_1 + w_2 + \sqrt{kt/h_1} + \sqrt{kt/h_2}}. \quad (5.15)$$

For uninsulated edges, i.e. the previously considered simple case, this expression reduces to equation 5.14.

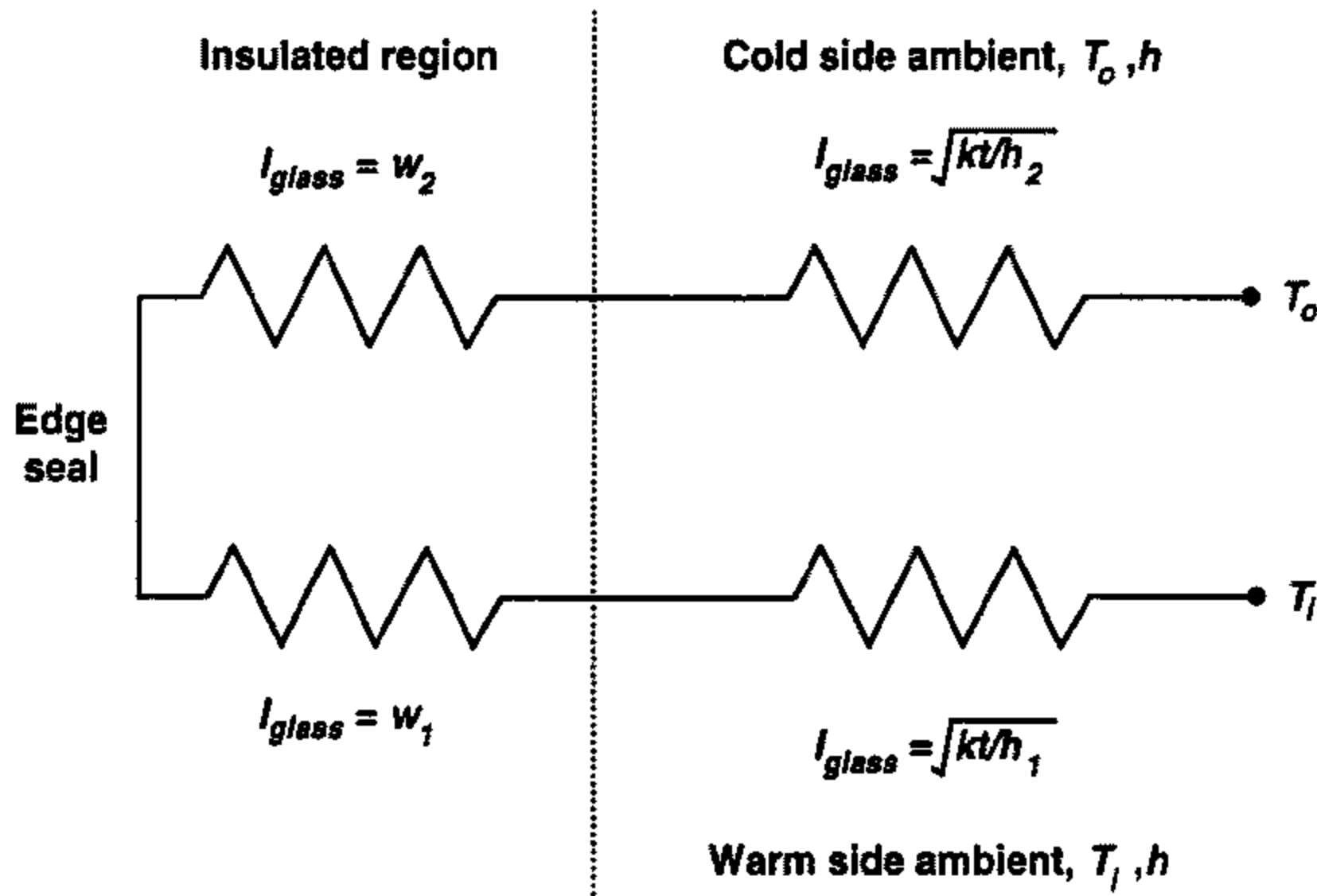


Fig. 5.3: The thermal resistance circuit for the edge conduction in a perfectly insulating vacuum glazing (i.e $U_{g-g}=0$) with adiabatically insulated surfaces extending over recess depths of w_1 and w_2 . The glazing is exposed to air temperatures of T_1 and T_2 , and to corresponding heat transfer coefficients of h_1 and h_2 . The length, l , of each glass component of the circuit is given. The thermal resistance for each component is equal to the corresponding length, divided by the product of the cross-sectional area of the glass sheet and the thermal conductivity of glass ($1.0 \text{ Wm}^{-1}\text{K}^{-1}$). The thermal resistance of the edge seal is not considered.

For a sample which is not perfectly insulating, an approximate solution can be obtained for the case of equal heat transfer coefficients, $h_1=h_2$, provided that the heat transfer between the sheets in the insulated region is neglected. In this case, simple exponential temperature dependencies occur in the uninsulated region with a characteristic distance given by equation 5.13, for which $h = h_1 + 2U_{g-g}$.

5.2.2 Finite Difference Solution to The Analytic Model

The one-dimensional analytic model discussed in the preceding section leads to two coupled differential equations for the two glass surface temperatures: equations 5.8 and 5.9. These equations can be expressed in finite difference form and solved numerically. Instead of considering an edge seal of finite thickness and width, the seal is modelled by applying the same boundary conditions given above for the continuities in the temperature and the heat flow at the origin: $T_1(0)=T_2(0)$; $dT_1(0)/dx=-dT_2(0)/dx$. As explained above for the analytic model, insulated surfaces are simulated by changing the heat transfer coefficients of the solution equations; the heat transfer coefficient over each insulated surface (h_1 or h_2) is replaced by a thermal conductance (h'_1 or h'_2) which is equal to the series combination of the insulation

thermal conductance and the ambient heat transfer coefficient (h_1 or h_2) to which the insulation is exposed. An adiabatically insulated surface is simulated with a zero heat transfer coefficient over the insulated region. The geometry and conditions represented by this model are shown in Fig. 5.4. This one-dimensional finite difference solution to the analytic model is also referred to below as the finite difference model (FDM). It is solved with a computer program written in Fortran code. As shown below, for the conditions considered in this study, sufficient convergence in the FDM results is achieved with a modelled glass sheet length of 200 mm, and with a uniform node spacing of 0.5 mm; although the total number of nodes can be reduced with a larger node density near the seal, this was not necessary and thus the nodes were not graded. All results from this model correspond to a glass thickness of $t=4$ mm (this thickness is used in the solution equations and for calculating the heat flows). The thermal conductivity of the glass is equal to that given in Appendix 1 for soda-lime glass: $1.0 \text{ Wm}^{-1}\text{K}^{-1}$.

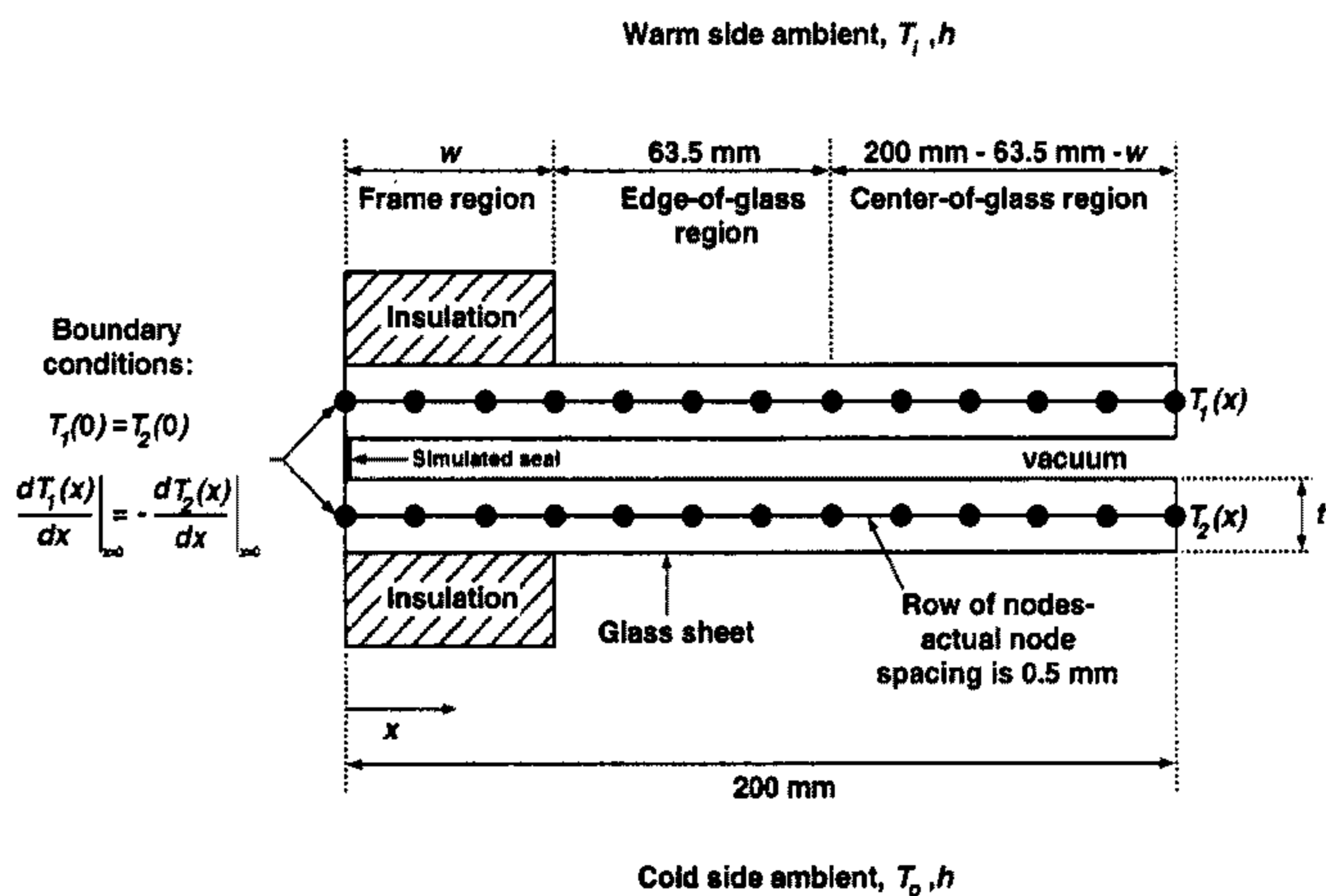


Fig. 5.4: The geometry represented by the finite difference solution to the analytic model for edge conduction. Each glass sheet is represented by a row of nodes. The temperatures along the warm and cold side rows of nodes are $T_1(x)$ and $T_2(x)$. Only the effect of an edge seal is simulated; the dimensions of an actual seal are not modelled. The boundary conditions for the solution, which are at the nodes at the origin (the position of the simulated seal) are indicated. The indicated regions are those over which the U-values are calculated, as described below in Section 5.2.4. In this figure, the insulation recess depth, w , is the same on both sides of the glazing. Not to scale.

5.2.3 Finite Element Model

A three-dimensional finite element model (FEM) of a square vacuum glazing has been constructed using the STRAND6 modelling package. The geometry of the model is shown in Figs. 5.5a and b. Due to symmetry considerations, it is only necessary to simulate a quarter section of the full glazing - as indicated in Fig. 5.6. The modelled quarter section corresponds to a 1 m by 1 m full glazing. It is noted that results could also be obtained from a one eighth triangular section of the glazing (for which the hypotenuse extends from the corner to the center of the glazing). However, since the required complex boundary conditions along the glazing diagonal can not be established with the modelling package, a one eighth section was not simulated.

As shown by Fig. 5.5b, a cross-section of the model represents the situation shown in Fig. 5.1 (without edge insulation): two sheets of 4 mm thick soda-lime glass are separated by an evacuated space, joined along their outer edges by a solder glass seal and exposed to ambient. Edge insulation, which is not shown in Figs. 5.5, is simulated by modifying the model in the manner described below. The edge seal is 6 mm wide, and the heights of the seal and evacuated space are uniform and equal to 0.2 mm. The thermal conductivities of the soda-lime glass and the solder glass are the same: $1.0 \text{ Wm}^{-1}\text{K}^{-1}$ (as given in Appendix 1). It is assumed that, in an actual glazing, the heat flows through the edges along the thicknesses of the seal and the glass sheets are insignificant. Therefore, in this model, these edges are defined to be adiabatic.

The internal heat transfer across the evacuated space is simulated as conduction through a slab between the glass sheets. The uniform thermal conductivity and thickness of this slab correspond to the glass-to-glass center-of-glass U-value, U_{g-g} , as defined by equation 5.3. The heat transfer with ambient at each external surface is simulated as conduction through a slab for which the uniform thermal conductivity and thickness correspond to the specified heat transfer coefficient. The temperature at the outer lateral surface of each external slab is equal to the required air temperature. As shown in Fig. 5.7, edge insulation on each side of the glazing is modelled by assigning a different uniform thermal conductivity to the corresponding external slab over the extent of the insulated region (which is equal to the width of the edge seal plus the insulation recess depth); this thermal conductivity and the thickness of the slab correspond to a thermal conductance which is equal to the series combination of the specified insulation conductance and the ambient heat transfer coefficient to which the insulation is exposed. Following the above notation, this thermal conductance for the external slab in the insulated region is denoted as h'_1 on the warm side and h'_2 on the cold side.

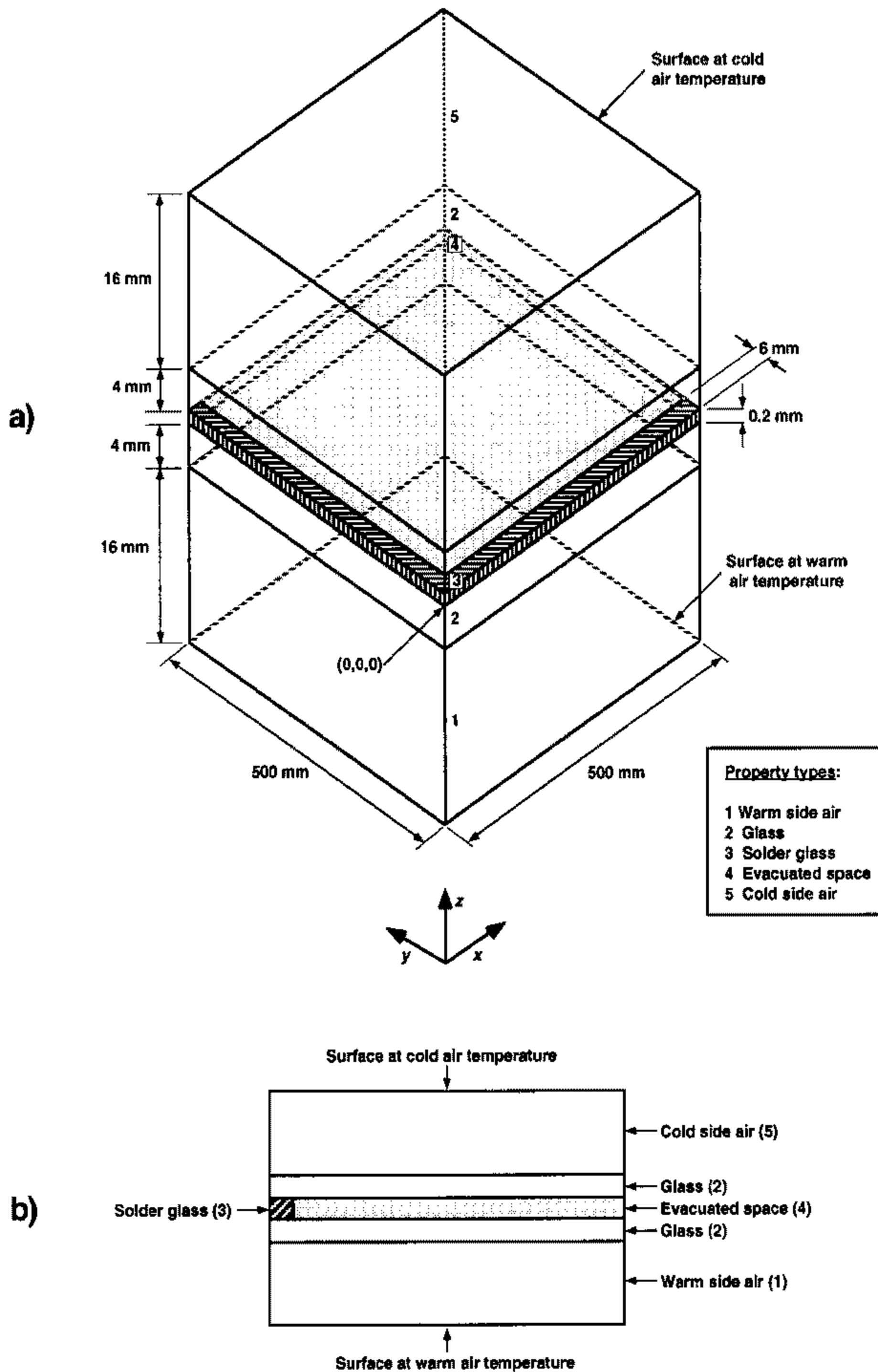


Fig. 5.5: The finite element model constructed with STRAND6 of a quarter section of a full 1 m by 1 m glazing: (a) full view, and (b) cross-sectional view (at a different scale). Two sheets of soda-lime glass (property type #2) are joined at their edges by a solder glass seal (hatched region, #3) and are separated by a slab representing an evacuated space (shaded region, #4). The heat transfer across the evacuated space is simulated as conduction through this slab, which has a thermal conductance equal to U_{g-g} . The heat transfer with ambient at the external glass surfaces is simulated as conduction through slabs which represent the warm and cold side air (property types #1 and #5, respectively). The temperatures at the outer lateral surfaces of these slabs are equal to the corresponding air temperatures. The thermal conductance of each external slab is equal to the heat transfer coefficient of the air to which it corresponds. These figures are not to scale.

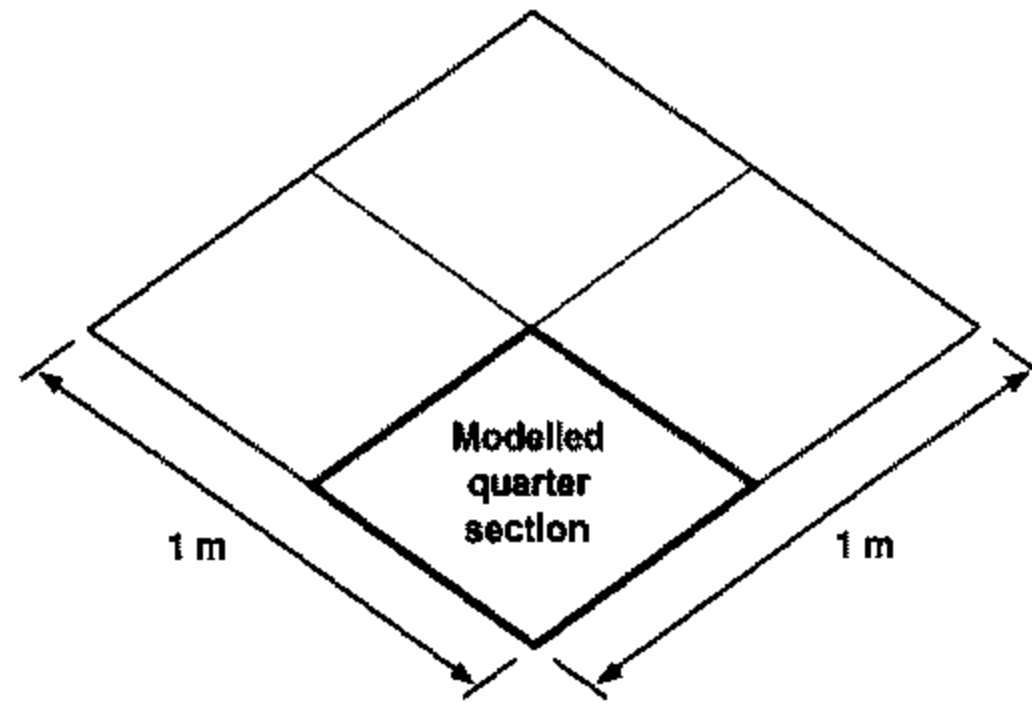


Fig. 5.6: Due to the symmetry of the full square glazing, it is only necessary to simulate a quarter section of the glazing. This figure shows the modelled quarter section of a 1 m by 1 m glazing.

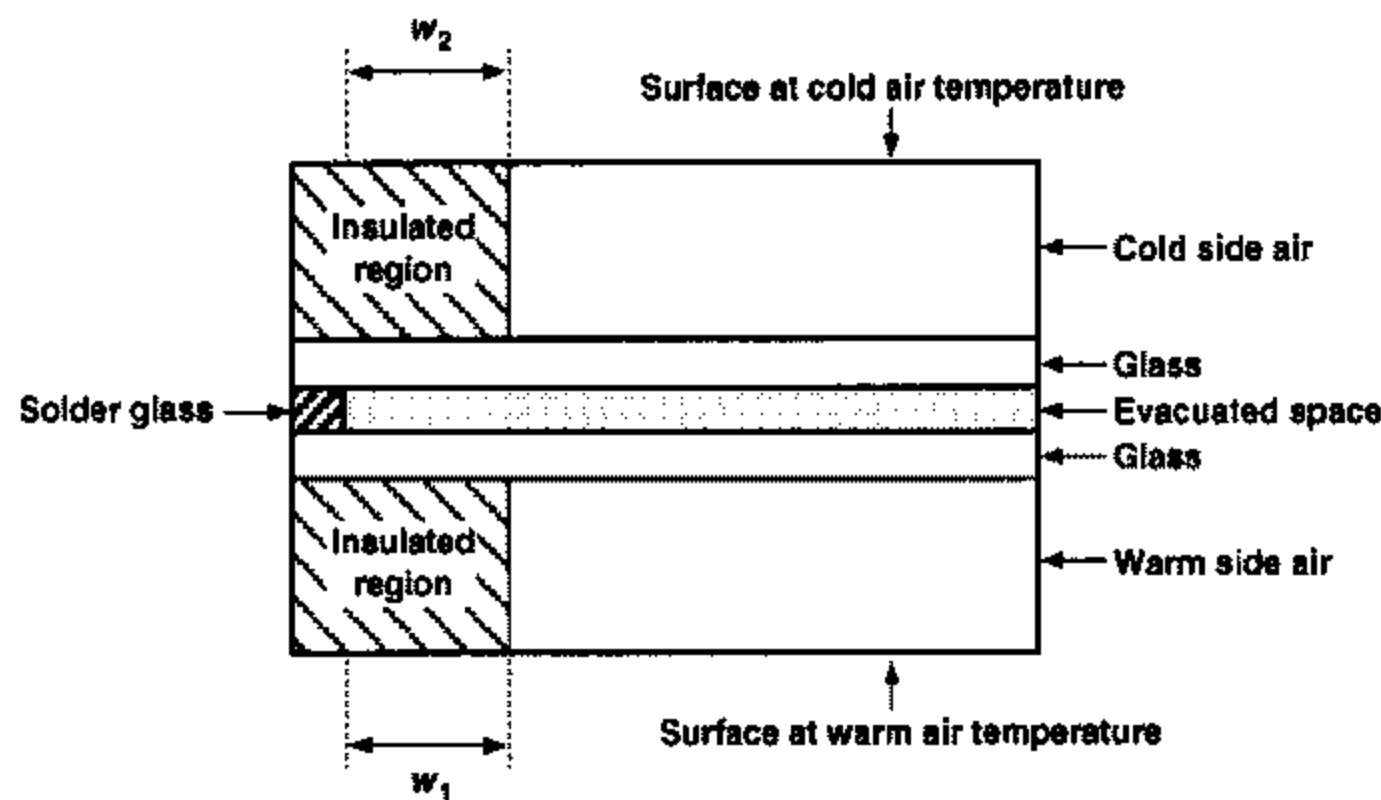


Fig. 5.7: A cross-section of the finite element model, as shown in Fig. 5.5b, except that insulated edges are considered. Edge insulation is simulated with the finite element model by changing the thermal conductivity of the air slabs over the extents of the insulated regions. For each side of the glazing, the extent of the insulated region is equal to the width of the edge seal plus the insulation recess depth. The recess depths are w_1 and w_2 on the warm and cold sides, respectively. Within the insulated region, the thermal conductance of each external slab is equal to the series combination of thermal conductance of the insulation and the heat transfer coefficient to which the insulation is exposed.

Since, in this finite element model and in the analytic and finite difference models presented above, the heat transfer coefficient over each insulated or uninsulated region is assumed to be uniform, these models do not simulate the effects associated with external convective flows which can lead to vertical temperature gradients across the glazing. However, as will be shown, such effects are not significant for the conditions considered in this study and thus the fact that they are not simulated is not a concern.

The finite element model simulates glass sheets of equal dimensions so that the edges of the sheets are coincident - like the structure shown in Fig. 5.1. In actual glazings, the dimension of one sheet is about 2 or 3 millimeters less than the other sheet so that there is a ledge at the edge of the glazing - as shown in Fig. 1.1. This ledge supports the molten solder glass while the seal between the sheets is being formed at the time of manufacture, and it is covered by a fillet of the extra fused solder glass that remains after this process. This actual edge configuration is not modelled because, due to the large thermal short circuiting effect of the edge seal, the extra length of glass and the external leftover solder glass do not have a significant influence on the edge conduction and the effects of this conduction.

The finite element model, like the finite difference and analytic models presented above, does not simulate the realistic situation with a frame around the glazing. Except for some of the experimental and modelling results presented in this chapter, frames are not considered in this thesis. There are two reasons for this. Firstly, it was decided that it would be best to obtain general, and easily comparable results which are not specific to any particular types of frames. Secondly, a proper consideration of frames would require the selection or design of an appropriate frame for vacuum glazing. Since this would be a significant work in itself, it was decided that any analyses with frames would be best covered in a separate study.

All of the FEM results presented in this chapter are external surface temperatures along an inner edge of the modelled quarter section - i.e. along a centerline of the full 1 m by 1 m glazing to which this section corresponds. Except where otherwise stated, the FEM results are for an edge seal width of 6 mm. The model contains 10 200 nodes. The node density decreases with distance from the edge.

5.2.4 Modelling of U-values

The thermal performance of fenestration systems can be evaluated on the basis of U-values (also referred to as U-factors). The 1989 American Society of Heating Refrigerating and Air-Conditioning Engineers (ASHRAE) Handbook of Fundamentals defines four window U-values [1] for the regions shown in Fig. 5.8 as:

U_{frame} : The U-value for the area of the frame, A_{frame} (the area between the edge of the window and the sightline - i.e. the area covered by the frame)

U_{edge} : The U-value for the edge-of-glass area, A_{edge} (which is calculated over a standard distance from the sightline - given below - towards the center of the window)

U_{cog} : The U-value for the center-of-glass area, A_{cog} (which is equal to the total window area minus A_{frame} and A_{edge})

U_{total} : The total U-value for the window, which is defined as:

$$U_{total} = \frac{A_{frame}U_{frame} + A_{edge}U_{edge} + A_{cog}U_{cog}}{A_{total}}, \quad (5.16)$$

where A_{total} is the entire window area, including that of the frame. The sightline is at the inner edge of the frame. According to the 1989 ASHRAE Handbook of Fundamentals, the distance from the sightline over which A_{edge} should be calculated is 65 mm [1]. However, in the present work, this distance is taken to be 63.5 mm (2.5 in.) because this is the more commonly used standard distance for calculating A_{edge} , and because it is equal to that which is specified by the FRAME3 model (described below) which is used to obtain comparative results [2]. Also, this distance of 63.5 mm is consistent with the distance for calculating the edge-of-glass area that is specified by the widely accepted WINDOW4 program (which is also described below) [3].

Window U-values are defined according to the net heat flow through the projected area of the warm side surface. The heat flow is summed over all elements between the location points which define the surface. The equation for each component (frame, edge-of-glass, and center-of-glass) is [4]:

$$U = \frac{Q_{warm\ side\ surface}}{\Delta T_{air-to-air} A_{projected}}, \quad (5.17)$$

where $Q_{warm\ side\ surface}$ is the above mentioned net heat flow, $\Delta T_{air-to-air}$ is the air-to-air temperature difference across the window, and $A_{projected}$ is the relevant area of the normal projection of the surface onto a plane perpendicular to the window.

The finite difference solution to the analytic model was used to calculate the U-values for a vacuum glazing with equal recess depths, w , of edge insulation on both sides of the glazing. For this calculation, the edge insulation is considered to represent an insulated "frame" that extends from the edge of the glazing over a distance of w

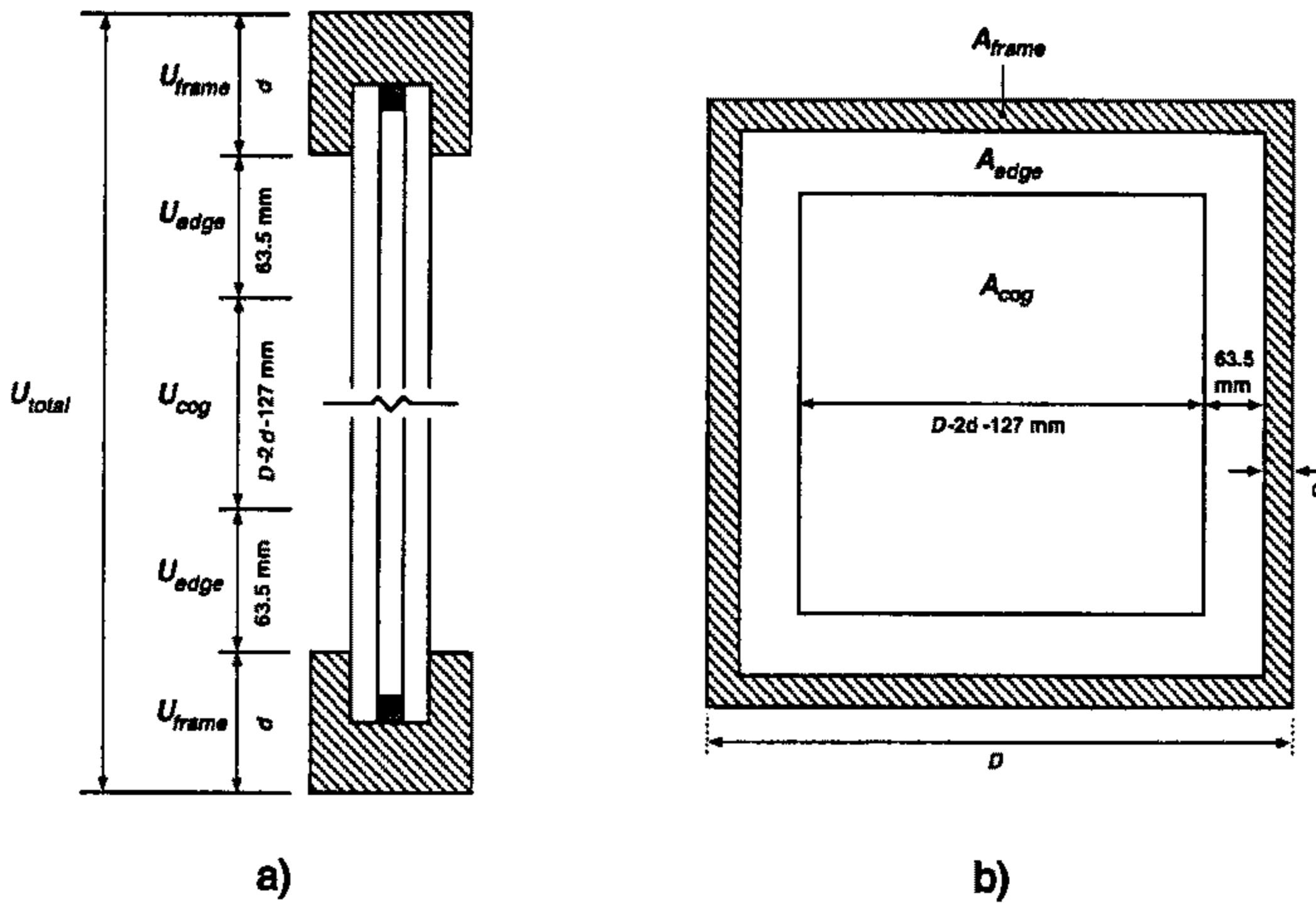


Fig. 5.8: The window U-values correspond to different regions of the window. The lineal extent of each region is shown in (a), and the corresponding full area of each region is shown in (b). The frame U-value, U_{frame} , is calculated over the area A_{frame} which is between the edge of the window and the sightline. The sightline is at a distance d from the edge of the window, where d is the extent of the frame. The edge U-value, U_{edge} , is calculated over the area A_{edge} which is between the sightline and 63.5 mm from the sightline. The center-of-glass U-value, U_{cog} , is calculated over the remaining area of the window, A_{cog} . The overall dimension of the square window considered in this figure is D . The same principles are applied to the calculation of the U-values for non-square windows.

from the sightline. Thus, U_{frame} is determined over the insulated region. The window U-values are calculated as follows. The elemental heat flows per unit length of edge are determined from the difference between the warm side glass and ambient temperatures. These elemental heat flows are summed over the extents of the frame, edge-of-glass, and center-of-glass regions (these extents are shown in Fig. 5.4) to give the respective regional one-dimensional heat flows per unit length of edge. For each region, the total heat flow is calculated by multiplying the one-dimensional heat flow per unit length of edge by the length of edge which corresponds to the lateral extent of the region (e.g. $4D-4w$ for the frame region of a square glazing with a dimension of D). The regional areas, as defined above and shown in Fig. 5.8 (but with d equal to w), are calculated on the basis of the specified size of the window. Each individual U-value (U_{frame} , U_{edge} , and U_{cog}) is calculated according to equation 5.17 by substituting into this equation the corresponding total heat flow and regional area for $Q_{warm\ side\ surface}$ and $A_{projected}$. The total U-value is calculated according to equation 5.16. It is noted that the model extends to only 200 mm from the edge where, for the

conditions considered in this study, the glass temperature is uniform. Therefore, total heat flows beyond this point, in the central area of larger glazings, are calculated on the basis of the difference between the warm side air and glass temperatures at 200 mm from the edge.

The above method for calculating the window U-values does not account for the enhanced heat flows that occur at the corners of actual glazings. However, this is consistent with the conventional approach to calculating U-values by area weighting results from two-dimensional simulations without any provisions for enhanced heat transfer at the corners [5].

It is recognized that the validity of using the one-dimensional model to calculate the window U-values is limited by the fact that the model does not simulate the complex two-dimensional heat flows near the edge of a glazing, and by the fact that it does not simulate an actual edge seal or frame. The purpose of the analysis with this simple model is not to provide accurate absolute results but rather to obtain estimations of the magnitudes of the effects of different insulation recess depths and thermal conductances, and also to give some insights into the physical processes involved.

Edge-of-glass region U-values for a vacuum glazing with different insulation widths and edge configurations were also calculated by Fred Beck of Lawrence Berkeley National Laboratory (LBNL) using a nationally accepted U.S. simulation methodology given by the National Fenestration Rating Council (NFRC) 100-91 procedure [5] based on WINDOW4 [6] and FRAME3 [2]. WINDOW4 is a public domain one-dimensional finite difference model for simulating the heat transfer through glazings and for calculating the total U-values of fenestration products. FRAME3 is a commercially available two-dimensional finite difference model for simulating the heat transfer through the frame and edge-of-glass regions of fenestration products. These two programs were used in combination to model the heat transfer through a vacuum glazing and, for comparison, the heat transfer through an atmospheric pressure glazing. All simulations with WINDOW4 and FRAME3 were performed for the ASHRAE winter conditions: air temperatures of 21.1 and -17.8 °C, and corresponding heat transfer coefficients of 7.41 and 29.00 Wm⁻²K⁻¹; the temperatures correspond to those specified by the above mentioned NFRC 100-91 procedure, and the heat transfer coefficients are determined from WINDOW4 and FRAME3 for the environmental conditions given in this procedure. The heat transfer coefficients only apply over the glass surfaces. The heat transfer coefficients over other surfaces, such as edge insulation, are similarly determined but differ from the coefficients over the glass surfaces because the different emittance properties of these

surfaces result in different radiative heat flows with the ambient. All WINDOW4 and FRAME3 calculations were performed for a glass thermal conductivity of $1.0 \text{ Wm}^{-1}\text{K}^{-1}$ and, except where otherwise specified, for a glass thickness of 4 mm. The window size for these calculations was equal to that which is listed by WINDOW4 as the generic ASHRAE residential size: 0.914 m by 1.220 m [6].

5.3 Experimental

Infrared thermography is the process of measuring temperatures with an infrared imaging radiometer [7]. It is non-invasive, and it can be used in a laboratory to obtain the surface temperatures of fenestration products [8]. Thermographic tests were performed on a vacuum glazing at the Lawrence Berkeley National Laboratory Infrared Thermography Facility to validate the analytic and finite element modelling for a number of edge configurations, and to explore the effectiveness of different methods of reducing the edge conduction. The development of the experimental techniques for these tests, the performance of the tests, and the analysis of the results were done by Fred Beck, Brent Griffith, Daniel Türlér and Dariush Arasteh of the Building Technologies Program, Energy and Environment Division, of Lawrence Berkeley National Laboratory, at Berkeley, California. Details about the apparatus, the reference emitter techniques and the issues associated with the thermographic techniques employed in these tests are described in detail by Griffith et al. [7].

A schematic diagram of the infrared (IR) test equipment is shown in Figs. 5.9a and b. A glazing sample is mounted vertically in an insulating mask wall (the test frame) which is situated between warm and cold environmental chambers in order to establish a temperature difference across the sample. Each chamber has a fixed temperature set point and thus, when the initial temperature variations have ceased, there is a steady state heat transfer across the glazing. In the cold chamber, there is a forced air flow across the glazing. This flow is directed parallel to the glazing surface, and upwards, between the glazing and a plenum which is about 100 mm from the sample. In the warm chamber, the air flow across the glazing occurs by natural convection and it is parallel to the glass surface, and downwards. Within the chambers, the air temperatures are measured with thermistor networks and type T thermocouples, and the air velocities are measured with hot wire anemometers.

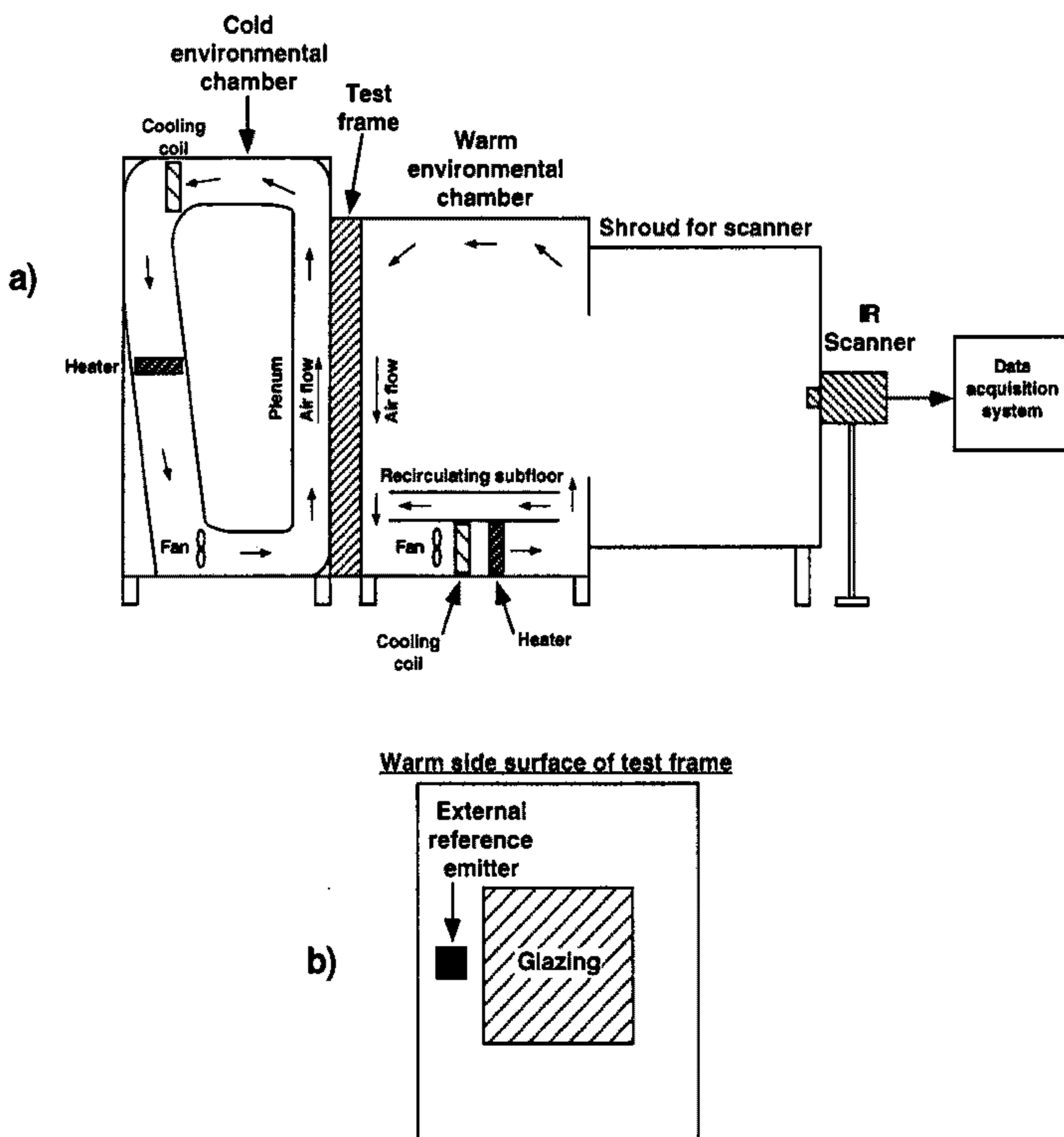


Fig. 5.9: Schematic diagram of the infrared thermography apparatus at Lawrence Berkeley National Laboratory: (a) cross-sectional view, and (b) warm side surface of the test frame. The glazing is vertically mounted in the test frame between the temperature controlled warm and cold environmental chambers. Air flows across each side of the glazing. Temperature measurements are made of the warm side external surface of the glazing (i.e. the surface shown in (b)) with an infrared imaging radiometer (IR scanner). An external reference emitter for the IR scanner measurements is in the vicinity of the glazing. For the two main tests described below, the surface of the emitter was about 40 mm closer to the IR scanner than the surface of the glazing. For most of the other tests, the emitter and glazing surfaces were co-planar.

An infrared imaging radiometer linked to a computer is used to measure the temperature of the glazing. The scanning type radiometer (or IR scanner) is located at the end of an adjustable shroud attached to the warm environmental chamber. Since glass is highly opaque in the infrared (beyond about 5 μm), and since the scanner detects radiation in the range of 8 to 12 μm , the results correspond only to the

temperatures at the external glass surface of the side of the glazing which faces the IR scanner: the warm side. An external reference emitter for the IR scanner measurements is located in the vicinity of the glazing. The emitter has a temperature controlled surface of a known emittance and temperature, and it aids in removing bias when making absolute temperature measurements [7]. For the two main tests described below, the surface of the emitter was about 40 mm closer to the IR scanner than the surface of the sample. For most of the other tests, the emitter and glazing surfaces were co-planar.

For all of the thermographic tests of this study, the set point temperatures of the warm and cold environmental chambers were 21.1 ± 0.3 °C and -17.78 ± 0.15 °C. The means of these temperatures correspond to those specified by the ASTM E 1423-91 standard for determining the steady state thermal transmittance of fenestration systems (based on exact conversions from the temperatures given by the standard in degrees Fahrenheit) [9]. For the two main tests described below, the warm chamber air velocity was $< 0.1 \pm 0.1$ ms⁻¹. The cold chamber air velocity was different for these tests: it was 4.3 ± 0.2 ms⁻¹ for one test, and it was 4.6 ± 0.2 ms⁻¹ for the other test.

The warm and cold side heat transfer coefficients corresponding to each main test were determined using standard fenestration test procedures [10] for the same air temperatures and velocities as during each glazing test, but with the glazing replaced by a calibrated transfer standard. This calibrated transfer standard consists of a 25 mm thick layer of expanded polystyrene foam insulation of a known thermal conductivity between two square sheets of uncoated glass which are 0.91 m in dimension and 4.8 mm in thickness [7]. Pairs of type T thermocouples are located directly across from each other on both sides of the foam. The heat flow across the sample can be determined from the thermocouple measurements and the thermal conductance of the foam. The warm and cold side heat transfer coefficients are calculated on the basis of this heat flow and the glass-to-air temperatures (the glass temperatures are based on the measured temperatures and the thermal conductances of the glass sheets). It is important to note that the radiative components of the heat transfer coefficients depend on the emittances of the external surfaces exposed to the air flows. Since, for both the calibrated transfer standard and the sample, these surfaces are uncoated glass, the heat transfer coefficients determined in this manner are applicable to the sample tests. For both main tests described below, the warm side heat transfer coefficient initially determined by this procedure was 9.3 ± 0.3 Wm⁻²K⁻¹. Due to the different cold side air velocities, there was a different cold side heat transfer coefficient for

each test: it was $31 \pm 2, -4 \text{ Wm}^{-2}\text{K}^{-1}$ for one test, and it was $30 \pm 2, -4 \text{ Wm}^{-2}\text{K}^{-1}$ for the other test. The heat transfer coefficients for the main tests are close to the ASTM standard fenestration test conditions: 8.3 and $30 \text{ Wm}^{-2}\text{K}^{-1}$ on the warm and cold sides, respectively [9].

It is noted that, after the modelling work that is presented below was completed, a re-examination of the calibrated transfer standard results showed that the warm side heat transfer coefficient was approximately $9.0 \pm 0.3 \text{ Wm}^{-2}\text{K}^{-1}$. It was decided not to redo the modelling on the basis of this new estimation of the warm side heat transfer coefficient because a simple analysis showed that the small difference between the initial and final estimations for this coefficient would not change the conclusions made below regarding the consistency between the theoretical and experimental results.

Infrared thermographic tests were performed on a 0.5 m by 0.5 m sample of vacuum glazing supplied by the University of Sydney (sample number 418). The edge seal of this sample is about 6 mm wide. The glazing consists of two sheets of K glass and it contains alumina pillars with a square cross-section. The pillar width and separation are 0.30 and 25 mm, respectively. The corresponding glass-to-glass center-of-glass pillar array U-value, as calculated by equation 5.2, is $U_{array}=0.54 \pm 0.01 \text{ Wm}^{-2}\text{K}^{-1}$. The pillar radius substituted into this equation corresponds to a circular cross-sectional pillar area which is equal to the square cross-sectional area of the pillars of the glazing sample (the justification for using this radius is given in Appendix 3). The glass-to-glass center-of-glass radiative U-value was calculated from measurements made with the thermal conductance apparatus described in Chapter 2. For internal glass surface temperatures of 22.0 and 2.1 °C, U_{rad} was determined to be $0.78 \pm 0.01 \text{ Wm}^{-2}\text{K}^{-1}$. For other internal glass surface temperatures (such as those for the infrared thermographic tests), this U-value must be adjusted according to an iterative solution based on equation 5.1 and the specified ambient conditions; an iterative solution is required because of the interdependency between the internal surface temperatures and U_{rad} . For the conditions considered in this study, the pillar array U-value is temperature independent.

Two main tests were performed on the vacuum glazing sample. For each test, there was foam insulation along the edges of one side of the glazing and extending over the width of the edge seal - as shown in Fig. 5.10. In one test, the edge insulation faced the cold chamber, and in the other test it faced the warm chamber. The edge insulation was 10 mm thick and had a thermal conductivity of $0.029 \text{ Wm}^{-1}\text{K}^{-1}$ (and

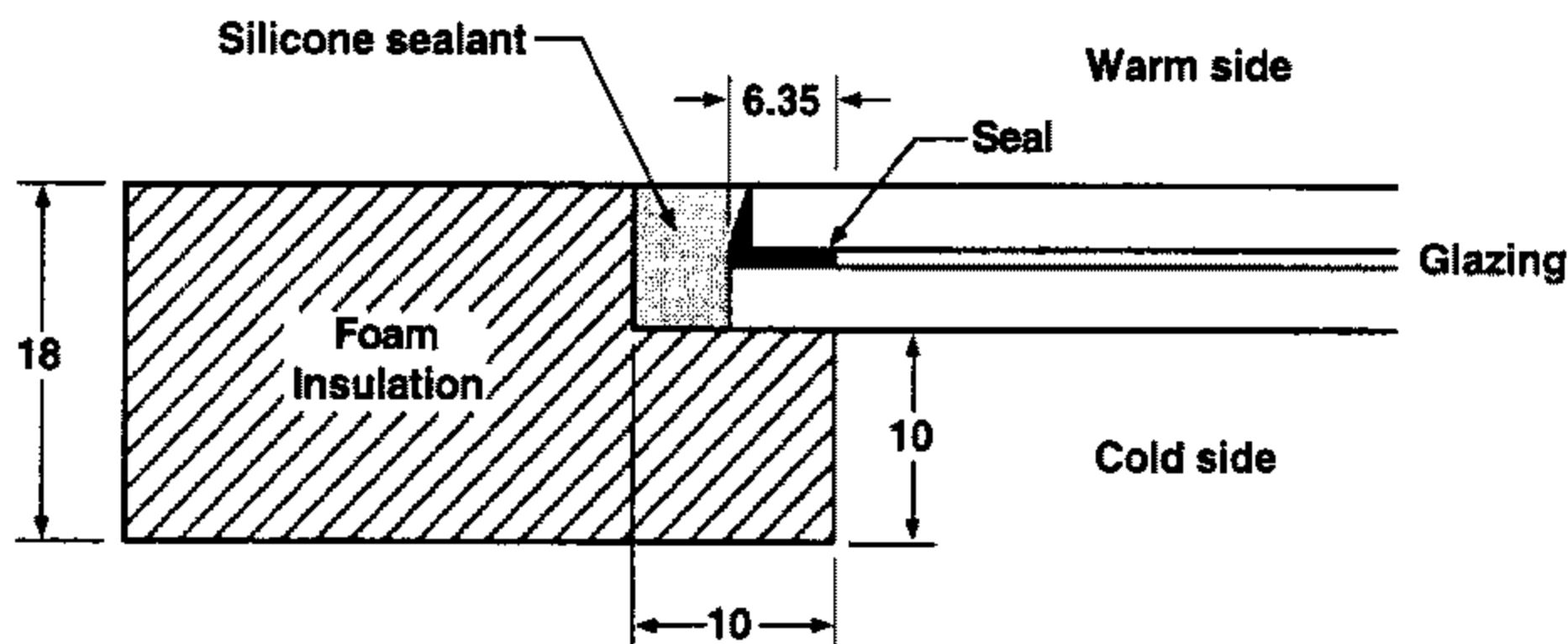


Fig. 5.10: The mounting and edge insulation configuration at the edges of the vacuum glazing for the two main IR thermographic tests. All dimensions in millimeters. Not to scale.

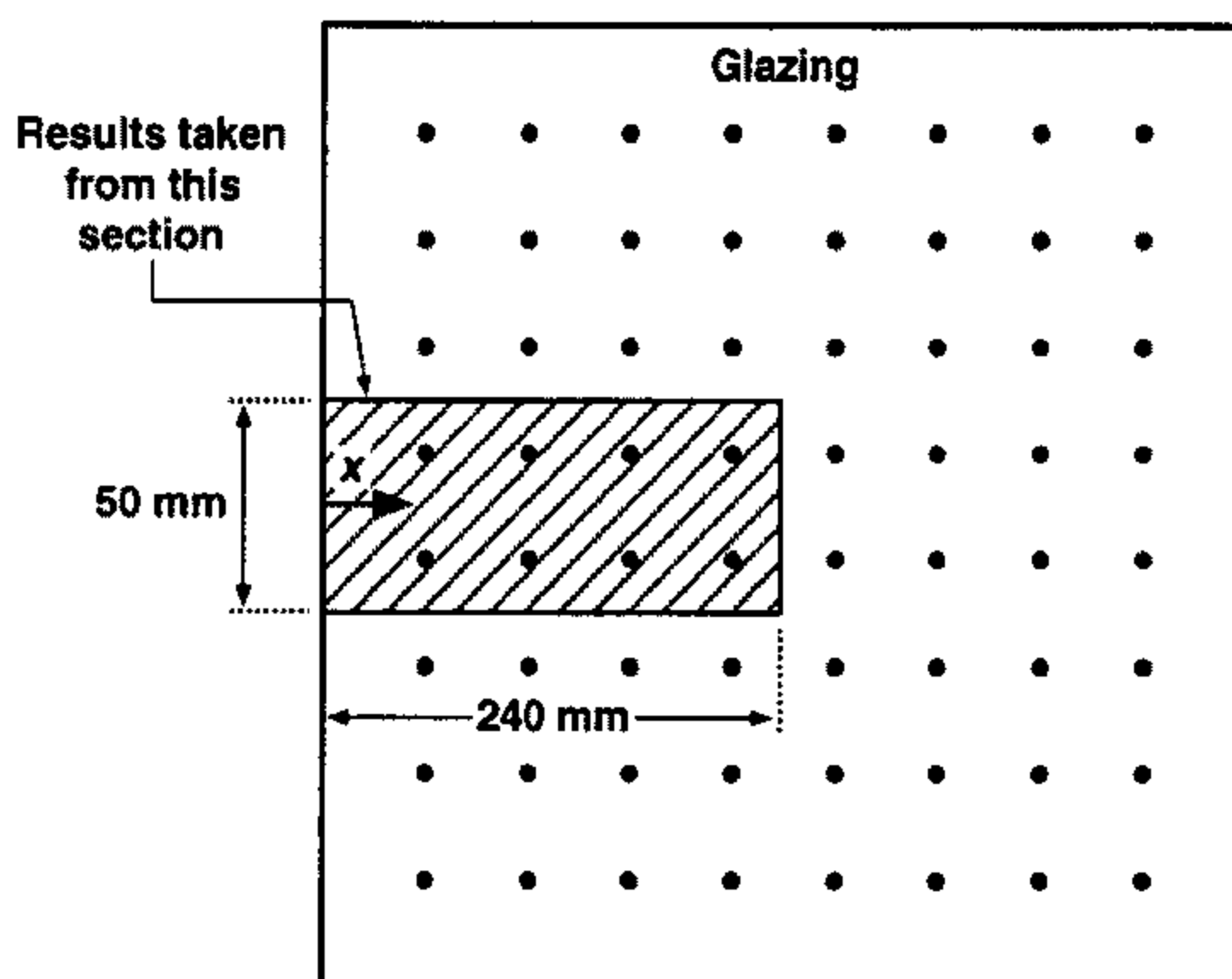


Fig. 5.11: For the two main IR thermographic tests, the temperature of the warm side external surface of the glazing was measured over about 240 mm from the edge of the sample across a 50 mm wide section near the mid-jamb centerline of the glazing. The borders of this section are half-way between pillar rows, and the section encloses the equivalent of two rows of unit cells. In order to reduce the noise in the data, the temperatures across the 50 mm wide section were averaged; each averaged result at position x is the average of the temperatures across the width of the section at this location. The pillars are represented as black dots. This figure shows fewer pillar rows than in the actual glazing sample (which has a pillar separation of 25 mm).

thus the thermal conductance of the edge insulation was $2.9 \text{ Wm}^{-2}\text{K}^{-1}$). The glazing was affixed with silicone sealant to an 18 mm thick mask wall composed of the same type of foam insulation. The temperature of the glazing was measured over about 240 mm from the edge of the sample across a 50 mm wide section near the mid-jamb centerline of the glazing - as shown in Fig. 5.11 (on the previous page). The borders of this section are half-way between pillar rows (which are separated by 25 mm), and thus the section encloses the equivalent of two rows of unit cells. In order to reduce the noise in the data, the temperatures across this 50 mm wide section were averaged. The averaged results are equivalent to the averages of several linescans from the edge that were taken at different positions across the width of the section. Only these averaged results are presented for the main tests. The absolute accuracy of the temperature measurement is $\pm 0.5 \text{ }^{\circ}\text{C}$, and the spatial resolution is $\pm 3.4 \text{ mm}$. The air temperatures, air velocities and heat transfer coefficients for the two main thermographic tests are presented in Tables 5.1 and 5.2. For the conditions of both tests, the glass-to-glass center-of-glass radiative U-value for the glazing, as calculated in the manner described above, is $U_{rad}=0.69 \pm 0.01 \text{ Wm}^{-2}\text{K}^{-1}$. Together with $U_{array}=0.54 \text{ Wm}^{-2}\text{K}^{-1}$, this gives a total glass-to-glass center-of-glass U-value of $U_{g-g}=1.23 \pm 0.02 \text{ Wm}^{-2}\text{K}^{-1}$ for these tests. Additional tests were made after these main tests in order to obtain presentable, yet less accurate, full- and partial-surface infrared thermographic images (thermograms). These additional tests were performed for the same sample orientations and for approximately the same conditions as for the main tests.

Four earlier tests on the vacuum glazing sample were performed with the glazing mounted in a fixed wood casement frame with and without a third pane of glass separated from the glazing by an air cavity. Both flush and recessed edge configurations were considered. The details of these configurations and the additional pane are given in the next section. These tests were done before the above described method for quantifying the heat transfer coefficients had been developed. Therefore, the corresponding heat transfer coefficients are not known, but they are thought to be approximately the same as those given in Tables 5.1 and 5.2. The air temperatures were the same as those listed in these tables. The results of these earlier tests are from single linescans across the glazing - i.e. unlike the data for the main tests, these results do not represent the averages of temperatures across a 50 mm wide section of the glazing.

	Warm side	Cold side
Air temperature (°C)	21.1 ± 0.3	-17.78 ± 0.15
Air velocity (ms ⁻¹)	4.6 ± 0.2	< 0.1 ± 0.1
Heat transfer coefficient (Wm ⁻² K ⁻¹)	31 +2,-4	9.3 ± 0.3

Table 5.1: Environmental conditions for the main vacuum glazing test with the edge insulation facing the cold chamber.

	Warm side	Cold side
Air temperature (°C)	21.1 ± 0.3	-17.78 ± 0.15
Air velocity (ms ⁻¹)	4.3 ± 0.2	< 0.1 ± 0.1
Heat transfer coefficient (Wm ⁻² K ⁻¹)	30 +2,-4	9.3 ± 0.3

Table 5.2: Environmental conditions for the main vacuum glazing test with the edge insulation facing the warm chamber.

5.4 Results

5.4.1 Temperature Profiles

Figure 5.12 presents vacuum glazing external glass surface temperatures obtained from the finite difference solution to the analytic model, and from the finite element model constructed with STRAND6. The location of the FEM results corresponds to a centerline of a 1 m by 1 m glazing. These results are for the environmental conditions given in Table 5.1: temperatures of 21.1 and -17.78 °C, and corresponding heat transfer coefficients of 9.3 and 31 Wm⁻²K⁻¹. The glass-to-glass center-of-glass U-value is 1.23 Wm⁻²K⁻¹. No edge insulation was simulated. It is recalled that the finite element model simulates an edge seal of finite width and thickness, whereas the finite difference model does not simulate such a seal. Therefore, for an appropriate comparison between the two sets of results, the FDM data have been shifted to the right by the width of the edge seal in the FEM: 6 mm. There is a close agreement between the two sets of modelling results, except near the edge. The divergence between the results at this location is due to the fact that, unlike the FDM, the FEM simulates direct air-to-air heat transfer across the width and through the thickness of an edge seal. This leads to temperatures at and near the edge of the FEM which are slightly different from those obtained with the FDM.

For a further comparison between the models, an insulated edge was considered. Figure 5.13 presents finite difference and finite element modelling results obtained with the same conditions as for Fig. 5.12, but with edge insulation which extends from the edge and over the cold side glass surface of each model by a distance equal to the width of the edge seal considered in the FEM: 6 mm. The thermal conductance of the insulation is 2.9 Wm⁻²K⁻¹, and thus the heat transfer coefficient over the insulated surface, h_2 as defined above, is 2.6 Wm⁻²K⁻¹. The conditions and edge insulation configuration considered for this figure are the same as those for one of the main infrared thermographic tests. Once again there is a close agreement between the two sets of modelling results, except near the simulated edge seal. The FDM results in this figure have also been shifted by 6 mm to the right for an appropriate comparison with the FEM results.

For the FEM results of Figs. 5.12 and 5.13, the minimum temperature differences across the edge seal are, respectively, 3.4 and 1.9 °C - or 8.7 and 5.1 % of the total air-to-air temperature difference across the glazing (38.9 °C). Clearly the temperature boundary condition for the analytic model and the FDM (the continuity in temperature across the simulated edge seal) is more appropriate with insulated edges.

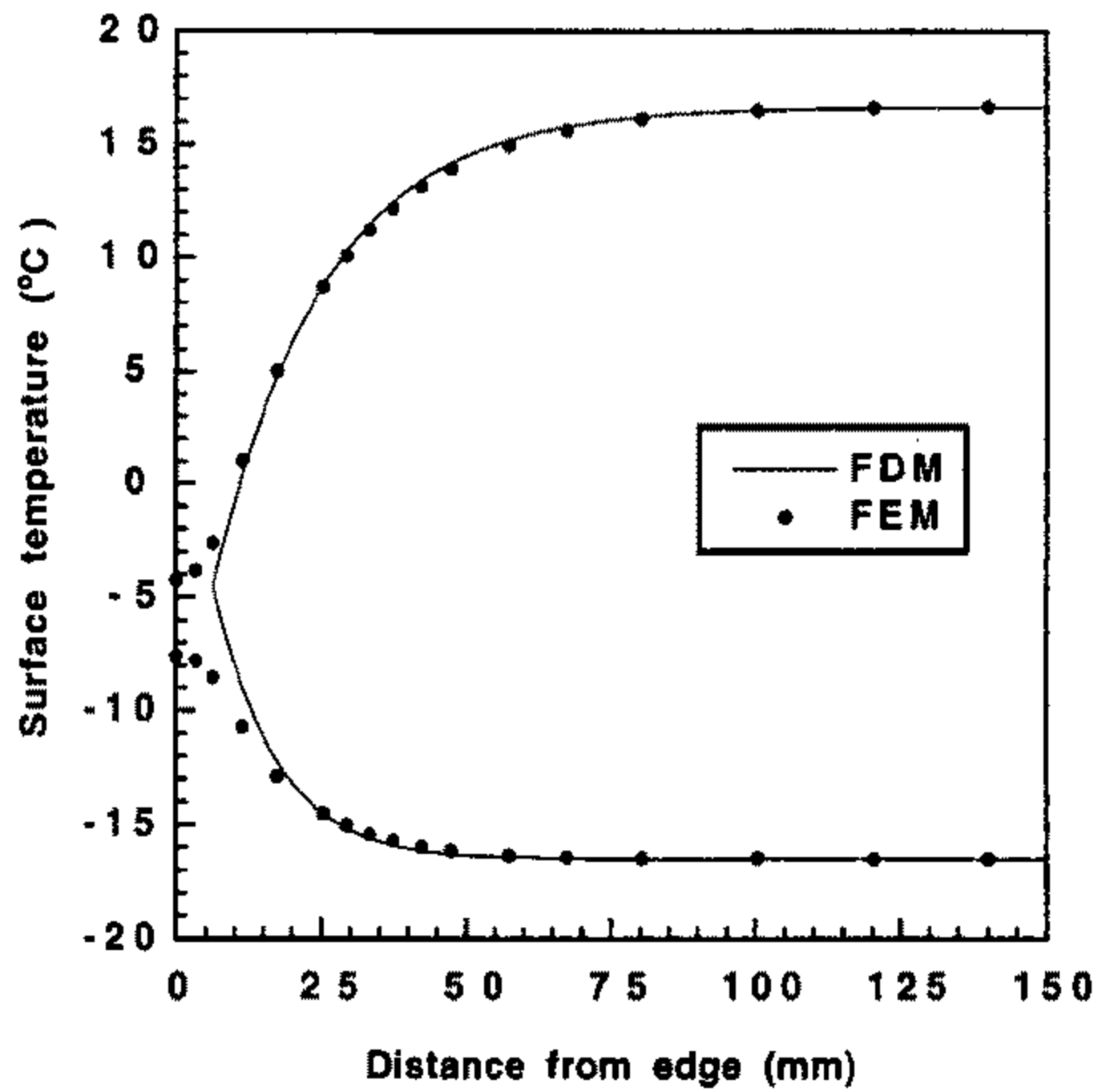


Fig. 5.12: Results from the finite difference model (FDM) and the finite element model (FEM) for the surface temperatures of a vacuum glazing exposed to air temperatures of 21.1 and -17.78 °C and corresponding heat transfer coefficients of 9.3 and 31 Wm⁻²K⁻¹. The value for U_{g-g} is 1.23 Wm⁻²K⁻¹. There is no edge insulation. Unlike the FEM, the FDM does not simulate an edge seal of finite dimensions. Therefore, for an appropriate comparison of the results, the FDM data have been shifted to the right by the width of the edge seal in the FEM: 6 mm. The FEM results are along the centerline of a 1 m by 1 m glazing.

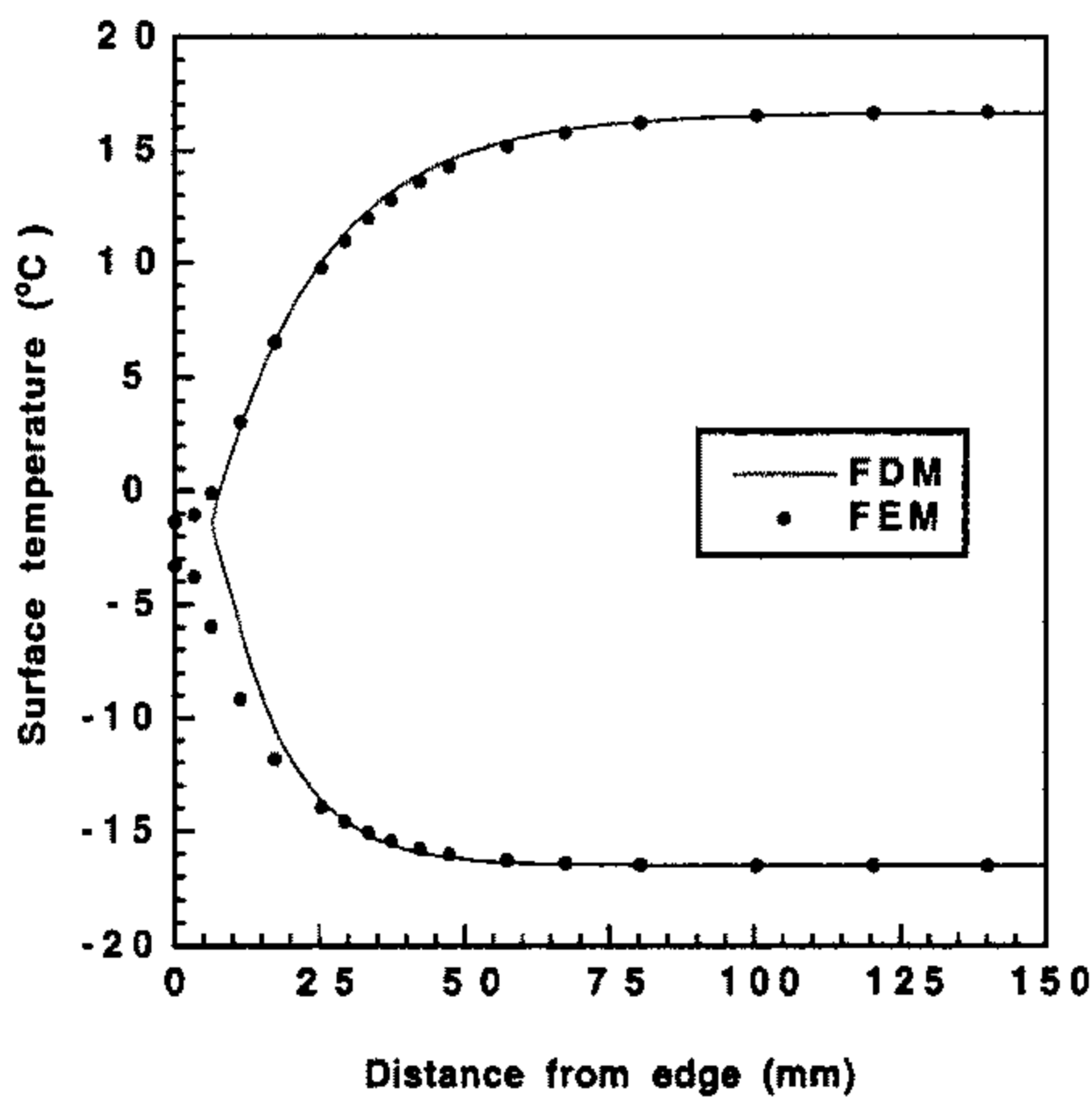


Fig. 5.13: Results from the finite difference model (FDM) and the finite element model (FEM) for the same conditions and glass-to-glass center-of-glass U-value considered for Fig. 5.12, but with edge insulation which extends from the edge and over the cold side glass surface of each model by a distance equal to the width of the edge seal considered in the FEM: 6 mm. The thermal conductance of the simulated edge insulation is 2.9 Wm⁻²K⁻¹.

This is because, as shown below, edge insulation decreases the edge conduction thereby reducing the temperature difference across the constant thermal resistance of the edge seal.

The close agreement between the results of the one-dimensional finite difference model and the results obtained from the more exact three-dimensional finite element model theoretically validates the one-dimensional solution, and the analytic model on which it is based. Since the finite difference and finite element models give equivalent results, except close to the simulated edge seal, either approach can be used to simulate the effects of edge conduction. However, for an *exact* calculation of the glass temperatures and heat flows close the seal, it is preferable to use the FEM. The finite difference model is easier to use than the finite element model for two reasons. Firstly, changes to the FDM can be easily made by a straightforward rewriting of the computer code, whereas most changes to the FEM must be made in a more time consuming graphics based package. Secondly, the FDM, by virtue of its one-dimensional nature, requires fewer nodes, and thus much less solution time, than the FEM. The FDM was used for many of the calculations below because of this greater ease of use and because, for these calculations, this model provides sufficiently accurate results near the simulated edge seal.

The one-dimensional analytic model and the corresponding FDM are based on the assumption that the temperature is uniform through the thickness of each glass sheet. This is of course a reasonable approximation of the actual situation because the thermal resistance through each sheet is much less than the other thermal resistances of the system. Results from the three-dimensional FEM justify this approximation. For the conditions considered in Figs. 5.12 and 5.13, the FEM results show that the temperature difference across the thickness of each sheet of glass varies from about 1 °C over the seal to 0.2 °C near the center of the glazing (the later result can also be easily predicted analytically); over most of each glass sheet, the temperature difference is less than 0.3 °C - which is a negligible 0.8 % of the total air-to-air temperature difference of 38.9 °C. Even for a larger glass-to-glass center-of-glass U-value of 4.4 Wm⁻²K⁻¹, this temperature difference is negligibly small (less than 0.8 °C over most of the glazing).

The following two thermograms of the vacuum glazing sample correspond to the same edge insulation configurations and to approximately the same environmental conditions given above for the two main tests on the sample. These thermograms are not quantitatively very accurate, and they were not used to generate the data sets

given below for the two main tests. The images are only presented in order to demonstrate several qualitative features about the IR thermographic measurements. For both thermograms, the top of the vertically oriented glazing is towards the top of the image.

Figure 5.14 is a thermogram of the entire warm side external surface of the vacuum glazing sample with the edge insulation facing the cold chamber (and thus the insulation is on the opposite side of the glazing to which these results correspond). This thermogram is actually a composite of two close-up images. The temperature scale is shown below the thermogram. The edge of the glazing is located at the edge of this image. It can be seen that most of the temperature variation due to the edge effect is limited to a narrow band around the edge of the glazing. A regular pattern of small cool spots is visible across the glazing. These spots are due to the temperature suppressions caused by the pillar conduction, and they are centered around the individual pillars. The slightly warmer temperatures at the middle of the glazing are associated with the outwardly displaced pillars around this location (which is used for measuring U_{rad}). Due to the greater pillar separation in this central region, the local U-value is lower. This lower U-value results in less heat transfer and thus a higher warm side surface temperature at this location.

The slight asymmetry in the extents of the edge effect along the top and bottom edges is probably related to the downward convective flow across the glazing. Aside from this small effect, the variation in the temperatures along all of the edges, and the small temperature suppressions around the pillars, the temperature across the glazing is fairly uniform. This indicates that there are no significant vertical temperature gradients due to convection. Therefore, for the conditions of the thermographic tests (i.e. the conditions in Tables 5.1 and 5.2), and presumably for the similar ASTM standard test conditions, it is appropriate to ignore localized convective effects when simulating the external heat transfer for vacuum glazings - as is done with the finite difference solution to the analytic model and the finite element model presented above, and with the modified finite element model presented in Section 4.3.

Figure 5.15 is a close-up thermogram of most of the left side of the glazing sample with the insulation facing the warm chamber. The edge of the glazing is at about the right edge of the left most light blue band, and the region to the left of this edge is the insulating mask wall (i.e. the test frame, which was not shown in Fig. 5.14). Several features seen in the previous thermogram can be more clearly seen in this image: the edge effect, the warmer temperatures at the center of the glazing (near the middle of

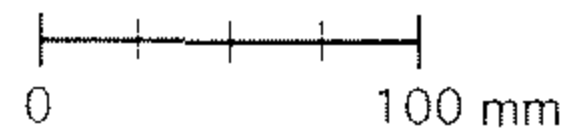
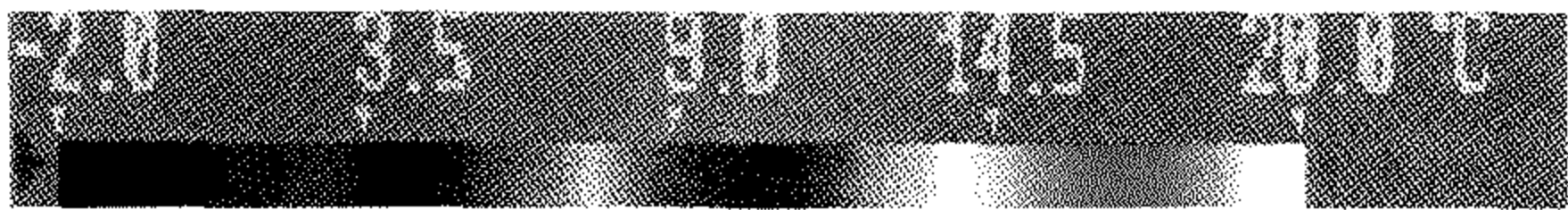
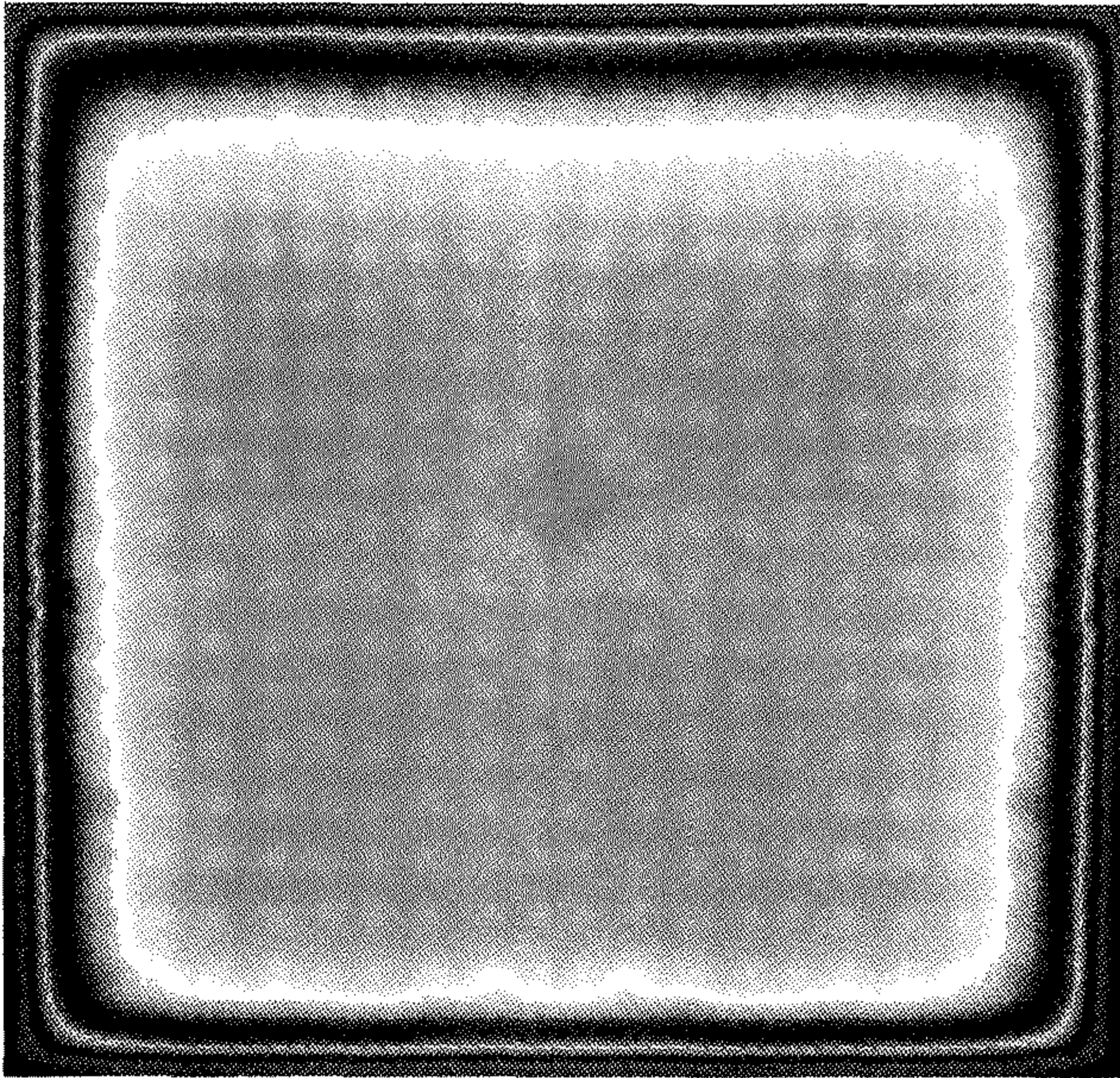
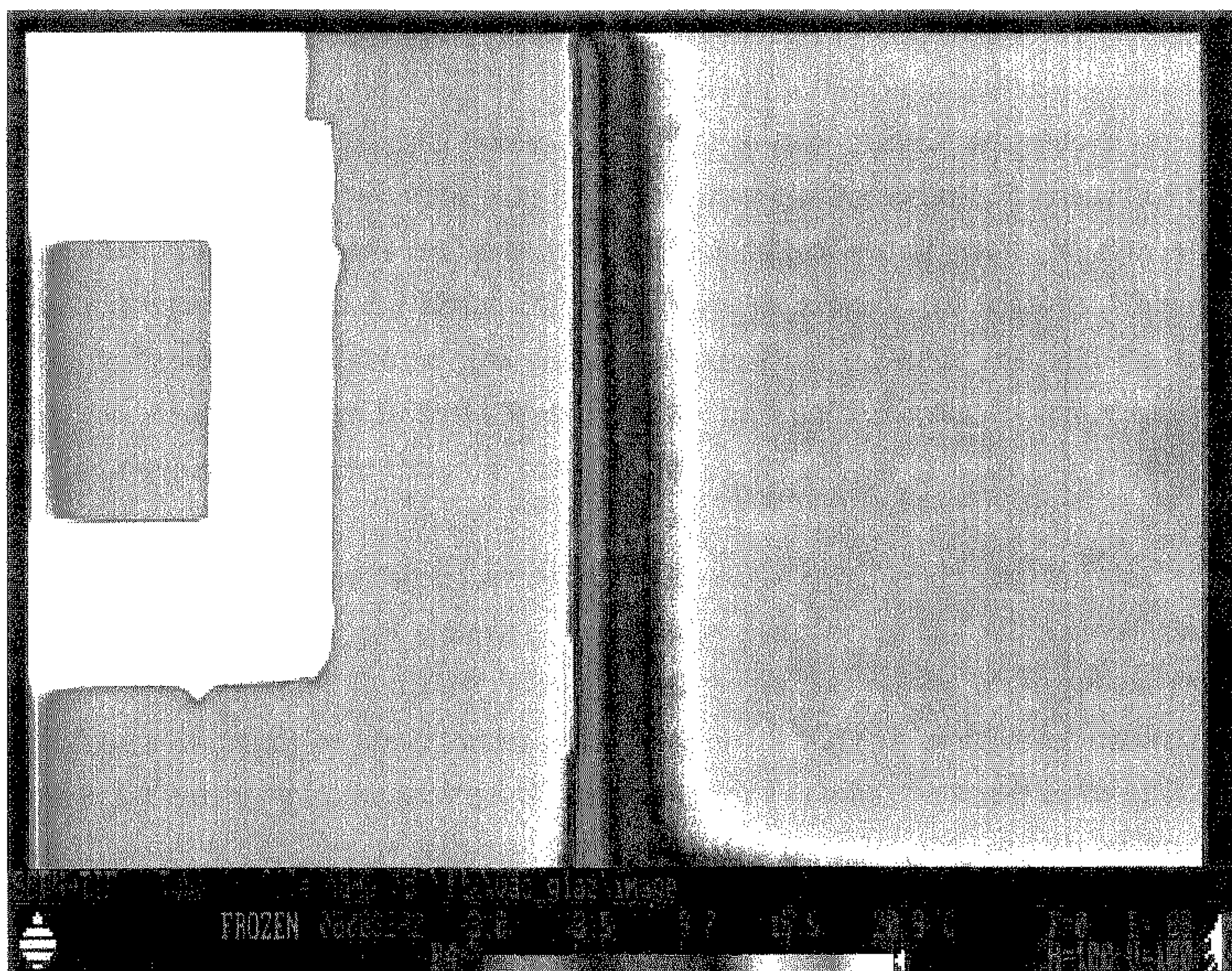


Fig. 5.14: Infrared thermogram of the entire warm side surface of the vacuum glazing for conditions which are approximately the same as those given in Tables 5.1 and 5.2, and for insulation covering the cold side edges over the width of the edge seal. The colour temperature scale is immediately below the image. The distance scale is approximate.



0 100 mm

Fig. 5.15: Infrared thermogram of a close-up view of most of the left side of the warm side surface of the vacuum glazing for conditions which are approximately the same as those given in Tables 5.1 and 5.2, and for insulation covering the warm side edges over the width of the edge seal. The edge of the glazing is at about the right edge of the left most blue band. The square to the left of the glazing is the external reference emitter. The colour temperature scale (which is different from the one of Fig. 5.14) is immediately below the image. The distance scale is approximate.

the right edge of this image), and the regular pattern of cool spots due to the temperature suppressions around the individual pillars. The separation between the centers of the cool spots, which is equal to the pillar separation of 25 mm, provides a useful scale with which to estimate the full extent of the edge effect: between about 75 and 100 mm from the edge of the sample.

The infrared thermographic results of the next two figures correspond to the previously mentioned two main tests on the vacuum glazing sample. It is recalled that the data from these tests are averages of the temperatures across a 50 mm wide section which extends for about 240 mm from the edge of the glazing. Only the results between the sightline (the coincident extents of the edge seal and the edge insulation) and 140 mm from the sightline (by which point the surface temperature is much more uniform) are shown in these figures.

Figure 5.16 presents infrared thermographic results for the main test with the edge insulation facing the cold chamber, and for the environmental conditions given in Table 5.1. The error limits for these IR thermographic results are obtained from shifting each result by the extremes of both the ± 0.5 °C error in the temperature and the ± 3.4 mm error in the location of the result. For example, the upper error limit for an IR result is obtained by shifting the result to the left by 3.4 mm, and up by 0.5 °C. The reverse is done for the corresponding lower error limit. It is recognized that this only approximates the exact profiles of the actual limits due to the fact that the error ranges of nearby results overlap. Also shown in this figure are finite element modelling results for the same environmental conditions as for the IR test, and for a glass-to-glass center-of-glass U-value of $U_{g-g}=1.23 \pm 0.02$ Wm⁻²K⁻¹; as explained above, this is the U-value for the sample which corresponds to the test conditions. The same edge insulation configuration as for the IR test is simulated with the model. The thermal conductance of the edge insulation is 2.9 Wm⁻²K⁻¹ and thus, following the above definition for the conductance over the modelled insulated surface, $h_2=2.6$ Wm⁻²K⁻¹. The error limits for the FEM results correspond to the modelled temperatures for the appropriate upper and lower limits of the environmental conditions and U_{g-g} that maximize and minimize the surface temperature. For example, the upper limits were determined with air temperatures of 21.4 and -17.63 °C, and with corresponding heat transfer coefficients of 9.6 and 27 Wm⁻²K⁻¹; also, for this calculation, $U_{g-g}=1.21$ Wm⁻²K⁻¹ and h_2 remains at 2.6 Wm⁻²K⁻¹. The error in the modelling results decreases with distance from the sightline because the changes to the environmental conditions and U_{g-g} (and thus to the heat flow across the glazing) have less of an absolute effect on the smaller glass-to-air temperature

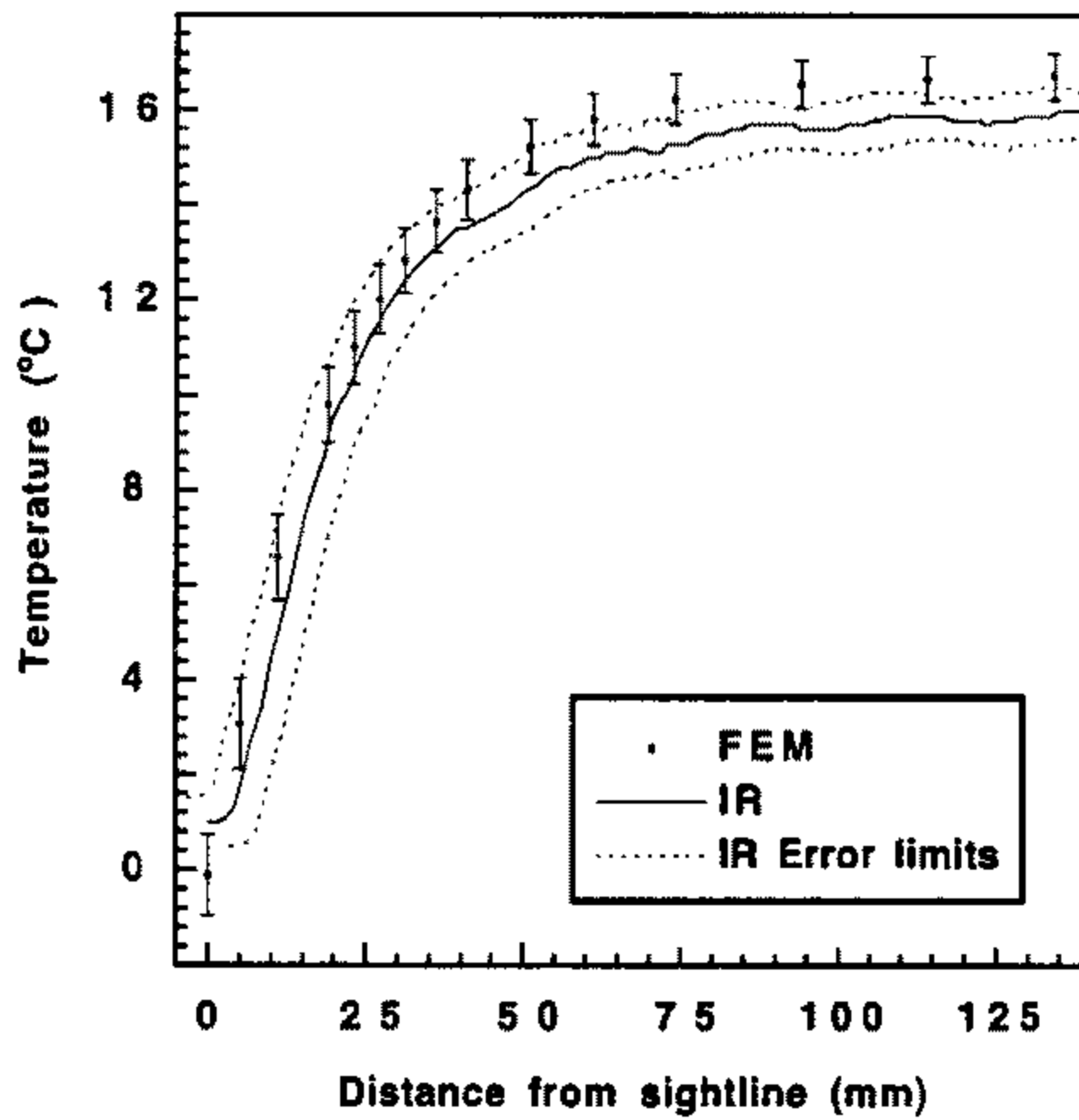


Fig. 5.16: Infrared thermographic results (IR) for the main thermographic test with the insulation covering the cold side edges over the width of the edge seal. The conditions for this test are given in Table 5.1. Also shown are finite element modelling results (FEM) for the same insulation configuration and conditions, and for the glass-to-glass center-of-glass U-value which corresponds to these conditions: $1.23 \text{ Wm}^{-2}\text{K}^{-1}$. The thermal conductance of the edge insulation is $2.9 \text{ Wm}^{-2}\text{K}^{-1}$.

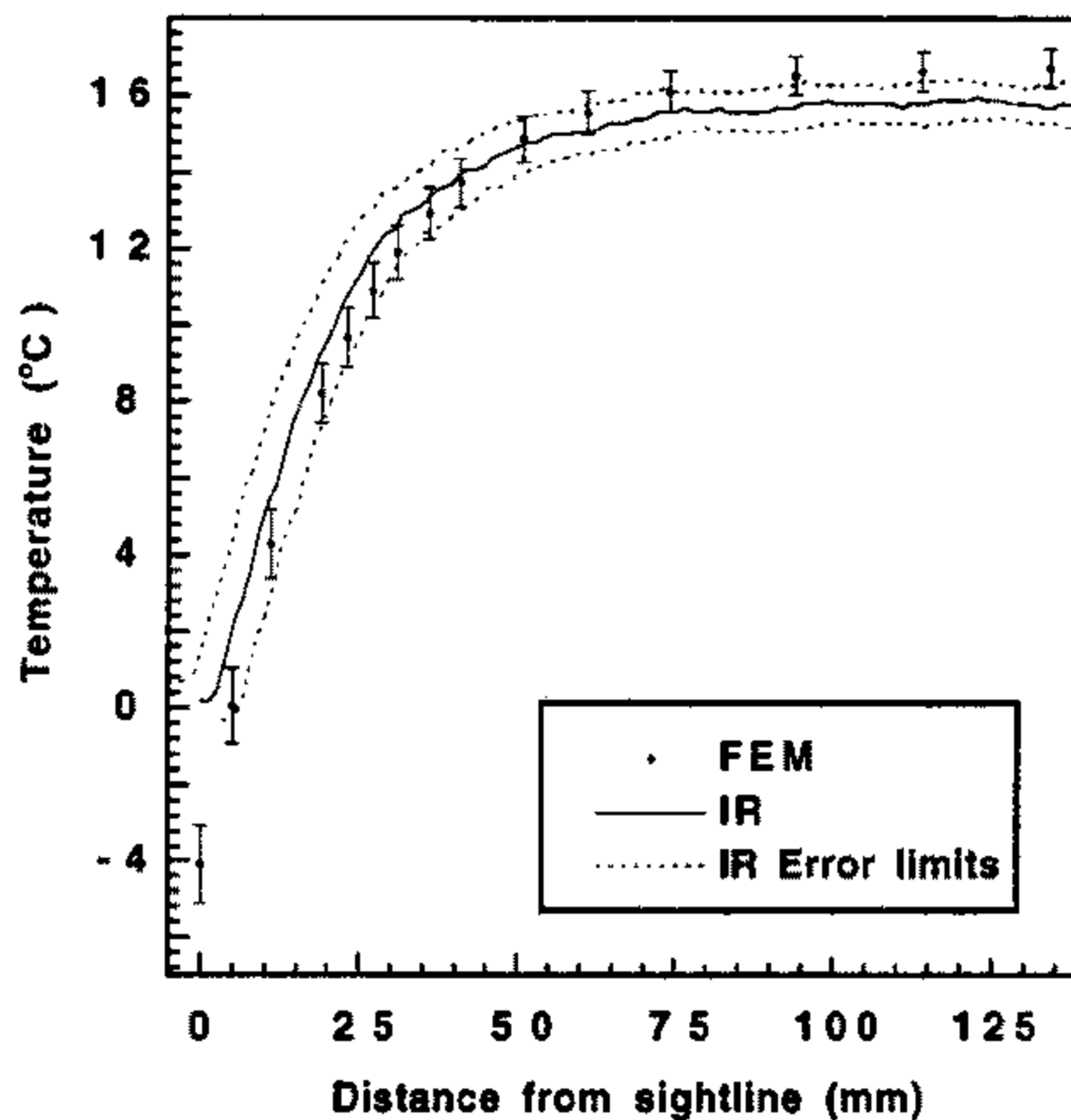


Fig. 5.17: Infrared thermographic results (IR) for the main thermographic test with the insulation covering the warm side edges over the width of the edge seal. The conditions for this test are given in Table 5.2. Also shown are finite element modelling results (FEM) for the same insulation configuration and conditions, and for the glass-to-glass center-of-glass U-value which corresponds to these conditions: $1.23 \text{ Wm}^{-2}\text{K}^{-1}$. The thermal conductance of the edge insulation is $2.9 \text{ Wm}^{-2}\text{K}^{-1}$.

differences away from the sightline.

Figure 5.17 presents infrared thermographic results for the main test with the edge insulation facing the warm chamber and for the environmental conditions given in Table 5.2. Also shown are finite element modelling results for the same conditions and edge insulation configuration, and for $U_{g,g}=1.23 \pm 0.02 \text{ Wm}^{-2}\text{K}^{-1}$. The error limits for both sets of results were determined as described above for Fig. 5.16. For this test, the temperature at the sightline is lower than that for the other test (with the edge insulation on the cold side of the glazing) as definitely indicated by the modelling results. This is because the insulation has less of an effect on the air-to-air heat transfer across the seal, and thus on raising the temperatures at and near the edge, when it is combined in series with the smaller of the two ambient heat transfer coefficients: the coefficient on the warm side of the sample.

In both Figs. 5.16 and 5.17, there is an agreement between the infrared thermographic measurements and the FEM results to within the respective errors. Therefore, the IR thermographic results experimentally validate the finite element model. It is noted that the finite element and finite difference modelling results of Fig. 5.13 correspond to the same environmental conditions and edge insulation configuration to which the IR results of Fig. 5.16 correspond. In fact, the same FEM results beyond the sightline are presented in both figures (for the sake of clarity, the FDM results were not included in the latter figure). Since there is a good agreement between the FEM and FDM results of Fig. 5.13, and since the IR and FEM results of Fig. 5.16 are in agreement for the same situation, the infrared thermographic results also validate the finite difference model and the analytic model on which it is based.

In both Figures 5.16 and 5.17, there is a small periodic variation in the infrared thermographic results beyond about 75 mm from the sightline. This variation is associated with the temperature suppressions due to pillar conduction and it is further discussed below. The variation is not observed closer to the edge of the glazing because it is much less than the temperature variation due to the edge effect. Both of these figures show that the edge effect extends to about 100 mm from the sightline; this is roughly consistent with the results of Fig. 5.15 which are for approximately the same conditions.

There are several ways in which the edge conduction - and thus the influence of the associated edge effect on the surface temperatures and the U-values - can be reduced: by insulating the edges, by recessing the edges into a sash, and by adding a third pane

of glass to the glazing which is separated from the sample by an air cavity. The effects of these methods on the edge conduction were analyzed in a direct manner by calculating the changes in this conduction, and in an indirect manner by considering the changes to the surface temperatures and the U-values. The results of this study are presented below.

5.4.2 Reduction of Edge Conduction with Edge Insulation

5.4.2.1 Effects of Edge Insulation on Edge Temperatures and Conduction

Figures 5.18a to f present finite element modelling results for the warm and cold side external glass surface temperatures as functions of the distance from the edge, and along a centerline of the glazing, for different insulation recess depths on both sides of the glazing. The results are for the ASTM standard test conditions, and a glass-to-glass center-of-glass U-value of $1.10 \text{ Wm}^{-2}\text{K}^{-1}$. This is a typical U-value for a vacuum glazing with two sheets of K glass, a pillar radius of 0.125 mm and a pillar separation of 25 mm, and which is exposed to the ASTM standard test conditions. A typical edge seal width of 8 mm is simulated. The insulation extends from the edge of the glazing by the same distance over both the warm and cold sides of the glazing. The insulation recess depth, w , is measured from the inner edge of the seal - as shown in Fig. 5.1. Therefore, the zero recess depth just covers the seal, and the recess depth for no insulation is the negative of the edge seal width: -8 mm. For each of Figs. 5.18, the sightline (i.e. the extent of the insulation) is indicated by the vertical dashed lines. The thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$.

The results of Figs. 5.18 demonstrate two important effects that the addition of edge insulation has on the glass temperatures associated with the edge effect. Firstly, the warm side glass temperature at the sightline increases with the recess depth. This is clearly shown by Fig. 5.19, which presents the sightline temperatures from Figs. 5.18, and also from modelling results for other recess depths (25 and 35 mm) that are not considered in Figs. 5.18. Under usual operating conditions, condensation in the view area of the vacuum glazing is most likely to occur at the external glass surface near the warm side sightline. Since the formation of such condensation depends on the temperature at this location and since this temperature is increased by the addition of edge insulation, the condensation resistance of the glazing increases with the insulation recess depth. However, as demonstrated by the results of Fig. 5.19, beyond a recess depth of a few millimeters, a smaller increase in the sightline temperature is achieved with each additional fixed increment of insulation. Therefore, there is a

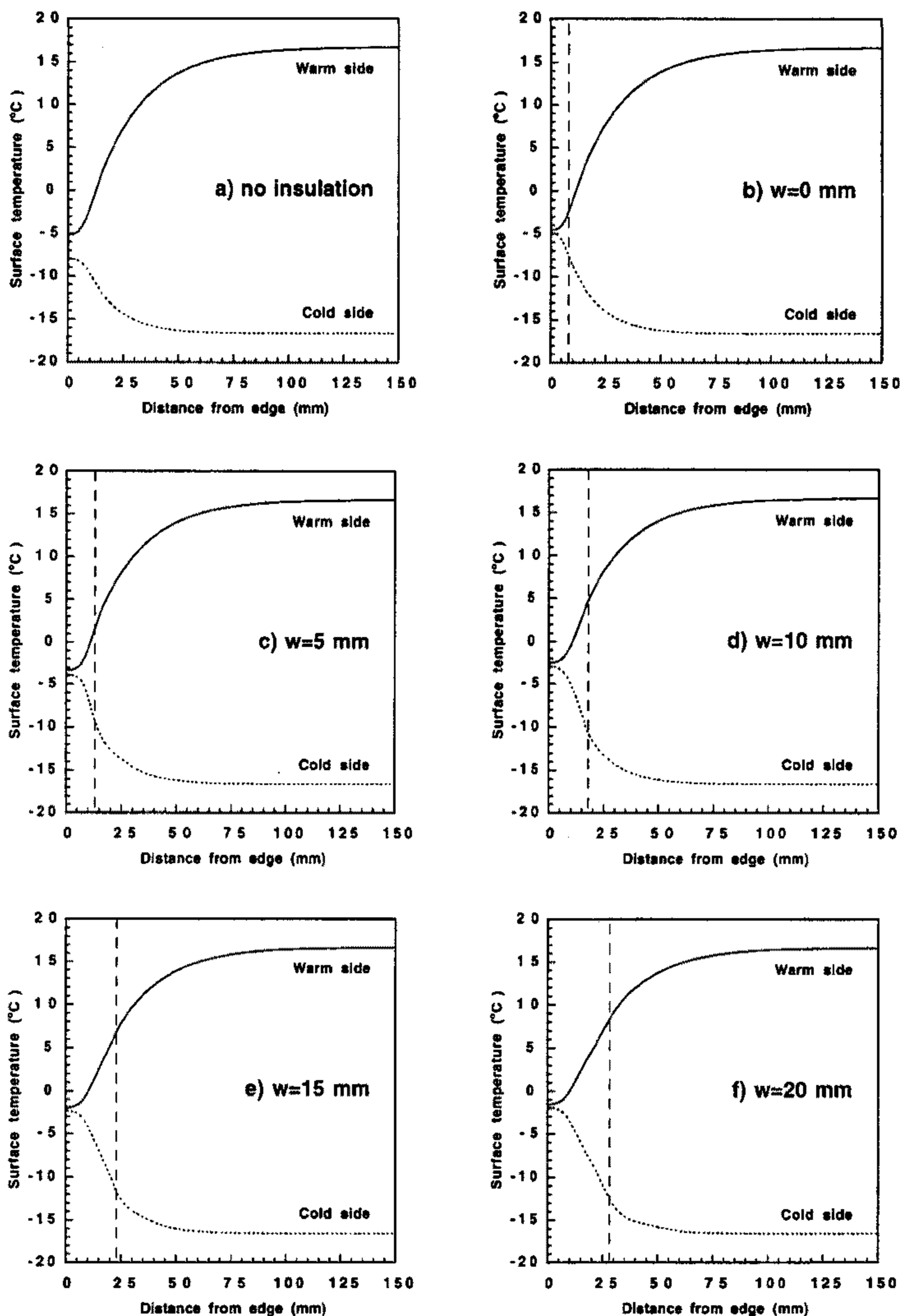


Fig. 5.18: Finite element modelling results for the external glass surface temperatures as functions of the distance from the edge, and along a centerline of a 1 m by 1 m glazing, for different insulation recess depths. The results are for the ASTM standard test conditions: air temperatures of 21.1 and -17.8 °C, and corresponding heat transfer coefficients of 8.3 and 30 Wm⁻²K⁻¹. The simulated U_{g-g} is 1.10 Wm⁻²K⁻¹. The insulation recess depth, w , is the same on both sides of the glazing, and it is measured from the inner edge of the 8 mm wide edge seal. The sightline (i.e. the extent of the insulation) is indicated by a vertical line of long dashes in (b) to (f). The thermal conductance of the insulation is 1.5 Wm⁻²K⁻¹.

diminishing point of return in the effect of edge insulation on the sightline temperature.

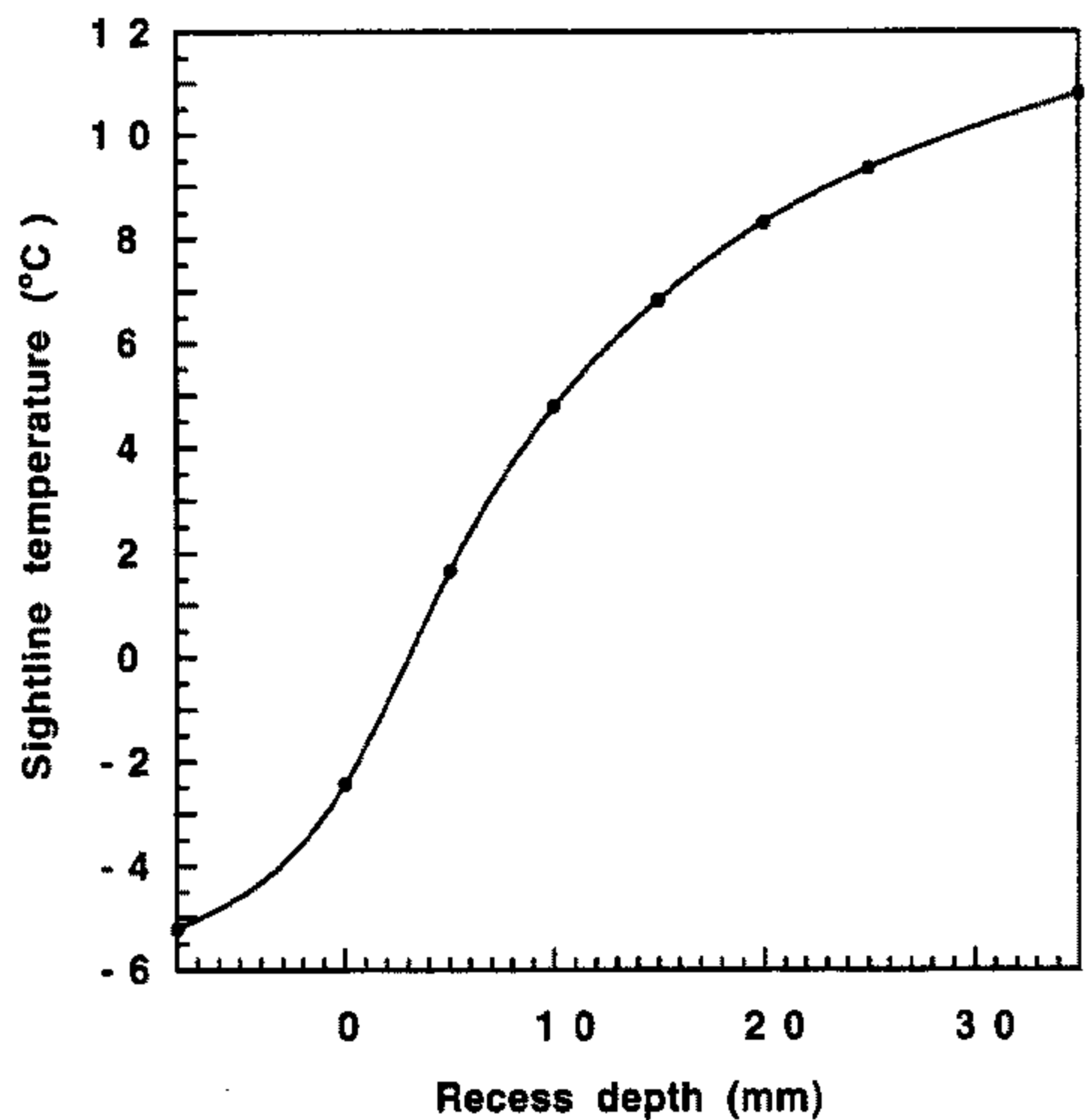


Fig. 5.19: Finite element modelling results for the sightline temperature at the warm side external glass surface as a function of the recess depth for the same conditions and edge insulation configuration considered in Figs. 5.18. Except for the results for $w=25$ and 35 mm, all of these temperatures are from Figs. 5.18. All of the data correspond to an edge seal of width 8 mm.

The second important effect of edge insulation on the glass temperatures is that it raises the temperature of the glass at the edge relative to the mean temperature of the entire glass structure of the glazing. As explained in Chapter 6, this reduces the temperature induced tensile stresses in the edge region of each glass sheet. This in turn has the effect of reducing the probability of structural failure for the glazing - as discussed in Chapter 8.

The addition of edge insulation also has an effect on the edge conduction. This effect is most easily studied with the finite difference solution to the analytic model. The conduction through the edge seal per unit length of edge can be calculated from the results of the FDM on the basis of the temperature gradient between the edge node and the first node from the edge for either simulated glass sheet (the boundary conditions ensure that the magnitude of the temperature gradient at this location is the

same on both sides of the glazing). The heat flow is calculated by multiplying this temperature gradient by the glass thermal conductivity ($1.0 \text{ Wm}^{-1}\text{K}^{-1}$) and the transverse dimension of the cross-sectional area of the glass sheet: the specified glass thickness (which for this study is 4 mm). This calculation gives the heat flow per unit length of edge along the glass sheet at this location - which is approximately equal to the heat that would flow through an actual edge seal. Figures 5.20a and b present lineal heat flows determined in this manner as functions of the insulation recess depth for the ASTM standard test conditions and for two glass-to-glass center-of-glass U-values: 1.20 and $4.40 \text{ Wm}^{-2}\text{K}^{-1}$. The first U-value is slightly larger than the typical value for a glazing made with two sheets of K glass, and the second U-value is slightly larger than the typical value for a glazing made with two sheets of uncoated glass - with both types of glazings having pillars which are of the typical radius of 0.125 mm and which are separated by the typical distance of 25 mm. The insulation extends from the edge of the model (i.e. the location of the simulated seal) over a recess depth, w , which is measured from this edge and is the same on both sides of the glazing - as shown in Fig 5.4. Three values of insulation thermal conductance, C_{ins} , are simulated: 0, 1.5 and $3 \text{ Wm}^{-2}\text{K}^{-1}$. The zero insulation conductance corresponds to adiabatically insulated glass surfaces. The results of these figures show that modest insulation recess depths can significantly reduce the edge conduction. There are two reasons why the edge conduction is less for the larger value of U_{g-g} . Firstly, for the larger U-value, the ratio of the thermal resistance along the glass sheets to that across the evacuated space is greater. Therefore, a smaller fraction of the heat flows along the glass sheets and through the simulated edge seal than across the evacuated space. Secondly, for this larger U-value, the difference between the warm and cold side center-of-glass temperatures is smaller, resulting in less conduction along the sheets and through the simulated seal.

The edge conduction for a perfectly insulating glazing (i.e. for $U_{g-g}=0$) with adiabatic insulation can be calculated with the analytic relationship given above by equation 5.15. Figure 5.21 presents results obtained from this analytic approach for the edge conduction per unit length of edge as a function of the insulation recess depth. The insulation recess depth is the same on both sides of the glazing, and the results correspond to the ASTM standard test conditions. Also shown are results from the finite difference model for the same conditions: adiabatic insulation, and glass-to-glass center-of-glass U-values of 0, 1.2 and $4.4 \text{ Wm}^{-2}\text{K}^{-1}$. As expected, there is a very close agreement between the exact analytic results and the FDM results for the case of a perfectly insulating glazing; the FDM results are only 1.5 to 0.7 % greater than the corresponding analytic data. This indicates that, for the finite difference model, the

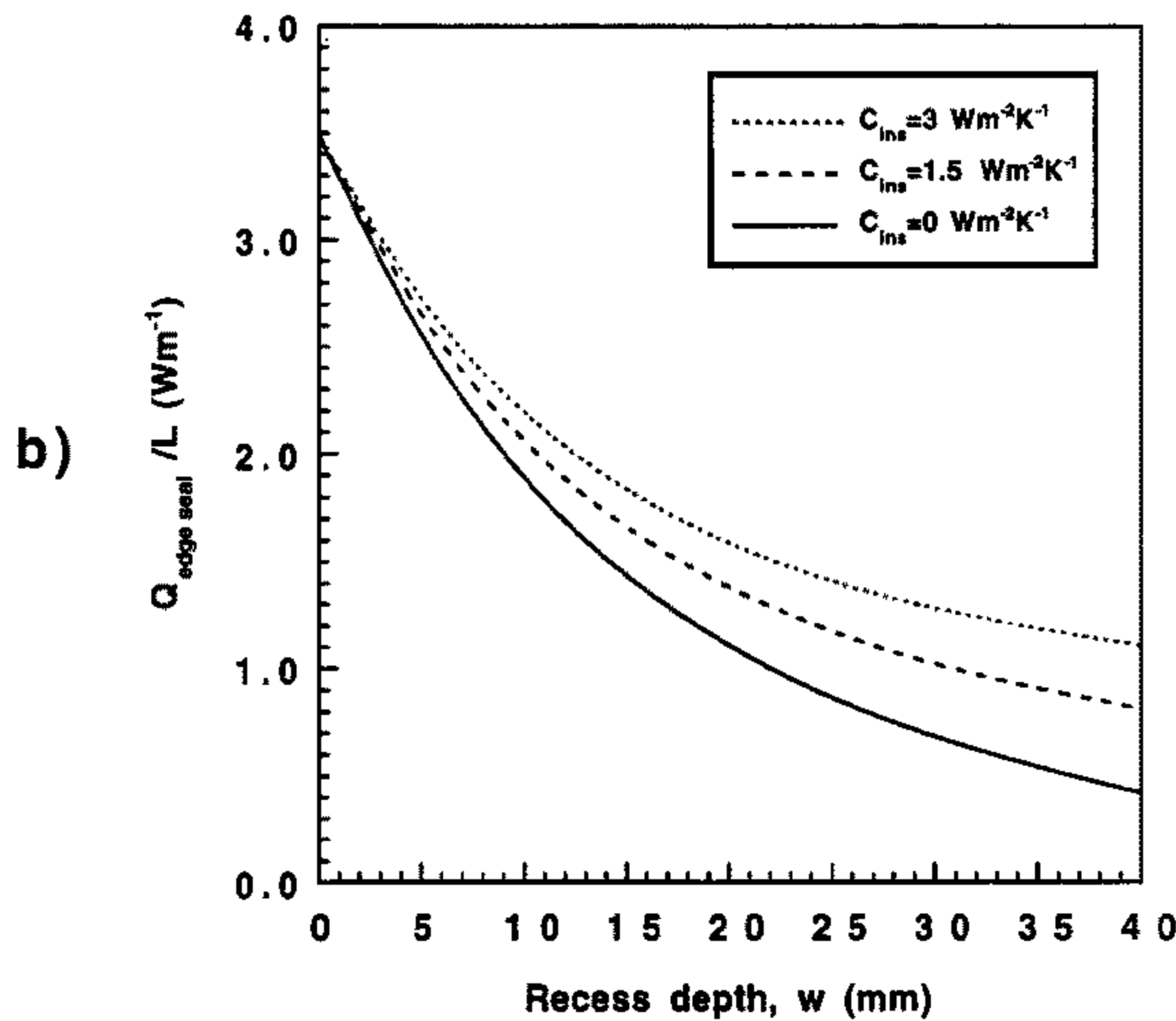
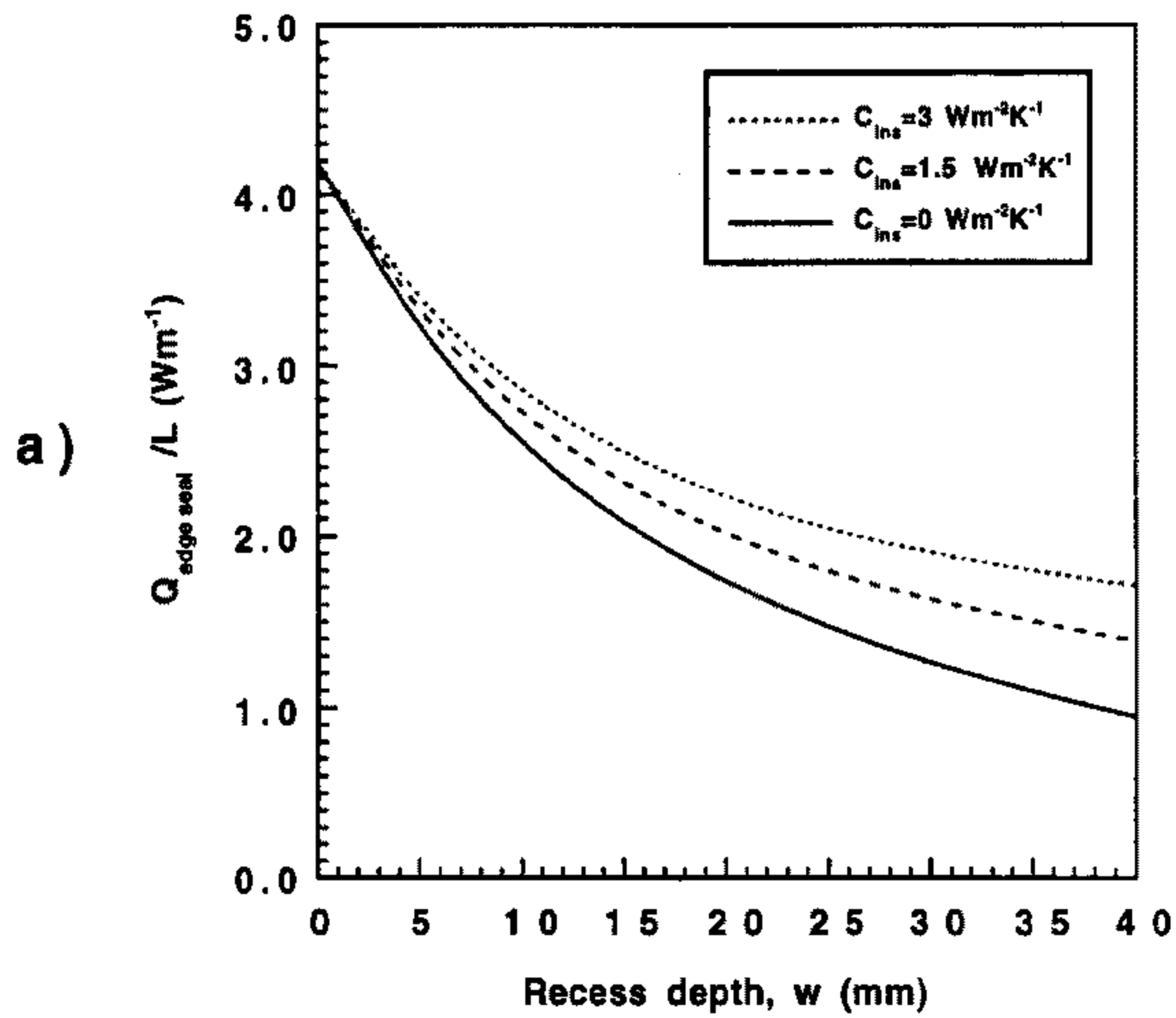


Fig. 5.20: Finite difference modelling results for the edge conduction per unit length of edge as a function of the insulation recess depth for the ASTM standard test conditions and for three values of insulation thermal conductance, C_{ins} . Results are presented for two glass-to-glass center-of-glass U-values (U_{g-g}): (a) $1.2 \text{ Wm}^{-2}\text{K}^{-1}$, and (b) $4.4 \text{ Wm}^{-2}\text{K}^{-1}$. The results are based on a glass thermal conductivity of $1.0 \text{ Wm}^{-1}\text{K}^{-1}$, and a glass sheet thickness of 4 mm. The recess depth is the same on both sides of the glazing.

length of the simulated glass sheets and the spacing of the nodes ensure that there is a good convergence in the results from this model. Thus, the exact analytic heat flows theoretically validate the finite difference model. For the FDM data, the departures from the results for the perfectly insulating case increase with U_{g-g} . The results of Fig. 5.21 show that the trend in the heat flow (i.e. the shape of the plots) is not significantly affected by the glass-to-glass center-of-glass U-value.

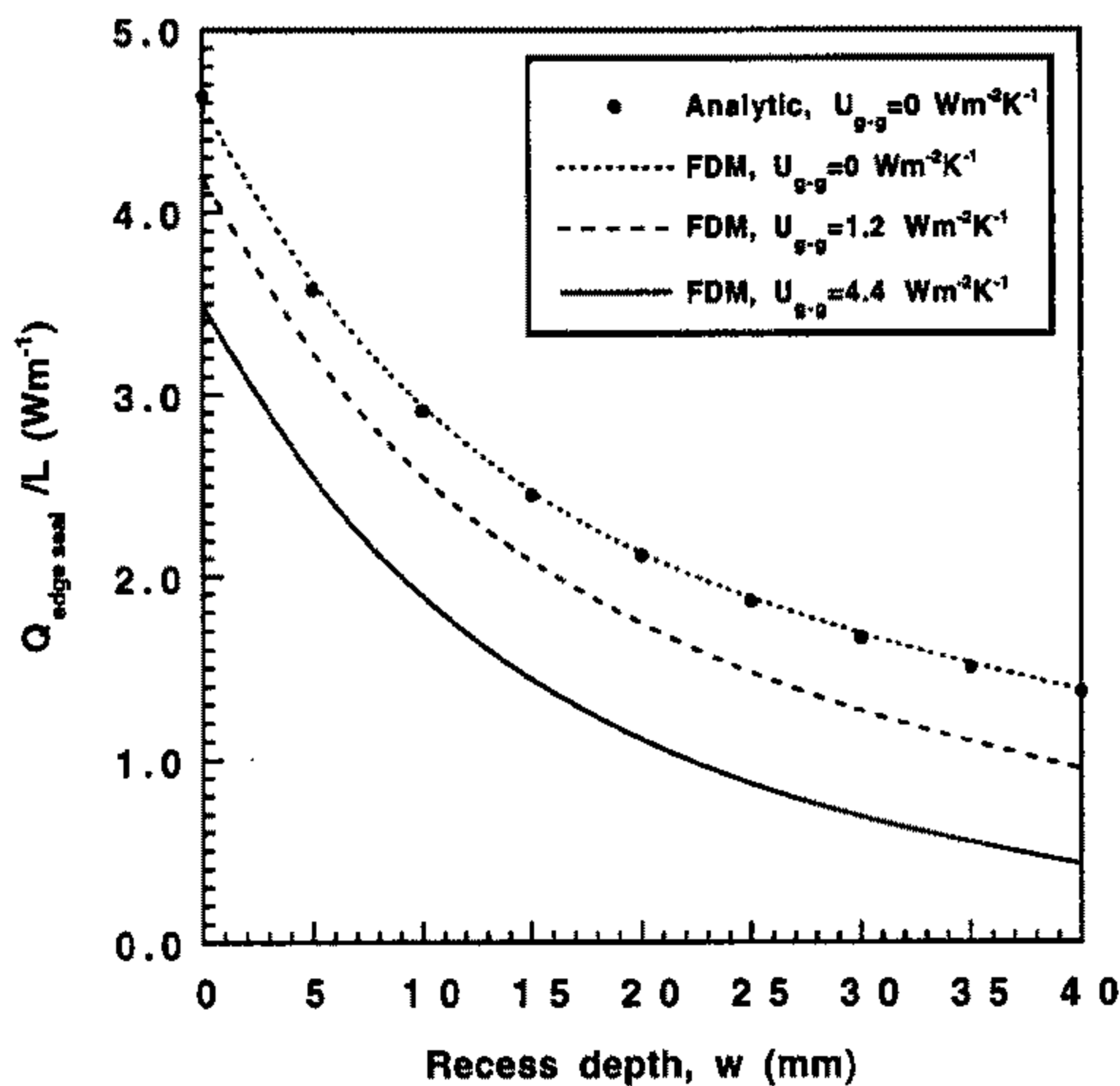


Fig. 5.21: Analytic results (Analytic) for the edge conduction per unit length of edge as a function of the insulation recess depth for a perfectly insulating glazing ($U_{g-g}=0$) which is exposed to the ASTM standard test conditions and which has the same recess depth of adiabatic insulation on both sides of the glazing. These analytic results are calculated by equation 5.15. Also shown are finite difference modelling results (FDM) for the same conditions and edge insulation configuration, but for three glass-to-glass center-of-glass U-values ($U_{g-g}=0, 1.2$ and $4.4 Wm^{-2}K^{-1}$).

5.4.2.2 Effects of Edge Insulation on U-values

It is also of interest to consider the effects of edge insulation on the window U-values. The standard distance of 63.5 mm from the sightline which is used to calculate A_{edge} (and thus U_{edge}) is based on the assumption that the significant portion of the edge effect does not extend beyond, or far beyond, this distance. However, there are two reasons why this is not so for vacuum glazings under standard test conditions. Firstly, the edge conduction flows through a seal of an unconventionally low thermal resistance. Secondly, the typical glass sheet thickness for vacuum glazings ($t=4$ mm)

is greater than the common glass sheet thickness for conventional glazings ($t=3$ mm) to which the standard distance presumably corresponds; since equation 5.13 indicates that the characteristic distance for the edge effect is proportional to $t^{1/2}$, the extent of the significant portion of the edge effect should be greater for vacuum glazings.

Some of the results presented thus far in this chapter provide examples of the extent of the edge effect in vacuum glazing. Figure 5.18a shows that the extent of the edge effect on the warm side of the glazing for the ASTM conditions and for no edge insulation is about 100 mm. This is approximately consistent with the extents of the edge effect for the infrared thermographic results presented in Figs. 5.15, 5.16 and 5.17 which are for similar conditions and with edge insulation on one side of the glazing. Therefore, as will be shown, in using the standard distance of 63.5 mm with the convention given in Section 5.2.4 for calculating the U-values, some of the edge effect is accounted for in the center-of-glass U-value.

It was decided to use the finite difference solution to the analytic model to determine approximate window U-values. In order to demonstrate the validity of using this simple one-dimensional solution to calculate U-values, edge-of-glass U-values determined with the FDM are compared with those obtained from a presumably more accurate two-dimensional model: FRAME3.

The edge-of-glass U-value for a vacuum glazing with different insulation recess depths was calculated with the FRAME3 program by Fred Beck of Lawrence Berkeley National Laboratory. The geometry and conditions for this simulation are as indicated in Fig. 5.22. The modelled conditions are the ASHRAE winter conditions given above: air temperatures of 21.1 and -17.8 °C, and corresponding heat transfer coefficients of 7.41 Wm⁻²K⁻¹ and 29.00 Wm⁻²K⁻¹ over the glass surfaces. There is a different warm side heat transfer coefficient of 7.60 Wm⁻²K⁻¹ over the insulation due to the different emittance properties of the insulation surface (as explained above). The glass-to-glass center-of-glass U-value of the glazing is 1.20 Wm⁻²K⁻¹. The base of the glazing rests on a slab of insulation of width 25.40 mm, length 59.00 mm and thermal conductivity 0.03 Wm⁻¹K⁻¹. There is a 13.34 mm thick layer of edge insulation of the same thermal conductivity on the warm side of the structure. The edge recess depth for this layer of insulation is measured from the inner edge of the 6 mm wide seal. Four recess depths were modelled: 0 mm, 6.35 mm (0.25 in.), 12.70 mm (0.50 in.) and 25.40 mm (1.00 in.). The U_{edge} values were calculated over the standard distance from the sightline (63.5 mm). The simulated size of the glazing was equal to the generic ASHRAE residential size of 0.914 m by 1.220 m [6]. The one-dimensional finite difference solution to the analytic model was solved for the

same ambient conditions, for an equivalent air-to-glass thermal conductance via the insulation ($h_i=1.74 \text{ Wm}^{-2}\text{K}^{-1}$) and for similar recess depths of 0, 6, 13 and 25 mm but with no insulation at the base of the glazing. Values for U_{edge} were calculated from the FDM results in the manner described above for the same glazing size simulated with FRAME3.

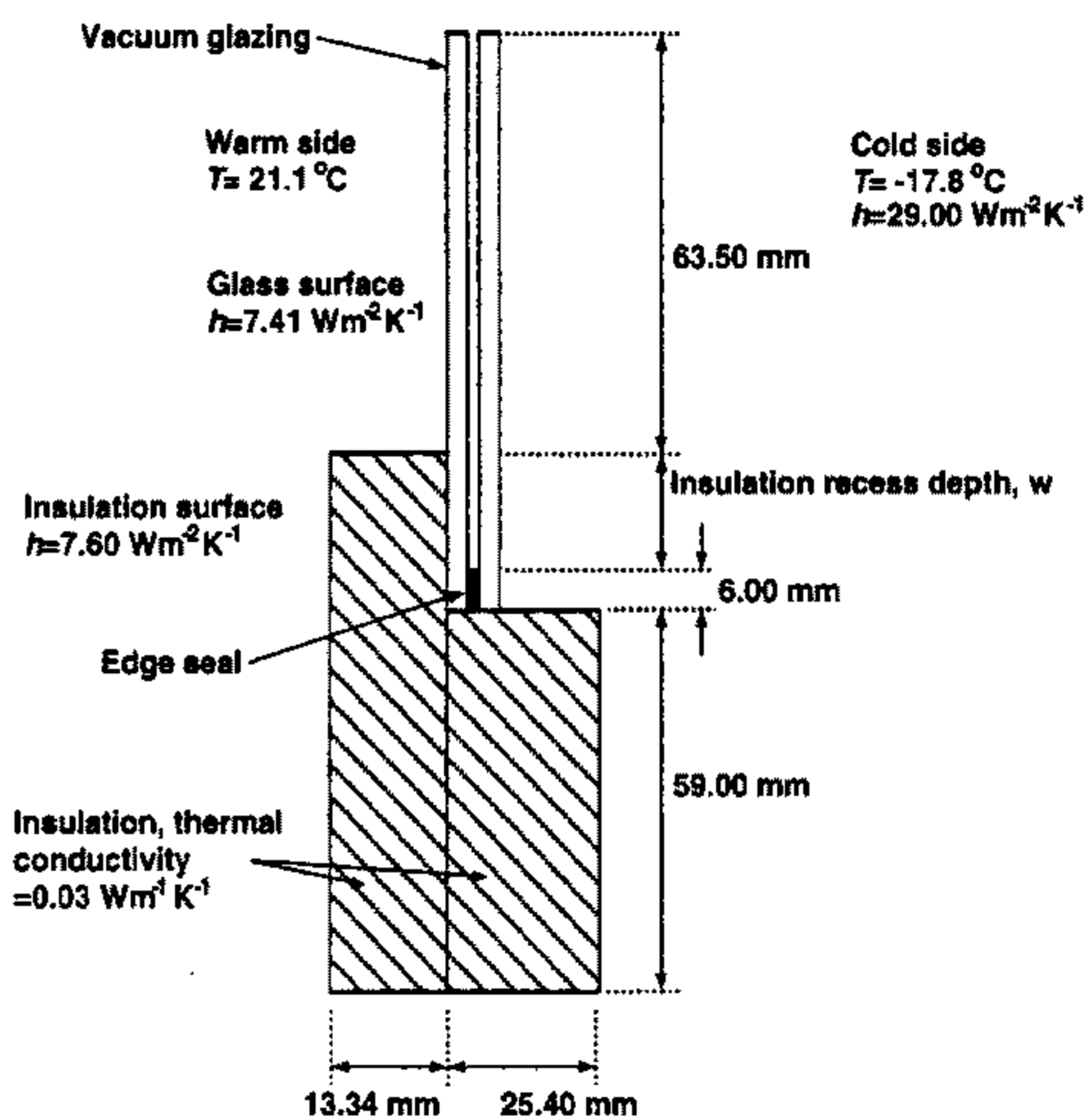


Fig. 5.22: The geometry and the conditions for the calculation of the vacuum glazing edge-of-glass U-values with FRAME3. Not to scale.

The edge-of-glass U-values obtained from the above analysis with the FDM and with FRAME3 are presented in Fig. 5.23. The FDM results are about 3 to 4 % greater than those from the FRAME3 model. This is most likely due to the fact that the FDM does not simulate an actual edge seal, the complex two-dimensional heat flows at the edge, or insulation at the base of the glazing. However, the trends of both sets of results are the same. Thus, the simple one-dimensional FDM can be used to determine approximate edge-of-glass U-values, and to study the relative effects of the edge insulation recess depth on U_{edge} . Since the edge-of-glass region is the location of the greatest changes in the temperature gradient, it can be expected that of all the individual U-values that can be calculated with the FDM (U_{frame} , U_{edge} , and U_{cog}), the edge-of-glass U-value, U_{edge} , should be in the greatest error. Therefore, since this U-value is so close to the presumably more exact FRAME3 results, it is assumed that the FDM can also be used to predict the other U-values with at least the same (and more likely a better) accuracy.

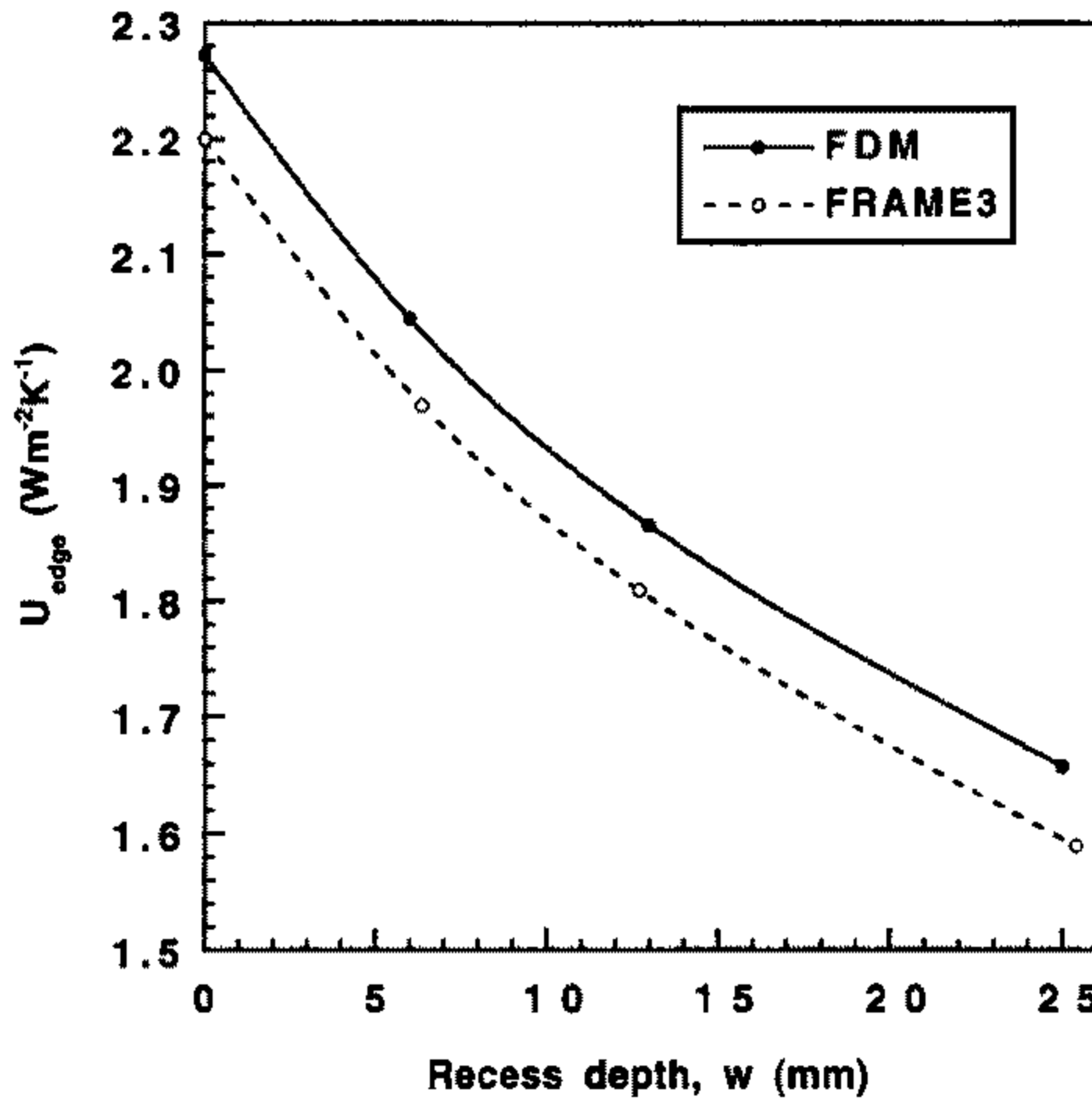


Fig. 5.23: Results from FRAME3 and the finite difference solution to the analytic model (FDM) for the edge-of-glass U-value as a function of the insulation recess depth. The geometry and conditions for the FRAME3 results are shown in Fig. 5.22. The FDM results are for the equivalent situation, but with no insulation at the base of the glazing. The results are for a glazing of size 0.914 m by 1.220 m.

Having thus established the validity of using the FDM to calculate U-values, this model was used to calculate window U-values as functions of the insulation recess depth. This was done for the ASTM standard test conditions, a glazing of size 1 m by 1 m, and insulation of the same recess depth on both sides of the glazing. The same glass-to-glass center-of-glass U-values, and the same insulation thermal conductances considered for Fig. 5.20 were simulated: $U_{g-g}=1.2$ and $4.4 \text{ Wm}^{-2}\text{K}^{-1}$, and $C_{ins}=0, 1.5$ and $3 \text{ Wm}^{-2}\text{K}^{-1}$.

Figure 5.24a is a plot of U_{frame} as a function of the insulation recess depth, for $U_{g-g}=1.2 \text{ Wm}^{-2}\text{K}^{-1}$ and for two values of insulation thermal conductance: 1.5 and 3 $\text{Wm}^{-2}\text{K}^{-1}$. In the limit of both sides of the glazing being completely covered in insulation of conductances 1.5 and 3 $\text{Wm}^{-2}\text{K}^{-1}$, the U-values over the frame region would be slightly greater than the resulting respective center-of-glass U-values of 0.43 and 0.60 $\text{Wm}^{-2}\text{K}^{-1}$. These are close to the values to which it appears that the plots of Fig. 5.24a will converge. The frame U-value for $C_{ins}=3 \text{ Wm}^{-2}\text{K}^{-1}$ is greater than that for $C_{ins}=1.5 \text{ Wm}^{-2}\text{K}^{-1}$ because more heat flows through the insulation with the larger thermal conductance. This also explains the same qualitative dependence of U_{edge} on the insulation thermal conductance, which is presented in Fig. 5.24b for the

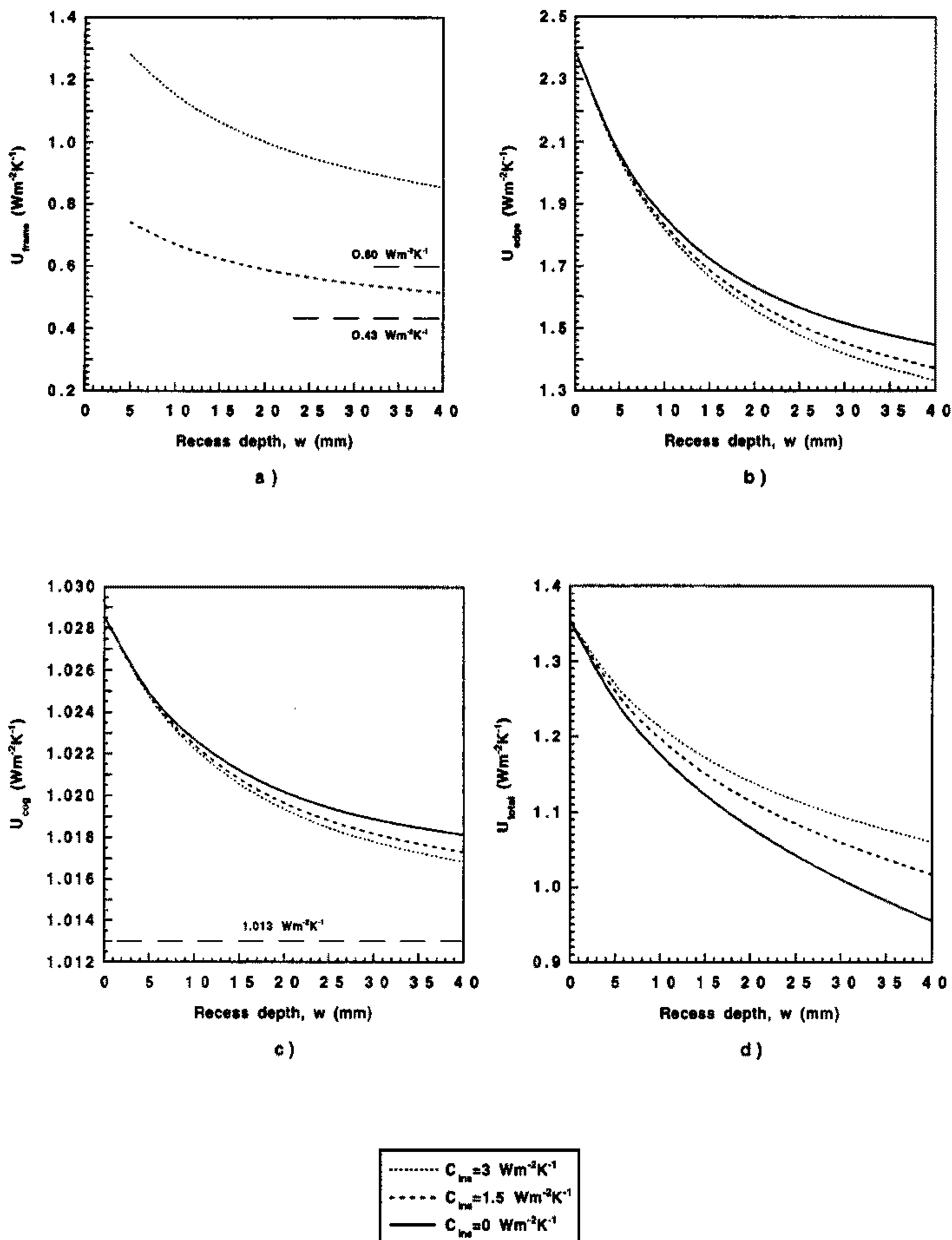


Fig. 5.24: U-values as functions of the insulation recess depth for different insulation thermal conductances (C_{ins}), for a glass-to-glass center-of-glass U-value (U_{g-g}) of $1.2 \text{ Wm}^{-2}\text{K}^{-1}$, and for the ASTM standard test conditions: (a) frame U-value, (b) edge-of-glass U-value, (c) center-of-glass U-value, and (d) total window U-value. The results are for a window of size 1 m by 1 m. The insulation recess depth is the same on both sides of the glazing.

same U_{g-g} and the same insulation conductances - plus an additional insulation conductance of $0 \text{ Wm}^{-2}\text{K}^{-1}$ (the zero conductance was not modelled for the calculation of U_{frame} since it obviously results in a zero frame U-value).

Figure 5.24c presents the center-of-glass U-value as a function of the insulation recess depth for the same values of U_{g-g} and C_{ins} modelled for Figs. 5.24a and b. As mentioned above, the edge effect extends beyond the standard definition of the edge-of-glass region (i.e. beyond 63.5 mm from the sightline). Therefore, some of the edge effect is accounted for in the center-of-glass U-value, as can be seen in Fig. 5.24c; even with an insulation recess depth of 40 mm, the U_{cog} results are above - but converging towards - the center-of-glass U-value that is expected in the absence of any influence from the edge effect: $1.013 \text{ Wm}^{-2}\text{K}^{-1}$ (this differs from the value for U_{g-g} because U_{cog} is an air-to-air quantity). Total window U-values were calculated according to equation 5.16 from the results in Figs. 5.24a to c, and they are presented in Fig. 5.24d.

The above calculation of the U-values was repeated for a larger glass-to-glass center-of-glass U-value: $4.4 \text{ Wm}^{-2}\text{K}^{-1}$. The corresponding results for U_{total} as a function of the insulation recess depth are presented in Fig. 5.25.

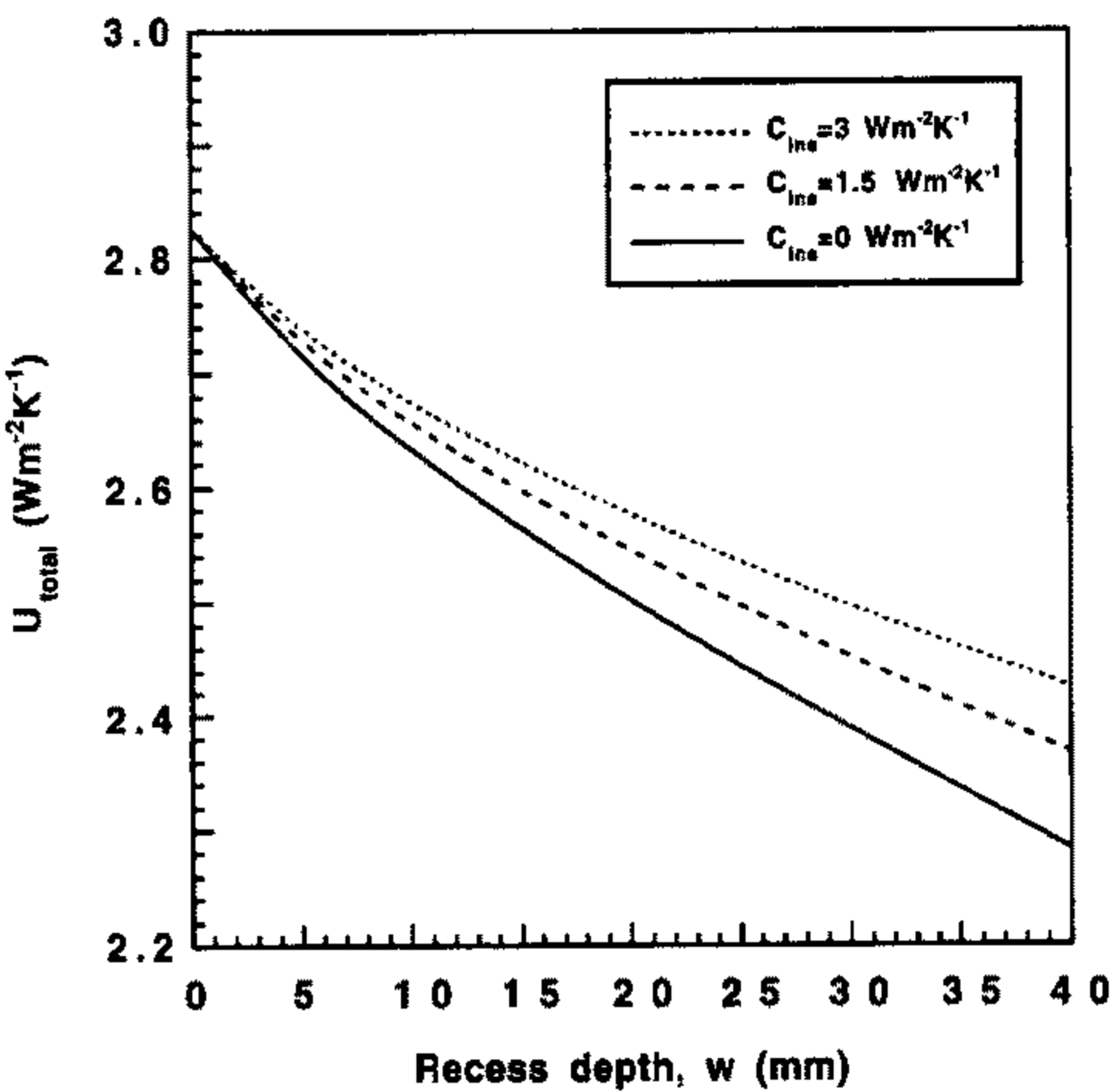


Fig. 5.25: The total window U-value as a function of the insulation recess depth for different insulation thermal conductances (C_{ins}), for $U_{g-g}=4.4 \text{ Wm}^{-2}\text{K}^{-1}$, and for the ASTM standard test conditions. The results are for a window of size 1 m by 1 m. The insulation recess depth is the same on both sides of the glazing.

5.4.3 Reduction of Edge Conduction with Recessed Edges

The results of Figs. 5.18 show how edge insulation can be used to increase the warm side sightline temperature, and thus the condensation resistance of the glazing. Similar increases can also be obtained by recessing the glazing edge into the surrounding sash - i.e. by essentially using the sash as edge insulation. The influence of flush mounted and recessed edge configurations on the edge effect of vacuum glazing was investigated with an infrared thermographic test at Lawrence Berkeley National Laboratory that was performed on the same sample considered above. The results are presented in Fig. 5.26a as the "Vacuum glazing" profile. As shown in Fig. 5.26b, the sample was mounted in a commercially available fixed wood casement frame with flush mounting on the left jamb (vertical edge) and with recessed mounting on the right jamb. For the former configuration, the sash was flush to the inner edge of the seal. For the latter configuration, a recessed sash was simulated by adding strips of wood to a similar flush mounting on both the warm and cold sides of the glazing; these strips extended 12.7 mm (0.5 in.) beyond the inner edge of the seal. The environmental conditions were approximately the same as those given in Tables 5.1 and 5.2. Infrared thermographic measurements were made across the glazing at about the mid-jamb height. The results, which as noted above are presented as the "Vacuum glazing" profile of Fig. 5.26a, are summarised in Tables 5.3 and 5.4. This temperature profile and the other one of this figure (explained below) are from single linescans only; there was no averaging of results over a 50 mm wide section and thus there is more noise in these results than in those of Figs. 5.16 and 5.17. The measured sightline temperature of the recessed side is 5.1 °C higher than that of the flush side. However, recessing the glazing edge into the sash results in a minimum sash surface temperature which is 0.7 °C lower than that of the sash surface on the flush side. The slight rise in the temperature at the middle of the sample is, as explained above for the similar effect seen in the thermograms presented in Figs. 5.14 and 5.15, associated with the outwardly displaced pillars around this location.

5.4.4 Reduction of Edge Conduction by The Addition of a Third Pane

Edge conduction and the associated edge effect can be significantly reduced by covering the entire vacuum glazing with a highly insulating layer on one or both sides. This condition can be established by adding a third pane of low emittance glass to the sash, thereby completely covering the warm side surface of the vacuum glazing. Infrared thermographic measurements were made at Lawrence Berkeley National Laboratory of the above described vacuum glazing sample which was modified in this manner. The third pane of glass was 3 mm thick and the coated

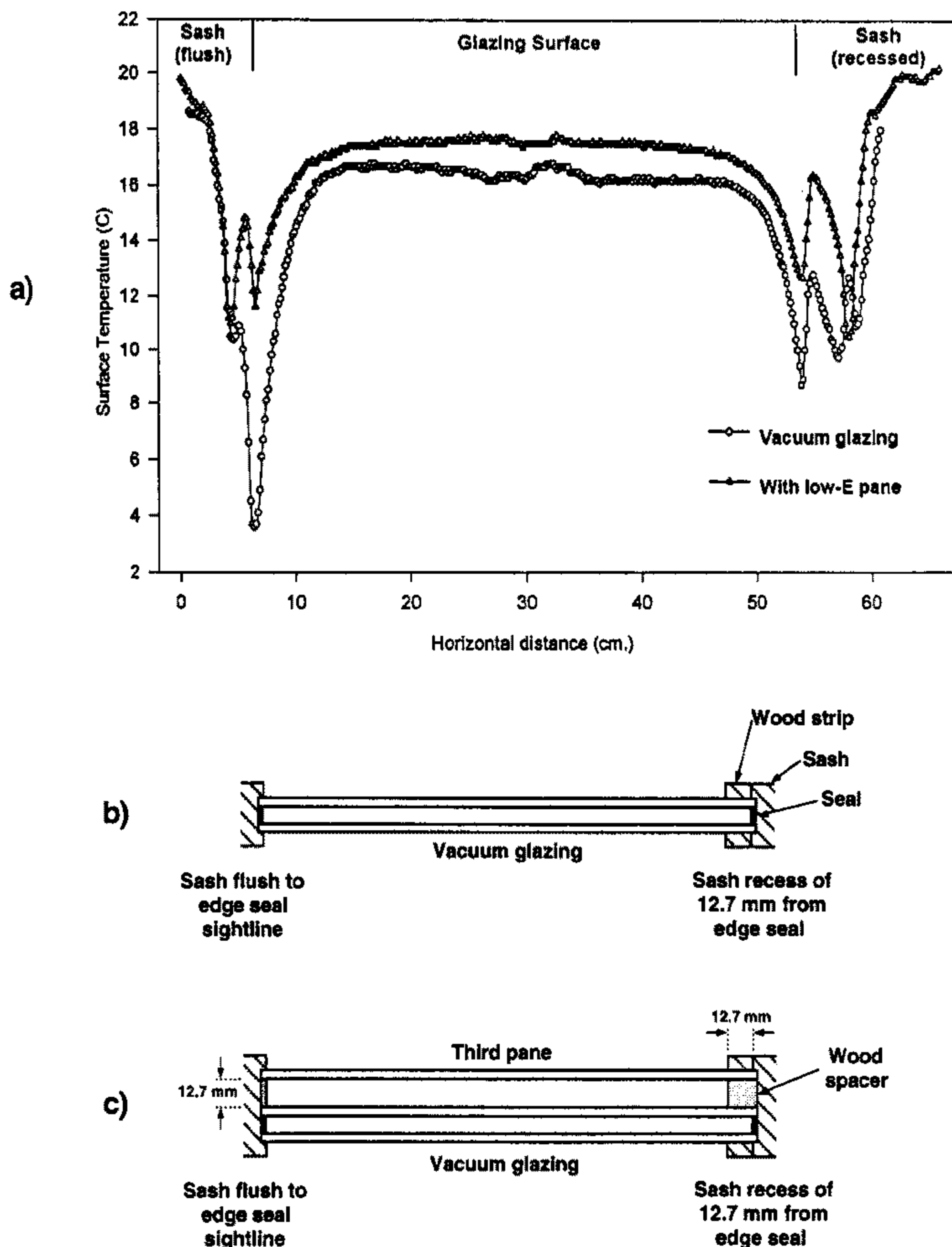


Fig. 5.26: (a) Infrared thermographic measurements of the warm side surface temperatures of a vacuum glazing with and without a low emittance third pane (low-E pane) of glass. The glazing was mounted in a fixed wood casement frame. The region beyond the sash is the insulating mask wall. The edge configurations are shown for the vacuum glazing alone (b), and for the vacuum glazing combined with a third pane (c). The third pane is separated from the glazing by an air cavity which is maintained by 12.7 mm (0.5 in.) thick wood spacers. The glazing edge seal, or spacer in the case of the third pane configuration, is flush to the sightline on the left jamb, and recessed into the sash by 12.7 mm on the right jamb of the sample. The recess is accomplished with wood strips and the recess depth is measured from the inner edge of the seal. In (a), the sightlines are indicated by the vertical lines at the top of the graph. The measurements were taken for environmental conditions which were approximately the same as those given in Tables 5.1 and 5.2. Figure (a) was prepared by Fred Beck of LBNL.

surface, which faced the vacuum glazing, had a hemispherical total emittance of 0.197. As shown in Fig. 5.26c, this pane was separated from the glazing by 12.7 mm (0.5 in.) thick wood glazing stops (spacers), creating a 12.7 mm air cavity between the pane and the vacuum glazing. This structure is essentially a hybrid of vacuum glazing and conventional glazing. As in the previous test, each side of the window had a different configuration; the left jamb spacers were flush with the edge seal sightline, and the right jamb edge seal was recessed by 12.7 mm into the sash.

Infrared thermographic measurements were made of the third pane side of this hybrid glazing for environmental conditions which were approximately the same as those given in Tables 5.1 and 5.2. The temperature profile is presented in Fig. 5.26a (as "With low-E pane") and the results are summarized in Tables 5.3 and 5.4. The magnitude of the temperature variation associated with the edge effect was significantly reduced by the addition of the third pane of glass. Compared with the results for just the vacuum glazing, the sightline temperature for the recessed edge configuration is 4.0 °C higher, and the average center-of-glass temperature is 1.0 °C higher. With the third pane of glass, the minimum sash temperatures are lower than the sightline temperatures because of the effect of the cold glazing edge behind the wood spacers. This effect could be reduced with the use of a more insulating spacer material between the vacuum glazing and the third pane. The need for a recessed glazing edge is virtually eliminated by the use of the third pane; the data show that the flush and the recessed sightline temperatures with the third pane are much closer to the center-of-glass temperature than those without the third pane, and that they are within 1.1 °C of each other.

5.4.5 Summary of LBNL Results

Tables 5.3 and 5.4 summarize the IR thermographic results from Lawrence Berkeley National Laboratory for the edge configurations considered in the previous two sections.

Fred Beck of Lawrence Berkeley National Laboratory also obtained modelling results of U-values and warm side external surface sightline temperatures for a vacuum glazing and, for comparison, for an atmospheric pressure superwindow [11] which were mounted in fixed wood casement frames with the same edge configurations considered in the previous two sections. This modelling was done with WINDOW4 and FRAME3 according to the NFRC 100-91 simulation procedure referred to above [5], and for the ASHRAE winter conditions. The generic ASHRAE residential

window size of 0.914 m by 1.220 m [6] was simulated. The modelled superwindow is a 25.4 mm (1 in.) thick four-layer glazing consisting of two low emittance films suspended between two glass sheets. All of the cavities are filled with a mixture of krypton and argon gas, and the edge of the superwindow is sealed with a thermally broken spacer. The air-to-air center-of-glass U-value for the superwindow, as determined by a WINDOW4 simulation, is $0.68 \text{ Wm}^{-2}\text{K}^{-1}$. The results from this analysis are presented in Table 5.5, and they are also reported by Simko, Collins, Beck and Arasteh [12].

The results of Table 5.5 show that recessing the edge into the sash can significantly reduce the frame and edge-of-glass U-values. The sash recess depth considered in this study (12.7 mm) is probably about the maximum depth that would be used by a manufacturer. This is because a diminishing point of return, like that shown above for the effect of the insulation recess depth on the sightline temperature, can be expected for the influence of the sash recess depth on the U-values. Furthermore, unless the size of the glazing is increased, a greater sash recess depth will reduce the view area of the window.

The greatest reduction in all of the U-values is obtained with the addition of the third pane of low emittance glass. Further reductions could be obtained with a low conductivity gas in the cavity between this pane and the vacuum glazing. A comparison among the sightline temperatures of Table 5.5 shows that the external surface of the hybrid glazing would have the best condensation resistance (provided that the air cavity was maintained free of moisture so as to prevent condensation on the internal cavity surfaces). Although the addition of a third pane of glass to a vacuum glazing improves the thermal performance of the product, this is accomplished at the expense of a key attractive feature of vacuum glazing: its thinness. However, a third pane could be combined with a vacuum glazing as an easily removable storm window that would probably be used only during the winter - as is done with some conventional glazings in cold climates. The additional weight of the third pane would necessitate more and stronger framing materials, as well as stronger hinges - the cost of which would have to be considered in determining if the addition of a third pane is economically advantageous.

	Temperature (°C)	
	Vacuum glazing only	With third low-E pane
Minimum sash temperature	10.4	11.0
Sightline temperature	3.6	11.6
Average center-of-glass temp.	16.5	17.5

Table 5.3: Infrared thermographic results for the warm side surface temperatures of a vacuum glazing with and without a third low emittance (low-E) pane of glass, mounted in a wood frame with the edge seal or spacer flush to the sightline. The results are from Fig. 5.26a and they are for environmental conditions which are approximately the same as those given in Tables 5.1 and 5.2.

	Temperature (°C)	
	Vacuum glazing only	With third low-E pane
Minimum sash temperature	9.7	10.5
Sightline temperature	8.7	12.7
Average center-of-glass temp.	16.5	17.5

Table 5.4: Infrared thermographic results for the warm side surface temperatures of a vacuum glazing with and without a third low emittance (low-E) pane of glass, mounted in a wood frame with the edge seal or spacer recessed from the sightline by 12.7 mm. The results are from Fig. 5.26a and they are for environmental conditions which are approximately the same as those given in Tables 5.1 and 5.2.

Configuration	U-values (Wm ⁻² K ⁻¹)				<i>T_{sightline}</i> (°C)
	<i>U_{frame}</i>	<i>U_{edge}</i>	<i>U_{cog}</i>	<i>U_{total}</i>	
Vacuum glazing, flush edge	2.23	2.09	1.00	1.52	-2.3
Vacuum glazing, recessed edge	2.07	1.58	1.00	1.42	6.0
Vacuum glazing, flush edge & third pane	1.69	1.52	0.75	1.14	8.8
Vacuum glazing, recessed edge & third pane	1.71	1.32	0.75	1.14	10.0
Superwindow, flush edge	1.65	1.06	0.68	1.01	10.0
Superwindow, recessed edge	1.57	0.87	0.68	0.99	13.2

Table 5.5: Simulation results obtained by Fred Beck for a vacuum glazing and a superwindow mounted in a generic wood casement frame. The sightline temperatures (*T_{sightline}*) are at the external warm side surface of the glazing.

5.4.6 Surface Temperature Variations Due to Pillar Conduction

It was noted above that there is a periodic variation in the infrared thermographic results of both Figures 5.16 and 5.17 beyond about 75 mm from the sightline. This variation is associated with the temperature suppressions due to pillar conduction (the suppressions around individual pillars can be seen in Figs. 5.14 and 5.15). In this section, the magnitude and period of the variation in these infrared thermographic results are compared with the magnitude and period of the variation in results obtained from the analytic and finite element modelling approaches presented in the last chapter. This is done in order to validate experimentally these approaches. It might be thought that it would be best to base this comparison on results from single linescans across the glazing. However, as shown by Fig. 5.26a, the noise in the infrared thermographic results for a single linescan is so great that it is very hard to discern the temperature variations due to pillar conduction. Due to this noise, it would be difficult to make a meaningful comparison between the experimental and modelling data. Therefore, for this analysis, only averaged surface temperatures, like those of Figs. 5.16 and 5.17, are considered.

Figure 5.27 presents infrared thermographic results of the same data set from which the results of Fig. 5.16 were taken. These results are from 112 to 203 mm from the sightline. Since, as indicated by the data of Fig. 5.16, the glass temperatures within this region should be beyond the influence of the edge effect, it is possible to see clearly a small variation of about 0.3 °C in the results. As previously mentioned, this variation is associated with the temperature suppressions due to pillar conduction. The periodicity of this variation is, as expected, approximately equal to the separation of the pillars in the vacuum glazing sample: 25 mm. It is important to note that these results are the averages of the temperatures across a 50 mm wide section of the glass surface - as shown in Fig. 5.11. Since this section encloses the equivalent of two rows of unit cells, the average temperatures between successive minima (e.g. the minima at 150 and 175 mm) represent the average temperatures across a unit cell of width 25 mm (i.e. between two rows of pillars separated by 25 mm). A method was presented in Section 4.4.1 for determining such temperatures across a unit cell on the basis of results from a finite element model for the analysis the effects of the heat conduction through a single pillar; results from this method are presented in Fig. 4.12. This same method was employed to obtain modelling results for the conditions corresponding to the infrared thermographic data of Fig. 5.27 except that, for this case, internal radiative heat transfer was simulated (the simulated radiative U-value was the same as that given above for the sample: 0.69 Wm⁻²K⁻¹) These modelling results are compared with the infrared thermographic data in Fig. 5.27. The modelling

results calculated across one unit cell are presented for many cycles, and the minima are approximately aligned with those of the IR data of the figure. The total variation in the modelled temperatures ($0.3\text{ }^{\circ}\text{C}$) is consistent with the variation in the measurements. The error for the modelling results is associated with the uncertainties in the environmental conditions. This error is approximately equal to that given for the FEM result of Fig. 5.16 which is furthest from the sightline ($\pm 0.5\text{ }^{\circ}\text{C}$ at 134 mm), the derivation of which is explained above. As previously mentioned, the absolute accuracy of the IR results is $\pm 0.5\text{ }^{\circ}\text{C}$ (or, for this case, only $\pm 1.3\%$ of the air-to-air temperature difference across the glazing). Thus, to within the respective ranges of error, the magnitude of the variation in the experimental and modelling results of Fig. 5.27 are in agreement. Due to the above agreement in both the period and magnitude of the variation in the experimental and modelling results for the surface temperature, the thermographic measurements provide an experimental validation of the finite element modelling technique.

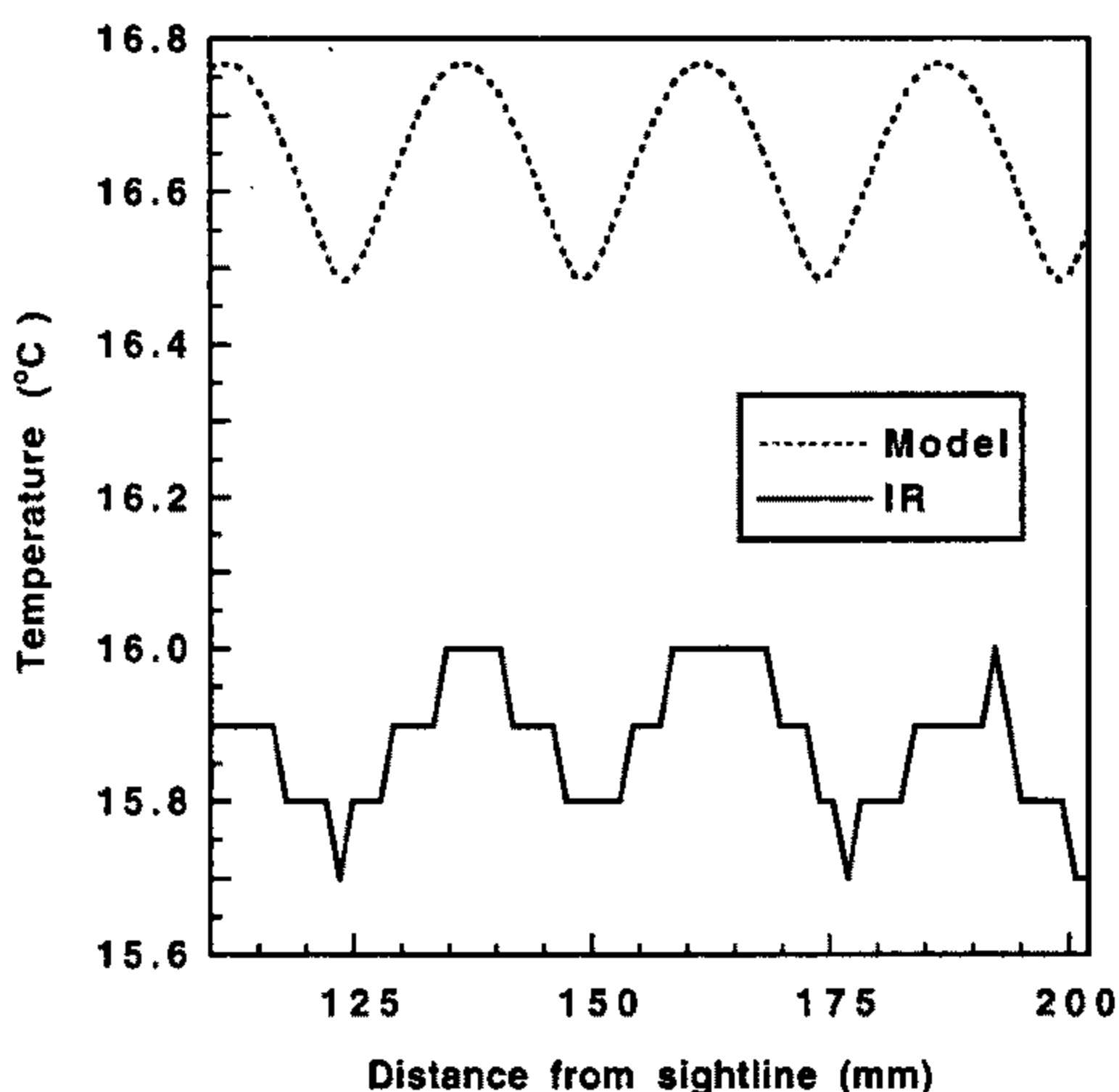


Fig. 5.27: Infrared thermographic results (IR) for the main thermographic test with the insulation covering the cold side edges over the width of the edge seal. The conditions for this test are given in Table 5.1. Also shown are finite element modelling based results (Model) for the same insulation configuration and conditions, and for the glass-to-glass center-of-glass U-value which corresponds to these conditions: $1.23\text{ Wm}^{-2}\text{K}^{-1}$. The thermal conductance of the insulation is $2.9\text{ Wm}^{-2}\text{K}^{-1}$. Both sets of results represent the average temperatures across a unit cell of pillars for $\lambda = 25\text{ mm}$. The modelling results were obtained according to the method described in Section 4.4.1.

It is noted that the above mentioned observed variation of 0.3 °C refers only to the average temperatures across a unit cell. The results of Figs. 4.11 and 4.12, which are for similar conditions, indicate that the point-to-point variation in the surface temperature is approximately twice as large as the peak-to-peak variation in the average temperature across a unit cell. Therefore, the results of Fig. 5.27 should correspond to a point-to-point temperature variation of about 0.6 °C. The point-to-point variation for these experimental results was not recorded. However, the predicted point-to-point variation of about 0.6 °C is approximately consistent with the same type of variation in the data for the infrared thermogram presented in Fig. 5.15 which is for similar conditions: 0.4 ± 0.1 °C.

It is recalled that, for the conditions considered for Fig. 5.27 (which are given in Table 5.1), the glass-to-glass center-of-glass U-value for the glazing is $1.23 \text{ Wm}^{-2}\text{K}^{-1}$. This U-value is the sum of a measured radiative U-value which is corrected according to an iterative solution based on equation 5.1, (first introduced as equation 3.3a of Chapter 3), and a pillar array U-value which is calculated by equation 5.2 (first introduced as equation 4.3 of Chapter 4) according to the pillar dimension and separation. The uniform warm side surface temperature predicted with this center-of-glass U-value and the conditions of Table 5.1 is 16.7 °C. The error for this temperature is associated with the uncertainties in the environmental conditions and it is approximately equal to that given for the modelling results of Fig. 5.27: ± 0.5 °C. To within the respective errors, this expected average temperature is in agreement with the average of the infrared thermographic results of Fig. 5.27 (15.9 ± 0.5 °C). Therefore, the infrared thermographic results provide an experimental validation of the following upon which the expected average temperature is based: the methods used to measure and correct the radiative U-value, and the analytic expression for calculating the pillar array U-value.

5.5 Conclusion

Edge conduction arises from the lateral heat flow along the glass sheets and through the thermally short circuiting edge seal. It decreases the thermal performance and the condensation resistance of the glazing. Edge conduction - and thus the influence of the associated edge effect on the surface temperatures and the U-values - can be reduced with edge insulation, recessed edges, and the addition of a third low emittance pane of glass to the glazing. Infrared thermography and several modelling

techniques have been used to study the effectiveness of these methods of reducing the edge conduction.

A one-dimensional analytic model has been developed to study the effects of edge conduction. It is based on the assumption that temperature gradients exist only along the planes, and not through the thicknesses, of the glass sheets. This model leads to coupled differential equations which can be expressed in finite difference form and solved numerically. For certain simple cases, these equations can be solved exactly.

The finite difference solution to the analytic model for edge conduction gives results that are in good agreement with those obtained from a more exact three-dimensional finite element model (which has also been developed to study the effects of edge conduction). However, close to the edge, there is a divergence between the finite difference and finite element modelling results because the finite difference model does not simulate an actual edge seal of finite width and thickness. Edge-of-glass U-values calculated with this finite difference model are close to those obtained at Lawrence Berkeley National Laboratory with a commonly used fenestration modelling package: FRAME3.

Finite element and finite difference modelling results show that the addition of insulation to the edges of the glazing reduces the edge conduction, and that it significantly increases the sightline temperature at the warm side external glass surface. Due to this effect on the warm side sightline temperature, edge insulation also improves the condensation resistance of the glazing.

Infrared thermographic tests were performed on a sample of vacuum glazing at Lawrence Berkeley National Laboratory. External glass surface temperatures obtained with the above mentioned finite element model are in agreement with the thermographic results. Therefore, the infrared thermographic measurements experimentally validate the finite element model. Since results from the finite element and finite difference models are in good agreement for the same conditions as for one of the thermographic tests, the infrared thermographic measurements also validate the finite difference model and the analytic model on which it is based. Infrared thermographic tests were also performed with the glazing mounted in a fixed wood casement frame for both flush and recessed edge configurations, as well as with and without a third pane of low emittance glass. The measurements show that significant increases in the sightline temperature at the warm side external glass surface can be achieved with these configurations - especially with the addition of a third pane of

low emittance glass. Modelling results show that these configurations also result in significant decreases in the frame and edge-of-glass U-values.

The finite element model presented in the previous chapter for the study of the effects of pillar conduction was used to obtain surface temperature profiles for the same conditions as for one of the infrared thermographic tests. Due to the agreement in the magnitude and variation of the measured and modelled results, the thermographic measurements provide an experimental validation of this finite element model.

The average center-of-glass surface temperature was predicted with a U-value calculated on the basis of a temperature corrected measured radiative U-value, and a pillar array U-value calculated from the analytic relationship first presented in the previous chapter. This average temperature agrees with that obtained from the infrared thermographic measurements. Thus, these measurements also validate the methods used to determine the above mentioned center-of-glass U-value.

Edge conduction in vacuum glazing causes thermal stresses in the edge region which can result in the failure of the glazing. It is therefore important to be able to quantify the thermal and consequent stress effects of edge conduction. Modelling techniques to calculate the former have been presented and experimentally validated in this chapter. In the next chapter, it is shown how the edge stresses can be calculated on the basis of the thermal results of the finite element model presented above.

5.6 List of Symbols

a	pillar radius
A_{cog}	area of center-of-glass region
A_{edge}	area of edge-of-glass region
A_{frame}	area of frame region
$A_{projected}$	normal projection of the surface onto a plane perpendicular to the window
A_{total}	total area of window
C_{ins}	thermal conductance of insulation
d	extent of frame
D	dimension of square glazing
h	heat transfer coefficient
h_1	heat transfer coefficient on warm side of glazing
h_2	heat transfer coefficient on cold side of glazing
h_i	thermal conductance equal to the series combination of the insulation conductance and the warm side heat transfer coefficient

h_2	thermal conductance equal to the series combination of the insulation conductance and the cold side heat transfer coefficient
k	thermal conductivity of glass
l	characteristic distance (written as "/" in figures)
L	length of edge (or width of glass sheet)
Q_{cond}	heat conduction along glass sheet
Q_{edge}	heat conduction through edge
Q_{ext}	heat transfer between glass and ambient
$Q_{warm\ side\ surface}$	heat transfer between ambient and the warm side of the glazing
Q_{int}	heat transfer across the glazing by internal processes
t	glass thickness
T_{ave}	average temperature of internal glass surfaces
T'	glass temperature at the inner edge of the seal for both sides of the glazing
T_i	warm (inside) ambient temperature
T_o	cold (outside) ambient temperature
$T_{sightline}$	sightline temperature
$T_1(x)$	temperature of warm side glass sheet
$T_2(x)$	temperature of cold side glass sheet
U_{array}	pillar array U-value
U_{cog}	center-of-glass U-value
U_{edge}	edge-of-glass U-value
U_{frame}	frame U-value
U_{g-g}	glass-to-glass center-of-glass U-value
U_{rad}	radiative U-value
U_{total}	total window U-value
w	insulation recess depth or width (the same on both sides of the glazing)
w_1	insulation recess depth on warm side of glazing
w_2	insulation recess depth on cold side of glazing
x	distance along glass sheet from the inner edge of edge seal (or from the location of the simulated edge seal in the finite difference model)

Greek Symbols

ϵ_{eff}	effective emittance of the internal glass surfaces
λ	pillar separation
σ	Stefan-Boltzmann constant
$\Delta T_{air-to-air}$	air-to-air temperature difference across glazing

5.7 References

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6. Stresses Due to A Temperature Difference Across Vacuum Glazing

6.1 Introduction

As will be shown in Chapter 8, the probability of failure for a vacuum glazing is a function of the stresses in the structure. It is therefore important to consider the origins and magnitudes of these stresses. Since some of these stresses arise from the temperature non-uniformities associated with the heat flows discussed in the previous chapters, the focus of this thesis now shifts to a study of the stress implications of the heat transfer processes in vacuum glazing.

The two sources of stress in vacuum glazing considered in this thesis are: atmospheric pressure, and the presence of a temperature difference across the glazing. An extensive modelling study [1] has shown that atmospheric pressure leads to essentially time independent stresses that are significant only in the vicinity of the pillars and edges. These stresses can be reduced to manageable levels with a properly designed pillar array. In contrast, the stresses due to a temperature difference across the glazing are more extensive and can be significant enough to result in the catastrophic failure of the structure. These thermal stresses are discussed in this chapter, and the stresses due to atmospheric pressure are dealt with in the next chapter. The results for both sources of stress are then combined for the probability of failure analysis presented in Chapter 8.

Due to the inherent difficulty in measuring localized stresses in vacuum glazing, especially within the glass and at the internal glass surfaces, it is preferable to rely on the use of a model for stress analyses - provided that the model has been properly validated. Thermal stresses were studied by Fischer-Cripps using a finite element model for a quarter section of a vacuum glazing [2]. This model was indirectly validated by measurements of the deflection of a glazing exposed to particular external conditions. However, since these test conditions were not well defined, there were large uncertainties in the results. Furthermore, the heat transfer coefficients to which the glazing was exposed were quite unrealistic (e.g. the heat transfer coefficient on one side was about $100 \text{ Wm}^{-2}\text{K}^{-1}$). Thus, a more accurate validation procedure has been devised.

In this chapter, a finite element model similar to that used by Fischer-Cripps is developed to calculate the thermal stresses in vacuum glazing. This model is directly validated by measurements of both surface deflections and strains which were made

under well defined conditions. Details of the measurement apparatus and procedure are presented. The work in this chapter is an extension rather than a duplication of the previous work on thermal stresses done by Fischer-Cripps; a more refined model which gives more accurate results is presented, and this model is experimentally validated in a significantly different and more exact manner. The validation of this model is crucial to the work of Chapter 8 because the model is used to predict stresses from which probabilities of failure for vacuum glazings are calculated.

6.2 Stresses Due to Temperature Differences

The presence of a temperature difference across a vacuum glazing causes dimensional differences between the two glass sheets. The differential thermal expansion and contraction of the warm and cold glass sheets, which are bonded together around the edges, places the sheets in compression and tension respectively. This leads to shear forces in the edge seal, where the sheets are joined. These shear forces produce torques relative to the mid-plane of each glass sheet which can cause the sheets to bend, as shown in Fig. 6.1. The large forces due to atmospheric pressure ensure that the sheets remain in contact through the array of support pillars and thus the sheets essentially bend together. The bending is such that the external surface of the warm glass sheet is convex, and is therefore in tension, while the internal surface of this sheet is concave and experiences a compressive stress. The reverse situation applies to the cold sheet. The stresses in the sheets, as a result firstly of compression or expansion, and secondly of bending, are therefore additive on the internal surfaces of the glass sheets, and in an opposite sense on the external surfaces.

An analytic model has been developed for calculating the bending and stresses of a vacuum glazing which is exposed to a uniform glass-to-glass temperature difference [3]. According to the model, the sample assumes a shape which is part of a sphere of radius $4t/(3\alpha\Delta T_{g-g})$, where t is the thickness of each equally sized glass sheet, α is the linear coefficient of thermal expansion for glass, and ΔT_{g-g} is the glass-to-glass temperature difference across the sample. The maximum resulting deflection, d , of a square sample of side L , between the middle of one edge and the centre is:

$$d = \frac{3L^2\alpha\Delta T_{g-g}}{32t} . \quad (6.1)$$

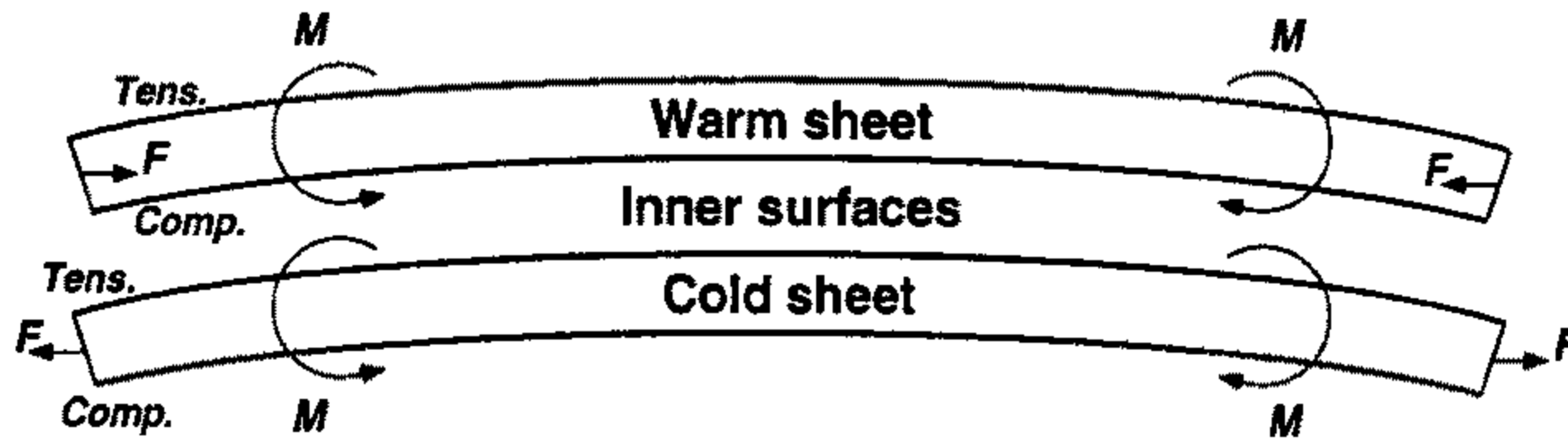


Fig. 6.1: If a temperature difference is applied across a vacuum glazing, the differential thermal expansion and contraction of the glass sheets leads to shear forces in the edge seal and torques, M , which cause the sheets to bend. The bending is such that the glass surfaces are under tensile and compressive stresses (Tens. and Comp.) as indicated. The torques actually occur at the edges.

The model predicts that the stress at the external surface of each glass sheet of an unconstrained sample of vacuum glazing is isotropic in the plane of the glass sheet, and is of magnitude:

$$\sigma = \frac{E\alpha\Delta T_{g-g}}{4(1-\nu)}, \quad (6.2)$$

where E and ν are the Young's modulus and the Poisson's ratio of glass. This stress is tensile on the warm side external surface, and it is compressive on the cold side external surface. The presence of the bonded edge seal significantly complicates the stresses near the edges and thus this simple analytic model provides only approximate results for the stresses and deflections near the center of the glazing. There are four ways in which the edge seal affects the stresses (and thus the deflections) in the edge region.

Firstly, as discussed in Chapter 5, the edge seal is essentially a thermal short circuit between the two glass sheets. Due to the resulting edge effect, the temperature at the edge of each glass sheet is significantly different from that near the center of the sheet. Consequently, the mean temperatures of the warm and cold sheets are, respectively, less and greater than they would be without the edge seal. Thus, the average temperature difference between the sheets is less than for the uniform assumption of the analytic model, and this results in reduced bending and stresses.

Secondly, depending on the external heat transfer coefficients and on the edge insulation (if any), there can be a difference between the temperature of the glass at the edge and the mean temperature of the entire glass structure of the glazing. This temperature difference leads to additional stresses in the edge region of each glass

sheet. If the edge temperature is greater or less than the mean temperature, then the additional stresses are, respectively, compressive or tensile. This is best understood if the glazing is considered to be a continuous plate of glass of thickness equal to that of the two glass sheets. For example, in Fig. 5.18a, the temperature of the glass at the edge ($\sim -6.6\text{ }^{\circ}\text{C}$, based on the average of the warm and cold side external glass surface temperatures) is less than the approximate mean temperature of the glazing ($\sim 0.4\text{ }^{\circ}\text{C}$, based only on the centerline external glass surface temperature profiles). Thus, the glass in the edge region contracts more than the glass elsewhere in the glazing. However, most of this contraction of the edge region glass is prevented by the rest of the glazing. The glass near the edge is therefore under greater tension than if its temperature was equal to the mean glazing temperature. Since the extra contraction is effectively constrained by the inner region of glass, the additional stresses are directed parallel to the edge. If the edge temperature is greater than the mean temperature, then the additional parallel stresses are compressive. The magnitude of the additional stress (tensile or compressive) depends on the difference between the mean and edge temperatures. This temperature difference is of course affected by the heat transfer coefficients: it is non-existent for symmetric heat transfer coefficients, and it increases with the inequality of the coefficients. The presence of insulation near the edges of vacuum glazing, by virtue of its effect on the edge temperature (demonstrated by Figs. 5.18), also influences the stresses in the edge region.

Thirdly, at the external surfaces of the edge of the glazing, there are additional stresses associated with the bending of the structure as a whole. Since the bonded edge is approximately twice as thick as either glass sheet, these bending stresses are twice as large as those due to bending in either glass sheet or, according to the approximate analytic model, three times the center-of-glass external surface stresses. Thus, for each glass sheet, the magnitude of the external surface stress at the edge is:

$$\sigma_{edge} = \frac{3E\alpha\Delta T_{g-g}}{4(1-\nu)} \quad (6.3)$$

Finally, the edge seal is essentially a stiffening bar along the edge. It therefore reduces the curvature of the glazing, and thus changes the magnitude and distribution of the stresses in the structure. A similar effect occurs if the edge is constrained by an external support such as a frame (for the same reasons given in the previous chapter, frames are not considered in the stress analyses of this thesis).

As explained in Chapter 8, the probability of failure for vacuum glazing is most strongly dependent on the stresses at the external glass surfaces. Therefore, since the purpose of this study is to provide data for a probability of failure analysis, only the external glass surface stresses are considered.

6.3 Modelling

A three-dimensional finite element model has been constructed with the STRAND6 modelling package for the analysis of the thermal stresses in a square vacuum glazing. The geometry of the model is presented in Fig. 6.2a. Figure 6.2b shows that, as with the finite element model of the previous chapter, only a quarter section of a the full square glazing need be modelled. Results could also be obtained from a one eighth section of the glazing. However, since the required boundary conditions along the glazing diagonal can not be established with STRAND6, this simulation approach was not employed.

The modelled quarter-section corresponds to a full glazing of size 500 mm by 500 mm. The model consists of two 4 mm thick sheets of soda-lime glass joined along their outer edges by a solder glass seal of height 0.2 mm and width 8 mm. Beyond the edge seal, the glass sheets are separated by a gap which represents the evacuated space, but which contains no elements. The height of this gap is 0.2 mm. Typical actual glass sheet separations are between 0.1 and 0.2 mm. Due to the relative widths of the glass sheets and the evacuated space, and due to the similarity in the material properties of the soda-lime glass and the solder glass, the modelling results are not strongly dependent on the simulated glass sheet separation. Properties for the two components of the model are listed in Table 6.1. References for the values in this table are given in Appendix 1.

Property	Soda-Lime Glass	Solder glass
Coefficient of thermal expansion ($10^{-6} \text{ }^{\circ}\text{C}^{-1}$)	8.0	7.0
Young's modulus (10^{10} Pa)	7.38	6.1
Poisson's ratio	0.21	0.22

Table 6.1: Material properties for the two components of the model shown in Figs. 6.2.

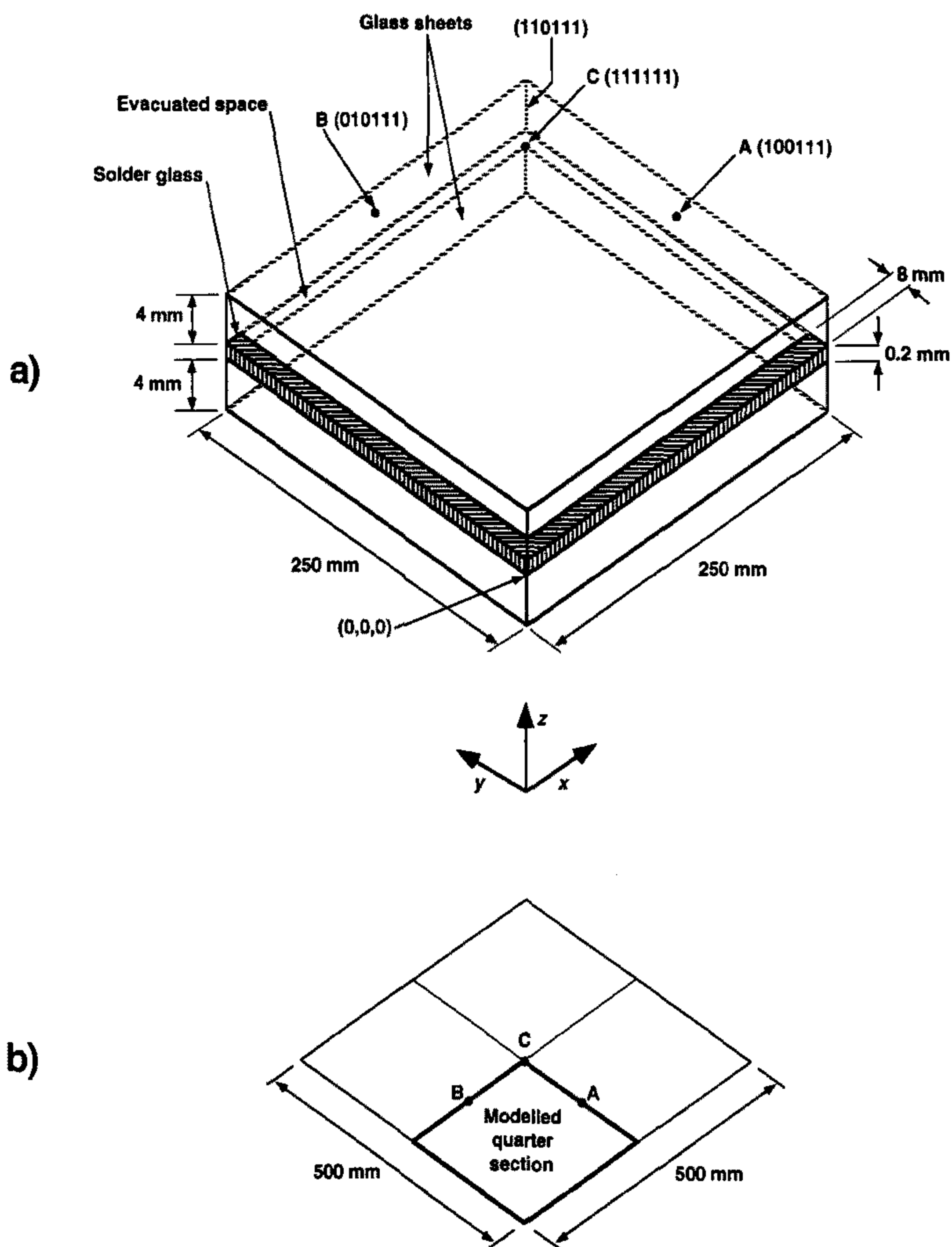


Fig. 6.2: (a) The model for the structural analysis of a quarter section of a vacuum glazing consists of two sheets of glass joined along the outer edges by an edge seal. There are no elements in the evacuated space between the sheets. As shown in (b), the model corresponds to a quarter section of a full 500 mm by 500 mm glazing. The freedom conditions for certain nodes are indicated, where "0" and "1" refer to free and fixed, respectively. Nodes A and B lie on the surfaces of symmetry (the inner edges) of the model; nodes on these surfaces of both sheets of glass are assigned the same freedom conditions as these nodes. Node C, on the internal surface of the lower sheet of glass, is the fully constrained reference node. All other nodes at the inner corner of the model have freedom conditions of (110111).

In an actual vacuum glazing, the two glass sheets are in contact via the support pillars. As shown in the next chapter, atmospheric pressure ensures that, to within a few micrometers, the separation between the two sheets is approximately uniform and equal to the pillar height. However, in this model, there are no elements in the gap between the two glass sheets to ensure a uniform separation when the sheets bend. Due to the small amounts of bending involved, and due to the common freedom conditions specified at the boundaries of the two glass sheets (described below), the sheets have approximately the same radius of curvature (i.e. they essentially bend together). Thus, to within a few micrometers, the width of the gap in this model is maintained at 0.2 mm with and without a temperature difference across the simulated glazing.

In order to ensure that the modelled quarter section is symmetric with the remainder of the full glazing, it is necessary to specify certain freedom conditions for the nodes on the surfaces of symmetry (i.e. the inner edges of the model). In STRAND6, the freedom conditions can be set to permit or prevent translation along, or rotation about lines parallel to the three coordinate axes. The freedom conditions are represented as a series of six numbers - either "0" for free, or "1" for fixed - which correspond to the DX, DY, DZ, RX, RY and RZ settings (e.g. (100111)); DX refers to translation in the x-direction, RX refers to rotation about a line parallel to the x-axis, etc.

Point A of Figs. 6.2a and b is on the surface of symmetry which is parallel to the y-axis. As the glazing bends, this point can translate in the z-direction or towards the center of the simulated glazing in the y-direction. However, by symmetry, it can not move in the x-direction. As with all of the nodes in this model, there is no rotation about the axes. Thus, the freedom conditions for this node, and for all other nodes which are on this surface of symmetry of the model, are (100111). Similarly, the conditions for node B and for all other nodes on the other surface of symmetry of the model are (010111). The nodes at the inner corner of the model (the center of the simulated glazing) can only move in the z-direction and thus they have freedom conditions of (110111) - with the exception of one point at this corner. In order to prevent rigid body motion, it is necessary to specify one node that is completely fixed (i.e. a node which has freedom conditions of (111111)) from which all deflections can be referenced. The node selected for this purpose is at the inner corner, on the internal surface of the lower sheet of glass (point C in Figs. 6.2). The default freedom conditions for all other nodes in the model are (000111).

Since thermal stresses are to be modelled as part of a structural solution, the temperatures of the nodes in this model must be specified. This requires a thermal solution for the required internal and ambient conditions. Clearly this can not be done with the present model because there are no elements for the internal evacuated space, or for the air on each side of the glazing. In Section 5.2.3, a model was presented for the thermal analysis of a quarter section of a 1 m by 1 m glazing. If this thermal model is scaled by one half in the lateral dimensions, and if the width of the edge seal is set at 8 mm, then the structural model described above represents the glass and solder glass components of this modified model. Thus, the structural solution proceeds according to the following steps. The modified thermal analysis model is used to solve for the temperature distribution among all of the nodes for the required conditions. Then the internal and external elements are deleted. What remains is the structural analysis model described above - glass and solder glass elements - with the node temperatures assigned from the thermal solution. The temperature to which the thermal expansions and contractions of the glass elements are referenced is specified. The freedom conditions are set as previously described, and then the model is run.

It is of interest to study the effects of edge insulation on the thermal stresses in vacuum glazing. Edge insulation is simulated in the thermal analysis model as described in Section 5.2.3: for each side of the glazing, an appropriate thermal conductivity is assigned to the external slab over the width of the insulation.

The structural analysis model is similar in construction to that used by Fischer-Cripps, but it contains about three times as many nodes: 5617 versus 1734. Most of the additional nodes are concentrated near the outer edges, where the stress gradients are larger and thus greater accuracy is required. Also, there is an extra row of elements in each sheet of glass. Since the addition of even more nodes does not change the results by more than 0.5 %, it is concluded that the node density of this model ensures that there is a sufficient convergence of the solution.

The structural analysis model provides results for stresses, strains and deflections. In order to test the validity of the model, the calculated results for any one of these quantities can be compared with experimental data. Since this model will be used to predict stresses for the probability of failure analysis in Chapter 8, it would be preferable to obtain comparative stress measurements. However, since it is difficult to make direct and accurate measurements of the stresses in a sample, it was decided to measure the associated external surface strains - which can be easily and accurately

measured with strain gages. External surface deflections were also measured in order to provide an extra set of data for the validation of the model.

The surface strain measurements were made on a sample of vacuum glazing which is exposed to various temperature differences. A temperature difference across the sample leads to two sources of strain in the glass sheets: the stresses described above due to the bending of the glazing, and the thermal expansion (or contraction) of the glass. Thus, the total local strain is:

$$\epsilon = \sigma_{bending} / E + \alpha(T_{glass} - T_o) , \quad (6.4)$$

where $\sigma_{bending}$ is the local stress due to the bending of the glazing, T_{glass} is the local glass temperature and T_o is the reference temperature for the calculation of the thermal expansion (i.e. the temperature at which the unstrained condition is defined). As explained below, due to the nature of the experimental technique, the strain associated with the thermal expansion of the glass (given by the second term of equation 6.4) is not measured; the results reflect only the strains due to the bending stresses (i.e. the first term of this equation). However, the structural model provides results which reflect the bending stresses and the thermal expansions together. Therefore, in order to obtain modelling results for an appropriate comparison with the experimental data, the thermal expansion component is subtracted from each modelled strain. This is done on the basis of the specified reference temperature, and the local temperature predicted by the thermal analysis model. Thus, all modelled strains presented in this chapter, like the measured strains, reflect only the bending stresses of the glass.

6.4 Experimental

6.4.1 Test Cells

Fischer-Cripps measured the surface deflection of a sample of vacuum glazing exposed to a temperature difference [1, 2]. This difference was established by placing one external surface of the sample in contact with water which was maintained at a constant temperature by a water bath, and by exposing the other external surface to air which was at another temperature. Forced convective flow at this exposed surface was established with a fan. The glazing was supported only at the corners in order to permit the glazing to bend in an essentially unconstrained manner. A bar was

mounted above and across the air side surface of the sample. The top surface of this bar provided a reference plane from which deflections across the sample could be measured by inserting a dial micrometer into holes along the bar in a manner similar to that described below.

As previously mentioned, in order to obtain comparative modelling results, the external conditions must be well known. However, for this early apparatus, there were large uncertainties in the estimations of the external heat transfer coefficients. This was because of fluctuations in the ambient conditions, the fact that the air flow velocity was not measured, and because the water and air flows were exposed to each other around the edges of the sample. Therefore, it was decided to build a new apparatus for which the external conditions could be more accurately determined.

A schematic diagram of the new apparatus is shown in Figs. 6.3. The glazing is mounted between warm and cold air cells. The air temperatures in these cells are set by the temperatures of the water flows in the covering plates. The water flow temperatures are maintained by temperature controlled water baths. In order to ensure that the sample is effectively unconstrained, it is supported only at the corners; the glazing rests on ball bearing tipped screws, the heights of which can be adjusted in order to level the sample. Foam insulation is placed around the edges of each cell. The cells are separated by a wooden divider which surrounds, but does not contact the glazing (as such contact could constrain the bending of the sample). The narrow gap between the divider and the edges of the sample is covered with a flexible seal of masking tape. Masking tape is ideal for this purpose because it has a hemispherical total emittance which is close to that of glass [4] and thus it does not significantly perturb the radiative heat transfer between the plate and glazing surfaces.

There are three plates for this apparatus: one at the bottom of the cold cell and two over the warm cell. These aluminum plates are mounted on a wood frame. The cold plate is fixed in the center of the frame and the two warm plates slide along rails at the top of the frame. The warm plates are brought together on opposite sides of the micrometer support bar (described below). The warm plates are large enough to cover completely the warm cell wherever the bar is located on the sample.

Water from Julabo F-20 temperature controlled water circulators enters the plates via copper header and feeder pipes. The two warm plates are connected in parallel to a single circulator. The minimum flow rate, which occurs in the inter-connected warm plates, is approximately 10 liters per minute. This is sufficient to ensure that, for the

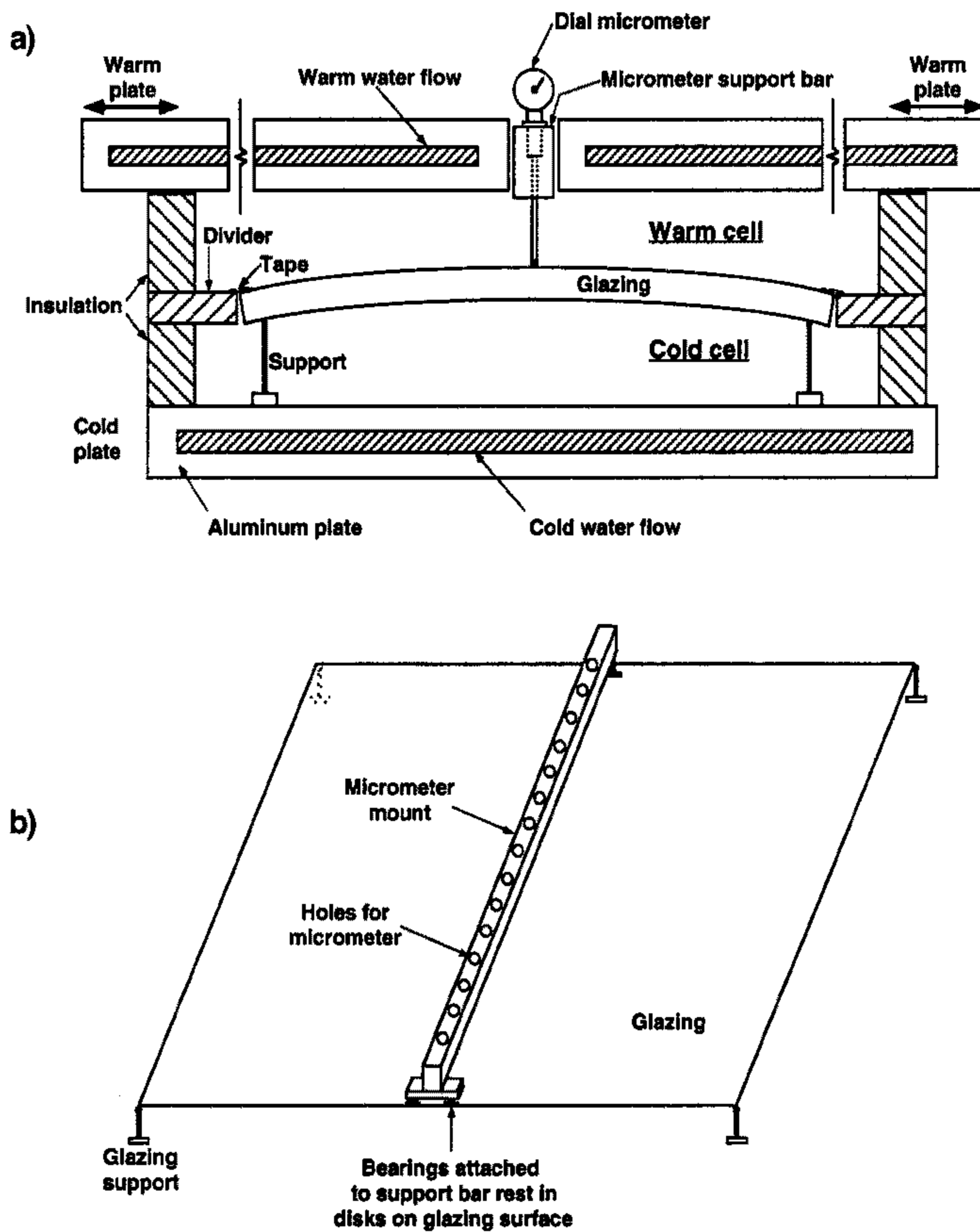


Fig. 6.3: (a) A sectional view of the experimental apparatus showing the glazing between the warm and cold cells. The temperatures of these cells are maintained by water flows through the moveable warm plates, and through the fixed cold plate. Deflection measurements are made with a dial micrometer on a support bar which, as shown in (b), is mounted on the glazing via ball bearings which rest in mounting disks on the glazing surface. The glazing is supported at the corners by ball bearing tipped screws. Only a few of the 32 holes in the bar are shown.

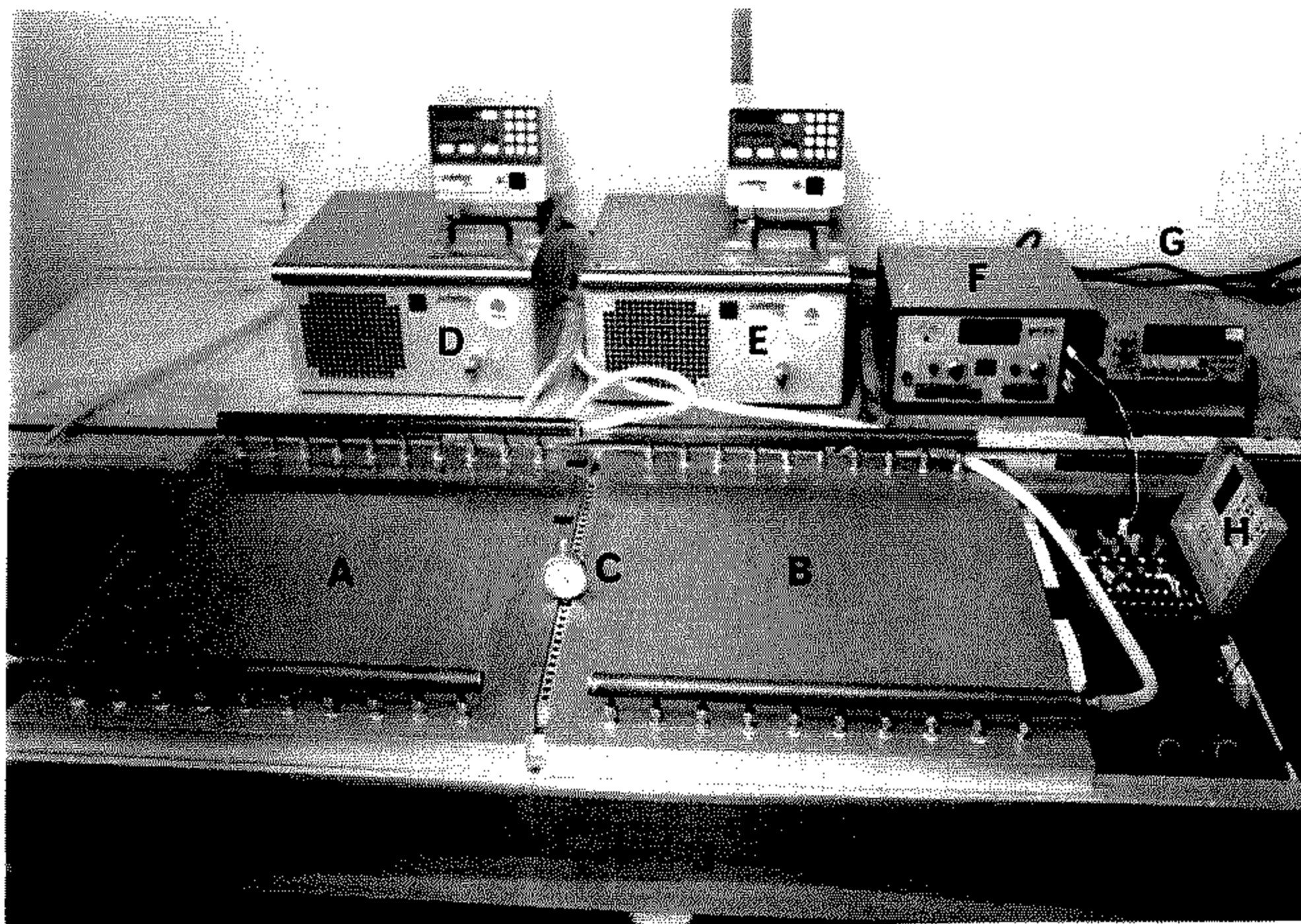


Fig. 6.4: A Photograph of the experimental apparatus with the warm plates (A and B) together as during a test. The dial micrometer (C) rests in one of the holes of the micrometer support bar which is between the warm plates. Behind the plates are the warm and cold water baths (D and E), the strain indicator (F), and the digital multimeter (G) for the thermocouple measurements. To the right of the plates are the switch and balance unit (H) and thermocouple switch box (I), to the right of which is the thermometer. The plates rest on a wooden frame (J). The tubes on top of the warm plates are the header and feeder pipes for the water flow to and from the warm water bath.

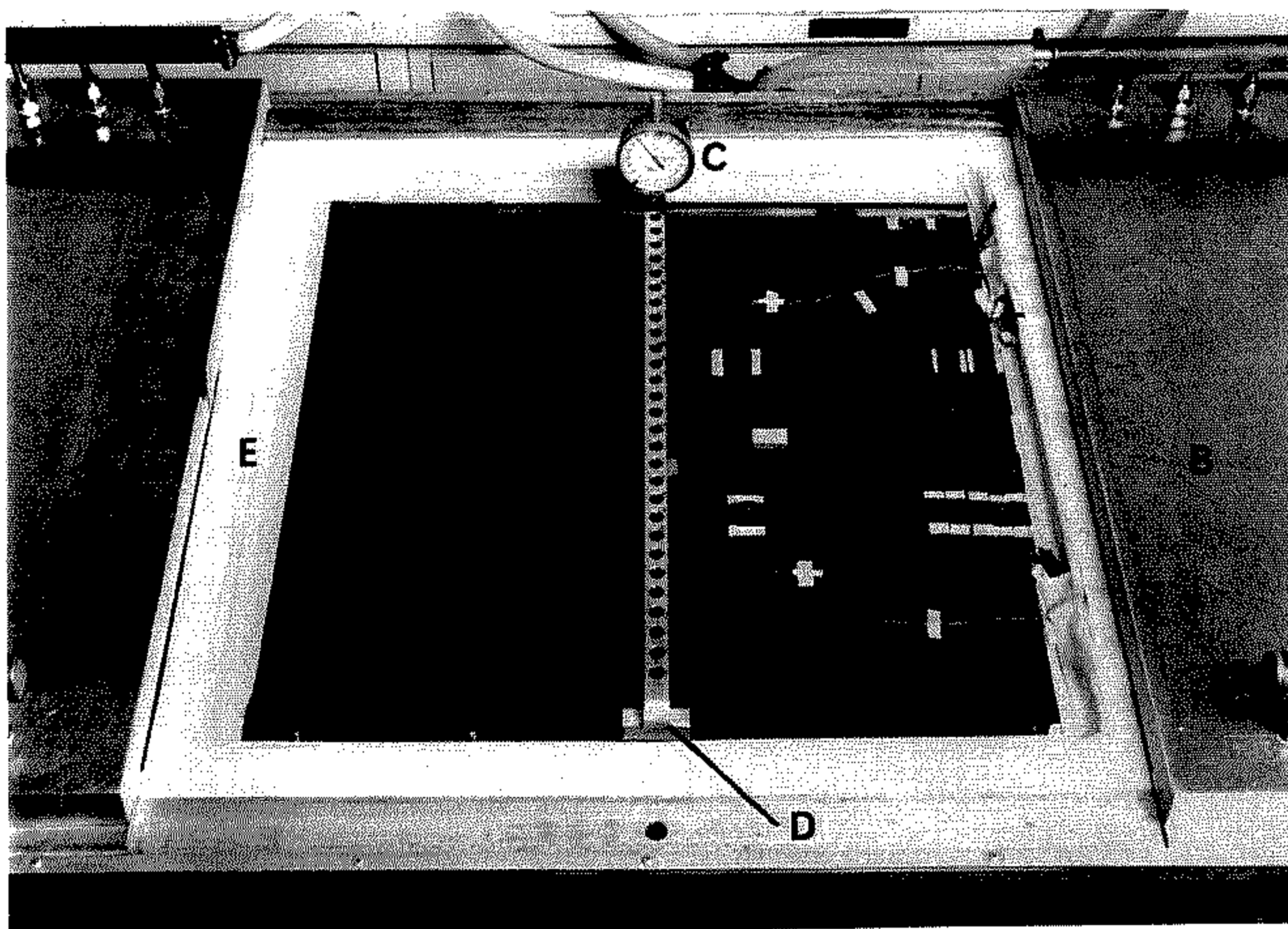


Fig. 6.5: A close-up view of the experimental apparatus with the warm plates (A and B) separated in order to show the sample, micrometer (C), micrometer support bar (D) and strain gages (and the gage connecting wires). The gages are to the right of the micrometer support bar, and they are taped to the sample. The warm cell insulation (E) is also visible.

plate dimensions and the intended operating conditions, the inlet and outlet flows are within 0.1 °C of each other. The warm and cold water flows are parallel and in opposite directions to each other.

In order to ensure that there would be easily measurable strains and deflections, it was decided to design the apparatus so that it would be possible to establish an external glass-to-glass temperature difference of about 20 °C across the type of vacuum glazing to be studied. This type of glazing consists of two sheets of K glass and pillars with a typical dimension and separation (0.2 and 25 mm, respectively); for internal glass surface temperatures of 23 and 3 °C, such a glazing has a glass-to-glass center-of-glass U-value of about 1.20 Wm⁻²K⁻¹. The total heat transfer coefficients in the cells must be large enough to establish the desired 20 °C external glass-to-glass temperature difference with plate temperatures for which the heat flow across the glazing will be stable: temperatures above the dew point but not too much greater than the ambient temperature.

Since it could be difficult to quantify the effects of convective heat transfer on the test conditions, it was decided to inhibit convection in the apparatus. This was done by locating the warm cell above the cold cell. Thus, in each narrow and well insulated cell, there are only two significant modes of heat transfer between the plate and glass surfaces: conductive and radiative.

In order to ensure that the heat transfer coefficients were sufficiently large, gross estimations of their conductive and radiative components were calculated during the design of the apparatus. More exact values are experimentally determined from data taken during the actual tests. For the expected mean temperature in each cell, the thermal conductivity of air is about 0.03 Wm⁻¹K⁻¹ which, for the vertical cell dimensions (20 and 21 mm for the cold and warm cells, respectively), corresponds to a conductive heat transfer coefficient of only about 1 Wm⁻²K⁻¹. The radiative heat transfer in each cell is essentially that between two plane parallel surfaces (the glass of the sample, and the plate) for which the radiative heat transfer coefficient (i.e. the radiative conductance or U-value) can be estimated by equation 3.3a. For the expected surface temperatures, the radiative heat transfer will be too small unless the aluminum faceplate is covered with a high emittance coating. There are special solar lacquers with directional total emittances of about 0.96 which are ideal for this purpose. However, it was decided that ordinary flat black paint, which has an estimated hemispherical total emittance of about 0.87, would suffice. This estimated emittance is based on the normal total emittance for such paint given by Holman, 0.96

to 0.98 [5], and a correlation of about 0.90 between the hemispherical and normal total emittances of dielectric surfaces [6]. Based on a hemispherical total emittance of 0.87 for a painted plate, and 0.84 for that of glass, and also based on a warm side mean surface temperature of about 40 °C, the warm side radiative heat transfer coefficient is calculated to be about 5 Wm⁻²K⁻¹. Therefore, the expected total heat transfer coefficient on the warm side of the glazing is about 6 Wm⁻²K⁻¹. The total heat transfer coefficient on the cold side, calculated as above but with a mean surface temperature of about 15 °C for this side, is about 5 Wm⁻²K⁻¹. These warm and cold side heat transfer coefficients are suitable for the temperature and heat transfer requirements of this experiment. Compared to the heat transfer coefficients for the apparatus used by Fischer-Cripps, these heat transfer coefficients of about 5 and 6 Wm⁻²K⁻¹ are closer to realistic standard test conditions.

The experimental determination of the exact total heat transfer coefficients is described below. It is based on faceplate and sample surface temperatures measured with type K thermocouples which are secured to the surfaces with masking tape. For each plate, thermocouples are set at three locations along the centerline that is parallel to the water flow: near the flow inlet, at the center of the plate, and near the flow outlet. One thermocouple is secured to each side of the sample in the middle of a unit cell of pillars near the center of the glazing. All of the thermocouple wires are connected to a switch box, from which they are connected to a Fluke 8842A digital multimeter for the measurement of the thermocouple voltages. The use of cold junctions to establish the thermocouple reference temperature is not required; this temperature is measured to a sufficient accuracy with a mercury-filled glass bulb thermometer which is placed next to the thermocouple switch box.

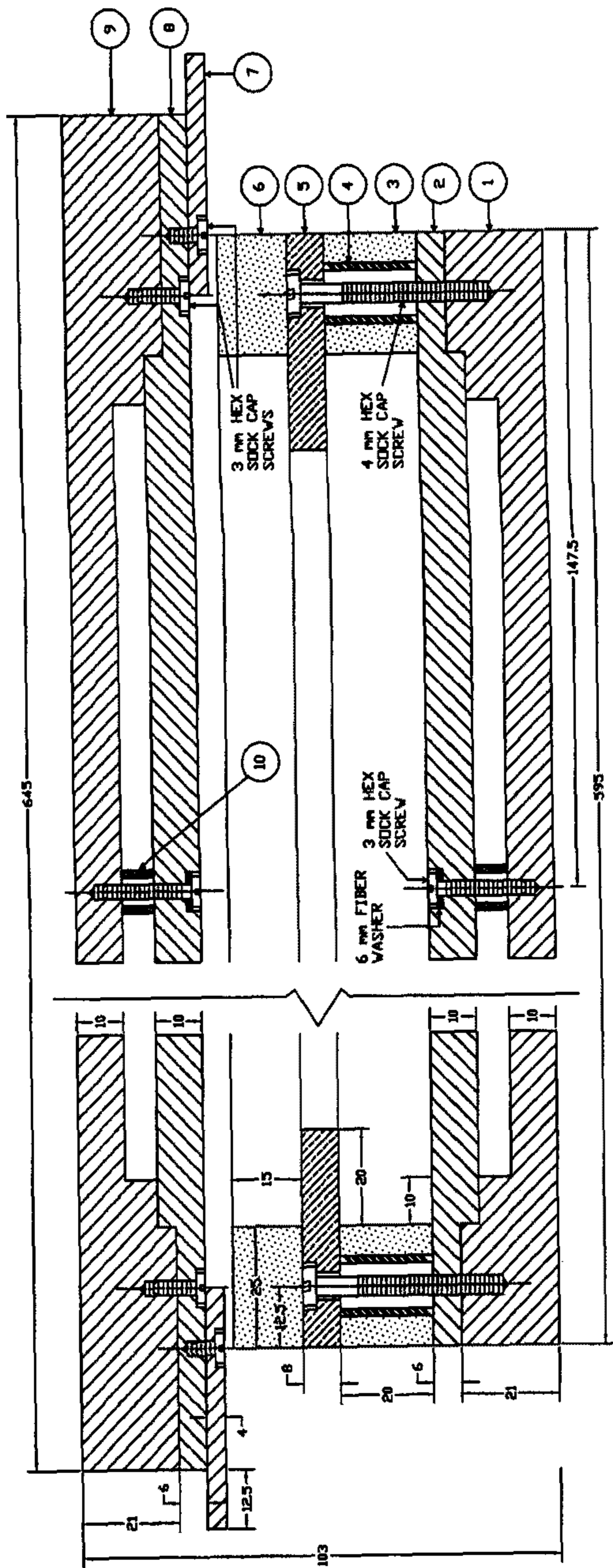
Engineering drawings of the apparatus are presented in Figs. 6.6 and 6.7. Seals between the water jackets and the faceplates are made with silicone sealant around the edges of the plates, and with fiber washers below the heads of the connecting screws.

6.4.2 Deflection Measurements

As mentioned above, the amount of bending in a sample of vacuum glazing can be determined by measuring the local vertical surface deflections with a dial micrometer. Such measurements require a fixed reference plane on which the micrometer can be mounted. In the present apparatus, the reference surface is the top side of an aluminum bar which is placed across and above the warm side surface of the glazing, as shown in Fig. 6.3b. A dial micrometer is inserted sequentially into each of the 32

Fig. 6.6: Engineering drawing of the experimental apparatus, sectional view parallel to the water flows (i.e. the flows are parallel to the page). The warm plates are supported at their bases by rails on a wooden frame (which are not shown). Also not shown are the glazing, glazing supports and micrometer support bar. All dimensions in millimeters. Scale 1:1.3

NO.	PART NAME	MATL	NO.	PART NAME	MATL
1	COLD WATER JACKET	ALUMINUM	6	WARM CELL INSULATION	STYROFOAM
2	COLD FACEPLATE	ALUMINUM	7	WARM PLATE BASE	ALUMINUM
3	COLD CELL INSULATION	STYROFOAM	8	WARM FACEPLATE	ALUMINUM
4	SPACER (TUBE 15 mm OD)	ALUMINUM	9	WARM WATER JACKET	ALUMINUM
5	INTER-CELL DIVIDER	WOOD	10	SPACER (TUBE 6 mm OD)	ALUMINUM



holes along the length of this bar in order to obtain a deflection profile. It is of course necessary to know the exact position of the reference surface with respect to the glass surface being measured. Thus, a special method was devised for supporting the bar.

In order to minimize the uncertainty in the relative position between the bar and the sample, it was decided to mount the bar directly onto the glass surface. The mounting method must preserve the horizontal alignment between the bar and the sample without constraining the bending of the sample. Thus, the bar is supported by the three-point mounting configuration shown in Fig. 6.8. At one end of the bar are two ball bearings, each of which is 16 mm from the deflection measurement transit. One of these bearings rests in the hemispherical cup of a steel disk which is secured to the glass surface with epoxy resin. The other bearing rests in a hemispherical cup of a steel disk which sits freely on the glass surface. At the other end of the bar is a screw with a ball bearing at its tip. This bearing rests in a notched disk which is secured to the glass with epoxy resin. The notch of this disk is perpendicular to the edge of the glazing. This configuration permits only one relative motion between the bar and the glazing; in response to the bending of the sample, the position of the bearing in the notched cup shifts a small and insignificant distance along the notch, thereby shifting the bar by an equivalent distance in the same direction. The height of the bearing tipped screw can be adjusted in order to level the bar.

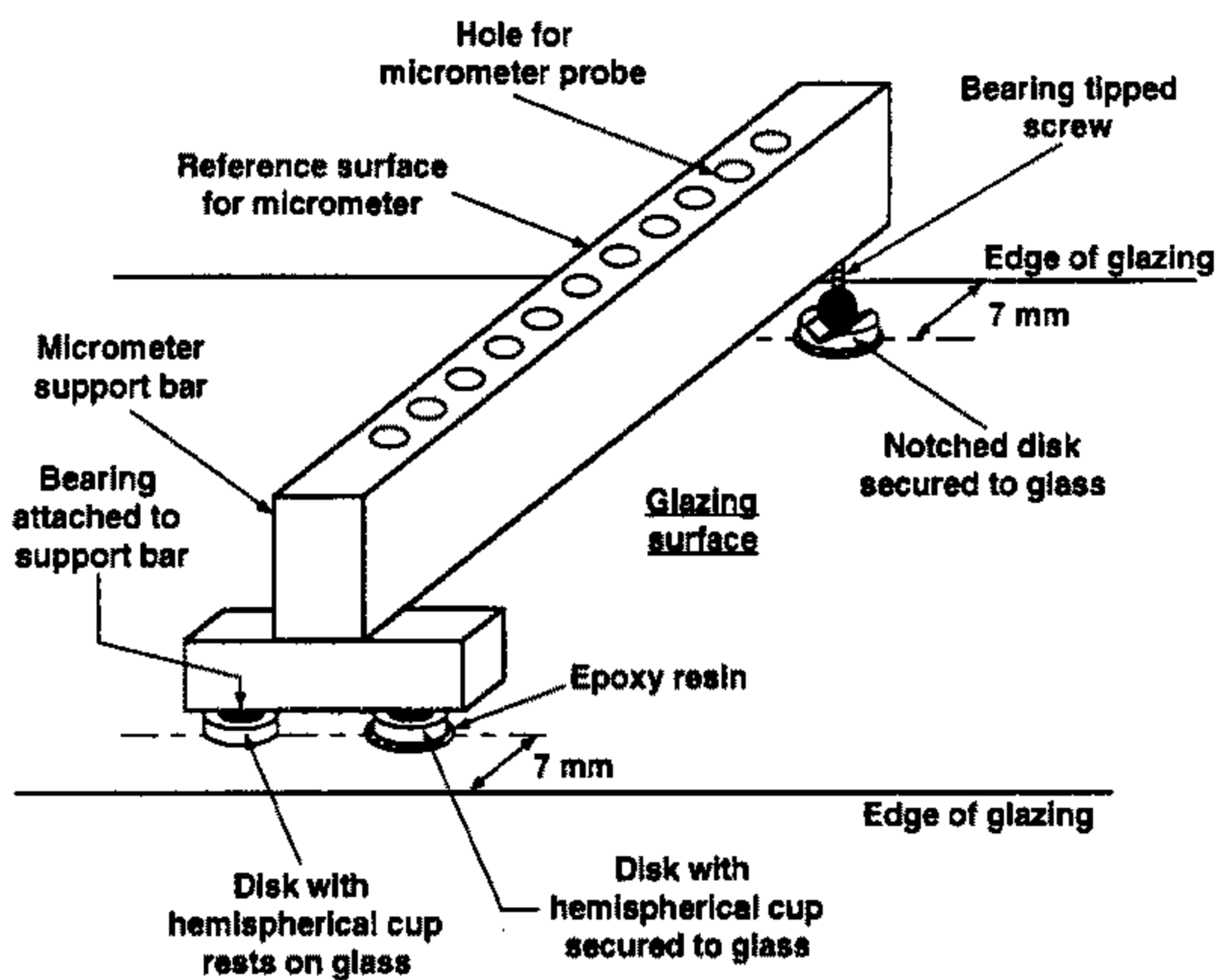


Fig. 6.8: The micrometer support bar is mounted on the glazing at three points, where bearings attached to the bar fit into mounting disks. At one end of the bar are ball bearings resting in the hemispherical cups of fixed and free disks. At the other end is a ball bearing in the notch of a fixed disk. The notch is perpendicular to the edge of the glazing. Only a few of the 32 holes in the bar are shown. The centers of the disks are located at 7 mm from the nearby glazing edges.

The micrometer support bar provides a reference surface for the measurement of the net deflections across the surface of the glazing. These net deflections are relative to the glass below the mounting disks which support the bar. Ideally, in order to obtain total absolute deflections, these mounting disks should be located where there is essentially no deflection of the sample surface: exactly at the edges of the glazing. Due to the finite size of the mounting disks (all three disks are 6 mm in diameter), this is not possible and thus the disks are located with their centers at 7 mm from the nearby edges. As shown in Fig. 6.9, there is a deflection at these positions. This means that the bar will be shifted by the same deflection, and that all measurements will include the effect of this shift or offset. Total deflections with respect to the edge of the glazing can be determined by correcting the measurements for this offset. For an appropriate comparison with the uncorrected measurements, modelling results must be shifted down by an offset equal to the predicted result along the location of the measurement transit at 7 mm from the edges. It is noted that, for the sake of simplicity, the deflections of all three mounting disks - including the two disks that are 16 mm off-axis from the measurement transit - are assumed to be the same.

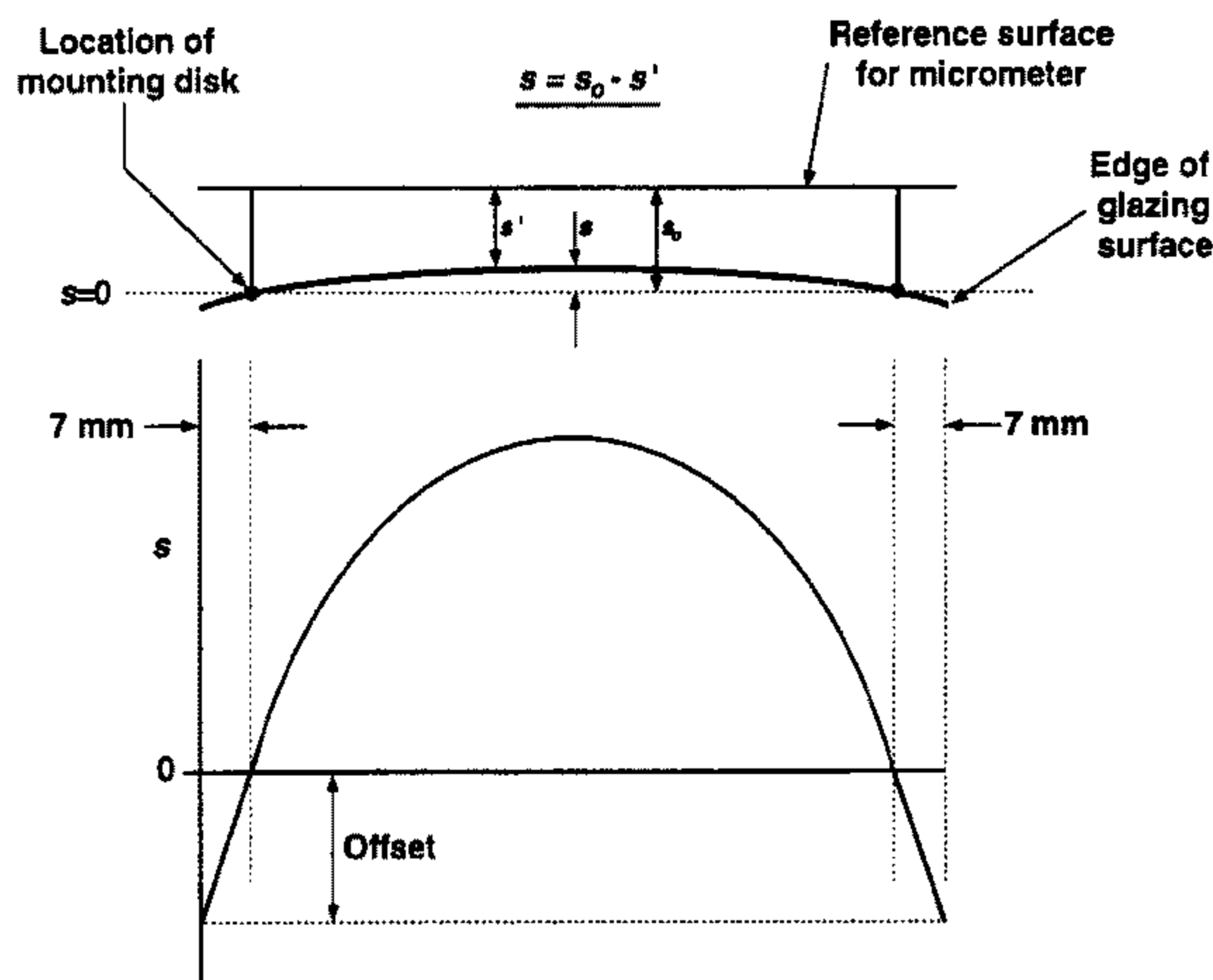


Fig. 6.9: The reference surface for the micrometer (the top of the support bar) is connected to the glazing surface via supports which rest in mounting disks, the centers of which are located at 7 mm from the edges of the glazing. The net deflection, s , is equal to the difference between the null deflection, s_0 , and the deflection under bending, s' . All deflection measurements are with respect to the glass below the disks and thus the net deflections at 7 mm from the edges are zero, and those between the disks and the nearby edges are negative. The total deflections with respect to the edges of the glazing can be determined by correcting the net deflections by the indicated offset.

Mounting disks are set on a 0.5 m by 0.5 m sample of vacuum glazing so that the micrometer support bar can be positioned to obtain deflection profiles along transits at 250, 125 and 20 mm from one edge of the glazing. The deflection measurements were made with a Mitutoyo 2109-10 dial gage micrometer to which was attached a ball bearing tipped probe. This micrometer has a resolution of 1 μm . Repeated measurements of a specific deflection are reproducible to about $\pm 2 \mu\text{m}$.

Since any contact between the micrometer support bar and the adjacent warm plates could constrain the bar and thus affect the measurements, there are uncovered gaps between the bar and plates. Since the width of these gaps is only about 1 mm, there is no significant convective flow between the warm cell and ambient through the gaps. Due to the narrowness of the gaps, the temperature of the bar is only slightly less than that of the warm plates. In order to ensure a fairly uniform radiative heat transfer across the warm cell, the lower surface of the support bar (which faces the sample) is covered with the same paint used for the plates. It is therefore assumed that the presence of the bar results in an insignificant perturbation to the warm side plate-to-glass heat transfer in the vicinity of the bar, and thus to the bending of the glass below the bar.

Due to the dimensional non-uniformities of the micrometer support bar, and due to the fact that the reference surface of the support bar is not exactly parallel to the glass surface, it is necessary to determine the net experimental deflections. This is done on the basis of two sets of measurements: one without a temperature difference across the glazing (the null deflections), and one with such a temperature difference. This difference method also eliminates the effect of the weight of the bar and micrometer on the deflections since this effect is essentially identical for each measurement. The variation in the glass thickness need not be considered because the typical variation in the thickness across the glass sheets used in vacuum glazing ($\pm 0.5 \mu\text{m}$) is less than the error in the deflection measurements.

In response to a change of the warm cell temperature, the micrometer support bar expands or contracts, thus affecting its usefulness as a reference surface. In order to minimize this effect, the temperature of the warm plates is kept constant for the measurements with and without a temperature difference across the glazing. Thus, only the temperature of the cold plate is changed. For the conditions considered in this experiment, the maximum change in the mean warm cell temperature due to adjustments in the temperature of the cold plate is about 3 °C. The consequent

thermal dimensional changes of the bar are insignificantly small: ~ 1 and 40 μm in the vertical and longitudinal dimensions, respectively.

As mentioned above, it was decided to design the apparatus so that it would be possible to establish an external glass-to-glass temperature difference across the sample of about 20 °C. Based on the glass-to-glass center-of-glass U-value of the sample, and based on the above estimations of the heat transfer coefficients in the cells, this requires a plate-to-plate temperature difference of about 30 °C. The constant temperature of the warm plate must be such that this temperature difference can be established with a cold plate temperature above the dew point; if the cold plate is at or below the dew point, the resulting condensation in the cold cell will change the heat transfer in this cell. Also, in order to minimize the heat transfer between the warm plate and ambient, and thus the effects of fluctuations in the air temperature on this heat transfer, the warm plate temperature must not be much greater than room temperature. With both of these considerations in mind, the warm plate temperature was set at about 45 °C.

6.4.3 Strain Measurements

The most straightforward way to measure small surface strains is to use strain gages. As shown in Fig. 6.10, a strain gage typically consists of a continuous resistive wire in a regular switchback grid pattern on a plastic backing. The measurement axis is parallel to the main grid lines. The gage is affixed with adhesive to the surface to be studied. When the surface is under strain (tensile or compressive) the gage, and thus the wire, deforms with the surface. This changes the length of each switchback of wire, thereby changing the total resistance of the gage. The strain is related to the change in the resistance by the gage factor, GF (or K):

$$GF = \frac{\Delta R/R_o}{\epsilon}, \tag{6.5}$$

where ΔR is the change in the gage resistance, R_o is the gage resistance at the reference or null strain, and ϵ is the strain. Most gages are also sensitive to strains that are perpendicular to the measurement axis. This transverse sensitivity is typically so small (only a fraction of a percent of the parallel sensitivity) that it is usually not considered.

For sensitive strain measurements, such as those in this experiment, there are three temperature dependent effects that must be considered. The first effect is the variation

of the gage factor with the temperature of the gage. This necessitates a temperature correction to the factor, which can be done on the basis of information provided by the gage manufacturer. The second effect is that of the gage temperature on the output of the gage - the thermal output. The strain reading can be corrected for this effect by the relationship between the thermal output and the temperature, which is also usually specified by the manufacturer. The third temperature dependent effect relates to changes in the resistances of the gage connecting wires. Changes in the wire temperatures during a test (e.g. due to variations in the ambient temperature to which the wires are partially exposed) cause changes in the resistances of the wires. These resistance changes are in addition to those due to the strain induced resistance changes of the gage, and they can be thought of as representing erroneous apparent strains. For very sensitive measurements, these errors can be as large as the strains which are to be detected. For each gage, it is possible to compensate for these errors, and for those associated with the thermal output, with the use of a second strain gage.

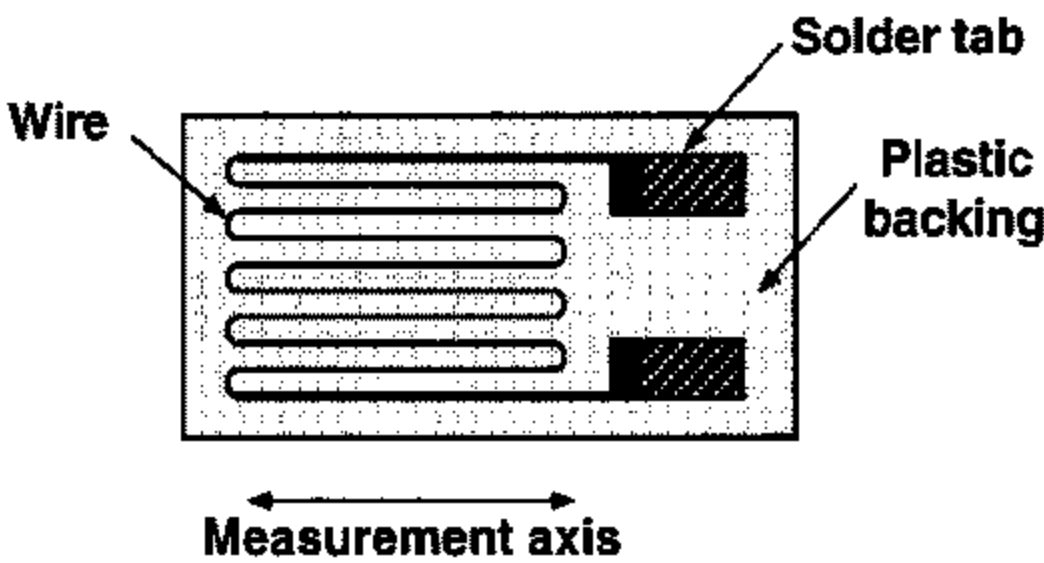


Fig. 6.10: The construction of a typical general purpose strain gage.

If the strains to be detected are very small, as is the case in this study, a greater sensitivity is obtained by connecting the gage into a bridge circuit. The bridge voltage provides an indication of the change in the gage resistance, and thus of the change in the strain. For this experiment, the temperature dependent effects on the strain measurements can be significant. Therefore, a common method of correcting for these effects is employed: a second identical strain gage is connected into each bridge circuit, as shown in Fig. 6.11. The second gage is placed in thermal contact with the surface being studied, but it is not rigidly attached to this surface. The second gage is therefore unaffected by the structural behaviour of the sample. This gage is located next to the measurement gage in order to ensure that the two gages are at approximately the same temperature. The wires from the two gages are equal in length and they are stranded together in order to ensure that they are exposed to the same temperature conditions. Thus, in two arms of the bridge circuit are identical

strain gages and lengths of connecting wire which are subjected to the same temperatures - and to the same changes in temperature. Therefore, the thermal outputs and the temperature induced changes in the wire resistances are the same for these two arms of the bridge. Since these arms are on opposite sides of the bridge, there are no consequent changes in the bridge voltage. Since the second gage compensates for these temperature dependencies, it is usually referred to as a compensating gage, but is it also known as a dummy gage. In the other two arms of the bridge circuit are precision resistors which are enclosed in the strain indicator. In reference to the portion of the bridge which is external to the strain indicator, this arrangement is commonly referred to as a half-bridge circuit.

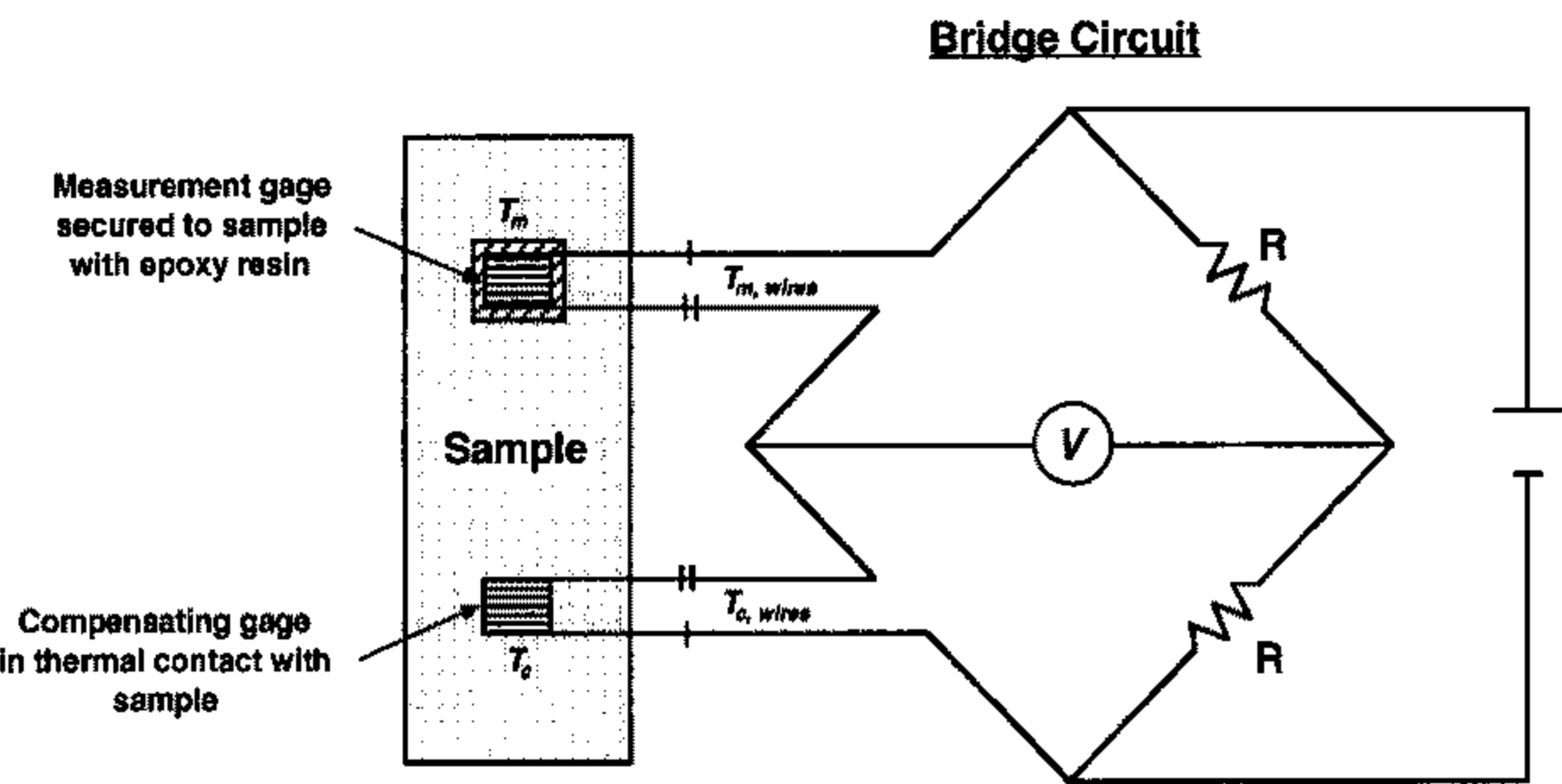


Fig. 6.11: A strain gage half-bridge circuit. The measurement gage is secured to the sample, and the compensating gage is placed in good thermal contact with the sample. The two gages are placed near each other so that their two temperatures, T_m and T_c , are approximately equal. The wires from the gages are equal in length, as indicated, and they are stranded together in order to ensure that the wire temperatures, $T_{m,wires}$ and $T_{c,wires}$, are equal.

The measurement gages are affixed to the glass surface of the sample with a special epoxy resin (AE-10). The compensating gages are similarly secured to small sections of glass microscope slides which are loosely taped to the sample about 50 mm from, and at the same distance from the edge as, the respective measurement gages. As mentioned above, this positioning is to ensure that the measurement and compensating gages are at approximately the same temperature. Thus, the compensating gages are in thermal contact with the sample via the microscope slide sections. It is assumed that the glass of the microscope slides has the same coefficient of thermal expansion as that of the sample. Therefore, in response to changes in the test temperatures, the slide sections and the sample experience the same local thermal

expansions or contractions, and thus the same consequent strains. These strains are detected by the measurement and compensating gages. However, as with the temperature dependent effects referred to above, these equal strains will not result in changes to the bridge voltage. Therefore, the only significant strain that is detected with the bridge circuit is that which is associated with the stress due to the temperature induced bending of the glazing (i.e. the strain which is given by the first term of equation 6.4)

For sensitive strain measurements, the signal to noise ratio can be improved by supplying a high voltage to the bridge circuit. However, unless the gage has a high resistance, the ohmic power dissipated in the gage can significantly alter the local temperature and thus the gage factor. It was decided to use a bridge input voltage of 15 V, for which an appropriate gage resistance is 1 k Ω .

All of the strain gages and the associated instrumentation used in this experiment were manufactured by Micro-Measurements Division of Raleigh, North Carolina. The type of strain gage used was CEA-06-250UW-10C. This is a general purpose gage made of constantan wire and a polyimide backing. According to a data sheet supplied by the manufacturer [7], at a reference temperature of 24 °C, the gage resistance is 1 k $\Omega \pm 0.3$ %, and the gage factor is 2.110 ± 0.5 %. With respect to the reference temperature, the gage factor changes by $(+ 1.2 \pm 0.2)$ % per 100 °C. The gage transverse sensitivity, $(+ 0.1 \pm 0.2)$ % of the strain along the measurement axis, is negligible for this experiment and thus it is not considered. The size of the measurement area (the grid of resistive wire) is 6.35 mm by 11.43 mm.

Figure 6.12 shows a schematic diagram of the circuit for the strain gages and instrumentation. Each measurement gage is connected into a half-bridge circuit with a compensating gage. Thin leadwires are attached to the copper tabs of the gages with a low melting point solder. Connections are made with stranded wires to a ten-channel switch and balance unit (SB 10), and from there to a strain indicator (model 3800). The switch and balance unit contains potentiometers for balancing the bridge circuits; there is a separate potentiometer for each circuit. The strain indicator converts the measured bridge voltage to a strain on the basis of the programmed gage factor.

Each of the ten channels of the switch and balance unit consists of a half-bridge circuit with a measurement gage and a compensating gage. Thus, the strain at only ten locations on the sample can be measured. The model can be sufficiently validated on the basis of results for only one side of the glazing. Since it is important to minimize

any strain gage temperature dependencies that might not be fully corrected for by the compensating gages, all of the gages are located on the side of the glazing which is exposed to the smallest changes in the mean cell temperature: the warm side (the mean cell temperature on this side changes less due to the constant warm plate temperature). Due the symmetry of the square sample, it is not necessary to measure the entire warm side surface. Figure 6.13 shows the locations of the gages, as well as the positions of the deflection measurement transits. Two rows of measurement gages are located 25 mm from, and on either side of, one half of a centerline. The gages of one row are oriented to measure strains parallel to the edge of the glazing, and the gages of the other row are positioned to measure strains normal to the edge of the glazing. This arrangement provides symmetric measurements of parallel and normal strains that should be close to the centerline values. Since preliminary modelling studies showed that the greatest changes in the parallel and normal strains occur close to the edge of the glazing, most of the gages are located near the edge. The same studies showed that the parallel strains are greater in magnitude (and thus, as explained in Chapter 8, more significant in the context of the probabilities of failure) than the normal strains, and that the normal strains are more constant after a shorter distance from the edge. Therefore, more gages are oriented to measure the parallel strain. A gage is also located near one corner of the glazing. The strains and deflections were measured in different regions of the sample in order to minimize the interaction between the different measurements. With a temperature difference across the sample, the strain readings are reproducible to about $\pm 2 \mu\epsilon$.

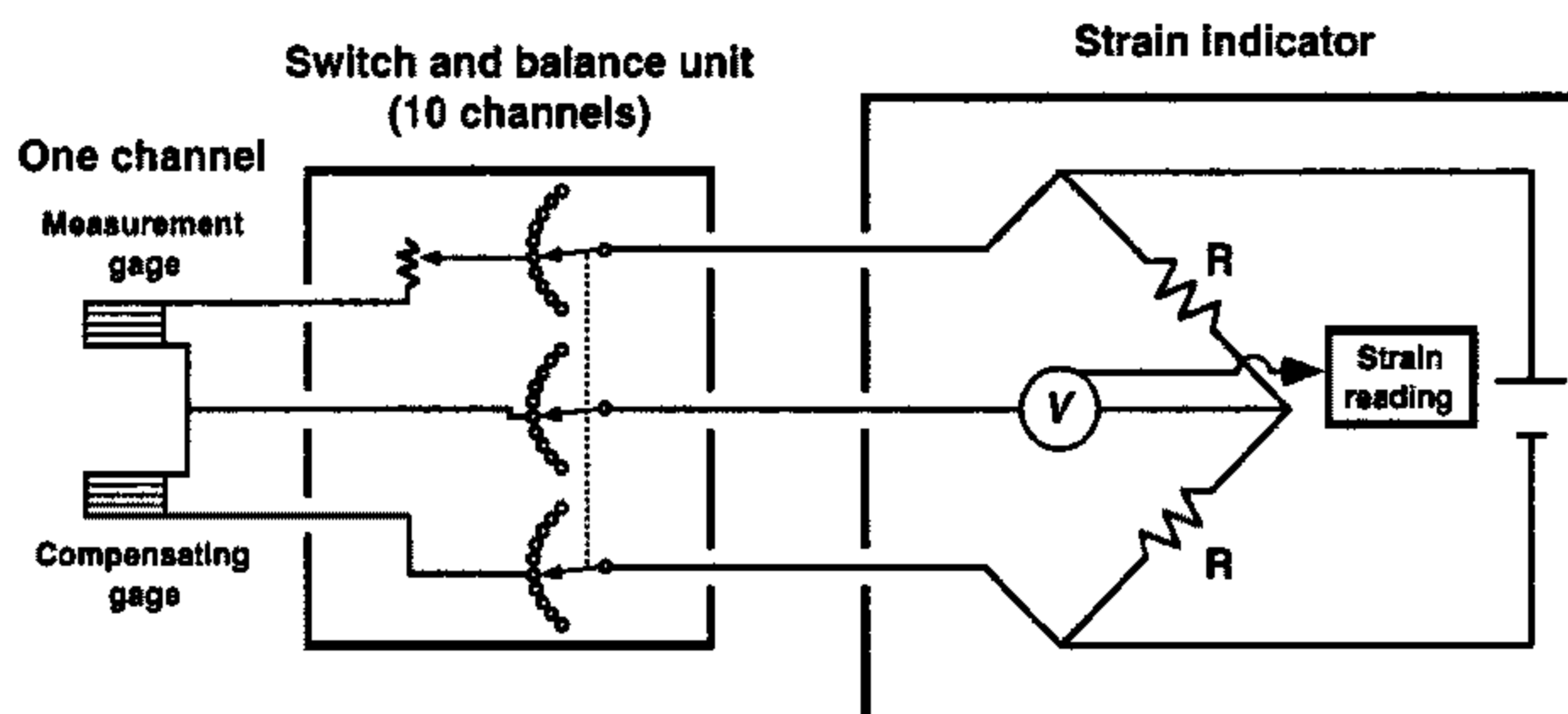


Fig. 6.12: A schematic diagram of the strain gage measurement instrumentation. The gages are connected into the bridge circuit via a ten-channel switch and balance unit. Only one channel, which consists of a measurement gage and a compensating gage, is shown in this figure. There is a bridge balancing potentiometer for the circuit associated with each channel. The bridge voltage is measured by the strain indicator and, based on the programmed gage factor, this voltage is converted to a strain reading.

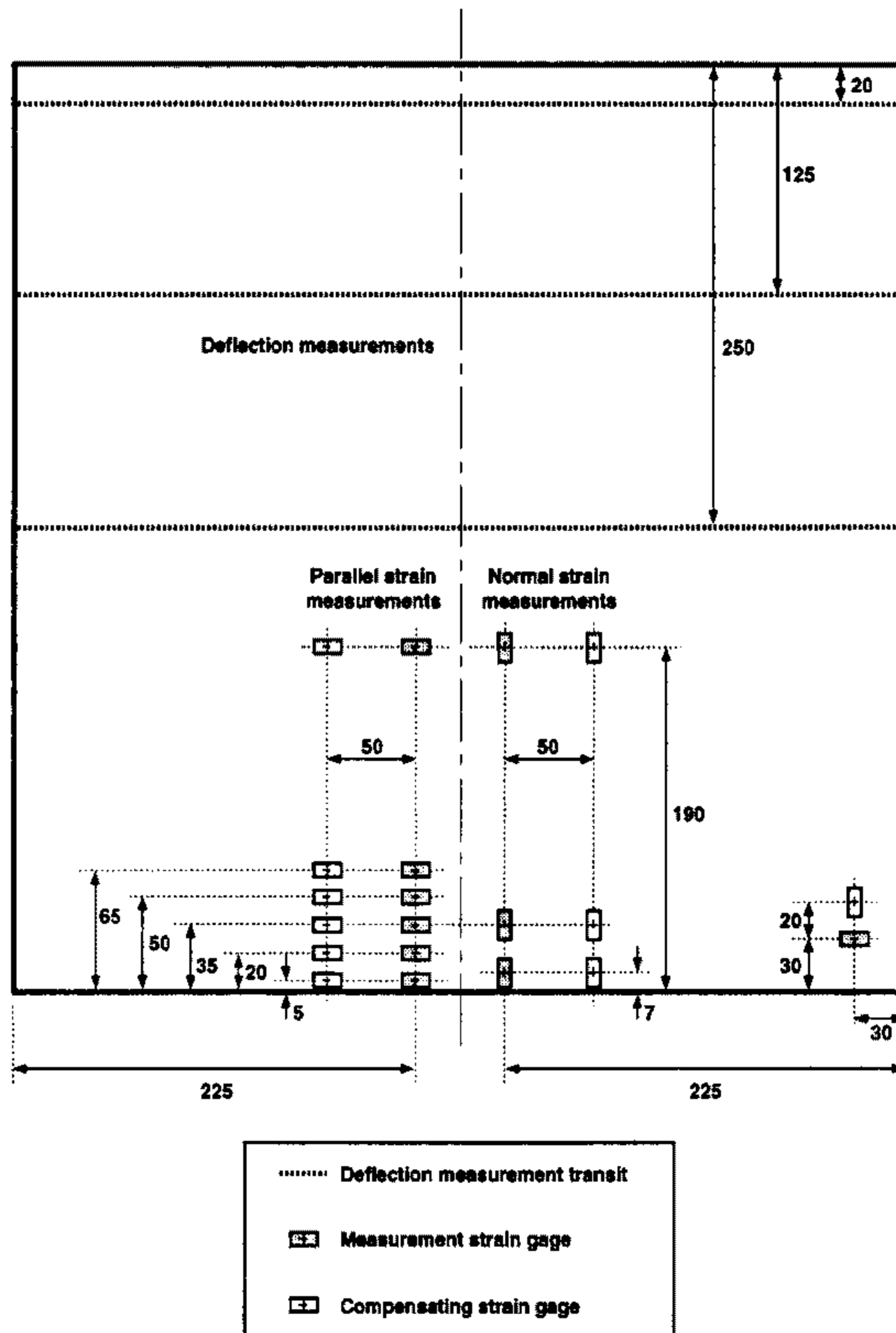


Fig. 6.13: Locations of the deflection measurement transits and the centers of the strain gages for the 500 mm by 500 mm sample of vacuum glazing. The positions of the mounting disks for the micrometer support bar are such that the bar can be located along any of the transits. The centers of the rows of gages which are separated by 50 mm are located at positions which are approximately between pillar rows. All dimensions are in millimeters.

The strain results are assumed to correspond to the locations at the centers of the gages. However, due to the finite gage size, each result actually reflects the average or effective strain over the entire measurement area of the gage. Thus, there is a spatial averaging associated with each measurement. The spatial resolution depends on the dimensions of the gage, and on the orientation of the gage with respect to the main strain gradient. For the strains parallel to the edge, the spatial resolution is ± 3 mm and, for the strains normal to the edge, this resolution is ± 6 mm.

As explained in the next chapter, atmospheric pressure leads to significant stresses near the edges and pillars. However, these stresses - and the associated strains - are not dependent on the temperature difference across the glazing, and they are therefore essentially constant. Thus, the atmospheric pressure induced strains are factored out in the calculation of the net experimental temperature induced bending strains (this calculation is described below). For the same reason, the external glass surface deflections due to atmospheric pressure (which are discussed in the next chapter) do not affect the net experimental deflection results.

6.4.4 Edge Insulation

In order to investigate the effects of edge insulation on the deflection and strain measurements, it was decided to perform some of the tests with insulated edges. As explained above, most of the heat transfer between the plate and glazing surfaces is radiative. Therefore, the most effective and practical way of insulating the edges of the sample is to cover the edges with a radiation shield - i.e. to insert low emittance surfaces between the plate and glass surfaces. This was the approach that was employed in this study. As shown in Fig. 6.14, the shield (which consists of cardboard with aluminum foil glued to both sides) is supported about 15 mm above the glass surface by pins which rest on the glazing. This simulates the effect of conventional edge insulation (e.g. strips of foam around the edges). With this arrangement, there is no contact between the shield and the strain gages or the connecting wires. Since it is difficult to access the cold side of the glazing once the glazing is in set in place, only the edges on the warm side are insulated in this manner.

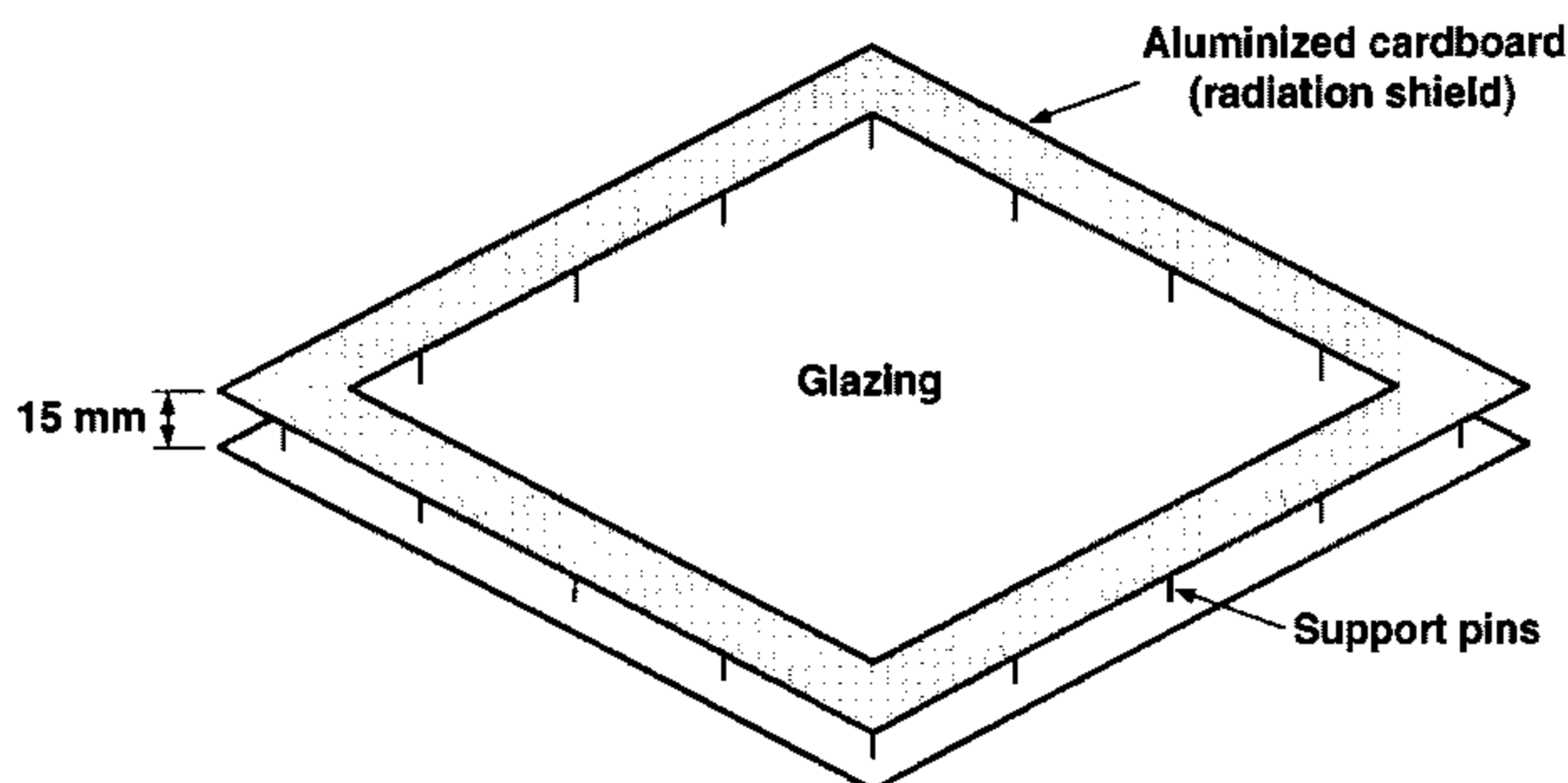


Fig. 6.14: Warm side edge insulation is simulated by a cutout sheet of aluminized cardboard supported 15 mm above the glass surface by pins which rest on the glazing.

6.4.5 Glazing Sample

One 0.5 m by 0.5 m sample of vacuum glazing (sample 467) was used for all of the deflection and strain measurements. It is made of two sheets of 4 mm thick K glass, and it contains ceramic pillars of square cross-section. The pillars are of dimension ~ 0.20 mm and of height ~ 0.1 mm, and they are in a regular array with a pillar separation of 25 mm. The width of the edge seal is about 8 mm. The thermal conductance apparatus described in Chapter 2 was used to evaluate the thermal performance of the sample; the radiative U-value was determined to be 0.77 ± 0.01 $\text{Wm}^{-2}\text{K}^{-1}$ for internal glass surface temperatures of 23.1 and 3.3 °C, and the pillar array U-value was determined to be 0.41 ± 0.01 $\text{Wm}^{-2}\text{K}^{-1}$.

6.5 Measurement Procedure

A thermocouple was secured with masking tape to each side of the sample in the middle of a unit cell of pillars near the center of the glazing. The sample was then placed on the ball bearing tipped support screws. The cold cell insulation and the inter-cell wooden divider were set in place with screws. The narrow gap between the sample and the surrounding divider was covered with masking tape; care was taken to ensure that there was no direct contact between the divider and the sample. The warm cell insulation was then placed on the divider and taped in place. The micrometer support bar was set in the appropriate mounting disks, and the two warm plates were brought together on opposite sides of the bar. A gap of about 1 mm was left between the bar and each warm plate. The wires from the strain gages were then connected to the switch and balance unit.

The two water baths were turned on and the temperature for each bath was set at 45 °C. When the cell temperatures had stabilized, the temperature of the cold side bath was adjusted until the warm and cold side glass surface temperatures were within 0.1 °C of each other (these temperatures were determined from the thermocouple readings in the manner described below). This indicated that there was almost no heat flow across the sample - i.e. that a condition of thermal equilibrium had been established. With no temperature difference across the sample, there was no temperature induced bending and thus no associated deflections or surface strains. Therefore, at the equilibrium condition, the null deflections were measured and each strain gage bridge circuit was balanced.

Before beginning the null deflection measurements, the dial micrometer was inserted into a hole near the center of the support bar for five minutes to bring it to approximately the same temperature as the warm cell. This was done in order to ensure that any significant thermal expansions of the micrometer springs, gears and probe occurred before - instead of during - the measurements. The micrometer was then inserted sequentially into each of the holes along the bar. For each measurement, the micrometer was lightly tapped into place and an average of three readings was taken. The measurements were reproducible to about $\pm 2 \mu\text{m}$.

With the zero temperature difference across the sample, three initial adjustments were made to the strain measurement instrumentation. Firstly, the amplifier output of the strain indicator was zeroed. Secondly, the gage factor reading on the indicator was set to an appropriate value for all of the gages. As mentioned above, the gage factor for each gage depends on the gage temperature, which itself depends on the test conditions and the position of the gage with respect to the edge. The above mentioned finite element model for the thermal analysis of a glazing quarter section was used to determine the expected glass temperatures at the gage locations on the basis of the estimated values for the intended test conditions. For the tests without edge insulation, these temperatures are between 30 and 43 °C, for which the corresponding gage factors are 2.111 and 2.113. For the tests with edge insulation, the temperature and gage factor ranges are 24 to 40 °C, and 2.110 to 2.112. The variation in the gage factor is so small that sufficient accuracy can be obtained by using the gage factor which corresponds to most of the measurements: 2.112. Thus, this was the value that was set on the strain indicator. Later modelling results which were based on the actual test conditions confirmed that this was an appropriate gage factor. Lastly, each half-bridge was zeroed as follows: the appropriate channel on the switch and balance unit was selected and then the corresponding potentiometer was adjusted until the strain reading on the indicator was close to zero. The strain reading was allowed to stabilize for five minutes and then it was recorded to an accuracy of $\pm 1 \mu\epsilon$.

Having thus established the null deflections and zeroed the half-bridge circuits, the temperature of the water flow for the cold plate was reduced until the desired external glass-to-glass temperature difference across the sample, as determined from the thermocouple readings, was attained. The glass bulb thermometer temperature and all of the thermocouple voltages were then recorded. Complete sets of deflection and strain measurements were made concurrently for external glass-to-glass temperature differences of 7.1, 11.3, 16.9 and 22.0 °C. Additional measurements were made near

the center and the edge of the glazing for external glass-to-glass temperature differences of 8.9, 14.6 and 18.6 °C.

Measurements of deflections with a temperature difference across the sample were made in the same manner as that described above. Net deflections were calculated by subtracting these readings from the null deflections. The accuracy for the net deflections is about $\pm 4 \mu\text{m}$.

Each strain measurement was made by selecting the appropriate channel, waiting five minutes for the strain reading to stabilize, and then recording the indicated strain. The net strain result is equal to the difference between this final measurement and the zero strain reading. Based on the fluctuations in the final and zero strain readings, the net results are accurate to about $\pm 3 \mu\epsilon$. All of the strain and deflection measurements presented in the next section are net results.

Before commencing the measurements with insulated edges, a test was performed to determine the thermal conductance for the heat flow between the warm side plate and glass surfaces via the radiation shield. The entire warm side of the glazing was covered with a sheet of the aluminized cardboard in the manner described above, and a plate-to-plate temperature difference of 31.2 °C was established. The plate-to-glazing thermal conductance across the radiation shield was calculated on the basis of the warm side plate and glass surface temperatures and the heat flow across the sample - which were determined in the manner described below. The same sheet of aluminized cardboard was then cut, after each test, to the required dimensions to insulate the edges of the glazing for three distances measured from the outside edge of the 8 mm wide seal: 45, 35, and 25 mm. Using the convention introduced in Chapter 5 for the insulation recess depth, which is measured from the inside edge of the seal, these widths correspond to recess depths of 37, 27, and 17 mm. Holes were cut in the shield for the micrometer support bar mountings and for the micrometer probe.

For most of the tests, the glass sheet temperatures at the initial equilibrium (i.e. null) conditions were about 44.6 °C. Therefore, for the comparative structural modelling work, the reference temperature for the calculation of the thermal expansion (or in this case, contraction) of the glass was set to 44.6 °C.

6.6 Results

6.6.1 Calculation of Test Conditions

In order to obtain modelling results for comparison with the experimental data, the following conditions for each test must be known: the plate and glass surface temperatures, the glass-to-glass center-of-glass U-value of the sample (U_{g-g}), and the heat transfer coefficients.

The plate and glass surface temperatures are determined from the thermocouple voltage readings. For type K thermocouples, and for the range of temperatures of interest in this experiment, the voltage-to-temperature conversion factor is 2.5×10^4 °CV⁻¹ [8]. This gives a temperature relative to that where the thermocouple wires are connected to conventional wires - at the switch box. The temperature at this location is measured with the glass bulb thermometer.

The pillar array and radiative components of U_{g-g} were given above as 0.41 ± 0.01 Wm⁻²K⁻¹ and 0.77 ± 0.01 Wm⁻²K⁻¹, respectively. The pillar array U-value is independent of temperature, but the radiative U-value depends on the internal glass surface temperatures. The above radiative U-value, U_{rad} , was measured for internal surface temperatures of 23.1 and 3.3 °C (for which the average temperature is 286.4 K). For other internal surface temperatures, this U-value must be corrected on the basis of equation 3.3a as follows:

$$U_{rad} = 0.77 \frac{T_{ave}^3}{(286.4)^3}, \quad (6.6)$$

where U_{rad} is in Wm⁻²K⁻¹, and T_{ave} is the average of the internal glass surface temperatures, in Kelvin, for the test being considered. The temperatures at the internal glass surfaces differ from those measured at the external glass surfaces by an amount which depends on the heat flow across the sample - which itself depends on the radiative U-value. A simple analytic estimation shows that, for the different sets of conditions of this experiment, the center-of-glass temperature difference across the thickness of each glass sheet varies between 0.04 and 0.11 °C. However, for all of the tests, it is sufficiently accurate to assume that the internal surface temperatures differ from the measured external temperatures by 0.1 °C. The glass-to-glass center-of-glass U-value for the sample is the sum of the pillar array U-value and the temperature corrected radiative U-value. The center-of-glass heat flow per unit area across the sample is approximately equal to the product of U_{g-g} and the difference between the

internal glass surface temperatures. This approach ignores the non-uniformity in U_{g-g} due to the heat conduction through the individual pillars. However, as explained in Chapters 4 and 5, the actual heat flows and average temperatures can be estimated to a very good accuracy by the use of a uniform U-value.

The heat transfer coefficients are calculated from the heat flow across the sample, as determined above, and from the differences between the respective faceplate and external glass surface temperatures. There are of course temperature variations across each plate ($\sim 0.1\text{ }^{\circ}\text{C}$) and between the same locations on the two warm plates ($\sim 0.1\text{ }^{\circ}\text{C}$). The surface temperature of the cold plate is taken to be equal to that measured at the center of the plate, and that of the two warm plates is taken to be equal to the average of the measured central temperatures of the warm plates. The small absolute uncertainties in the plate temperatures can lead to large errors in the heat transfer coefficients. This is because the coefficients are based on small differences between the plate and glazing surface temperatures. The errors in the coefficients are less for larger heat flows across the glazing. However, in all cases, the accuracy to which these coefficients are determined is adequate to permit a sensitive comparison between the experimental and modelling results for the deflections and strains.

Deflection and strain measurements were made for the external glass-to-glass temperature differences, ΔT_{g-g} , which are listed in Table 6.2. The corresponding glass-to-glass center-of-glass U-values, plate surface temperatures and heat transfer coefficients are also presented in this table. The errors for the heat transfer coefficients were evaluated on the basis of the variations in the measured values of the plate surface temperatures, and on the basis of the error associated with the value of U_{g-g} for each test. As expected, the warm and cold side heat transfer coefficients are close to the respective estimated values of 6 and 5 $\text{Wm}^{-2}\text{K}^{-1}$ given in Section 6.4.1, and the cold side transfer coefficient decreases with lower cold side temperatures.

It is noted that the internal glass surface temperatures, the glass-to-glass center-of-glass U-values and the heat transfer coefficients can also be calculated by a more exact iterative approach. This approach is based on the measured temperatures for the plate and the external glass surfaces, and on an initial assumption for the radiative U-value. To within the errors given in Table 6.2, this approach yields the same results as those obtained by the simpler method given above.

The errors for the finite element modelling results are associated with the uncertainties in the test conditions. These errors can be quantified in the same manner as was done for the modelling results obtained for the comparison with the infrared thermography data of Figs. 5.16 and 5.17 of Chapter 5: by running the model for the appropriate upper and lower limits of the test conditions which maximize or minimize the results. However, these errors are not considered in this study. This is because the errors are most strongly dependent on the uncertainty in the external glass-to-glass temperature difference; since this uncertainty is so small (as shown in Table 6.2), the resulting errors are negligible in comparison with the experimental errors.

ΔT_{g-g} ($\pm 0.1\text{ }^{\circ}\text{C}$)	U_{g-g} ($\pm 0.02\text{ Wm}^{-2}\text{K}^{-1}$)	Plate surface temperatures ($\pm 0.1\text{ }^{\circ}\text{C}$)		Heat transfer coefficients ($\text{Wm}^{-2}\text{K}^{-1}$)	
		Warm side	Cold side	Warm side	Cold side
7.1	1.41	44.6	34.7	6.8 ± 0.5	6.9 ± 0.3
8.9	1.40	44.6	32.1	6.7 ± 0.2	6.7 ± 0.3
11.3	1.38	44.6	28.7	6.7 ± 0.2	6.5 ± 0.2
14.6	1.36	44.6	23.9	6.7 ± 0.2	6.3 ± 0.3
16.9	1.35	44.6	20.7	6.8 ± 0.2	6.1 ± 0.2
18.6	1.34	44.6	18.2	6.7 ± 0.2	6.0 ± 0.3
22.0	1.32	44.5	13.2	6.7 ± 0.2	5.7 ± 0.2

Table 6.2: The test conditions.

6.6.2 Deflection Results

Figure 6.15 presents experimental and modelling results for the warm side surface deflections along the centerline of the glazing. Data are given for four glass-to-glass temperature differences across the sample, and for the corresponding conditions listed in Table 6.2. The modelling results of this figure and, except where otherwise stated, all other modelled deflections in this chapter are presented so that the predicted deflections are zero at the locations of the centers of the mounting disks for the micrometer support bar (7 mm from the edges). The rationale for this is explained above in Section 6.4.2. As a result of the offset shifts, the deflections between these positions of the disk centers and the nearby edges of the glazing are negative. The model only provides results for along one half of the centerline (i.e. from 0 to

250 mm); the data presented for the other half of the centerline are, by symmetry, simply the mirror image of the modelling results. In general, there is a good agreement between the experimental and modelling results; for each temperature difference, most of the results agree to within the experimental error of $\pm 4 \mu\text{m}$.

Figure 6.16 shows experimental and modelling results from Fig. 6.15, and from measurements taken at other temperature differences, for the deflection at the center of the glazing as a function of ΔT_{g-g^*} . Since there was no micrometer hole at the center of the measurement transit, the experimental results were determined from a quadratic fit to the deflections measured near to, and on both sides of, the center. To within the experimental error of $\pm 4 \mu\text{m}$, most of the measurements agree with the predicted deflections. Both the experimental and modelling results show that, with respect to points 7 mm from the mid-edges, the center deflection is proportional to the temperature difference across the glazing. This is consistent with the analytic model (equation 6.1) which relates the deflection between the middle of one edge and the center of a square glazing to ΔT_{g-g^*} .

Fig. 6.17 presents results for the edge-to-center deflection obtained from equation 6.1, and from the finite element model. Unlike the results of the previous two figures, these deflections are referenced to the edges (instead of to points 7 mm from the edges) and are therefore total absolute deflections. The results of this figure indicate that the analytic model provides only approximate results for the center-of-glass deflection; in fact, it consistently overestimates these deflections. This is because, as explained above in Section 6.2, the model does not account for the effects of the edge seal. However, the more exact finite element modelling results indicate that the analytic model correctly predicts a linear relationship between the edge-to-center deflection and ΔT_{g-g^*} .

Figure 6.18 presents experimental and modelling results for deflections along transits at 20 and 125 mm from the edge of the glazing. The results are for $\Delta T_{g-g^*}=22.1^\circ\text{C}$, and for conditions which agree with those given in Table 6.2 for the 22.0°C temperature difference. Thus, these data can be compared with those given in Fig. 6.15 for deflections along the centerline (250 mm from the edge) for $\Delta T_{g-g^*}=22.0^\circ\text{C}$. As expected, there is less bending closer to the edge. The experimental and modelling results at 250 mm from the edge are in very close agreement with the corresponding results at 125 mm from edge - so close that, for the sake of clarity, the centerline results are not presented in Fig. 6.18. The maximum experimental deflection at the centerline is only about $3 \mu\text{m}$ greater than that at 125 mm. However,

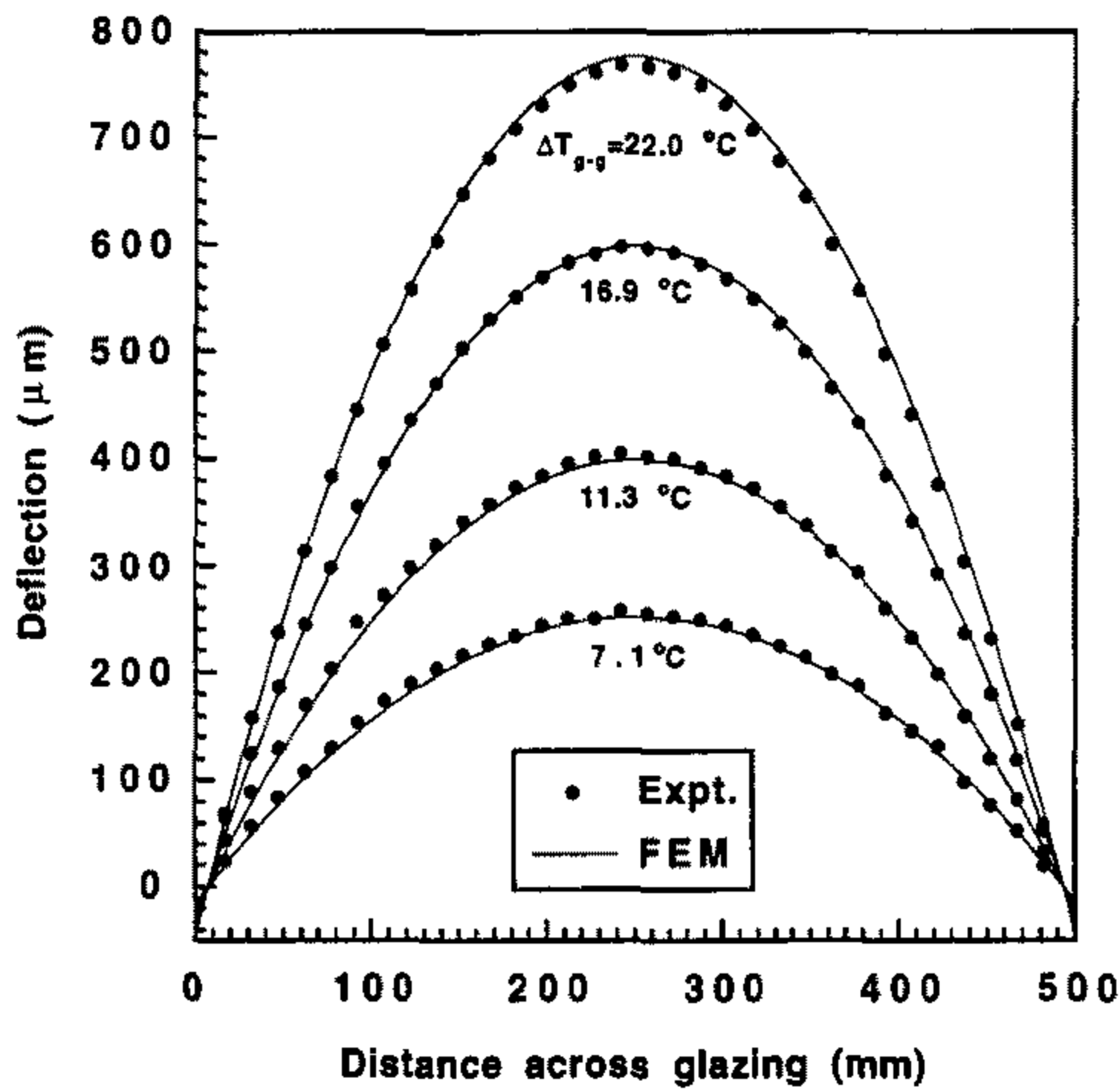


Fig. 6.15: Warm side external surface deflections along the glazing centerline (250 mm from the edge) for four glass-to-glass temperature differences across the glazing (ΔT_{g-g}). The corresponding conditions are given in Table 6.2. The finite element modelling results (FEM) are presented so that the deflections are zero at the same locations as the reference points for the experimental results (Expt.). The experimental results are accurate to $\pm 4 \mu m$.

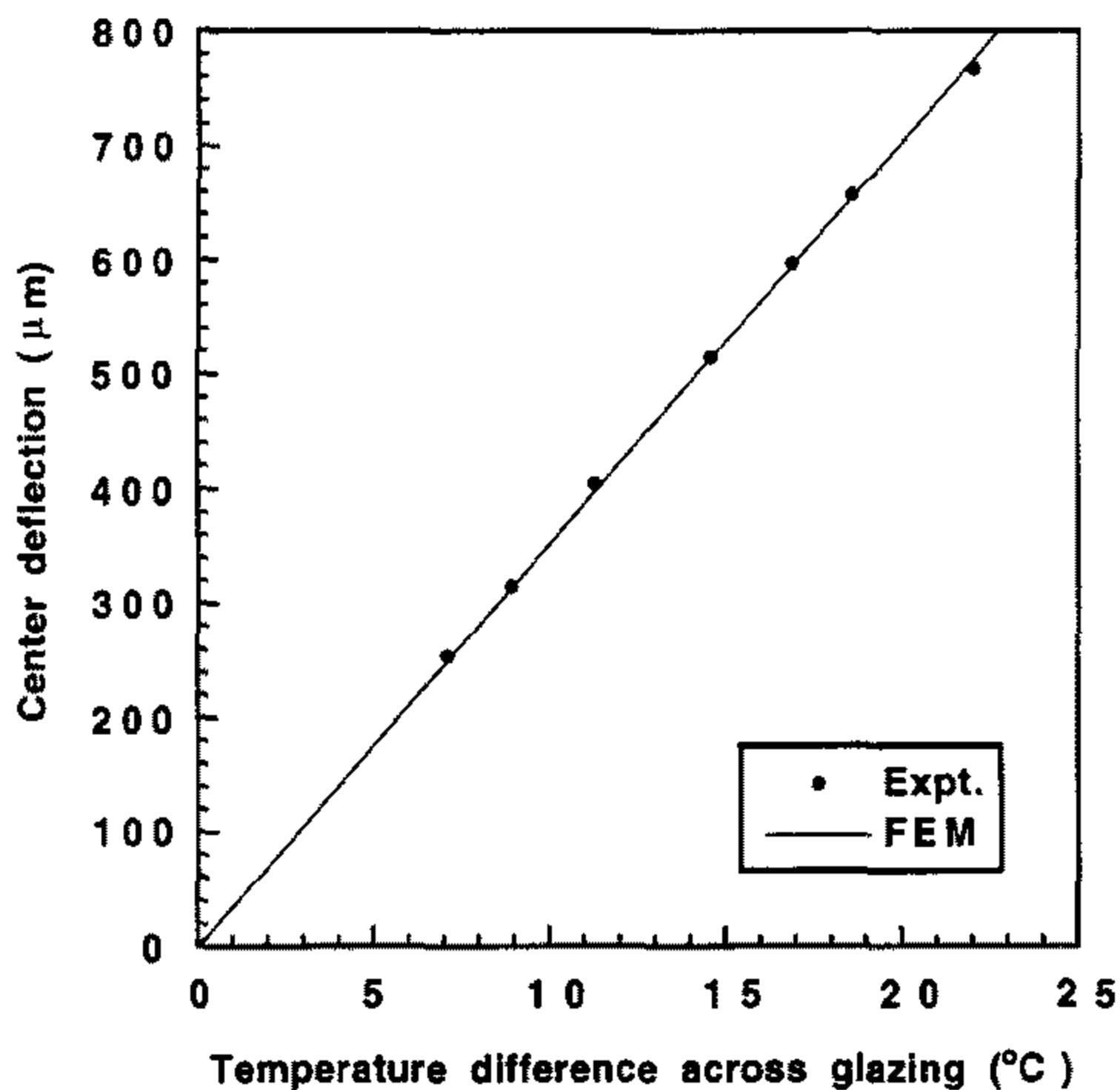


Fig. 6.16: Experimental and finite element modelling results (Expt. and FEM) for the warm side center-of-glass deflection as a function of ΔT_{g-g} . The corresponding conditions are given in Table 6.2. The experimental results are accurate to $\pm 4 \mu m$.

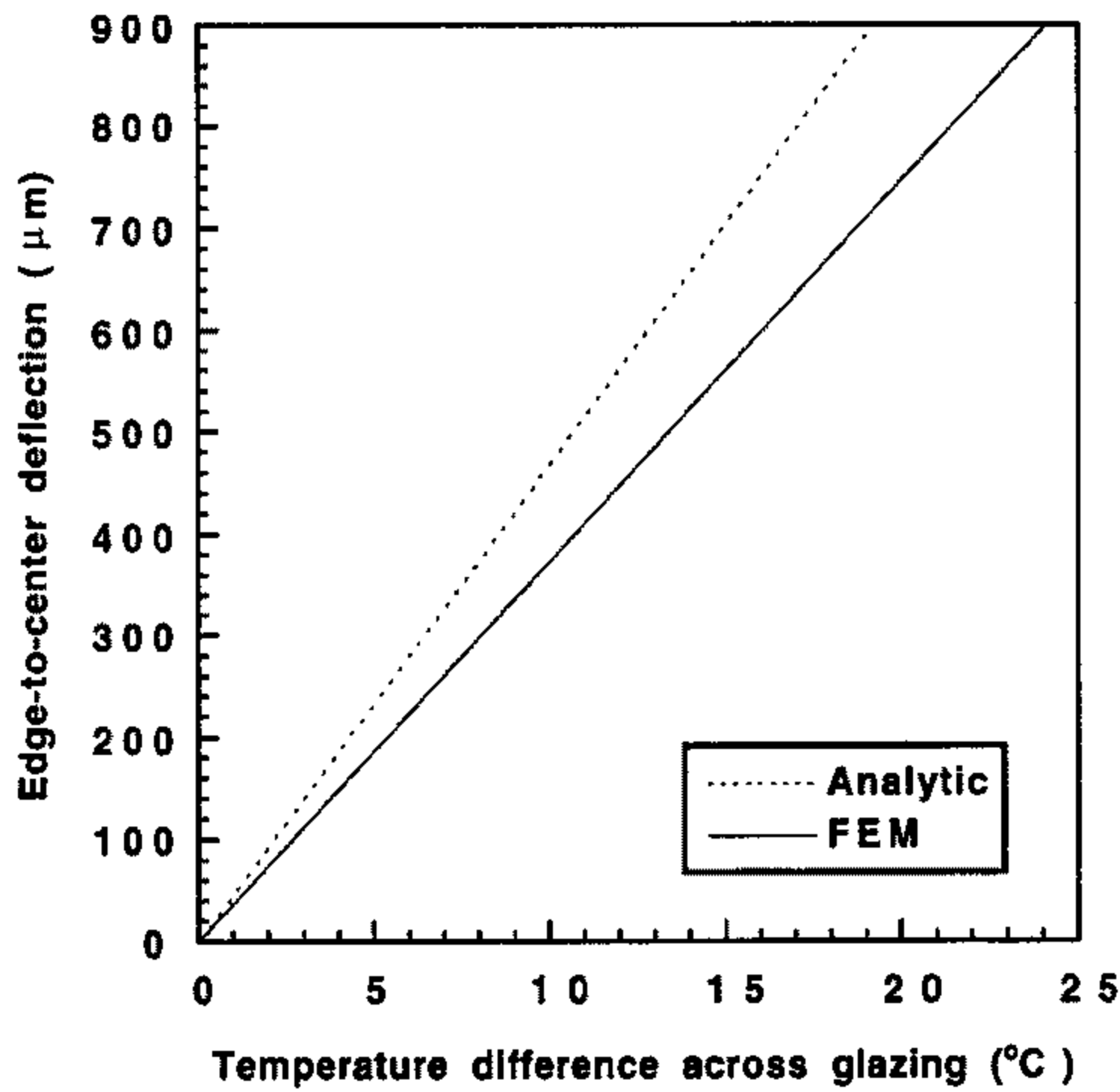


Fig. 6.17: Warm side external surface deflections from the middle of an edge to the center of a glazing as a function of ΔT_{g-g} , as calculated by the analytic model of equation 6.1 (Analytic) and by the finite element model (FEM) for a vacuum glazing of size 500 mm by 500 mm. The results were obtained for the conditions given in Table 6.2.

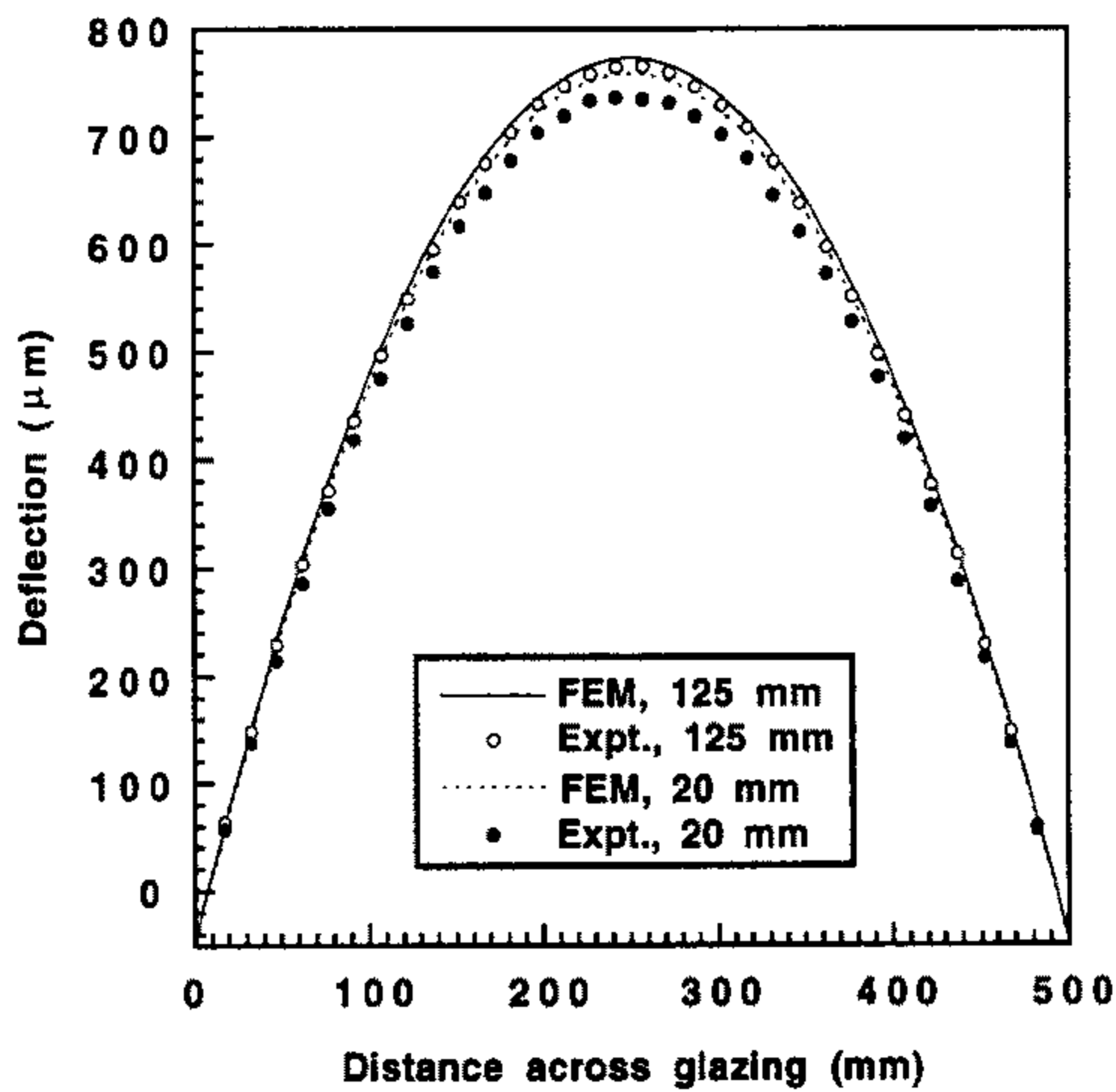


Fig. 6.18: Warm side external surface deflections at 125 and 20 mm from the edge of a 500 mm by 500 mm sample of vacuum glazing for $\Delta T_{g-g}=22.1$ °C. The test conditions are in agreement with those given in Table 6.2 for $\Delta T_{g-g}=22.0$ °C. Experimental and finite element modelling results (Expt. and FEM) are shown.

there is a greater reduction in the bending between 125 and 20 mm, with the maximum experimental deflections differing by about 30 μm . It is noted that the results of this figure represent the bending along individual transits with respect to points 7 mm from the edges, and not the total absolute deflections with respect to a common reference point such as the center or a corner of the glazing. The total deflection with respect to a corner increases with proximity to the center of the glazing.

In Fig. 6.18, there is generally a good agreement between the experimental and modelling results at both distances from the edge. However, there is less consistency between the two sets of results at 20 mm from the edge. The reasons for this discrepancy are not known but two possible ones are as follows. Firstly, the thermal perturbation to the local heat flow due to the presence of the slightly cooler bar is, as explained above, assumed to be small and to have an insignificant effect on the bending of the glazing. However, if the bar is placed along a single edge (or, in this case, at 20 mm from an edge) instead of closer to the center of the sample, this perturbation might significantly affect the bending because of the asymmetry in the conditions at the edges. Secondly, the offset that is applied to the modelling results (the predicted deflection along the measurement transit at 7 mm from the edge) is assumed to correspond to the deflection of all three mounting disks for the micrometer support bar - including the two disks at one end that are 16 mm on either side of the measurement transit. However, this assumption might be less valid for transits near the edge where, due to the much larger deflection gradients there, these two mounting disks might have significantly different deflections.

6.6.3 Strain Results

As explained above, all of the experimental and modelled strain results presented in this chapter reflect only the stresses due to the temperature induced bending of the glazing.

Figures 6.19a to d present experimental and modelling results for the warm side surface strains for four temperature differences across the glazing, and for the corresponding conditions listed in Table 6.2. The results are for the strains parallel and normal to the edge of the glazing, along a transit at 225 mm from the edge. For this comparison, the spatial resolutions of the measurements are insignificantly small and thus they are not represented. To within the accuracy of the strain measurements ($\pm 3 \mu\epsilon$), most of the experimental data agree with the modelling results. However,

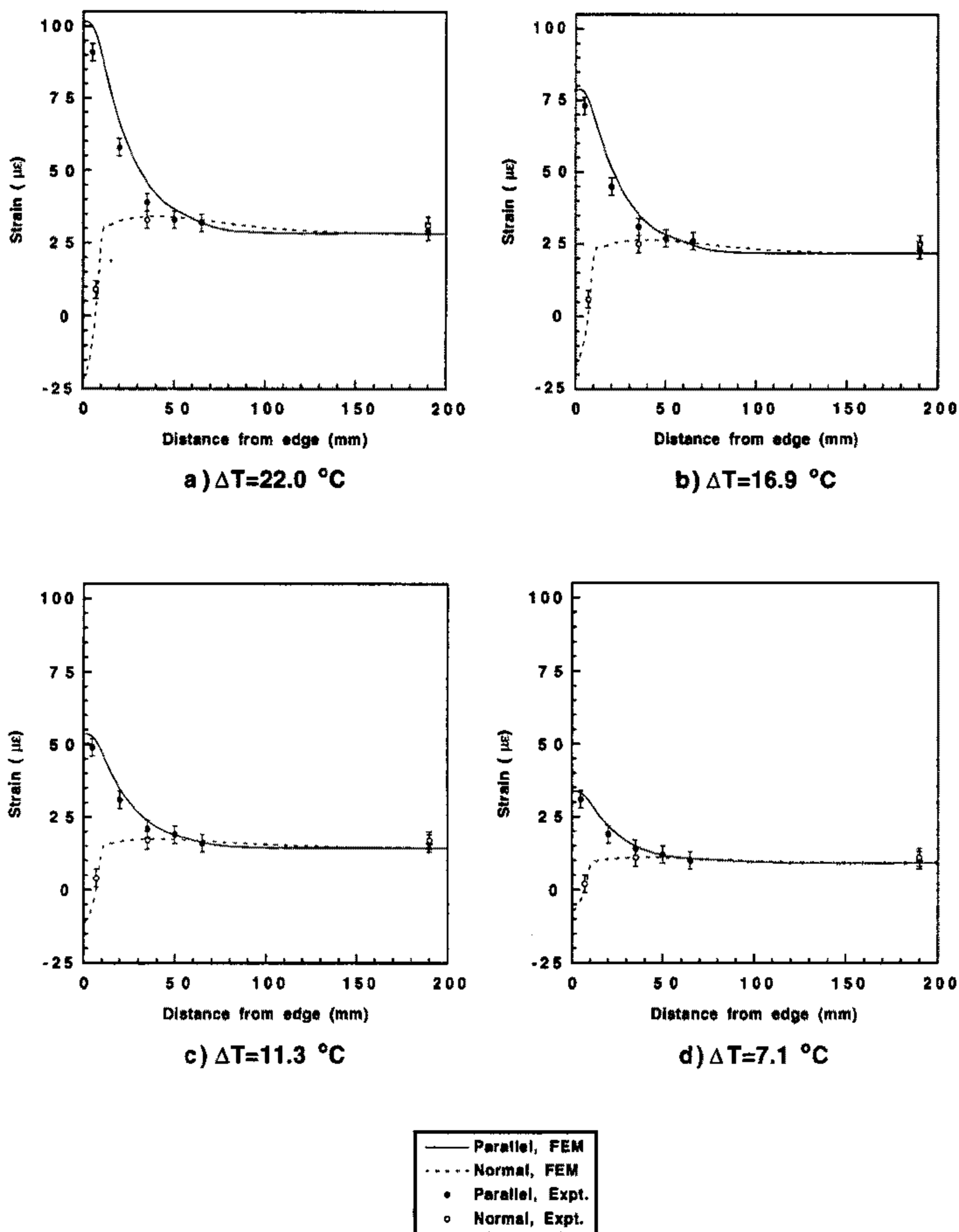


Fig. 6.19: Experimental and finite element modelling results (Expt. and FEM) for the warm side surface strains parallel and normal to the edge, along a transit at 225 mm from the edge. The results are for four different temperature differences across the glazing, and for the corresponding conditions given in Table 6.2.

most of the parallel strain measurements near the edge are slightly less than the predicted results. This is probably associated with the density of gages in this region; for the parallel strain measurements, most of an area of about 70 mm by 70 mm near the edge is covered by five measurement gages and the sections of microscope slides for the accompanying compensating gages (as indicated in Fig. 6.13). Clearly this covering will disrupt the local heat flow to the glazing, and thus affect the strain measurements. The better agreement between the experimental and modelling results for the normal strain is probably due to the fact that the gages for the normal strain measurements are more spread out.

Within the region that was measured, the experimental results are consistent with the trends seen in the modelling data. The more extensive modelling results show that, at the edge, the parallel strain is tensile and that it is much larger in magnitude than the normal strain, which is compressive. These strains change with distance from the edge: the parallel strain decreases but remains tensile, while the normal strain increases, becomes tensile and then slightly decreases. Beyond some distance from the edge, these two tensile strains are equal and uniform. The changes in strain are mostly due to the effects of the edge seal, which are described above in Section 6.2.

The variation in the normal surface strain near the edge is related to the stress at the edge of the glazing. At the edge, there is no force directed normally to the glass sheets - i.e. there is nothing pushing normally against the edge of the glazing. Thus, the normal stress - and the normal strain associated with this stress - must decrease to zero with proximity to the edge (as will be explained, the stress is actually slightly non-zero). However, Figs. 6.19 show that there is a significant compressive normal strain at the edge. This is due to the contraction of the glass in the direction perpendicular to the large parallel tensile strain at this location - i.e. there is a lateral compressive normal strain associated with the axial tensile parallel strain. In each of Figs. 6.19, the ratio of the magnitudes of the normal and parallel strains at the edge is within 0.5 % of 0.22. This is consistent with the fact that, in the absence of any other source of lateral strain, the ratio between the magnitudes of the lateral and axial strains at the edge must be equal to the Poisson's ratio for the glass: 0.22.

The fact that the strains near the center of the glazing are isotropic indicates that the bending of the glazing in this region is spherical in shape. In contrast, the inequality of the parallel and normal strains in the edge region indicates that the bending of this part of the glazing is non-spherical. Near the edge, the parallel strains are much larger in magnitude than the normal strains. Therefore, the strain in this region is essentially

uniaxial and thus the bending should be more cylindrical than spherical in shape. This is consistent with both the experimental and modelled deflection results of Fig. 6.15. At the center of the glazing, the deflection profiles are obviously spherical since, like spherical profiles of small curvature, they can be approximated by parabolas. However, near the edges, the deflections can be approximated by straight lines (this is best shown by the more extensive modelling results of Fig. 6.15). Thus, there is a straightening of the deflection near the edge - like that due to a transition from a spherical to a mostly cylindrical bending of the glazing in this region.

The modelling results of Figs. 6.19 indicate that the strain measurements at 190 mm from the edge should be well within the central region of constant strain. Therefore, these measurements are assumed to represent the strains at the center of the sample (250 mm from the edge). Figure 6.20 presents the center-of-glass strain as a function of ΔT_{g-g} . The experimental results are from Figs. 6.19, and from measurements taken at other temperature differences across the glazing. Parallel and normal measured strains are compared with the isotropic strains determined from the structural analysis finite element model, and from the simple analytic model for stress given by equation 6.2. This figure demonstrates the uniaxial nature of the central strains; to within the experimental error of $\pm 3 \mu\epsilon$, the measured parallel and normal strains are in agreement with each other. To within this same accuracy, the experimental data also agree with the finite element modelling results. As with the deflection results of Fig. 6.17, it is apparent that the analytic model provides only approximate results for the center-of-glass strain (and, in fact, it overestimates this strain). This is because of the effects of the edge seal (described above in Section 6.2 in the context of the associated stresses) which are not accounted for by this model. However, as indicated by the definitive linear trend of the more exact finite element modelling results, the analytic model correctly predicts a linear relationship between the center strain and ΔT_{g-g} .

Figure 6.21 presents the parallel edge strain as a function of ΔT_{g-g} . Only measurements at 5 mm from the edge are available, and thus they are compared with finite element modelling results for the same location. The analytic strains are calculated from equation 6.3 and they correspond to the strains right at the edge. Since the modelling data of Figs. 6.19 show that there is little change in this strain over the first few millimeters from the edge, it is appropriate to compare results for 5 mm from the edge with results at the edge. The experimental and finite element modelling results are in agreement at the smaller temperature differences. As with the central strains of Fig. 6.20, the analytic model overestimates the edge strain. This is

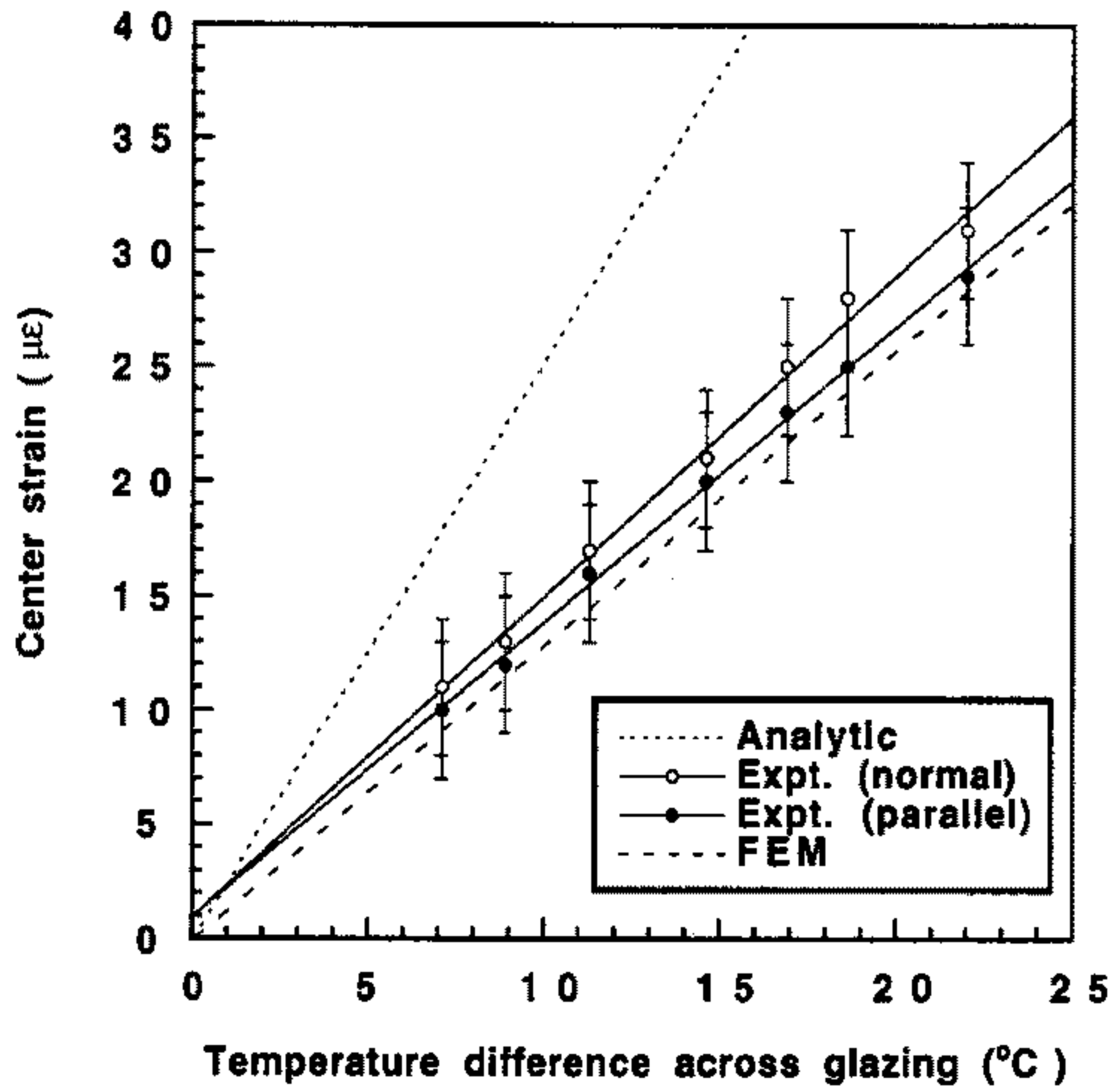


Fig. 6.20: Center-of-glass strain as a function of ΔT_{g-g} . The corresponding conditions are given in Table 6.2. Experimental strains (Expt.), parallel and normal to the edge, are compared with the isotropic strains calculated from the finite element model (FEM), and from the simple analytic model given by equation 6.2 (Analytic). Lines of linear regression are shown for the experimental results.

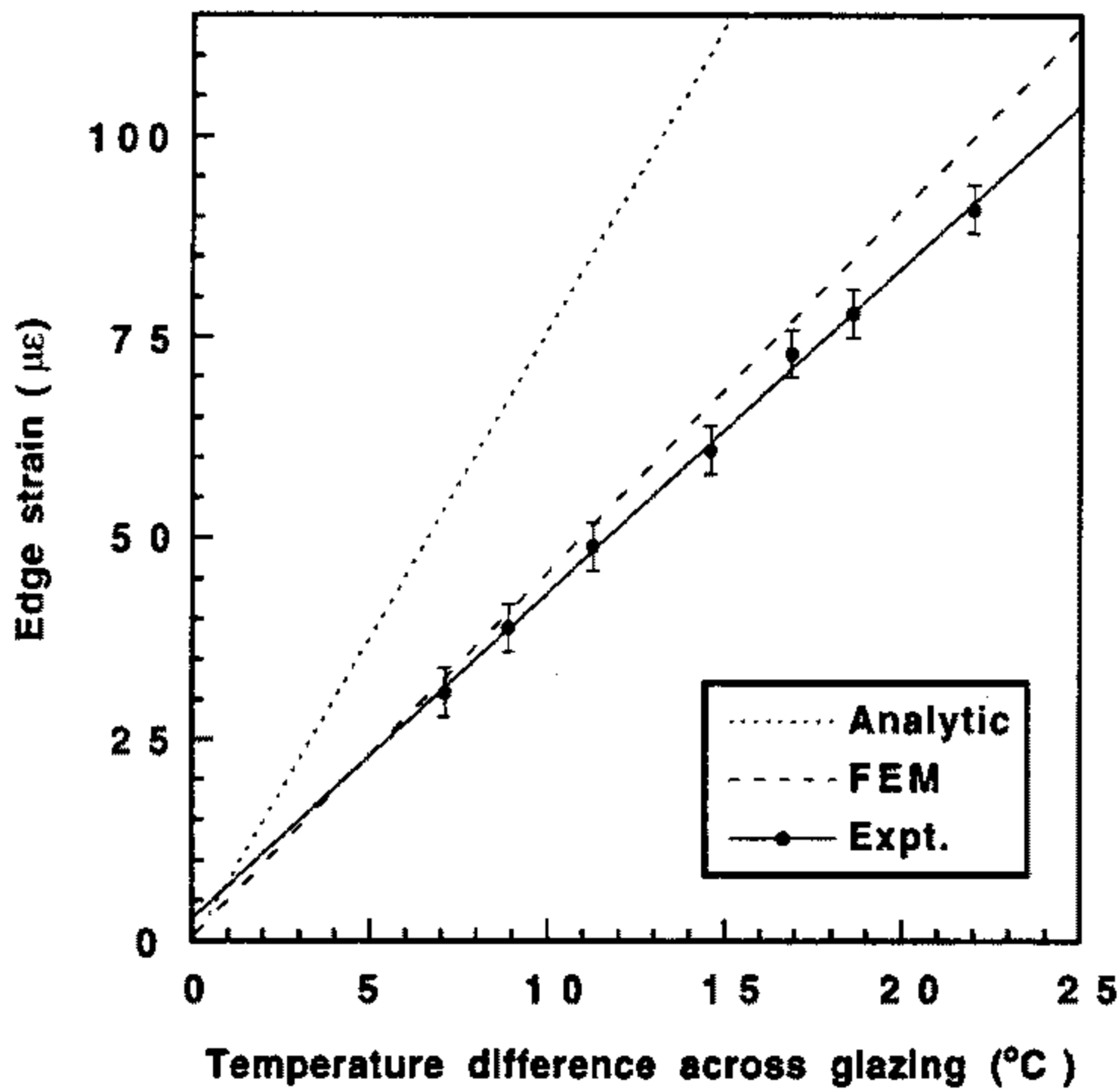


Fig. 6.21: Edge strain parallel to the edge as a function of ΔT_{g-g} . The corresponding conditions are given in Table 6.2. Experimental and finite element modelling results (Expt. and FEM) at 5 mm from the edge are shown with the strain calculated from the simple analytic model given by equation 6.3 (Analytic) for the strain at the edge. A linear regression fit to the experimental results is shown.

because of the effects of the edge seal which are not accounted for by this model. However, the analytic model correctly predicts a linear relationship between the edge strain and ΔT_{g-g} - as indicated by the definitive linear trend of the more exact finite element modelling results.

The center of the strain gage near the corner of the glazing is located at 30 mm from each edge. Thus, by symmetry, this gage effectively measures both the parallel and normal strains at this position. Figure 6.22 presents experimental and finite element modelling data for the strain at this location near the corner as a function of ΔT_{g-g} . The two sets of results are in agreement at the smaller values of ΔT_{g-g} . The strains measured at this location near the corner are less than the parallel strains near the centerline at 30 mm from the edge (as indicated by the experimental results of Figs. 6.19). This is consistent with the modelling results presented below which show that the corners of the glazing are regions of reduced stress - and thus strain. This occurs for the same reason given above for the decrease in the normal stress at the edge of the glazing: the absence of any external force against the edge of the glazing, directed normally to the edge. At each corner, the stress parallel to one edge is also, by symmetry, normal to the other edge - i.e. both axial stresses at the corners are normal stresses. Therefore, like the normal stresses at the edges elsewhere in the glazing, both of these axial stresses are close to zero (were it not for the finite thicknesses of the glass sheets, they would be exactly zero). Consequently, near the corners - and only near the corners - there must be a reduction in both the parallel and normal stresses, and thus a reduction in the associated strains.

It is recalled that all of the experimental strains presented above are net results which are calculated from measurements with and without the strains associated with the temperature induced bending stresses; these measurements are respectively made with and without a temperature difference across the sample. If the zero strain readings are correct, then a linear fit to the net results should yield no strain for a zero ΔT_{g-g} . In Figs. 6.20, 6.21 and 6.22, the y-intercepts for the linear regression fits to the measurements are within the experimental error ($\pm 3 \mu\epsilon$) of zero. Each of these intercepts corresponds to a separate strain gage bridge circuit. A similar check of the variation in strain with ΔT_{g-g} for the other seven bridge circuits showed the same consistency between the experimental and expected values for the y-intercepts. Therefore, to within the experimental error, the net results indicate that all of the zero strain readings are correct.

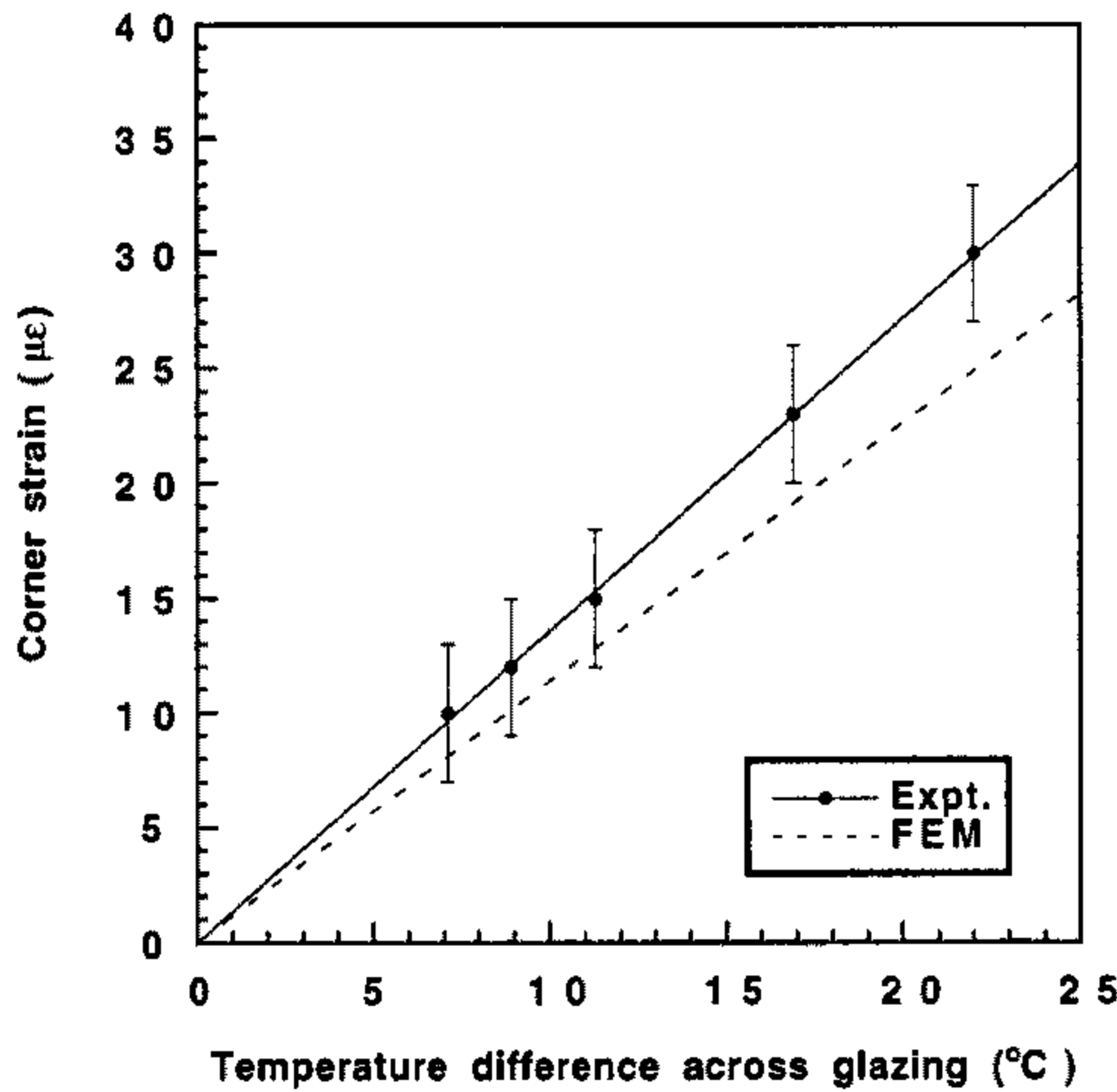


Fig. 6.22: Experimental and modelling results (Expt. and FEM) for the warm side strain near a corner, at a location 30 mm from each edge, as a function of ΔT_{g-g} . The corresponding conditions are given in Table 6.2. A linear regression fit to the experimental results is shown.

6.6.4 Results with Insulated Edges

The plate-to-glass thermal conductance via the radiation shield was determined from the test with the entire glazing covered with the sheet of aluminized cardboard. Based on the warm side plate and glass surface temperatures of 44.5 and 31.0 °C, and on the calculated heat flow across the glazing, this thermal conductance was determined to be $1.35 \pm 0.05 \text{ Wm}^{-2}\text{K}^{-1}$. This is the thermal conductance that was used in the thermal analysis finite element model to simulate the edge insulation (i.e., following the notation introduced in the previous chapter, $h_i = 1.35 \pm 0.05 \text{ Wm}^{-2}\text{K}^{-1}$).

Figures 6.23a and b present modelling and experimental results for the parallel strains at 225 mm from the edge for a ΔT_{g-g} of 22.0 °C, and for conditions that are in agreement with those listed in Table 6.2 for this temperature difference. Four insulation recess depths are considered: -8, 17, 27 and 37 mm. As previously explained, the insulation covers the 8 mm wide edge seal and extends beyond the inner edge of the seal by the recess depth (i.e. the recess depth is measured from the inner edge of the seal). The results for the -8 mm case (which corresponds to no edge insulation) are from Fig. 6.19a. Close to the edge, the modelled strain increases with

the insulation width, while the reverse situation applies beyond about 65 mm from the edge. The experimental data confirm the former trend, and it appears that they indicate either a similar cross over or a convergence in the strain profiles beyond 65 mm from the edge; to within the accuracy of the measurements ($\pm 3 \mu\epsilon$), it is not possible to tell whether these results cross over or converge.

As previously discussed, the experimental strains close to the edge are less than the predicted results. The increase in the edge strain with the insulation recess depth is due to the fact that only the warm side of the glazing is insulated. As explained at the end of Section 5.2.1, the addition of edge insulation to only the warm side of the glazing lowers the edge temperature with respect to the mean glazing temperature. As noted in Section 6.2, this increases the additional parallel tensile stresses in the warm side edge region.

Figure 6.24 presents the normal strains that were measured along with the parallel strains of Fig. 6.23b. Modelled normal strains are also shown in Fig. 6.24. There is a better agreement between the experimental and modelling results for the normal strains than there is for the parallel strains. However, once again, the measurements near the center of the glazing are greater than the predicted strains and, to within the experimental error, most of these central strains are in agreement with each other. The slight non-uniformities in the modelling results within the insulated regions are probably related to the presence of the insulation.

Centerline deflections were measured along with the strain results of the previous two figures. These deflections are compared with modelling results in Figs. 6.25a and b. Both sets of results show that the overall glazing deflection decreases with the insulation recess depth. This is consistent with the decrease in the central strain with the recess depth which is definitively indicated by the modelling results of Figs. 6.23a and 6.24.

It is noted that the edge insulation was modelled as slab with a thermal conductance equal to the conductance between the plate and glass surfaces via the radiation shield. This conductance was assumed to be uniform over the insulated region, and to be constant for all of the insulated edge tests. For several reasons this is not quite correct, and thus the way in which the insulation was simulated could be partly responsible for the discrepancies between the experimental and modelling results of Figs. 6.23 and 6.24. Due to the variation of the glass temperature near the edge, and due to the changes in the test temperatures, the plate-to-glass thermal conductance

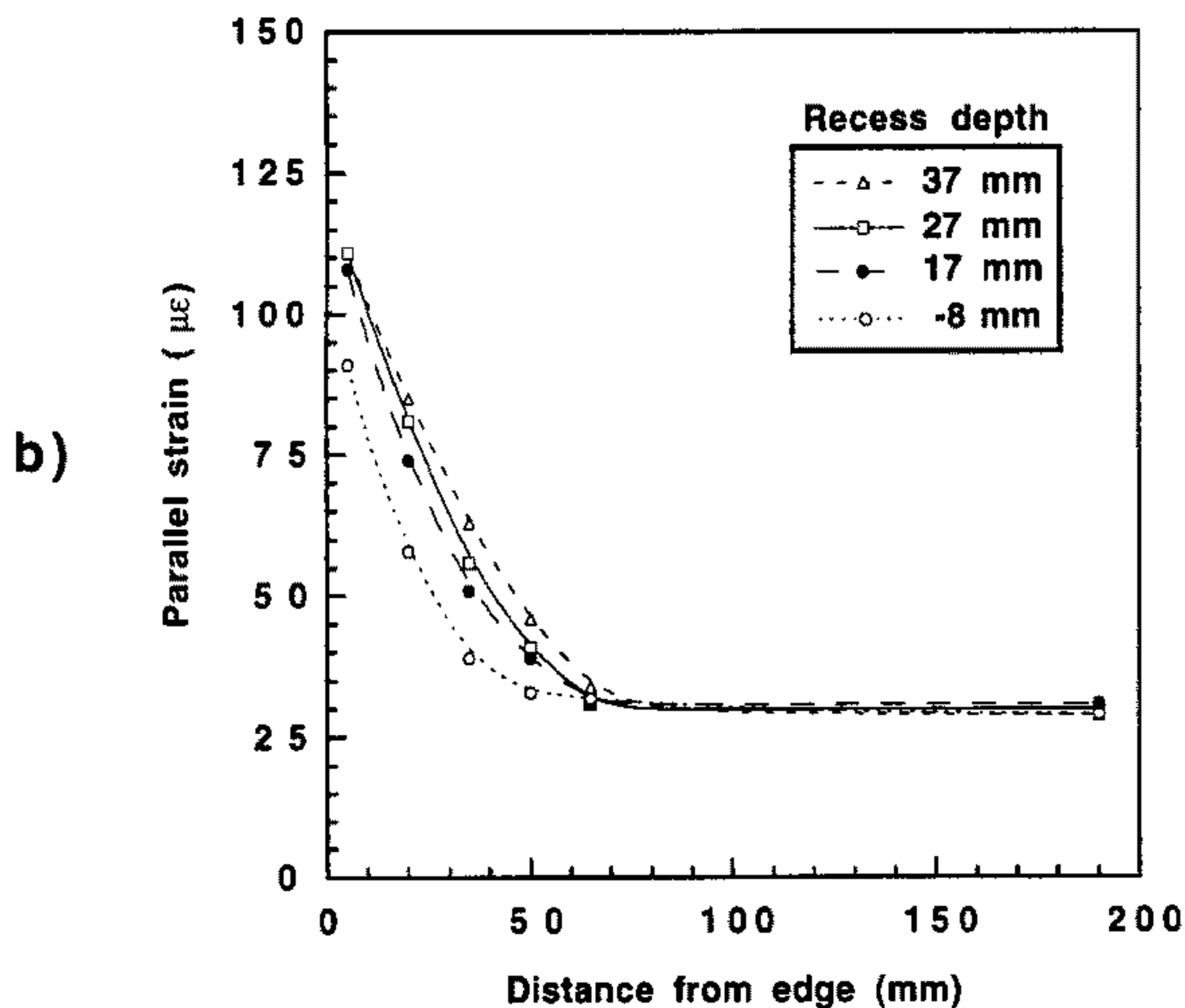
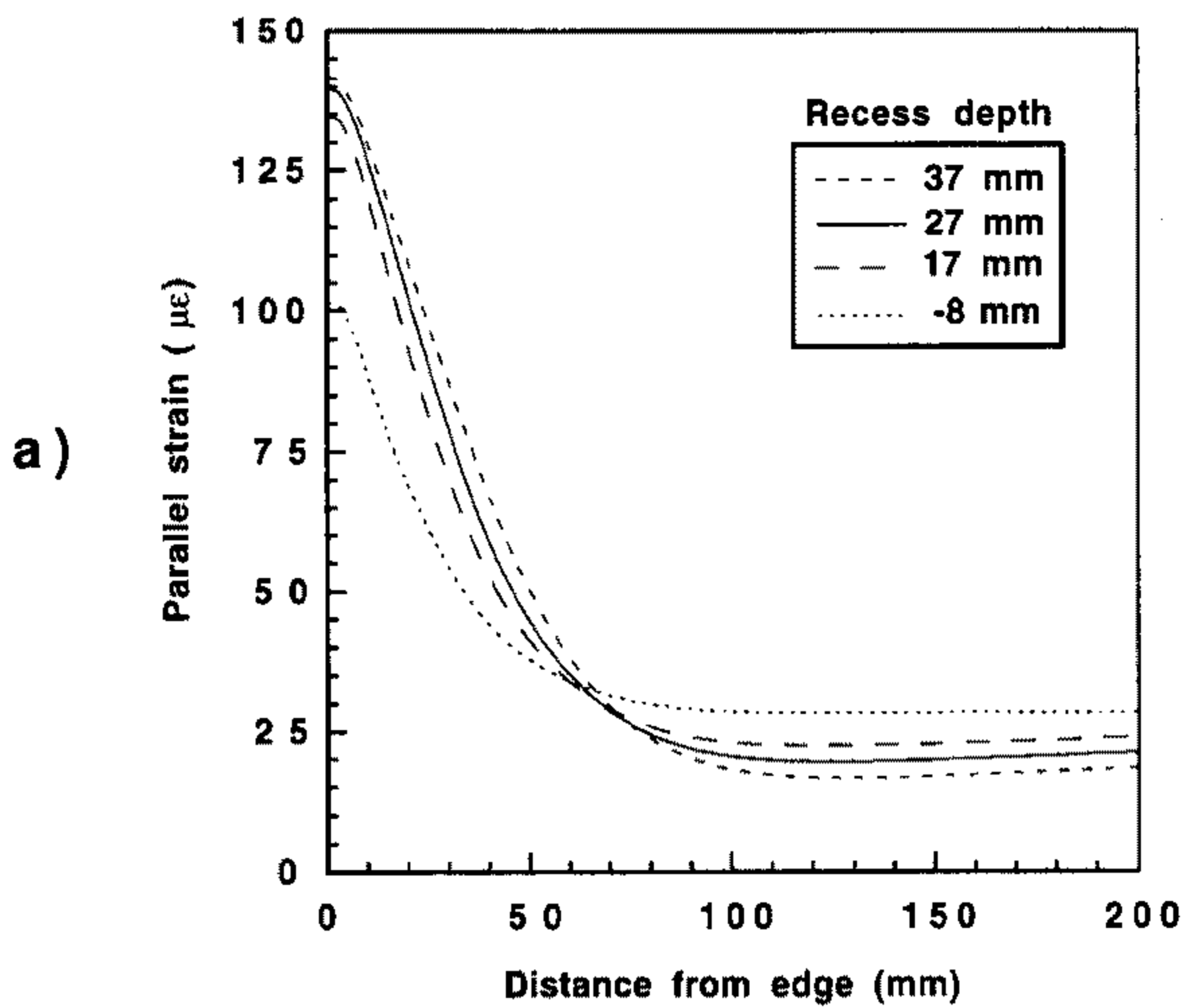


Fig. 6.23: Warm side strain parallel to the edge, at 225 mm from the edge, for $\Delta T_{g-g} = 22.0$ °C, and for four recess depths of warm side insulation: (a) modelling results, and (b) experimental results. The corresponding conditions are given in Table 6.2. The recess depth is measured from the inner edge of the 8 mm wide edge seal, and thus the recess depth of -8 mm corresponds to no edge insulation. The experimental results are accurate to ± 3 $\mu\epsilon$.

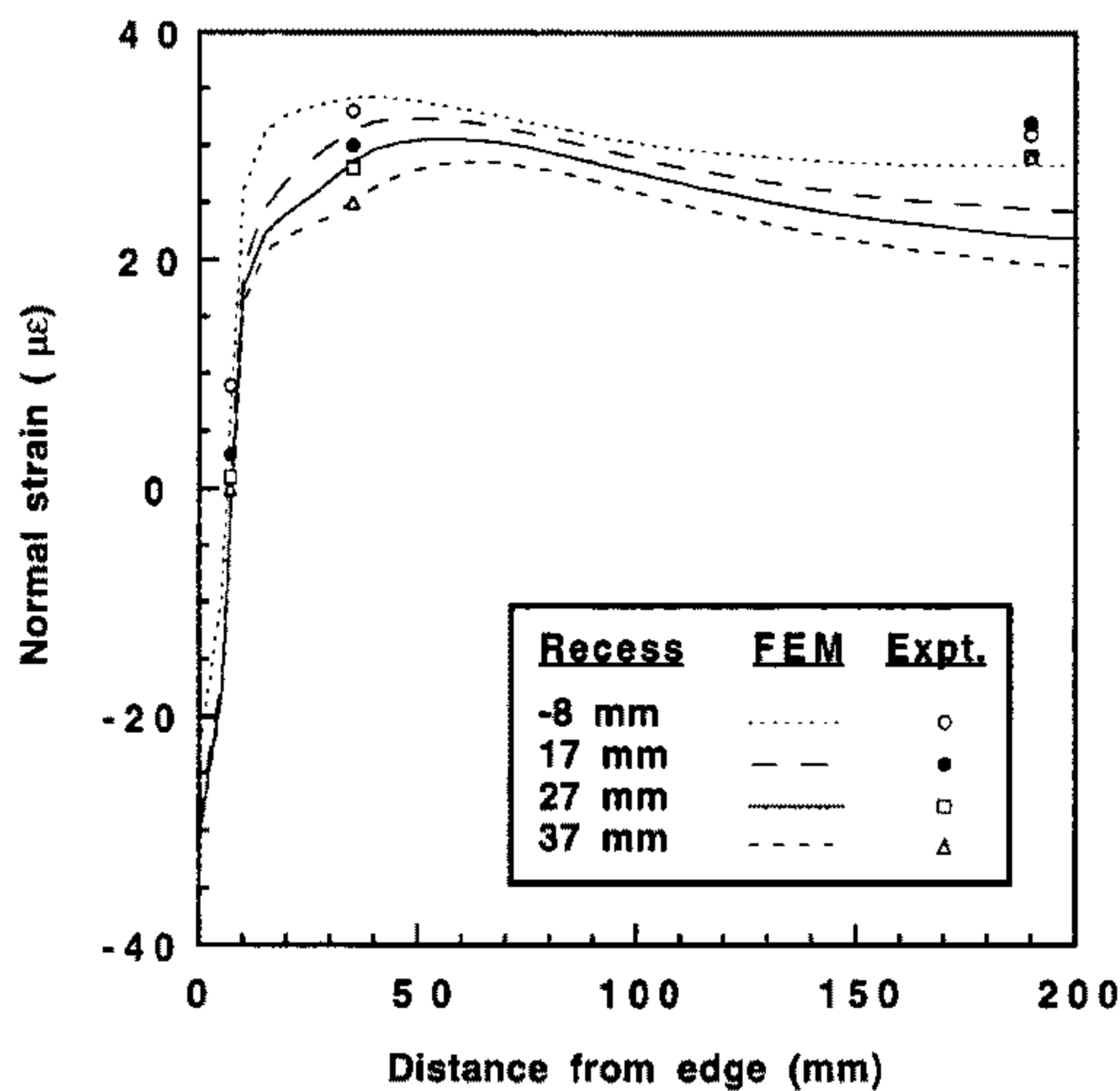


Fig. 6.24: Experimental and modelling results (Expt. and FEM) for the warm side strain normal to the edge, at 225 mm from the edge, for $\Delta T_{g-g} = 22.0$ °C, and for four recess depths of warm side insulation. The corresponding conditions are given in Table 6.2. The recess depth is measured from the inner edge of the 8 mm wide edge seal, and thus the recess depth of -8 mm corresponds to no edge insulation. The experimental results are accurate to ± 3 $\mu\epsilon$.

across the radiation shield is neither uniform nor constant; this conductance should be adjusted for the temperature dependencies of the system surface emittances. However, since these dependencies were not known, the thermal conductance given above - which was determined at temperatures close to those of the insulated edge tests - was used for the model. Another reason why the thermal conductance is not uniform over the insulated region is that, near the inner edge of the radiation shield, there is some radiative heat transfer at non-normal angles between the plate and glass surfaces; this heat transfer could be reduced by decreasing the distance between the shield and glazing. However, there is not much room to do so because the wires to the strain gages are also some distance above the surface of the glazing.

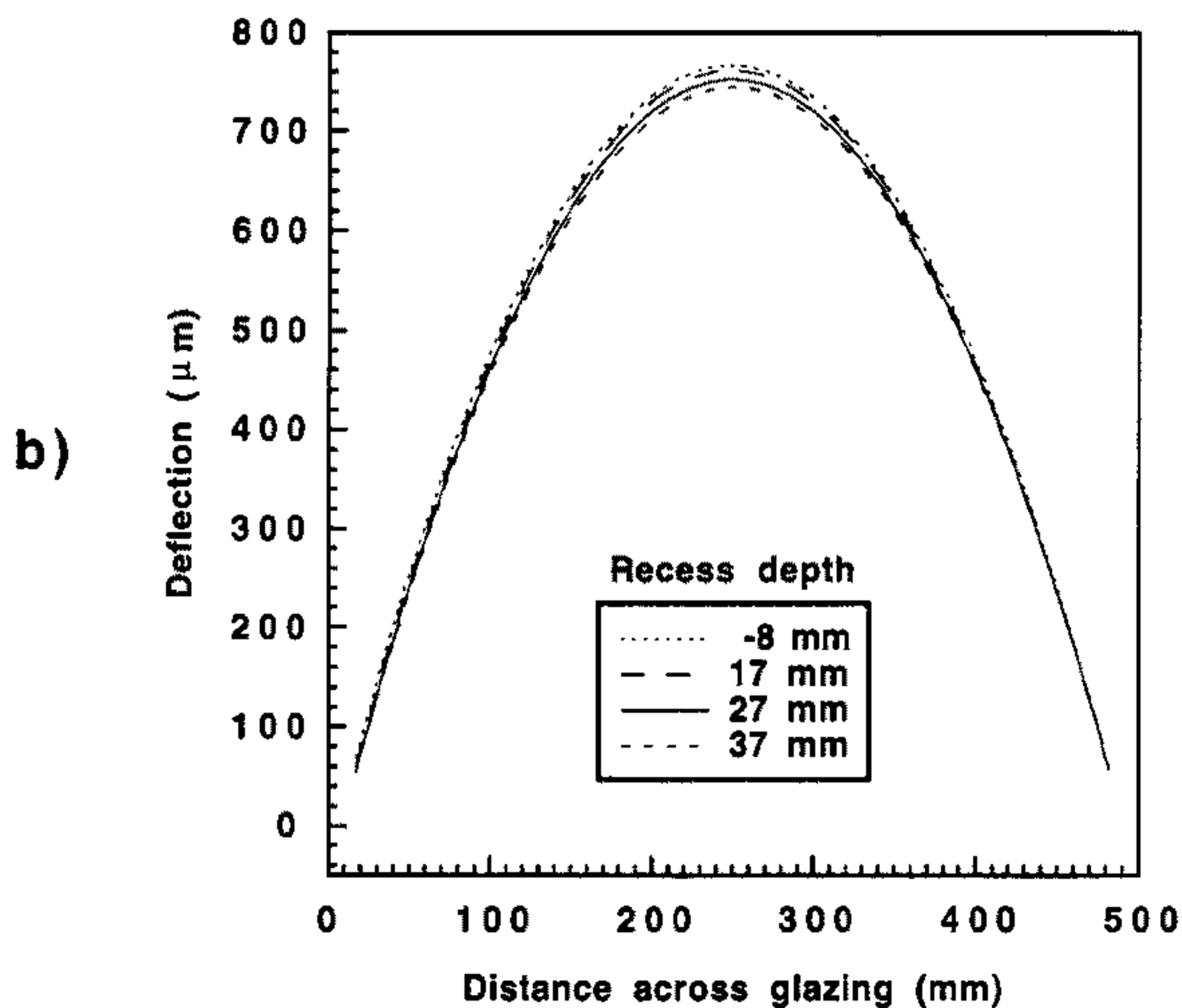
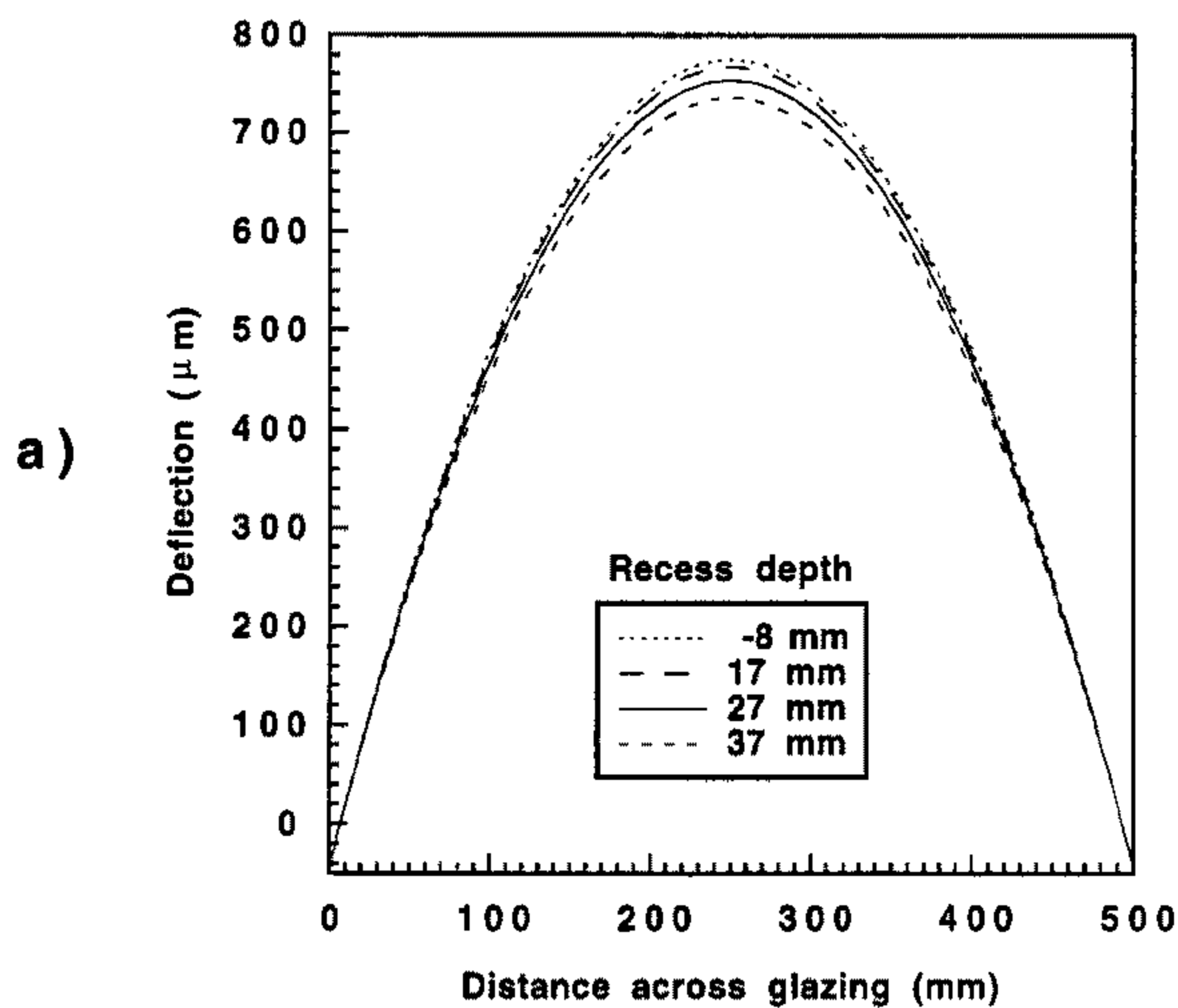


Fig. 6.25: Warm side external surface deflections along the glazing centerline (250 mm from the edge) for $\Delta T_{g-g} = 22.0\text{ }^{\circ}\text{C}$, and for four recess depths of warm side insulation: (a) modelling results, and (b) experimental results. The corresponding conditions are given in Table 6.2. The recess depth is measured from the inner edge of the 8 mm wide edge seal, and thus the recess depth of -8 mm corresponds to no edge insulation. The experimental results are accurate to $\pm 4\text{ }\mu\text{m}$.

6.6.5 Complete Modelling Results

Thus far, only the strains on the warm side of the glazing have been presented. This is because the experimental validation of the model is based on warm side measurements. In order to provide a complete overview of the situation on both sides of the glazing, modelling results are now also presented for the cold side. Since the data will be referenced in Chapter 8 in the context of probability of failure calculations, several changes were made to the modelling analysis. Firstly, the thermal solutions correspond to more realistic ambient conditions - the ASTM standard fenestration test conditions first introduced in Chapter 4: air temperatures of 21.1 and -17.8 °C, and respective heat transfer coefficients of 8.3 and 30.0 Wm⁻²K⁻¹ [9]. Secondly, the glass-to-glass center-of-glass U-value is 1.10 Wm⁻²K⁻¹. This U-value corresponds to a typical vacuum glazing (one with two sheets of K glass, a pillar radius of 0.125 mm and a regular pillar separation of 25 mm) which is exposed to the above mentioned ASTM conditions. Thirdly, the reference temperature for the calculation of the thermal contraction of the glass is 21.1 °C. Fourthly, the size of the modelled glazing is closer to that which is expected to be commonly used in actual installations: 1 m by 1 m. Except for this change in the overall glazing dimension, the structural analysis model is the same as that described in Section 6.3; the corresponding model used for the thermal analysis step of the solution procedure is identical to that presented in Section 5.3, except that the width of the edge seal is 8 mm (this is a more typical width for the seal). Fifthly, since stresses and not strains are used in the probability of failure calculations, the results are presented as stresses. Lastly, in order to demonstrate the relationship between the glass temperatures and the stresses in the edge region, some glass sheet temperature profiles are shown along with the stress results.

Contour plots of the external surface stresses on both sides of an uninsulated glazing are presented in Figs. 6.26a and b. The glazing edges (i.e. the outer edges of the model) are along the bottom and left borders of the plots (which are, respectively, along the x-axis and the y-axis of the model). The stresses shown in these plots are parallel to the x-axis. Due to the geometry of the model, the stresses parallel to the y-axis are symmetric with the plotted stresses. In these figures, the stresses near the glazing edge along the x-axis are parallel to the edge, and those near the glazing edge along the y-axis are normal to the edge. These figures show that, near the edges, the parallel and normal stress contours are fairly uniform between the glazing centerlines and about 150 mm from the glazing corner (which is at the lower-left corner of each plot). As explained above, there is a reduction in the stresses in the corner region of

the glazing. The slight non-uniformities in the individual contours are associated with the grading in the node density of the model.

Temperature and stress profiles along the glazing centerline (i.e. along the inner edges of the model) for the conditions considered in Figs. 6.26 are presented in Figs. 6.27a and b. The results for Fig. 6.27b were obtained by appropriately smoothing the corresponding stresses of Figs. 6.26; this smoothing was done in order to remove the above mentioned non-uniformities in the stresses of Figs. 6.26 that are due to the grading in the node density. Thus, the results of Fig. 6.27b are not exactly the same as the stresses shown in Figs. 6.26. Figures 6.27 demonstrate the relationship between the glass temperatures and the stresses in the edge region. In this case, the edge temperature is less than the mean glazing temperature. As explained in Section 6.2, this leads to *additional* tensile parallel stresses in the edge region, which is why the parallel stress profiles are asymmetric and skewed in the direction of positive (tensile) stress. The presence of maximum and minimum normal stresses on the warm and cold sides, respectively, was not revealed in a previous modelling study [1] which showed only asymptotically increasing and decreasing normal stresses on these sides; this was probably because the earlier work was based on a model with a much lower node density.

As expected, the warm side stresses of Figs. 6.26a and 6.27b show similar trends to those seen in the above results for the parallel and normal strains. As previously explained in the context of the associated strain, the normal stress at the edge is expected to be zero. However, due to the finite thickness of the glass sheet, it is slightly compressive (-0.2 MPa). The trends in the stresses on the cold side are of the opposite sense to those on the warm side, but they occur for the same reasons as those for the trends in the warm side stresses.

Figures 6.28a and b present centerline temperature and stress profiles for same conditions considered for Figs. 6.27, but with 33 mm of edge insulation on both sides of the glazing. This width of insulation is measured from the outer edge of the 8 mm wide seal, and thus the insulation recess depth is 25 mm. The thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$. A comparison between the results of Figs. 6.27a and 6.28a shows that, as discussed at the end of Section 5.2.1, the addition of edge insulation with the same recess depth to both sides of the glazing decreases the difference between the mean and edge temperatures of the glazing. As explained in Section 6.2, this leads to a decrease in the *additional* parallel tensile surface stress on each side of the glazing. In Fig. 6.28b, this is exhibited by the drop in the tensile

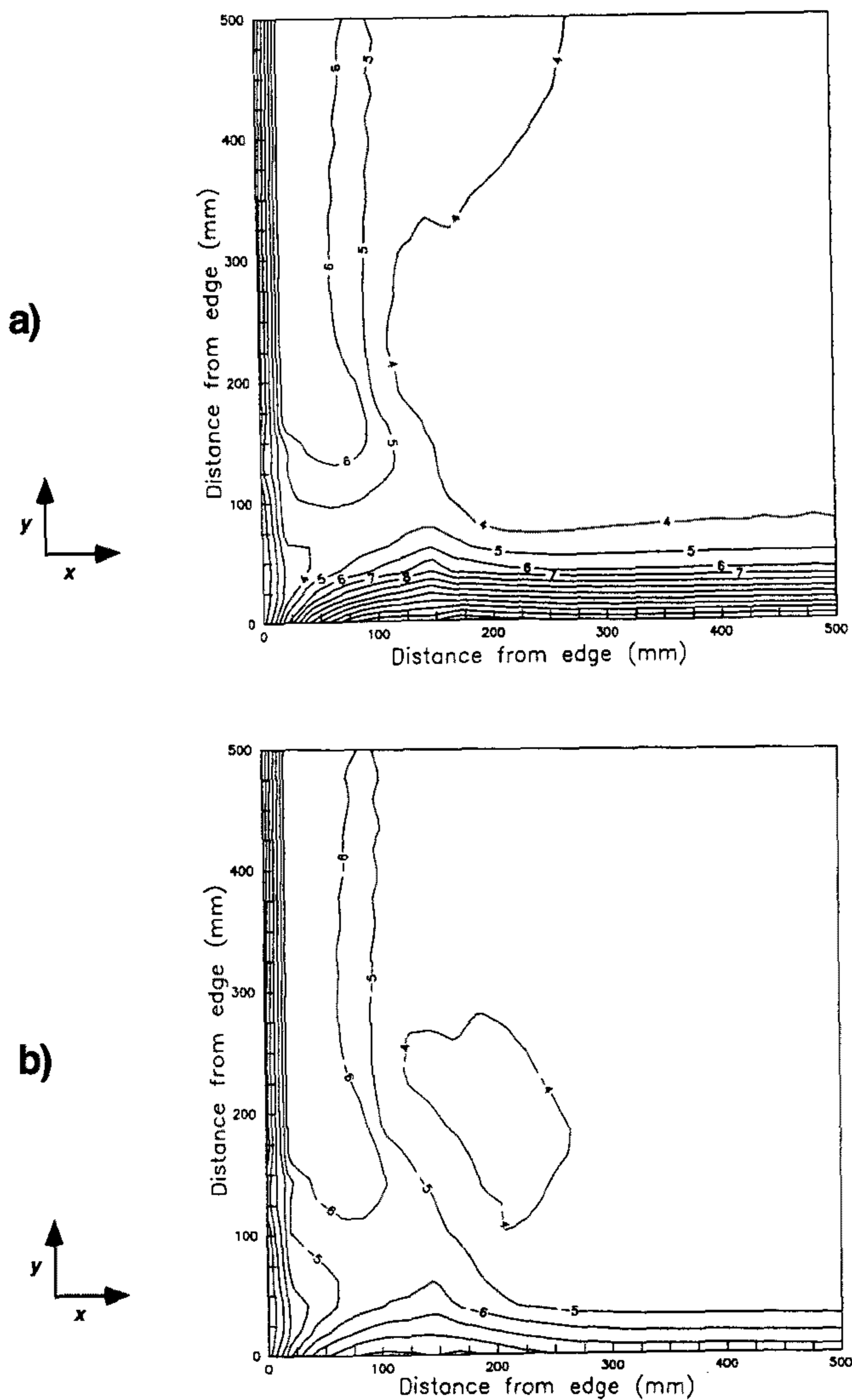


Fig. 6.26: Contour plots of the external surface stress parallel to the x-axis for both sides of a quarter section of a 1m by 1m glazing, and for the ASTM standard test conditions and $U_{g-g}=1.10 \text{ Wm}^{-2}\text{K}^{-1}$: (a) warm side, and (b) cold side. The x-axis is parallel to the lower border of each plot. The edges of the glazing (i.e. the outer edges of the model) are along the labelled borders, and the glazing corner is at the lower-left corner of the plots. The contour interval for both plots is 1 MPa.

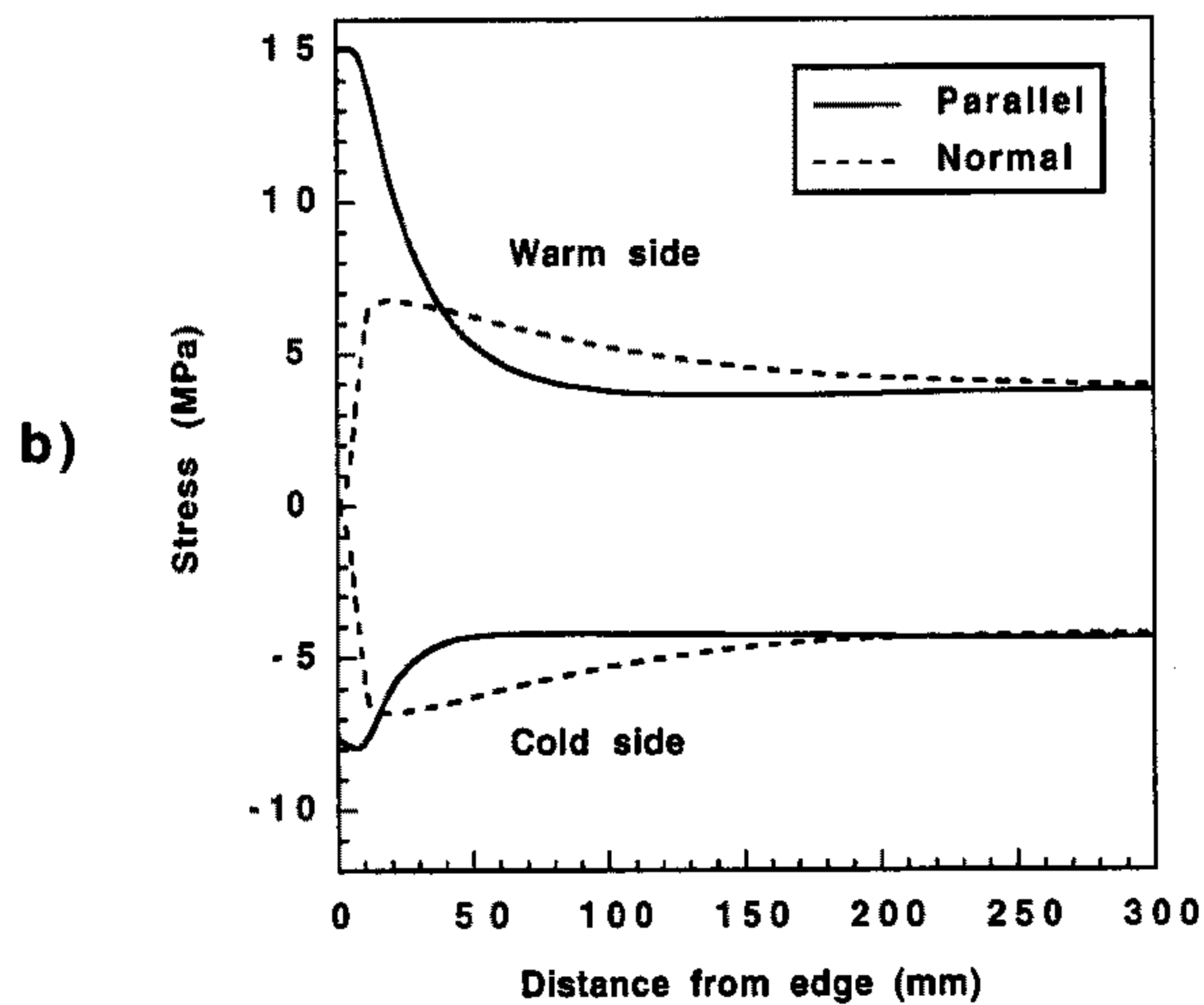
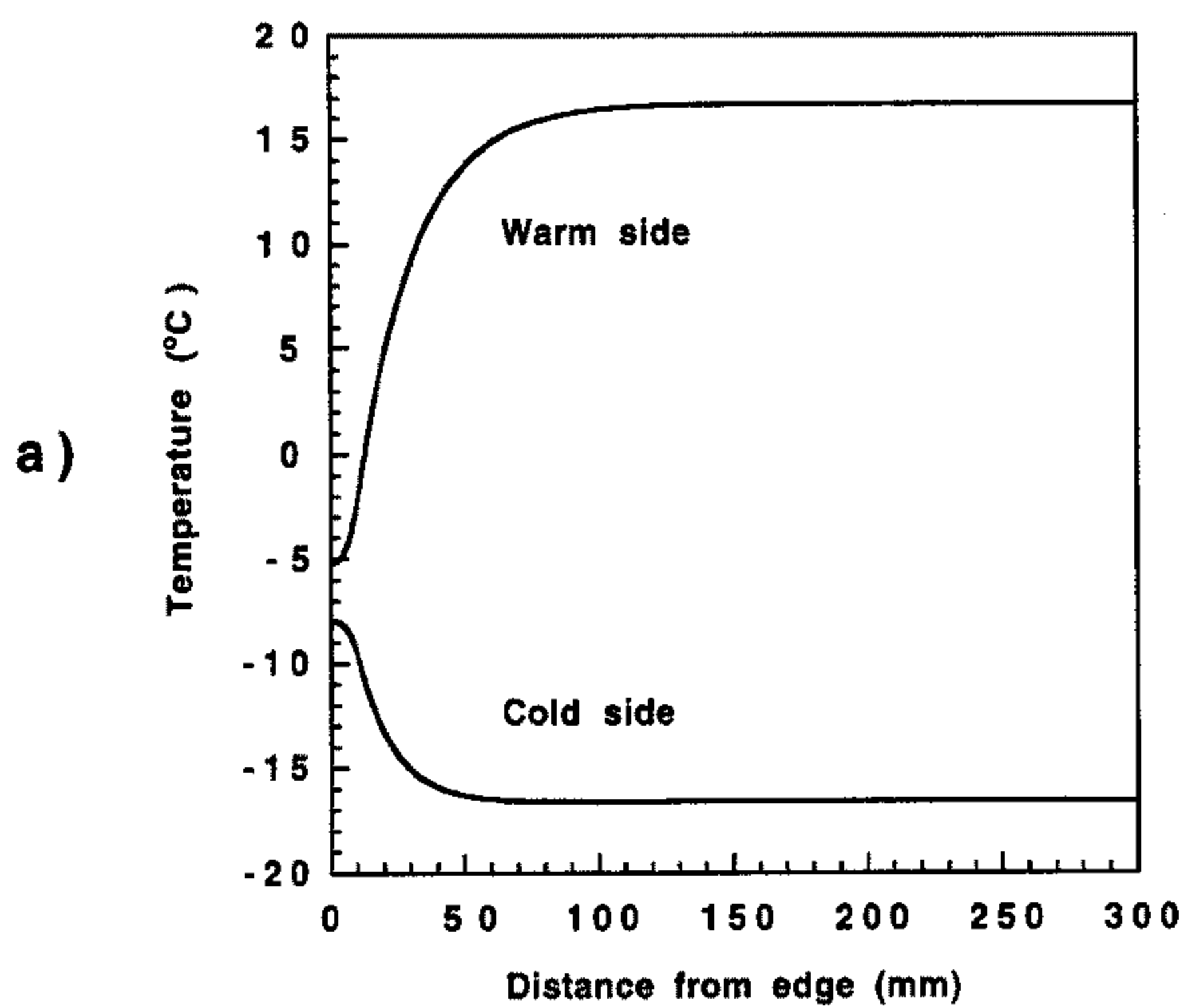


Fig. 6.27: (a) The centerline external surface temperatures, and (b) the centerline external surface stresses for the results presented in Fig. 6.26.

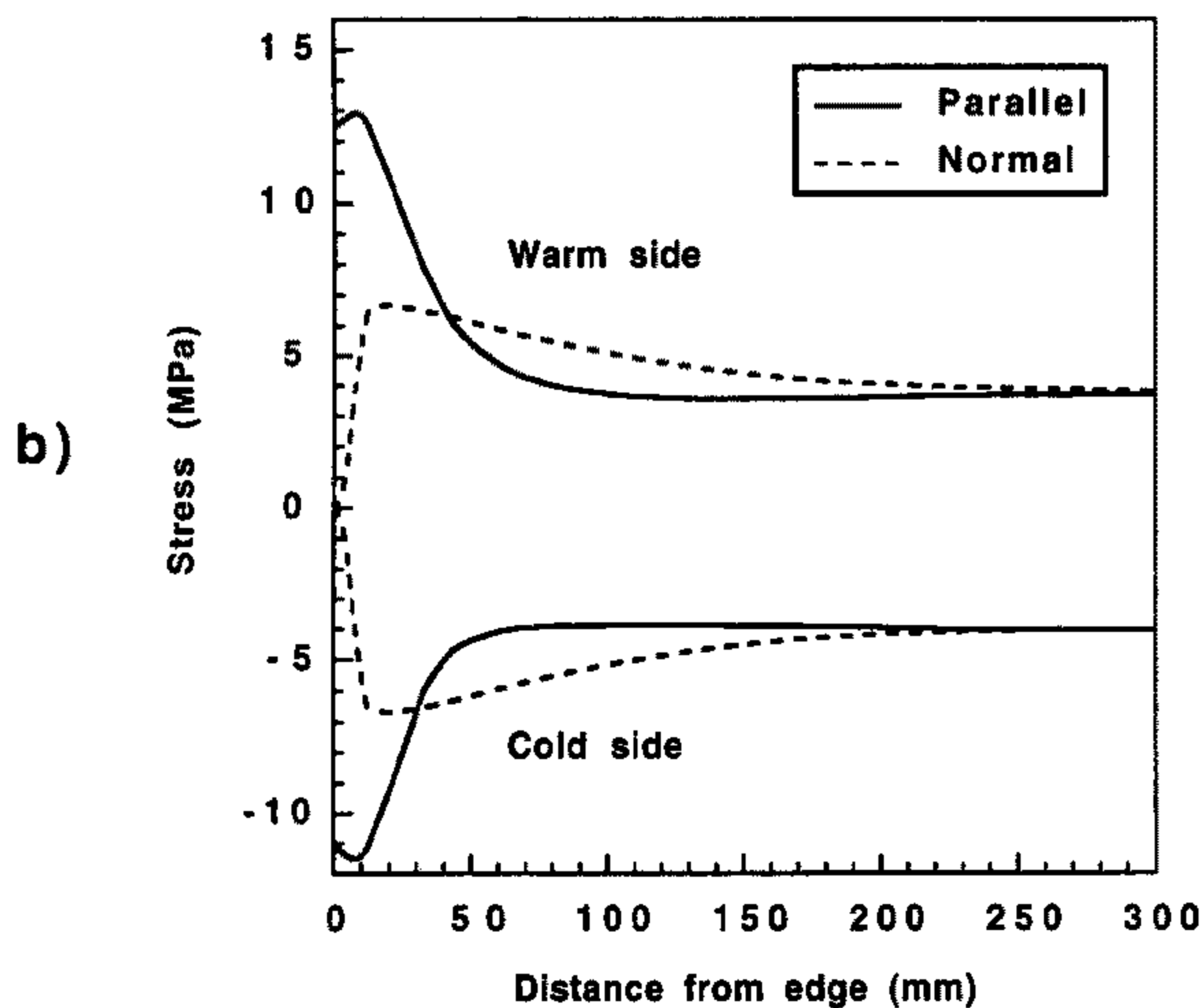
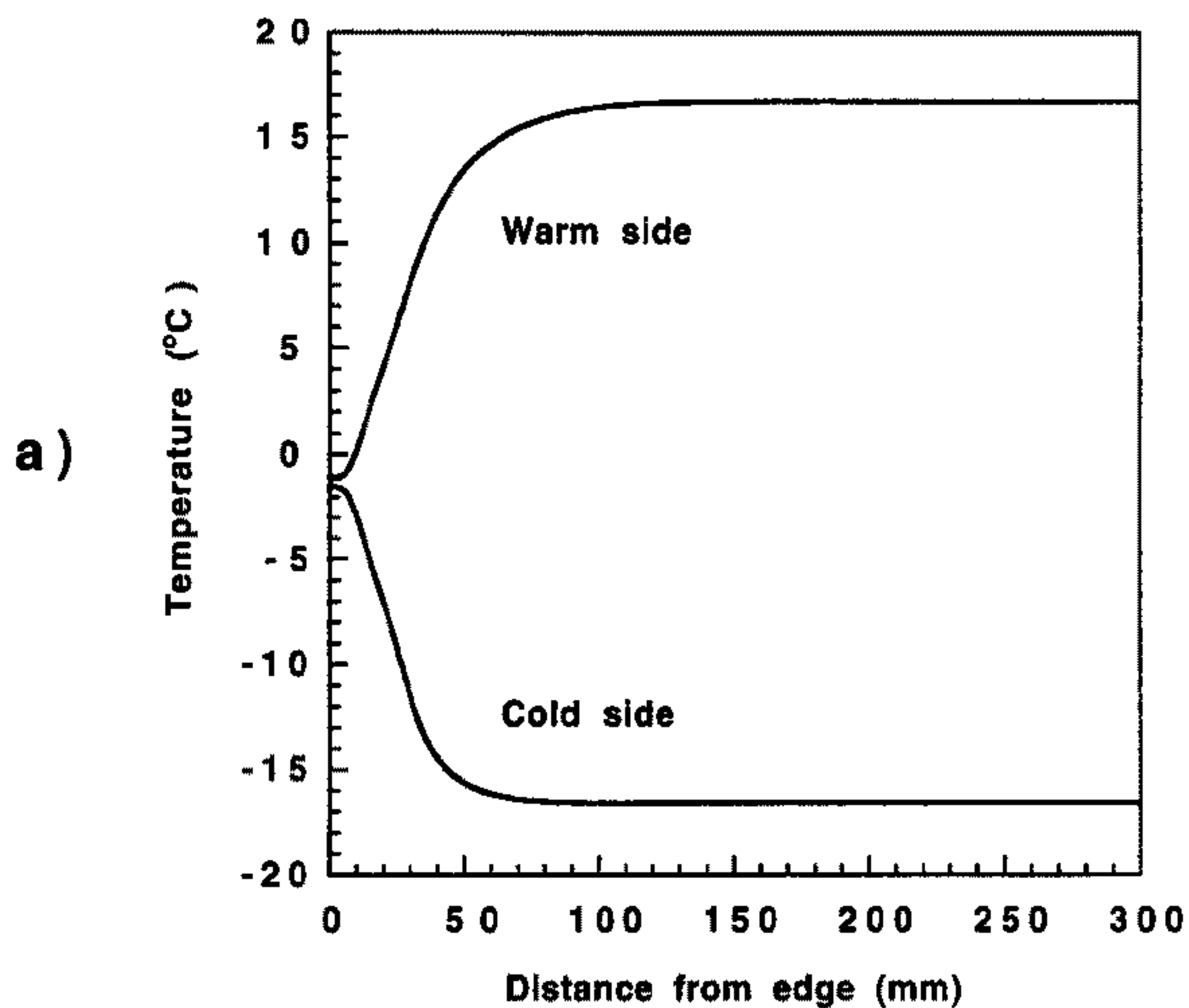


Fig. 6.28: The centerline external results for the ASTM test conditions, $U_{g-g}=1.10 \text{ Wm}^{-2}\text{K}^{-1}$, and an edge insulation recess depth of 25 mm on both sides of the glazing: (a) surface temperatures, and (b) surface stresses. The thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$.

stress on the warm side, and by the rise in the compressive stress on the cold side. These changes to the warm and cold side stresses are approximately equal in magnitude. However, as will be explained in Chapter 8, what is most important in the context of the probability of failure of the glazing is the change in the warm side tensile stress. The fact that the normal stresses are virtually unchanged by the thermal effect of the insulation indicates that, as stated in Section 6.2, the difference between the edge and mean temperatures is only responsible for additional stresses which are directed parallel to the edge.

The effect of edge insulation on the edge stress is further investigated by considering other insulation recess depths. Centerline surface temperature profiles for the same conditions considered in Figs. 6.27 and 6.28 and for several recess depths, w , are presented in Figs. 5.18a to f (the results for Figs. 6.27a are from Fig. 5.18a). These figures clearly show that the addition of edge insulation to both sides of the glazing raises the temperature of the edge, T_{edge} , relative to the mean glazing temperature, T_{mean} (i.e. it decreases the difference between the mean and edge temperatures). The warm side centerline surface stresses were calculated for each value of w considered in Figs. 5.18 and 6.28 (and also for $w=35$ mm) and some of these stress results are presented in Fig. 6.29. This figure further demonstrates that the parallel tensile surface stress at the edge decreases with the addition of edge insulation to both sides of the glazing. The expected correlation between $T_{mean}-T_{edge}$ and the edge stress as functions of the recess depth is presented in Fig. 6.30.

As explained in Section 6.2, if the edge temperature is greater than the mean glazing temperature, then the *additional* edge stresses will be compressive instead of tensile. This will result in a reduction of the *regular* tensile temperature induced edge stress on the warm side of the glazing. As explained in Chapter 8, such a reduction will usefully decrease the probability of glazing failure. For the conditions considered in Figs. 6.27 and 6.28, this can not be accomplished by increasing the equal insulation recess depth on both sides of the glazing; with increasing recess depth, the edge temperature approaches but does not exceed the mean glazing temperature. Neither can this be accomplished by insulating only the warm side of the edge because, as previously mentioned, this has the opposite effect on the edge temperature. However, a simple analysis based on the thermal resistance circuit presented in Fig. 5.3 shows that the edge temperature can be made greater than the mean temperature if only the cold side of the edge is insulated. This will reduce the warm side tensile edge stress and thus the probability of glazing failure. However, from the point of view of reducing the edge conduction, the installation of edge insulation to only one side of

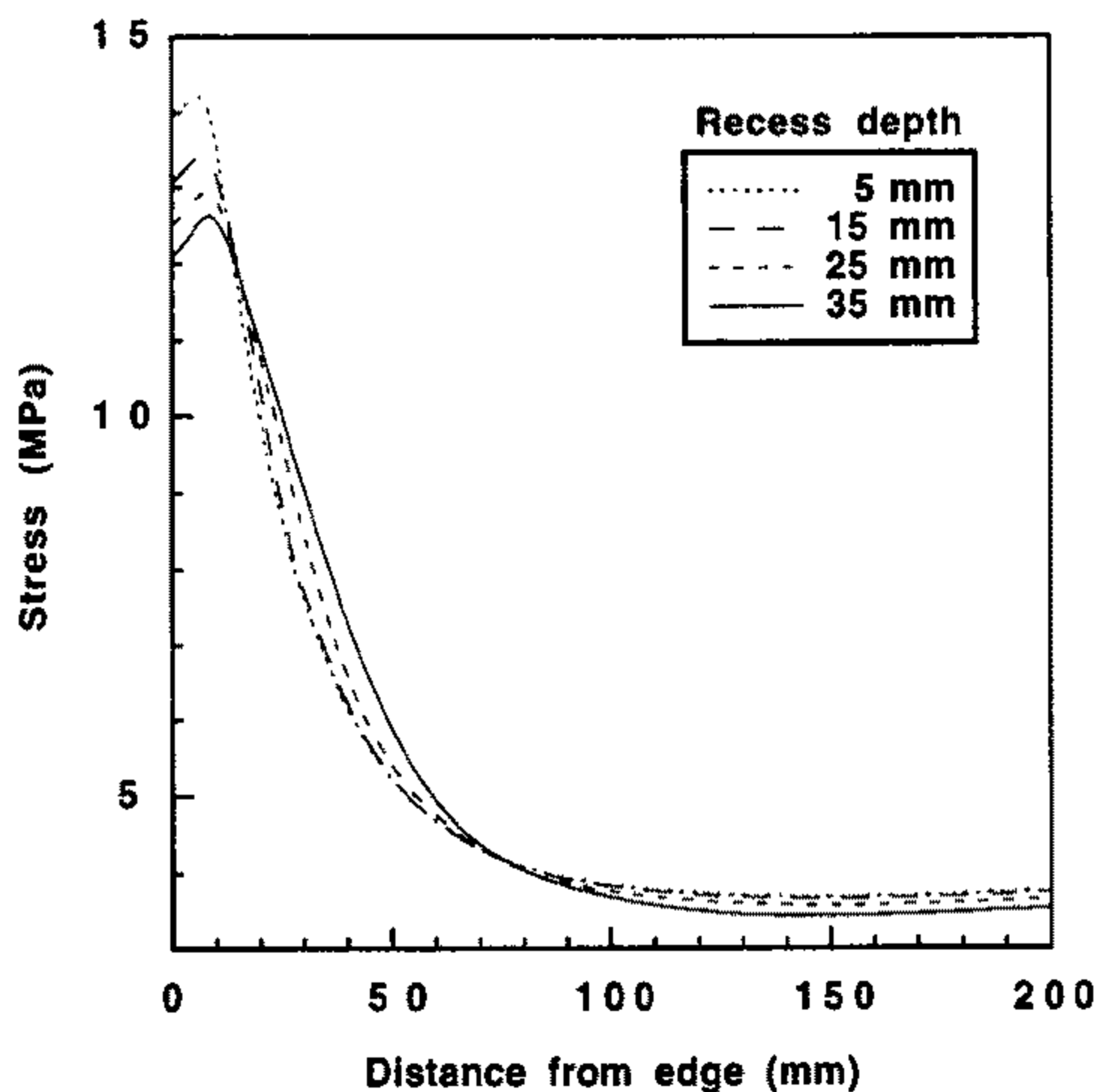


Fig 6.29: Centerline warm side parallel external surface stresses for the ASTM test conditions, $U_{g-g}=1.10 \text{ Wm}^{-2}\text{K}^{-1}$, and different recess depths. The recess depth is the same on both sides of the glazing. The thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$.

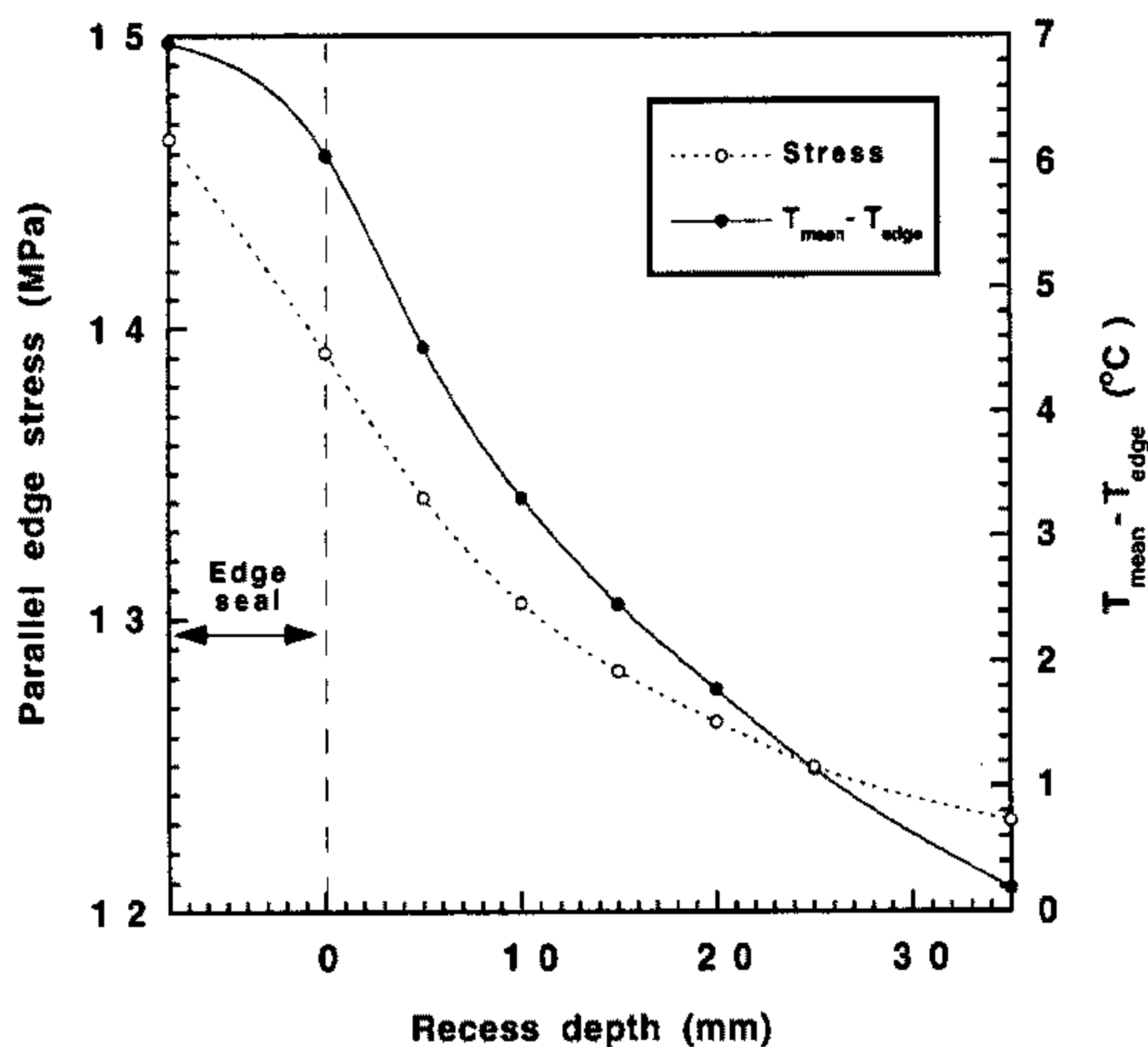


Fig. 6.30: Centerline results for the warm side parallel external surface stress at the edge and the difference between the mean and edge temperatures of the glazing ($T_{mean}-T_{edge}$) as functions of the insulation recess depth. The recess depth is the same on both sides of the glazing. The results are for the ASTM test conditions, $U_{g-g}=1.10 \text{ Wm}^{-2}\text{K}^{-1}$, and an insulation thermal conductance of $1.5 \text{ Wm}^{-2}\text{K}^{-1}$. The extent of the 8 mm wide edge seal is indicated.

the glazing is not the most effective use of the insulation.

Thus far, only the direct stresses parallel to the axes of the model (i.e. to the edges of the glazing) have been considered because they best assist with the physical understanding of the stresses due to the presence of a temperature difference across the structure. However, the shear stresses must also be considered since these stresses are used together with the direct stresses in order to calculate the probabilities of glazing failure. Following a common notation for stresses, the direct external glass surface stresses parallel to the x-axis and the y-axis can be respectively expressed as σ_{xx} and σ_{yy} . The corresponding shear stress is expressed as σ_{xy} (this shear stress is symmetric with σ_{yx}) [10].

Figures 6.31a and b present results for the shear stress, σ_{xy} , at the warm and cold side external glass surfaces for the same dimensions and properties given for Figs. 6.26. A detailed analysis of these plots is not required, but there are two points about the shear stresses that are important to note for the probability of failure calculations of Chapter 8. Firstly, unlike the direct stresses presented in Figs. 6.26, the shear stresses are not fairly uniform along most of the edges. Secondly, the shear stresses are much smaller in magnitude than most of the direct stresses.

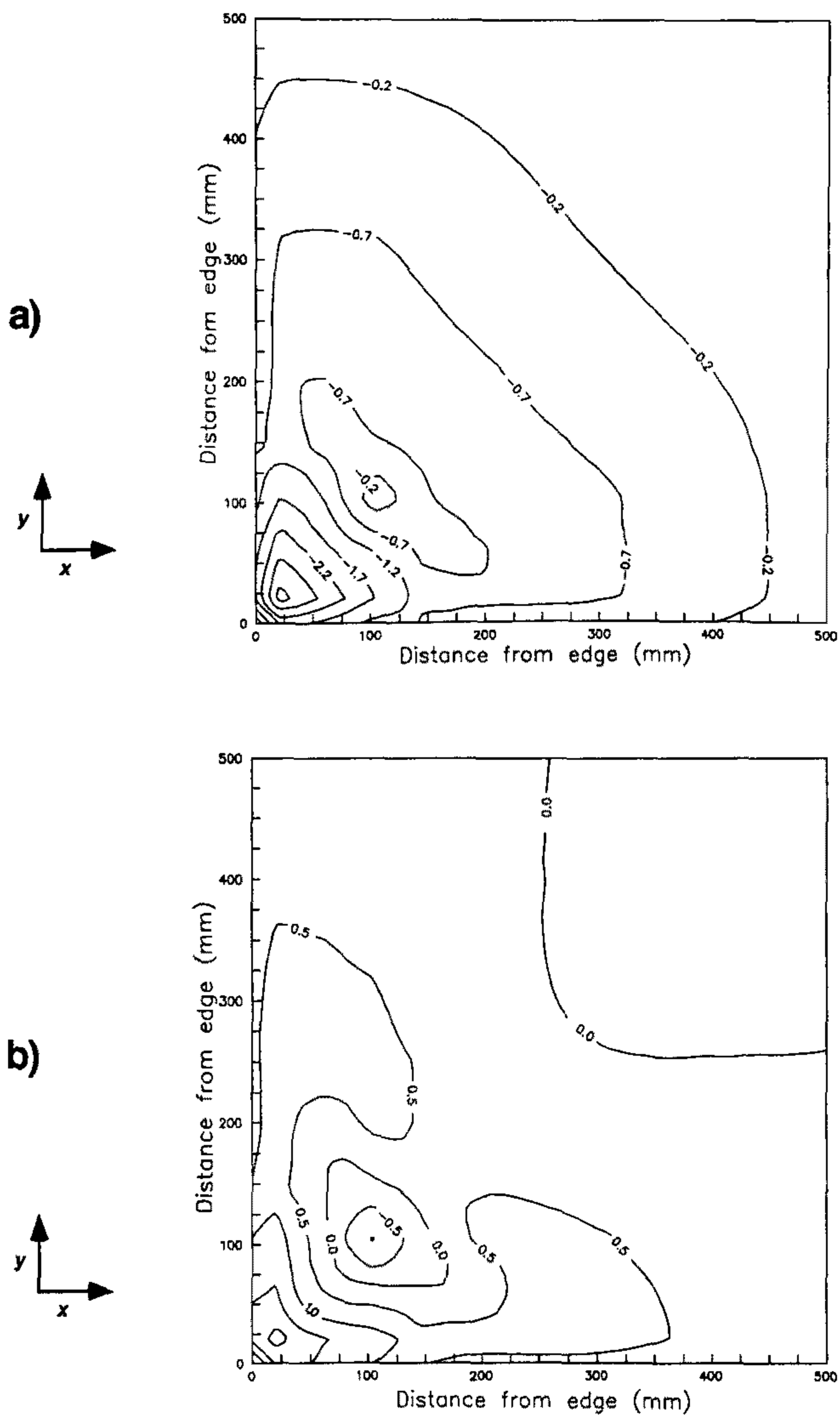


Fig. 6.31: Contour plots of the external surface shear stress (σ_{xy}) for both sides of a quarter section of a 1m by 1m glazing, and for the ASTM standard test conditions and $U_{g-g} = 1.10 \text{ Wm}^{-2}\text{K}^{-1}$: (a) warm side, and (b) cold side. The edges of the glazing (i.e. the outer edges of the model) are along the labelled borders, and the glazing corner is at the lower-left corner of the plots. The contour interval for both plots is 0.5 MPa.

6.7 Conclusion

Since the probability of failure for a vacuum glazing is a function of the stresses in the structure, it is important to consider the sources and magnitudes of these stresses. One source of glazing stress, the presence of a temperature difference across the glazing, has been studied in this chapter. A temperature difference across the glazing results in a differential thermal expansion and contraction of the two glass sheets. This can cause the sheets to bend, thereby leading to stresses within the glass.

A finite element model has been developed for the structural analysis of a vacuum glazing which is exposed to a glass-to-glass temperature difference. The solution procedure relies on the determination of the node temperatures with a corresponding thermal analysis model which has elements for the simulation of the internal and external heat transfer processes.

In order to obtain data with which to validate the model, an apparatus was constructed for the measurement of warm side surface deflections and strains under well defined conditions. In this apparatus, a 0.5 m by 0.5 m sample of vacuum glazing was supported in an essentially unconstrained manner between isolated warm and cold cells. The cell temperatures were maintained by water flows in the plates which covered the cells. The deflections were measured with a dial micrometer, for which the reference surface was a bar coupled to the warm side surface of the glazing via a three-point mounting configuration; this configuration effectively preserved the alignment between the bar and glazing without constraining the bending of the glazing. Surface strains were measured with strain gages which were connected into bridge circuits. Compensating gages were utilized in order to minimize the effects of the temperature dependencies of the gages.

In general, there is a good agreement between the experimental and finite element modelling results for both the deflections and the strains. It is clear that the deflection measurements constitute the most sensitive test of the model since the accuracy of these data is much greater than that of the strain measurements (e.g. 0.5 % versus 3 % for the respective minimum errors). However, to within the accuracy obtained, most of the deflection and strain measurements agree with the modelling results, and all of these measurements are qualitatively consistent with the trends predicted by the model.

The effects of edge insulation on the deflections and strains in vacuum glazing were investigated. An accurate comparison between the experimental and modelling results for the insulated case was not possible due to the uncertainties in the thermal conductance presented by the insulation. However, the measurements are roughly consistent with the modelling data in showing that the addition of edge insulation to the warm side of the glazing decreases the overall glazing deflection, and that it increases the warm side parallel tensile strain at the edge. Modelling results show that the addition of such insulation decreases the warm side center-of-glass strain. The addition of insulation to both sides of the edge decreases the difference between the mean and edge glazing temperatures of the glazing. This results in a reduction in the warm side parallel tensile stress at the edge.

Experimental and finite element modelling results were compared with data obtained from a previously developed simple analytic model for determining the temperature induced edge-to-center deflections, and the strains at the edge and at the center of a vacuum glazing. The comparisons indicate that the analytic model correctly predicts that there should be a linear relationship between these quantities and the temperature difference across the glazing. However, this simple model provides only approximate results for the deflections and strains (in fact, it overestimates these quantities) because it does not include the influence of several effects associated with the edge of the glazing.

On the basis of the good agreement between the experimental and finite element modelling results, it is concluded that the above mentioned finite element model for the structural analysis of a vacuum glazing has been validated. It is therefore appropriate to use this model to determine the stresses in vacuum glazing under realistic conditions (such as the ASTM standard test conditions) for a probability of failure analysis. There is one other issue that must be addressed before beginning such a failure analysis. Since the probability of failure is a function of all of the stresses in the glazing, it is first necessary to consider the other source of stress in vacuum glazing: atmospheric pressure. This is the subject of the next chapter.

6.8 List of Symbols

d	maximum amount of bending between the middle of one edge and the center of a square sample of vacuum glazing
E	Young's modulus
GF	strain gage gain factor
h_i	thermal conductance equal to the series combination of the insulation conductance and the warm side heat transfer coefficient.
L	dimension of a square sample of vacuum glazing
M	torque
R_o	reference resistance of strain gage
s	net glass deflection
s_o	null glass deflection
s'	glass deflection with a temperature difference across the glazing
t	thickness of glass sheet
T_{ave}	average of internal glass surface temperatures
T_c	temperature of compensating strain gage
$T_{c, wires}$	temperature of wires to compensating strain gage
T_{edge}	temperature of glass at edge of glazing
T_{glass}	local glass temperature
T_m	temperature of measurement strain gage
T_{mean}	mean temperature of entire glazing
$T_{m, wires}$	temperature of wires to measurement strain gage
T_o	reference temperature
U_{g-g}	glass-to-glass center-of-glass U-value
U_{rad}	radiative U-value
w	insulation recess depth

Greek symbols

α	coefficient of thermal expansion
ϵ	strain
ν	Poisson's ratio
σ	external glass surface stress
$\sigma_{bending}$	stress due to the bending of the glazing
σ_{edge}	external glass surface stress at the edge of the glazing
σ_{xx}, σ_{yy}	direct stress
σ_{xy}, σ_{yx}	shear stress
ΔR	change in strain gage resistance
ΔT_{g-g}	external glass-to-glass temperature difference across the glazing (i.e. the temperature difference between the external glass surfaces)

6.9 References

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7. Stresses Due to Atmospheric Pressure

7.1 Introduction

Since the probability of failure for a vacuum glazing is dependent on all of the stresses in the structure, it is important to consider both the stresses due to the presence of a temperature difference across the glazing, and the stresses caused by atmospheric pressure. The former stresses were studied in the previous chapter, and the latter stresses are the subject of this chapter. The results from both of these studies are combined for the failure analysis of the next chapter of this thesis.

The modelling of atmospheric pressure stresses can be greatly simplified by taking advantage of the symmetry of the glazing, and of the uniform periodicity of the stresses. A simple model is presented which gives results which are comparable to those obtained with a more exact fully three-dimensional simulation. The simple model is then used to calculate the stresses due to atmospheric pressure for different pillar spacings and material properties.

It is important to note that the main purpose of this study is to demonstrate the general method for calculating the atmospheric pressure induced stresses rather than to provide a comprehensive set of stress results. Only typical pillar properties and dimensions are simulated in order to obtain results for the probability of failure study in the next chapter of this thesis.

7.2 Stresses Due to Atmospheric Pressure

After the evacuation of the internal gap of a vacuum glazing, there is a pressure difference of about 10^5 Pa across each sheet of glass. The consequent forces are supported by the pillars and the edge seal. Atmospheric pressure therefore leads to the following stresses identified in Fig. 7.1: compressive stresses in the pillars, stresses associated with the indentation of the pillars into the internal glass surfaces, and localized tensile stresses at the external glass surfaces above the pillars and near the edge seal. Atmospheric pressure also causes the sheets to come together from their pre-evacuation positions by a distance which depends on the two processes which are indicated in Fig. 7.1: the compressive deformation of the pillars from their original height (which is the same as that of the fused edge seal), and the indentation of the

pillars into the glass sheets. Both of these processes affect the stresses due to atmospheric pressure.

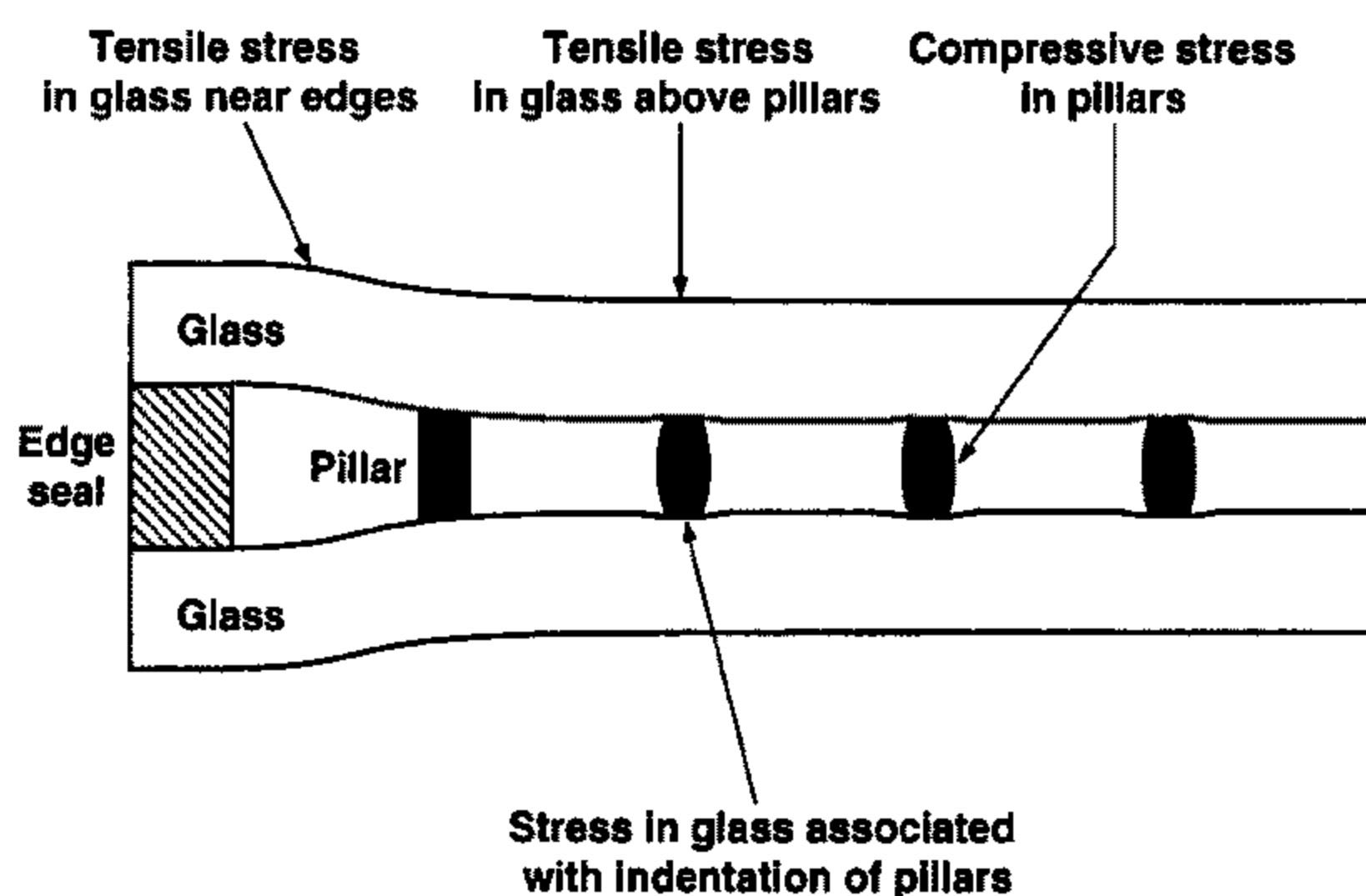


Fig. 7.1: After evacuation, the two glass sheets of a vacuum glazing come together under the influence of atmospheric pressure, leading to the indicated stresses. The pillars are compressed and they indent into the inner glass surfaces. The pillars closest to the edge are compressed by only a small amount (before evacuation, all of the pillars are as high as the fused edge seal). The variation in the deformation across the external glass surfaces is much less than that at the internal surfaces, and thus it is not shown.

In the glass at and around the points of contact between the pillars and the glass sheets, there are large stresses associated with the indentation of the pillars into the internal glass surfaces. This indentation is mostly elastic, although there is a small amount of plastic deformation of the glass (which is about three orders of magnitude less than the elastic deformation) [1]. Thus, the extent of the indentation is governed by the Young's modulus and, to a much lesser extent, by the hardness of the glass. The associated stresses are similar to those which occur in classical Hertzian indentation experiments in which a hard sphere is pressed into the surface of a brittle material and causes a conical crack to form [2]. This is referred to as conical indentation fracture. Under the right conditions, such a crack can grow and ultimately lead to structural failure. Auerbach's law [3], which gives the relationship between the radius of a spherical indenter and the force required to initiate a crack, can be used to calculate the minimum force necessary for a cone crack to form. For vacuum glazing, this minimum force is greatly increased by the friction between the pillar and the glass, and by the virtual absence of water vapour in the evacuated space (water is known to assist with crack propagation in glass [4]). Thus, for typical glazing designs, the large internal surface stresses associated with pillar indentation have never been

observed to lead to cone cracks - or to the structural failure that such cracks can cause [5]. Also, these stresses have never been observed to lead to failure at the small cracks in the glass near the corners of square shaped pillars (such cracks are caused by stress concentrations at these locations) [5, 6], or at the small crescent shaped cracks that can form in the glass around both square and circular shaped pillars [6]. The latter cracks can be caused by the shear forces in the glass near the pillars that result from the non-uniform heating of one or both glass sheets or, less commonly, from the complex or large bending of the glazing [6]. Since the internal surface stresses around the pillars do not lead to the structural failure of glazings, and since the results from this study will be used for the probability of failure calculations of the next chapter, these internal surface stresses can be safely ignored. Therefore, of the stresses in the vicinity of the pillars, only the stresses at the external glass surfaces (which are smaller than the stresses at the glass internal surfaces) need be considered.

The external glass surface stresses around the pillars, which are tensile, result from the elastic bending of the glass sheets over the pillars. The maximum external surface stress, which occurs directly above the pillars, can be estimated with an analytic model given by Timoshenko and Woinowsky-Krieger [7]. In this model, the maximum isotropic tensile stress at the center of a circular plate (i.e. a disk) with a clamped edge and a concentrated force, F_d , at the center is:

$$\sigma = (F_d / t^2)(1 + \nu)(0.485 \ln(a/t) + 0.52) , \quad (7.1)$$

where t is the disk thickness, ν is the Poisson's ratio of the disk material, and a is the disk radius. The stress corresponds to the surface of the disk which is on the opposite side of the applied force. The stress above a pillar in a vacuum glazing can be estimated from this relationship by assuming that the disk and central load represent a single glass sheet of thickness t , and the reaction force above a central support pillar. For this estimation, the area of the disk must be set equal to the area of a square unit cell of the glazing (which is of dimension λ) in order to ensure that the central force on the disk is equal to the force on the unit cell due to atmospheric pressure, P (i.e. $\pi a^2 = \lambda^2$ so that $F_d = P\lambda^2$). A limitation of this analytic approach is that it does not simulate the uniform load on the external surface of a vacuum glazing [8].

Since the edge seal is more rigid than the pillars, the pillars near the seal are compressed less, and thus exert smaller forces on the glass sheets, than the pillars elsewhere in the glazing [5]. Therefore, away from the edge, there is greater pillar deformation and indentation. Consequently, as shown in Fig. 7.1, the glass sheets are

closer together in the region away from the edge than at the comparatively rigid edge seal of fixed height. This causes a significant bending of the glass sheets near the seal and thus large tensile stresses at the convex external glass surfaces above the inner edge of the seal. Due to the curvature of these surfaces with respect to the edge of the glazing, the associated tensile stresses are mostly oriented normally to the edges. These stresses extend for a short distance towards the center of the glazing. Whereas the pillar deformation (and thus the edge stresses) can be reduced by using pillars of an appropriate strength, it is not possible to reduce significantly the pillar indentation (and the edge stresses) by using certain types of glass for the glazing. This is because there is not much variation in the Young's modulus and hardness among the glasses that are suitable for vacuum glazing; tempered glass is unsuitable because, for typical glass sheet separations, the large variations in the thickness of this type of glass can lead to contact between the sheets across the evacuated space (a larger than typical separation was used for the measurements on tempered glass which were described in Chapter 3). The use of strong pillars is also important in order to ensure that the large compressive stresses within the pillars do not lead to the structural failure of the supports. Since the pillars currently used in vacuum glazing are of a sufficient strength to prevent such failure, the compressive stresses in the pillars are not considered in this study.

Clearly, if the thickness of the solder glass seal could be adjusted to match the pillar deformation and indentation that occurs during evacuation then, in response to the coming together of the glass sheets, there would be no bending of the sheets near the edges. This would prevent the establishment of the tensile external surface stresses near the edge seal. Theoretically, this could be accomplished by evacuating the glazing at a high temperature during the manufacturing process while the solder glass is soft. However, in order to obtain an adequate internal vacuum, the solder glass must be completely fused - and thus fixed in thickness - prior to evacuation [5].

For the stresses near the edge seal, only the external glass surface stresses need be considered. This is because, as explained in Chapter 8, there is a low probability of glazing failure from the corresponding internal surface stresses due to the low number of surface flaws, and due to the virtual absence of water vapour at these isolated internal surfaces.

7.3 Finite Element Model

7.3.1 Modelling Approach

Two requirements must be met for the accurate simulation of the stresses due to atmospheric pressure. Firstly, in order to obtain results for the localized stresses around the pillars, it is necessary to simulate individual pillars; unlike with the previously presented thermal and structural models for vacuum glazing, the pillar array can not be represented as an inter-pane slab with equivalent uniform bulk properties. Secondly, the model must be large enough to ensure that the results for pillars away from the edges of the simulated glazing are not affected by the boundary conditions at these edges. Meeting both of these requirements can result in a model with an impracticably large number of nodes - unless several simplifications are made.

One such simplification involves the representation of the glass sheets and the pillars. As shown in Fig. 7.2a, the situation to be modelled is two soda-lime glass sheets of finite thickness separated by pillars of finite height and cross-section. Since atmospheric pressure acts equally on both symmetric sides of the structure, thereby leading to the same deformations and stresses on these sides, only a half-thickness of the glazing need be modelled. Therefore, it is sufficient to simulate one sheet of glass supported by half-height pillars which are set on a partially fixed plane which represents the mid-plane surface of symmetry of the glazing. As discussed below, this requires the specification of certain freedom conditions for the nodes at the pillar mid-heights which are on this surface of symmetry. The total number of nodes can be further reduced by simulating the glass sheet with thin shell plate elements. These are two-dimensional elements which simulate the behaviour of three-dimensional brick elements based on assigned membrane and bending thicknesses. For thin shell plate elements, results can be obtained for the equivalent lower, middle, and upper surfaces of the simulated three-dimensional structure (which, in this case, is the glass sheet). Another reduction in the number of nodes can be accomplished by representing the pillars as one-dimensional beam elements with an appropriate stiffness. This is justified on the basis that, as explained below, the shear stresses in the pillars are not significant. By using the above modelling approaches, the actual situation of Fig. 7.2a can be simulated by the arrangement shown in Fig. 7.2b.

Only square glazings are considered in this study. As with the previously described models for vacuum glazing, the present model can be simplified even more by making use of the lateral surfaces of symmetry of the full square glazing. Due to the

symmetry of the square glazing, it is sufficient to simulate only a quarter-section of the structure. However, unlike the previous models, the quarter-section can correspond to a glazing of less than the actual full size. Due to the uniform periodicity of the stresses around the pillars far from the edges, and due to the uniformity of the periodic stresses along the edges, the modelled quarter-section need only be large enough to obtain representative results for these regions; the stresses beyond the modelled dimensions can be assumed to be the same as these representative regional results. It is also possible to economize on the nodes by grading them so that the node density is greatest at the locations of the largest stress gradients: around the beams, and near the glazing edges. It is noted that results could also be obtained from a one eighth-section of glazing.

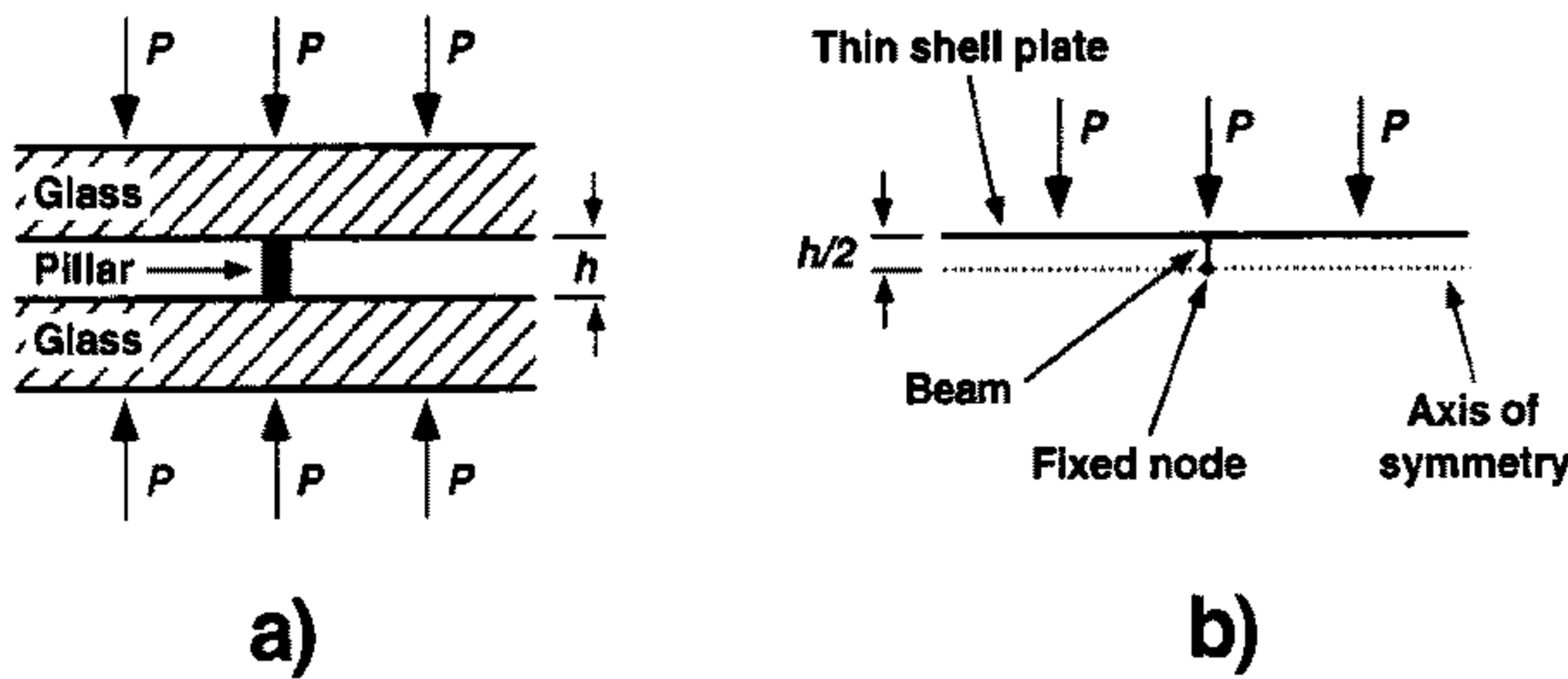


Fig. 7.2: (a) The situation to be modelled is a glazing consisting of two sheets of glass of finite thickness separated by finite-sized pillars of height h . Each external surface is subjected to atmospheric pressure, P . (b) The glass and pillar components can be simulated with thin shell plate and beam elements, respectively. Only a half-thickness of the glazing need be modelled.

Figure 7.3 shows the geometry of the finite element model constructed with STRAND6. The square quarter-section of glazing is six unit cells in length. Each unit cell away from the outer edges has a beam (or actually the equivalent of a quarter-section of a beam which is common to the adjacent cells) in each of the four corners. Beam elements of height 0.1 mm represent the half-pillars (i.e. the corresponding total pillar height is 0.2 mm). The beam separation is λ . The outer edges of the model represent the edge seal and thus the model simulates a 6 pillar by 6 pillar array. A pressure equal to atmospheric pressure (101.3 kPa) is applied in the negative z -direction to each thin shell plate element.

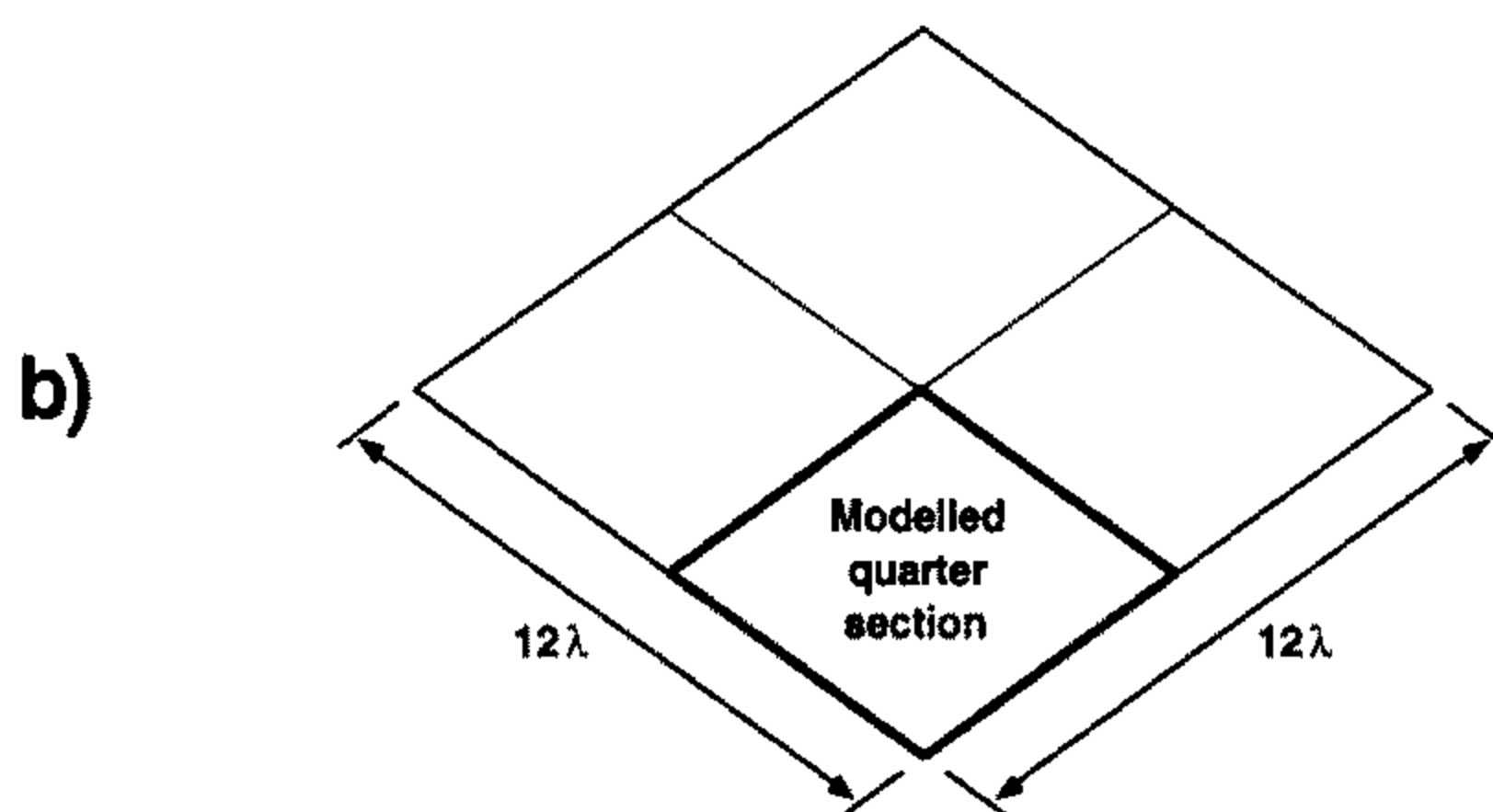
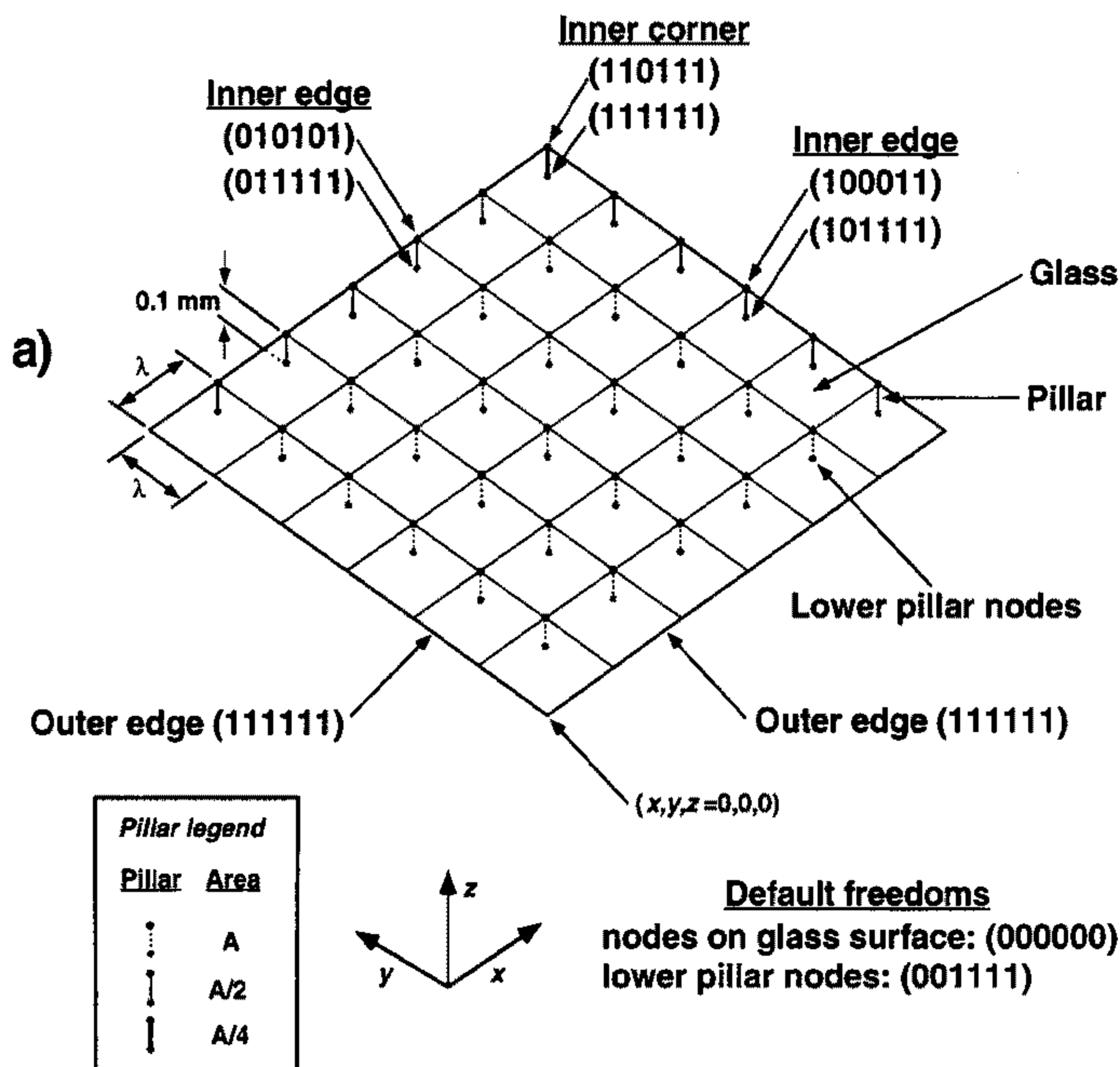


Fig. 7.3: (a) The geometry of the finite element model for the analysis of the stresses due to atmospheric pressure. A surface of thin shell plate elements (a glass sheet) is supported by beam elements (pillars) which are of height 0.1 mm and which are separated by a distance of λ . The freedom conditions for the nodes are as indicated, where "0" and "1" refer to free and fixed, respectively. The edge seal is simulated with fully fixed nodes. The simulated pillar (beam) cross-sectional areas are also indicated. The square unit cells are outlined. (b) Due to the symmetry of the full square glazing, only a quarter-section of the structure need be modelled.

It is recalled from Section 6.3 that the node freedom conditions are represented as a series of six numbers - either "0" for free or "1" for fixed - which correspond to the DX, DY, DZ, RX, RY, and RZ settings; DX refers to translation in the x-direction, RX refers to rotation about a line parallel to the x-axis etc. The default conditions for nodes on thin shell plate elements are the fully free conditions: (000000). For nodes on thin shell plate elements which are neither fully free nor fully fixed, the rotational conditions (the last three settings) are the opposite of the translational conditions (the first three settings) for the corresponding axes.

Since, relative to the pillars, the solder glass edge seal of a vacuum glazing is very rigid, the glass bonded with the seal can be assumed to experience only a negligible deflection due to atmospheric pressure [5] - i.e. this glass can be assumed to be completely fixed. Thus, the structure of the edge seal and the glass bonded with the seal is essentially equivalent to a built-in or clamped edge around the glazing. It is therefore possible to simulate the edge seal by fully fixing the nodes at the outer edges of the model - i.e. by assigning freedom conditions of (111111) to these nodes. Like the finite difference model for edge conduction that was presented in Section 5.2.2, this approach does not simulate an actual edge seal of finite width. However, it does provide appropriate results for the external glass surface stresses from the center of the glazing to right above the inner edge of such a seal. Due to the built-in nature of the edge around the glazing, the external glass surface stress over the edge seal changes from being a mostly direct (and tensile) stress right above the inner edge, to being an almost entirely shear stress (which rapidly drops to near-zero over the seal). As a result of this transition in the nature of the stress, the direct stress falls from the value above the inner edge of the seal to near-zero over the seal. These drops in the shear and direct stresses can be expected to be exponential with distance from the inner edge and, due to the rigidity of the bonded edge structure, to occur over a very short distance; they can therefore be approximated as step functions. Thus, in this study, the tensile and shear stresses in the glass over the edge seal are assumed to be zero. As shown in Chapter 8, such simple approximations of the stresses over the seal do not significantly affect the probability of failure calculations (which are based on the stresses over the entire glazing). It is noted that, as shown in the previous chapter, the bending caused by a temperature difference across the glazing results in large tensile stresses over the edge seal. Since atmospheric pressure does not cause a similar overall bending of the glazing, it does not lead to such stresses at this location.

For glass surface nodes at the inner edge of the model which is parallel to the y-axis, the freedom conditions are (100011). The reasons for the three translational

conditions (the first three conditions) are the same as those given for the translational conditions of the nodes on the analogous axis of symmetry for the structural model presented in Section 6.3. As mentioned above, since these nodes of the present model are neither fully free nor fully fixed, the rotational conditions (the last three conditions) are the opposite of the translational conditions for the corresponding axes. Similarly, for glass surface nodes on the other inner edge (the edge which is parallel to the x-axis), the freedom conditions are (010101). The glass surface node at the inner corner must have the same fixed freedoms as the adjoining inner edges: (110111).

Due to the symmetry on each side of the mid-plane of the glazing, the nodes representing the pillar mid-heights must not move in the vertical direction (i.e. along the z-axis) of Fig. 7.3. However, these nodes must be able to move laterally in order to match the translation associated with the bending of the glass sheet. If this were not the case, there would be unrealistic - and very large - shear forces in the simulated pillars (i.e. the beams). Thus, for the lower nodes of the beam elements, the freedom conditions are (001111) - except along the inner edges and at the inner corner where, as indicated in Fig. 7.3, the DX and DY translational conditions match the above described symmetry of the model.

Although the simulated pillars are one-dimensional beam elements, they must have an assigned cross-sectional area - which is chosen to be circular. Since the lateral axes of symmetry (the inner edges of the model) intersect pillar rows, there are different circular cross-sectional areas for the beams of this model, as shown in Fig. 7.3. For the beams not on the edges of the model, the cross-sectional area is equal to that of the pillars, A , which is defined by the specified pillar radius. For the beams along the lateral axes of symmetry, the cross-sectional area is $A/2$, and for the beam at the intersection of these axes (at the inner corner) the area is $A/4$. The selection of the material properties for the beams is discussed below.

In order to simulate a glass sheet of 4 mm thickness (which is the only glass thickness considered in this study), the membrane and bending thicknesses of the thin shell plate elements are set at 4 mm. Except where otherwise specified, the Young's modulus of elasticity and the Poisson's ratio for these elements are equal to the corresponding values given in Appendix 1 for soda-lime glass: $E=7.38 \times 10^{10}$ Pa, and $\nu=0.21$.

7.3.2 Pillar-Glass System Stiffness

Fischer-Cripps developed a fully three-dimensional finite element model for the analysis of the deflections and stresses in a unit cell of a vacuum glazing which are caused by atmospheric pressure [5, 9]. The model consists of a half-thickness and quarter-section of a single square unit cell of stainless steel pillars. This model is referred to as the "fully" three-dimensional model because both the glass sheet and the pillar are represented by three-dimensional brick elements - as opposed to the model described above which has a three-dimensional arrangement of two-dimensional thin shell plates and one-dimensional beams. As with the plate and beam model, the model developed by Fischer-Cripps simulates the actual situation with only one sheet of soda-lime glass and a half-height section of a pillar. All glass and pillar deformations are assumed to be elastic, and thus the fully three-dimensional model simulates only elastic deformations. The material properties for this model are given in Table 7.1 (the origins of the glass material properties, which differ from the referenced values listed in Appendix 1, are unknown).

Property	Soda-lime glass	Stainless steel
Young's modulus (10^{10} Pa)	7.00	21.0
Poisson's ratio	0.22	0.33

Table 7.1: The material properties used by Fischer-Cripps for the fully three-dimensional model [9].

As previously explained, the distance by which the two glass sheets come together due to atmospheric pressure depends on the compressive deformation of the pillars, and on the indentation of the pillars into the glass sheets. Figure 7.4c shows finite element modelling results obtained by Fischer-Cripps from the fully three-dimensional model described above [5, 9]. The modelled glass thickness is 4 mm. These results are for stainless steel pillars for which the pillar radius, half-height and separation are 0.125 mm, 0.1 mm and 25 mm, respectively. Fischer-Cripps obtained results for other pillar radii and separations but only data for these ones are considered in the present study. The net deflections (deformations) of the internal and external glass surfaces due to atmospheric pressure are presented as functions of the distance from a pillar along what would be a line between adjacent pillars in a multiple pillar simulation (i.e. as functions of the distance over an equivalent row of pillars). As shown in this figure, the total deflection of the external glass surface immediately above the pillar ($d_{total}=-3.41\text{ }\mu\text{m}$) is the sum of the deflection due to

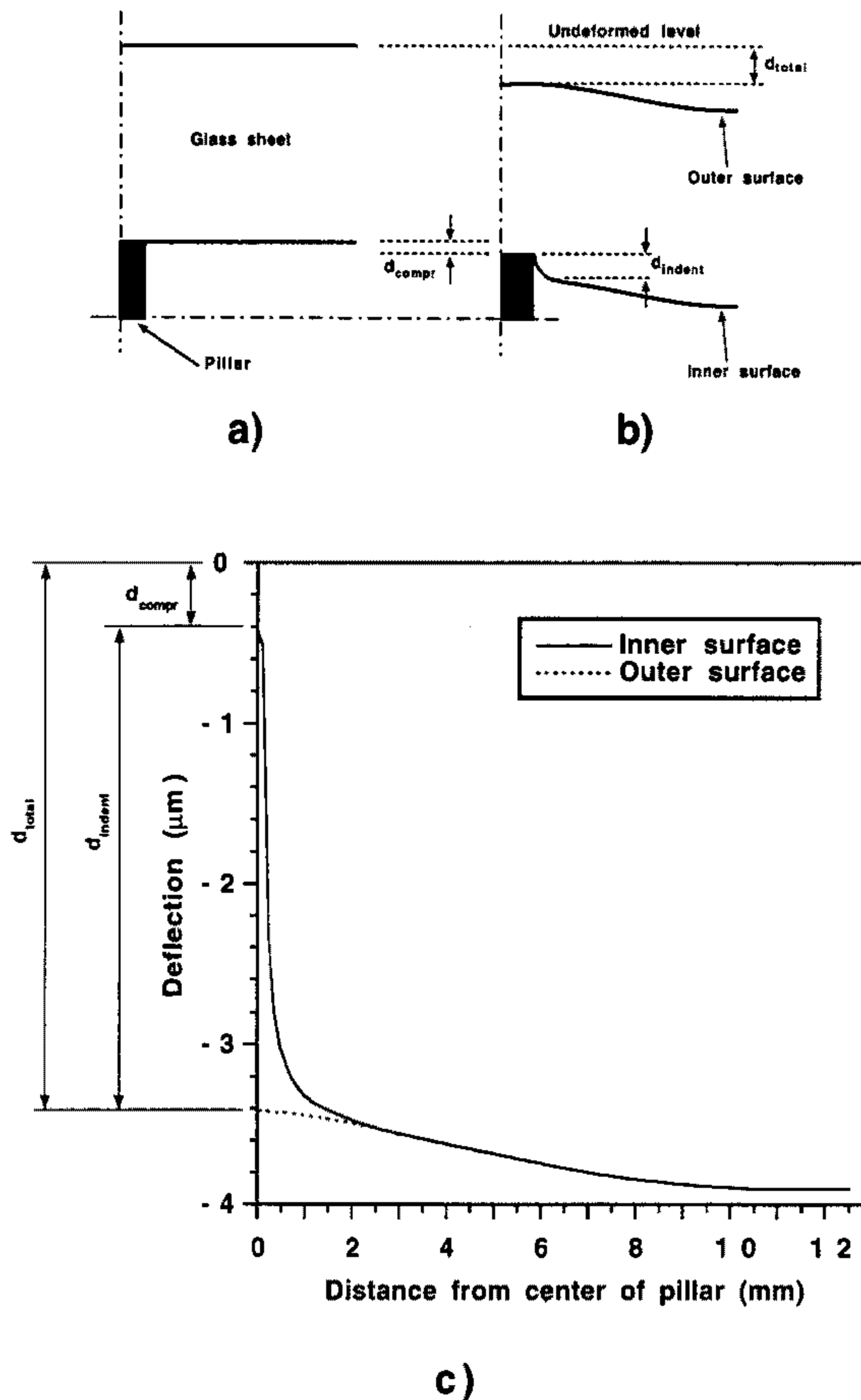


Fig. 7.4: (a) A cross-section of an undeformed glass sheet and a half-height, half-width pillar prior to evacuation. (b) After evacuation, the glass immediately above the pillar deflects by a distance d_{total} which is the sum of the deflection due to the compressive deformation of the pillar, d_{compr} , and the deflection due to the indentation of the pillar into the glass, d_{indent} . (c) These distances are shown in the modelling results of Fischer-Cripps for the deflections of the internal and external (i.e. inner and outer) glass surfaces along a line between adjacent pillars of a unit cell [5, 9]. Beyond about 3 mm from the pillar, the profiles for the two surfaces are coincident. These results are for stainless steel pillars of radius, half-height and separation of 0.125 mm, 0.1 mm and 25 mm, respectively. The glass thickness is 4 mm. The properties for the pillar and glass elements are given in Table 7.1.

the pillar compression ($d_{compr}=-0.39 \text{ }\mu\text{m}$), and the deflection due to the pillar indentation ($d_{indent}=-3.02 \text{ }\mu\text{m}$).

For the thin shell plate and beam element model described above, the stiffness of the simulated pillars (i.e. the beam elements) must reflect the separate deflections due to the deformation and indentation of the pillars so that, far from the outer edges of the model, the correct total deflection, d_{total} , is obtained above the pillars. Since the pillar deformation and indentation components of this total deflection depend on the pillar and glass material properties, the beam stiffness essentially represents the stiffness of the pillar-glass system. The stiffness coefficient for each beam is:

$$k_{beam} = \frac{E_{beam}A_{beam}}{h_{beam}} , \quad (7.2)$$

where E_{beam} , A_{beam} and h_{beam} are the Young's modulus, cross-sectional area and height of the beam. The beam stiffness is set by specifying these parameters within the model. It is recalled that the beam cross-sectional area depends on the beam location and that it is either A , $A/2$ or $A/4$ (where A is equal to the cross-sectional area of each beam not on an inner edge of the model).

Both the beam stiffness coefficient and the beam Young's modulus can be derived from experimental or other modelling results as follows. Each beam can be thought of as a spring, with a certain spring constant, below the single simulated glass sheet. Each unit cell of the glass sheet has an area equal to the square of the pillar separation, λ , and thus it is subjected to a force, F , of:

$$F = P\lambda^2 , \quad (7.3)$$

where P is the atmospheric pressure. Since each square unit cell contains a quarter-section of each of the four corner simulated pillars, there is the equivalent of one beam per unit cell. Thus, the force supported by each beam is equal to F . Therefore, by Hooke's law, the beam spring constant (i.e. the beam stiffness coefficient) is:

$$k_{beam} = \frac{F}{-d_{total}} , \quad (7.4)$$

where d_{total} is the total negative deflection of the external glass surface above the pillar - as described above.

Assuming that the deformation of the pillar-glass system is elastic:

$$E_{beam} = \frac{\sigma}{\epsilon}$$

$$= \frac{F/A_{beam}}{-d_{total}/h_{beam}}, \quad (7.5)$$

where F is defined by equation 7.3. This expression is equivalent to equation 7.2. For $\lambda=25$ mm, a pillar radius of $a=0.125$ mm, $d_{total}=-3.41$ μm and $h_{beam}=0.1$ mm, the Young's modulus for a beam not on the inner edges of the model (i.e. for $A_{beam}=A$, where $A=\pi a^2$) is $E_{beam}=3.79 \times 10^{10}$ Pa.

One other material property to be specified for the beam elements is the beam shear modulus, G_{beam} :

$$G_{beam} = \frac{E_{beam}}{2(1+\nu)}, \quad (7.6)$$

where ν is the beam Poisson's ratio. The modelling results are essentially independent of the exact value of ν chosen for the beams. This is because the shear stresses in the beam elements should be insignificant due to the geometry and the Young's modulus of the beams, and due to the freedom conditions at the lower nodes of the beams. This is also true for the shear stresses within actual pillars. Since most of the pillar-glass system deflection is due to the indentation of the pillar into the glass ($d_{indent} > d_{compr}$, as shown in Fig. 7.4c), and since this indentation is mostly governed by the material properties of the glass, the beam Poisson's ratio is assumed to be equal to that of glass. In this case, since the modelling results obtained with this set of beam properties are compared with those from the fully three-dimensional model, the Poisson's ratio for glass is taken to be the same as the one used for the fully three-dimensional model: 0.22. Substituting the above values of E_{beam} and ν into equation 7.6 gives $G_{beam}=1.56 \times 10^{10}$ Pa.

The above values for E_{beam} and G_{beam} are for beams which are not on the inner edges of the model - i.e. for beams with a cross-sectional area equal to that of the pillars ($A_{beam}=A$). Other sets of moduli must be calculated for the beams on the inner edges, including the beam at the inner corner of the model, on the basis of the corresponding beam cross-sectional areas of $A/2$ and $A/4$. Only the moduli for beams which are not on the inner edges are recorded below.

The beam properties calculated above relate to a pillar-glass system with stainless steel pillars. Stainless steel was used for pillars in early designs of vacuum glazing, but current designs employ pillars made from stronger materials such as alumina (which is a ceramic). Stainless steel pillars are considered below, but only for a comparison with benchmark modelling results because no such results are available for other types of pillars. All other analyses in this chapter correspond to the stronger and currently used alumina pillars. A separate set of beam properties must be calculated for a system with these pillars.

Measurements of the pillar-glass system stiffness for alumina pillars were made by Dominique Jeker at The University of Sydney [10]. In Jeker's experiment, a square alumina pillar of dimension 0.27 mm and height 0.2 mm was placed between two 4 mm thick sheets of soda-lime glass. The deflection at one of the external glass surfaces above the pillar was measured for different compressive forces on the pillar-glass system. The results represent the total deformation of the pillar and the two sheets of glass due to the applied load. Typical data are presented in Fig. 7.5. The slope of the linear regression fit to the data, which is equal to the stiffness coefficient for the pillar-glass system, is $9.30 \times 10^6 \text{ Nm}^{-1}$. Since the plate and beam element model simulates only half-pillars and one sheet of glass, the modelled deflections should be one half of these measured results. Thus, the equivalent stiffness coefficient for the beams is twice the experimentally determined value - i.e. $k_{beam} = 18.6 \times 10^6 \text{ Nm}^{-1}$. The corresponding Young's modulus of elasticity for the beams is calculated from equation 7.5 in the manner described above, but with d_{total} determined from equations 7.3 and 7.4 with $k_{beam} = 18.6 \times 10^6 \text{ Nm}^{-1}$ and with the specified pillar separation. The circular cross-sectional area of the beams is set equal to the square cross-sectional area of the alumina pillars (the choice of beam cross-sectional shape does not significantly affect the results). For a square pillar of dimension 0.27 mm and full-height 0.20 mm, and for a pillar spacing of 25 mm, $E_{beam} = 2.55 \times 10^{10} \text{ Pa}$. For the calculation of the beam shear modulus, the Poisson's ratio is taken to be 0.21 - which is equal to the Poisson's ratio for soda-lime glass (as given in Appendix 1) and which is equal or nearly equal to several estimations of that for alumina (0.21 to 0.22 [11]). Thus, for this alumina pillar and glass system, the equivalent beam shear modulus, as calculated by equation 7.6, is $G_{beam} = 1.05 \times 10^{10} \text{ Pa}$. The above results for the beam Young's modulus and the beam shear modulus would also correspond to an alumina pillar with a cylindrical cross-sectional area equal to the area of the square alumina pillar considered above; the equivalent radius of such a pillar is 0.15 mm. In the next chapter, only cylindrical pillars of the more typical radius of 0.125 mm are

considered (together with the same typical height of 0.20 mm). However, since no experimental results are available for the determination of the stiffness of a pillar-glass system with pillars of this radius, it is assumed that the above moduli can also be applied to this system. Therefore, for an alumina pillar and glass system with pillars of radius and full height of 0.125 and 0.20 mm, respectively, it is assumed that $E_{beam} = 2.55 \times 10^{10}$ Pa and that $G_{beam} = 1.05 \times 10^{10}$ Pa. Since the results of Fischer-Cripps indicate that the stiffness of a pillar-glass system is approximately inversely proportional to the pillar radius [5], the error in this assumption could be about 20%. In the context of the main purpose of this study, which is to demonstrate the general method for calculating the atmospheric pressure induced stresses, this is an appropriate approximation.

The thin shell plate and beam element model simulates only elastic deformations because, as mentioned above, the beam properties are based on the assumption that the deformation of the pillar-glass system is elastic. Since, as previously noted, the deformation of the glass component of the system is essentially elastic, the validity of this assumption depends on the nature of the pillar compression. This assumption is not correct for a system with stainless steel pillars due to the significant plastic deformation of such pillars in vacuum glazing [6]. However, in this work, it is appropriate to assume that stainless steel pillars deform elastically. This is because, in order to validate the plate and beam model, results for stainless steel pillars are compared with those obtained from the fully three-dimensional model which simulates only elastic effects. It is desirable to base this validation on elastic deformations because the plate and beam model is later only used to obtain results for an elastic pillar-glass system: one with alumina pillars. Jeker's results show that, to within the experimental error of $\pm 1 \mu\text{m}$, the deformation of a system with alumina pillars is elastic over a wide range of loads [10]. This range includes all but one of the forces considered in the present study: 162 N. No data are given for this force of 162 N (which corresponds to the load on a unit cell with a pillar separation of 40 mm) but it is assumed to cause only elastic deformation of alumina pillars.

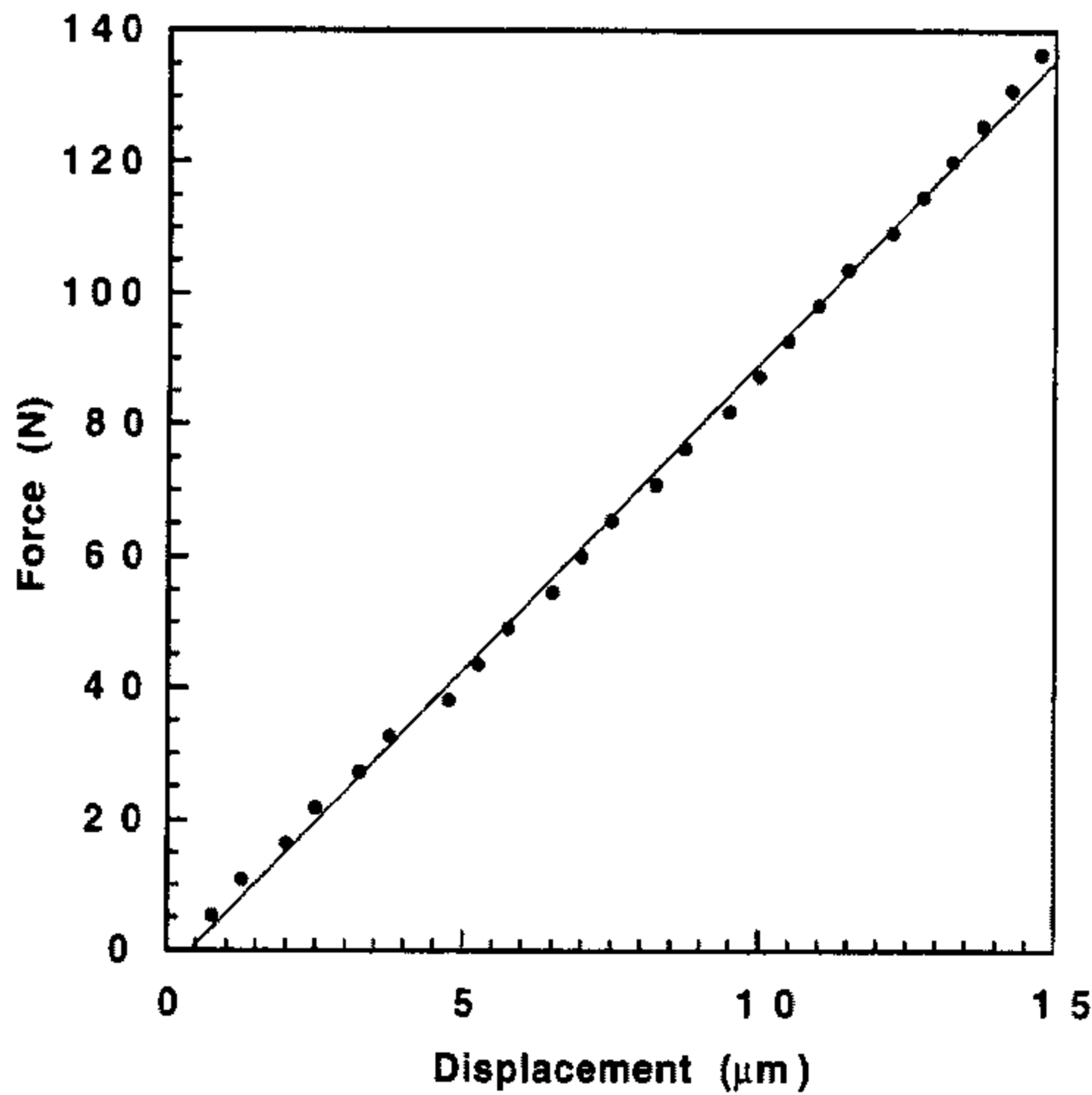


Fig. 7.5: Measurements made by Jeker for the relationship between the applied compressive force and the external glass surface deflection for a system of two sheets of 4 mm thick glass separated by a 0.2 mm high alumina pillar with a square cross-section of dimension 0.27 mm [10]. A linear regression fit to the data is shown. The data are accurate to $\pm 1 \mu\text{m}$.

7.3.3 Node Density

For the thin shell plate and beam element model, the node density must be sufficient to ensure agreement with results obtained from the more exact fully three-dimensional modelling approach. However, the total number of nodes must not exceed that which can be processed with the available computer memory (this type of structural model can require a significant amount of memory for the solution calculations). As noted above, the total number of nodes can be reduced by grading the nodes so that the node density is greatest at the locations of the largest stress gradients: around the beams, and near the glazing edges.

Fischer-Cripps used the above described fully three-dimensional model of a quarter-section of a unit cell to calculate the external glass surface stress along the half-diagonal of the cell. Stresses obtained with this model are within 3 % of results from the analytic approach given by equation 7.1, and within 2 % of results from a more approximate analytic solution (which corresponds to the stress in a disk with simply supported edges) [9]. Therefore, it is assumed that results from the fully three-dimensional model can be used as benchmarks for the theoretical validation of the

simpler thin shell plate and beam element model. For this validation, the properties of the thin shell plate elements are the same as those given in Table 7.1 for soda-lime glass ($E=7.00 \times 10^{10}$ Pa, and $\nu=0.22$). A plate shear modulus need not be specified. The pillar radius, half-height and separation are 0.125 mm, 0.1 mm, and 25 mm, respectively. The properties of the beam elements are as calculated above on the basis of the results presented in Fig. 7.4c for stainless steel pillars of the same dimensions and a Poisson's ratio of 0.22 ($E_{beam}=3.79 \times 10^{10}$ Pa, and $G_{beam}=1.56 \times 10^{10}$ Pa). The simulated glass thickness is 4 mm.

The thin shell plate and beam element model was run for several node densities - both with and without node grading around the pillars. The best agreement with the benchmark three-dimensional modelling results of Fischer-Cripps [5, 9] was obtained with 240 nodes per unit cell away from the outer edges of the model, and with these nodes graded from a spacing of 3.125 mm ($\lambda/8$) near the middle of each cell, to 0.125 mm ($\lambda/20$) around the pillars at the corners of each cell. The results obtained with this node density are compared with the benchmark results in Fig. 7.6a (the curve fit to the plate and beam modelling data is explained below). This figure presents the external glass surface stress parallel to the x-axis as a function of the distance from a pillar along adjacent half-diagonals of two unit cells. Since these results are also along the diagonals of the two symmetric models and since, as shown in Figs. 7.6b and c, the lateral axes of both models are parallel to the perpendicular edges of the unit cells, the results also represent the stresses parallel to the y-axis. For the plate and beam model, the external surface is defined as the equivalent upper surface of the simulated glass sheet. The results for this model are from unit cells which are far from the edges of the glazing. Since the fully three-dimensional model of a quarter-unit cell only provides results for along a half-diagonal of a cell, a mirror image of these results is also presented in order to give a complete curve.

In Fig. 7.6, there is a very good agreement between the two sets of results - except within ± 1.5 mm of the simulated pillar where the stress obtained from the simple plate and beam element model is much greater than that obtained from the fully three-dimensional model. It might be thought that this discrepancy could be resolved by simply increasing the node density around the beam. However, as explained in Appendix 4, due to the way in which the stress is calculated at the connection between the plate and beam elements, this has the opposite effect. Clearly some appropriate method must be used to determine the significant stresses close to the pillars. A common solution to such a problem is to estimate the results near the beams on the basis of the physical understanding of the situation, or on the basis of more

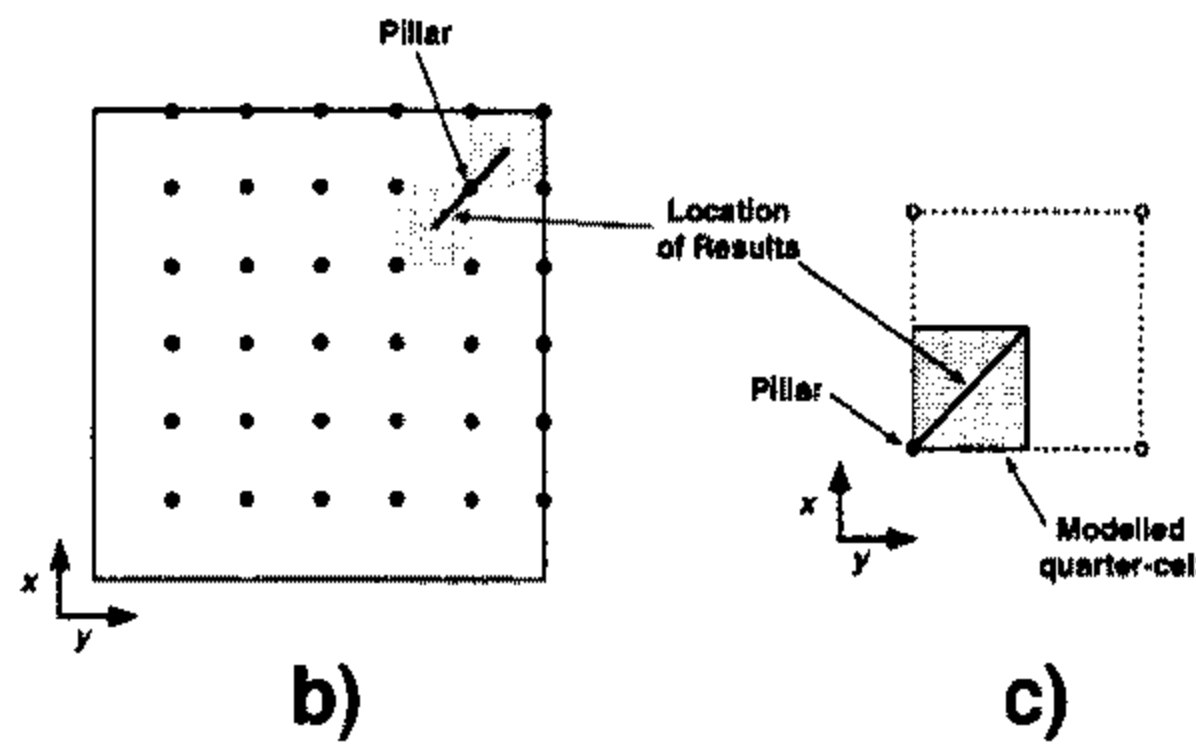
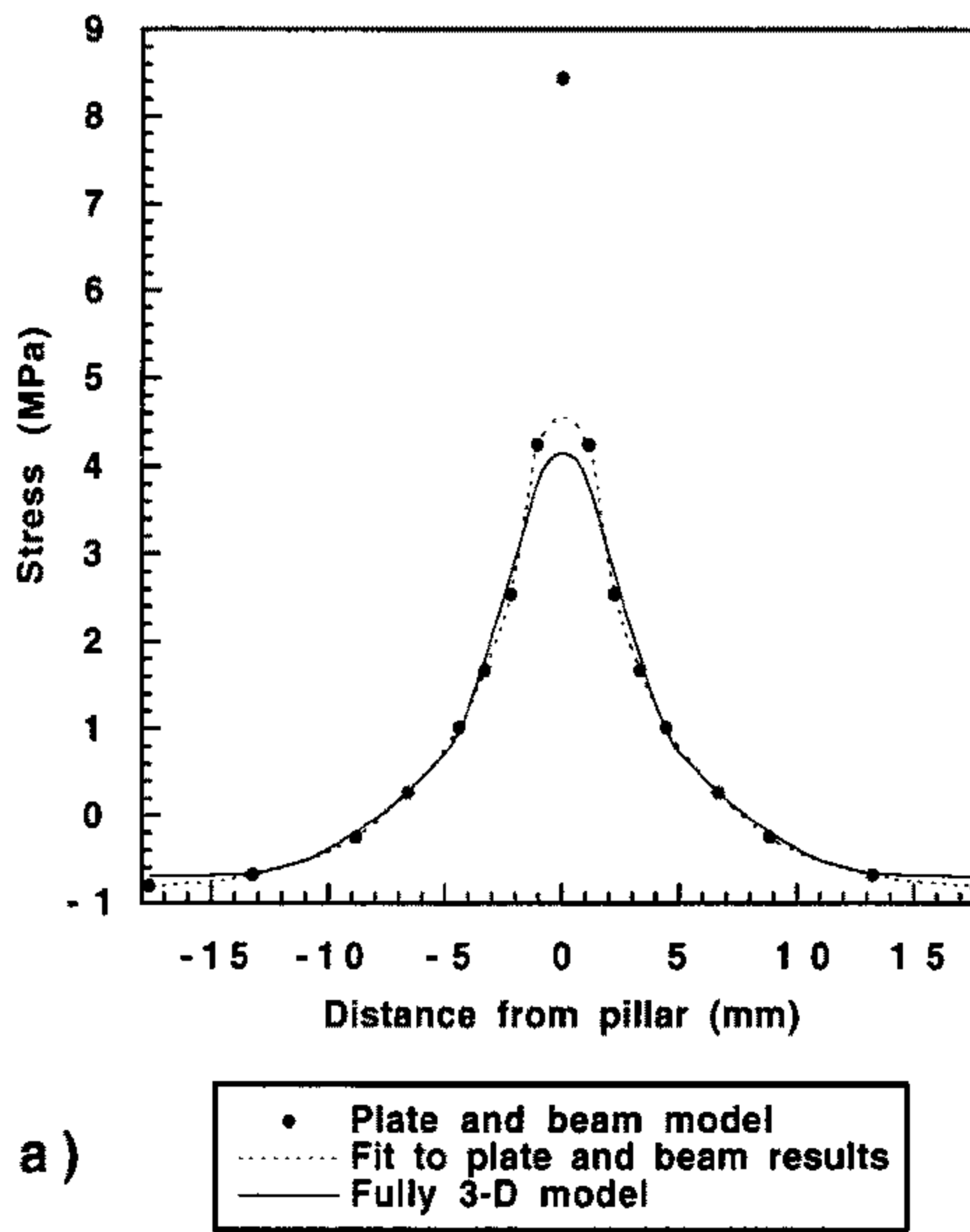


Fig. 7.6: a) The external glass surface stress parallel to both the x-axis and the y-axis as a function of the distance from a pillar along adjacent half-diagonals of two unit cells. The results are for stainless steel pillars of radius, half-height and separation of 0.125 mm, 0.1 mm and 25 mm, respectively. The glass thickness is 4 mm. Modelling results obtained with the thin shell plate and beam element model (Plate and beam model, (b)) are compared with the fully three-dimensional modelling results of Fischer-Cripps (Fully 3-D model, (c)) [5, 9]. The former results are for plate properties which are equal to the glass properties given in Table 7.1, and for beam properties of $E_{beam}=3.79 \times 10^{10}$ Pa, and $G_{beam}=1.56 \times 10^{10}$ Pa. The latter results are for the properties given in Table 7.1. The mirror image of the Fully 3-D results over a single half-diagonal is also shown. The fit to the plate and beam results is to all of these results except for the data point over the pillar (at distance=0).

accurate modelling data [12]. In this case, since the analytically verified benchmark results agree with the other results over most of the distance from the pillar, they are assumed to provide an appropriate guide for the stresses that the simple model should give close to the pillar. An interpolated curve fit to all of the thin shell plate and beam modelling results except the data point immediately above the pillar closely matches the fully three-dimensional modelling results, as shown in Fig. 7.6. Therefore, for the modelling results presented in the next section, the stresses right over the pillars are obtained from interpolated curve fits to all of the data points except for the points above the pillars. The stresses at all other locations are the actual modelling results. It is noted that this procedure leads to a slight overestimation of the stress above the pillars. However, it is assumed that this does not significantly affect the probability of failure calculations.

Despite the difficulties with calculating the stresses above the pillars, the thin shell plate and beam element model correctly predicts the deflections of the glass sheets due to atmospheric pressure. This is because the deflection calculations are not limited by the nature of the plate and beam element connections. For example, the model gives a value of d_{total} which is consistent with that on which the beam material properties are based: $-3.41 \mu\text{m}$. Also, the modelled glass deflection mid-way between adjacent pillars ($-3.84 \mu\text{m}$) is within 2 % of the result from the fully three-dimensional model ($-3.90 \mu\text{m}$ at 12.5 mm in Fig. 7.4c).

The results at the other region of large stress gradients - near the outer edges of the model (i.e. near the edges of the simulated glazing) - are not strongly dependent on the node density. For example, with the previously described node density of 240 nodes per unit cell and for $\lambda=25 \text{ mm}$, the stress which is normal to and at the outer edge of the model (and which is also on the glazing centerline) is 4.272 MPa. A doubling of the cell node density results in less than a 0.1 % increase in this stress. Further increases in the node density yield even smaller changes in the normal stress. The same is true for the stress parallel to the edge at this location. It is therefore concluded that the node density given above is sufficient to ensure a convergence in the results near the outer edges of the model, and that it is not necessary to grade the nodes in this region.

On the basis of the above results, the following node density was chosen for the model: 240 nodes per unit cell, graded from a node spacing of $\lambda/8$ near the middle of each cell to $\lambda/20$ around the pillars. The model consists of a total of 6836 nodes. This

was close to the maximum number of nodes for which this structural model could be solved with the available computing resources.

7.4 Results

All of the finite element results presented in this section are from the thin shell plate and beam element model. The plate and beam properties that were used in Section 7.3.3 correspond to the properties used by Fischer-Cripps, and they were only considered for the purpose of obtaining comparative modelling results. All of the modelling results in the present section were obtained from the plate and beam element model with the plate properties equal to those for soda-lime glass which are given in Appendix 1 ($E=7.38 \times 10^{10}$ Pa, and $\nu=0.21$). The beam Young's modulus and the beam shear modulus are equal to the experimentally derived values that were given in Section 7.3.2 for an alumina pillar and glass system: $E_{beam}=2.55 \times 10^{10}$ Pa, and $G_{beam}=1.05 \times 10^{10}$ Pa. It is recalled that these moduli are assumed to correspond to a system with pillars of radius and height of 0.125 and 0.20 mm, respectively. Therefore, all modelling results are for a pillar radius of 0.125 mm, and a pillar height of 0.20 mm; the corresponding height of the beams (i.e. the simulated half-pillars) of the model is 0.10 mm. The simulated glass sheet thickness is 4 mm. Except where otherwise stated, the results are along one of the inner edges of the model (i.e. along a centerline of the full square glazing and over a row of pillars). Results are presented only for the equivalent external glass surface of the simulated glass sheet because, as noted above, it is the external glass surface stresses that have the greatest effect on the probability of glazing failure. It is recalled that the model does not simulate an actual edge seal of finite width. However, as previously explained, the deflections and stresses over such a seal are assumed to be zero. A seal with the typical width of 8 mm is considered in this study by presenting these zero results for 8 mm beyond the outer edges of the model. All distances along the centerline of the glazing are measured from the inner edge of the seal (i.e. from the outer edge of the model) so that the pillar locations can be easily identified at integer multiples of λ . Including the region over the edge seal, the theoretical results correspond to a full square glazing of a dimension equal to 12 times the pillar separation plus two edge seal widths (e.g. a dimension of 316 mm for $\lambda=25$ mm). As explained below, for the conditions considered in this study, the same results would be obtained at the equivalent locations of larger glazings.

Figure 7.7 presents results for the deflection of the external glass surface as a function of the distance from the inner edge of the seal along the glazing centerline and over a row of pillars. As previously explained, the deflection of the glass over the indicated width of the edge seal is assumed to be zero, and the glass sheets are closer together away from the seal. The correlation between the locations of the pillars (which are separated by 25 mm) and the locations of the periodic minimum negative deflections is as expected. It was previously mentioned that the pillars near the seal are compressed less, and thus they exert smaller forces on the glass sheets, than the pillars elsewhere in the glazing. This is demonstrated by the fact that the pillar closest to the seal (at 25 mm) has only a small effect on the deflection results and, as shown below, on the stresses at the external glass surface. Due to the non-uniformity in the forces exerted by the pillars on the glass sheets, the average surface deflection is greatest between the second and third pillars from the edge seal (which are at 50 and 75 mm). The maximum negative glass deflection near the location of the second row of pillars is consistent with the approximate experimental and modelling results reported by Fischer-Cripps for stainless steel pillars [5]; the experimental results were obtained by a technique similar to that described below, and the modelling results were based on a very simple approach in which a half-thickness pillar array below a glass sheet was represented by an elastic slab with equivalent uniform material properties. As noted in Section 7.3.1, this modelling approach does not provide results for around individual pillars. However, the data of Fig. 7.7 (which are from a more exact model) confirm that it does give a correct qualitative description of the trend in the deflection profile near the edge of a glazing.

Deflection measurements were made on a sample of vacuum glazing which contains alumina pillars. The square glazing consists of glass sheets of thickness 4 mm and dimension ~ 300 mm, and the edge seal width is about 8 mm. The pillars have a square cross-section of dimension 0.20 mm, and they are 0.12 mm in height. The first pillar rows are at 12.5 mm from the inner edges of the seal, and beyond these rows the pillar separation is 25 mm. The glass-to-glass thickness was measured with a wide reach micrometer along a centerline of the glazing which, as for the results of Fig. 7.7, is coincident with a row of pillars. The net deflections of the external glass surfaces due to atmospheric pressure were calculated by subtracting from the measurements the thickness over the edge seal. The edge seal result was used as the reference because, as previously mentioned, there is a negligible pressure-induced glass deflection over the relatively rigid seal. It is important to note that this procedure only gives deflections relative to this reference; it does not provide absolute results for the net change from the pre-evacuation surface positions because

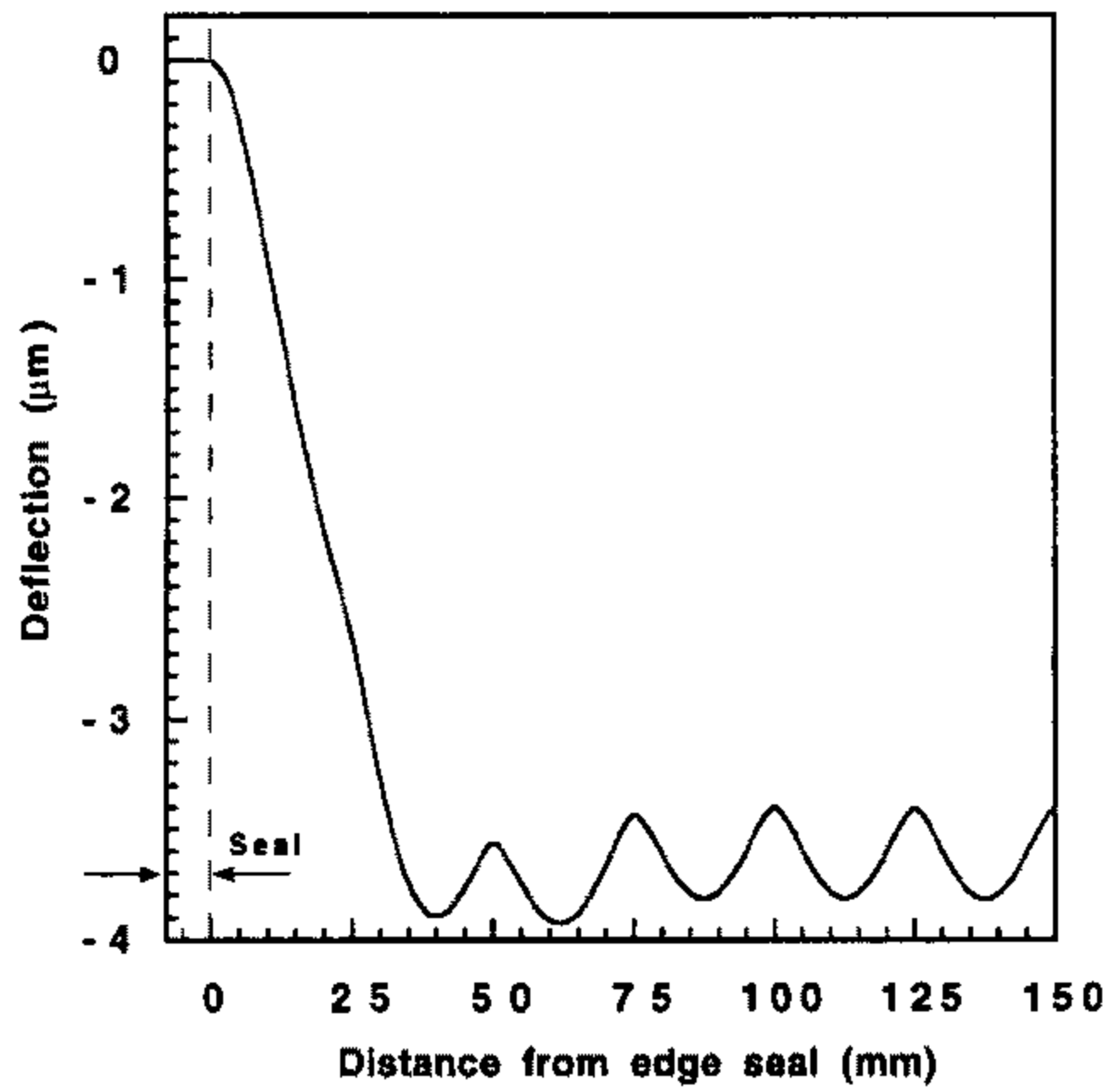


Fig. 7.7: Modelling results for the deflection of the external glass surface as a function of the distance from the inner edge of an 8 mm wide edge seal, along the glazing centerline and over a row of pillars. The width of the seal is as indicated. The results are for alumina pillars of radius, height and regular separation of 0.125, 0.20 and 25 mm, respectively. The pillars are located at integer multiples of 25 mm. The simulated glass thickness is 4 mm. The properties for the beam elements are $E_{beam} = 2.55 \times 10^{10}$ Pa, and $G_{beam} = 1.05 \times 10^{10}$ Pa, and the properties for the plate (i.e. glass) elements are $E = 7.38 \times 10^{10}$ Pa, and $\nu = 0.21$.

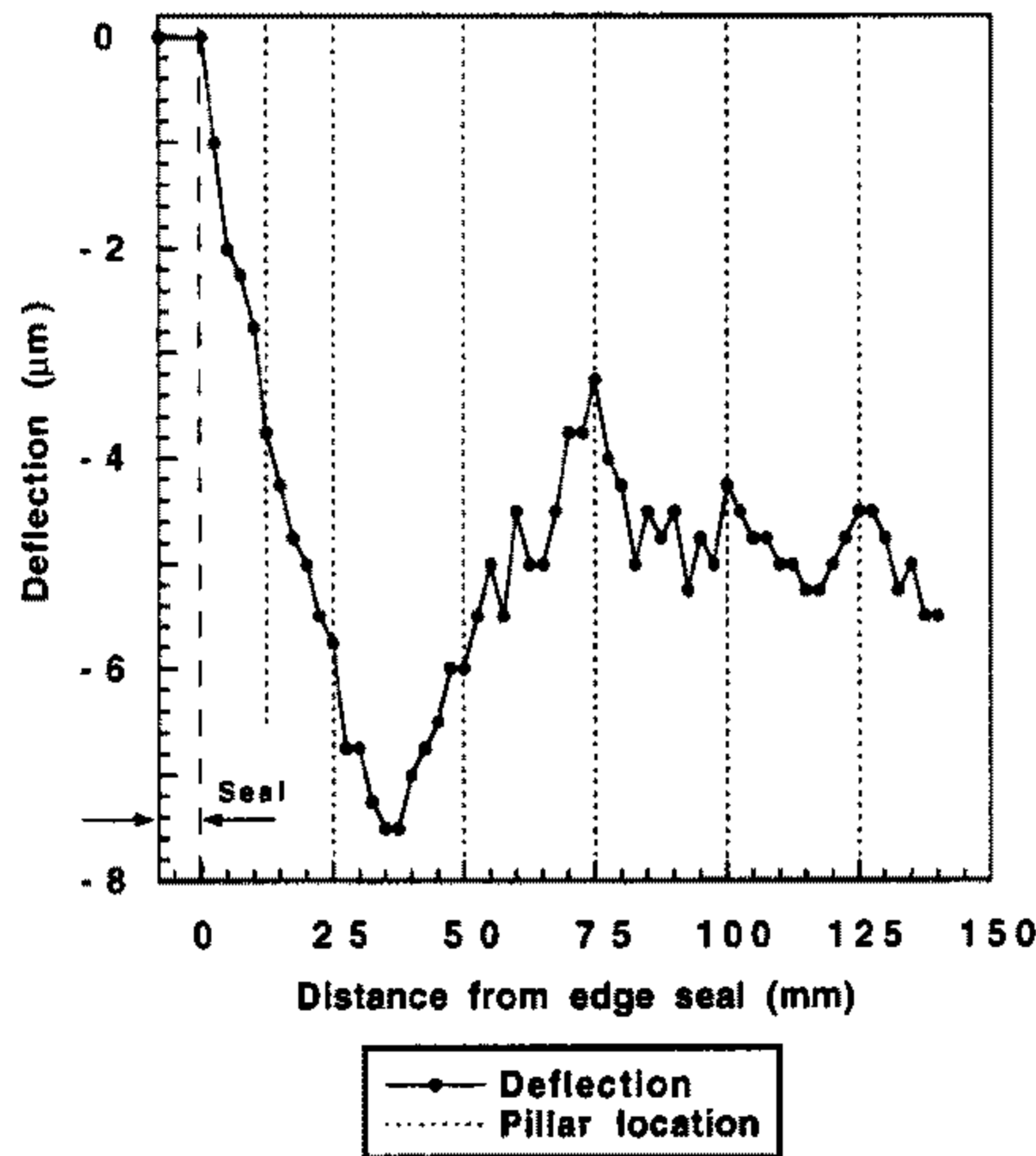


Fig. 7.8: Experimental results for the deflection of the external glass surface as a function of the distance from the inner edge of an 8 mm wide edge seal, along the glazing centerline and over a row of pillars. These results correspond to a half-thickness of a square glazing which is of dimension ~ 300 mm, and which contains square alumina pillars. The pillars are of dimension 0.20 mm and height 0.12 mm, and the glass thickness is 4 mm. The locations of the pillar rows are indicated by the vertical dashed lines.

it does not account for variations in the glass thickness. However, for the glass used in vacuum glazing, such variations across *each* glass sheet are usually less than $\pm 0.5 \mu\text{m}$. Therefore, neglecting any variations in the original pillar height, the sheet-to-sheet deflections are accurate to about $\pm 1 \mu\text{m}$. Since the deformations of the two glass sheets should be symmetric about the mid-plane of the glazing, the deflections for a single sheet and for half-height pillars (i.e. for a half-thickness of the glazing) can be assumed to be one half of the net deflection results. Such experimental deflections for a half-thickness of the glazing are presented in Fig. 7.8. Although the glazing size, pillar dimensions and pillar layout are somewhat different from the more typical conditions considered for Fig. 7.7, these experimental data do provide a qualitative validation of several aspects of the modelling results of Fig. 7.7 (which also correspond to a half-thickness of a glazing). Firstly, the experimental and modelling deflections are of the same order of magnitude; the experimental deflections are slightly larger because they are for alumina pillars of a lower stiffness; the stiffness coefficient, which is given by equation 7.2, is more affected by the smaller pillar cross-sectional area than by the smaller pillar height (it is recalled that the moduli for the modelling results were derived from measurements on a pillar-glass system with a square pillar of dimension 0.27 mm and of height 0.2 mm). Secondly, as with the modelling results, there is a maximum negative deflection between the second and third pillar rows from the edge (for the experimental data, the second row is at 25 mm). Thirdly, it appears that there are periodic minimum negative deflections at the pillar locations of 75, 100 and 125 mm. However, since the variation in the deflection beyond 75 mm is approximately within the error associated with the data ($\pm 1 \mu\text{m}$), these results do not definitely confirm the trend in the modelling results over the pillars of this region. It is noted that measurements prior to evacuation would permit the calculation of net absolute deflections. However, since the reproducibility of each thickness measurement is about $\pm 0.5 \mu\text{m}$, there would be no reduction in the error associated with the final results.

Modelling results for the external glass surface stresses normal and parallel to the outer edge of the model (i.e. normal and parallel to the edge of the glazing), and along the glazing centerline for $\lambda=25 \text{ mm}$ are presented in Fig. 7.9. These results are for the same conditions as those for the deflections of Fig. 7.7. As explained above, the larger stress at the inner edge of the seal is the normal stress, and both the normal and parallel stresses are assumed to be zero over all of the seal. The smaller stresses in the edge region (mostly between the first and second pillars) are associated with the above mentioned enhanced bending due to the non-uniform pillar-glass forces near the edge; this bending increases the concavity of the external glass surface

between these pillars, and thus the magnitude of the compressive stress at this location. A comparison with the results of Fig. 7.7 shows that, as expected, the minimum stress (which is compressive) occurs at the location of the maximum concave curvature: half-way between the first and second pillars from the edge seal (at 37.5 mm). Far from the edge, both orientations of stress are symmetric about the pillars but the trends in the stress profiles are different. This is simply a reflection of the symmetry of the locations of the nearby pillars with respect to the transit to which the results correspond. Along the centerline of the glazing, and thus over a row of pillars, the only points about which there is a symmetry in the locations of the surrounding nearby pillars of the model are the points above the pillars of this row. In between the pillars, the normal and parallel stresses should be different by virtue of the asymmetry in the locations of the nearby pillars.

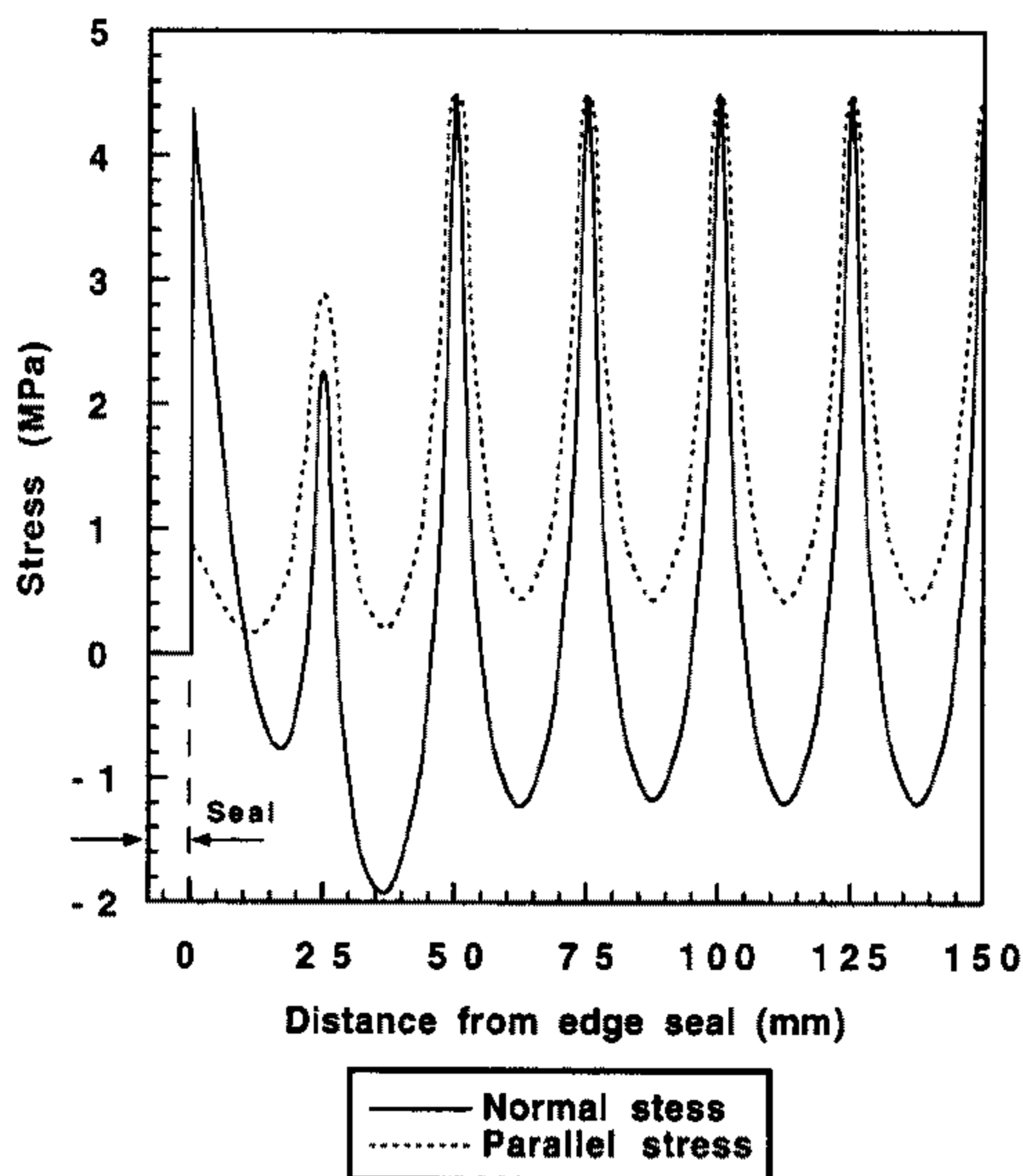


Fig. 7.9: The external glass surface stresses normal and parallel to the edge of the glazing, along the centerline of the glazing and over a row of alumina pillars, for the same dimensions and properties given for Fig. 7.7.

In principle, the strains associated with the external surface stresses due to atmospheric pressure could be measured with strain gages in a manner similar to that which is described in Chapter 6. However, the results of Fig. 7.9 indicate that this would require gages with very small absolute and spatial resolutions - especially for measurements away from the edges. Although average glass surface strains could be easily measured, it is unlikely that useful localized results could be obtained without specialized - and very expensive - strain gages. Therefore, the experimental validation of the model is only based on the above qualitative comparison between the modelled and measured deflections.

A contour plot of the external glass surface stress parallel to the x-axis for $\lambda=25$ mm is presented in Fig. 7.10. In this figure, the stress is parallel to the outer edge of the model which is coincident with the bottom border of the plot, and it is normal to the outer edge of the model which is coincident with the left border of the plot. The results are symmetric with the stress parallel to the y-axis. The region over the edge seal, where the stresses are assumed to be zero, is not shown (i.e. only the region bounded by the inner edges of the seal is shown). This figure further demonstrates that, near the edges, the normal stress is greater than the parallel stress. The reason for the reduction in the stresses near the corner of the glazing (i.e. the outer corner of the model) is the same as one of the reasons given in Section 6.6.3 for the similar reduction in the temperature-induced bending stresses: the symmetry of the normal and parallel stresses at the corner. The fact that atmospheric pressure leads to reduced corner stresses might seem surprising considering the fate of the square windows of the De Havilland Comet Mark 1 (this was the first commercial jet-powered passenger aircraft, and it was manufactured in the U.K. in the early 1950's). The pressure differential across the windows led to enhanced stresses around the window corners which, due to the repeated cycling of the cabin pressure, ultimately resulted in structural failure. However, in the case of these windows, the problems actually occurred at the corners of the square supports surrounding the structures (the square holes in the fuselage - i.e. the window frames) and not in the glazings. This is as expected: with forces perpendicular to the lateral dimensions, the corners of square holes in plates (e.g. these window frames) are regions of stress concentration, whereas the corners of square plates (e.g. vacuum glazings) are regions of stress reduction.

In Figs. 7.7, 7.9 and 7.10, there is a uniformity in the periodic deflection and stress results beyond 75 mm from the edge seal. This indicates that the model is large enough to ensure that, away from the outer edges of the model and for the above conditions, the results around the pillars are unaffected by the fully fixed conditions at

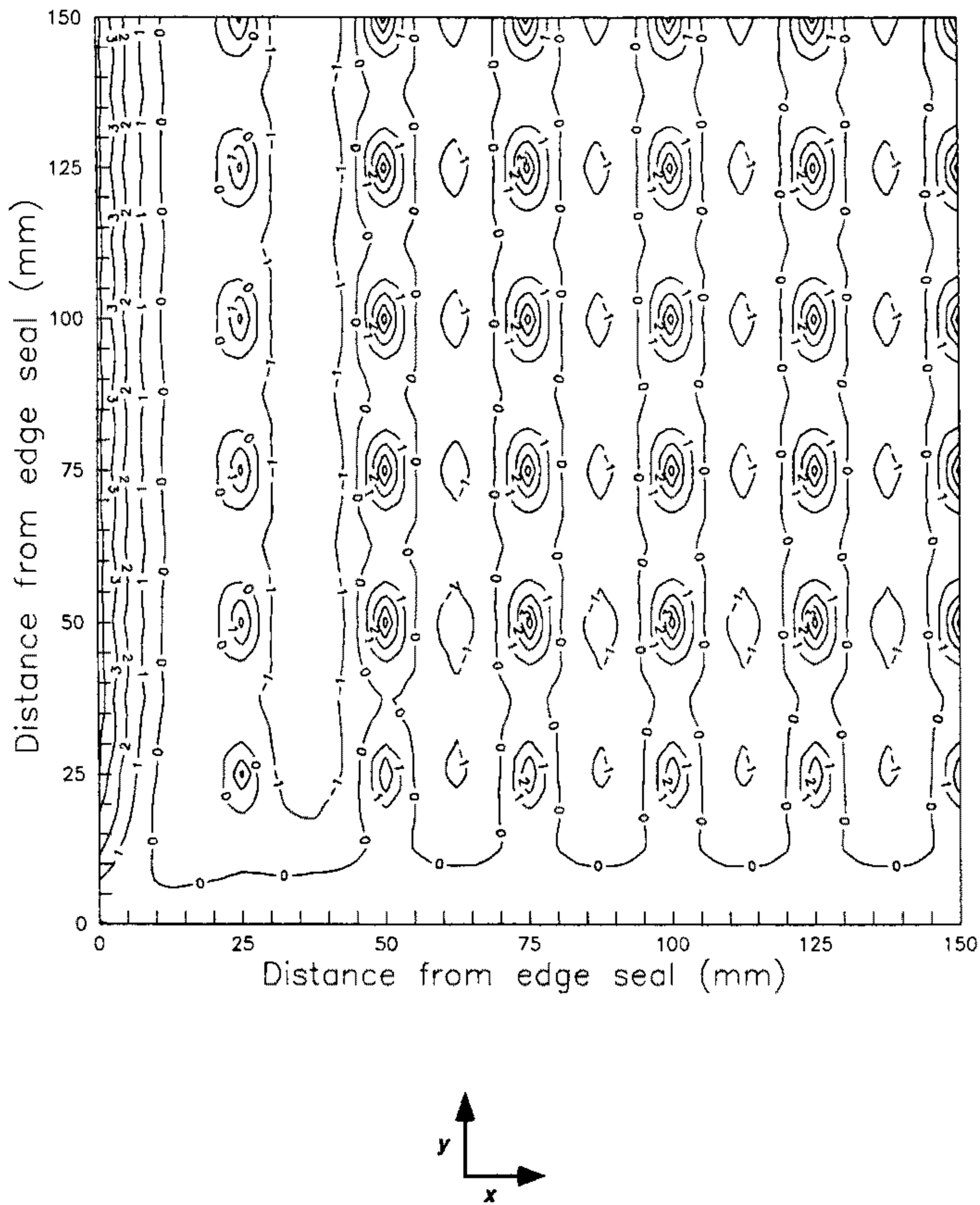


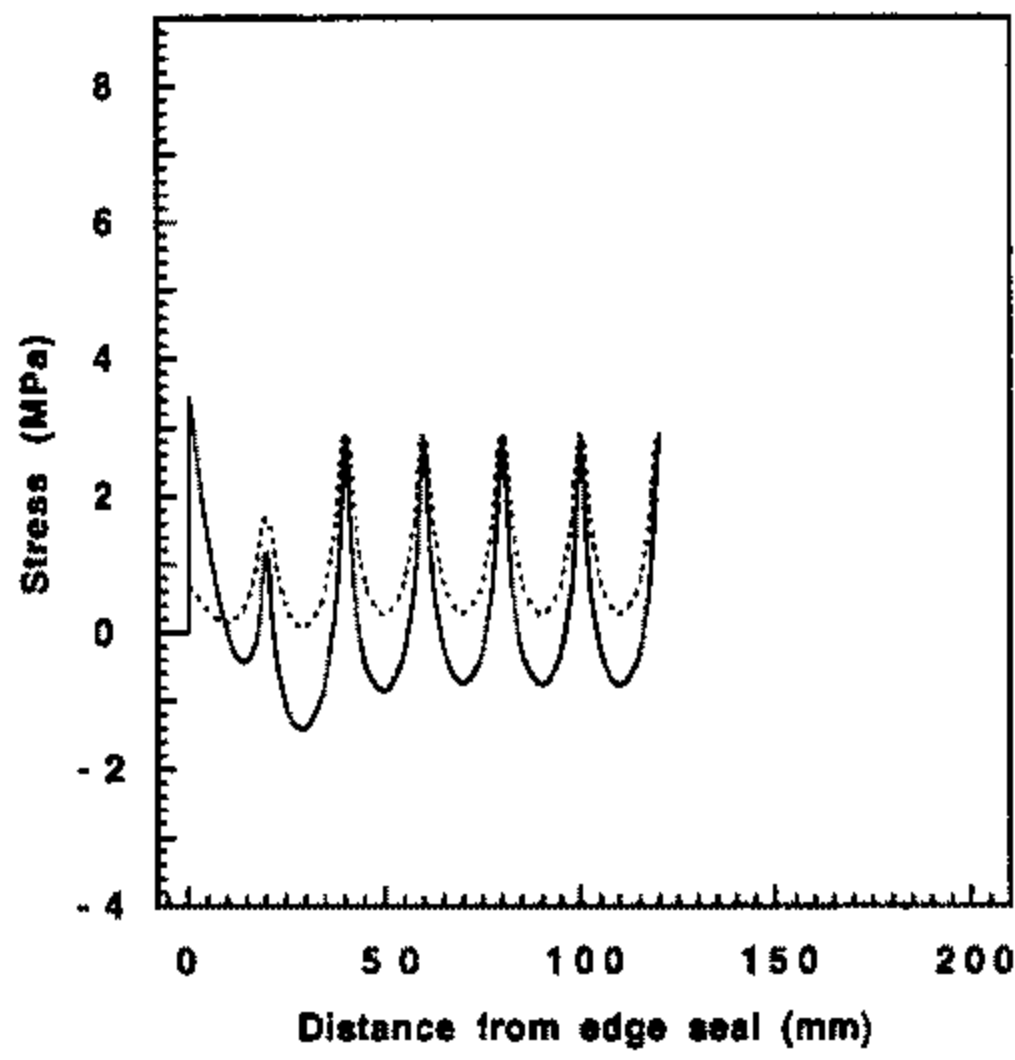
Fig. 7.10: A contour plot of the external glass surface stress parallel to the x-axis (which is coincident with the bottom edge of this plot) for a quarter-section of a glazing with alumina pillars. The pillar separation is 25 mm, and the pillars are located at the centers of the oval shaped patterns of concentric contours (i.e. the pillars are at integer multiples of 25 mm). These results are for the same dimensions and properties given for Fig. 7.7. The region over the edge seal is not shown. The stress contour interval is 1 MPa.

the outer edges. Furthermore, as demonstrated by the results of Fig. 7.10, the periodic results along the region within 75 mm (i.e. 3λ) of the edge seal and about 50 mm (i.e. 2λ) from the corner of the glazing are fairly uniform. Therefore, for the conditions considered above, the model is large enough to provide representative results for both the central and edge regions of larger glazings. This means that the same results would be obtained at the equivalent locations of larger glazings (e.g. along a centerline which is coincident with a row of pillars).

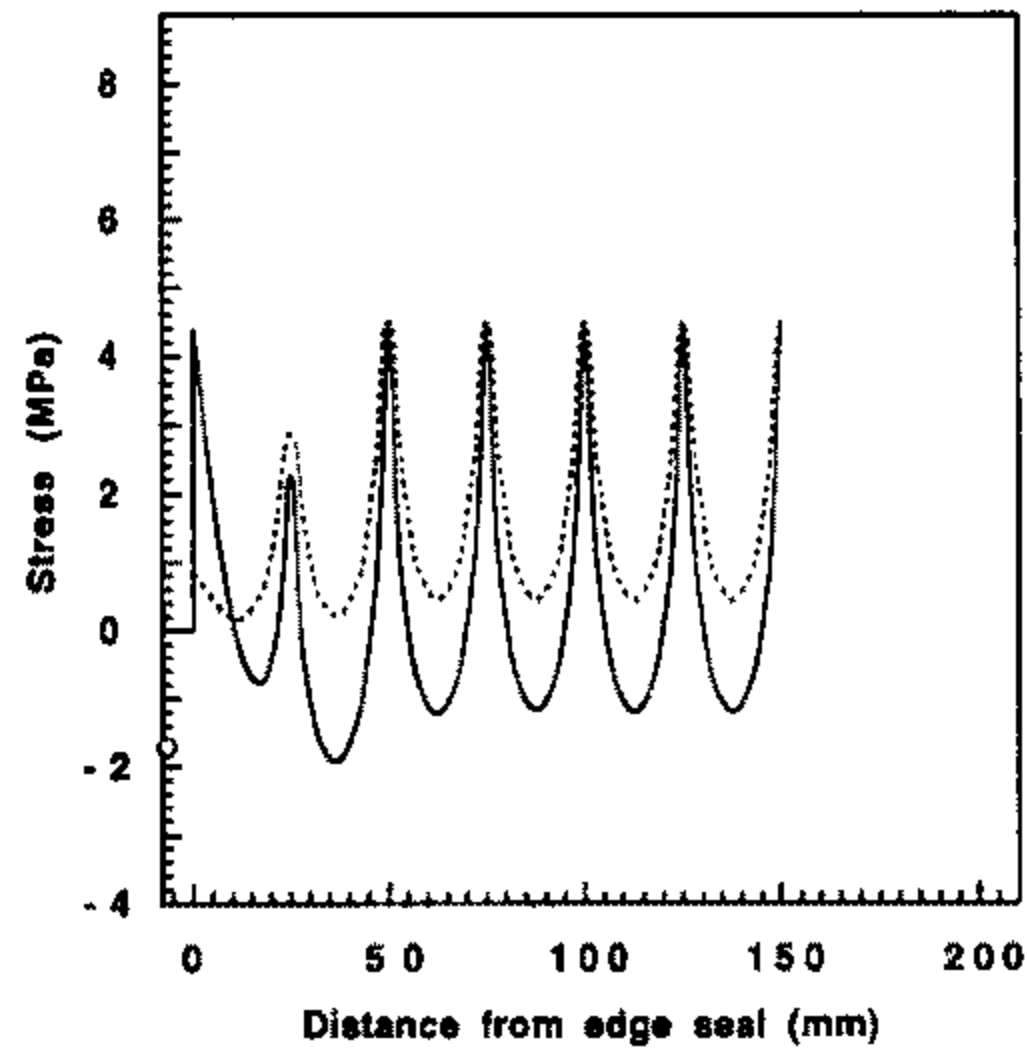
Figures 7.11a to d present a comparison of the normal and parallel external glass surface stresses for different pillar separations: 20, 25, 30 and 35 mm. The results for $\lambda=25$ mm are from Fig. 7.9. Since the stiffness of the pillar-glass system is unaffected by the pillar separation, all of the modelling results of Figs. 7.11 are for the same beam Young's modulus of elasticity and the same beam shear modulus given above for a pillar separation of 25 mm. The results of these figures show that the larger tensile stresses occur over the pillars and at the inner edges of the seal.

Figure 7.12 presents the stress as a function of the distance from a pillar along adjacent half-diagonals of two unit cells far from the edges of the glazing. Since these half-diagonals also lie along the diagonal of the symmetric glazing, these results represent both the stresses normal and parallel to the edge of the glazing (i.e. the stresses parallel to the x-axis and the y-axis). Results are given for different pillar separations. The maximum stress, which occurs above the pillar and is isotropic, is plotted in Fig. 7.13 as a function of the pillar separation. Also plotted in Fig. 7.13 is the isotropic stress predicted by the analytic relationship of equation 7.1 in the manner described above with $P=101.3$ kPa, $t=4$ mm and $\nu=0.21$. These data are presented on a log-log scale in order to highlight the functional relationship between the stress and the pillar separation. Since the modelling data can be approximated by a straight line for which the slope is 1.96, the stress above the pillar is approximately dominated by a λ^2 dependence. This is roughly consistent with equation 7.1, as demonstrated by the fact that the slope of the linear fit to the analytic data of Fig. 7.13 is 2.43.

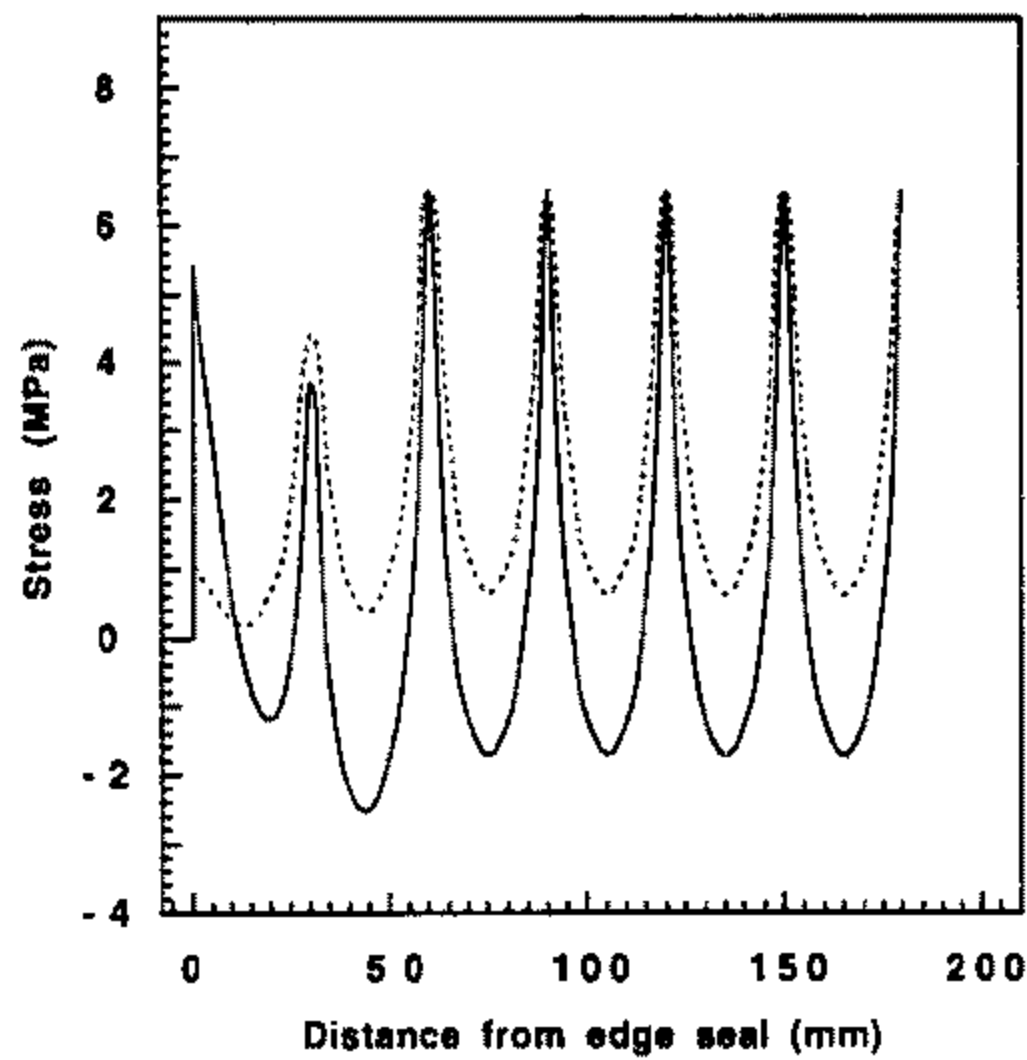
The results of Figs. 7.11 show that the stresses above the inner edge of the seal also increase with the pillar separation. These stresses - both those normal and parallel to the edge - are presented in Figure 7.14 as functions of the pillar separation.



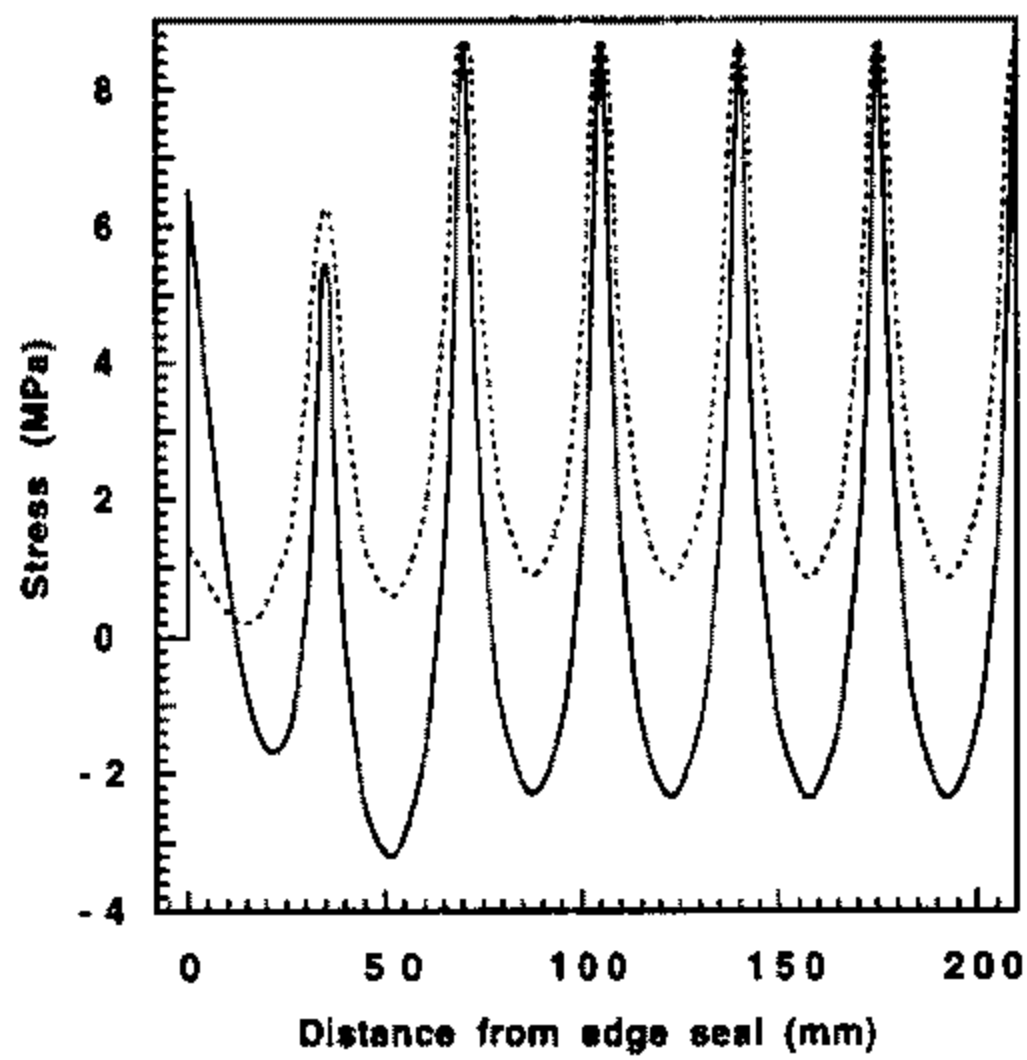
a) $\lambda=20$ mm



b) $\lambda=25$ mm



c) $\lambda=30$ mm



d) $\lambda=35$ mm



Figs. 7.11a to d: The external glass surface stresses normal and parallel to the edge of the glazing, along the centerline of the glazing and over a row of alumina pillars, for different pillar separations, λ . Aside from the pillar separation, these results are for the same dimensions and properties given for Fig. 7.7.

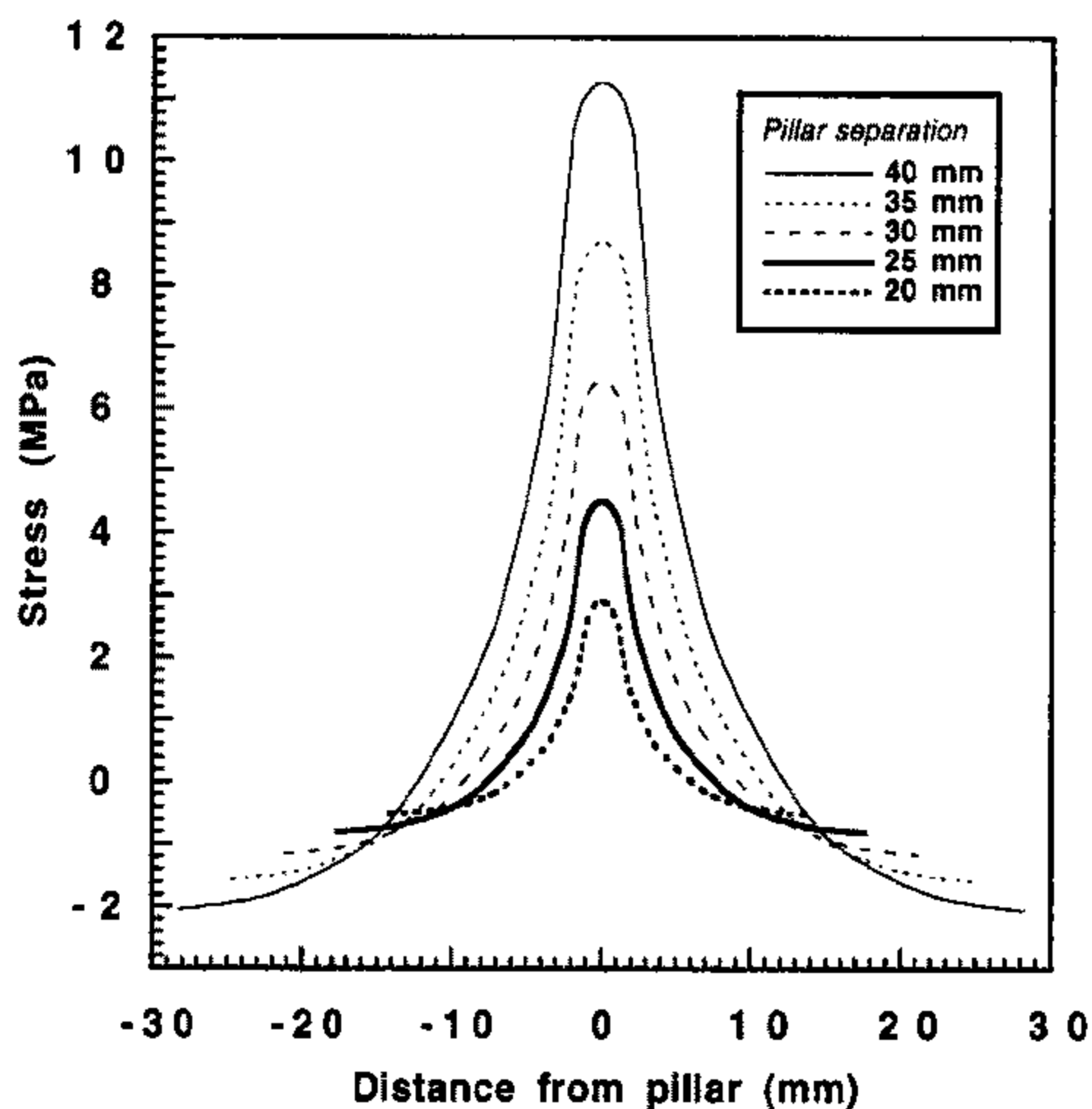


Fig. 7.12: The external glass surface stress which is both normal and parallel to the edge of the glazing as a function of the distance from an alumina pillar along adjacent half-diagonals of two unit cells for different pillar separations. Aside from the pillar separation, these results are for the same dimensions and properties given for Fig. 7.7.

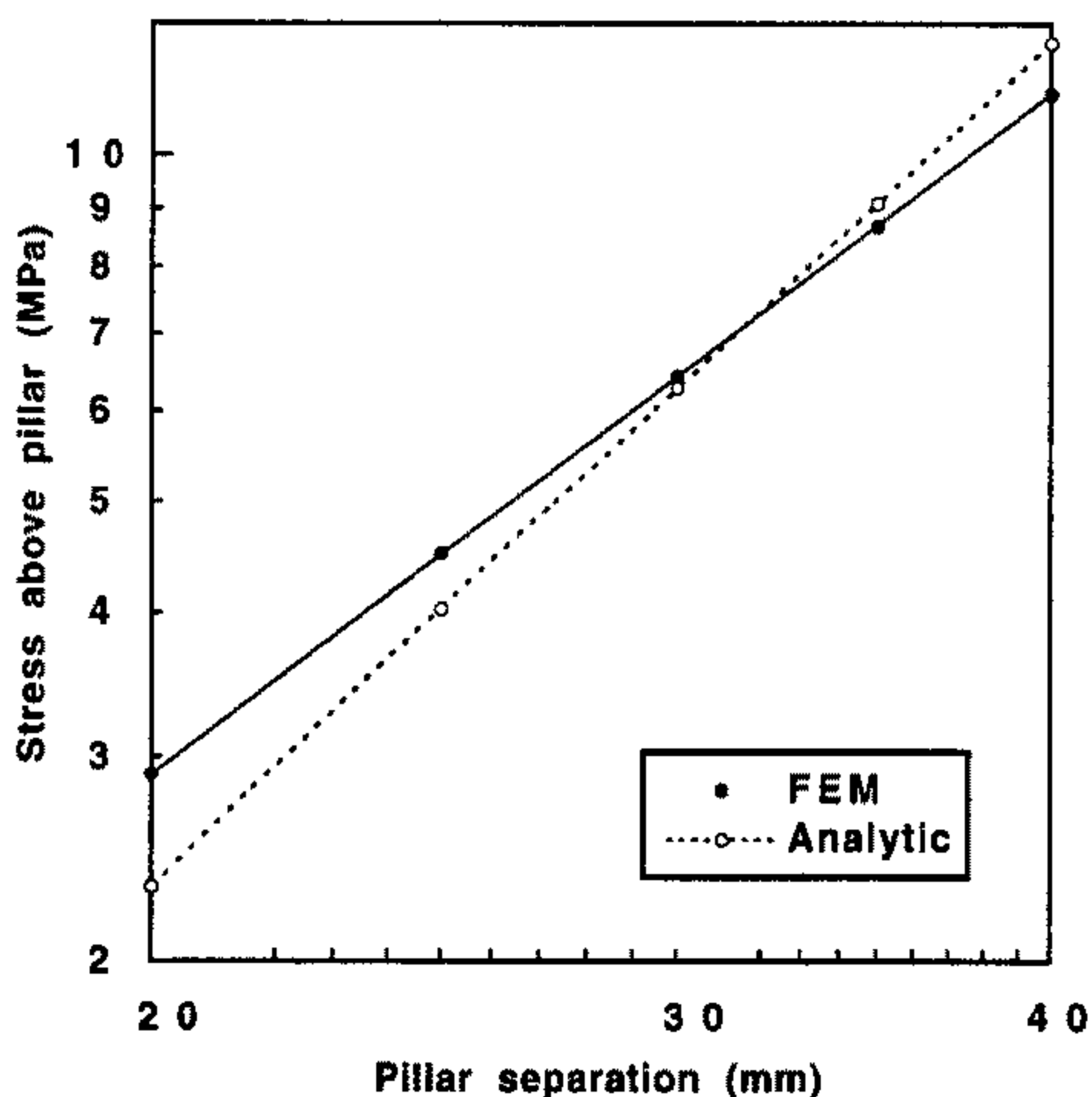


Fig. 7.13: The isotropic external glass surface stress above an alumina pillar, as predicted by the finite element model (FEM) and by the analytic relationship given by equation 7.1 (Analytic), as functions of the pillar separation. Lines of linear regression are fitted to the data. Aside from the pillar separation, these results are for the same dimensions and properties given for Fig. 7.7.

All of the above modelling results of this section are for typical and approximately typical pillar-glass system stiffnesses. Figure 7.15 presents the centerline external glass surface stresses for a pillar-glass system which is ten times less stiff than the typical alumina pillar and glass system considered for Figs. 7.7, 7.9 and 7.10: the beam cross-sectional area and the beam height are the same, but $E_{beam}=2.55 \times 10^9$ Pa. Based on the same Poisson's ratio of 0.21, the corresponding beam shear modulus is $G_{beam}=1.05 \times 10^9$ Pa. The pillar separation is 25 mm. These results can be compared with those of Fig. 7.9 which are for the same pillar separation but which correspond to the previously considered larger stiffness. Due to the smaller system stiffness, the glass sheets bend together more, leading to greater stresses near the edge of the glazing; the order of magnitude decrease in the stiffness leads to a threefold increase in the normal stress above the inner edge of the seal, and to an increase in the tensile stresses over the first 20 mm from the seal. Also associated with the increased bending of the sheets is a more extensive region of lower stresses - to about the fourth pillar at 100 mm. The reduction in the system stiffness is further reflected by the fact that the first pillar from the edge seal (at 25 mm) has a smaller effect on the stress profiles. Far from the edge, the stresses over the pillars are close to the results obtained with the larger stiffness. This is as expected because the stresses above the pillars in this region should be more influenced by the bending of the glass over the individual pillars than by the distance by which the glass sheets come together. The large edge stresses in Fig. 7.15 highlight the importance of selecting pillars of an appropriate minimum strength. In order to ensure acceptable edge stresses, and thus probabilities of failure, the pillars in highly insulating vacuum glazings must be made from materials which do not deform plastically for compressive stresses of less than 1.5 GPa (plastic deformation leads to greater pillar compression and thus to greater edge stresses than elastic deformation) [5].

Thus far, only the direct stresses parallel to the axes of the model have been considered because they best facilitate the physical understanding of the response of the glazing structure to atmospheric pressure. However, the shear stresses must also be considered since these stresses are used in conjunction with the direct stresses to calculate the probabilities of failure for the glazing. As noted at the end of the previous chapter, the direct external glass surface stresses parallel to the x-axis and the y-axis can be respectively expressed as σ_{xx} and σ_{yy} . The corresponding shear stress is expressed as σ_{xy} (this shear stress is symmetric with σ_{yx}) [13].

Figure 7.16 presents the shear stress, σ_{xy} , at the external glass surface for the same dimensions and properties given for Fig. 7.7. As with Fig. 7.10 (which is for the same

conditions), only the region bounded by the inner edges of the seal is shown. It is recalled that the shear stress in the glass over the seal is assumed to be zero. The results are presented as colour contours in order to best highlight the fine structure around the pillars (which are located at the centers of the cloverleaf shaped patterns). The stress interval between each colour is about 0.0542 MPa (this interval best highlights the stresses).

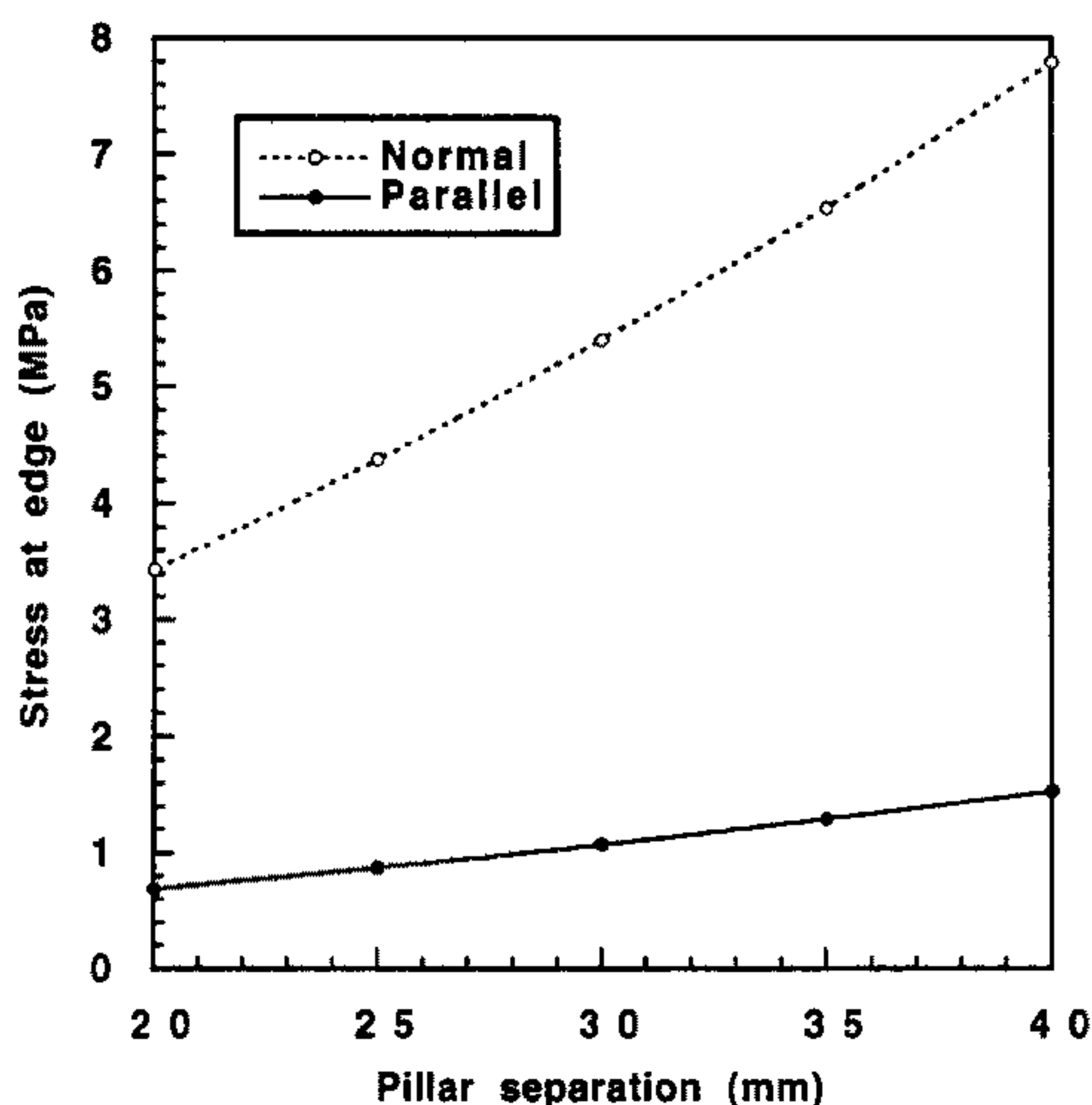


Fig. 7.14: Finite element modelling results for the stresses at the external glass surface above the inner edge of the edge seal - both the stresses normal and parallel to the edge - as functions of the pillar separation. Aside from the pillar separation, these results are for the same dimensions and properties given for Fig. 7.7.

The results of Fig. 7.16 show that there is a uniformity in the periodic shear stress results beyond 75 mm from the edge seal. Also, the periodic results along the region within 75 mm of the edge seal and about 50 mm from the outer corner are fairly uniform. Therefore, as with the previously considered direct axial stresses, the model can be used to obtain representative shear stress results for both the edge and central regions of larger glazings.

It is of interest to compare the magnitudes of typical external surface stresses due to atmospheric pressure (e.g. the stresses of Fig. 7.9), with those of typical external surface stresses due to the bending caused by a temperature difference across a

typically sized vacuum glazing which is exposed to standard fenestration test conditions (e.g. the stresses of Fig. 6.27b). Over the edge seal, both orientations of the pressure induced stress are zero, whereas those of the temperature induced stress are not. Near the seal, the maximum tensile pressure stress (which is normal to the edge) is less than the maximum tensile temperature stress (which is parallel to the edge). The pressure stresses over the pillars are slightly larger than the tensile isotropic center-of-glass stresses due to a temperature difference. However, over most of both the warm and cold sides of the glazing, the typical pressure stresses (both tensile and compressive) are smaller in magnitude than the typical temperature stresses. For the larger (and atypical) stiffness corresponding to the results of Fig. 7.15, the tensile pressure stresses near the edge are comparable to the tensile temperature stresses at this location. The effects of the combined pressure and temperature stresses on the probability of failure calculations are discussed in the next chapter.

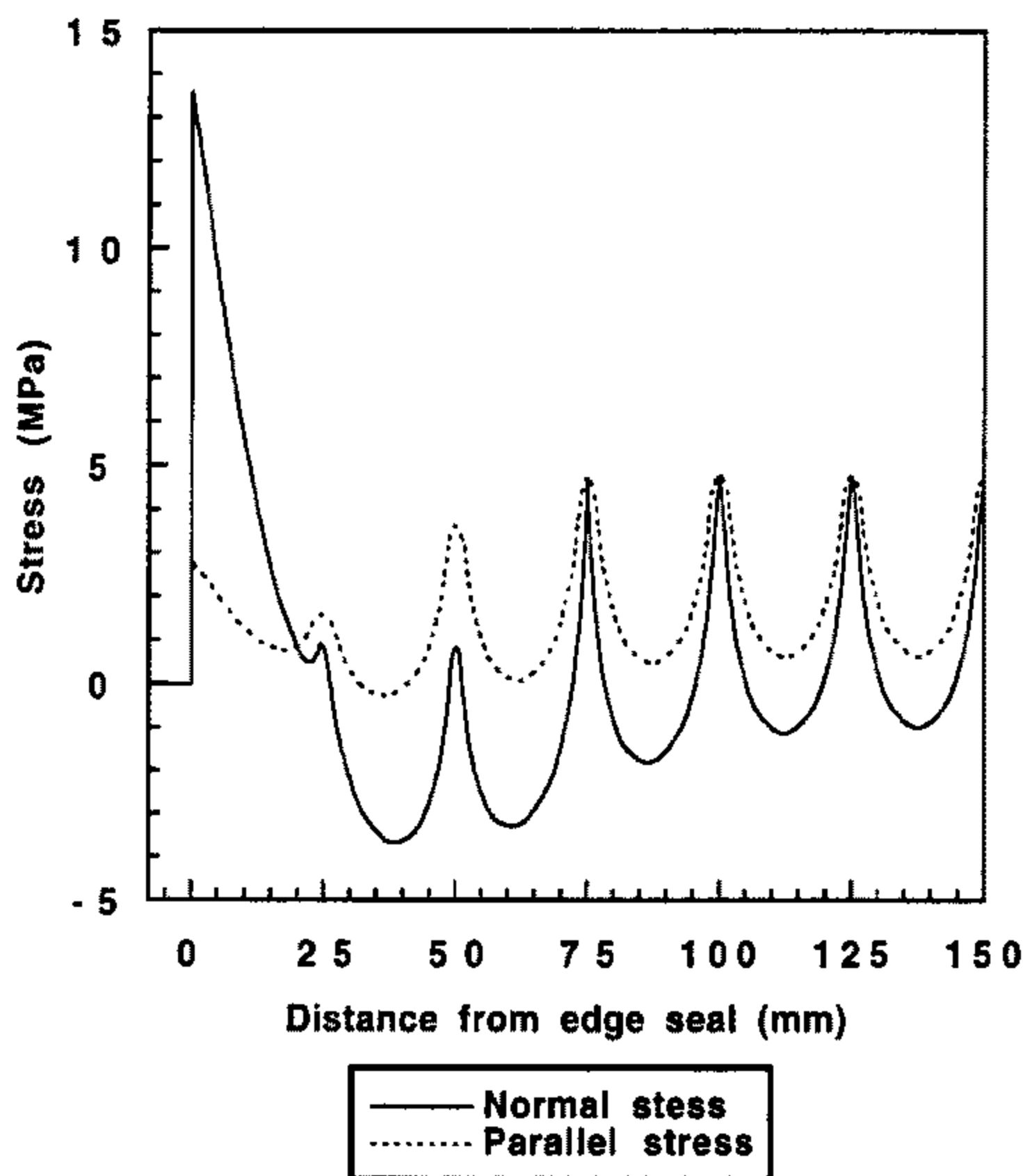


Fig. 7.15: The external glass surface stress normal and parallel to the edge, along the centerline of the glazing and over a row of pillars. The pillar radius, half-height and separation are 0.125 mm, 0.1 mm and 25 mm, respectively. The simulated glass thickness is 4 mm. These results are for a pillar-glass system which is ten times less stiff than the glass and alumina pillar system considered for Figs. 7.7, 7.9 and 7.10. The properties for the beam elements are $E_{beam} = 2.55 \times 10^9$ Pa, and $G_{beam} = 1.05 \times 10^9$ Pa, and the properties for the plate (i.e. glass) elements are $E = 7.38 \times 10^{10}$ Pa, and $\nu = 0.21$.

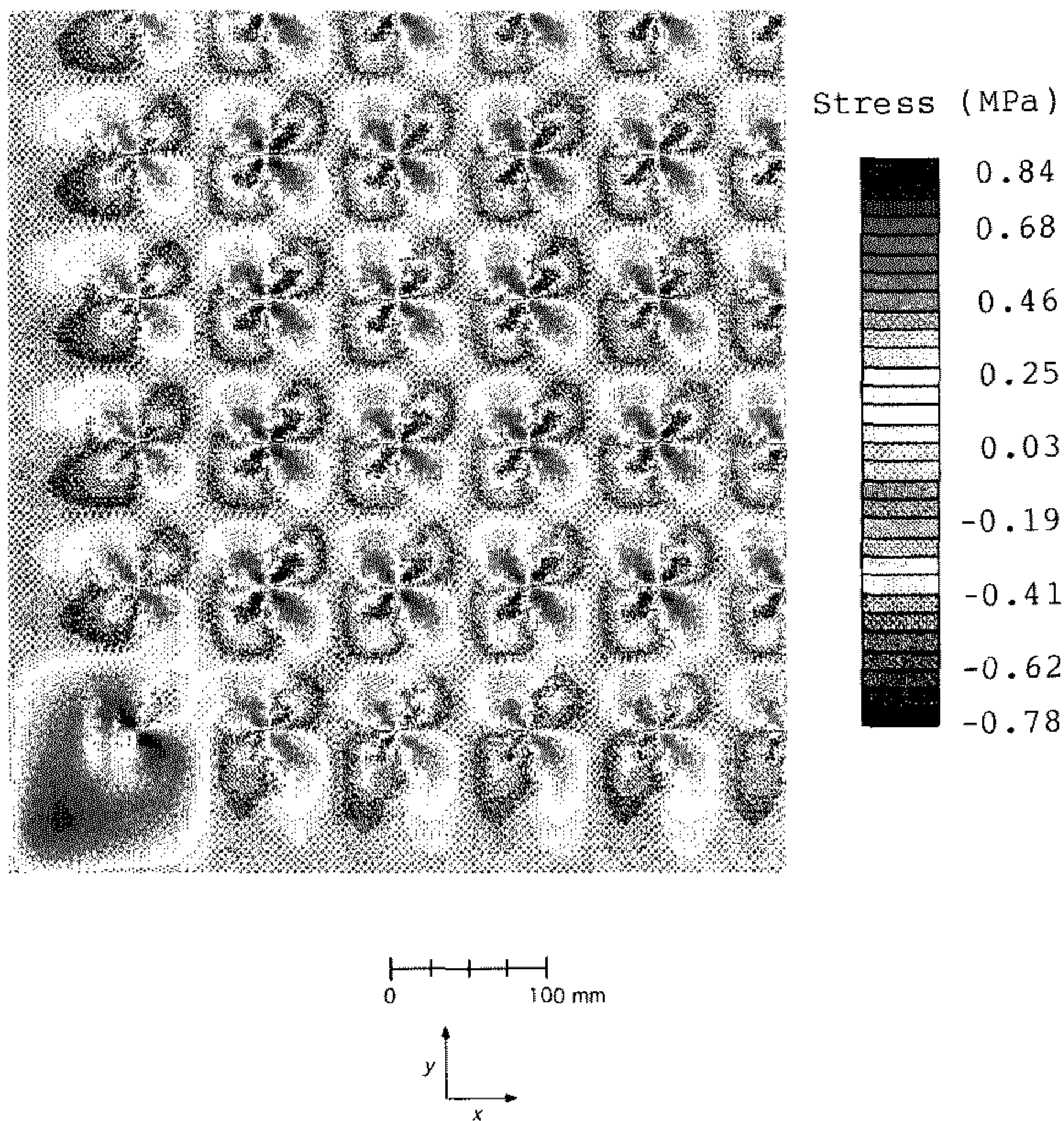


Fig. 7.16: A colour contour plot of the atmospheric pressure induced shear stress, σ_{xy} , at the external glass surface of a quarter-section of a vacuum glazing with alumina pillars. The pillar separation is 25 mm. These results are for the same dimensions and properties given for Fig. 7.7. The region over the edge seal is not shown. The pillars are located at the centers of the cloverleaf shaped patterns. The glazing corner is at the lower left corner of the image. The stress interval between each coloured region is about 0.0542 MPa.

7.5 Conclusion

Atmospheric pressure leads to compressive stresses in the pillars, stresses at the internal glass surfaces associated with the indentation of the pillars into the glass sheets, and tensile stresses at the external glass surfaces above the pillars and near the edge seal. It also causes the glass sheets to come together through the compression of the pillars, and through the indentation of the pillars into the glass surfaces; these two processes affect the stresses due to atmospheric pressure.

A finite element model has been developed for the analysis of the stresses in vacuum glazing that are caused by atmospheric pressure. In order to reduce the required number of nodes, several simplifications are made. The glass is modelled with thin shell plate elements, and the pillars are simulated with beam elements of appropriate material properties. Only a portion of a full square glazing need be modelled due to the vertical and lateral surfaces of symmetry of the glazing, and due to the uniform periodicity of the stresses. The stresses around pillars away from the edges of the glazing are, over most of the distance from these pillars, close to the results obtained by a fully three-dimensional modelling approach. However, due the nature of the connections between the plate and beam elements, the modelled stresses close to the pillars are not correct. Thus, in this region, post-processing of the results is employed in order to estimate appropriately the glass stresses.

The modelling results show that, near the edge seal, the pressure induced stress normal to the edge is much greater than that parallel to the edge. The stresses near the seal and the stresses above the pillars vary with the pillar separation. The functional relationship between the modelled isotropic external glass surface stress above the pillars and the pillar separation is roughly consistent with that predicted by a simple analytic approach (this stress is approximately proportional to the square of the pillar separation). For highly insulating vacuum glazings, the stresses due to atmospheric pressure for typical pillar properties and spacings are mostly smaller in magnitude than the stresses due to the bending caused by a temperature difference across a typically sized glazing which is exposed to standard fenestration test conditions.

In this chapter and the previous one, modelling results have been presented for the stresses in vacuum glazing from two separate sources: the presence of a temperature difference across the glazing, and atmospheric pressure. In the next chapter, these results are combined for an analysis of the probability of glazing failure.

7.6 List of Symbols

a	radius of disk
A	cross-sectional area of pillars
A_{beam}	cross-sectional area of beams
d_{compr}	distance by which pillar is compressively deformed
d_{indent}	distance by which pillar indents into inner glass surface
d_{total}	total deflection of the external glass surface immediately above a pillar
E	Young's modulus of elasticity
E_{beam}	Young's modulus of elasticity for beam elements
F	force exerted on each unit cell, and on each supporting pillar
F_d	concentrated central force applied to disk
G_{beam}	shear modulus for beam elements
h_{beam}	height of beam elements
k_{beam}	stiffness coefficient, or spring constant, of beam elements
P	atmospheric pressure
t	thickness of disk

Greek Symbols

ϵ	strain
λ	pillar separation
σ	stress
σ_{xx}, σ_{yy}	direct stress
σ_{xy}, σ_{yx}	shear stress
ν	Poisson's ratio

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8. The Probability of Failure for Vacuum Glazing

8.1 Introduction

In the last two chapters, results were presented for the stresses in vacuum glazing from the following external effects: the presence of a temperature difference across the glazing, and atmospheric pressure. These results - and others obtained from the same finite element models - are now combined to calculate the total stresses in the structure for the purpose of determining the total probability of glazing failure. The entire solution procedure is described in detail below, and then the effects of the pillar separation and the insulation recess depth on the failure probabilities are investigated. The results from this analysis are required for the glazing design study which is presented in the next chapter of this thesis.

The probability of glazing failure is dependent on many factors that are not fully considered in this study. These include the glazing dimensions, the pillar radius, the glass surface flaw parameters and the edge insulation thermal conductance. A complete investigation into the effects of all of these factors would be a very extensive work which would be best covered in a separate study. Therefore, the present study is only based on approximately typical conditions in order to demonstrate the method of calculation in general terms, and to obtain representative results and trends. The same approach can be used to obtain results for other conditions.

8.2 The Probability of Glass Failure

Cracks leading to the structural failure of glazings originate at small imperfections called flaws. These flaws are present at the surfaces of all glass specimens and they can be caused by rough handling or environmental abrasion. A surface flaw leads to large microscopic concentrations of the macroscopic stress. The concentrated stress can accelerate the growth of the flaw and eventually lead to catastrophic structural failure. Since glass is very strong under compression, such failure of glazings via the growth of a flaw almost always occurs under tensile stress [1]. Therefore, glass failure prediction models for glazings are based on tensile stresses.

If glass is protected from external mechanical damage, it will have few surface flaws and it will therefore have a low probability of failure. This is why glass optical fibers,

which are encapsulated at the time of manufacture, are so strong. Since the external surfaces of a glass sheet provide a similar protective covering for the sub-surface material, sub-surface stresses need not be considered in failure analyses such as the present study. Since, as noted in the last two chapters, the internal glass surfaces of vacuum glazings are isolated from external damaging processes and from water vapour (which assists with crack propagation in glass [2]), the stresses at these internal surfaces also need not be considered in this failure analysis. Therefore, in this study of the probability of failure for vacuum glazings, only the stresses at the external glass surfaces are considered.

Most of the modelling work of the previous two chapters concerned the direct surface stresses in vacuum glazings because results were required for a comparison with measurements of similarly oriented associated strains, and because the study of these stresses assisted with the physical understanding of the origins of the stresses in vacuum glazing. Surface shear stresses were also briefly considered. However, it is the maximum and minimum principal stresses which must be considered for probability of failure calculations. The principal stresses act on planes on which there are no shear stresses [3]. The maximum and minimum principal stresses (respectively σ_{11} and σ_{22}) are calculated from the total direct stresses (σ_{xx} and σ_{yy}) and from the total shear stress (τ_{xy}) according to [3]:

$$\sigma_{11}, \sigma_{22} = \frac{(\sigma_{xx} + \sigma_{yy})}{2} \pm \sqrt{\left(\frac{\sigma_{xx} - \sigma_{yy}}{2}\right)^2 + \tau_{xy}^2} . \quad (8.1)$$

For this study of vacuum glazing, the total direct and shear stresses are the sums of the corresponding stresses due to the two main sources of stress: the presence of a temperature difference across the glazing, and atmospheric pressure.

Several different glass failure prediction models are reviewed by Fischer-Cripps and Collins [4]. The approach which is used in the present study is based on the method presented by Beason and Morgan [5] because, as noted below, it accounts for several important factors. The approach also includes a modification to this method that was developed by Fischer-Cripps and Collins to account for delayed failure from long term stresses [4]. Fischer-Cripps used this modified model to calculate the probability of failure for vacuum glazing [1, 6]. However, for the sake of simplicity, this analysis was based on the assumption of uniform and uniaxial stresses near the edges. The more exact solution procedure presented below correctly accounts for the non-uniform and biaxial nature of these stresses.

The total probability of glass failure can be expressed as a continuous probability distribution of the form [5]:

$$P_{f,total} = 1 - e^{-\beta} , \quad (8.2)$$

where β is the risk of failure function. According to Beason and Morgan, this function must reflect the following three factors which affect the strength of glass: the magnitude and duration of the tensile surface stress, the surface area of the glass exposed to this stress, and the geometries and orientations of the surface flaws with respect to the stress [5]. These factors are accounted for in the following expression [5]:

$$\beta = kA(c\sigma'_{11})^m , \quad (8.3)$$

where k and m are the surface flaw parameters, A is the glass surface area, c is the biaxial stress correction factor, and σ'_{11} is the equivalent maximum principal tensile stress. All of these terms are described below.

The equivalent maximum principal tensile stress, σ'_{11} , can be used to calculate the probability of delayed failure due to the long term application of a corresponding maximum principal tensile stress, σ_{11} . This is based on the assumption that the probability of long term delayed failure at σ_{11} is equal to the probability of instantaneous failure at σ'_{11} . Fischer-Cripps and Collins found that previous methods for calculating the equivalent stress (such as the method presented by Beason and Morgan [5]) were inadequate and thus they developed their own approach [4]. This approach is based on a consideration of the stress intensity factor, K , which governs the growth of glass flaws. Instantaneous failure occurs if K is greater than or equal to a critical value, i.e. if $K \geq K_{Ic}$ [1, 4]. Delayed failure can occur when non-critical flaws grow to a critical size. It is assumed that no such growth occurs for $K < K_{Isc}$, where K_{Isc} is the sub-critical crack growth stress intensity factor (which is also referred to as the static fatigue limit) [1, 4]. Under this condition, it is assumed that there is no delayed failure. Therefore, the relationship between the stresses corresponding to the delayed and instantaneous probabilities of failure is [1]:

$$\sigma'_{11} = \sigma_{11} \left(\frac{K_{Ic}}{K_{Isc}} \right) . \quad (8.4)$$

For soda-lime glass, the data of Wiederhorn et al. indicate that, at crack velocities for effective failure, $K_{Ic}=0.75 \text{ MPa m}^{1/2}$ [7]. Atkins and Mai report that $K_{Ic}=0.8 \text{ MPa m}^{1/2}$ [8]. In the present study, K_{Ic} is taken to be the average of these two values: $0.78 \text{ MPa m}^{1/2}$. Estimates of K_{Isc} for soda-lime glass can be determined from published data on the growth of cracks in glass. Since a glazing can be exposed to wet conditions (particularly on the side exposed to the outside ambient) both aqueous and moist air environments should be considered. A value of $K_{Isc}=0.25 \text{ MPa m}^{1/2}$ is implied by the data of Wiederhorn and Bolz for glass in water at 25°C [9]. The results of Wan et al. for glass in air at 50 % relative humidity suggest that $K_{Isc}=0.37 \text{ MPa m}^{1/2}$ (based on the elastic properties of soda-lime glass given in Appendix 1) [10]. Wiederhorn's measurements for glass in air at 50% relative humidity indicate that $K_{Isc}=0.30 \text{ MPa m}^{1/2}$ [11]. In the present study, K_{Isc} is arbitrarily taken to be equal to the above value attributed to just Wiederhorn: $0.30 \text{ MPa m}^{1/2}$.

If the maximum and minimum principal stresses are unequal, the probability of failure due to each flaw depends on the orientation of the flaw, θ , with respect to the stress field. This dependence is accounted for by the biaxial stress correction factor, c [5]:

$$c = \left[\frac{2}{\pi} \int_0^\alpha (\cos^2(\theta) + n \sin^2(\theta))^m d\theta \right]^{1/m}, \quad (8.5)$$

where n is the ratio of the minimum to the maximum principal stresses (i.e. $n=\sigma_{22}/\sigma_{11}$), and m is one of the surface flaw parameters described below. If the minimum principal stress is tensile, $\alpha = \pi/2$. For a compressive minimum principal stress [5]:

$$\alpha = \tan^{-1}(|n|^{-1/2}). \quad (8.6)$$

Beason and Morgan give values for c as a function of n for integer values of m ranging from 4 to 15 [5]. The results for $m=9$ are presented in Table 8.1. Values of c for non-tabulated values of n can be determined with sufficient accuracy by linear interpolation.

The glass surface flaw parameters k and m are determined on the basis of experimental results for the failure of glass plates. They reflect the characteristics of the surface flaws and thus they are strongly dependent on the weathering and other damaging processes to which the glass has been exposed. The parameter m indicates the spread in the statistical distribution of the flaw sizes (it varies inversely with the spread), and k indicates the absolute size and density of the flaws [4]. The parameter m is dimensionless and the parameter k is quoted in units of $\text{mm}^{2m-2} \text{ N}^{-m}$ [5]. Both the

magnitude and units of k are dependent on m . It is possible to express k in a form for which the units - but not the magnitude - are independent of m . However, in order to highlight the interdependence of the magnitudes of the two surface flaw parameters, k is usually expressed in the form such as the one given above [5]; this form is based on a conversion from the English units version of k which is used with stresses expressed in psi. In the present study, k is expressed in an equivalent form with units of $\text{m}^{-2}\text{Pa}^{-m}$ in order to be consistent with the fact that the stresses substituted into equation 8.3 are in Pa.

n	c
1.00	1.00
0.80	0.92
0.60	0.88
0.40	0.86
0.20	0.84
0.00	0.83
-0.20	0.82
-0.40	0.81
-0.60	0.81
-0.80	0.80
-1.00	0.80

Table 8.1: Values for the biaxial stress correction factor, c , as a function of n for $m=9$ [5]; $n=\sigma_{22} / \sigma_{11}$.

Several sets of surface flaw parameters for both new and weathered soda-lime glass are presented in Table 8.2. Each set of results for weathered glass corresponds to glass plates taken from the windows of a different building; the windows had been in service for 20 to 25 years. Brown’s value of $m=7.3$ [14] is based on the failure results of Bowles and Sugarman [15]. Brown calculated a value for k on the basis of this m value and the results of Kaiser [16].

Since current design procedures are based on the strength of new glass instead of on the strength of weathered glass [5], this study is based on the surface flaw parameters for new glass. These parameters are taken to be equal to Abiassi’s results listed in Table 8.2 for new glass: $m=9$, and $k=1.32 \times 10^{-69} \text{ m}^{-2}\text{Pa}^{-6}$. It should be noted that the failure prediction results of Beason and Morgan indicate that it might be more appropriate to base failure analyses on the surface flaw parameters for weathered

glass [5]. It should also noted that there can be significant differences in the surface flow parameters of soda-lime glass samples of different manufacture or origin (as indicated by the data sets of Table 8.2, which are for different batches of glass); therefore, it might be best to measure and then to use surface flow parameters for the particular brands of glass that are employed in vacuum glazing and that have been exposed to the glazing manufacturing process. However, due to the extensive experimental work that would be required to determine the flow parameters, this is not a practical option. Furthermore, it is recalled that the purpose of this study is to obtain representative results for approximately typical conditions and, in so doing, to demonstrate the method of calculation. The same approach can be used to obtain results for other (and possibly more appropriate) surface flow parameters. In this context, the exact choices for the surface flow parameters are not crucial to this study.

Reference	Glass type	<i>m</i>	<i>k</i>
Beason and Minor [12]	Weathered	6	6.33 x 10 ⁻⁴⁵ m ² Pa ⁻⁶
Abiassi [13]	Weathered	6	3.01 x 10 ⁻⁴⁵ m ² Pa ⁻⁶
Abiassi [13]	Weathered	5	9.06 x 10 ⁻³⁸ m ² Pa ⁻⁵
Abiassi [13]	New	9	1.32 x 10 ⁻⁶⁹ m ² Pa ⁻⁶
Brown [14]	New	7.3	5 x 10 ⁻⁵⁷ m ² Pa ⁻⁶

Table 8.2: Glass surface flow parameters for soda-lime glass.

The surface flow parameters considered above correspond to the region remote from the edges of the glass sheets. Due to the damage that is inherent in the cutting of the sheets, the nature of the surface flaws is very different at and close to the edges than elsewhere across the sheets (the flaws are presumably more numerous and larger at and near the edges). Therefore, the surface flow parameters within the edge region can be expected to differ from those of the central region of the glazing. An exact failure analysis should take into account these different parameters - especially since, as shown below, the greatest contribution to the total probability of failure comes from the region near the edges. However, since the surface flow parameters for cut edges are unknown, and since they are probably strongly dependent on the cutting technique, they are not considered in this analysis. Therefore, *m* and *k* are assumed to be uniform across the external glass surfaces. This is consistent with the failure calculation approaches of Beason and Morgan [5] and Fischer-Cripps and Collins [4].

If the glass surface stresses are highly non-uniform (as is the case in vacuum glazing), it is necessary to calculate the probability of failure on the basis of many small individual elements over which the stresses are fairly uniform. The elemental probability of survival, $P_{s,e}$, is calculated according to the exponential term of equation 8.2, with β determined as per equation 8.3 on the basis of the elemental area A_e and the elemental equivalent maximum principal tensile stress, $\sigma'_{11,e}$ (which is calculated from the average direct and shear stresses across the element):

$$P_{s,e} = e^{-kA_e(c_e\sigma'_{11,e})^m}, \quad (8.7)$$

where the subscript "e" denotes the elemental quantity. The total probability of failure for the glazing, $P_{f,total}$, is equal to unity minus the product of all of the individual elemental probabilities of survival:

$$P_{f,total} = 1 - P_1 \cdot P_2 \cdot P_3 \cdot \dots \cdot P_N, \quad (8.8)$$

where N is the total number of elements. This can also be expressed as:

$$P_{f,total} = 1 - e^{-k \sum_{i=1}^N A_e (c_e \sigma'_{11,e})^m}. \quad (8.9)$$

8.3 Solution Procedure

Only glazings of size 1 m by 1 m are considered in this study. The solution procedure is explained below in the context of a pillar separation of $\lambda=25$ mm. The same approach is used to obtain results for other separations. A key step in the solution procedure is the combination of the stresses due to a temperature difference across the glazing, and the stresses due to atmospheric pressure. In order to simplify greatly this step, several assumptions are made.

It was noted in Section 6.6.5 that the temperature induced direct stresses along the edges of the glazing are fairly uniform except within about 150 mm from the glazing corners (as shown in Figs. 6.26). In this study, for the sake of simplicity, the non-uniformities in the corner regions are ignored and the stresses along the edges are assumed to be equal to the corresponding centerline results (which are presented in Fig. 6.27b). The results along each edge are assumed to extend to the lines of symmetry for the direct stresses: the diagonals of the square glazing. Based on these

assumptions, the temperature induced stresses of Figs. 6.26 are represented as shown in Figs. 8.1. It is recalled from Section 6.6.5 that the centerline results for Fig. 6.27b were appropriately smoothed in order to remove the slight non-uniformities in the stresses of Figs. 6.26 that result from the grading in the node density of the model; therefore the centerline results of Fig. 6.27b - and thus the results of Figs. 8.1 - are not exactly the same as the corresponding stresses shown in Figs. 6.26. However, these smoothed results are a reasonable representation of the centerline stresses.

It is recalled from Section 7.4 that, beyond about the width of two unit cells (2λ) from the glazing corners, there is a uniform periodicity in the atmospheric pressure induced direct stresses along the edges of the glazing. This is clearly shown in Fig. 7.10. As with the temperature induced stresses, the non-uniformities in the pressure induced stresses of the corner regions are ignored in this study; across the unit cells within 3λ from the edges, the stress profiles are assumed to be the same as those across the cells away from the corner and at the same distance from the edge. Again, the direct stresses along each edge are assumed to extend only to the glazing diagonals. As mentioned in Section 7.4, there is a uniform periodicity in the stresses across the unit cells beyond 3λ from the edges. Therefore, the pressure induced external glass surface stresses of Fig. 7.10 are represented as shown in Fig. 8.2. As explained in Section 7.3.1, the results of this figure can be extrapolated to dimensions beyond those which are shown in order to obtain the stresses across the entire glazing. The atmospheric pressure stresses at the external glass surfaces are the same on both sides of the glazing and, as explained in Section 7.3.1, they are assumed to be zero over the edge seal.

The decision to ignore the non-uniform stresses in the corner regions is justified on two grounds. Firstly, for the situation of unconstrained edges (which is what is considered in this study), these stresses occur over a small portion of the total glazing area. For the temperature and pressure induced stresses, the corner regions of non-uniform stress respectively represent 9 and 1 % of the total area of the 1 m by 1 m glazings considered in this study. Secondly, and more importantly, within these regions, the total stresses - and thus the contributions to the total probability of failure - are less than elsewhere in the glazing. Therefore, the above assumptions relating to the uniformity of the stresses along the edges leads to a slight overestimation of the total probability of failure. This overestimation, which is estimated below, can be thought of as representing a fractional safety factor that is inherent in the solution procedure.

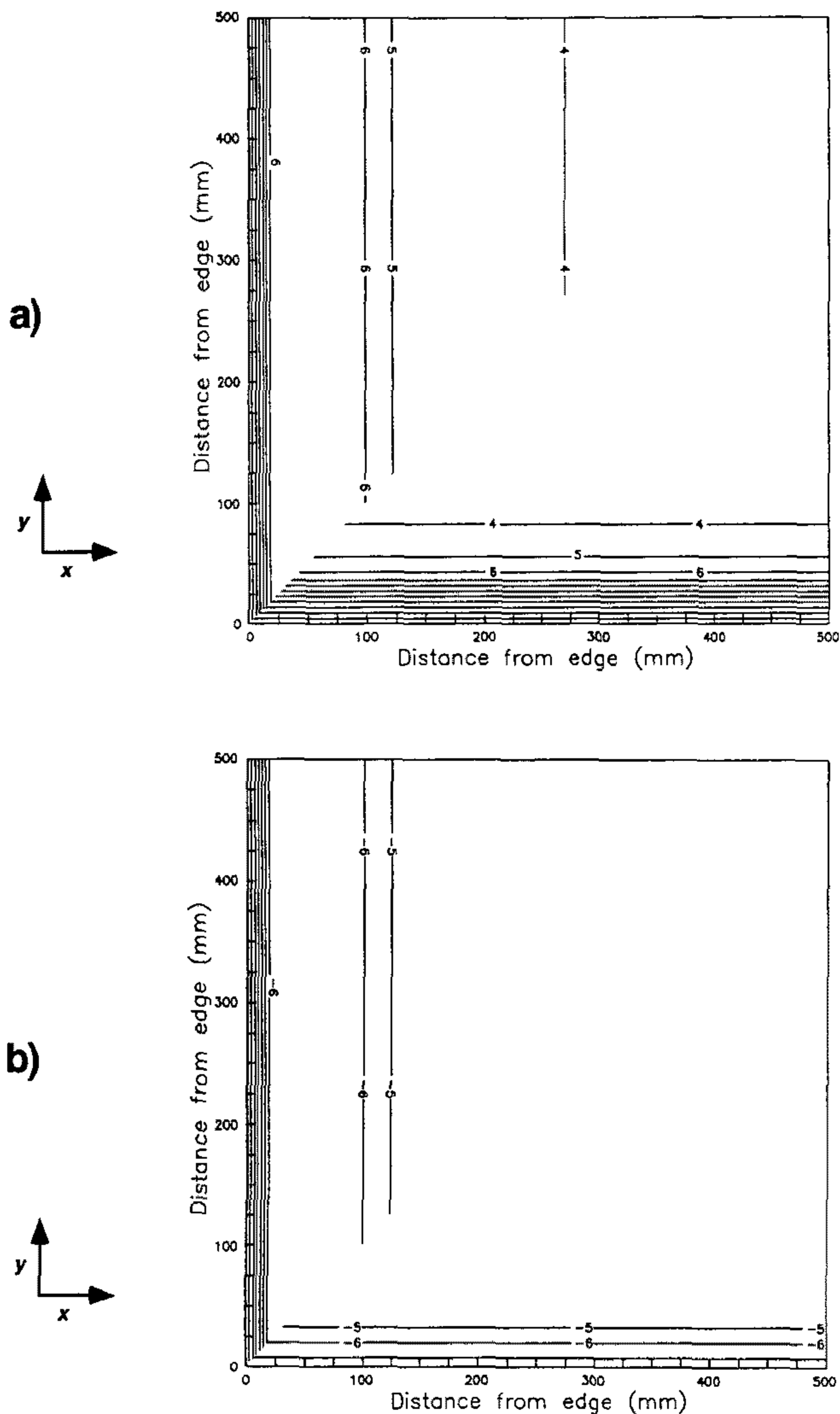


Fig. 8.1: A simplified representation of the temperature induced external surface stresses of Figs. 6.26 (which have been appropriately smoothed) for (a) the warm side, and (b) the cold side of a glazing quarter-section, as functions of the distance from the glazing edge. These stresses are parallel to the x-axis; the stresses parallel to the y-axis are symmetric with these results. The stresses along the each edge are assumed to be uniform and equal to the corresponding centerline results, and to extend to the glazing diagonals. The results correspond to a full 1 m by 1 m glazing, the ASTM standard fenestration test conditions and $U_{g-g}=1.10 \text{ Wm}^{-2}\text{K}^{-1}$. The edge seal width is 8 mm and the glass thickness is 4 mm. The stress contour interval for both plots is 1 MPa. In each plot, the stresses along the left border approach zero with proximity to the edge, and the stresses along the bottom border increase in magnitude with proximity to the edge.

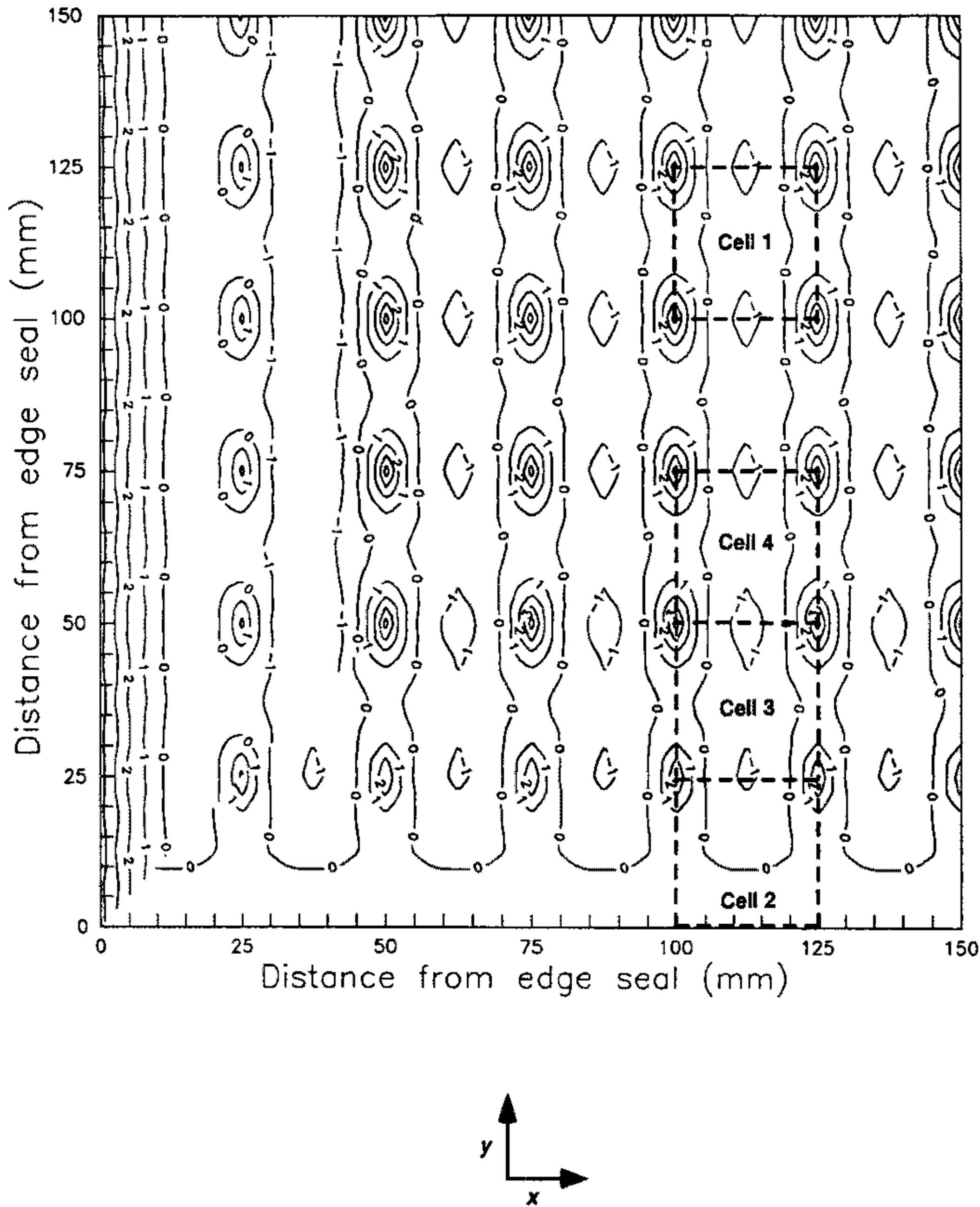


Fig. 8.2: A simplified representation of the pressure induced stress results of Fig. 7.10 for a corner section of vacuum glazing, as functions of the distance from the inner edge of the edge seal. The region over the seal, where these stresses are assumed to be zero, is not shown. The external glass surface stresses parallel to the x-axis are presented, and the stresses parallel to the y-axis are symmetric with these results. Four representative unit cells are bordered by dashed lines and labelled. There is a uniform periodicity in the stresses across the unit cells beyond 3λ from the edge seal (e.g. cell 1). Within 3λ from the edge seal, the stresses along each edge are assumed to be uniformly periodic and to extend to the glazing diagonals. The stress profiles across the unit cells of this edge region are equal to those across the cells which are at the same distance from the edge: cells 2, 3 and 4 (based on the results of Fig. 7.10 away from the corners). The results correspond to alumina pillars of radius 0.125 mm, height 0.20 mm, and separation 25 mm. The simulated glass thickness is 4 mm. The contour interval is 1 MPa.

Figures 6.31 and 7.16 show that the shear stresses due to the two sources of stress are not uniform along the edges. However, in this study, the non-uniformities in these stresses are ignored; for each sheet of glass, the shear stress from each source is assumed to be uniform and equal to the average corresponding stress across the sheet. This simplification is justified on the basis that, over most of the glazing, the shear stresses are about an order of magnitude less than the direct stresses; therefore, as indicated by equation 8.1, the shear stresses do not have a significant influence on the calculation of the maximum and minimum principal stresses - and thus on the calculation of the total probability of failure. The error associated with this assumption is estimated below. The average temperature and pressure induced shear stresses were determined from Figs. 6.31 and 7.16 and the results are presented in Table 8.3. For the temperature induced shear stresses, the large non-uniformities near the glazing corner were ignored in order to provide results which apply over *most* of each side of the glazing. The average pressure induced shear stress is negligibly small (0.003 MPa) and thus it is assumed to be zero - as tabulated. The results of Table 8.3 are for $\lambda=25$ mm. For the other pillar separations considered in this study, the average temperature induced shear stresses change very little (by less than about 0.1 MPa), and the average atmospheric pressure induced shear stresses are negligible (they range from -0.015 to 0.007 MPa). Therefore, for all of the pillar separations of this study, the shear stresses for each side of the glazing are assumed to be uniform and equal to the total results presented in Table 8.3.

It is noted that since the tabulated total shear stresses for the warm and cold sides of the glazing are equal in magnitude (at least for this case), and since the shear stress is only used in equation 8.1 (where it is squared), either total value could be applied to both sides of the glazing. It is also noted that, if both orientations of the direct stress are small in magnitude (approximately less than 2 MPa), the shear stress can be significant in calculating the principal stresses. In such a situation, the assumed uniform shear stresses can lead to large and unrealistic errors in the principal stresses. However, since the direct stresses are so small, the corresponding elemental maximum principal stress - and thus the elemental probability of failure - can also be expected to be small. Due to the relative magnitudes of the temperature and pressure induced stresses, this situation only occurs on the cold side of the glazing which, as explained below, makes no contribution to the total probability of failure for the pillar separations considered in this study. Therefore, this error associated with the assumption of uniform shear stresses can be ignored.

The two orientations of direct stress that were considered in the previous two chapters were parallel to the x- and the y-axes of the finite element models. For both models, these axes lie along the outer edges of the simulated glazing quarter-sections. Since only square glazings were considered, the stresses parallel to the x-axis are symmetric with the corresponding stresses parallel to the y-axis. Due this symmetry, and due to the nature of equation 8.1, it does not matter which of these stresses corresponds to which total direct stress (σ_{xx} or σ_{yy}) of equation 8.1. However, as the subscripts suggest, the stresses parallel to the x-axis and the y-axis are taken to correspond to σ_{xx} and σ_{yy} , respectively. The symmetry in the direct stresses and the assumption of uniform shear stresses ensure that the same maximum and minimum principal stresses - and thus the same elemental probabilities of survival - are obtained at the same distance from each edge of the glazing. It is recalled that, for the purpose of assisting with the understanding of the stresses, the stresses parallel to the x- and y-axes were also previously referred to as the stresses parallel and normal to the edge, respectively.

Type of stress	Average shear stress, τ_{xy} (MPa)	
	Warm side	Cold side
Temperature	-0.5	0.5
Pressure	0.0	0.0
Total	-0.5	0.5

Table 8.3: The estimated average temperature and pressure induced shear stresses for both sides of the glazing based on the results of Figs. 6.31 and 7.16 for $\lambda=25$ mm. The temperature induced results correspond to most of the glazing surface. For all of the pillar separations considered in this study, the shear stresses for each side of the glazing are assumed to be uniform and equal to the total results given in this table.

The temperature induced direct stresses can be determined from a single set of one-dimensional centerline results. In contrast, there is a two-dimensional variation in the pressure induced direct stresses across each glass sheet. This variation necessitates a two-dimensional solution for the probability of failure. However, it is possible to take advantage of the periodicity in the variation of the pressure induced stresses in order to simplify the solution. The results of Fig. 8.2 show two regions within which the pressure induced stresses are uniformly periodic: beyond 3λ from the edge seal, and within 3λ from the edge seal. For the first region, the stress profile across each unit cell is the same. Therefore, it is only necessary to consider one representative unit cell from this region - e.g. cell 1 of Fig. 8.2. For the second region, there is a uniform periodicity in the stresses along the edges (the apparent non-uniformity in the cells

along the glazing diagonal is not a concern due to the symmetry of the stresses). This periodicity corresponds to the individual unit cells along the edges and thus one need only consider a representative section of this region - e.g. the section which is comprised of cells 2, 3 and 4 of Fig. 8.2. As explained above, due the symmetry of the direct stresses and due to the assumed uniformity of the shear stresses, the same maximum and minimum stresses will be obtained regardless of the edge to which this section is adjacent. Therefore, in order to calculate the total probability of failure for the entire external surface area, it is only necessary to consider the above four representative unit cells for each side of the glazing.

Each of the four representative cells corresponds to a certain number of cells within or beyond the same distance from the edge seal of the glazing. For example, for a 1 m by 1 m glazing with an edge seal width of 8 mm, and for $\lambda=25$ mm, there are a total of 1549.21 cells within the 984 mm by 984 mm area bounded by the inner edge of the seal. Of this number, a total of 153.44 cells can fit within a 25 mm wide strip around the edge of the area (i.e. within λ of the edge seal), 145.44 cells can fit within a 25 mm wide strip adjacent to the first strip (i.e. between λ and 2λ from the edge seal), and 137.44 cells can fit within a 25 mm wide strip adjacent to the second strip (i.e. between 2λ and 3λ from the edge seal). This leaves 1112.89 cells beyond 3λ from the edge seal. Therefore, the number of unit cells, N_{cell} , corresponding to each of cells 1, 2, 3, and 4 of Fig. 8.2 is, respectively, 1112.89, 153.44, 145.44, and 137.44 cells. In actual glazings, the pillars closest to the edge seal (i.e. the pillars of the first rows from the edge) are generally spaced either closer to or further from the seal than the regular pillar separation so that the unit cells beyond these pillars are of a uniform size (of λ by λ). This avoids the presence of fractional unit cells over most of the glazing. Such a non-uniform spacing of the pillars closest to the edge will affect the nearby stresses, as indicated by the results of Figs. 7.11. However, for the sake of simplicity, and for the sake of consistency in the solution procedure for different pillar separations, the actual non-uniform spacing is not considered in this analysis and the values of N_{cell} are determined as in the above example. Furthermore, the stresses over complete cells are assumed to apply over the extents of any fractional cells indicated by non-integer values of N_{cell} . The techniques presented below can be used to calculate results for different positions of the pillars closest to the edge seal.

For both the warm and cold side external surfaces of the glazing, each of the four representative unit cells is divided into a regular array of 25 elements by 25 elements. For $\lambda=25$ mm, the size of each element is 1 mm by 1 mm. The temperature and pressure induced stresses at the nodes which define the corners of the elements are

determined by interpolating among the respective finite element modelling results (which correspond to different node positions and densities). For the pressure induced stresses, the interpolation is performed on results for which the stresses right over the pillars have been determined from the modelling data in the manner described in Section 7.3.3. As explained in Section 7.3.1, the atmospheric pressure induced direct and shear stresses at the external glass surfaces over the edge seal are assumed to be zero. For the sake of simplicity, the temperature induced stresses beyond cell 4 (i.e. beyond 3λ plus the width of the edge seal) are assumed to be uniform and equal to the center-of-glass results: 3.9 MPa on the warm side, and -4.3 MPa on the cold side.

For each element, the elemental probability of survival is calculated according to the following procedure:

1. The elemental temperature induced direct stresses parallel to the x- and y-axes are set equal to the glazing centerline results which correspond to the mid-element distance from the edge of the glazing.
2. The elemental pressure induced direct stresses parallel to the x- and y-axes are set equal to the averages of the corresponding stresses at the four corner nodes of the element.
3. The total elemental direct stresses parallel to the x- and y-axes, $\sigma_{xx,e}$ and $\sigma_{yy,e}$, are calculated by adding the corresponding temperature and pressure induced results determined from the above two steps.
4. The total elemental shear stress, $\tau_{xy,e}$, is set equal to the total value given in Table 8.3 for the side of the glazing to which the element corresponds. Due to the assumption of uniform shear stresses, the elemental shear stresses are actually the same for each side of the glazing.
5. The results for $\sigma_{xx,e}$, $\sigma_{yy,e}$, and $\tau_{xy,e}$ are substituted into equation 8.1 to obtain the elemental maximum and minimum principal stresses, $\sigma_{11,e}$ and $\sigma_{22,e}$.
6. As explained above, it is assumed that maximum principal stresses which are compressive do not contribute significantly to the total probability of failure. A zero maximum principal stress (which of course does not occur if non-zero values for $\tau_{xy,e}$ are used) yields a zero probability of failure, as indicated by equations 8.2 and 8.3. Therefore, if $\sigma_{11,e} \leq 0$, the elemental probability of survival is set to 1 and no further calculations are required. However, if $\sigma_{11,e}$ is tensile, the solution proceeds according to the following steps.
7. The elemental equivalent maximum principal tensile stress, $\sigma'_{11,e}$, is calculated according to equation 8.4 on the basis of $\sigma_{11,e}$ and the previously given values of $K_{Ic}=0.78 \text{ MPa m}^{1/2}$ and $K_{Isc}=0.30 \text{ MPa m}^{1/2}$.
8. The elemental biaxial stress correction factor, c_e , is determined from Table 8.1 according to the ratio $n=\sigma_{22,e} / \sigma_{11,e}$. Linear interpolation is used to determine c_e for non-tabulated values of n .
9. If all of the unit cells of the glazing were subdivided in the previously described manner, the above stresses would also apply to the elements at the same locations

within all of the unit cells which correspond to the representative cell being considered. In order to obtain failure results for the entire glazing surface, the elemental area is multiplied by the appropriate value of N_{cell} to give the total elemental area, $A_{e,total}$. For example, with $\lambda=25$ mm, the total elemental area corresponding to an element in cell 1 is $A_{e,total}=(1 \text{ mm}^2) \times 1112.89$, or $1.11289 \times 10^{-3} \text{ m}^2$. The sum of all of the total elemental areas, including those over the seal (which are referred to below), is equal to the total external surface area of the glazing.

10. The above values for $\sigma_{11,e}$, c_e , and $A_{e,total}$ are substituted into equation 8.7 along with the previously given values of $m=9$ and $k=1.32 \times 10^{-69} \text{ m}^2\text{Pa}^{-6}$ in order to obtain the elemental probability of survival ($A_{e,total}$ is substituted for A_e in this calculation).

The above ten steps are repeated for all elements of the warm and cold side surfaces of each representative unit cell. A similar procedure is applied over the edge seal except that, since the pressure induced stresses are zero over this region, only the temperature induced stresses need be considered (i.e. step 2 above is skipped). Furthermore, since the temperature induced stress profile is uniform over the seal, only a single section of eight 1 mm by 1 mm elements over the 8 mm width of the seal need be considered; the corresponding total elemental areas depend on the total number of similar elements that can be placed at the same distances from the edge of the glazing.

The total probability of failure, P_{total} , is determined from all of the elemental probabilities of survival according to equation 8.8. The entire solution procedure is equivalent to equation 8.9. A computer program was written in Fortran code to calculate the total probability of failure according to the entire procedure described above.

Different pillar separations are considered in this study in order to investigate the effect of λ on $P_{f,total}$. Since the pillar array component of the glass-to-glass center-of-glass U-value, U_{g-g} , varies with λ (as per equation 4.3), it is necessary to determine U_{g-g} for each simulated pillar separation. This is done according to the iterative procedure mentioned half-way through Section 5.3 on the basis of the ASTM standard fenestration test conditions, a typical glass-to-glass radiative U-value of $0.80 \text{ Wm}^{-2}\text{K}^{-1}$ (for two sheets of K glass with internal surface temperatures of 23 and 3 °C), and a typical pillar thermal resistance of $4 \times 10^3 \text{ KW}^{-1}$. The resulting value for U_{g-g} is used in the thermal analysis step of the modelling to obtain the temperature induced stresses. Values of U_{g-g} determined in this manner for the pillar separations considered in this study are presented in Table 8.4.

The effect of the edge insulation recess depth on the total probability of glazing failure is also investigated. It is assumed that the insulation is mounted in such a way that it

does not constrain the edges of the glazing. Therefore, edge insulation only affects the temperature induced stresses. With insulated edges, the solution procedure is thus as described above, but it is based on the appropriate centerline temperature induced stresses for the specified recess depth, thermal conductance and arrangement of the insulation. Temperature induced stress results for several recess depths of insulation are presented in Figs. 6.28b and 6.29.

λ (mm)	U_{g-g} (Wm ⁻² K ⁻¹)
20	1.32
25	1.10
30	0.98
35	0.90
40	0.86

Table 8.4: Total glass-to-glass center-of-glass U-values, U_{g-g} , for the values of pillar separation, λ , considered in this study. The results are determined from an iterative solution on the basis of the ASTM standard fenestration test conditions, a typical glass-to-glass radiative U-value of 0.80 Wm⁻²K⁻¹ (for two sheets of K glass with internal surface temperatures of 23 and 3 °C), and a typical pillar thermal resistance of 4 x 10³ KW⁻¹.

8.4 Results

All of the results of this study are based on the previously given values for K_{lc} , K_{lsc} , m and k . The results also correspond to a glazing with a size of 1 m by 1 m, the above typical radiative U-value for two sheets of K glass, a glass thickness of 4 mm, an edge seal of width 8 mm, and alumina (ceramic) pillars of radius 0.125 mm and of height 0.20 mm. The temperature and pressure induced stresses are determined from the finite element models presented in the previous two chapters. Each set of temperature induced stresses corresponds to the ASTM standard fenestration test conditions, and to the glass-to-glass center-of-glass U-value given in Table 8.4 for the specified pillar separation. For λ =25 mm, the pressure induced stresses are based on the Young's modulus and shear modulus given in Section 7.3.2 for simulated pillars (beams) which correspond to alumina pillars with the above dimensions: E_{beam} =2.55 x 10¹⁰ Pa, and G_{beam} =1.05 x 10¹⁰ Pa. All of the results are based on the thermal and elastic properties for soda-lime and solder glass presented in Appendix 1.

The computer program for the solution procedure was tested with the following uniform stresses on both sides of the glazing: x- and y-components of the temperature induced stress equal to 5 MPa, x- and y-components of the pressure induced stress equal to 5 MPa, and a shear stress of 1 MPa. For this test, the pressure induced stresses were, for the sake of simplicity, applied over the edge seal. The total probability of failure calculated by the program is 0.01706. This probability can be compared with the exact expected result. The expected total probability of survival is determined from equation 8.7 by substituting for the elemental quantities the following values which apply to the entire glazing: $\sigma'_{11}=28.60$ MPa, $c=0.9273$, and $A=2\text{m}^2$ (the total external surface area of the 1 m by 1 m glazing). The resulting expected total probability of survival, 0.98301, corresponds to a total probability of failure of 0.01699 ($P_{f,total}=1-P_{s,total}$). Therefore, the program gives a probability of failure which is only 0.4 % greater than the expected result. This discrepancy is likely due to the floating point error associated with the calculations and it is so insignificantly small that the usual solution to such a situation - the use of double-precision - need not be employed. It is concluded that the close agreement between the calculated and expected results validates the computer solution.

The following six figures, Figs. 8.3 to 8.8, present contour plots of the intermediate elemental results of the solution procedure for a pillar separation of $\lambda=25$ mm. These figures show results along a representative section of the glazing which extends for a distance of 5λ along the edge, and a distance of 6λ plus the width of the edge seal ($w_{seal}=8$ mm) from the edge. These plots provide representative results for near and remote from the edge which apply over the entire glazing. It is noted that there are two expected discontinuities in the results. One discontinuity is at the inner edge of the seal (at 8 mm) and it results from the assumption that there are no pressure induced stresses over the seal. The other discontinuity occurs at $3\lambda+w_{seal}$ (83 mm) from the edge because, beyond this location, the temperature induced stresses are assumed to be uniform and equal to the center-of-glass stresses. Depending on the contour locations (i.e. on the specified contour intervals), these discontinuities lead to non-uniformities in some of the nearby contours due to the interpolation routines of the plotting program. The latter discontinuity is particularly pronounced and it reflects the fact that, as shown in Fig. 6.27b, the warm side temperature induced stresses near 83 mm are slightly larger than the center-of-glass values. As will be explained, this does not affect the calculation of the total probability of failure. In Figs. 8.3 to 8.8, the x- and y-axes are respectively parallel to the horizontal and vertical borders of the plots.

It is at this point possible to predict the contribution from the cold side of the glazing to the total probability of failure. The addition of the cold side temperature induced stress results of Fig. 6.27b to the corresponding pressure induced stress results of Fig. 7.9 shows that, except immediately over the pillars away from the edges, there is no location along a row of pillars at which the maximum principal stress will be tensile. This is also true along other transits between pillar rows, where the pressure induced stresses are less than those shown in Fig. 7.9. Therefore, aside from points above certain pillars, the stresses on the cold side are entirely compressive. The tensile stresses over the pillars are so small (< 0.2 MPa) that they can be expected to result in negligible contributions to the total probability of failure; in fact, due to the exponent underflow associated with the very small risk of failure functions over these pillars, the corresponding elemental probabilities of survival are calculated to be 1. The rest of the cold side of the glazing also makes no contribution to the total probability of failure since, as previously noted, compressive stresses do not lead to failure. Therefore, as was indeed found to be the case from the calculations for $\lambda=25$ mm, there is no contribution to $P_{f,total}$ from any element on the cold side of the glazing. This is also true for the other pillar separations considered in this study; even for the larger pillar separations, the increased cold side stresses over the pillars are not large enough to yield risk of failure functions that result in elemental probabilities of survival other than unity. Since the stresses on the cold side of the glazing do not affect $P_{f,total}$, they are not important in the context of this study. Therefore, no results are presented below for the cold side.

Figures 8.3 and 8.4 respectively present the total direct stresses parallel to the x- and y-axes for the warm side of the glazing (i.e. σ_{xx} and σ_{yy}). These results were determined from the combination of the temperature and pressure induced stresses (step 3 above). The pillars are at the centers of the oval shaped patterns.

Figures 8.5 and 8.6 respectively present the warm side minimum and maximum principal stresses (i.e. σ_{22} and σ_{11}) which correspond to the results of Figs. 8.3 and 8.4, and to the uniform shear stresses of Table 8.3. These results were determined according to step 5 of the above solution procedure. It is noted that the minimum and maximum principal stresses are entirely tensile on the warm side of the glazing. Results for the equivalent maximum principal tensile stress multiplied by the biaxial stress correction factor (i.e. $c\sigma_{11}$) as per step 7 above are presented in Fig. 8.7.

The logarithm of the elemental probability of failure across the warm side external surface of the glazing is presented in Fig. 8.8 (the elemental probability of failure is

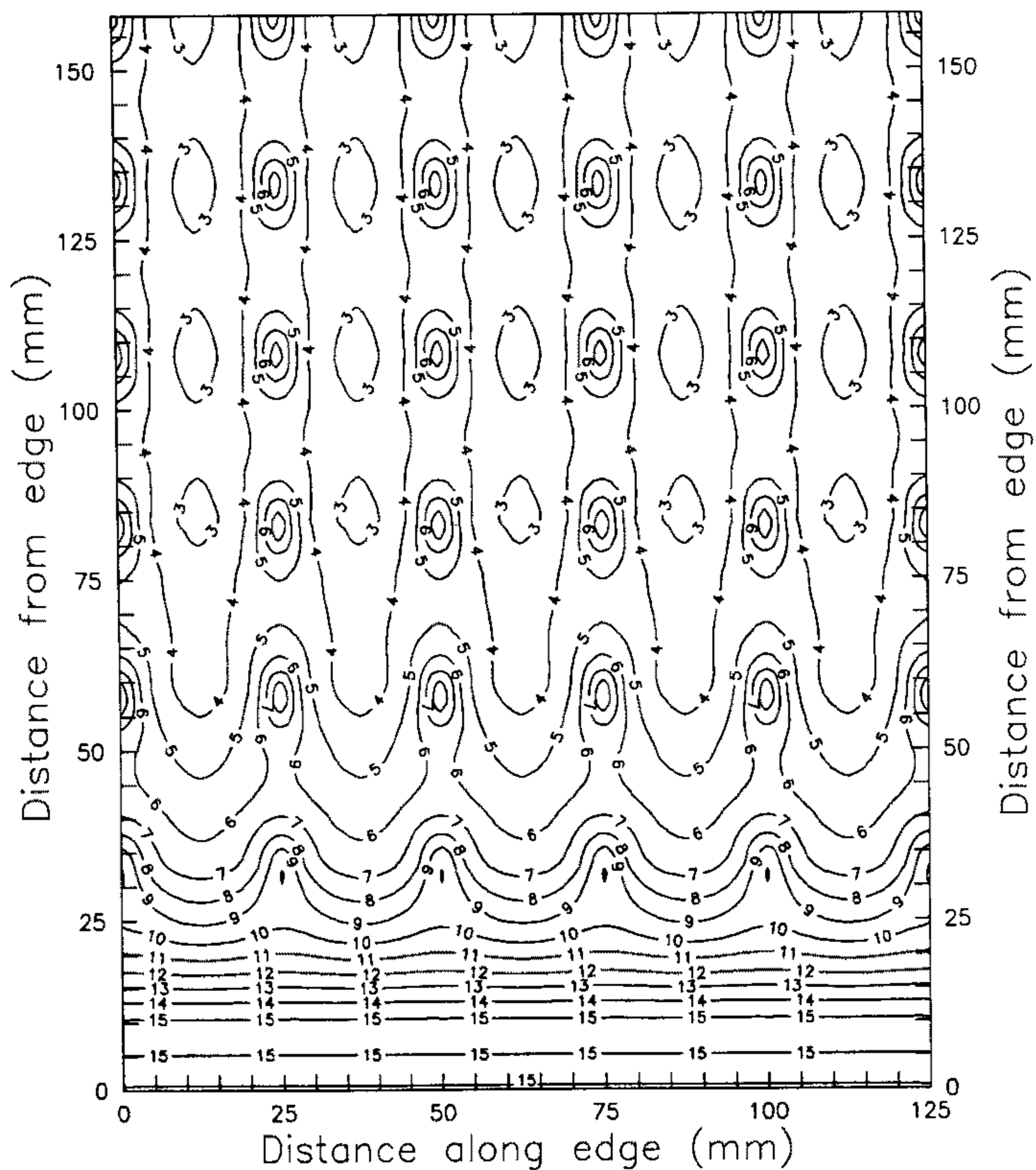


Fig. 8.3: The total direct stress parallel to the x-axis, σ_{xx} , for a section of the warm side of a vacuum glazing for the dimensions and conditions given for Figs. 8.1 and 8.2. These results were determined by combining the results of Figs. 8.1 and 8.2. The x-axis is parallel to the horizontal borders of the plot. The stress contour interval is 1 MPa.

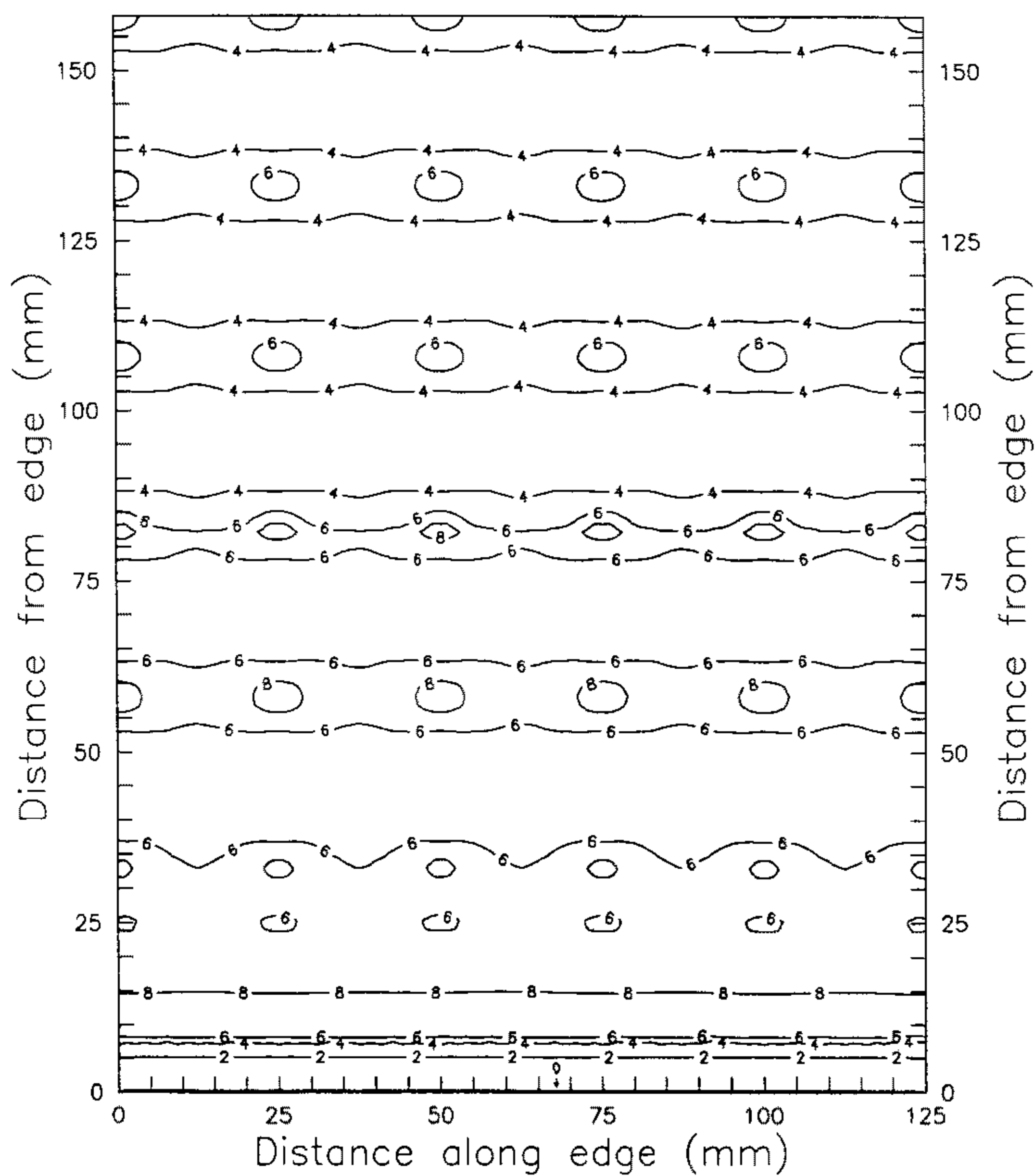


Fig. 8.4: The total direct stress parallel to the y-axis, σ_{yy} , for a section of the warm side of a vacuum glazing for the dimensions and conditions given for Figs. 8.1 and 8.2. These results were determined by combining the results of Figs. 8.1 and 8.2. The y-axis is parallel to the vertical borders of the plot. The slight non-uniformities in the contours near 8 and 83 mm from the edge are explained near the beginning of this section. The stress contour interval is 1 MPa.

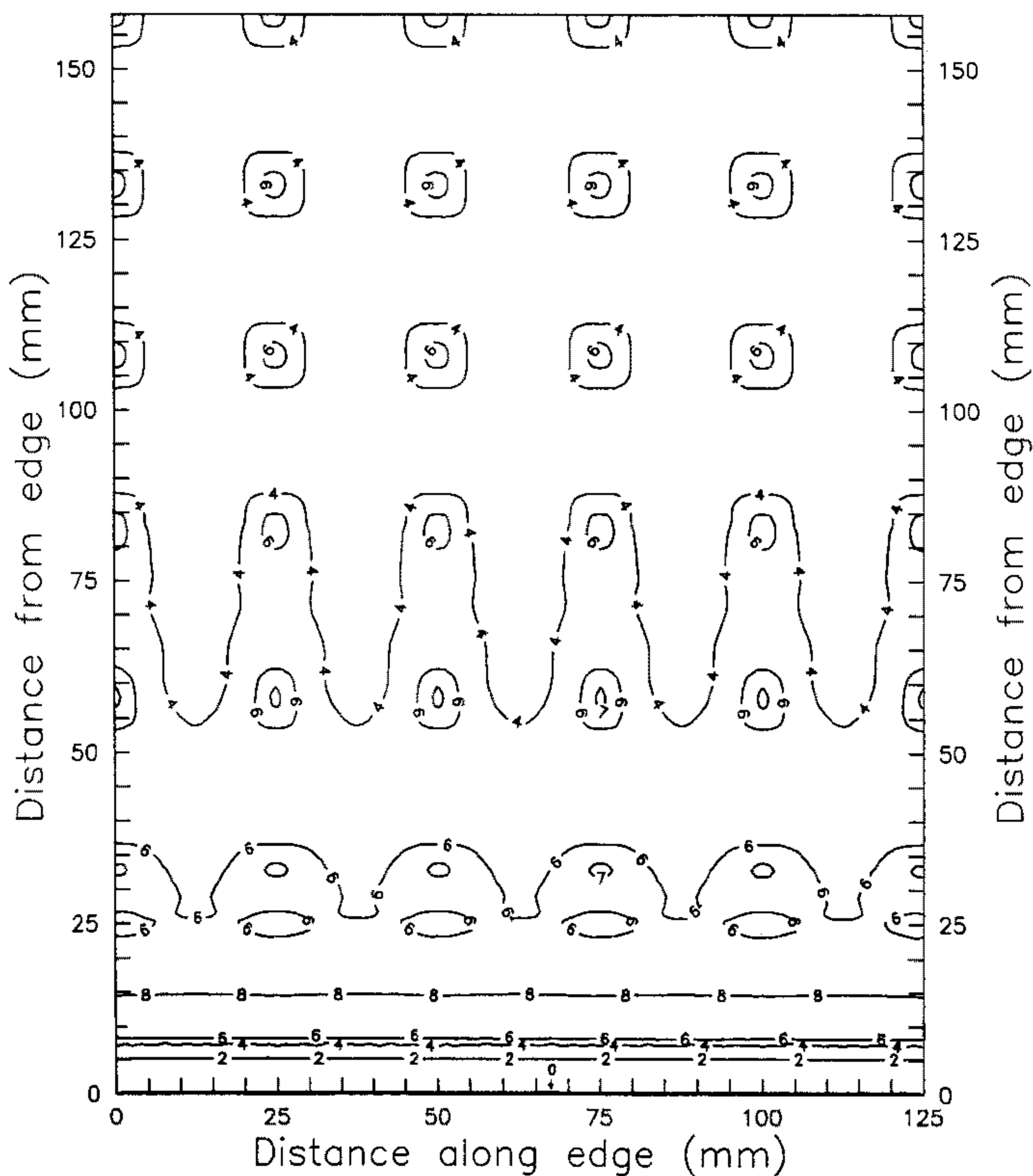


Fig. 8.5: The minimum principal stress, σ_{22} , for a section of the warm side of a vacuum glazing for the dimensions and conditions given for Figs. 8.1 and 8.2. These results were determined by substituting into equation 8.1 the direct stresses of Figs. 8.3 and 8.4, and the shear stresses of Table 8.4. The slight non-uniformity in one of the contours near 8 mm from the edge is explained near the beginning of this section. The stress contour interval is 1 MPa.

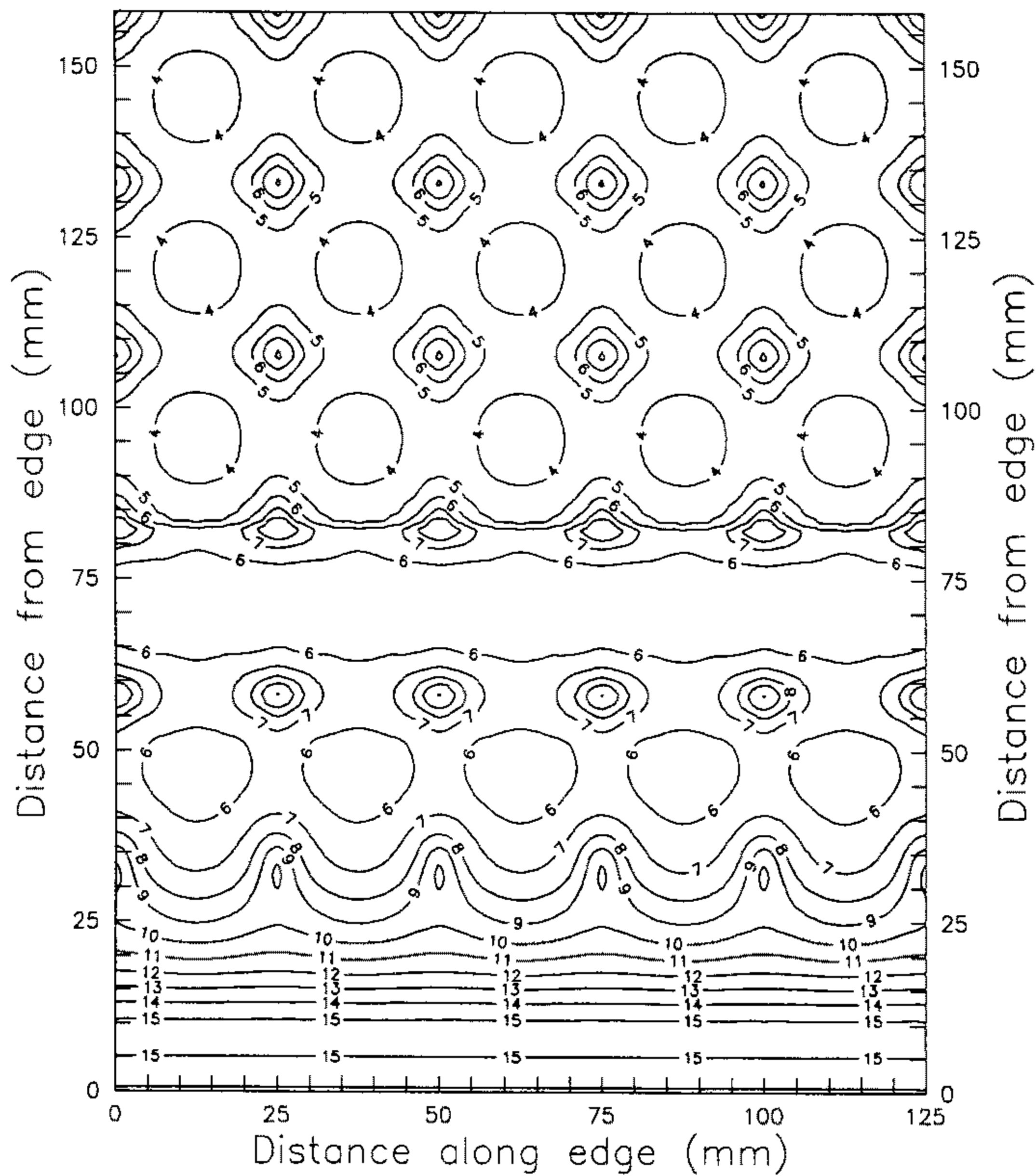


Fig. 8.6: The maximum principal stress, σ_{11} , for a section of the warm side of a vacuum glazing for the dimensions and conditions given for Figs. 8.1 and 8.2. These results were determined by substituting into equation 8.1 the direct stresses of Figs. 8.3 and 8.4, and the shear stresses of Table 8.4. The non-uniformity in the contours at about 83 mm is explained near the beginning of this section. The stress contour interval is 1 MPa.

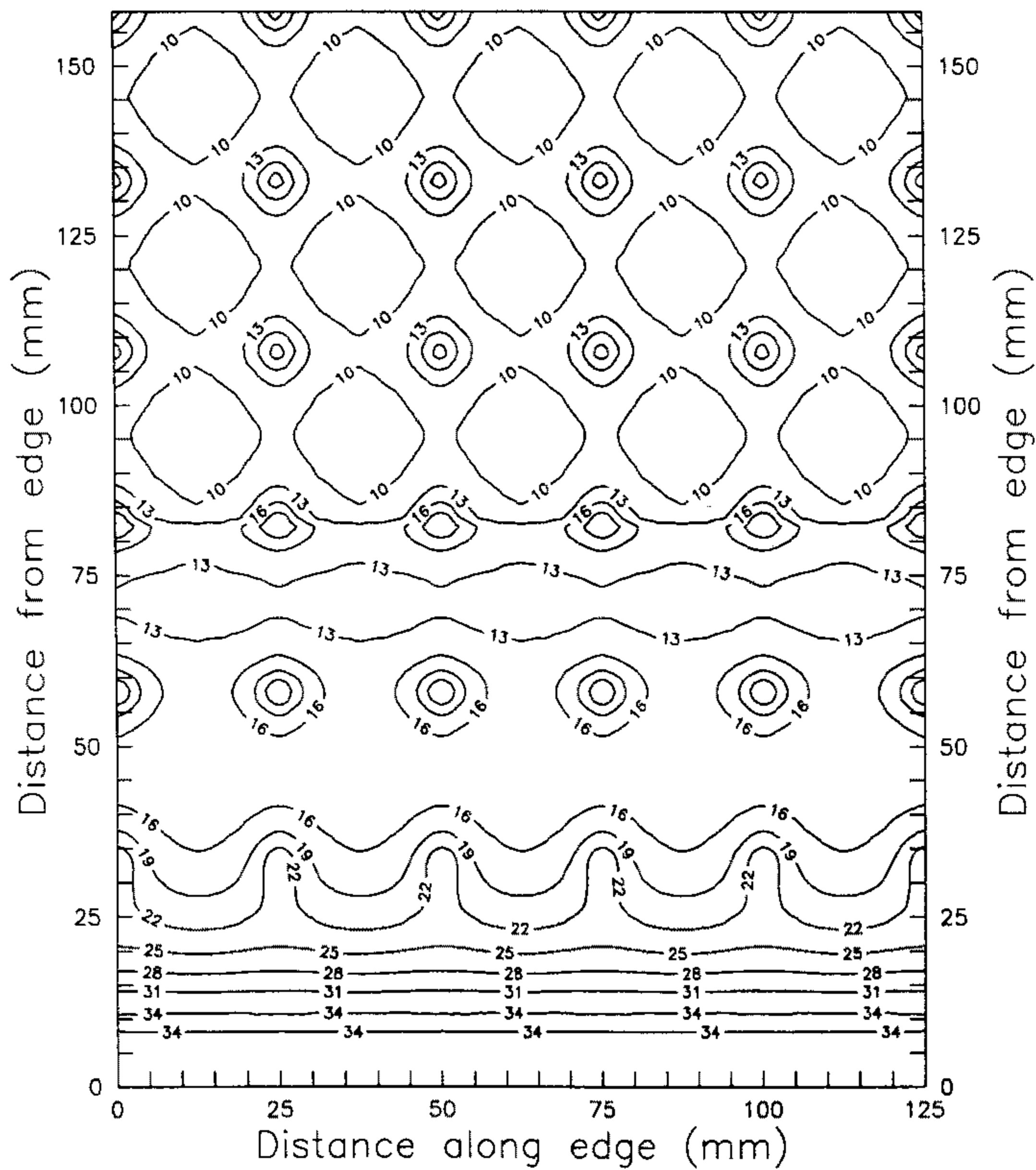


Fig. 8.7: The equivalent maximum principal tensile stress scaled by the biaxial stress correction factor, $c\sigma'_{II}$, for a section of the warm side of a vacuum glazing for the dimensions and conditions given for Figs. 8.1 and 8.2. These results are based on the stresses of Figs. 8.5 and 8.6, $K_{Ic}=0.78 \text{ MPa m}^{1/2}$, $K_{Isc}=0.30 \text{ MPa m}^{1/2}$, and on biaxial stress correction factors determined from Table 8.1. The non-uniformity in the contours at about 83 mm is explained near the beginning of this section. The stress contour interval is 3 MPa.

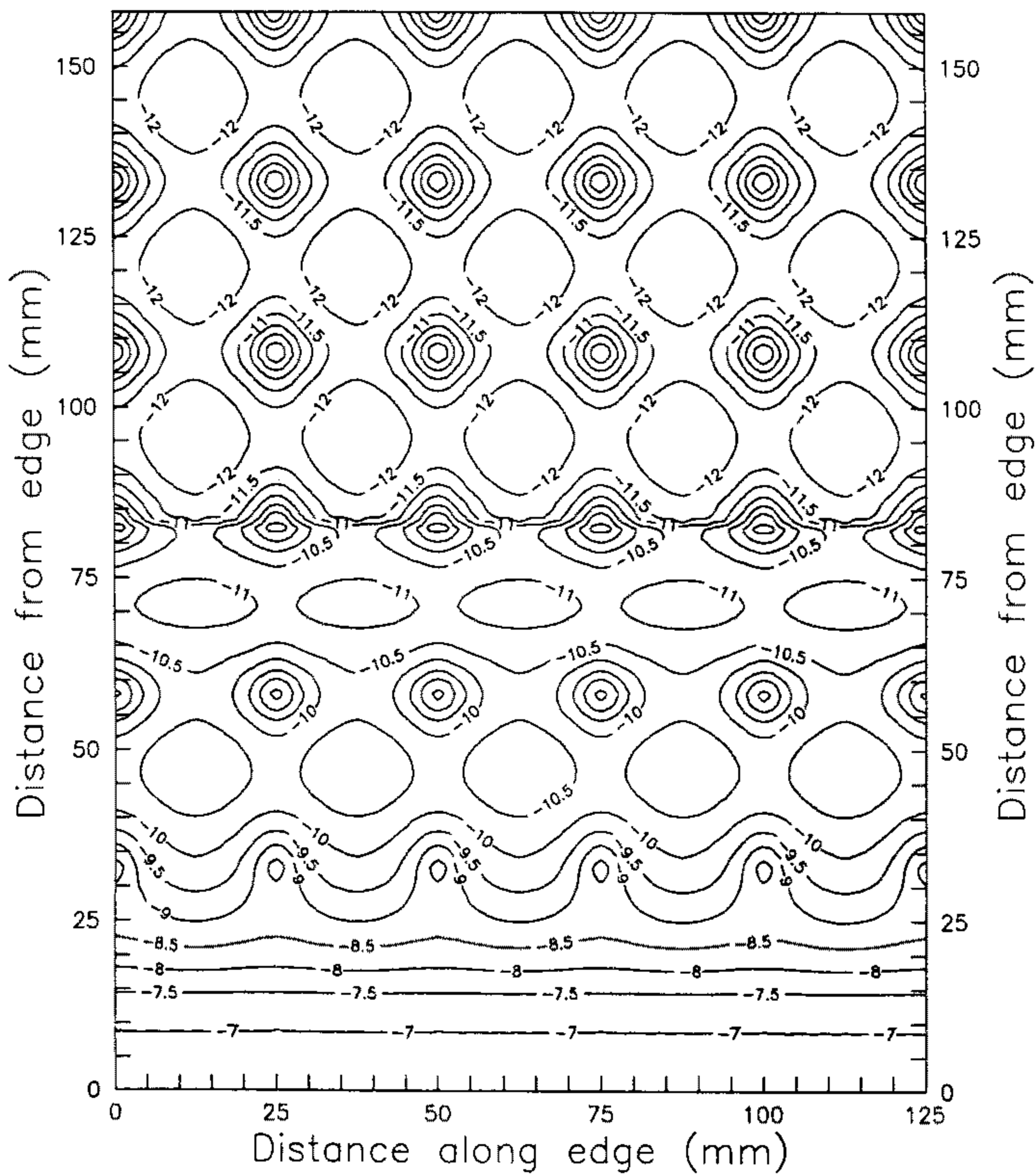


Fig. 8.8: The log of the elemental probability of failure for a section of the warm side of a vacuum glazing for the dimensions and conditions given for Figs. 8.1 and 8.2. These results are based on the results of Fig. 8.7, $m=9$, and $k=1.32 \times 10^{-69} \text{ m}^2\text{Pa}^{-6}$. The size of the elements is 1 mm by 1 mm and thus the results are per mm^2 . The non-uniformity in the contours at about 83 mm is explained near the beginning of this section. The contour interval is 0.5 mm^{-2} .

equal to unity minus the elemental probability of survival). Since the elemental size is 1 mm by 1 mm, this plot represents the results for $\log(P_f \text{ per mm}^2)$. As expected from the results of Fig. 8.7, and from equations 8.2 and 8.3, the probability of failure is greatest where the biaxially corrected equivalent maximum principal tensile stresses are the largest: near the edge of the glazing and, to a lesser extent, over the pillars. The corresponding total probability of failure is 0.0040 for the entire 1 m by 1 m glazing. This means that 4 out of 1000 vacuum glazings of this size are predicted to fail catastrophically over a long period of time (and thus to undergo a complete loss of the internal vacuum). It is noted that the glazing considered above has a total external surface area of 2 m² - half of which does not contribute to the total probability of glazing failure. For the sake of convenience, the total probability of failure is expressed per area of glazing size, i.e. $P_{f,total} = 0.0040 \text{ m}^{-2}$. All total probabilities given in this study correspond to a glazing of size 1 m by 1 m.

Figures 8.9 present plots of the results from Fig. 8.7 for over a row of pillars, and midway between a row of pillars (e.g. along transits at 100 mm, and at 112.5 mm along the edge in Fig. 8.7). The corresponding results for $\log(P_f \text{ per mm}^2)$ from Fig. 8.8 are presented in Figs. 8.10. The cumulative probability of failure as a function of the distance from the edge of the glazing is shown in Fig. 8.11. This figure is based on the combined probability of all of the elements at each incremental distance from the edge. The total probability of failure for all of the elements across the glazing is indicated by the horizontal dashed line.

In Figs. 8.3 to 8.11, the first and second rows of pillars from the edge are located at 33 and 58 mm. Figure 8.11 shows that most of the contribution to the total probability of failure of 0.0040 m⁻² occurs within the first row of pillars (i.e. between 0 and 33 mm from the edge), and that about one half of this contribution comes from the region over the edge seal (i.e. between 0 and 8 mm from the edge). This figure also shows that almost all of the small remaining contribution to the total probability of failure occurs between the first and second rows of pillars (i.e. between 33 and 58 mm from the edge). Clearly, the stresses beyond the second row of pillars (i.e. beyond 58 mm) are not large enough to lead to significant elemental probabilities of failure. Therefore, for $\lambda = 25 \text{ mm}$, it is the stresses close to the edge (mostly those within the location of the first row of pillars) that are the most significant in the context of the glazing failure. It should therefore be possible to reduce the probability of failure by lowering the stresses within this region by using edge insulation. This approach is investigated later in this section.

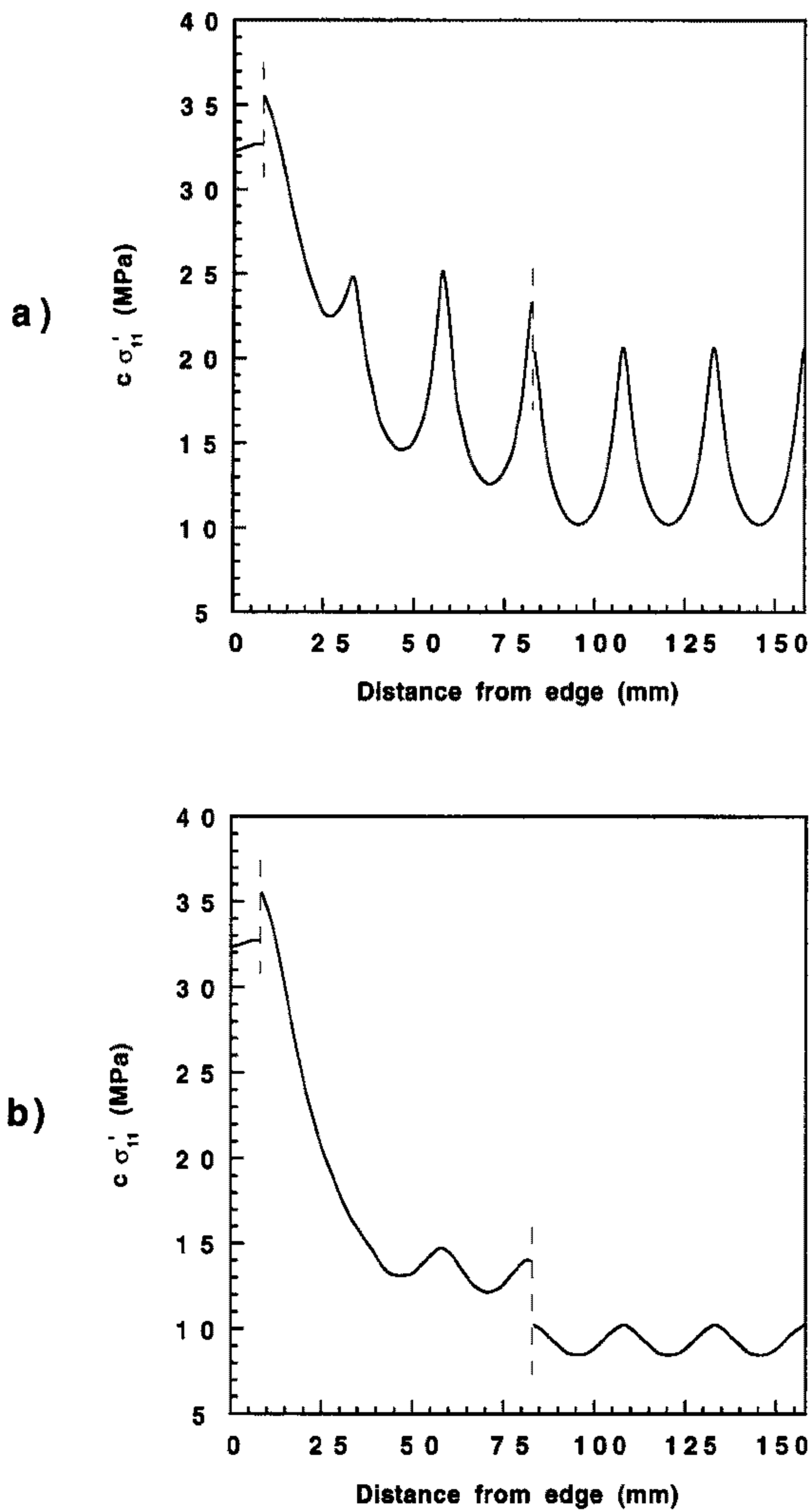


Fig. 8.9: Results for the biaxially corrected equivalent maximum principal tensile stress, $c\sigma_{11}$, from Fig. 8.7 for (a) over a row of pillars, and (b) mid-way between a row of pillars (e.g. along transits at 100 mm and 112.5 mm along the edge in Fig. 8.7). The discontinuities at 8 and 83 mm (explained above) are indicated as dashed lines.

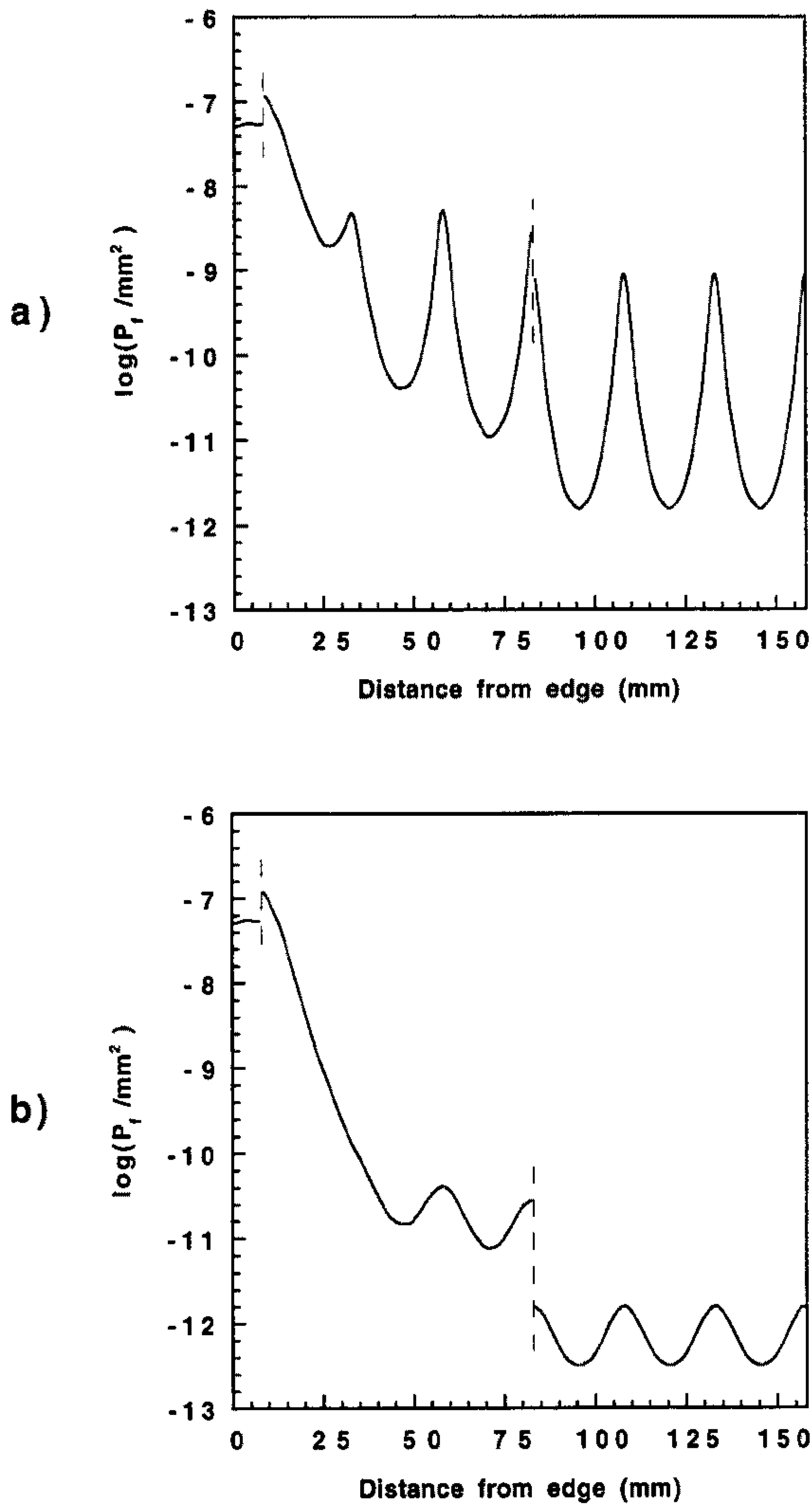


Fig. 8.10: Results for the log of the elemental probability of failure from Fig. 8.8 for (a) over a row of pillars, and (b) mid-way between a row of pillars (e.g. along transits at 100 mm and 112.5 mm along the edge in Fig. 8.8). The discontinuities at 8 and 83 mm (explained above) are indicated as dashed lines.

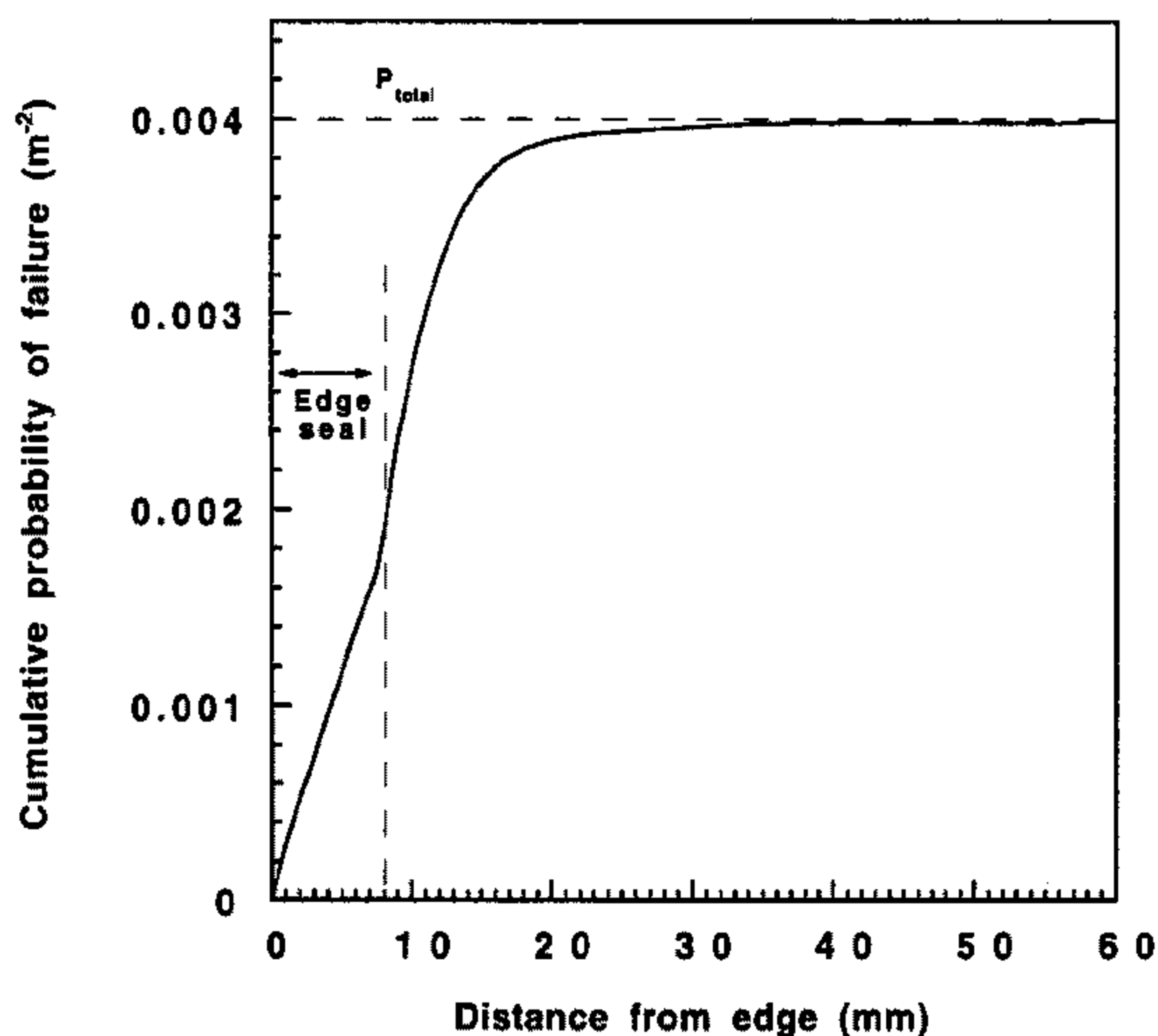


Fig. 8.11: The cumulative probability of failure as a function of the distance from the edge of the glazing, based on the results of Fig. 8.8. The total probability of failure for the entire glazing (0.0040 m^2) is indicated by the horizontal dashed line, and the location of the inner edge of the seal is indicated by the vertical dashed line.

It is important to note the sensitivity of the total probability of glazing failure to the assumptions made in Section 8.3 concerning the uniformities of the stresses. Firstly, the solution procedure is based on the assumptions that the temperature and pressure induced direct stresses along the edges are, respectively, uniform and uniformly periodic. It is recalled that these assumptions were made so that it would not be necessary to consider the corresponding non-uniform stresses within about 150 and 50 mm from the glazing corners. Since the direct stresses at the corners are essentially zero, the direct stresses within the respective corner regions actually range from zero to about the assumed values. Thus, in order to simulate the extreme case with the least possible stress, the corner region direct stresses (as well as the corresponding shear stresses) can be set equal to zero. This is done for both the temperature and pressure induced stresses over the extent of the most non-uniform region of the former stresses: 50 mm from the corners. When this is done, the total probability of failure changes from 0.0040 m^2 to 0.0037 m^2 . Therefore, the assumptions concerning the uniformities in the direct stresses along the edges result in a *maximum* overestimation of the total probability of glazing failure by about 7.5 %. In actuality, the corner region stresses are mostly non-zero and thus, for this analysis, they could be better simulated as being

equal to stresses somewhere between zero and the corresponding assumed values. Thus, the actual percentage error can be expected to be somewhat less than the above maximum estimate.

Secondly, as explained above, the temperature and pressure induced shear stresses - and thus the total shear stresses - for each external glass surface are assumed to be uniform and equal to the average shear stresses across the surface (the total values are -0.5 and 0.5 MPa on the warm and cold sides respectively). In actuality, the temperature induced shear stress varies from 0 to -0.7 MPa over most of the warm side external surface of the glazing, and from 0 to 0.7 MPa over most of the cold side external surface (if the large non-uniformities near the glazing corners are not considered). For each side, the pressure induced shear stress varies from -0.5 to 0.5 MPa over most of the glazing (again ignoring the large non-uniformities near the corners). Thus the magnitude of the total shear stress varies from 0 to 1.2 MPa on each side of the glazing (only the variation in the magnitude need be considered since the shear stress is squared in equation 8.1). For the conditions considered in this study, only the variation in the warm side total shear stress can affect the calculation of the total probability of failure. With assumed uniform warm side shear stresses of 0 and -1.2 MPa, the total probability of failure is calculated to be 0.0039 m² and 0.0042m², respectively. Therefore, the assumption of uniform average shear stresses leads to an error of about 2.5 to -5.0 % in the total probability of failure.

Thirdly, the pressure induced direct and shear stresses at the external glass surfaces over the edge seal are assumed to be uniformly zero. As noted in Section 7.3.1, this is based on the expectation that the pressure induced stresses at each external glass surface above the inner edge actually decrease exponentially to near zero over a very short distance along the width of the seal. In the worst case, it is expected that significant pressure induced direct stresses over the seal might extend to about 1 mm from the inner edge of the seal. This situation can be simulated by setting these stresses at this extent to be equal to 1/e of the corresponding values above the inner edge of the seal. The pressure induced shear stress can be assumed to be small enough to be kept at zero for this case. If this is done, the total probability of failure changes from 0.0040 m² to 0.0041 m². Therefore, the assumption of uniformly zero pressure induced stresses over the edge seal results in an underestimation of the total probability of failure by about 2.5 %.

Lastly, it is assumed that the temperature induced stresses are uniform beyond $3\lambda + w_{seal}$ (83 mm) from the edge of the glazing. As previously noted, this is not quite

correct. However, since this leads to only a small error in the total direct stresses, and since the stresses beyond 83 mm from the edge are well below the levels which lead to significant contributions to the total probability of failure, the error in this assumption is of no consequence to the calculation of $P_{f,total}$.

The above error discussion is based on results for a pillar separation of 25 mm. For larger pillar separations, the shear stresses are even smaller in magnitude than the direct stresses and thus the error associated with the assumption of uniform shear stresses decreases with λ . The absolute value of the error in the assumption of uniform pressure induced stresses increases with the pillar separation. However, this error represents a smaller fraction of $P_{f,total}$ which also increases with λ - but at a larger rate. Thus, the percentage error in $P_{f,total}$ due to this assumption decreases with the pillar separation. As will be discussed below, the error due to the assumption of uniform temperature induced stresses beyond $3\lambda + w_{seal}$ increases with λ . The overall conclusion from this discussion of the errors is that the solution procedure leads to a net error in the total probability of failure of a few percent, and that it should be possible to determine overall trends in the variation of $P_{f,total}$ with λ .

The above solution procedure was also performed for pillar separations of 20, 30, 35 and 40 mm. The temperature and pressure induced stresses were determined from the appropriately modified finite element models used to obtain the stresses for $\lambda=25$ mm. In determining the temperature induced stresses, the glass-to-glass center-of-glass U-value was adjusted according to the values given in Table 8.4. The assumed uniform center-of-glass stresses were set equal to the corresponding modelling results. The pressure induced stresses were based on the beam elastic properties given above. Results from the calculations for different pillar separations are presented below.

The results over a row of pillars for $c\sigma_{11}$, and for $\log(P_f \text{ per mm}^2)$ are presented in Figs. 8.12 and 8.13 together with the comparable results from Figs. 8.9 and 8.10 for $\lambda=25$ mm. Plots of the cumulative probability of failure for all of these pillar separations are given in Figs. 8.14. Figure 8.15 shows the total probability of failure as a function of the pillar separation. The results of Figs. 8.12 to 8.15 can be interpreted as follows. As the pillar separation is increased, both the pressure and temperature induced direct stresses become larger. This leads to larger elemental values for the equivalent maximum principal tensile stress, σ'_{11} , and thus, as indicated by equation 8.3, to larger elemental risk of failure functions, β . Consequently, the probability of failure increases with λ . It is noted that since there are fewer unit cells in the glazing for larger separations, an increase in the pillar separation slightly lowers the various

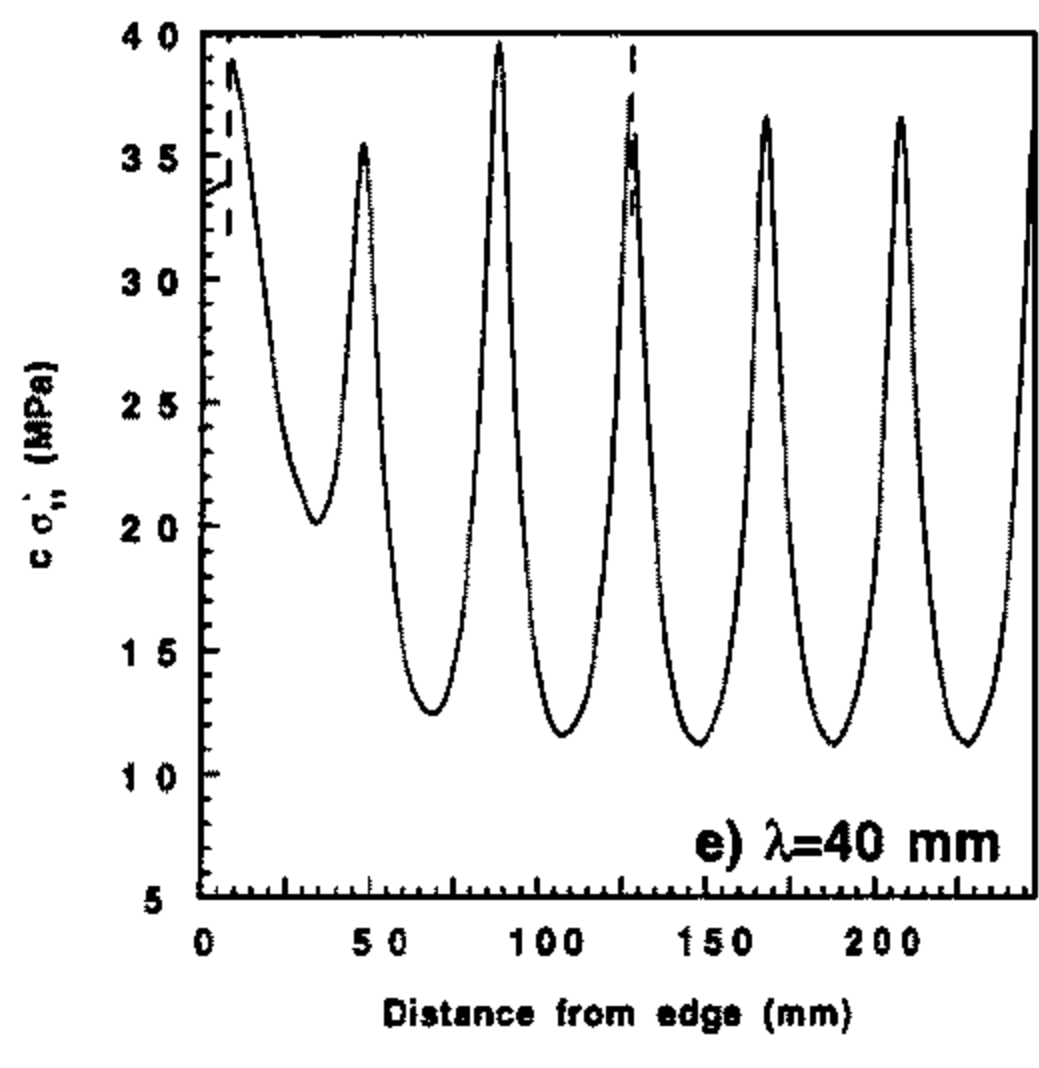
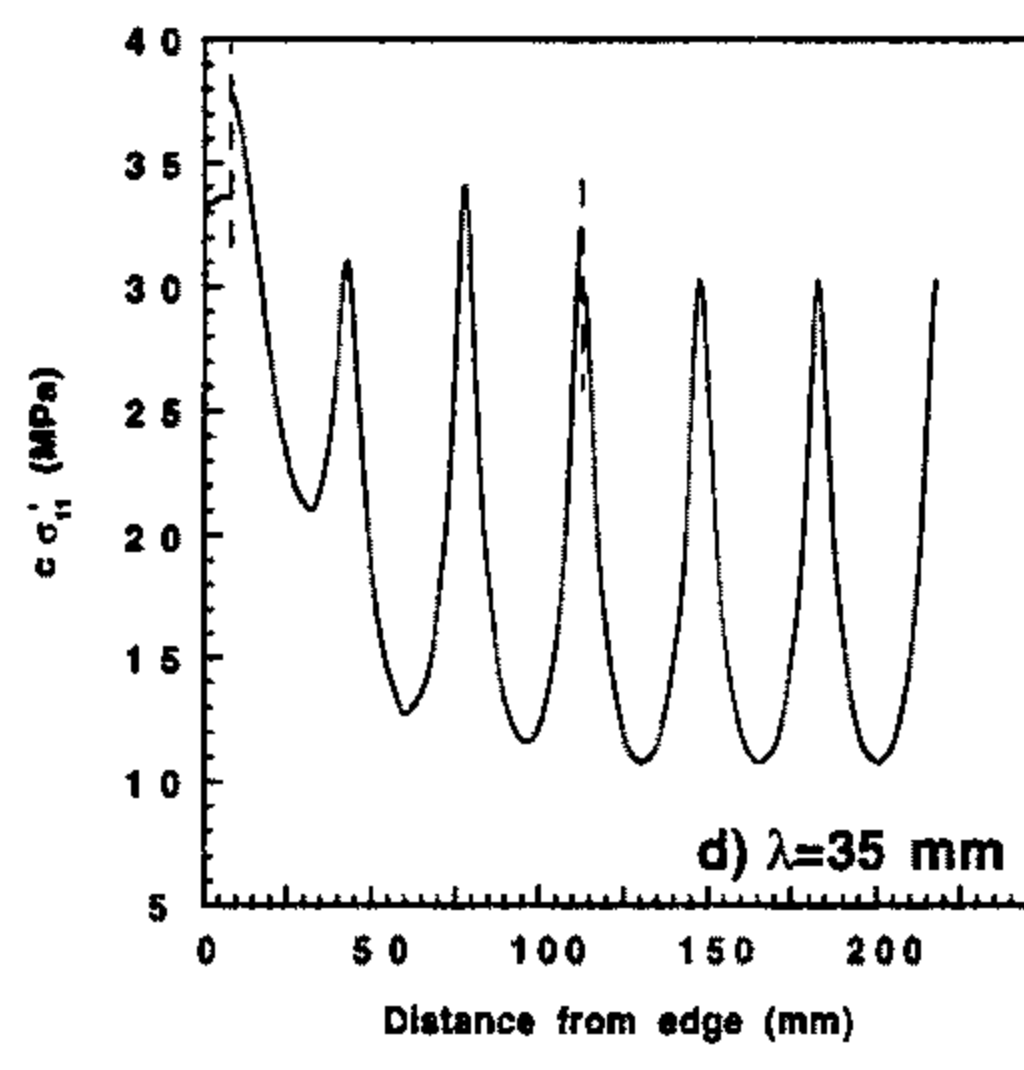
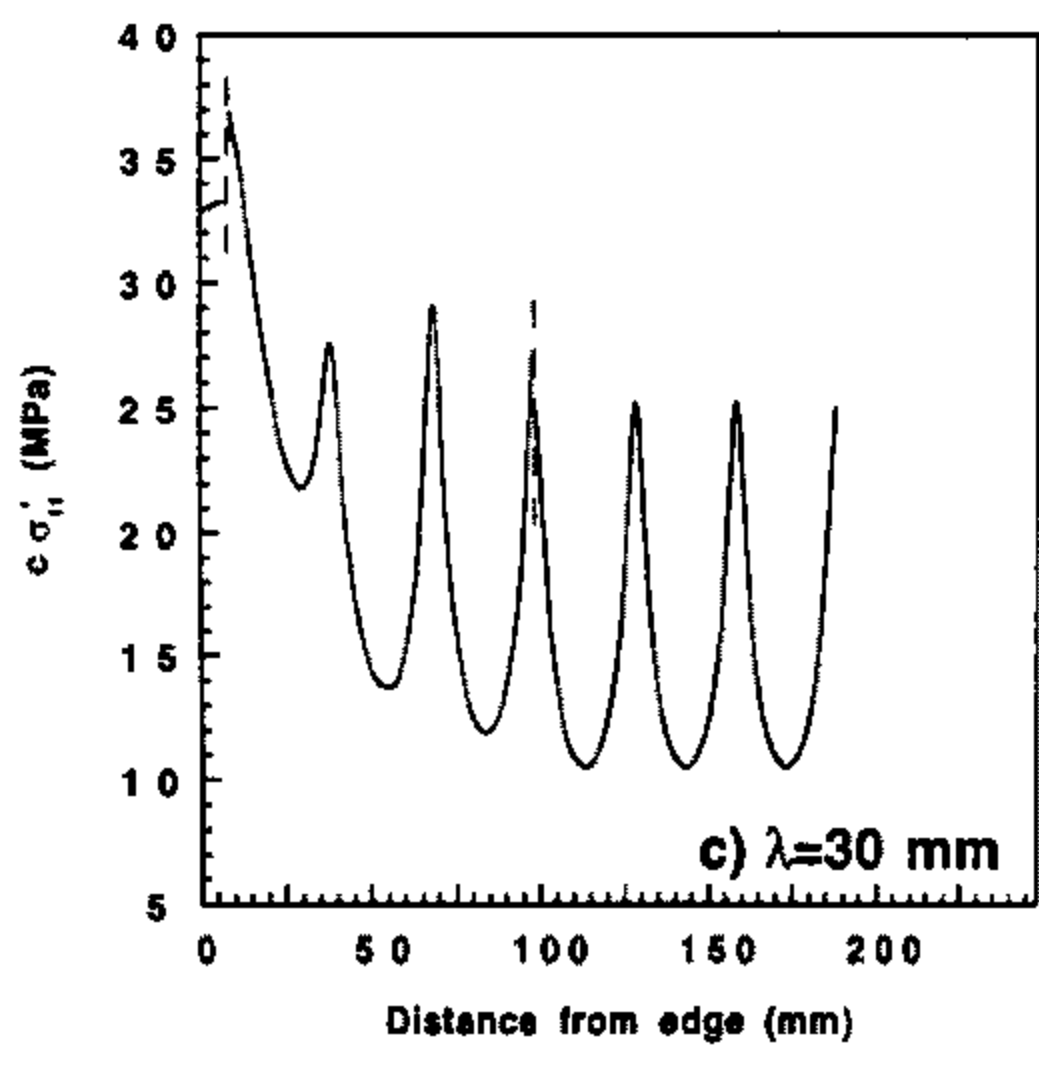
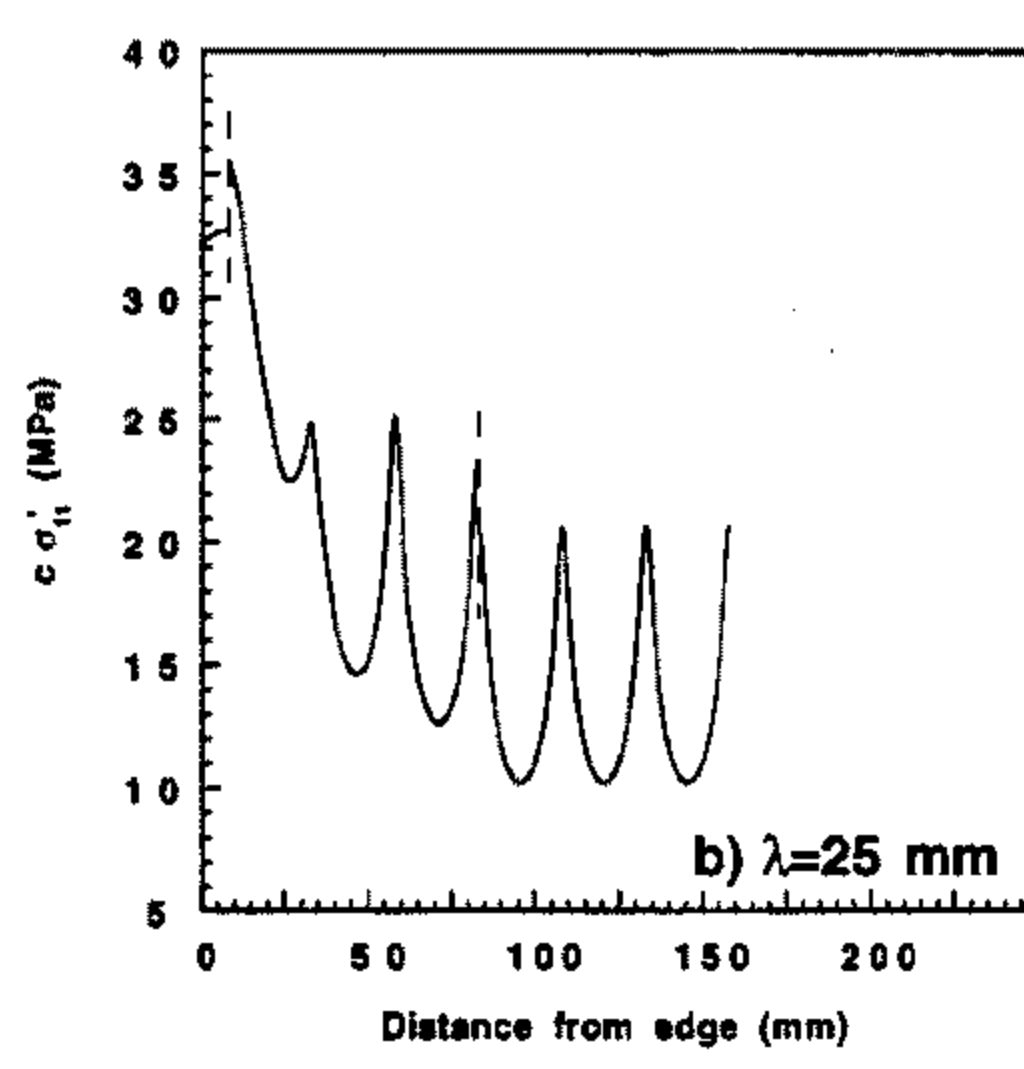
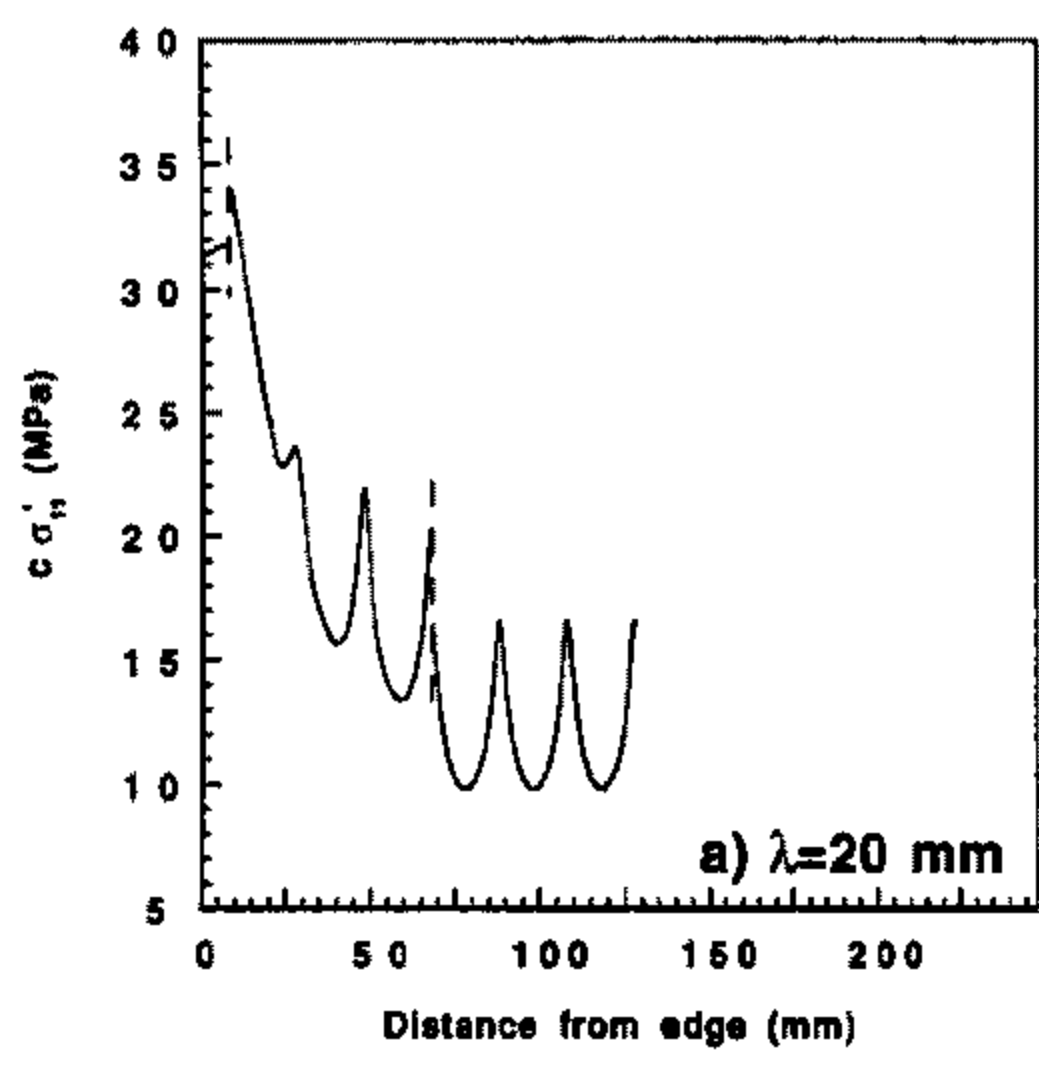


Fig. 8.12: Results for the biaxially corrected equivalent maximum principal tensile stress, $c\sigma_{11}$, over a row of pillars for different pillar separations. Aside from the pillar separation and the corresponding values for U_{g-g} as given in Table 8.4, the results are for the same dimensions and conditions given for Figs. 8.1 and 8.2.

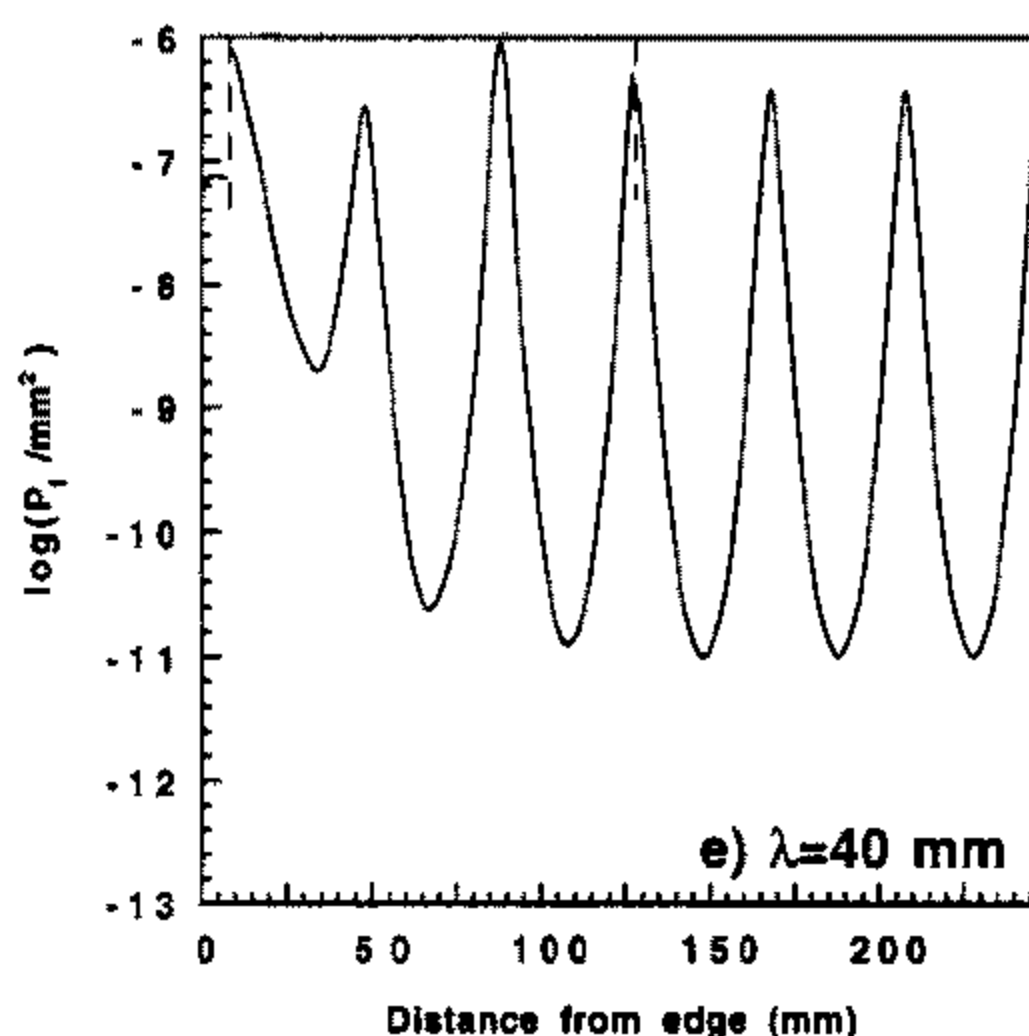
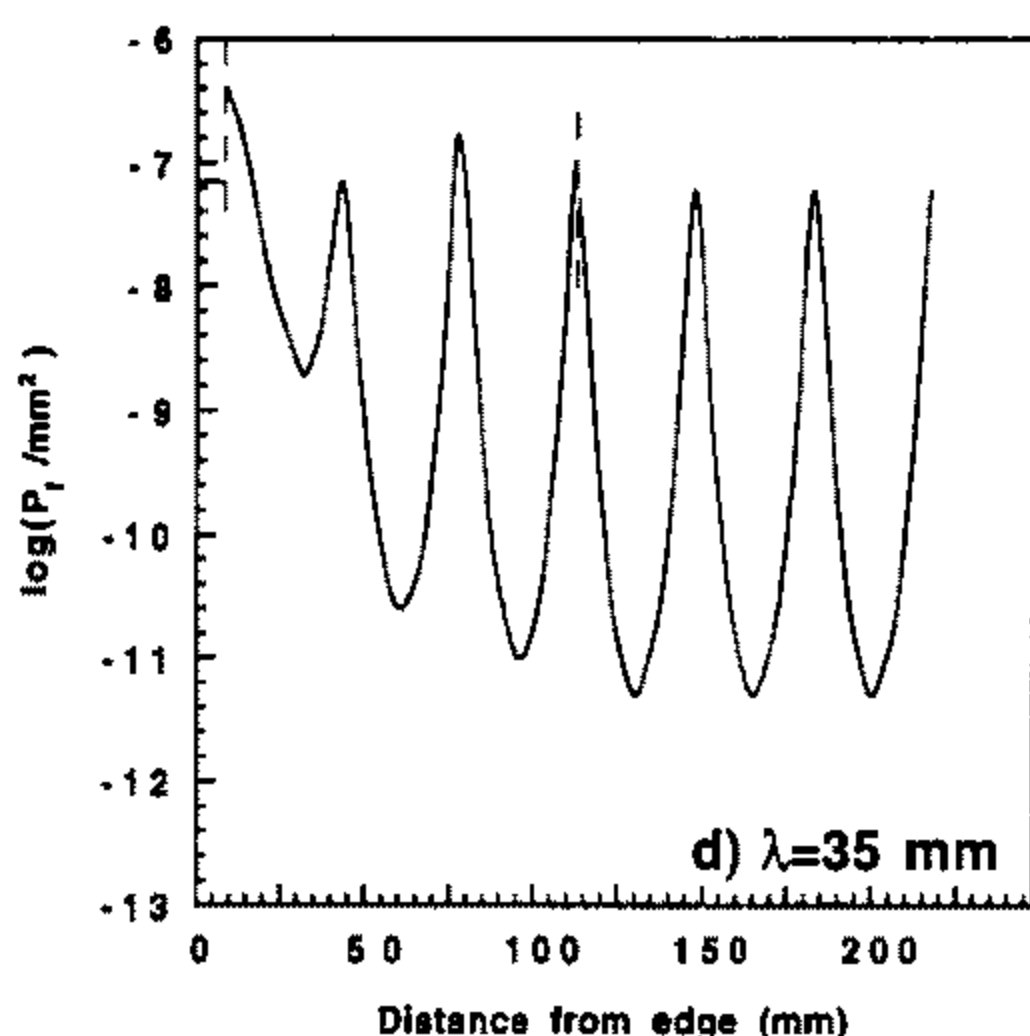
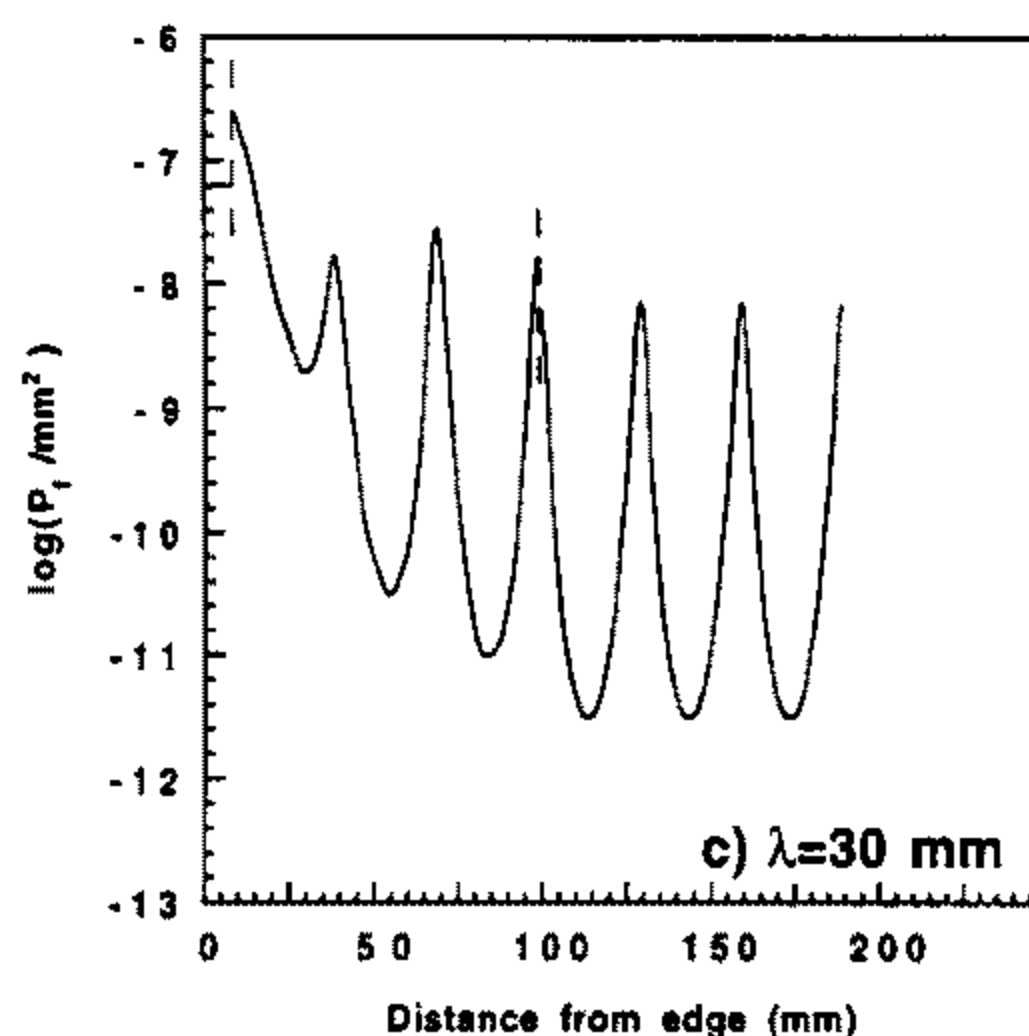
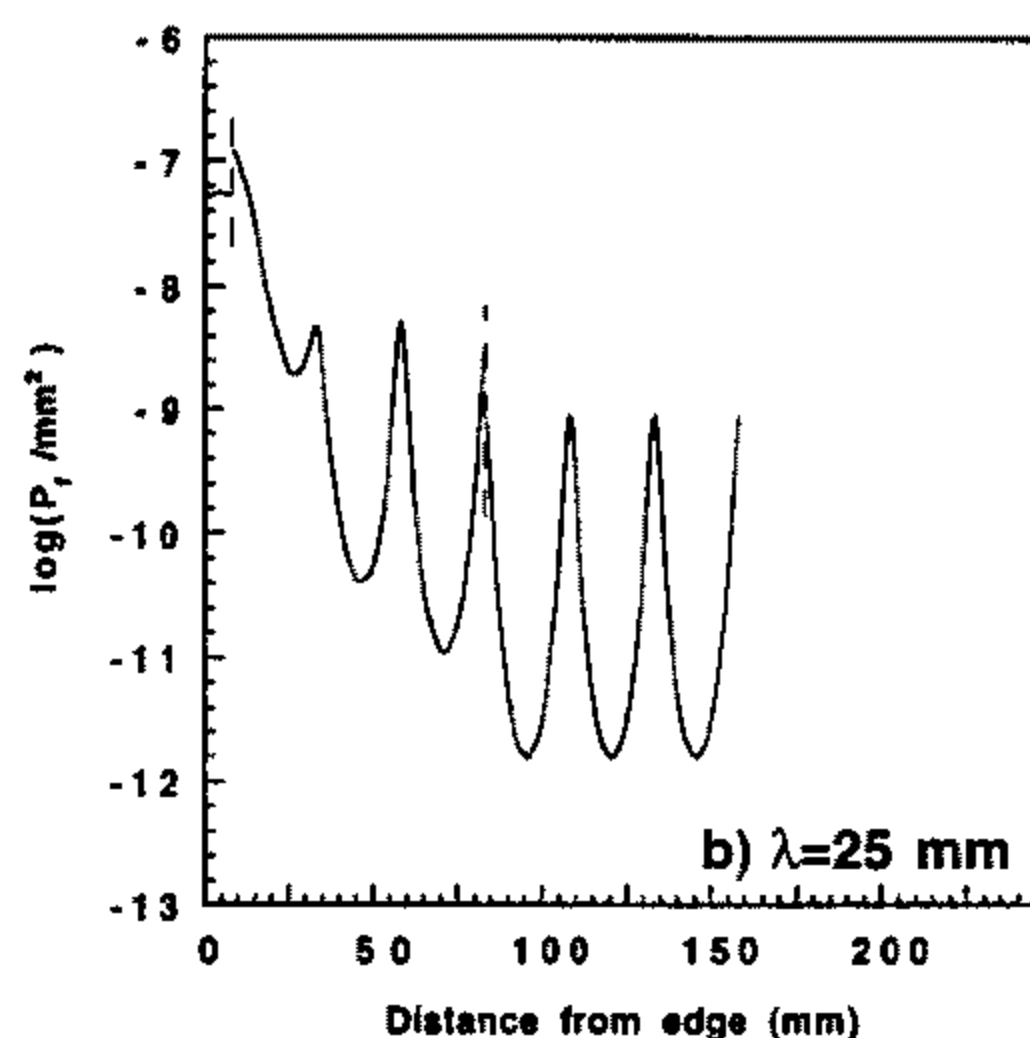
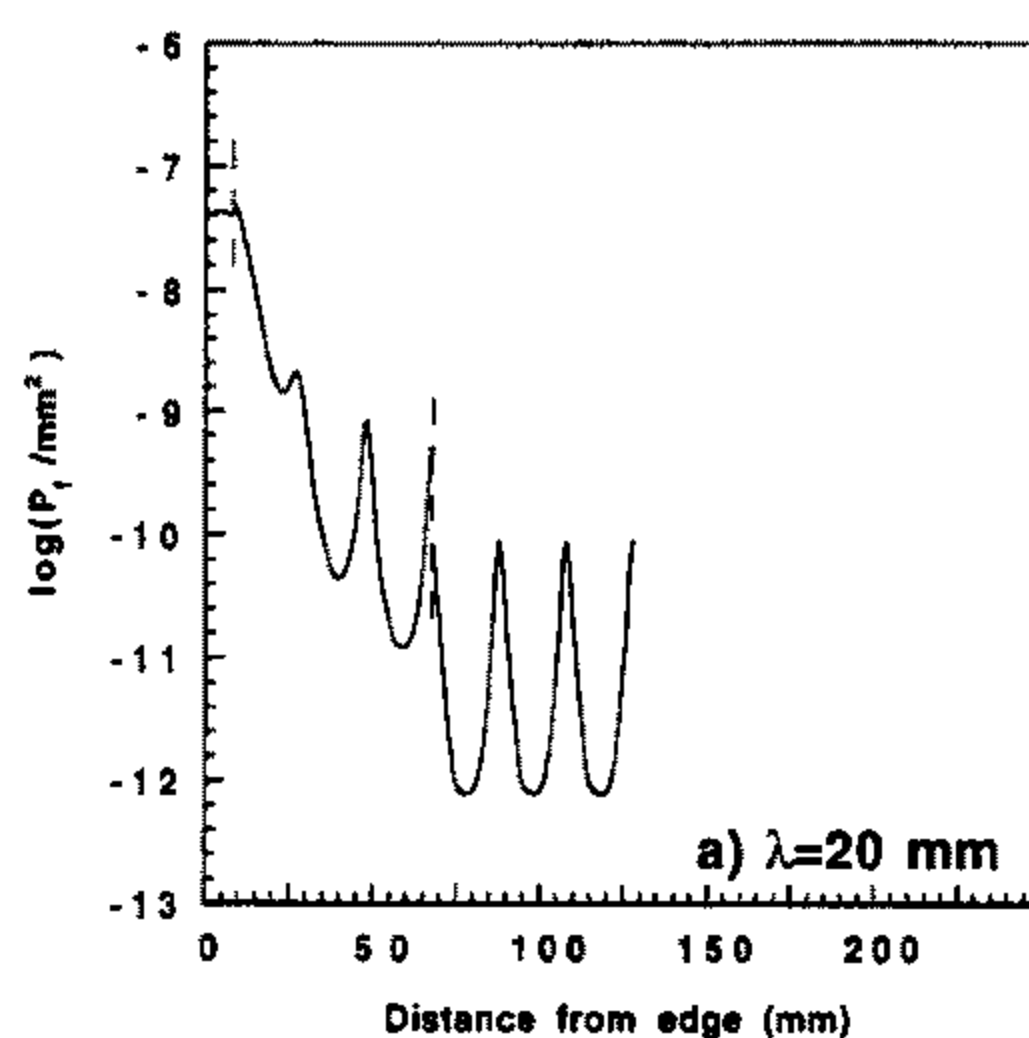


Fig. 8.13: Results for the log of the elemental probability of failure over a row of pillars for different pillar separations. Aside from the pillar separation and the corresponding values for U_{g-g} as given in Table 8.4, the results are for the same dimensions and conditions given for Figs. 8.1 and 8.2.

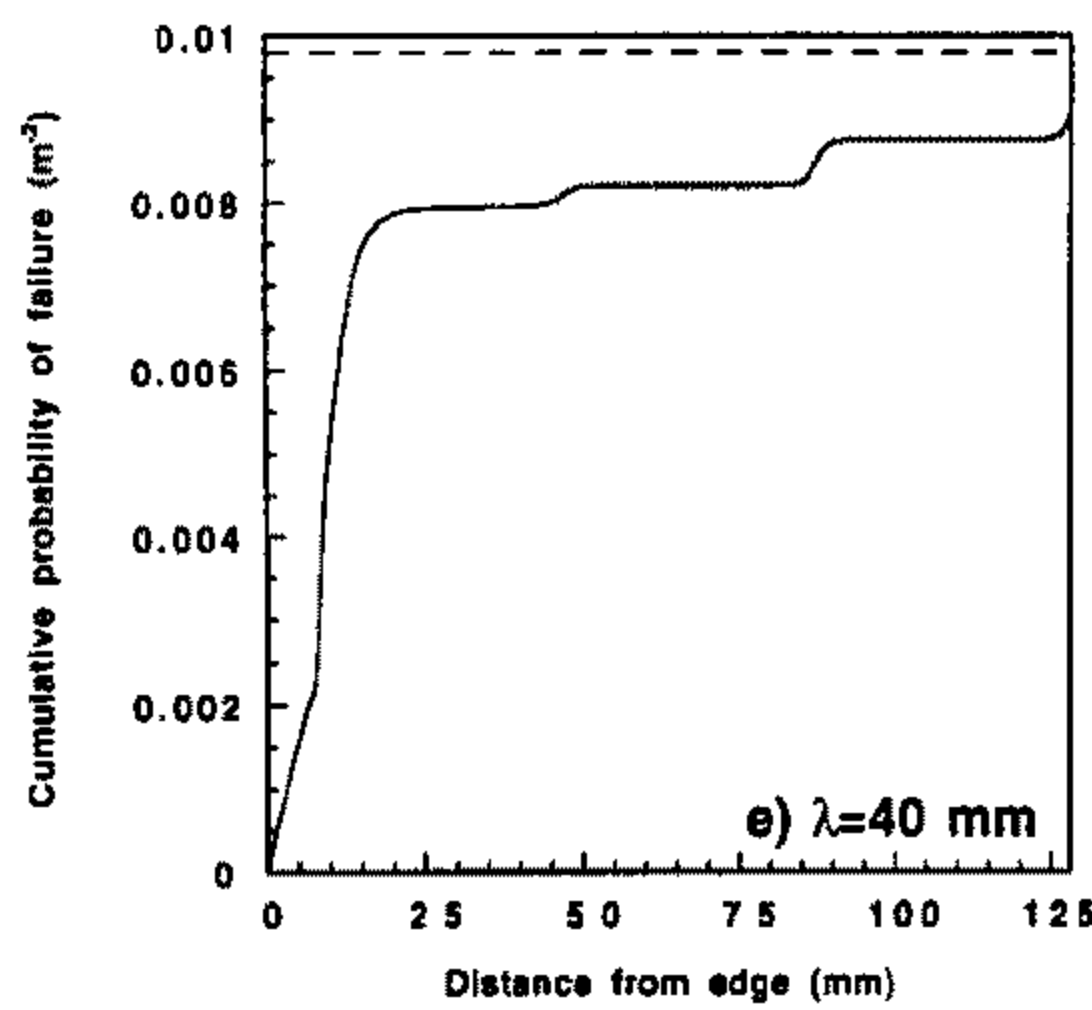
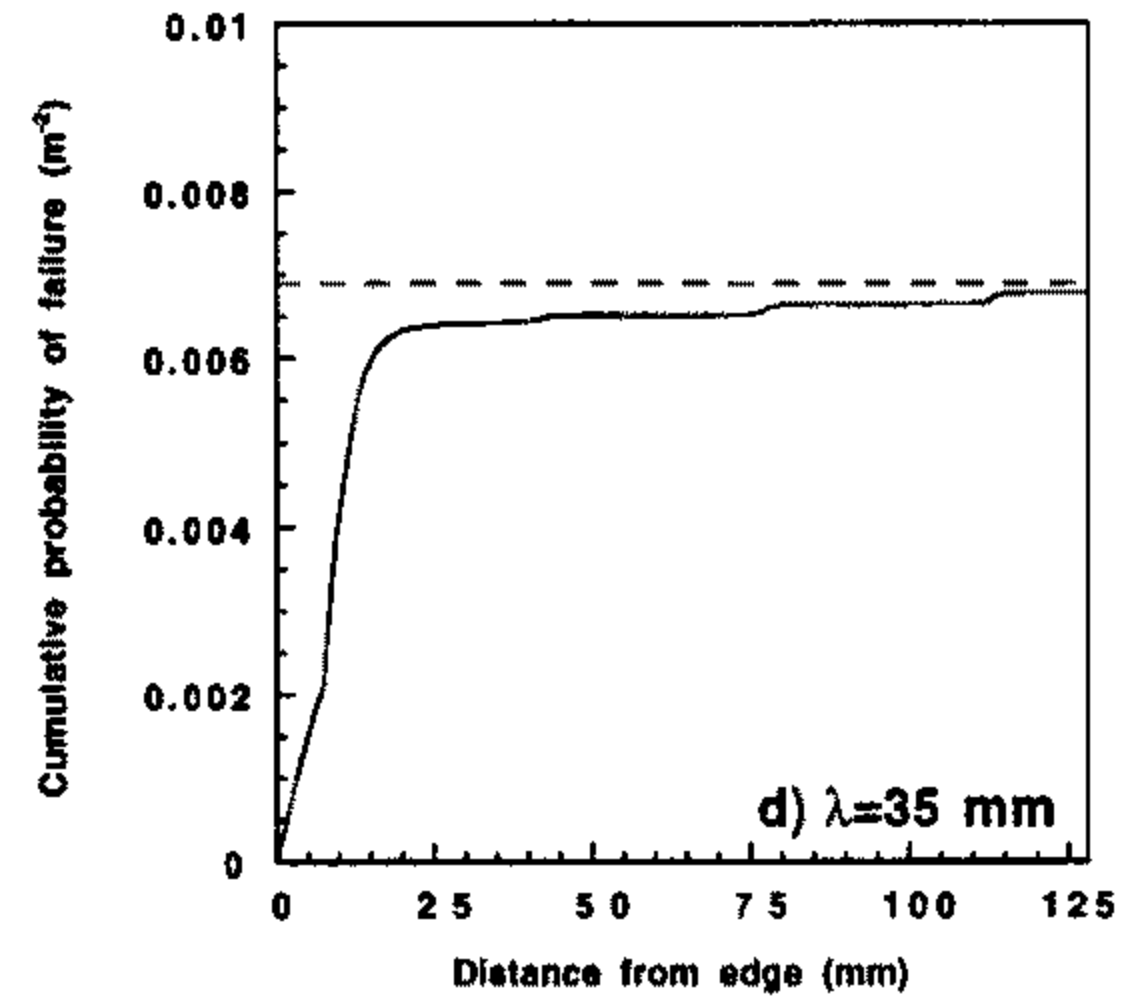
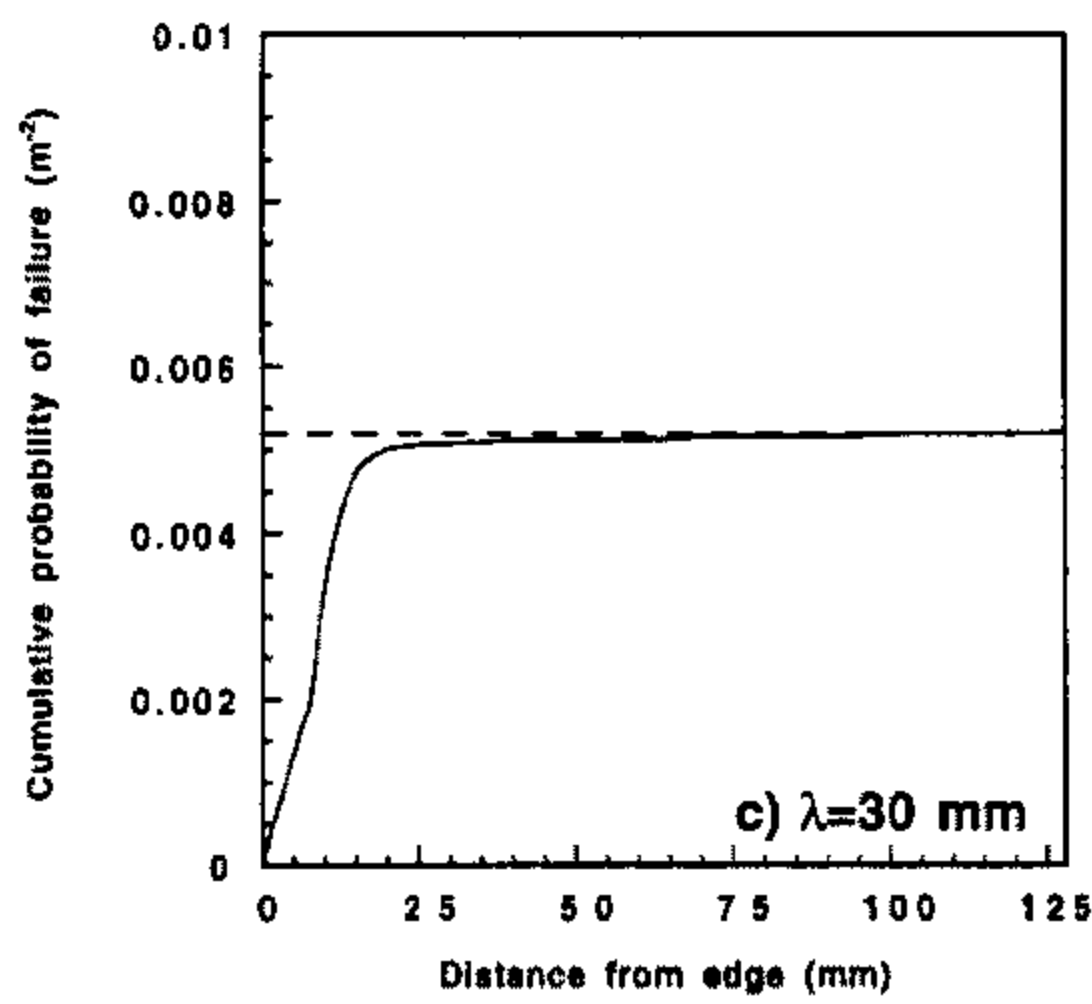
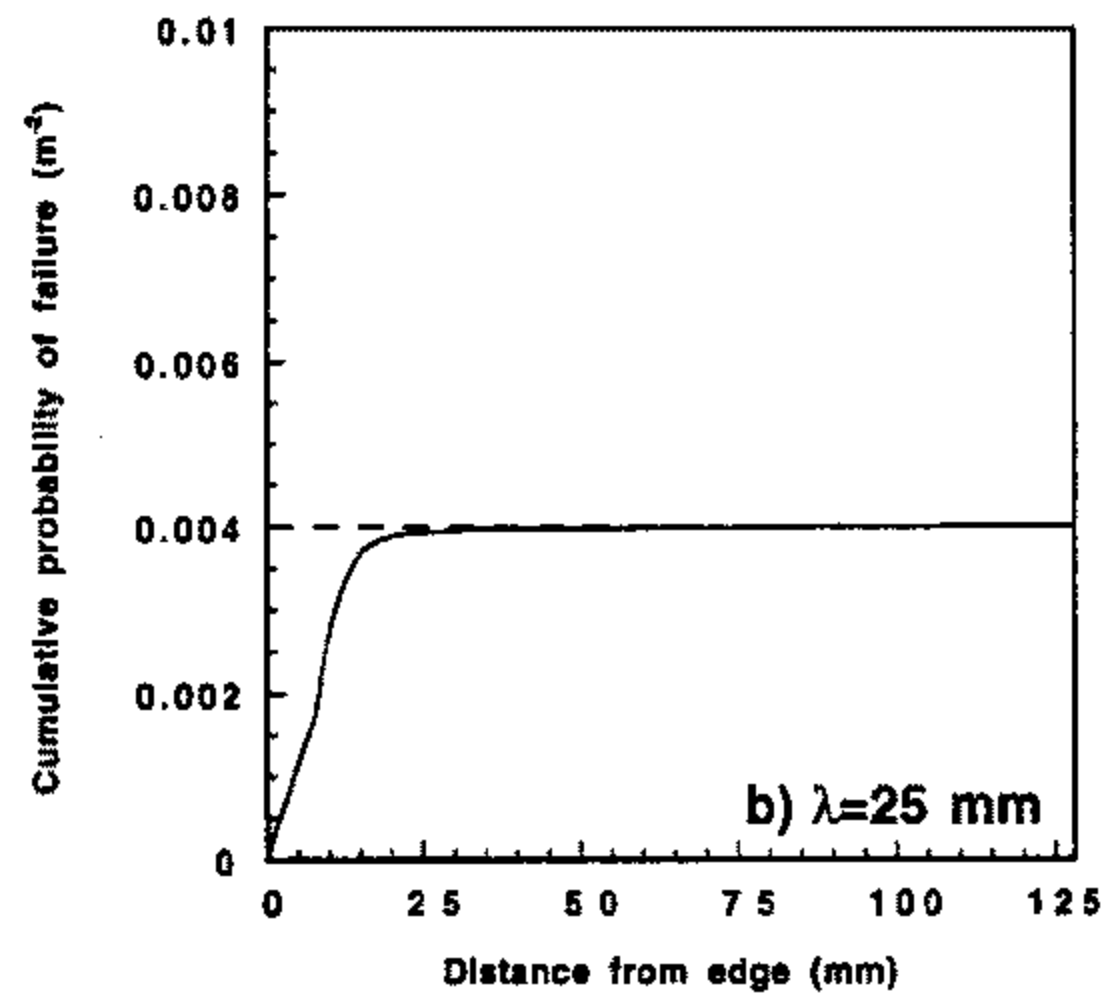
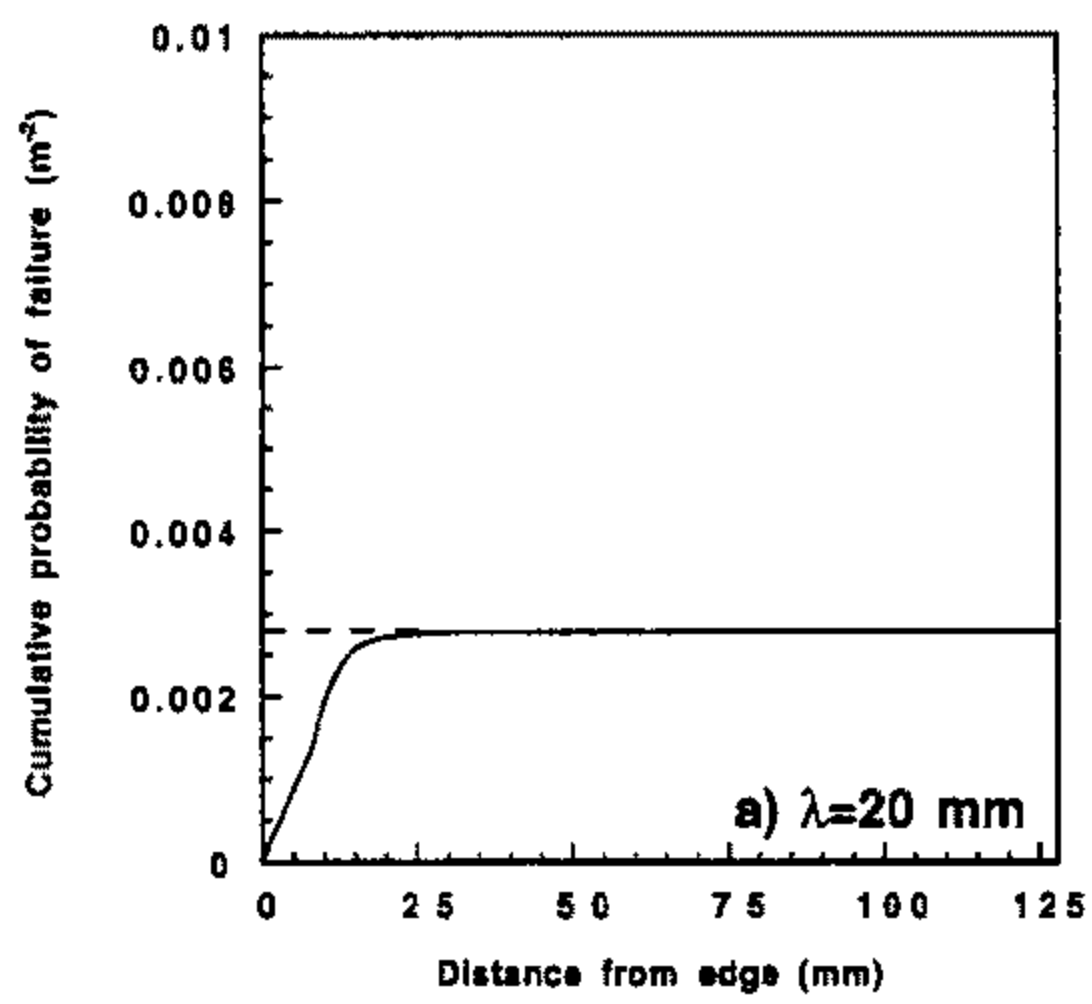


Fig. 8.14: Results for the cumulative probability of failure for different pillar separations. The total probabilities of failure for the entire glazing (including the region beyond that which is considered in these plots) are indicated by the horizontal dashed lines. Aside from the pillar separation and the corresponding values for U_{g-g} as given in Table 8.4, the results are for the same dimensions and conditions given for Figs. 8.1 and 8.2.

values for N_{cell} . This reduces the total elemental area contribution to the elemental values for β . However, the effect of the pillar separation on β is dominated by the much larger increase in σ'_{11} (this increase is magnified by the fact that σ'_{11} is raised to the power of $m=9$).

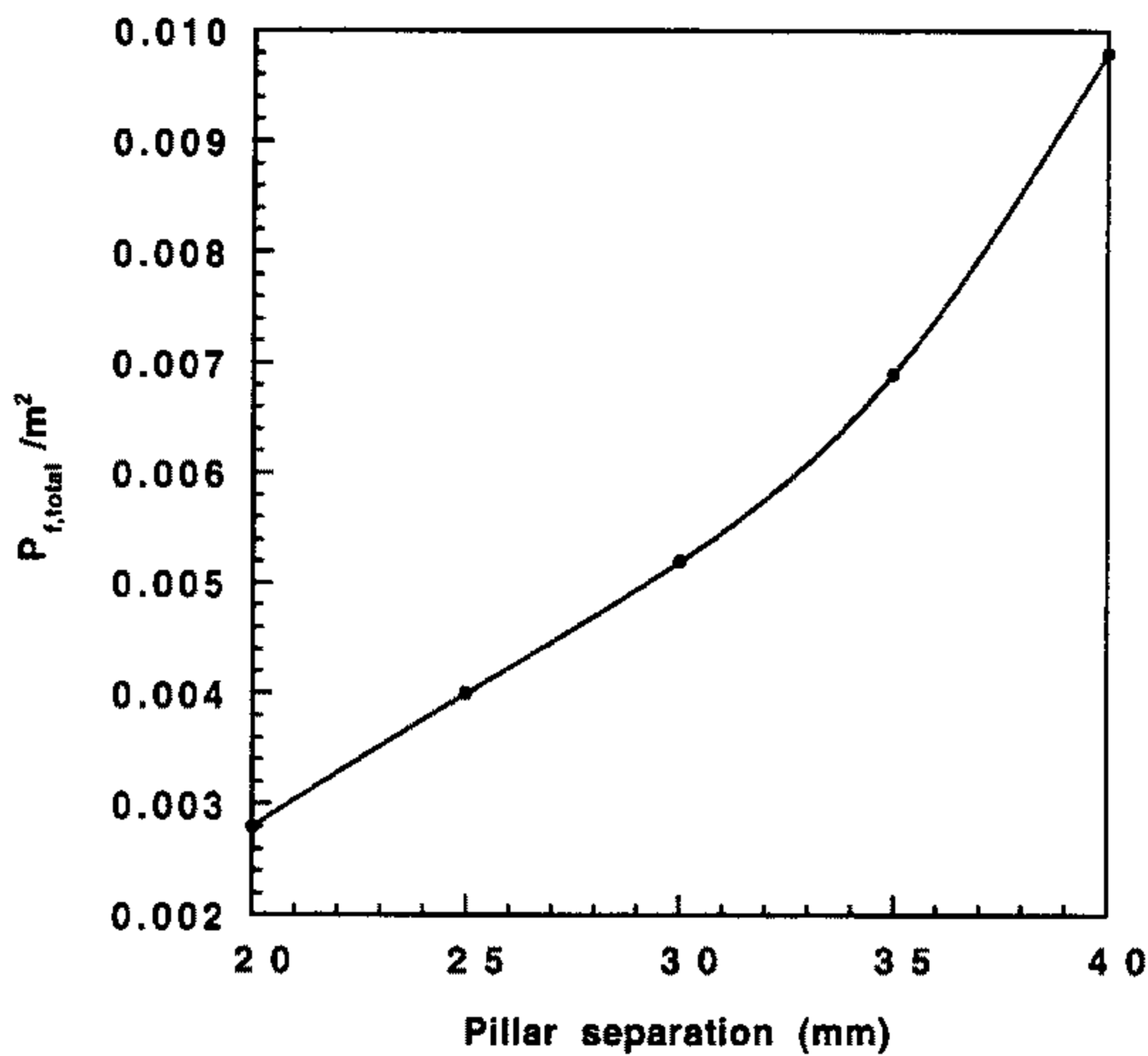


Fig. 8.15: The total probability of glazing failure as a function of the pillar separation. Aside from the pillar separation and the corresponding values for U_{g-g} as given in Table 8.4, the results are for the same dimensions and conditions given for Figs. 8.1 and 8.2.

The increase in the pressure induced direct stresses with λ is clearly shown by Figs. 7.11 and 7.12. It is important to note that significant increases in these stresses occur in the region which makes the largest contribution to the total probability of failure: near the edge seal. The stresses over the pillars also increase with λ , thereby increasing the local elemental probabilities of failure. The effect of this on the total probability of failure can be seen in Figs. 8.14; with increasing pillar separation, the stresses over the pillars become large enough to lead to significant contributions to the total probability of failure. This effect is just barely discernible in Fig. 8.14c for $\lambda=30$ mm, and it can be clearly seen for the two larger pillar separations of 35 and 40 mm. In fact, for the latter two separations, the stresses over the pillars in the region beyond $3\lambda + w_{seal}$ from the edge contribute significantly to P_f . It is because results over these pillars are not shown in Figs. 8.14d and e that the plots of these figures are below the total probabilities (indicated as dashed lines). Since the pillars of this region do

make a significant contribution to the total probability of failure, the assumption of uniform temperature induced stresses beyond $3\lambda + w_{seal}$ can lead to a slight underestimation of $P_{f,total}$. However, even if the small error due to this assumption were accounted for, the obvious non-linearity in the results of Fig. 8.15 for the larger pillar separations would still exist (and perhaps be more pronounced).

The increase in the temperature induced stresses with the pillar separation has a subtle origin. As indicated in Table 8.4, U_{g-g} decreases with λ . This results in a larger glass-to-glass temperature difference across the glazing. Since, as explained in Section 6.2, it is this temperature difference which causes the bending which gives rise to the temperature induced stresses, the overall stresses increase with λ . A decrease in U_{g-g} also results in a slight increase in the mean temperature of the glazing (due to the relative magnitudes of the heat transfer coefficients, the increase in the warm glass sheet temperature is greater than the decrease in the cold glass sheet temperature), and it also leads to a slight - but smaller - increase in the temperature of the glass at the edge of the glazing. These temperature changes are seen in the finite element modelling results, and they are consistent with a simple analytic solution to this situation. Thus, the difference between the mean temperature of the glazing, T_{mean} , and the temperature of the edge, T_{edge} , increases. As discussed in Section 6.2, the additional tensile stress at the edge (which, for the conditions considered in this study, occurs on the warm side of the glazing) increases with $T_{mean} - T_{edge}$. Therefore, as the pillar separation is increased, the temperature induced stresses increase throughout the glazing and, more importantly, within the region which has the greatest influence on the total probability of failure: over and near the edge seal. It is recalled that it is assumed that the only stress over the seal is the temperature induced stress. Although this stress over the seal increases by a small percentage over the range of pillar separations considered in this study (by about 7 % from $\lambda=20$ to 40 mm), the exponential dependence of the risk of failure function on the corresponding results for the equivalent maximum principal tensile stress (as per equation 8.3) leads to a significant increase in the total probability of failure. This effect can be seen in Figs. 8.14 by the larger cumulative probabilities of failure over the extent of the seal (from 0 to 8 mm) for the larger pillar separations.

Figure 8.15 shows that, for larger pillar separations, there is an obvious non-linear relationship between the total probability of failure and λ . This is explained as follows. As noted above, larger pillar separations lead to increases in the stresses near the edge and over the pillars - and thus to increases in the corresponding equivalent maximum principal tensile stresses at these locations. The exponential dependence of the risk of

failure function on these principal stresses ensures that there will be even larger - and non-linear - increases in the local elemental probabilities of failure, and thus in the total probability of failure, with λ . Since the stresses over the pillars only make significant contributions to $P_{f,total}$ at larger pillar separations, this non-linearity is most pronounced for larger values of λ . The power law dependence between the stresses over the pillars and λ (shown in Fig. 7.13) contributes to this non-linearity.

As mentioned above, it should be possible to reduce the total probability of failure with edge insulation. As noted in Section 6.6.5, and as shown in Figs. 6.28b and 6.29, the addition of edge insulation to both sides of the glazing lowers the additional tensile stress at the edge on the warm side of the glazing. This in turn results in a lower total probability of failure. The fact that such insulation also increases the compressive stress on the cold side is inconsequential to the calculation of the total probability of failure - the contribution from this side to $P_{f,total}$ remains zero. Figure 8.16 presents results for the total probability of failure as a function of the insulation recess depth (measured from the inner edge of the seal) for a pillar separation of 25 mm. The recess depth is the same on both sides of the glazing and the thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$. The probabilities of failure were calculated according to the same procedure as above except that different temperature induced stress results were used (e.g. the results of Fig. 6.28b for a recess depth of 25 mm). Figure 6.29 shows the warm side stress profiles that were used for three of the other recess depths. The corresponding stress profiles for the cold side, which are not shown in this figure, were also included in the analysis. However, as previously noted, the stresses on the cold side did not lead to any contributions to $P_{f,total}$.

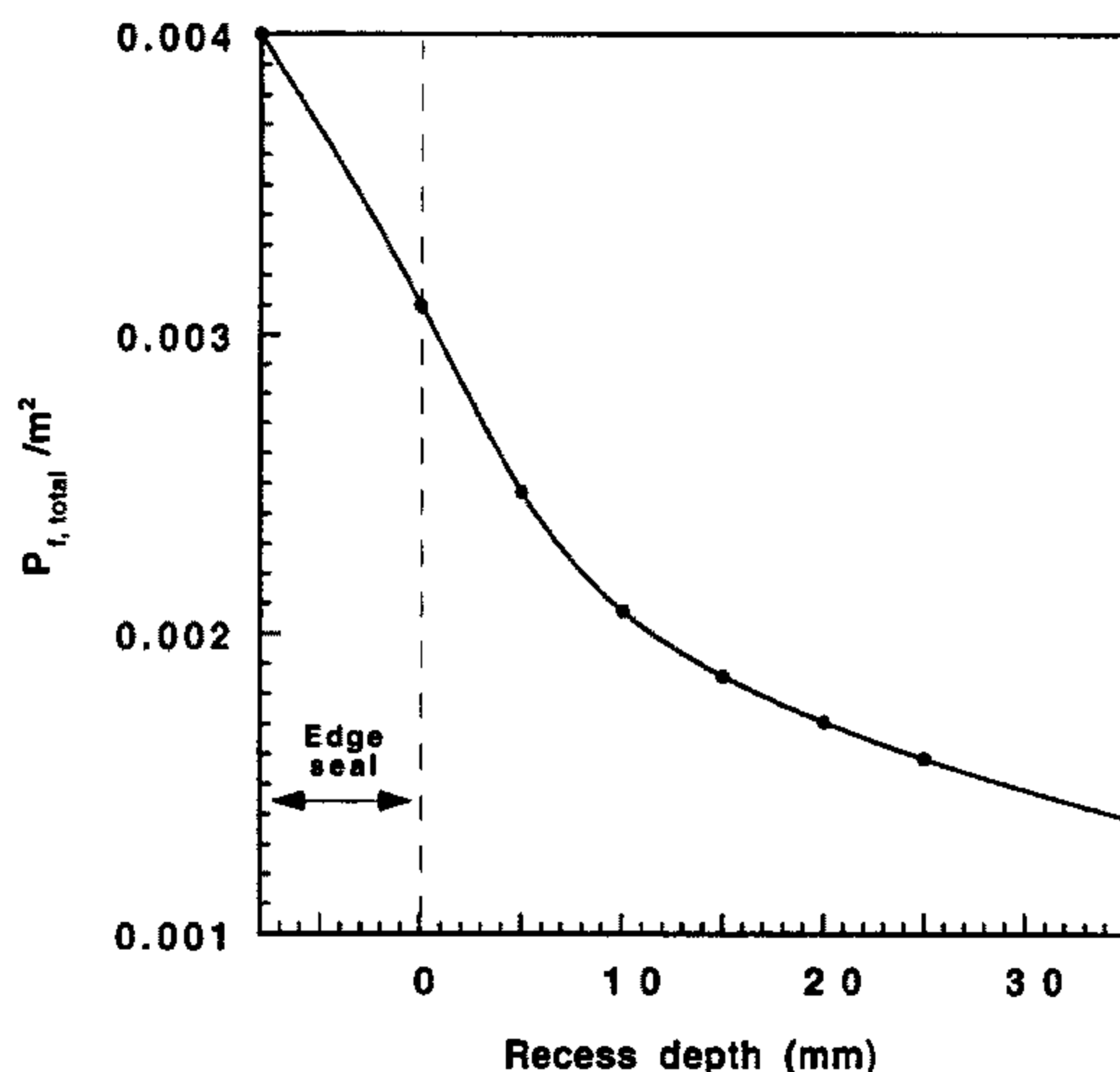


Fig. 8.16: The total probability of glazing failure as a function of the insulation recess depth for a pillar separation of 25 mm. The recess depth is the same on both sides of the glazing, and it is measured from the inner edge of the 8 mm wide seal. The thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$. The results correspond to the dimensions and conditions given above for Figs. 8.1 and 8.2.

8.5 Conclusion

The glass failure prediction model presented by Beason and Morgan [5] and modified by Fischer-Cripps and Collins [4] has been used to predict the probability of failure for vacuum glazing. This modified model accounts for the following factors which affect the strength of glass and thus the total probability of glazing failure, $P_{f, total}$: the magnitude and duration of the tensile external surface stresses, the area of the external surfaces exposed to these stresses, and the nature of the surface flaws. The analysis was based on the combination of the stresses due to the presence of a temperature difference across the glazing, and the stresses due to atmospheric pressure. These two types of stress were determined from the finite element models presented in the last two chapters. In order to simplify greatly the solution procedure, the profiles of the temperature induced direct stresses were assumed to be uniform along the edges, and the profiles of the pressure induced stresses were assumed to be uniformly periodic along the edges. For each side of the glazing, the total shear stress was assumed to be

uniform and equal to the average value on the side being considered. Also, the pressure induced stresses over the edge seal were assumed to be zero, and the stresses beyond three pillar rows from the inner edge of the seal were assumed to be uniform and equal to the center-of-glass values. All of these assumptions led to a net error in $P_{f,total}$ of only a few percent. It is noted that this error is probably less than that which is associated with the uncertainties in the glass surface flaw parameters (which were not considered).

On the cold side of the glazing, the combined stresses are compressive except right over the pillars where these stresses are tensile and small in magnitude. Since compressive stresses do not contribute to the total probability of failure, and since the cold side tensile stresses are too small to make significant contributions to this probability, the cold side of the glazing is not significant in the context of the probability of failure for vacuum glazings. This was found to be the case for all of the pillar separations considered in this study (which were between 20 and 40 mm). Larger pillar separations would of course lead to greater cold side tensile stresses over the pillars. However, it is much more likely that such separations would lead to catastrophic failure near the warm side edges than at any location on the cold side of the glazing. Thus, the following conclusions concern only the warm side of the glazing where, for the conditions considered in this study, the total direct stresses (and, more importantly, the corresponding maximum principal stresses) are entirely tensile.

The most important conclusion from this work is that the greatest contribution to the total probability of failure occurs over the region near the warm side edges due to the large tensile stresses at this location. This was found to be the case for all of the pillar separations considered in this study. For example, for a pillar separation, λ , of 25 mm, an edge seal of width 8 mm, and for the pillars, glazing dimensions and conditions considered above, most of the contribution to $P_{f,total}$ comes from the warm side region within the first row of pillars from the edge of the glazing (i.e. within 33 mm from the edge). It is noted that the actual position of the first row of pillars from the edge is somewhat arbitrary and that it is influenced by the desire to obtain uniformly sized unit cells elsewhere in the glazing. In this study, it was assumed that the first row of pillars was at a distance λ from the inner edge of the seal. The techniques presented above can be used to obtain results for other positions of this row.

The above conclusion concerning the contribution to $P_{f,total}$ from the warm side edge region implies that catastrophic failure of the glass in vacuum glazing is most likely to occur near the edges on the warm side of the glazing. This theoretical prediction is

consistent with the observation that all spontaneous failures (i.e. non-impact related failures) of vacuum glazings occur as cracks which initiate at and propagate along the edges - usually very close to the edge seal. This has been observed in early versions of vacuum glazings in which the pillars provided inadequate levels of interpane support near the edges; such poor support led to intolerably large stresses and thus to failures at this location. The initiation of failure near the edges is also probably assisted by the different nature of the flaws at and near the cut edges of the glass sheets of the glazing (the flaws are presumably more numerous and larger). This was not considered in the above study.

Since the warm side stresses near the edges have such a strong influence on the total probability of failure, it should be possible to reduce this probability by reducing these stresses. As discussed in Section 6.6.5, this can be accomplished by insulating the edges. An investigation into the variation of $P_{f,total}$ with the extent of edge insulation has shown that significant reductions in $P_{f,total}$ can be achieved with edge insulation on both sides of the glazing. As also noted in Section 6.6.5, for the conditions considered in this study, even greater reductions in the edge stresses (and thus presumably in $P_{f,total}$) can be accomplished by insulating only the cold side of the glazing. However, since such a configuration of insulation is not the most effective way of reducing the edge conduction, it is unlikely to be used in actual glazings and thus it was not considered in this work.

It is also possible to reduce the warm side tensile edge stresses by other means. One possible approach is to mount the glazing in a frame which forcibly constrains the edges. This can lead to compressive stresses that partially counteract the warm side tensile edge stresses. Another option is to deform the entire glazing by a small and barely noticeable amount at the time of manufacture prior to the formation of the fused edge seal and the evacuation process by applying a force to the middle of the structure. As a result, one external surface will be slightly concave, and the other one will be slightly convex. The concave external surface will contain compressive stresses which will partially counteract the warm side tensile edge stresses that occur under service conditions - provided that the glazing is installed so that the pre-stressed concave surface faces the warm side environment. The other external surface will of course be slightly convex and contain tensile stresses. However, as long as these stresses are smaller in magnitude than the compressive stresses expected for this surface under service conditions, they will not affect the total probability of failure. The above approaches to reducing the tensile edge stresses could be investigated in future studies of the stresses in vacuum glazing.

In this chapter, issues relating to the design of vacuum glazing have been indirectly raised. It has been noted that, as the pillar separation is increased, the stresses in the glazing - and thus the total probability of failure - increase. However, an increase in the pillar separation does have the desirable effect of reducing the center-of-glass U-value, thereby improving the thermal performance of the product. Thus, in order to design a glazing with both a low total probability of failure and a good thermal performance, one must pay particular attention to the design of the pillar array. Such design considerations are discussed in the next chapter. Also presented in that chapter is a comparison of the improvements to both the total probability of failure and the thermal performance of the glazing that can be accomplished with edge insulation.

8.6 Symbols

A	glass surface area
c	biaxial stress correction factor
E_{beam}	Young's modulus of elasticity for beam elements
G_{beam}	shear modulus for beam elements
m	surface flaw parameter indicating the statistical distribution of flaw sizes
k	surface flaw parameter indicating the absolute size and density of flaws
K	stress intensity factor
K_{Ic}	critical value of K at which instantaneous failure occurs
K_{Isc}	sub-critical crack growth stress intensity factor
n	ratio of the minimum principal stress to the maximum principal stress
N_{cell}	number of unit cells corresponding to a specified representative unit cell
P_f	probability of failure
P_s	probability of survival
T_{edge}	temperature of the glass at the edge of the glazing
T_{mean}	mean temperature of the glazing
U_{g-g}	glass-to-glass center-of-glass U-value
w_{seal}	width of edge seal

Greek Symbols

β	risk of failure function
λ	pillar separation
θ	orientation of flaw with respect to the stress field
σ_{xx}	total direct stress parallel to x-axis
σ_{yy}	total direct stress parallel to y-axis
σ_{11}	maximum principal stress
σ'_{11}	equivalent maximum principal tensile stress
σ_{22}	minimum principal stress
τ_{xy}	shear stress

Common Subscripts

e	denotes elemental quantity
total	total quantity

8.7 References

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9. The Design of Vacuum Glazing

9.1 Introduction

In order to be commercially acceptable, vacuum glazing must be designed so that the product has both a good thermal performance and a low probability of failure. Several factors which affect the achievement of these goals have been discussed in the previous chapters. The thermal performance is influenced by the emittances of the internal glass surfaces, the design of the pillar array and the use of edge insulation. In the following design study, the surface emittances are not considered; it is assumed that the glazing is manufactured with the type of glass that currently provides the lowest practicably achievable radiative U-value for vacuum glazing: K glass. If the use of edge insulation is treated as a separate case (as is done in this study), then the glazing thermal performance can be considered to be only influenced by the design of the pillar array. For the sake of convenience, this thermal performance is gaged in terms of the pillar array component of the glass-to-glass center-of-glass U-value. As discussed in the previous chapter, the total probability of glazing failure is also influenced by the design of the pillar array. Therefore, the following study of the considerations involved in designing a vacuum glazing that meets the above two goals focuses on the design of the pillar array - specifically on the radius and the separation of the pillars.

Certain design criteria for vacuum glazing have been described in detail by Collins and Fischer-Cripps [1], Fischer-Cripps et al. [2], and Collins et al. [3]. It is not intended to repeat these descriptions in the present work. Instead, the criteria are briefly mentioned and then built upon by being combined with design limitations determined from the probability of failure results of this thesis. This leads to a more comprehensive set of design criteria for vacuum glazing. Also, a novel comparison of the design implications of edge insulation is presented.

The criteria discussed in this chapter are presented in the context of pillars with circular cross-sections since most current designs of vacuum glazing are made with cylindrical pillars. The corresponding design criteria for pillars of square cross-section can be established according to the same approaches described below.

9.2 The Design of The Pillar Array

As noted in the last chapter, if the pillar separation is increased, the thermal performance of a vacuum glazing improves, but the total probability of failure becomes larger. Therefore, there are trade offs involved in determining the most desirable design of the pillar array that satisfies the two goals given above. It is noted that the thermal performance and the probability of failure of a vacuum glazing also depend on the pillar radius. Thus, the design trade offs are best represented as a set of curves which reflect the design criteria as functions of both the pillar separation and radius. Detailed explanations of the first three of the following four criteria can be found in the literature [1, 2, 3].

The first design criterion concerns the pillar array component of the overall glazing thermal performance. The dependence of the glass-to-glass center-of-glass pillar array U-value, U_{array} , on the pillar separation and radius (λ and a) is given by equation 4.3. Figure 9.1a shows a curve which represents the corresponding relationship between λ and a for a fixed value of U_{array} which, for this study, is taken to be the typical value of $0.40 \text{ Wm}^{-2}\text{K}^{-1}$. This relationship is also based on a glass thermal conductivity of $1.0 \text{ Wm}^{-1}\text{K}^{-1}$. Since all values of λ and a above the curve yield a pillar array U-value of less than $0.40 \text{ Wm}^{-2}\text{K}^{-1}$, the curve sets limits on the pillar separation and radius.

The issue of the atmospheric pressure induced compressive stresses within the pillars was briefly noted in Section 7.2. Since current designs of vacuum glazing employ pillar materials which are of sufficient strength to withstand the expected compressive stresses, this issue was not considered in the previous analyses of this thesis. However, for the sake of completeness, it is considered in this design study in order to specify another design criterion that must be met. If the compressive stresses within the pillars are too large, the pillars are likely to deform either plastically (in the case of metallic pillars) or catastrophically (in the case of ceramic pillars). For both cases, the end result will be that there will be insufficient inter-pane support and thus larger stresses near the edges. This in turn will lead to a larger total probability of glazing failure. It is therefore necessary to ensure that:

$$\frac{P\lambda^2}{\pi a^2} < S, \quad (9.1)$$

where S is the specified maximum allowable compressive stress in each pillar, and P is atmospheric pressure. This relationship can be derived from equation 7.3 and the

cross-sectional area of the pillar (πa^2). Generally, S must be less than about 1 to 2 GPa [2, 3]. In this thesis, for the sake of convenience, S is taken to be 1.3 GPa. Figure 9.1b shows the relationship between λ and a determined from equation 9.1 for this stress. Values of λ and a below this curve satisfy the compressive stress design criterion.

The possibility of conical indentation fracture occurring due the indentation of the pillars into the inner glass surfaces of a vacuum glazing was also briefly noted in Section 7.2. This type of fracture could lead to glazing failure. As with the compressive pillar stresses, conical indentation fracture was not considered in any of the previous calculations of this thesis because, for typical glazing designs, it has never been observed and it is thus not a concern. However, for the sake of completeness, it is considered in this study. The criterion to prevent conical indentation fracture can be determined from Auerbach's law [4] which relates the radius of a spherical indenter to the force necessary to initiate a conical crack. This relationship can be modified for an indenter which has a flat surface (e.g. a pillar). This leads to the following relationship between the pillar separation and the radius for which conical indentation fracture will not occur in a vacuum glazing:

$$\lambda < 27.6a^{3/4} \quad (9.2)$$

where λ and a are in meters [1, 3]. This relationship is presented in Fig. 9.1c. The conical indentation fracture design criterion sets further limits on the values for λ and a .

The fourth design criterion is the maximum acceptable total probability of glazing failure, $P_{f,total}$. This criterion can be determined on the basis of the probability results presented in Fig. 8.15 of the last chapter. Since these results are for only one pillar radius (0.125 mm), there are not enough data to define a λ versus a curve below which $P_{f,total}$ will be less than a specified value. However, it is possible to use the results to define a series of points for $a=0.125$ mm below which $P_{f,total}$ will be less than certain values. This is shown in Fig. 9.1d. For example, for $a=0.125$ mm, all pillar separations below point B ($\lambda=25$) yield a probability of failure less than 0.0040 m⁻². It is noted that the probabilities of failure in this figure correspond to a glazing of size 1 m by 1 m, and a glass thickness of 4 mm. All of the other design criteria (Figs. 9.1a to c) are independent of the glazing area and the glass thickness.

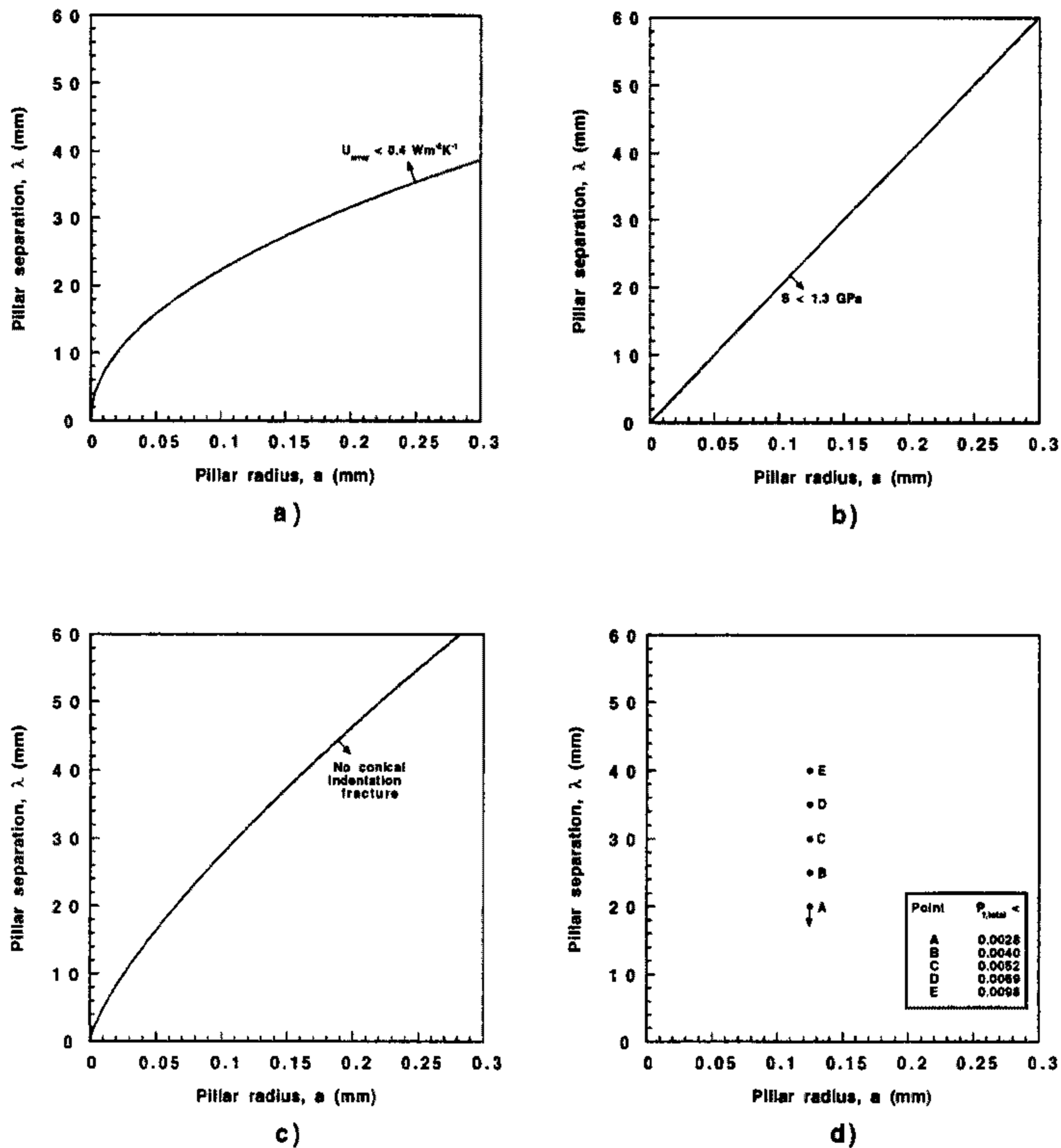


Fig. 9.1: Plots of the pillar separation versus the pillar radius which define the following four design criteria: (a) that the pillar array U-value, U_{array} , be less than $0.40 \text{ Wm}^{-2}\text{K}^{-1}$, (b) that the compressive stress in each pillar, S , be less than 1.3 GPa , (c) that the pillar indentation does not lead to conical indentation fracture, and (d) that, for a glazing of size 1 m by 1 m , the total probability of glazing failure, $P_{f,total}$, be less than certain specified values. For each of the first three criteria, the region of acceptable values of pillar separation and radius are as indicated by the arrows - i.e. either above or below the respective plot. The results of (d), which are per m^2 of glazing size, correspond to the dimensions and conditions given in Figs. 8.1 and 8.2, a glass thickness of 4 mm , and only to a pillar radius of $a=0.125 \text{ mm}$. For each point in (d), pillar separations at $a=0.125 \text{ mm}$ which yield a probability below the corresponding value for $P_{f,total}$ are below the point.

The curves and points of Figs. 9.1a to d can be combined to give an overall graphical representation of the design criteria, as shown in Fig. 9.2. For $a=0.125$ mm, the only pillar separation that meets all of the design criteria (to within 0.09 mm) is $\lambda=25$ mm, which is at point B. Thus, point B represents the optimum design point that satisfies the above requirements for a pillar radius of 0.125 mm. Of course a single design point limits the design options. It would be preferable to obtain a range of possible values for both pillar parameters which satisfy the design criteria. This can be done by establishing the relationship between λ and a for the total probability of failure criterion - i.e. to represent this criterion as a curve instead of as a series of points like in Fig. 9.1d. This would require repeating the calculations of Chapters 6, 7 and 8 for other pillar radii. Due to the significant additional work that this would involve, it was decided not to do such calculations for this thesis. However, it is of interest to describe how, as part of any future studies in this area, the methods of calculation presented in the previous chapters could be used to obtain the required results.

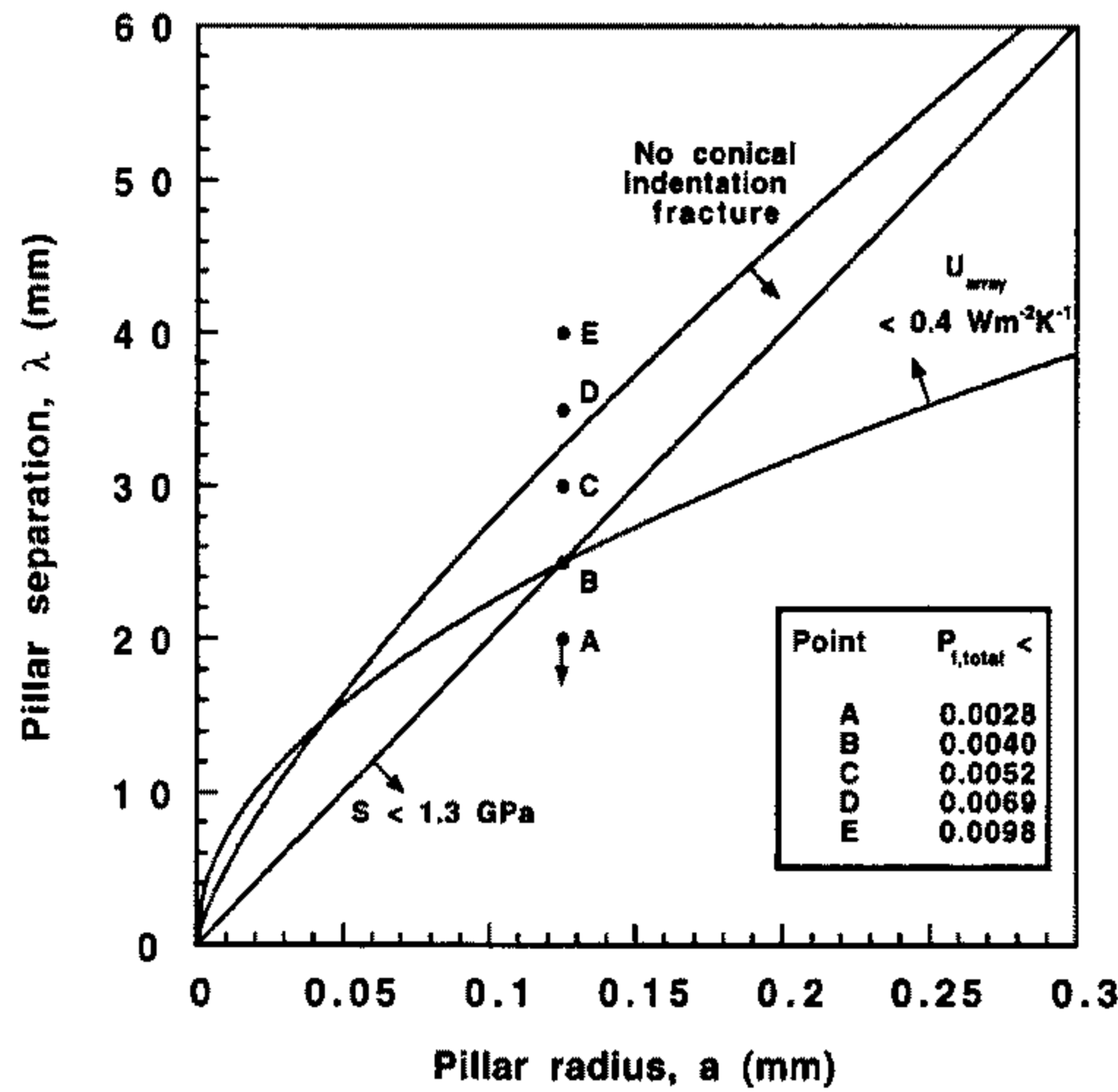


Fig. 9.2: The results of Figs. 9.1a to d can be combined to give a comprehensive design plot from which pillar separations and radii which meet all of the above four design criteria can be determined. The total probability of failure results correspond to the dimensions and conditions given in Figs. 8.1 and 8.2, a glass thickness of 4 mm, and only a pillar radius of 0.125 mm. For this radius, the only pillar separation that satisfies all of the design criteria is 25 mm (at point B). Thus point B represents the optimum design point for $a=0.125$ mm.

The temperature induced stresses would be calculated on the basis of glass-to-glass center-of-glass U-values determined for the different values of a (and λ) according to the iterative method mentioned half-way through Section 5.3. The pressure induced stresses would be calculated on the basis of the elastic properties of the simulated pillars (i.e. E_{beam} and G_{beam}) given in Section 7.3.2 for alumina pillars of radius 0.125 mm; these properties would be modified for the different values of λ and a according to equations 7.3 to 7.6. For each set of λ and a , the temperature and pressure induced stresses would be combined and the total probability of failure would be determined as per the procedure presented in Section 8.3. This analysis would yield a curve for each pillar radius of the total probability of failure as a function of the pillar separation (like the curve shown in Fig. 8.15). The resulting curves would be presented together in order to determine the pillar separation and radius corresponding to the specified maximum allowable total probability of failure (e.g. by noting the values of λ and a at which the curves intersect a horizontal line at the specified $P_{f,total}$). These values would then be used to form a design curve which, as a more complete failure design criterion, establishes the relationship between λ and a for the selected probability of failure; pillar separations and radii below the curve would lead to probabilities below the maximum value. This design curve would be combined with the other design curves from Figs. 9.1a to c to reveal what pillar dimensions - if any - satisfy all of the design criteria. Ideally, there would be a region of satisfactory pillar parameters. The size of such a region could of course be changed by adjusting the limiting values for U_{array} , S or $P_{f,total}$.

It should be noted that the exact values of $P_{f,total}$ which are used to establish the total probability of failure design criterion depend on many specific details of the glazing design. These include the glazing dimensions, the glass sheet thickness, the emittances of the internal surfaces, the use of edge constraints, the distance of the first row of pillars from the edge, the glass flaw parameters and the values of the stress intensity factor for instantaneous failure and sub-critical crack growth. These factors also depend on the manufacturing methods since they can be influenced by the pre-stressing of the glass sheets. It would be useful to investigate the dependence of $P_{f,total}$ on these details in order to develop a very comprehensive set of probability of failure criteria curves. These curves could be combined with the curves for the other three criteria given above to determine the design options for a wide range of possible situations. It is recommended that such an extensive investigation be done in a future study of the design of vacuum glazing.

9.3 The Design Implications of Edge Insulation

As noted in the previous chapters, the addition of edge insulation to a vacuum glazing improves the thermal performance of the glazing and also reduces the total probability of glazing failure. It is of interest to compare the results for these effects together on the same graph in order to demonstrate the dual implications of edge insulation. Such a graph could be used by a vacuum glazing designer to determine the insulation recess depth that would result in a certain U-value and/or a certain probability of failure.

It is recalled from Section 5.4.2.2 that edge insulation affects all of the window U-values: the frame, edge-of-glass and center-of-glass U-values. Therefore, in studying the thermal implications of edge insulation, it is most convenient to consider the effect of the insulation on the U-value which reflects all of the above U-values: the total window U-value, U_{total} . As explained in Section 5.2.4, in calculating the total U-value for vacuum glazing, edge insulation is considered to represent an insulated frame. Therefore, the frame component of U_{total} is calculated over the insulated region. Figure 9.3 shows how U_{total} varies with the insulation recess depth. The recess depth is the same on both sides of the glazing (i.e. both the warm and cold side edges are covered by equal extents of insulation), and it is measured from the inner edge of the seal. The thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$ and the glass-to-glass center-of-glass U-value, U_{g-g} , is the typical value of $1.10 \text{ Wm}^{-2}\text{K}^{-1}$. These finite difference modelling results for U_{total} were determined in the same manner used to obtain the results of Fig. 5.24d (which are for a slightly larger than typical U_{g-g} of $1.20 \text{ Wm}^{-2}\text{K}^{-1}$).

Figure 9.3 also shows the results from Fig. 8.16 for the variation in the total probability of failure as a function of the insulation recess depth for the same insulation configuration, insulation thermal conductance and U_{g-g} given above. The two plots of this figure demonstrate that significant improvements to the thermal performance and the total probability of failure of the glazing can be achieved with modest insulation recess depths. They also demonstrate that the rates of these improvements decrease with the recess depth.

The analyses used to determine the results of Fig. 9.3 (which correspond to only a pillar radius of 0.125 mm) could be used to obtain a more complete set of results for other pillar radii. Such a set of results could be used to establish insulation related design criteria for the maximum values of the total window U-value and the total probability of failure in terms of λ and a . This could be done in the same manner as described at the end of the last section on the basis of the corresponding maximum

values, and the procedure would lead to other design curves pertaining to edge insulation that could be added to the four design curves presented in Fig. 9.2.

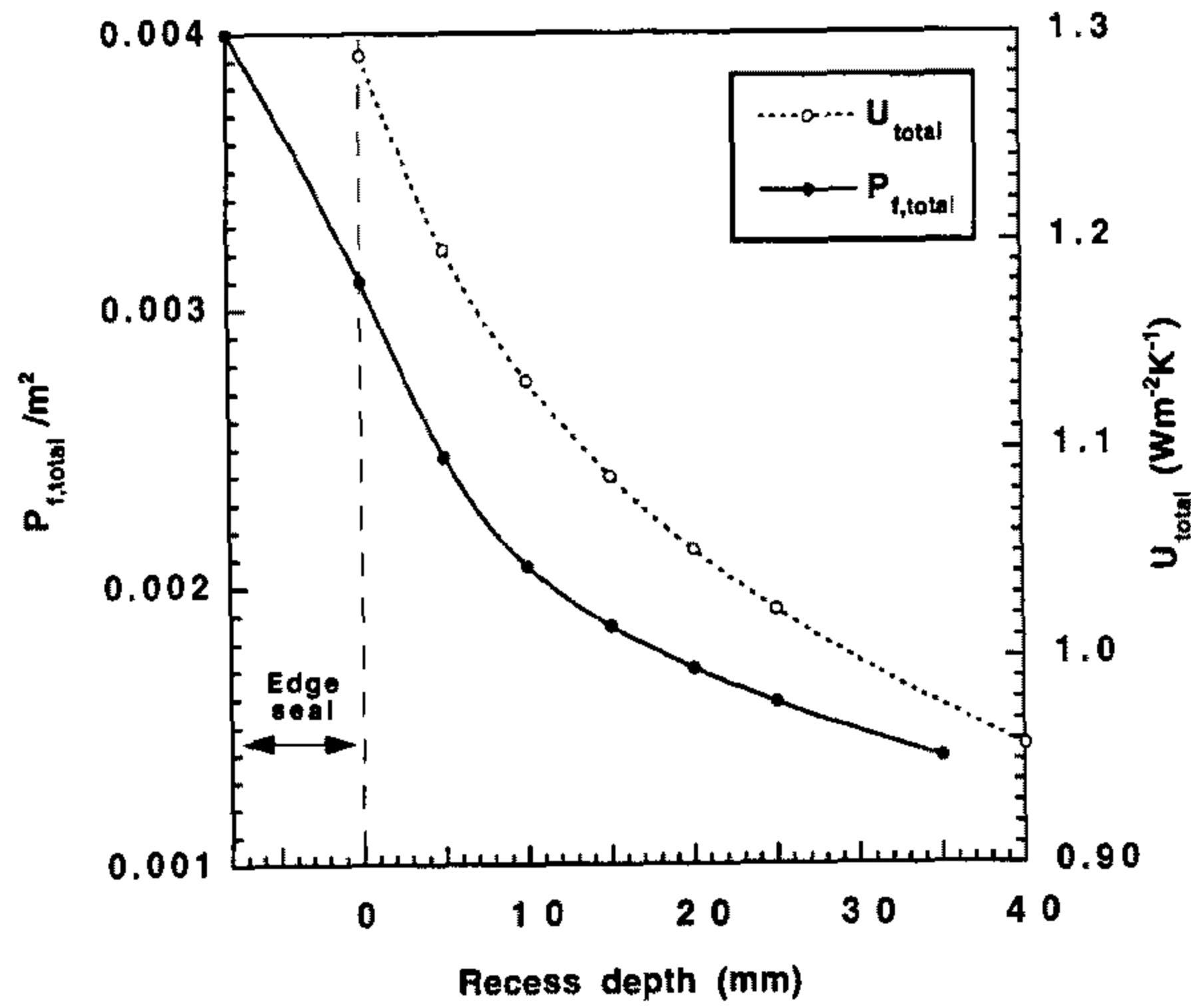


Fig. 9.3: Results for the total probability of glazing failure ($P_{f,total}$) and the total window U-value (U_{total}) as functions of the insulation recess depth. The recess depth is measured from the inner edge of an 8 mm wide edge seal (the location of which is indicated). The probabilities are for a glazing of size 1 m by 1 m and a glass thickness of 4 mm. Both sets of results correspond to the dimensions and conditions referred to in Figs. 8.1 and 8.2, except that an edge seal of finite was not simulated in obtaining the U_{total} data. The recess depth is the same on both sides of the glazing and the thermal conductance of the insulation is $1.5 \text{ Wm}^{-2}\text{K}^{-1}$.

As noted in Section 8.5, for the conditions considered in this study, even greater reductions in $P_{f,total}$ can be obtained if the edges are insulated on only the cold side of the glazing. However, such a use of edge insulation is not the most effective way of reducing the edge conduction - and thus the total window U-value. Therefore, cold side only insulation is ideal for reducing the total probability of glazing failure, but it is not ideal for improving the glazing thermal performance. It is recommended that, as part of a future design study, the above analysis be repeated for cold side only insulation in order to obtain a plot similar to Fig. 9.3. Such a plot would show if it is possible to obtain acceptable values for $P_{f,total}$ and U_{total} by insulating just the cold side edges with less insulation than would be required if both the warm and cold side edges were insulated. If this is possible, then cold side only insulation could be used to save

on insulation material costs and to increase the view area. It would also be worthwhile to investigate the situation in which the edges on both sides of the glazing are insulated, but in which the cold side edges are better insulated than the warm side edges (e.g. with thicker insulation). Depending on the insulation thicknesses, it might be possible to obtain greater improvements to *both* $P_{f,total}$ and U_{total} with such insulation than with the same insulation on both sides.

9.4 Conclusion

It is important that vacuum glazings be designed to have both a good thermal performance and a low probability of failure. If the radiative U-value of a vacuum glazing is not considered, and if edge insulation is considered separately, the main design issues to address concern the pillar array. It has been shown in previous chapters that if the pillar separation is increased, the glazing thermal performance improves but the total probability of glazing failure becomes larger. Since the thermal performance and probability of failure also depend on the pillar radius, a full design study should consider both the pillar separation and the pillar radius. The effective design of vacuum glazing involves the selection of these two pillar parameters according to four design criteria.

For the study of the design of a vacuum glazing without edge insulation, it is convenient to gage the glazing thermal performance in terms of the pillar array U-value. The pillar array U-value design criterion is set by specifying a maximum desired value for U_{array} . Similarly, the total probability of failure criterion is defined by the choice for the maximum acceptable value for $P_{f,total}$. Two other criteria concern the requirement not to exceed a maximum compressive stress in the pillars, and the requirement to prevent the formation of conical indentation fractures. Each of the above four design criteria can be represented by a relationship between the pillar separation and the pillar radius. The relationships can be combined to create a comprehensive design plot which shows which values - if any - of the pillar separation and radius satisfy all of the criteria. The probability of failure results determined in the last chapter can only be used to define a series of possible design points for a single pillar radius on such a combined plot. However, it has been shown how the techniques presented in the previous chapters can be used to establish a more extensive failure criterion curve.

Another issue to be considered in the design of vacuum glazing is the use of edge insulation. As previously discussed in this thesis, the addition of edge insulation to a vacuum glazing improves the thermal performance of the glazing and decreases the total probability of glazing failure. These effects of edge insulation have been compared together in a single figure. Such a figure could be used by a glazing designer to select a desirable extent of edge insulation.

It is noted that there has been no discussion of what might be actual acceptable values for the probability of failure for a vacuum glazing in specific applications. This is because there are no well defined industry standards and the actual probability of failure depends on the specific applications [5]. It is important to note that the uncertainties in some of the factors which affect the probability of failure calculations - especially the glass flaw parameters - can lead to large potential errors in the absolute values of the probabilities of failure. All that can be said at this stage is that the analysis presented in this chapter is a good indication of the way in which the probability of failure of vacuum glazing depends on various design options. The validation of this method of analysis and the determination of acceptable values for commercial applications of vacuum glazing must await a properly conducted field trial of many samples of vacuum glazing. It is possible that such a field trial will be performed within the next few years.

In this chapter, results from the modelling work and from the probability of failure calculations of the previous chapters have been brought together for a study of the considerations involved in the design of vacuum glazing. This study effectively shows how the theoretical concepts of this thesis can be applied to the practical design of actual vacuum glazing products. There are of course many areas of further investigation which can add to the understanding of how to design vacuum glazings. Some of these areas have been noted, and are they briefly mentioned again in the next - and last - chapter of this thesis.

9.5 List of Symbols

a	pillar radius
E_{beam}	Young's modulus of elasticity for beam elements
G_{beam}	shear modulus for beam elements
P	atmospheric pressure
$P_{f,total}$	total probability of glazing failure
S	compressive stress within each pillar
U_{array}	pillar array U-value
U_{g-g}	glass-to-glass center-of-glass U-value
U_{total}	total window U-value
λ	pillar separation

9.6 References

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10. Conclusions and Recommendations for Further Work

10.1 Conclusions

In order to be commercially acceptable, vacuum glazing must be designed so that the heat flows are small enough to result in a good glazing thermal performance, and so that the stresses lead to a low probability of glazing failure. This thesis is a study of three interrelated issues concerning vacuum glazing: the heat transfer processes, the stresses, and the probability of failure. Each of the following three heat flows across the glazing is separately investigated by experimental and modelling techniques: the radiative heat transfer between the internal glass surfaces, the conduction through the support pillars, and the lateral heat flow along the glass sheets and through the edge seal (the edge conduction). The stresses due to a temperature difference across the glazing are then calculated with an experimentally validated finite element model on the basis of the understanding of the heat flows across the glazing. The other type of stress in vacuum glazing considered in this work, the atmospheric pressure induced stress, is determined from a model which is qualitatively validated on the basis of the associated surface deflections of a vacuum glazing. The temperature and pressure induced stresses are then combined to determine the total stresses in vacuum glazing, from which the total probability of glazing failure is calculated. The thermal, stress and failure results of this thesis are tied together in a design study in which a set of design criteria are specified and the factors which influence these criteria are quantified.

Most of the validation of the thermal modelling work of this thesis depends on the heat flow measurements made with a specially designed guarded hot plate apparatus. This apparatus is referred to as the thermal conductance apparatus, or TCA. The TCA consists of a hot plate and an isothermal cold plate on either side of a glazing or a simulated glazing sample. The hot plate consists of an isothermal guard which surrounds a metering piece. The heat flow across the evacuated space of the sample is determined by measuring the amount of power - the null power - that must be dissipated in a resistor in the metering piece in order to obtain equal temperatures at specific locations in the metering piece and the guard. The temperatures are measured with thermistors which are incorporated into a Wheatstone bridge circuit.

In order to ensure that sufficiently accurate measurements could be made with the TCA, the following issues were addressed. The thermal resistances across the apparatus were minimized by using a metal of a high thermal conductivity, and by

making the layers of epoxy resin (which bond the metal plates to glass face plates) as thin and as consistent as possible. Significant parasitic heat flows from the central metering piece were avoided by selecting an appropriate diameter for the guard, and by thermally terminating the wires from the resistor core to the metering piece before they were connected to the circuitry at the guard. This was also accomplished with a high thermal resistance between the metering piece and the guard. Another issue that was considered was where the thermistors should be located, and this issue is related to the effective measurement area.

Due to the thermal resistance of the glass below the metering piece, the heat flow from this piece spreads out to an area at the evacuated space which is greater than the geometric area of the metering piece. The former area, which is referred to as the effective measurement area, must be well known in order to obtain accurate U-value results from the TCA heat flow measurements. This area is best determined by a calibration procedure which compares the experimental and theoretical results for the heat flow across a sample with a well defined thermal conductance. The sample used for this calibration consisted of two sheets of uncoated glass separated by an evacuated space, for which the heat flow could be accurately predicted according to the exact method mentioned below. The effective area determined in this way, $1.45 \pm 0.01 \times 10^{-4} \text{ m}^2$, is known sufficiently accurately for distinguishing among the three heat flows across the evacuated space of vacuum glazing at locations remote from the edges: the radiative heat transfer, the pillar conduction and the gas conduction (the latter of which is usually negligible). Based on the error in the effective area and on the uncertainty in the temperature difference across the evacuated space, the accuracy for the per unit area heat flow measurements is estimated to be $\pm 1.4\%$. The TCA measurements indicate that, as expected, there is a linear relationship between the voltage output from the thermistor bridge circuit and the power input to the metering piece resistor. With the use of an appropriate correction for the pillar conduction, the heat flows measured with the TCA can be used to determine the glass-to-glass center-of-glass U-value associated with each of the following two heat flows: the interpane radiative heat transfer, and the conduction through the array of support pillars.

The interpane radiative heat transfer across vacuum glazing can be significantly reduced by coating one or both internal glass surfaces with a low emittance coating. The high temperature manufacturing process for vacuum glazing limits the choice of suitable glass coatings (the glazing is brought to a temperature of about 500 °C for about an hour in order to melt the solder glass for the edge seal). The only economical

transparent low emittance coating that has been found to withstand the high temperature glazing manufacturing process without significant degradation of the optical and thermal properties of the coating is pyrolitically deposited doped tin oxide. The particular brand of doped tin oxide coated glass considered in this thesis is the brand which is most commonly used in vacuum glazing: K glass. Most vacuum glazings are made with two sheets of K glass, while others are made with either two sheets of uncoated glass or a combination of uncoated glass and K glass. Therefore, the study of the interpane radiative heat transfer concerned the following two surfaces: uncoated glass and K glass.

The radiative heat flow across the evacuated space of vacuum glazing can be considered to be radiative heat transfer between plane parallel surfaces. This heat transfer is usually calculated on the basis of the hemispherical total emittances of the surfaces, and also on the basis of the average of the surface temperatures. This is the approach given in a standard for calculating the radiative heat transfer between plane parallel glass surfaces [1]. Another method involves an integration over angle of the directional total emittances in order to obtain an effective emittance for the system. A third and a more exact method, which is presented in this thesis, calculates this radiative heat transfer according to an integration over both wavelength and angle of the directional spectral emittances of the surfaces and the black body spectra at the specified surface temperatures. Accurate measured heat flows that were obtained with a guarded hot plate apparatus which is similar to the TCA (but which has a larger effective measurement area) were compared with the results predicted with each of the above three approaches. The first and second approaches underestimated the actual heat flow by as much as 5% and 4%, respectively. However, to within the range of the respective errors ($\pm 1.4\%$) the results of the third method agreed with the experimental data. Smaller area measurements made with the TCA of heat flows between combinations of uncoated and coated glass surfaces are, to within the estimated uncertainties, in agreement with results calculated by the exact method and by the second of the two methods described above - the one involving an integration over angle. These small area measurements indicate that, while the standard method for calculating the radiative heat transfer between uncoated glass surfaces (the first of the above methods) is in error, it gives quite reasonable results for other combinations of surfaces (to within 2% of the limits of the uncertainties in the calculated and experimental results). This suggests that the standard should be revised for uncoated glass surfaces. It is therefore concluded that, in order to calculate accurately the radiative heat transfer between the internal glass surfaces of vacuum glazing, the above exact method should be used. However, for most of the calculations of this thesis, the

first of the above methods (the one based on the total hemispherical emittance) was used because it provides sufficiently accurate results.

The calculation of the interpane radiative heat transfer according to the above exact method requires a knowledge of the directional spectral surface emittances, which can be calculated from the appropriate optical constants. For uncoated glass, optical constants determined from reflectance measurements can be found in the literature. For K glass, the optical constants can be determined on the basis of the Drude model and reflectance spectrum measurements according to a novel method developed by Qi-Chu Zhang of the University of Sydney. The radiative heat transfer between two K glass surfaces, and between a K glass surface and an uncoated glass surface have been calculated on the basis of this method. The agreement with heat flows measured with the TCA validates this method.

Another heat flow across the glazing - the conduction through the support pillars - has also been studied by theoretical and experimental techniques. A simple analytic approach shows that, for a system with a single pillar between two sheets of glass, the thermal resistance of the system can be expressed as $1/2k_{glass}a$, where k_{glass} is the thermal conductivity of glass and a is the pillar radius. This expression is based on the assumptions that the thermal resistance for the heat flow in the pillar is very small and is thus negligible in comparison with the thermal resistance for the heat flow which spreads out in the glass, and that the glass sheets are of a semi-infinite thickness. Finite element modelling results and TCA measurements which reflect the effect of the actual thermal resistance for the heat flow in the pillar and the effect of a finite glass thickness show that the analytic solution provides a close estimation of the actual thermal resistance for the system with a single pillar. The U-value for the entire array of pillars is inversely proportional to the product of the thermal resistance for an individual pillar and the square of the pillar separation - i.e. $U_{array} = 2k_{glass}a/\lambda^2$.

A finite element model for the analysis of the effects of the heat conduction through a pillar has been developed. Pillar thermal resistances calculated on the basis of the results from this model are consistent with experimental results and they are close to thermal resistances obtained with the less exact analytic expression given above. This finite element model for an individual pillar is particularly useful for calculating the temperature non-uniformities at the external and internal glass surfaces due to the pillar conduction. These results can be used to determine the glass surface temperature across an array of pillars. Modelling results show that, for typical pillar dimensions and spacings and for the ASTM standard fenestration test conditions, the significant glass

surface temperature non-uniformities due to the heat conduction through individual pillars mostly occur close to the pillars - especially for the non-uniformities at the internal surfaces. The average external glass surface temperature determined on the basis of this exact modelling approach is in very good agreement with that based on a pillar array U-value calculated by the above analytic approach.

The temperature non-uniformities at the internal surfaces which are caused by pillar conduction slightly reduce the local radiative heat transfer around the pillars. However, this localized suppression of the radiative heat transfer has a negligible effect on the total center-of-glass U-value for typical glazings.

Edge conduction arises from the lateral heat flow along the glass sheets and through the thermally short circuiting edge seal. The edge conduction - and thus the influence of the associated edge effect on the surface temperatures - can be reduced with edge insulation, recessed edges and the addition of a third pane of low emittance glass to the glazing. Several modelling techniques and infrared thermography were used to study the effectiveness of these methods of reducing the edge conduction. A one-dimensional analytic model was developed to study the effects of edge conduction, and it was expressed in finite difference form and solved numerically. Temperature profile results from the finite difference solution are in good agreement with results obtained from a more exact three-dimensional finite element model which was also developed for an edge conduction analysis. The results of the two modelling approaches do however slightly diverge close to the edge due to the fact that the simple one-dimensional finite difference model does not simulate an actual edge seal of finite width and thickness. However, the finite difference model can be used to obtain reasonable estimations of the window U-values; for example, edge-of-glass U-values calculated on the basis of the results from this model are close to the corresponding edge-of-glass U-values obtained at Lawrence Berkeley National Laboratory with a commonly used fenestration modelling package (FRAME3). Finite difference and finite element modelling results show that the addition of insulation to the edges of vacuum glazing reduces the edge conduction, and that it significantly increases the sightline temperature at the warm side external glass surface. Due to this effect on the warm side sightline temperature, edge insulation also improves the condensation resistance of the glazing.

Infrared thermographic tests were performed on a sample of vacuum glazing at Lawrence Berkeley National Laboratory. External glass surface temperatures obtained with the above mentioned finite element model for the analysis of edge

conduction are in agreement with the infrared thermographic results. Therefore, the infrared thermographic measurements experimentally validate the finite element model. Since results from the finite element and finite difference models are in good agreement for the same conditions as for one of the thermographic tests, the infrared thermographic measurements also validate the finite difference model and the analytic model on which it is based. Infrared thermographic tests were also performed with the glazing mounted in a fixed wood casement frame for both flush and recessed edge configurations, as well as with and without a third pane of low emittance glass. The measurements show that significant increases in the sightline temperature at the warm side external glass surface can be achieved with these configurations - especially with the addition of the third pane of glass. Modelling results show that these configurations also result in significant decreases in the frame and edge-of-glass U-values.

The infrared thermographic results were also used to validate the above mentioned finite element model which was used for the analysis of the pillar conduction. The magnitude and period in the variation of the external glass surface temperatures predicted by this model are in agreement with the infrared thermographic results for the same conditions. The average centre-of-glass surface temperature obtained from the infrared thermographic measurements was found to be consistent with the corresponding temperature calculated on the basis of a predicted glass-to-glass center-of-glass U-value. This U-value is the sum of a temperature corrected measured U-value, and a pillar array U-value calculated by the analytic relationship given above. Therefore, the infrared thermographic measurements also validate the methods used to determine the radiative and pillar array U-values.

The main rationale for the analysis of the stresses in vacuum glazing was to provide results for calculations of the probability of glazing failure. Since the sub-surface glass and the internal glass surfaces of vacuum glazing are relatively well protected from processes and water vapour which can assist with failure-inducing crack growth and propagation, it was only necessary to consider the results for the stresses at the external glass surfaces for the stress and failure studies of this thesis.

A temperature difference across a vacuum glazing results in a differential thermal expansion and contraction of the two glass sheets. This can cause the sheets to bend, thereby leading to stresses within and at the surfaces of the sheets. These stresses were calculated for an unconstrained vacuum glazing with a finite element model on the basis of the above understanding of the heat transfer processes in the glazing. The modelling solution is based on temperatures predicted with the finite element model

referred to above for the analysis of edge conduction (this model also appropriately simulates the interpane radiative heat transfer and the conduction through the pillar array). The stress simulation model shows that the external surface temperature induced stresses at the center of the glazing are isotropic. It also confirms the simple analytic understanding of the temperature induced stresses in that it shows that the external surface stresses near the edges are much greater in magnitude than the external surface stresses near the center of the glazing, and that these larger edge region stresses are mostly parallel to the edge of the glazing. These edge region stresses are fairly uniform along the edges except for near the glazing corners where the stresses decrease to zero. For a typical vacuum glazing and for the ASTM standard fenestration test conditions, the edge temperature is less than the mean glazing temperature. This leads to additional parallel stresses at the edges. The external surface stresses on the warm side of the glazing are entirely tensile, and the external surface stresses on the cold side of the glazing are entirely compressive - except for the external surface stresses at the edge which are normal to the edge; these normal stresses are slightly compressive on the warm side and they are slightly tensile on the cold side. The additional parallel tensile stresses at the edges can be reduced with edge insulation.

In order to obtain data with which to validate the model for the temperature induced stresses, an apparatus was constructed for the measurement of the warm side external surface deflections and strains under well defined conditions. In this apparatus, a 0.5 m by 0.5 m sample of vacuum glazing was supported in an essentially unconstrained manner between isolated warm and cold cells. The cell temperatures were maintained by water flows in the plates which covered the cells. The deflections were measured with a dial micrometer, for which the reference surface was a bar coupled to the warm side surface of the glazing via a three-point mounting configuration; this configuration effectively preserved the alignment between the bar and the glazing sample without constraining the bending of the sample. Surface strains were measured with strain gages which were connected into bridge circuits. Compensating gages were utilized in order to minimize the effects of the temperature dependencies of the strain gages. In general, there is a good agreement between the experimental and finite element modelling results for both the deflections and the strains. It is clear that the deflection measurements constitute the most sensitive test of the model since the accuracy of these data is much greater than that of the strain measurements (e.g. 0.5 % versus 3 % for the respective minimum errors). However, to within the accuracy obtained, most of the deflection and strain measurements agree with the modelling results, and all of these measurements are qualitatively consistent

with the trends predicted by the model. The modelling and experimental results show that the center-of-glass strain is isotropic and proportional to the glass-to-glass temperature difference across the glazing, and that both the parallel strain at the edge of the glazing and the deflection between the edge and the center of the glazing are proportional to this same temperature difference. The results obtained with simple analytic models are consistent with these trends.

Atmospheric pressure leads to compressive stresses in the pillars, stresses at the internal glass surfaces associated with the indentation of the pillars into the glass sheets, and tensile stresses at the external glass surfaces above the pillars and near the edge seal. It also causes the glass sheets to come together through the compression of the pillars, and through the indentation of the pillars into the glass surfaces; these two processes affect the stresses due to atmospheric pressure. Only the pressure induced stresses which are likely to lead to the failure of current designs of vacuum glazing were studied in this thesis - the stresses at the external glass surfaces. These stresses were simulated with a finite element model. In order to obtain appropriate results with a reasonable number of nodes, several simplifications were made with this model. Due to symmetry considerations, it was only necessary to model a half thickness of a vacuum glazing. The glass sheet was simulated with two-dimensional thin shell plate elements, and the individual pillars were simulated by beam elements which had appropriate material properties based on experimental results for the stiffness of a typical pillar-glass system. The experimental validation of this model was based on the results for the deflections associated with the pressure induced stresses since the deflections are much easier to measure than the strains associated with the stresses. The modelled deflections are qualitatively consistent with measured deflections for somewhat different conditions in that they are of the same order of magnitude, that they are not significantly influenced by the first row of pillars from the edge, and that there is a maximum negative deflection between the second and third rows of pillars from the edge due to the non-uniform forces exerted by the pillars on the glass near the edge.

The pressure induced stresses calculated with the above model are, over most of the glazing, close to the results obtained by a fully three-dimensional finite element model. Some appropriate post-processing of the results over the individual pillars is required in order to account for the nature of the connections between the plate and beam elements. The modelling results show that the largest external glass surface stresses due to atmospheric pressure are near the edge seal and over the pillars. The results also show that, near the edge seal, the pressure induced stress normal to the

edge is much greater than that parallel to the edge. The stresses near the seal and the stresses above the pillars vary with the pillar separation. The functional relationship between the modelled isotropic external glass surface stress above the pillars and the pillar separation is roughly consistent with that predicted by a simple analytic approach (this stress is approximately proportional to the square of the pillar separation). For highly insulating vacuum glazings, the stresses due to atmospheric pressure for typical pillar properties and spacings are mostly smaller in magnitude than the stresses due to the bending caused by a temperature difference across a typically sized glazing which is exposed to the ASTM standard fenestration test conditions.

The modelling results for the stresses due to the presence of a temperature difference across the glazing and for the stresses due to atmospheric pressure were combined for a study of the probability of failure of vacuum glazing. This study was based on a glass failure prediction model which accounts for the following factors which influence the probability of failure: the magnitude, orientation and duration of the applied stress, the surface area of the glass exposed to this stress, and the surface flaw parameters. As explained above, it was only necessary to consider the stresses at the external glass surfaces for this study. The total probability of glazing failure was calculated on an elemental basis; the elemental probabilities of failure were determined from the maximum and minimum principal stresses and the total shear stresses which corresponded to the sums of the temperature and pressure induced direct and shear stresses. For the conditions and pillar separations considered in this study, the combined stresses on the cold side of the glazing are compressive - except right over the pillars where there are tensile stresses which are very small in magnitude. It is noted that compressive stresses do not contribute to the total probability of failure, and that the cold side tensile stresses are too small to make significant contributions to this probability. Therefore, for conditions and dimensions considered, the cold side of the glazing is not significant in the context of the probability of failure. Thus, the following conclusions concern only the side of the glazing where the combined stresses are entirely tensile.

The most important conclusion from this study is that the greatest contribution to the total probability of failure comes from the region near the warm side edges due to the large tensile stresses at this location. This was found to be the case for all of the pillar separations considered in this study. For example, for a uniform pillar separation of 25 mm, an edge seal width of 8 mm, and for the pillars, glazing dimensions and conditions that were considered, most of the contribution to $P_{f,total}$ comes from the warm side

region within the first row of pillars from the edge of the glazing (i.e. within 33 mm from the edge).

The above conclusion concerning the contribution to $P_{f,total}$ from the warm side edge region implies that catastrophic failure of the glass in vacuum glazing is most likely to occur near the edges on the warm side of the glazing. This theoretical prediction is consistent with the observation that, in early designs of vacuum glazing, many spontaneous failures (i.e. non-impact related failures) of the glazings occurred as cracks which initiated at and propagated along the edges - usually very close to the edge seal. This is because, in these early designs, the pillars provided inadequate interpane support near the edges; such poor support led to intolerably large stresses and thus to failures at this location. The initiation of failure near the edges is also probably assisted by the different nature of the flaws at and near the cut edges of the glass sheets of the glazing (the flaws are presumably more numerous and larger in this region). This was not considered in the probability of failure study because the surface flaw parameters for cut edges were unknown, and because they are probably strongly dependent on the cutting technique.

Since the warm side stresses near the edges have such a strong influence on the total probability of failure, it should be possible to reduce this probability by reducing these stresses. This can be accomplished by insulating the edges because edge insulation reduces the additional parallel tensile stresses near the edges. An investigation into the variation of $P_{f,total}$ with the extent of the edge insulation has shown that significant reductions in $P_{f,total}$ can be achieved with edge insulation on both sides of the glazing. For the conditions considered in this study, even greater reductions in the edge stresses (and thus presumably in $P_{f,total}$) can be accomplished by insulating only the cold side of the glazing. However, since such a configuration of insulation is not the most effective way of reducing the edge conduction, it is unlikely to be used in actual glazings and thus it was not considered.

Results from the heat transfer and stress modelling work and from the probability of failure calculations were brought together for a study of the considerations involved in the design of vacuum glazing. The study showed how the theoretical concepts of this thesis can be applied to the practical design of actual vacuum glazing products. Four design criteria were defined for this study: the U-value for the array of support pillars, the maximum acceptable total probability of failure, the maximum compressive stress in the pillars, and the prevention of conical indentation fractures at the internal glass surfaces in contact with the pillars. Each of these criteria was represented by a

relationship between the pillar separation and the pillar radius. The relationships were combined to create a comprehensive design plot which shows which values of the pillar separation and radius satisfy all of the criteria. Since edge insulation improves the thermal performance of vacuum glazing while reducing the total probability of the failure of the glazing, the use of this insulation permits a wider choice of satisfactory pillar separations and radii. As noted below, there are several other factors which influence the design of vacuum glazing which can be considered.

It has been shown that, for the ASTM standard fenestration test conditions, a typical vacuum glazing can have a glass-to-glass center-of-glass U-value as low as about $1.10 \text{ Wm}^{-2}\text{K}^{-1}$. The corresponding air-to-air center-of-glass U-value is $0.94 \text{ Wm}^{-2}\text{K}^{-1}$. It is of interest to compare this air-to-air U-value with the U-values for conventional glazings - which of course depend on the design details of the products. For example, as noted in Chapter 1, for a standard argon filled double glazing with a low emittance coating on one internal surface, it is possible to achieve an air-to-air center-of-glass U-value of $1.36 \text{ Wm}^{-2}\text{K}^{-1}$ and a corresponding total window U-value of $1.59 \text{ Wm}^{-2}\text{K}^{-1}$ [2]. This particular glazing has a 13 mm wide gas cavity, and the hemispherical total emittance for the coated glass surface is 0.04. These U-values are for the ASHRAE test conditions (which are similar to the ASTM conditions). The total window U-value for this glazing is the lowest value quoted by the manufacturer, and it is for a special design of a stainless steel spacer between the glass sheets, for the glazing mounted in a vinyl frame, and for the NFRC casement AA size (610 mm by 1220 mm). For a vacuum glazing of this size which has the above center-of-glass U-value of $0.94 \text{ Wm}^{-2}\text{K}^{-1}$ and no edge insulation, and which is exposed to the ASTM conditions, the total window U-value is about $1.36 \text{ Wm}^{-2}\text{K}^{-1}$; this U-value is based on an appropriate scaling of the individual U-values used for the result given in Fig. 9.3 for an uninsulated 1 m by 1 m glazing (for which the recess depth is zero). Therefore, in terms of both the center-of-glass and total window U-values, vacuum glazing has a better thermal performance than this particular double glazing. It is important to note that lower U-values for conventional glazings can be achieved with additional separated panes of glass and with suspended low emittance transparent thin films between the panes. For example, under the ASHRAE conditions, total window U-values of 0.68 to $0.74 \text{ Wm}^{-2}\text{K}^{-1}$ can be obtained with some designs of triple pane glazings which are filled with mixture of krypton and air and which have low emittance surfaces [3]. Another matter for comparison is the cost of manufacture. Garrison and Collins have shown that the cost of manufacture for vacuum glazing with one or two sheets of K glass is comparable to that for standard gas filled double glazing with the same number of K glass sheets [4].

It has also been shown that, for a vacuum glazing with a glass-to-glass centre-of-glass U-value of $1.10 \text{ Wm}^{-2}\text{K}^{-1}$ which is exposed to the ASTM standard fenestration test conditions, the stresses in a 1 m by 1 m vacuum glazing with no edge insulation lead to a probability of failure of about 0.004 m^{-2} . This probability can be reduced with edge insulation. A comparison with quoted acceptable probabilities of failure for conventional glazings is not made since there are no well defined industry standards and the actual probability of failure depends on the specific applications [5].

10.2 Recommendations for Further Work

In this thesis, several factors which affect the heat transfer processes and the stresses in vacuum glazing - and thus the probability of glazing failure - have been considered. These include the emittances of the internal surfaces, the use of edge insulation, the pillar separation and - to a limited extent - the pillar radius. Of course there are many other factors which can be considered, and the ones noted above could be further investigated. However, it was not intended that this thesis should provide a comprehensive examination of a wide range of design options. Instead, a limited set of options and conditions were considered in order to obtain typical results and to demonstrate the methods for calculating the heat transfer processes, the stresses and the probabilities of failure for vacuum glazing; these same methods can be applied to study other issues as part of any future work.

One issue which should be considered is the use of frames. Except for some of the results presented in the chapter on edge conduction, frames were not considered in this thesis. This is because it was decided that it would be best to obtain general and easily comparable results which are not specific to any particular types of frames. Furthermore, a proper consideration of frames would require a significant effort in the selection or the design of an appropriate frame for vacuum glazing. It is important to note that a frame would affect the edge stresses - and thus the total probability of failure for vacuum glazing. Any future work on frames could be combined with a re-examination of the thermal and stress implications of edge insulation by considering the more realistic situation of an insulated frame.

The understanding of the stresses in vacuum glazing would be further enhanced by simulating the atmospheric pressure induced stresses for different pillar radii. The

results could then be used to calculate the corresponding probabilities of failure which could be used to develop a more comprehensive plot of the design options.

Only the stresses due to the presence of a temperature difference across a glazing and the stresses due to atmospheric pressure were simulated in this thesis. There are other stresses which should be considered. These include the stresses due to wind, impacts and time-dependent illumination. These stresses were not considered in the present work due to the difficulty in simulating their transient nature. However, they are important in the context of the probability of glazing failure under service conditions. For example, the thermal shock due to the sudden illumination of a vacuum glazing (e.g. such as when the sun is no longer obscured by clouds) could conceivably cause glazing failure.

Most of the theoretical methods presented in this thesis have been validated by experimental results. However, no such validation has been provided for the method of calculating the total probability of glazing failure. This is because such a validation would require conducting an extensive field trial of many samples of vacuum glazing under realistic conditions. It would certainly be worthwhile to perform such a trial in order to confirm the understanding of the probability of failure and to gain some empirically based insights into how to reduce significantly this probability.

10.3 References

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Appendix 1: Properties of Glass

It was necessary to know the thermal and elastic properties of soda-lime and solder glass for the modelling work of this thesis. The modelling results are compared with measurements of samples made with Pilkington float sheet glass (a soda-lime glass), and solder glasses manufactured by Nippon Sheet Glass (NSG) and Schott Glaswerke. Thus, it is preferable to use the properties of these products. Some of the required properties for these brands of glass are available, while the other properties are assumed to be equal to the properties for similar products.

A1.1 Soda-Lime Glass

Values for the thermal conductivity of soda-lime glass quoted in the literature include $1.03 \text{ Wm}^{-1}\text{K}^{-1}$ at 21°C [1], $1.05 \text{ Wm}^{-1}\text{K}^{-1}$ [2], and $1 \text{ Wm}^{-1}\text{K}^{-1}$ [3]. Several participants of the International Energy Agency Task 18 (Advanced Glazing Materials) use a thermal conductivity of $1.0 \text{ Wm}^{-1}\text{K}^{-1}$ [4]. For the modelling work in this thesis, the thermal conductivity of glass is taken to be $1.0 \text{ Wm}^{-1}\text{K}^{-1}$ and to be independent of temperature.

For Pilkington float sheet glass, the coefficient of thermal expansion is $8.0 \times 10^{-6} ^\circ\text{C}^{-1}$, and the Young's modulus is $7.38 \times 10^{10} \text{ Pa}$ [5]. These values are close to those given for unspecified brands of soda-lime glass [6, 7, and 8]. A reference for the Poisson's ratio of the Pilkington product was not found, but references [6] and [8] specify a value of 0.21 for soda-lime glass. This was the Poisson's ratio used in the modelling work.

A1.2 Solder Glass

Solder glass is essentially soda-lime glass, but with the lime replaced by lead oxide. It is also known as lead-alkali silicate glass, or lead-silicate glass. The lead oxide acts as a fluxing agent and it lowers the melting point of the solder glass to below that of soda-lime glass [9]. Solder glass is commonly used to seal cathode ray tubes.

For this thesis, thermal measurements were performed on a glazing made with Schott Glaswerke G017-693-K3 solder glass (glazing sample 418), and measurements of temperature induced deflections and strains were performed on a glazing made with solder glass from Nippon Sheet Glass (glazing sample 467). Thus, for the comparative modelling work, the thermal conductivity must be known for both types

of solder glass, but the coefficient of thermal expansion, Young's modulus and Poisson's ratio need only be known for the NSG product. No properties were available for the Schott solder glass.

For the NSG solder glass, only the coefficient of thermal expansion is reported: $7.33 \times 10^{-6} \text{ }^{\circ}\text{C}^{-1}$ [10]. References for the thermal conductivity, Young's modulus and Poisson's ratio of other solder glasses are given in Table A1.1. Reference [8] is for an unspecified lead-silicate glass used in electrical applications, and reference [11] is for Schott Glass type 8095, which is a solder glass used for sealing cathode ray tubes.

Reference	Thermal conductivity ($\text{Wm}^{-1}\text{K}^{-1}$)	Young's modulus (10^{10} Pa)	Poisson's ratio
[8]		6.1	0.21
[11]	0.91	6.1	0.22

Table A1.1: Properties of solder glass

The thermal conductivities of the Schott G017-693-K3 and NSG solder glasses are probably similar to the thermal conductivity of the Schott product of reference [11] - which is close to the thermal conductivity of soda-lime glass. Therefore, the thermal resistance of the thin and narrow solder glass edge seal can be expected to be much less than the thermal resistances of the adjacent components of the model: the two glass sheets, and the evacuated space. Thus, the exact choice of thermal conductivity for the solder glass will not significantly affect the thermal modelling solution. It is therefore a sufficient approximation to assume that the thermal conductivities of the Schott G017-693-K3 and NSG solder glasses are equal to the thermal conductivity of soda-lime glass: $1.0 \text{ Wm}^{-1}\text{K}^{-1}$.

Due to the relatively small dimensions of the modelled edge seal, and due to the fact that the structural properties of solder glass are so close to those of the soda-lime glass elements above and below the seal, the structural modelling solution is not strongly dependent on the exact choices for the solder glass structural properties. Thus, for the solder glass manufactured by NSG, the Young's modulus and Poisson's ratio are taken to be the averages of the values given in Table A1.1: $6.1 \times 10^{10} \text{ Pa}$, and 0.22.

A1.3 Summary of Properties Used in Models

Table A1.2 gives a summary of the thermal and elastic properties for soda-lime and solder glass that were used in the modelling work of this thesis.

Property	Soda-lime glass	Solder glass
Thermal conductivity (Wm ⁻¹ K ⁻¹)	1.0	1.0
Coefficient of thermal expansion (10 ⁻⁶ °C ⁻¹)	8.0	7.33
Young's modulus (10 ¹⁰ Pa)	7.38	6.1
Poisson's ratio	0.21	0.22

Table A1.2: The properties used in modelling work of this thesis.

A1.4 References

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Appendix 2: The Thermal Resistance of The Epoxy Resin and Glass

The following simple analytic model for determining the thermal resistance of the glass and epoxy resin below and between the metering piece and the guard of the TCA, $R_{m-g}(glass+epoxy)$, provides a resistance which is approximately consistent of the more exact finite element modelling result given in Section 2.2.3. The two series components of this thermal resistance, $R_{m-g}(glass)$ and $R_{m-g}(epoxy)$, are considered separately.

The glass component of the glass and epoxy resin thermal resistance, $R_{m-g}(glass)$, can be estimated in the following manner. As shown in Fig. A2.1, the heat flow through the glass from the metering piece at temperature T_m to the guard at temperature T_g can be thought of as occurring through a series of annular rings of an *average* radius (measured from the center of the metering piece) that is equal to the mid-gap radius, r_{gap} . The cross-section of each ring resembles a hemispherical groove in the glass, of radius r and of thickness dr . The radius r is measured from the mid-point of the gap and it varies from one half the gap width at the glass, $g/2$, to the total thickness of the measurement side glass below the metering piece and guard, t . A single annular ring can be straightened to give an annulus of height πr , thickness dr , and circumference $2\pi r_{gap}$, as shown in Fig. A2.2. The total heat flow through all of these rings of glass is:

$$Q = \int_{g/2}^t \frac{k_{glass} 2\pi r_{gap} (T_m - T_g) dr}{\pi r} \quad (A2.1)$$

$$= 2k_{glass} r_{gap} (T_m - T_g) \ln\left(\frac{t}{g/2}\right),$$

where k_{glass} is the thermal conductivity of the glass. Thus, the thermal resistance of the glass is:

$$R_{m-g}(glass) = \frac{1}{2k_{glass} r_{gap} \ln\left(\frac{t}{g/2}\right)} \quad (A2.2)$$

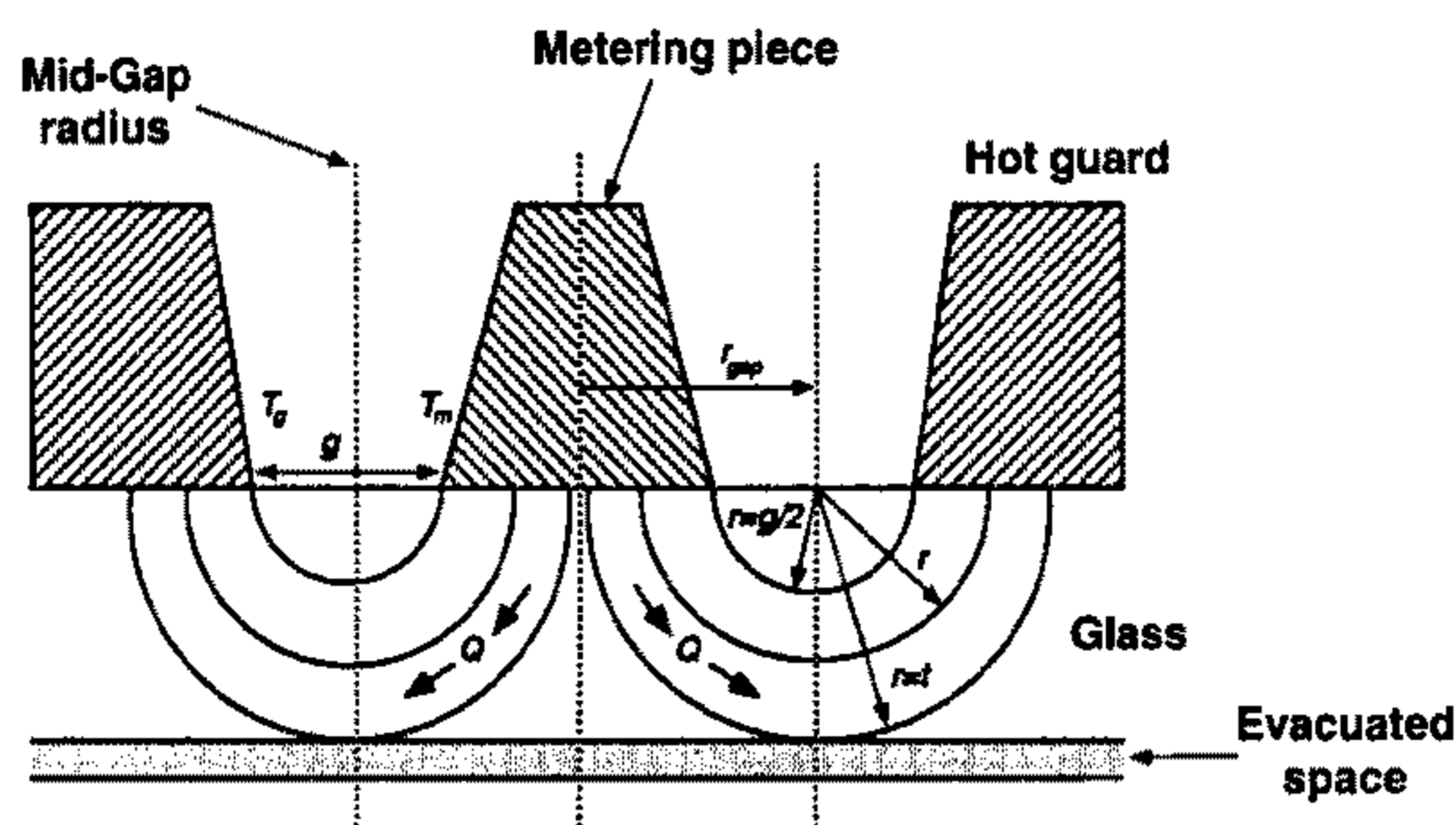


Fig. A2.1: An analytic model for determining the thermal resistance between the metering piece and the guard through the glass, $R_{m-g}(glass)$. A heat flow Q is assumed to flow between the metering piece and the guard through annular rings of average radius r_{gap} (measured from the center of the metering piece), the cross-section of which resemble hemispherical grooves in the glass of radius r and thickness dr (the dimension dr is not shown in this figure).

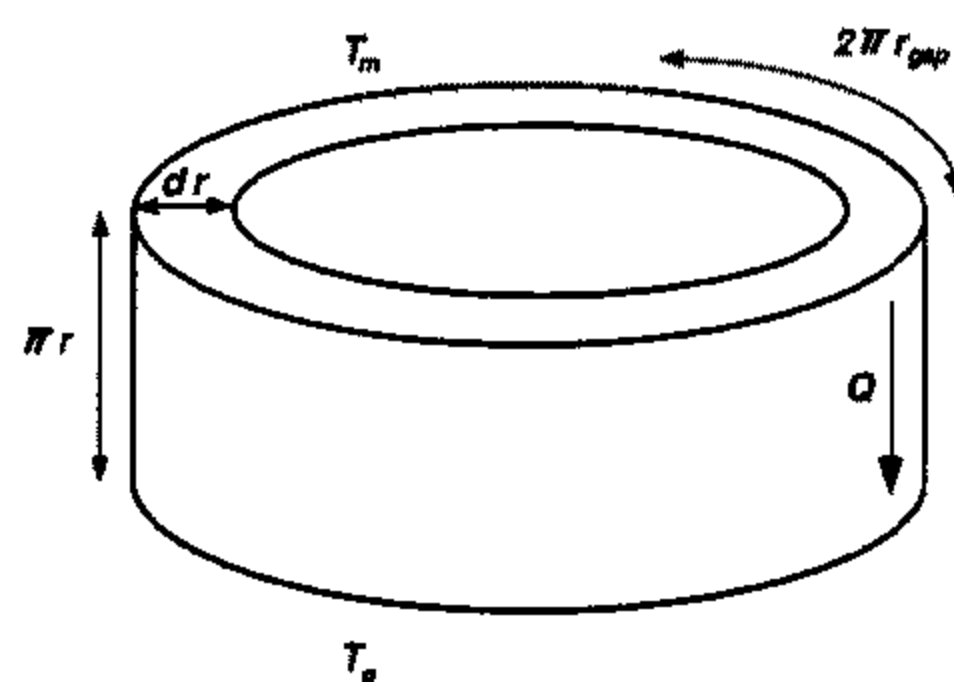


Fig. A2.2: Each annular ring shown in Fig. A2.1 can be straightened into an annulus of height πr , thickness dr , and circumference $2\pi r_{gap}$.

Substituting into equation A2.2, $k_{glass}=1.0 \text{ Wm}^{-1}\text{K}^{-1}$ (from Appendix 1), the TCA dimensions of $r_{gap}=7 \text{ mm}$, $g=1 \text{ mm}$, and the typical total measurement side glass thickness of $t=6 \text{ mm}$ gives a thermal resistance of 28.7 KW^{-1} .

The thermal conductance of the epoxy resin, $C_{m-g}(epoxy)$, is approximately equal to the total thickness of the epoxy resin layers beneath the metering piece and the guard ($0.11 \text{ mm} + 0.10 \text{ mm}$) divided by the thermal conductivity of the resin ($0.20 \text{ Wm}^{-1}\text{K}^{-1}$). This gives $C_{m-g}(epoxy)=952 \text{ Wm}^{-2}\text{K}^{-1}$. The corresponding thermal resistance is approximately equal to the inverse of the product of this thermal conductance and the area through which the heat flows between the metering piece to the guard. According to the simplistic representation of this heat flow that is shown in Fig. A2.1, this area is equal to the geometric area of the metering piece ($1.54 \times 10^{-4} \text{ m}^2$). Thus, $R_{m-g}(epoxy)=6.8 \text{ KW}^{-1}$.

The thermal resistance of the glass and epoxy resin below and between the metering piece and the guard is equal to the series combination of the above values for $R_{m-g}(glass)$ and $R_{m-g}(epoxy)$ - i.e. $R_{m-g}(glass+epoxy) = 35.5 \text{ KW}^{-1}$. This is approximately consistent with the more exact finite element modelling result of 34.9 KW^{-1} that is given in Section 2.2.3. The above analysis shows that the largest contribution to $R_{m-g}(glass+epoxy)$ comes from the glass component of this thermal resistance - i.e. $R_{m-g}(glass)$.

Appendix 3: Pillar Thermal Resistance

For a system with only two defined temperatures, the absolute heat flow, Q , can be expressed as:

$$Q = kS\Delta T_{total} , \quad (A3.1)$$

where k is the thermal conductivity of the medium in which the heat flows, S is the heat conduction shape factor, and ΔT_{total} is the difference between the defined temperatures [1]. Holman [2], gives the shape factor for the geometry shown in Fig. A3.1: a thin disk of radius a is parallel to, and at a distance D from, the isothermal surface of a semi-infinite medium of thermal conductivity k . The disk is at a uniform temperature of T_o , and the medium is at a temperature of T at large distances from the disk. For the case of $D=0$, S is given by:

$$S = 4a . \quad (A3.2)$$

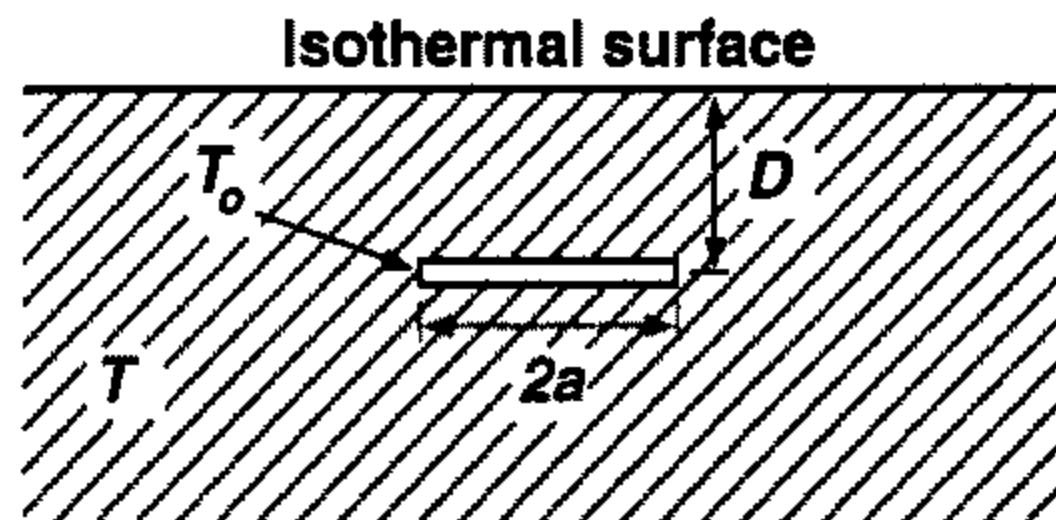


Fig. A3.1: A thin horizontal disk of radius a is parallel to, and at a distance D from, the isothermal surface of a semi-infinite medium. The disk is at a uniform temperature of T_o , and the medium is at a temperature of T at large distances from the disk.

Figure A3.2 shows two structures which are in contact with each other along their isothermal surfaces. Each structure is like the one shown in Fig. A3.1 but for $D=0$. For each structure, the difference between the temperature of the disk and the temperature of the medium at large distances from the disk is equal in magnitude but opposite in sign to that of the other structure. Therefore, the spreading heat flows in the structures are symmetric but in opposite directions with respect to the disks. Although the structures are actually thermally isolated by the isothermal surfaces, the symmetric heat flows into and out of the disks can be thought of as representing a "continuous" heat flow from one medium to the other via the disks (in fact, due to the symmetry of the

heat flows, the surfaces of contact would still be isothermal if the disks were thermally connected). This "continuous" heat flow, Q , is equal in magnitude to that in either structure but it corresponds to twice the temperature difference: that between the two media, ΔT_{total} (which is equal to $2T$). Therefore, according to equation A3.1, the shape factor for the "continuous" heat flow is one half of that given by equation A3.2 - i.e. $S=2a$.

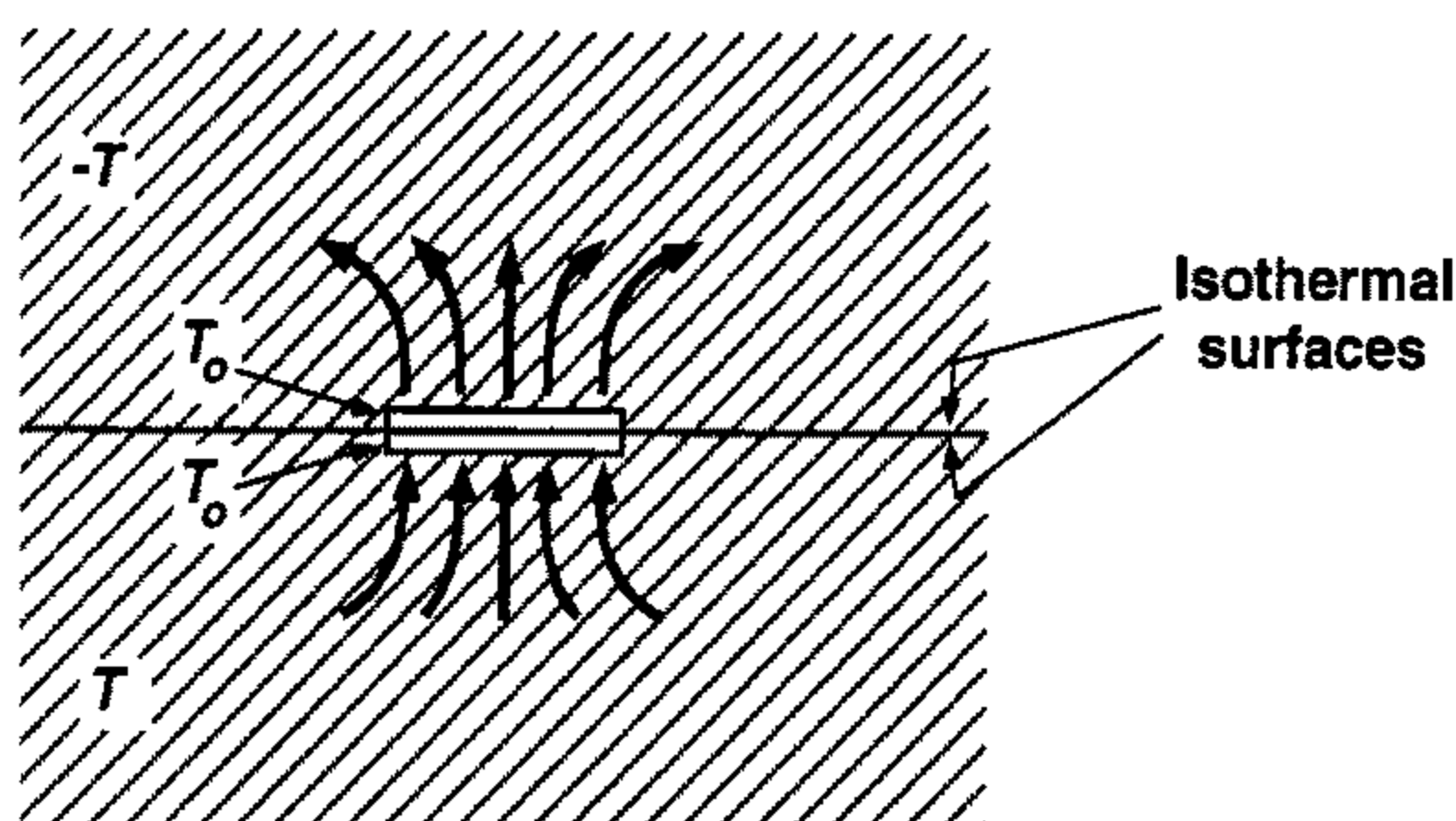


Fig. A3.2: Two structures like the one shown in Fig. A3.1, but for $D=0$, placed in contact along their isothermal surfaces. The disk temperatures are both T_0 . The medium temperatures at large distances from the disks are equal in magnitude but are opposite in sign (T and $-T$). Thus, the heat flows in the structures are symmetric but are in opposite directions with respect to the disks. Simplified heat flows are shown.

In a vacuum glazing, the dimensions of the cylindrical pillars are much less than those of the glass sheet, and thus the pillar can be assumed to be a thin horizontal disk and the glass sheet can be assumed to be a semi-infinite medium. Since the glass sheets are separated by a vacuum, they are essentially thermally well isolated - except at the edges (which need not be considered in this analysis), and at the pillars. Thus, the internal glass surfaces around the pillars can be assumed to be isothermal. As discussed in Section 4.4.1, the temperature variations across the external glass surfaces are only a few percent of the total glass-to-glass temperature difference (for the realistic ASTM standard fenestration test conditions). Thus, the external glass surface temperatures can be assumed to be approximately uniform. Since the glass sheets are considered to be semi-infinite media, these uniform temperatures are essentially at large distances from the pillar. The difference between the external glass surface temperatures, ΔT_{g-g} , can therefore be assumed to be analogous to the difference in the medium temperatures, ΔT_{total} , that was considered above. As a result of all of the above assumptions, the heat flow through a cylindrical pillar of radius a between two sheets

of glass (as shown in Fig. A3.3a below) is analogous to the "continuous" heat flow through the disks of radii a in the arrangement shown in Fig. A3.2. Therefore, the shape factor for the heat flow through the pillar is $S=2a$. Substituting this shape factor into equation A3.1, and noting that the thermal resistance is equal to the total (i.e. glass-to-glass) temperature difference divided by the heat flow ($\Delta T_{g-g}/Q$), the following is obtained for the thermal resistance of a pillar-glass system with a cylindrical pillar:

$$R_{pillar\ cyl.} = \frac{1}{2ka} . \quad (A3.3)$$

Occasionally, pillars of square cross-section are used in vacuum glazing. According to an expression given by Holman [2], the shape factor for the case of a thin square plate of dimension w at the isothermal surface of a semi-infinite medium (i.e. the situation considered in Fig. A3.1 but for a square plate at $D=0$) is:

$$S = \frac{\pi w}{\ln(4)} . \quad (A3.4)$$

By the same method outlined above for determining the thermal resistance for a pillar-glass system with a pillar of circular cross-section, this shape factor can be used to obtain the following thermal resistance for a system with a pillar of square cross-section:

$$R_{pillar\ sqr.} = \frac{2 \ln(4)}{k\pi w} . \quad (A3.5)$$

For cylindrical and square pillars of equal cross-sectional areas, $w^2 = \pi a^2$ and thus:

$$\begin{aligned} \frac{R_{pillar\ cyl.}}{R_{pillar\ sqr.}} &= \frac{\pi^{3/2}}{4 \ln(4)} \\ &= 1.003 . \end{aligned} \quad (A3.6)$$

Therefore, the thermal resistances for pillars of equal circular and square cross-sectional areas differ by only 0.3 %.

The derivation of the shape factor for the heat flow through a pillar-glass system is based on the assumption that, as stated above, this flow is analogous to the "continuous" heat flow between the two semi-infinite media of Fig. A3.2. Whereas the

latter flow is symmetric in each medium, the heat flow in each glass sheet is usually asymmetric. This is because, under practical conditions for a vacuum glazing, the external heat transfer coefficients are not equal and thus there is an asymmetric spreading of the heat flow in the sheets. Due to the relative thermal conductivities of the pillar and glass, most of the temperature drop in each glass sheet occurs in the immediate vicinity of the pillar. Thus, in this region, the flows in the sheets spread out very little relative to the glass thickness and are therefore approximately symmetric (as indicated by the shapes of the isotherms in Fig. 4.8 of Chapter 4). Since it is the heat flow close to the pillar that makes the largest contribution to the total (spherically integrated) flow in each sheet, it may be assumed that the flow through the pillar-glass system is essentially symmetric about the pillar, and thus that it is analogous to the "continuous" heat flow of the ideal system.

It is important to note that the above analytic approach for determining the thermal resistance of a pillar-glass system is based on the assumption that the glass sheets are of semi-infinite thickness. This is a good approximation if the pillar radius is much less than the glass thickness (i.e. if $a \ll t$). In an actual glazing, the relative dimensions are such that the finite glass thickness does affect the pillar thermal resistance - but only by a small amount. The effect of a finite glass thickness can be understood as follows. If semi-infinite glass sheets (represented by the media of Fig. A3.2) are truncated in thickness, the heat flow in each sheet passes through less glass and thus the thermal resistance for the system is reduced. Since most of the temperature drop in each sheet occurs in the immediate vicinity of the pillar (as mentioned above) instead of at large distances from the pillar, the imposition of a finite thickness results in only a small reduction in the difference between the external glass surface temperatures (ΔT_{g-g}) - and thus in the thermal resistance.

The effect of a finite glass thickness on the pillar thermal resistance varies with the pillar radius because, as the pillar diameter becomes somewhat greater than the finite glass thickness (i.e. $a > t$), the nature of the heat flow in the glass changes from being a mostly spreading flow within the sheets (as considered above) to being a mostly parallel flow between the sheets. These flows are shown in Figs. A3.3a and b, respectively. The thermal resistance for the former flow is given by equation A3.3, and that for the latter flow is approximately equal to the series combination of the thermal resistances of the two glass sheets over the area of the pillar:

$$R_{parallel, glass} = \frac{2t}{k_{glass} \pi a^2}, \quad (A3.7)$$

where t is the glass sheet thickness, and the thermal resistance of the pillar is assumed to be negligible because the pillar is much thinner and more conductive than the glass sheets. It can be assumed that both the spreading and parallel heat flows are present together and thus the total thermal resistance is equal to the parallel combination of the thermal resistances given by equations A3.3 and A3.7. The *inverses* of these two resistances, and the *inverse* of the total resistance, are presented in Fig. A3.4 as functions of a/t for $t = 4$ mm and $k_{\text{glass}} = 1 \text{ Wm}^{-1}\text{K}^{-1}$. As expected, for $a \ll t$, the total thermal resistance is slightly less than, but can be approximated by, the relationship for the spreading heat flow. However, with increasing pillar radius, the parallel heat flow becomes significant and the total resistance becomes increasingly less than that predicted by equation A3.3. At values of a/t greater than those shown in this figure, the parallel heat flow is the dominant flow and thus the total thermal resistance can be approximated by equation A3.7. Therefore, the effect of a finite and fixed glass thickness on the pillar thermal resistance varies with the pillar radius. This is clearly demonstrated by the results of Fig. 4.5 of Chapter 4; these results are for a glass thickness of 4 mm and a pillar radius of between 0 and 0.35 mm - and thus for values of a/t between 0 and about 0.09. In this figure, there is only a slight variation in the total (i.e. pillar) thermal resistance with the pillar radius. This is consistent with the results of Fig. A3.4 for the same range of a/t .

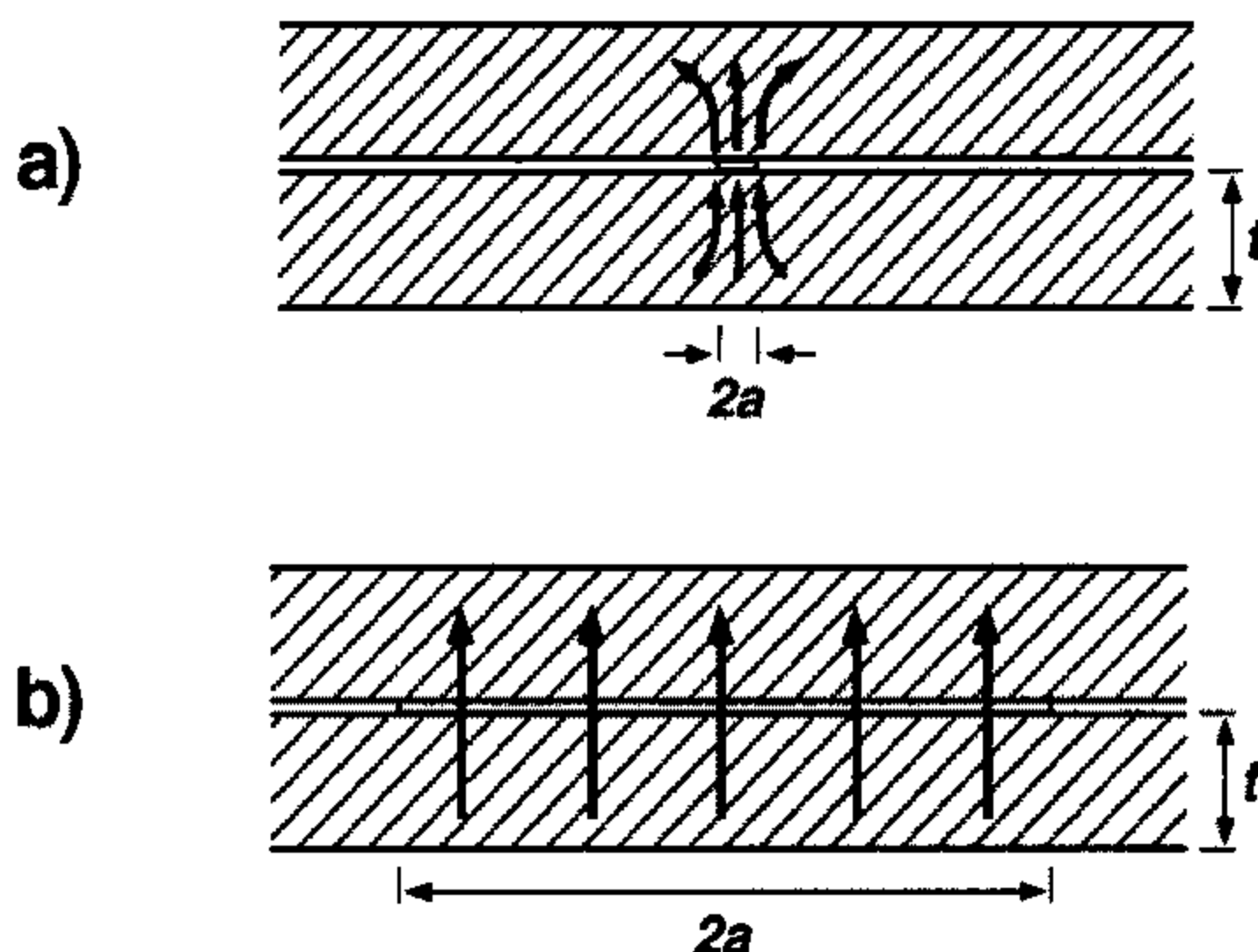


Fig. A3.3: (a) For a pillar with a radius, a , which is much less than the glass sheet thickness, t (i.e. $a \ll t$), the heat flow is mostly a spreading flow within the glass sheets. (b) For a pillar with a radius which is somewhat greater than the glass sheet thickness (i.e. $a > t$), the heat flow is mostly a parallel flow between the glass sheets. Simplified heat flows are shown.

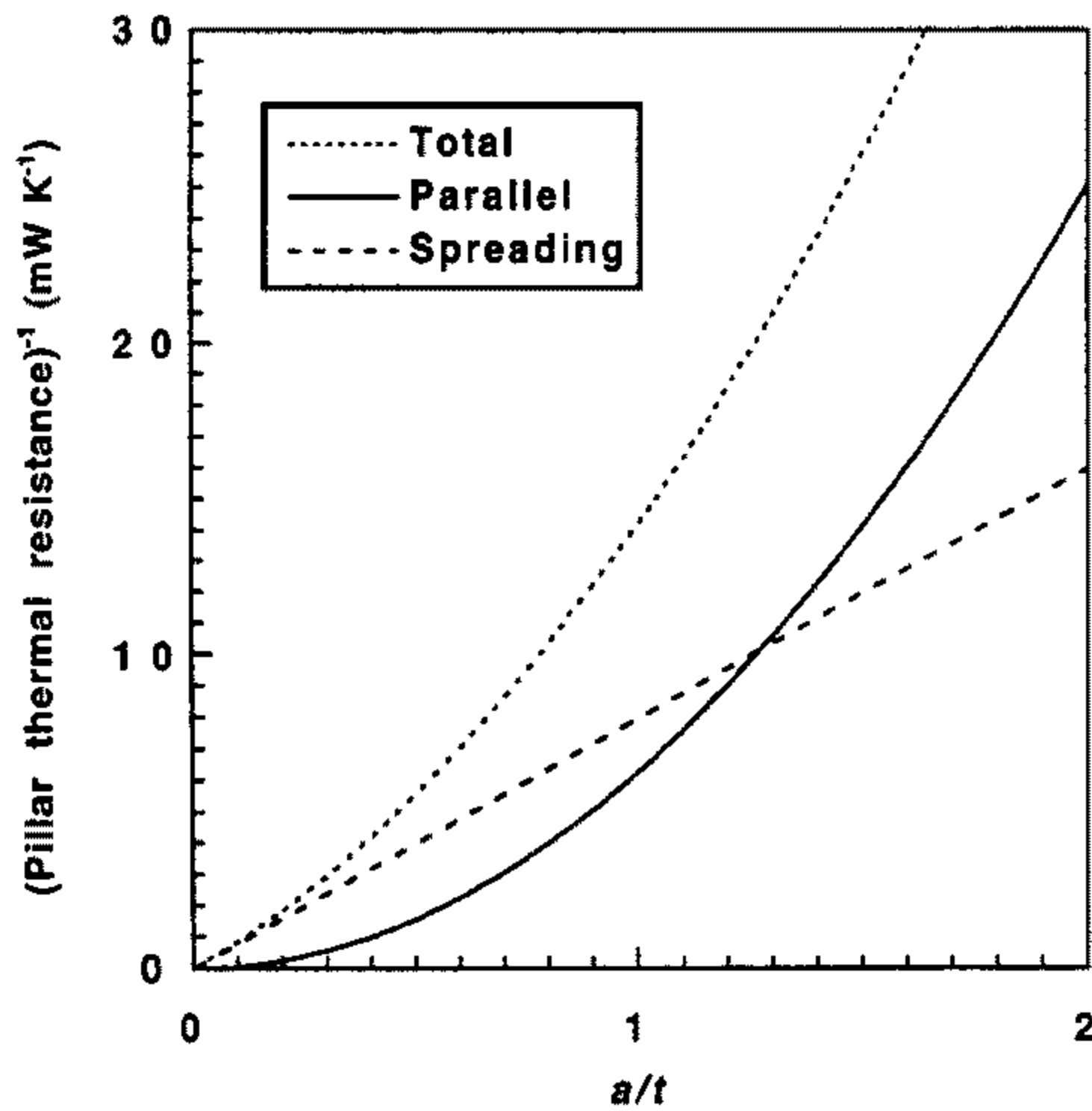


Fig. A3.4: The variation of the inverses of three thermal resistances with the ratio of the pillar radius to the glass thickness (a/t) for $t=4$ mm and $k_{glass}=1.0 \text{ Wm}^{-1}\text{K}^{-1}$. The results for the parallel heat flow (Parallel) are calculated by equation A3.7, and those for the spreading heat flow (Spreading) are calculated by equation A3.3. The sum of these results is equal to the inverse of the total thermal resistance (Total).

The analytic approach for determining the thermal resistance of the pillar-glass system is also based on the assumption that the glass sheets are of semi-infinite extent. This is justified on the basis that, as shown by the temperature profiles in Figs. 4.9 of Chapter 4, most of the effects of the heat flow through the system occur within a short radial distance from the pillar (about 25 mm, for the realistic conditions considered in these figures).

References

1. J.P. Holman, *Heat Transfer* (SI Metric Edn), p. 77. McGraw-Hill Book Company, Singapore (1989).
2. Holman, p.80.

Appendix 4:

Stresses at Thin Shell Plate and Beam Element Connections

The situation of interest is shown in Fig. A4.1: a pressure, P , is applied normally to two-dimensional thin shell plates, and the resulting load is supported by a one-dimensional beam element which is below and perpendicular to the plates. The connection between the plate and beam elements occurs at a common node, and thus the reaction force associated with the beam acts only on this node. Therefore, the stress corresponding to this force is a function of the area of the node. Since nodes are defined to be infinitesimally small, this stress should be infinite. However, due to the system stiffnesses, there is in fact a finite, but unrealistically large, stress at this node. Thus, a profile of the stress over many plates around the beam exhibits a singularity right above the beam (i.e. at this connecting node) [1].

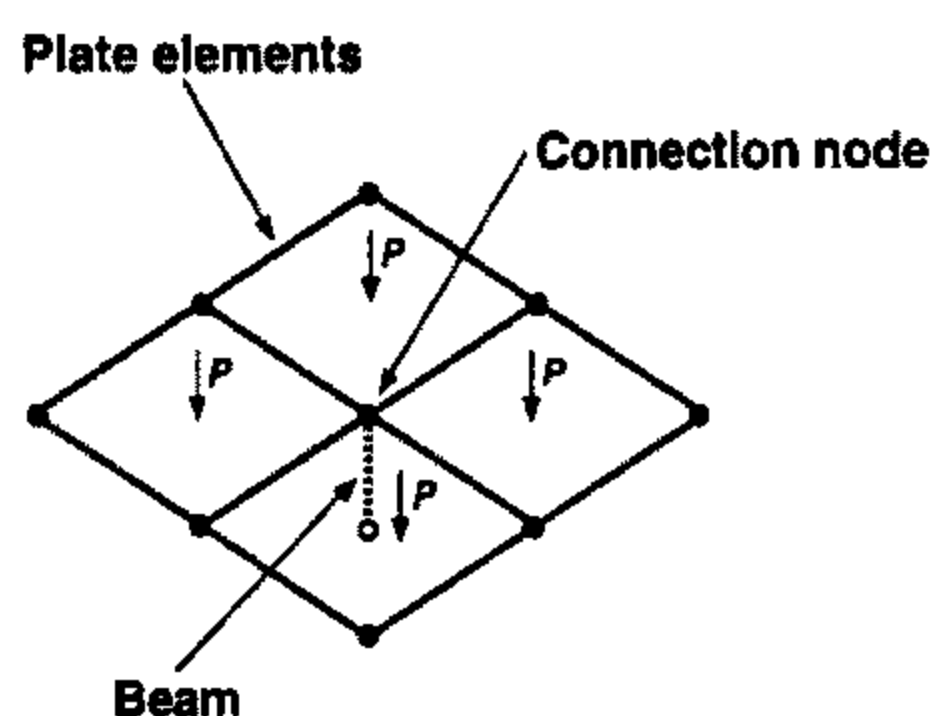


Fig. A4.1: A pressure, P , is applied to thin shell plate elements supported from below by a beam element. The plates and the beam are connected at a common node.

The large stress above the beam varies with the difference between the beam and plate stiffnesses [1]. The latter stiffness depends on three parameters of the beam: the Young's modulus, the cross-sectional area, and the height. As shown by Equation 7.2 of Chapter 7, the beam stiffness coefficient varies with the first two parameters, and it varies inversely with the third one. It is important to note that the cross-sectional area is only used for determining the beam stiffness and that it does not refer to the area of contact between the plate and beam elements (this contact area is defined by the above mentioned area of the connecting node).

Refining the node mesh around the beam by the subdivision of the plate elements does not reduce the magnitude of the stress above the beam. In fact, the opposite occurs. This is because, in solving for the nodal stresses, the stresses at the nodes of each

element are averaged over the dimensions of the element. As shown in Fig. A4.2, the subdivision of an element which is adjacent to the beam (e.g. the element which is defined by nodes 1, 2, 3 and 4) results in several smaller elements - one of which is adjacent to the beam (in this example, the element defined by nodes 1', 2', 3' and 4). Compared with the corner nodes of the larger element, the nodes of this smaller beam adjacent element are closer together. Also, the corner nodes not in contact with the beam are closer to the node at which the unrealistically large stress is calculated (node 4, which connects the plate and beam elements). Therefore, the averaging of the stresses at the four corner nodes of the smaller element over the reduced elemental dimensions results in larger stresses at all of these nodes. This exacerbates the singularity in the stress profile.

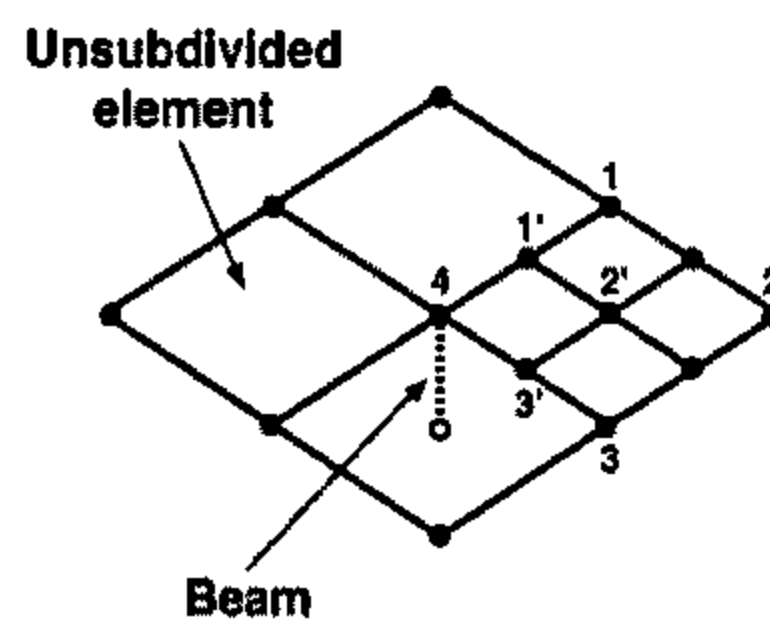


Fig. A4.2: The subdivision of an element adjacent to the beam (e.g. the one defined by nodes 1, 2, 3 and 4) results in smaller elements (e.g. the element defined by nodes 1', 2', 3' and 4, which is also adjacent to the beam). Compared with the corner nodes of the larger element, those of the latter element are closer together - and closer to the unrealistically large stress which is calculated at the plate-beam connecting node (node 4).

Reference

1. G. Carè, of G+D Computing, Sydney, Australia, Private communication.

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