NATURAL CONVECTION SUPPRESSION IN SOLAR COLLECTORS
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PROEFSCHRIFT

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He said it is as important to learn the important questions as it is the important answers.
It is especially important to learn the questions to which there may not be good answers.

Chaim Potok
uit "In the beginning"

aan mijn moeder en oom Eddy voor Ineke
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CHAPTER 1

GENERAL INTRODUCTION

1.1 Introduction and motivation.

During the last decades, methods have been developed for using the sun as an energy source. In the 1960's solar cells were developed to convert solar radiation directly into electrical energy for application in spacecrafts. Nowadays hundreds of satellites and in the future, undoubtedly, complete space laboratories are, and shall be, dependent on solar energy. However it lasted up to the energy crisis of 1973/74 before large research and development programmes were set up for the use of this energy source for application in space heating and industry.

Within the European Community a programme started to utilise solar energy for active and passive heating of buildings. In the Netherlands solar research programmes were initiated by the "Nationaal Onderzoek Zonneenergie". During these years a major part of the research programme has been the development of an active solar system for the supply of domestic hot water and house heating. A major component of such a solar system is the solar collector which converts the solar radiation into heat. The heat is transferred to a heat storage by means of a closed circuit by the so-called heat transporting fluid. It is clear that the efficiency of the solar collector determines the maximum efficiency of the solar system and it is therefore extremely important to use collectors with the highest possible efficiency.

However the collector efficiency is dependent on the collector's working temperature. For example, for domestic hot water supplies a storage tank temperature of about 60°C will be convenient, whereas for heating purposes the working temperature should be about 60°C to 80°C. The efficiency of solar collectors decreases with the increase of the working temperature due to heat losses to the surroundings. Usually the heat loss of solar collectors is expressed in an overall heat loss factor, which is a measure of the total heat loss by thermal radiation, convection and conduction.

For a spectral selective flat plate solar collector, with an already signifi-
cantly reduced thermal radiation heat loss, the total heat loss factor lies in the range of 4 to 5 W/m²K. For spectral selective flat plate collectors this results in efficiency values of only 0.10 to 0.20 at the high temperature range. The resulting heat losses for spectral selective flat plate collectors are mainly due to natural convective heat transfer between the absorber and cover plate of the collector. It is our aim to decrease the heat loss factor of flat plate solar collectors by reducing this natural convective heat loss.

### 1.2 Qualitative formulation of the problem

A solar collector consists of an absorber plate, placed in a box on a layer of thermal insulation material, with as cover, a transparent plate (usually made of glass). The so-called heat transporting fluid flows through the absorber and transports the heat to a heat storage (usually a watertank). Incoming solar radiation heats the absorber plate to a temperature above the ambient temperature. Due to the temperature difference with the environment, heat losses occur from the absorber plate to the environment by different mechanisms. The heat losses from the absorber can be divided into back and top heat losses. To the back (and the sides) the heat losses occur by thermal conduction through the insulation.

To the top the heat losses are a result of heat transfer from the absorber to the cover, and from the cover to the surroundings. Heat is transferred to the cover by thermal radiation and by natural convection. Natural convection is initiated due to the density differences of air in the enclosure, which are due to the temperature difference between the absorber and the cover. These density differences force the air to flow upwards near the (hot) absorber and to flow downwards near the (cold) cover. This circulating flow transfers heat from the absorber to the cover. The outer side of the cover is cooled by forced and natural convection of the outside air and by thermal radiation to the sky.

For most flat plate solar collectors the thermal radiative heat loss from the absorber has been reduced by application of so-called spectral selective layers. Research investigations concerning the spectral selectivity of materials have been described, for example, by Van der Leij (1979). Once radiative heat losses have been restricted, the natural convective heat loss from the absorber forms the largest heat loss. To minimize this convective heat loss it is important to know the process determining parameters. These are the temperature difference between the absorber and cover; the distance between the absorber and cover; the angle of inclination of the collector; the fluid properties and the specific geometry of the enclosure.

Basically three methods can be applied to reduce the convective heat transfer.

1. The enclosure could be evacuated so that natural convection and even air conduction is eliminated completely. Evidently vacuum collectors require special provisions for the collector housing. That is the reason why this alternative, until now, has been too expensive.

2. One can use multi-cover collectors. By using two or more cover plates the enclosure is divided into a number of air layers (each with optimum spacing), which increase the insulating value to the top of the collector and to the environment. A disadvantage of this method is that the incoming solar radiation which eventually reaches the absorber will be decreased by approximately a factor $T_{c1} \times T_{c2} \times T_{c3} \times \ldots$ where $T_{c(i)}$ represents the solar transmittance coefficients of the cover plates.

3. The air-filled enclosure can be divided into a large number of cells. Due to the reduced dimensions of each cell, in comparison to the single enclosure, the amount of viscous forces acting on the air in each cell is increased. If one dimensions the cells correctly, the development of natural convection can be shifted to larger temperature differences. This gives the opportunity to increase the plate distance and therefore to increase the insulating value of the air layer. The structures used to divide the enclosures in a large number of convecting cells are generally called Convection Suppressing Devices (abbrev. CSD's) and in special cases honeycombs.

It is obvious that CSD's should be made of thin, for solar radiation transparent material so that the loss of insolation and the conductive heat loss through the material is very small compared to the gain which is reached by the suppression of natural convection.

A large advantage of using CSD's instead of vacuum collectors is that the application of CSD's in solar collectors can be added very easily to the production process of flat plate solar collectors. For the production of vacuum collectors a completely different production process is needed.

Our investigation concerned the application of CSD's in flat plate solar collectors. This resulted in an numerical and experimental study of natural convection in very small aspect ratio enclosures, where a three dimensional model has been used to calculate natural convection. Moreover the interac-
tion of radiative, convective and conductive heat transfer in small aspect ratio enclosures has been a major scientific goal.

1.3 Outline of the investigation.

In chapter 2 of this thesis we will give a brief review of flat plate solar collectors, honeycombs and slit structures. In chapter 3 the theoretical part of our investigation shall be described. Basically two parts can be distinguished. The first part concerns the numerical method used to solve the equations which govern the three dimensional natural convection flow. In this part a comparison shall be given between the results which have been derived with a two and three dimensional natural convection model. Parameter variation has been performed to find the cell aspect ratios, which should be used for convection suppressing devices. These calculations have been performed according to the independent mode analysis, i.e. assuming that natural convection, thermal radiation and wall conduction are three heat transfer modes independent of each other.

The second part concerns the coupled radiative, convective and conductive heat transfer that occurs in a single cell of the aspect ratios selected. Parameter variation has been performed for black collectors (noted BB-case), spectral selective collectors (noted SB-case) and spectral selective collectors with a thermal radiation reflecting layer on the inner cover surface (noted SS-case).

Chapter 4 describes the experimental work. Flow visualization experiments have been performed to investigate the flow structure in slit honeycombs. Moreover heat transfer measurements have been performed for slit honeycombs considering convective heat transfer and total heat transfer as well. In addition the solar transmissivity of slit structures has been measured by means of a small solar simulator and finally experiments shall be described considering the influence of the application of slit structures on the total heat loss of flat plate solar collectors.

Our investigation has been supported by the "Foundation for Fundamental Research of Matter" in a programme of "Technical Physics". In the framework of this programme the utilization of a CSD in a practical solar collector has been a major goal of our work.

In chapter 5 the design and the manufacturing of two prototype slit honeycomb solar collectors is described. The advantages and disadvantages of the various components in the manufacturing process shall be discussed. Finally in chapter 6 the overall conclusions of the investigation shall be given.
CHAPTER 2

SOLAR COLLECTORS

2.1 Review of flat plate solar collectors

The flat plate solar collector is basically a for solar radiation black surface, mounted in a box with a transparent cover at the top and appropriate thermal insulation at the back (and sometimes at the sides as well). It is placed at a convenient angle to the daily motion of the sun (in the Netherlands approximately 45 degrees South). A large number of slightly different flat plate collector models exist but the schematic view given in fig. 2.1 represents the basic geometry.

Through the absorber flows the heat transporting fluid (water, glycol). Typical values for the thickness of the insulation layer at the back of the collector and of the distance between the absorber and cover are respectively 0.05 m and 0.015 m. Most flat plate collectors have areas of \( \approx 1.5 \) m\(^2\).

Over the years considerable effort has been taken to improve the efficiency and to increase the output temperatures of flat plate collectors. Examples of these technological developments are spectral selective layers and multi-
cover collectors. A different approach is the use of cylindrical evacuated tube collectors.

Restricting ourselves to flat plate solar collectors with single glass covers the energy balance of the collector for stationary conditions can be written as

\[
\langle E \rangle \cdot \alpha \cdot A_p \cdot Ap = Q_u + Q_1
\]  

(2.1)

here \( \langle E \rangle \) represents the rate of incident solar radiation on a unit area, \( \alpha \) and \( \alpha_p \) the transmittance of the cover and the absorptivity of the absorber both for solar radiation, \( A_p \) the aperture area of the collector, \( Q_u \) the useful heat gain and \( Q_1 \) the total heat loss due to reradiation, convection and conduction. A complete analysis of the energy balance of flat plate solar collectors has been given by Duffie and Beckmann (1989) and for stationary conditions this analysis leads to the so-called Hottel-Whillier performance equation (Hottel and Whillier (1958)).

\[
\eta = \frac{Q_u}{A_p \langle E \rangle} = \frac{1}{F_R} \left( \alpha \cdot \alpha_p - \frac{\theta - \theta_a}{L} \langle E \rangle \right)
\]  

(2.2)

\( T_i \) is the fluid inlet temperature, \( T_a \) the ambient temperature, \( F_R \) the heat removal factor of the solar collector and \( U_L \) the total heat loss factor.

Let us consider the total heat loss of the collector. The heat loss can be written as

\[
Q_1 = Q_t + Q_{re}
\]  

(2.3)

here \( Q_t \) represents the top heat losses and \( Q_{re} \) the back and side heat losses of the collector. For steady state conditions the top heat losses \( Q_t \) can be written as

\[
Q_t = Q_{rd,0} + Q_{cv,0}
\]  

(2.4a)

The indexes \( rd, cv \) refer respectively to radiation and convection from the outer cover surface. This heat loss is equal to the heat loss between the absorber to the cover plate.

\[
Q_t = Q_{rd,0} + Q_{cv,0}
\]  

(2.4b)

The different heat losses can be represented by heat transfer coefficients so that

\[
Q_{cv,0} = h_{cv,0} (T_c - \theta_a) A_p
\]  

(2.5a)

The radiative heat transfer has been calculated as radiative heat transfer between two infinite parallel flat plates.

\[
Q_{rd,0} = h_{rd,0} (T - \theta_a) A_p = \frac{1}{\frac{1}{\kappa} + 1/\kappa} \left( \frac{\theta^4 - \theta^4_a}{\kappa} \right) A_p
\]  

(2.5b)

\[
Q_{cv,0} = h_{cv,0} (T_c - \theta_a) A_p
\]  

(2.5c)

\[
Q_{rd,0} = h_{rd,0} (T - \theta_a) A_p = \frac{1}{\frac{1}{\kappa} + 1/\kappa} \left( \frac{\theta^4 - \theta^4_a}{\kappa} \right) A_p
\]  

(2.5d)

with \( T_{sg} \) as the apparent radiant sink temperature, resulting from the assumption that an inclined collector shall view both the sky and the ground at sky temperature and ambient temperature respectively (see Cooper (1981)). Because the radiative and convective heat losses are related to two different sink temperatures the analysis becomes more complicated. Moreover the back and side heat losses are not necessarily related to equal ambient temperatures \( T_a \) and the absorber and aperture area are usually not equal either. However for simplicity let us assume this.

The back and side heat losses are considered as one heat loss and can be written as

\[
Q_{re} = \frac{\lambda'_{ins}}{D_{ins}} (T_c - \theta_a) A' = h_{re, p} (T_a - \theta_a) A'_{pc}
\]  

(2.6)

with \( \lambda', D_{ins}, A'_{pc} \) as effective values for thermal conductivity, thickness and area of the back and side insulation.
Garg and Datta (1984) review different empirical relations to calculate the top heat losses. However for stationary conditions the top heat losses can be calculated from eq. (2.4) by using relations for the radiative and convective heat transfer in the spacing between the absorber and the cover. The internal convection coefficient \( h_{cv,i} \) has been given by Hollands (1976), dependent on the Rayleigh number and the inclination angle \( \phi \).

$$
\text{Nu} = \frac{1}{\lambda_{\theta}} = 1 + 1.44 \left( \frac{1700}{\text{Racos}\phi} \right) + \left( \frac{1700(\sin1.08 \phi)^{1.6}}{\text{Racos}\phi} \right)
$$

\( (2.7a) \)

where \( \left( X \right)^{\#} \) indicates that the argument inside the brackets should be taken equal to zero if the argument is negative

\( (1X)^{\#} = 0.5 (X + 1X) \).

$$
\text{Nu} = \left( 1 + 0.25 \left( \frac{\text{Racos}\phi}{0.33} \right) ; 0.039 \left( \text{Racos}\phi \right)^{\max} \right)
$$

\( (2.7b) \)

for \( 75 \leq \phi \leq 90 \)

The subscript "max" indicates that at a given Rayleigh number the maximum should be taken of one of the three quantities separated by the semicolons. \( \text{Ra} \) represents the Rayleigh number based on the absorber to cover spacing \( D \).

$$
\frac{g \beta_{\theta}(T_h - T_c) D^3}{V_{\theta} \alpha_{\theta} \text{Ra}}
$$

\( (2.8) \)

where \( g, \beta_{\theta}, V_{\theta} \), and \( \alpha_{\theta} \) represent gravity, thermal expansion dynamic viscosity and thermal diffusivity of the fluid. The range of Rayleigh numbers as it might occur for solar collectors has been given in fig. 2.2. In the figure lines of constant Rayleigh number has been given for varying plate distance and temperature difference. For plate distances of \( D \geq 0.015 \) m the Rayleigh number shall not be larger than \( \approx 10^4 \) and heat transfer by natural convection shall be nihil. The heat transfer by the air from the absorber to the cover shall be primarily by conduction. For plate distances of 0.04 m up to 0.06 m the Rayleigh number shall vary between \( 10^4 \) and \( 10^6 \). Our investigations concerning natural convection suppressing in the enclosure of solar collectors have been performed for this range of Rayleigh numbers.

The internal radiation coefficient \( h_{rd,i} \) can be calculated from eq. (2.5b) which gives in combination with \( h_{cv,i} \) and \( h_{re} \) the total heat loss coefficient \( U_L \) of the collector for stationary conditions.

$$
U_L = \frac{1}{h_{cv,i}} + \frac{1}{h_{rd,i}}
$$

\( (2.9) \)

One of the assumptions that have been made to calculate \( U_L \) is that natural convection and radiation do not interact. The top heat loss has been calculated by summing up the individual contributions. This last assumption is called the independent mode analysis (abbrev. IMA).

Despite the different sink temperatures the total heat loss coefficient is usually correlated with the temperature difference \( T_p - T_a \). In fig 2.3 an example has been given of the top heat loss coefficient for a SB- and a BB-collector. The distance \( D \) has been taken at 0.015 m and the heat loss by natural convection has been calculated according to eq. (2.7). The figure shows that for realistic values of the temperature difference (in practical operating circumstances \( AT \approx 60 K \)) the radiative heat transfer forms the major heat loss for the BB-collector, whereas for the SB-collector the convective heat loss is dominant. Therefore effective suppression of
natural convection, using convection suppressing devices, can lead to a significant increase of collector performance.

2.2 Honeycombs and slit structures

In 1929 Russian engineers appeared to have been the first to investigate the use of a transparent insulation, formed by stacked wired meshes or vertical sheets of corrugated paper, over a solar collector. The results have been reported by Veinberg (1959). Later in 1961 Francia used transparent glass cylinders to reduce the reradiation heat loss of medium and high temperature collectors. His work initiated a large number of research projects concerning cellular insulation (Hollands (1965), Perrot (1967), Buchberg et al. (1971)). In the early stage the investigations concerned basically the reduction of reradiation heat losses of the absorber, based on the idea that for large spacings the cell structure acts as thermal radiation shield. However with the application of spectral selective coatings on the absorbers, the natural convective heat loss becomes dominant.

The use of honeycomb structures for the suppression of natural convection heat losses was recognized and research was directed to this purpose (Edwards (1969)). In a paper Tabor (1969) discussed the use of cellular insulation in flat plate solar collectors. He formulated the requirements for the materials to be used as honeycombs. Such materials should have besides low costs per unit area:

1. high stability to solar radiation and to the maximum temperatures to be encountered.
2. high transparency to solar radiation or alternatively the material should have very high (specular) reflection for solar radiation.
3. high opacity for thermal radiation.

In the following years honeycomb structures made of different shapes and different materials were examined on their ability of heat loss reduction of flat plate solar collectors. Tien and Yuen (1975) investigated the radiation characteristics of square-celled honeycombs theoretically. Edwards and Field (1978) considered both the solar and infrared radiation properties of slit CSD's. An example of a so-called cross slope slit CSD can be seen in fig. 2.4. Buchberg and Edwards (1976) considered the design of cylindrical glass honeycombs and showed that non-selective black absorbers, applied with a glass honeycomb, were markedly superior to single glazed selective black collectors and double glazed non-selective black collectors. Other shapes of honeycomb structures have been investigated, for instance sinusoidal (McMurtrin and Buchberg (1981)) or hexagonal shaped honeycombs (Marshall et al. (1977)).

Because the main natural convection flow in the solar collector is directed up the absorber and down the cover plate, flow restrictions of this direction should suppress natural convection adequately (the so-called cross slope CSD, see 2.4). Based on this idea Hollands et al. (1978) and Guthrie and Charters...
(1982) applied slit structures to reduce convective heat transfer. Moreover, slit structures impede the incoming solar radiation less than cell structures.

Most of the above-mentioned experimental honeycombs have been made of glass of various thickness. For instance, the sinusoidal honeycomb had the smallest thickness, but was still \( 0.15 \times 10^{-3} \) m thick. Guthrie and Charters used \( 1.1 \times 10^{-3} \) m thick glass strips. The use of glass has the advantage of good thermal and UV stability, since most of the used plastics (see Marshall (1977)) lack the high temperature stability. However, using plastics, the honeycomb wall thickness can be easily reduced with an order of magnitude, resulting in less conduction heat losses through the material.

A more practical problem of the application of honeycombs in solar collectors has been investigated experimentally by Edwards, Arnold, and Catton (1976). They considered the effect of gaps between the honeycomb and absorber and/or cover plate to gain knowledge about the limits for designing CSD's.

All the investigators had in common that they used the so-called independent mode analysis to predict the total heat transfer. Basically, two different interpretations can be given for the independent mode analysis:

1. Natural convection and wall conduction are not affected by the radiative interaction. Both quantities can be calculated according to the equations given above. The radiative heat transfer is calculated as for the infinite parallel plates model. This is equal to the assumption that the CSD walls are completely transparent or specular reflective for thermal radiation.

2. Natural convection and wall conduction are again calculated as for point 1, but now the CSD acts as thermal radiation shield and the heat transfer by thermal radiation shall be reduced. Since the analysis described in point 1 results in a calculated heat transfer which is equal to the total heat transfer in the spacing of a non-CSD flat plate collector, we adopt this formulation as the independent mode analysis.

Buchberg et al. (1971) mentioned the heat transfer mode coupling by stating "it appears to be necessary to couple the radiation-convection-conduction modes to improve the prediction of cell wall temperature". Especially for spectral selective solar collectors, the experimental results were significantly worse than predicted, according to the independent mode analysis (IMA). The deviations have been due mostly to experimental accuracy or an insufficient radiation model. Only Hollands (1979) explained the deviations by doubting the IMA, and assumed the heat transfer modes to be coupled.

We shall see that the independent mode analysis is not correct and leads to very optimistic predictions, especially for spectral selective collectors. Therefore, coupled heat transfer has to be considered.
CHAPTER 3
NUMERICAL COMPUTATIONS ON HONEYCOMBS AND SLIT STRUCTURES

3.1 Introduction.

To calculate the total heat transfer in the enclosure, formed by the absorber, the cover and the side walls of the collector, it is usual to calculate radiative heat transfer and convective heat transfer separately. To evaluate the total heat transfer in the partitioned enclosure, it is only necessary to consider a single cell or slit of the honeycomb, due to symmetry. In Holland such a cell shall be normally inclined at an angle of 45 degrees with respect to the horizontal and shall have almost no convection inside.

In our case the natural convection flow is considered to be laminar and stationary and for the case of slit structures two dimensional flow might be expected. If a cell structure is used as the convection suppression device a three dimensional calculation of natural convection is necessary. In both cases the hot and cold walls may be considered isothermal. Calculation of the convective heat transfer is possible by solving the governing equations which are the Navier-Stokes equations, the continuity equation and the energy equation.

The radiative heat transfer must also be considered.

In a non-partitioned enclosure of a solar collector the dimensions are such that the problem can be considered as radiative heat transfer between two parallel infinite plates. The influence of the side walls is then neglected. It is evident that in a partitioned enclosure this assumption can not be true (except for the case when the side walls material is completely transparent or specular reflective for the thermal radiation, the so-called IMA).

The side walls shall normally absorb, transmit and reflect the incoming thermal radiation. In addition, each side wall itself emits thermal radiation. Therefore the calculation of the radiative heat transfer is much more compli-
ated than for parallel infinite plates. The optical properties of the side wall material and the temperature distribution of the side walls are important parameters in the heat transfer process. Moreover the radiative heat transfer is an important factor in the energy equation for the side walls. This means that, due to the very small thickness of the side walls, the temperature of the side walls shall also be determined by radiative heat transfer. As a consequence the side wall boundary condition for the natural convection flow changes. Instead of a linear temperature profile (as in the case of perfectly conducting side walls) the temperature is determined by wall conduction, thermal radiation and convective heat transfer. Due to this interaction of conduction, radiation and convection it is not possible to obtain the total heat transfer by summing up the three separately calculated heat transfer parts but. It is therefore necessary to consider the three heat transfer mechanisms simultaneously.

In this numerical investigation both approximations have been considered. The non-coupled convection calculations indicate in what range the values of the aspect ratios \( A_1 = A_2 = \frac{h}{D} \) are to be expected, to obtain efficient suppression of natural convection.

A more detailed parameter study has been performed, concerning the design of the solar collector, with the programme for the coupled heat transfer calculations.

In chapter 3.2 a description has been given for the non-coupled heat transfer analysis for both the two and three dimensional cases.

The calculation procedure for radiative heat transfer has been described in chapter 3.3 and finally chapter 3.4 concerns the coupled heat transfer analysis.

The coupled heat transfer and radiative heat transfer have been considered only in two dimensions.

3.2 Computation of natural convection.

3.2.1 Formulation of the problem.

In the non-coupled analysis convective cells have been considered as shown in fig. 3.1.

Two walls are assumed isothermal at different temperatures, \( T_h \) and \( T_c \), respectively \( (T_h > T_c) \). The distance between the isothermal walls is \( D \). The height of the enclosure is \( h \). In a flat plate solar collector, where a convection suppression device is used, \( h \) represents the distance between two horizontal partition walls. In the three dimensional case (honeycomb) the length of the enclosure (in the solar collector the distance between two vertical partition walls) is given by \( L \). For the two dimensional case the length is assumed to be much larger than \( D \) and the flow is considered to be two-dimensional. The enclosures have an angle of inclination \( \theta \), with \( \theta \) measured from the horizontal (see fig. 3.1). The natural convection flow is dependent on the angle of inclination \( \theta \), the geometry of the enclosure (represented by the aspect ratios \( A_1 = \frac{h}{D} \) and \( A_2 = \frac{L}{D} \)), the fluid properties and the boundary conditions.

In principle all the fluid properties are functions of temperature. However if the fluid (air) is considered incompressible and the flow is assumed laminar and stationary, the governing equations, after application of the Boussinesq approximation, are reduced to:

\[
\nabla \cdot \mathbf{v}' = 0
\]

\[
P \left( \nabla \cdot \mathbf{v}' \right) = \mu \nabla^2 \mathbf{v}' - \nabla p' + \rho(T) g
\]

\[
P \left( C_p \left( \rho'_g \mathbf{v}' \right) \right) = \lambda \nabla^2 T
\]
in which \( v', p' \) and \( T \) represent the fluid variables velocity, pressure and temperature respectively and \( \rho, \gamma, \rho \gamma, c_p \) and \( \lambda \) are the density, dynamic viscosity, gravity, specific heat at constant pressure and thermal conductivity, respectively. All the fluid properties are taken at the mean temperature \( T_\theta \) \( (T_\theta = (T_h + T_c)/2) \), which is denoted by the subscript \( \theta \), except for the density in the buoyancy term. The density is derived from the equation of state:

\[
\rho(T) = \rho_\theta (1 - \rho_\theta (T - T_\theta))
\]  

(3.4)

where \( \rho_\theta \) represents the coefficient of thermal expansion at \( T_\theta \). The Boussinesq approximation is frequently used in natural convection problems. Schinkel (1988) showed that the corrections, which should be made for adopting the Boussinesq approximation, are negligible in this situation. Using eq. (3.4), and subtracting the static pressure, the non-dimensional form of the equations (3.1 - 3.3) is obtained by using \( D \) and \( g \) as characteristic values for length and gravity. The characteristic values for the velocity \( (v_c) \), pressure \( (p_c) \) and temperature \( \theta \) are:

\[
v_c = \frac{g \rho_\theta (T_h - T_c) D^2}{\rho_\theta}
\]

\[
p_c = \frac{\rho_\theta g (T_h - T_c) D}{T_h - T_c}
\]

\[
\theta = \frac{T - T_c}{T_h - T_c}
\]

The non-dimensional equations for the three dimensional case are:

\[
\begin{aligned}
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} &= 0 \\
Gr(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z}) &= -\frac{2}{\rho c_p} \frac{\partial p}{\partial x} + \sin(\theta) (\theta - \theta_0)
\end{aligned}
\]

(3.6a)

\[
Gr(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z}) = -\frac{2}{\rho c_p} \frac{\partial p}{\partial x} + \sin(\theta) (\theta - \theta_0)
\]

(3.6b)

\[
Gr(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z}) = -\frac{2}{\rho c_p} \frac{\partial p}{\partial y} + \cos(\theta) (\theta - \theta_0)
\]

(3.6c)

with \( x, y, z, u, v, w, p, \theta \) are the non-dimensional variables and where \( Gr \) is the Grashof number and \( Pr \) is the Prandtl number defined by:

\[
Gr = \frac{g \rho_\theta (T_h - T_c) D^3}{\nu_\theta \rho_\theta \gamma}
\]

(3.8)

\[
Pr = \frac{v_\theta}{a_\theta}
\]

(3.9)

with \( v_\theta \) and \( a_\theta \) are the kinematic viscosity and the thermal diffusion at the reference temperature. In this investigation air is the considered fluid and the Prandtl number is assumed constant \( (Pr = 0.711) \). It is convenient to define the Rayleigh number:

\[
Ra = \frac{g \rho_\theta (T_h - T_c) D^3}{\nu_\theta \rho_\theta \gamma} = Gr \cdot Pr
\]

(3.10)

The no-slip condition for the velocities is imposed on all the walls:
The boundary conditions for the temperature are:

\[ \theta = 1 \quad \text{for} \quad y = 0 \]  
\[ \theta = 0 \quad \text{for} \quad y = 1 \]  

(3.12a)

(3.12b)

For the non-coupled case the side walls have been assumed to be perfectly conducting (abbrev. pc). This means that the temperature is defined by:

\[ \theta = 1 - y \quad \text{for} \quad x = 0 \quad \text{and} \quad x = A \]

(3.12c)

The non-dimensional equations and boundary conditions for the two dimensional case have been obtained by deleting the terms with \( w \) and \( z \) in equations (3.5 - 3.12).

The non-dimensional equations and the boundary conditions for the three dimensional case possess two symmetry properties. First the symmetry with respect to the midplane \( z = 0.5 A \), secondly the symmetry with respect to the center point \( (0.5 A, 0.5 A, 0.5 A) \).

The symmetry is defined by:

\[ u(x,y,z) = u(x, y, A - z) \]
\[ v(x,y,z) = v(x, y, A - z) \]
\[ w(x,y,z) = -w(x, y, A - z) \]
\[ p(x,y,z) = p(x, y, A - z) \]
\[ \theta(x,y,z) = \theta(x, y, A - z) \]  

(3.13)

The centro-symmetry by:

\[ u(x,y,z) = -u(A - x, 1 - y, A - z) \]
\[ v(x,y,z) = -v(A - x, 1 - y, A - z) \]
\[ w(x,y,z) = -w(A - x, 1 - y, A - z) \]
\[ p(x,y,z) = p(A - x, 1 - y, A - z) \]
\[ \theta(x,y,z) = 1 - \theta(A - x, 1 - y, A - z) \]  

(3.14)

The governing equations for the non-coupled heat transfer in the two dimensional case possess only the symmetry with respect to the center point \( (0.5 A, 0.5 A) \). These symmetry properties have been used in both cases to reduce the computational effort required to solve the governing equations. Applying these symmetry properties for the three dimensional case it is only necessary to solve the equations (3.5 - 3.7) in a fourth part of the enclosure. For the two dimensional enclosure a symmetrical half of the enclosure remains.

3.2.2 Review of literature: numerical investigations.

Buoyancy driven flows are often important and dominant in heat and mass transport processes. Examples are thermal insulation of buildings and heat and water and pollutants dispersion in estuaries. Consequently analytical and numerical investigations concerning natural convection have been frequently described in literature. In general the geometry of the specific problem can be converted into a cylindrical or rectangular enclosure with two isothermal opposite walls \( (T_h \text{ and } T_c \text{.)} \). Basically two cases can be distinguished, one with the temperature gradient aligned with the gravity field (isothermal walls horizontally, \( \phi = 0 \text{ or } \phi = 180 \) degrees) and one with the temperature gradient perpendicular with the gravity field (isothermal walls vertically, \( \phi = 90 \) degrees). Analytical progresses have been made by Batchelor (1954), Elder (1965) and Gill (1966) for enclosures with large height to width ratios \( (A_h >> 1) \). Numerical investigations for large aspect ratios have been described extensively in literature. A review of research, both experimental and numerical investigations, is given by Catton (1978). Recently Schinkel (1988) reported experimental and numerical work for natural convection flow in inclined enclosures with moderate aspect ratios.

Natural convection in enclosures with small aspect ratios has received rela-
tively little attention in literature. In 1974 an analysis of such a flow
was given by Cormack, Leal and Imberger (1974). They showed that
the flow in a small aspect ratio enclosure ($A_x \ll 1$, $\theta = 90^\circ$) is composed of two major regions: (1) in the core region a parallel
counter flow; (2) the end region with a length of order $h$, turning the paral-

cCore region

end region

Figure 3.2: Flow in a vertical small aspect ratio enclosure.

cel counter flow around by 180 degrees (see fig. 3.2)

They obtained an asymptotic result for the overall Nusselt number valid
only in the limit $A_x \to 0$.

Numerical results for a water-filled enclosure reported by Cormack, Leal
and Seinfeld (1974) showed an excellent agreement with their analysis. Experimen-
tal results reported by Imberger (1974) completed their investigation.

Catton et al. (1974) investigated numerically the heat transfer in small and
moderate aspect ratio enclosures ($0.2 \leq A_x \leq 20$) and for large Prandtl
fluids ($\text{Pr} \to \infty$). They found with decreasing aspect ratio an increase in
heat transfer up to $A_x = 1$. For $A_x < 1$ they found a significant decrease of
heat transfer. No heat transfer relations have been given by Catton (1974).

Bejan and Tien (1978) considered the heat transfer problem for small but fini-
ite $A_x$ analytically. They developed an approximation theory for predicting
the net heat transfer between the two ends of the enclosure.

Three models have been distinguished:

1. the regime of vanishing Rayleigh numbers ($\text{Ra} \to 0$)
2. the intermediate regime
3. the boundary layer regime

The three regimes have been given in fig. 3.3.

The solid lines indicate the place of this investigation. The theory of
Bejan and Tien is only applicable for vertical enclosures ($\theta = 90^\circ$) and lam-
nar natural convection.

The three models of heat transfer have been described by one general expres-
sion for the calculation of the Nusselt number.

\[
\text{Nu}_0 = 1 + \left( \frac{(Ra A_x^4)}{362880} \right) - n \frac{\text{Ra} \text{A}_x^{-0.4}}{362880} \right)^{1/n}
\]

with $n = -0.386$.

A review of analytical work on natural convection in rectangular vertical
enclosures is given by Bejan (1988). A summarized result, relating the Nus-
sett number with the aspect ratio $A_x$ for different Rayleigh numbers, is
given in fig. 3.4. It can be seen that for very small aspect ratios ($A_x \ll 1$) the Nusselt number reaches the limit $\text{Nu} = 1$. This means that the heat
transfer is due to conduction only and convective heat transfer does not
occur. Again the results are restricted for the cases where the isothermal
walls are vertical ($\theta = 90^\circ$).

A few numerical investigations have been performed for enclosures with
aspect ratio less than 1. As already mentioned Cormack, Leal and Seinfeld
(1974) investigated a water filled cavity with adiabatic side walls. They
assumed the flow to be two dimensional for the parameter range considered:
$10^3 \leq \text{Gr} \leq 2 \times 10^7$ and $0.85 \leq A_x \leq 1$.

Catton et al. (1974) investigated numerically the heat transfer for two
dimensional enclosures of various aspect ratios but derived no heat transfer
relations. Said and Trupp (1979) investigated at which aspect ratio a maxi-
Figure 3.4: Dependence of Nusselt on $A_x$ for different Rayleigh numbers, according to Bejan (1980).

...imum heat transfer by natural convection occurs. Therefore they analysed gas-filled vertical enclosures with aspect ratio varying from 0.5 to 5. Other two dimensional numerical investigations for natural convection have been performed by Lee and Sernas (1980), Inaba et al. (1981), Shiralkar and Tien (1981), Cho Chung and Trefethen (1982).

A summary of the parameter range of these investigations of the above mentioned authors is given in table 3.1.

Except for the investigation of Catton ($60 < A_x < 1165$) all the investigations have been performed for vertical enclosures ($\theta = 90$). Several authors, for instance Catton and Cormack, use different definitions for the Rayleigh number. In our investigation the Rayleigh number is based on the distance between the two isothermal walls, defined by $D$. To compare the different investigations it is convenient to define the same Rayleigh number. Therefore the ranges of Rayleigh numbers given in the table for the above mentioned authors have been converted to Rayleigh numbers based on $D$, for aspect ratio $A_x = 0.1$. As shown in table 3.1 almost all the investigations concerned adiabatic (ad) enclosures and relatively high Rayleigh numbers. This is not surprising if one bears in mind that they have been primarily concerned with natural convective flow instead of suppression of natural convection. For instance Cho Chung and Trefethen (1982) considered a vertical stack of inclined cavities of small aspect ratio, which conducts heat upwards but not downwards. This property is due to convective heat transport in the cavities at relatively high Rayleigh numbers. However the parameter ranges taken by Inaba, Shiralkar and Tien and Lee and Sernas show agreement with the parameter range considered in our investigation. Inaba obtained for enclosures with adiabatic side walls and for Prandtl numbers between 1 and $10^3$ the following heat transfer relations:

$$\text{Nu}_0 = 0.223 \text{Pr}^{0.624} \text{Ra}^{0.25} A_x^{-0.66} \quad 0.2 < A_x < 1 \quad (3.16a)$$

$$\text{Nu}_0 = 0.280 \text{Pr}^{0.824} \text{Ra}^{0.25} A_x^{-2.25} \quad 0.03 < A_x < 0.2 \quad (3.16b)$$

A comparison between these heat transfer results is shown in fig. 3.5. Shiralkar and Tien obtained for $A_x = 0.1$ heat transfer results, which have been given also in fig. 3.5 together with the results of Lee and Sernas for adiabatic side walls. The figure shows the large deviation between the results of Inaba and the other authors. The deviations might occur from the...
differences in the Prandtl number, but more likely are the deviations due to the different numerical schemes used. Inaba uses an upwind finite difference method. Shiralkar and Tien use an exponential differencing scheme and Lee and Sernas a central difference scheme. The results for $A_x = 0.1$ of Lee and Shiralkar are in good agreement with each other.

Wirtz and Tseng (1979) report numerical results for inclined low aspect ratio enclosures. They considered an enclosure with aspect ratio 0.5 and varied the angle of inclination from 0 (heated from below) to 180 degrees (heated from above). They considered different side wall boundary conditions, i.e. adiabatic (ad), perfectly conducting (pc) and isothermal (is) side walls. Most of their calculations have been performed for $Pr = 6.32$. For $\theta = 90^\circ$ they obtained a heat transfer relation valid for $A_x = 0.5$, $0.1 \leq Pr \leq 100$, $10^5 \leq Ra \leq 10^7$ and adiabatic side walls:

$$Nu_\theta = 0.082 \frac{Pr}{0.345} (Ra)^{0.2} + Pr$$

In fig. 3.5 the results from this relation are shown for $Pr = 0.7$. The results agree well with the results of Lee for $A_x = 0.4$. Again the results of Inaba compare poorly. For $Pr = 6.32$ and adiabatic side walls Wirtz obtained the coefficients to correct equation (3.17) for other angles of inclination according to a power law. However no coefficients have been given for $Pr = 0.7$.

Even less numerical investigations have been reported on three dimensional natural convective heat transfer in enclosures. One of the few three dimensional numerical calculations have been performed by Aziz and Hellums (1976) and by Ozoe et al. (1976). They considered the case $\theta = 0\,^\circ$, the fluid heated from below, for a cubic box at specific Rayleigh numbers. Later Ozoe et al. considered also other geometries (1977) and small inclination angles ($0 \leq \theta \leq 30\,^\circ$, (1979)). In all cases the calculations have been done for low Rayleigh numbers ($Ra \leq 10^4$) and using the vorticity streamfunction approach.

More recently Chan and Banerjee (1979) developed a method to solve the governing equations in finite difference form, based on the "marker and cell" method. So far results with this method have been obtained only for $\theta = 0\,^\circ$ of the box with perfectly conducting and adiabatic and obtained results for Rayleigh numbers ranging from $10^4$ up to $10^5$. The largest deviations from an identical two dimensional situation have been found for the case with perfectly conducting side walls.

Results for an enclosure with aspect ratios $A_x = 2$, $A_z = 2$ showed even a completely different flow pattern in comparison with the corresponding two dimensional calculation.

Three dimensional numerical calculation of natural convection in a vertical enclosure ($\theta = 90\,^\circ$) have been done by Mallinson and deVahl Davis in 1977. They showed that the three dimensional motion for small Prandtl fluids is dominated by an inertial interaction of the circulating flow with the end walls. For $6 \times 10^4 \leq Ra \leq 10^6$ the end effect penetrates a distance less than 0.6 times the distance $D$ between the isothermal walls. In fig. 3.6 an example is given of a streamline, obtained by tracing a 'particle' of the flow. This figure demonstrates the end wall effect.

For enclosures of $A_x > 1.2$ a region might be expected where the flow is nearly two dimensional. Heat transfer results for adiabatic enclosures in the form of average Nusselt number as a function of the Rayleigh number obtained by Mallinson and deVahl Davis have been given in fig. 3.7. As shown in the figure the effect of the three dimensional motion on the heat transfer is still relatively small. For
Figure 3.6: Air Flow in an enclosure $A_x = 1$, $A_z = 2$, $Ra = 1.5 \times 10^5$. Streamline obtained by particle tracing.

Figure 3.7: Heat transfer results for air ($\phi = 90$) as obtained by Mallinson and deWahl Davis.

Low Rayleigh numbers the differences in heat transfer between the two and three dimensional calculations are of the order of 4 - 5 percent.

Calculations have been performed also for $Ra = 5 \times 10^5$ and varying length to depth ratios ($A_z$). For $A_z = 1, 2, 5$ and $\infty$ (2-dim. calculation) the Nusselt numbers for $Ra = 5 \times 10^5$ are 7.37, 7.43, 7.52 and 7.58, respectively. Although the effect is still small the decrease in heat transfer with decreasing $A_z$ indicates the suppression of the flow by the side walls.

Interaction between radiation, convection and conduction is essential for various heat transfer processes. For example the interaction of radiation with natural convection of a transparent fluid between vertical flat plates has been described by Carpenter, Briggs, Sernas (1976) and by Sparrow, Shah and Prakash (1980). More in the field of solar engineering is the work of Liu and Sparrow (1980) concerning the interaction of convection and radiation in an air-operated solar collector. Here the radiative-convective interaction can cause a significant increase of collector efficiency.

Directly related with our work is that of Hatfield and Edwards (1982) and the project of Hollands et al. (1979). Hatfield and Edwards have been interested primarily in the influence of radiation on the onset of natural convection in slits heated from below. They used the Galerkin method to determine the critical Rayleigh number at which convection starts and natural convective heat transfer increases. Along with numerical results, experimental data, obtained by holographic interferometry, have been also reported. Although all of their results concern the horizontal geometry where the fluid is heated from below, some of their significant results shall be discussed here.

First they define two dimensionless quantities $H$ and $N$. These parameters account for the relative strength of wall conduction and radiation in comparison with fluid conduction. These two parameters, which characterize a situation, have been defined as:

$$H = \frac{\lambda_w h}{\lambda_w h + \lambda_h h} \quad \text{(3.18a)}$$

$$N = \frac{4 \cdot \delta T_{w)} h}{\lambda_h h} \quad \text{(3.18b)}$$

with $\lambda_w$, $h_w$ and $\lambda_h$, $h$ the thermal conductivity and thickness of the wall and air respectively.

To illustrate the effect of wall radiation on the temperature field fig. 3.8 shows the dimensionless temperature $\theta$ as function of the dimensionless distance for a slit structure of aspect ratio $A_x = 0.25$, $\epsilon_{hc} = 1$, $H = 0$ and $N = 5$. The solid line corresponds to the temperature profile for suppressed side wall radiant exchange or if the side walls are perfectly conducting. The dashed line represents the temperature of the side walls at $x = 0$ and $x = A_x$. The dotted line shows the temperature profile of air at $x = 0.5 A_x$ when radiative exchange occurs. The side wall radiative exchange tends to an almost uniform temperature on the side walls with temperature jumps.
near the hot and cold wall. Conductive heat transfer in the fluid and the side walls oppose this tendency. This results in the temperature profiles as shown in fig. 3.8. The deviation from the linear profile (solid line) is most pronounced when the side wall emissivity is high, the emissivity of the hot and cold wall is low, the wall conduction parameter \( H \) is low and the radiation-conduction parameter \( N \) is high. The steep temperature gradients in the fluid near the hot and cold wall (see fig. 3.8) result in an increase of natural temperature profile. The results of Hatfield and Edwards (1982) obtained with holographic interferometry showed clearly the increase of natural convective heat transfer.

In fig. 3.9 the \( N_u - Ra \) relationship has been given for the initial part of the curve for natural convection for two small aspect ratio enclosures. For curve 1 (\( A_x = 0.25 \)) the critical Rayleigh number \( (Ra_c) \), the Rayleigh number at which convective heat transfer becomes important is smaller than for the enclosure with aspect ratio \( A_x = 0.14 \). Moreover the heat transfer by the air at the isothermal walls for \( Ra < Ra_c \) is greater than the heat transfer based on a linear temperature profile \( (N_u = 1.2 \text{ for } A_x = 0.25 \text{ and } Ra < Ra_c) \). This indicates the influence of the radiation on the heat transfer. For \( A_x = 0.14 \) Hatfield obtained different heat transfer figures through the air at the hot and cold wall of the enclosure. Due to the different emissivities of the hot and cold wall \( (\epsilon_h = 0.1 \text{ and } \epsilon_c = 0.9, \text{ respectively}) \) the measured Nusselt numbers are not equal. The higher radiative flux at the cold wall results in a lower conductive heat flux at the cold wall and vice versa for the hot wall. The critical Rayleigh numbers obtained at the hot and cold wall are approximately equal. Although the wall conduction parameter \( H \) is fairly high (0.75 and 1.7 respectively) the influence of the radiative heat exchange at the side walls on the natural convective heat transfer is yet considerable. Based on the experimentally derived critical Rayleigh numbers Hatfield and Edwards conclude that side wall radiative exchange in the slit stabilizes the fluid. However, when both radiation and wall conduction effects are important it is possible for wall conduction to be destabilizing at low values of the wall conduction parameter. This behaviour is due to the opposing tendencies of side wall radiation and conduction on the base flow temperature field.

The investigation of Hollands et al. (1979) concerned the use of natural convection suppression devices (CSD) to reduce heat losses from flat plate solar collectors. The third report (Phase III) of this investigation describes an experimental, analytical and numerical study on the radiative-conductive coupling of the heat transfer in CSD's. The experimental part of the Phase III report concerns total heat loss measurements across hexagonal honeycombs of different aspect ratio and varying hot \( (\epsilon_h) \) and cold \( (\epsilon_c) \) wall emissivity. Three sets of emissivities have been used:

1. BB-case: \( \epsilon_h = \epsilon_c = 0.08 \)
2. SB-case: \( \epsilon_h = 0.085, \epsilon_c = 0.08 \)
3. SS-case: \( \epsilon_h = \epsilon_c = 0.085 \)

The cases 1 - 3 represent respectively a black solar collector, a spectral selective and a spectral selective collector with an infrared reflective...
coating on the cover of the collector.

Total heat transfer measurements under stratified air conditions ($\phi = 180^\circ$)

**Figure 3.10:** Total heat transfer coefficients measured for stratified air situations by Hollands (1979).

for the three cases have been given in fig. 3.10.

The diameter of the hexagonal honeycomb was fixed at $0.95 \times 10^{-2}$ m and the aspect ratio has been varied by changing the distance $D$. It is noted that for given hot and cold wall emissivities the heat transfer decreases with decreasing $A_x$. This effect has been expected since both the conductive and radiative heat transfer decreases with increasing $D$. Furthermore for a specific honeycomb the heat transfer has its highest value for the BB-case, is less for the SB-case and has a lowest value for the SS-case. Although qualitatively in good agreement with the independent mode analysis (in which conduction and radiation have been calculated separately), quantitative large differences have been found between the measurements and the predictions according to the independent mode analysis. To account for the radiative exchange at the side walls, both numerical and analytical theories have been developed by Hollands et al. The theories describe the coupled heat transfer by radiation and conduction in hexagonal honeycombs and slit structures. The hexagonal honeycomb has been modelled as a cylindrical cell of equal area as the hexagon. Calculations have been performed for both diffuse and specular reflective side walls. For the slit structures the diffuse model only has been analyzed. Some results shall be discussed now.

The results have been obtained with the assumption that natural convection has been suppressed totally (so-called stagnant air).

In table 3.2 numerical predictions for the total heat transfer coefficient $h_t$ have been given for slit structures and compared with experimental results. The slit structures used for the experiments have been made of polyethelene film.

Although significant differences have been found (last column of table 3.2), the agreement with the measured $h_t$ values has been found to be satisfactory and therefore a specular analysis has not been performed by Hollands et al. Predictions of the total heat transfer for the hexagonal honeycombs have been given for the specular reflectivity model as well as for the diffuse reflectivity model. The comparisons of the experimentally derived heat transfer coefficients with the numerically predicted ones have been given in fig. 3.11 for the BB-, SB- and SS-case.

Based on the good correlation between the specular reflectivity model and the measurements for the BB-case, they concluded that the side walls must be considered specular reflective. According to Hollands the lack of agreement in the SB- and SS-case is due to the fact that the side wall material (Mylar, $\varepsilon_{hc} = 0.44$) is not gray, as assumed in the analysis. All the predictions obtained with the specular model are within $20\%$ of the measured $h_t$ values. Using the diffuse model larger deviations have been obtained.

With the analytical theory the effect of the side wall emissivity ($\varepsilon_{hc}$) has been investigated for a cylindrical honeycomb of $A_x = 0.2$ and $D = 0.051$ m. The side walls have been assumed specular reflective and heat conduction by the honeycomb material has been neglected. The results obtained from five different cases have been shown in fig. 3.12.

For the BB-case large reduction of heat transfer has been found for high side wall emissivity. For spectral selective collectors low heat transfer has
Figure 3.11: Comparison between \( h_t \) measurements and numerical derived \( h_t \) values.

Figure 3.12: The dependence of the total heat loss coefficient on the emissivity of the honeycomb (Hollands (1979)).

Table:

<table>
<thead>
<tr>
<th>( \varepsilon_h )</th>
<th>( \varepsilon_c )</th>
</tr>
</thead>
<tbody>
<tr>
<td>BB 06</td>
<td>88</td>
</tr>
<tr>
<td>SS 06</td>
<td>88</td>
</tr>
<tr>
<td>SB 06</td>
<td>88</td>
</tr>
<tr>
<td>SS 06</td>
<td>88</td>
</tr>
<tr>
<td>0.6</td>
<td>88</td>
</tr>
<tr>
<td>0.5</td>
<td>88</td>
</tr>
<tr>
<td>0.2</td>
<td>88</td>
</tr>
</tbody>
</table>

been predicted for very small values of \( \varepsilon_{hc} \). For high values of \( \varepsilon_{hc} \) the difference between black and spectral selective collectors become negligible. Moreover for the SB- and SS-case maximum heat transfer has been predicted at a specific \( \varepsilon_{hc} \) value. These maxima are a result of the radiation-conduction coupling. The reduction in heat transfer is therefore less than predicted by the independent mode analysis. However in comparison with a non-honeycomb spectral selective collector \( (h_t \approx 4 \text{ W/m}^2\text{K}) \) an improvement has still been obtained.

In comparison with the theoretical analysis of Hollands our numerical model has been extended on three essential points:

1. Heat conduction of the slit structure material in axial direction has been considered.
2. A mixed specular-diffuse reflectivity model has been used.
3. Natural convection of air in the slit structure has been considered.

Marcus (1983) obtained qualitative good agreement in comparison with the results of Hollands. He used a simplified analytical treatment for the coupled heat transfer. His analysis is based on the passage transmittance function as introduced by Edwards and Tobin (1967). Heat transfer results derived with this method showed also that for BB-collectors large heat transfer reductions can be obtained, due to the suppression of reradiation losses through the honeycomb. For SB-collectors a significant decrease of heat transfer may be expected only for small aspect ratios \( (A_x \leq 0.1) \).

3.2.3 The three dimensional analysis.

A numerical method has been used to solve the governing equations which describe the natural convection problem. For the two dimensional case several numerical methods have been described in literature. For instance Catton (1974) used the Galerkin method, whereas Shiralkar and Tien (1981) used an exponential difference scheme to obtain solutions for basically the same problem. In a few cases the finite element method has also been used. However the Finite Difference Method (FDM) is without doubt the method most applied for natural convection problems. The solutions for the three dimensional natural convection problems, as referred to in chapter 3.2.2, have been obtained by using a finite difference method. A finite difference method has been used also in this investigation. However contrary to others, who used the stream function-vorticity formulation of the governing equations (Mallinson and devahl Davis (1977), Ozoe (1976)), in our analysis the primitive variables (velocity, pressure, temperature) have been solved.

In 1972 Patankar and Spalding presented a finite difference method to solve the governing equations in the hydrodynamic formulation for recirculating flows, named TEACH. Schinkel (1980) used this method for the same parameter
range of the aspect ratio $(A_x \neq 1)$, as considered by us in this investigation for the two-dimensional case. In our case $(A_x \neq 1)$ less convergence problems may be expected. The extension of the TEACH programme to a three-dimensional formulation is straightforward. This makes the method preferable above a stream function–vorticity formulation analysis, where a three-dimensional approach leads to the introduction of a three-dimensional vector potential for the velocity and a three-dimensional vorticity vector. For the TEACH programme the extension results in five equations which have to be solved instead of six equations for the stream function–vorticity approach. The TEACH method as developed by Patankar and Spalding (1972) and as used by Schinkel (1980) is adopted for this investigation.

In the following chapters the method is described briefly for the three-dimensional natural convection problem.

3.2.3.1 The TEACH method.

The governing equations (3.5 - 3.7) possess a common form and therefore it is possible to present them by one general differential equation. If the dependent variable is denoted by $\xi$ the general differential equation reads:

$$
\frac{\partial}{\partial x_i} (a \xi) = \frac{\partial}{\partial x_i} (G \frac{\partial}{\partial x_i}) + S \xi
$$

with $u_i$ the velocity component in the $i$-direction. $F$ and $G$ are constants and $S \xi$ represents a source term. A description of the constants and variables as obtained for the governing equations is given in table 3.3.

Using equation (3.19) it is only necessary to construct one numerical solution procedure to solve the governing equations. The aim of the numerical method is to calculate the values of the variables at a set of chosen points (called the grid points). Algebraic equations (called the discretization equations) have been derived by integrating the governing differential equation over a subdomain surrounding each grid point. These subdomains are called the control volumes. In fig. 3.13 a typical grid point $P$ is shown. The control volume is constructed around point $P$. The other grid points $E, W,$

<table>
<thead>
<tr>
<th>eq.</th>
<th>$\xi$</th>
<th>$F$</th>
<th>$G$</th>
<th>$S$</th>
</tr>
</thead>
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<td>0</td>
<td>0</td>
</tr>
<tr>
<td>3.6a</td>
<td>$u$</td>
<td>Gr</td>
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<td>$-\frac{3}{\rho} \frac{\partial}{\partial x} (\xi - \bar{\xi})$ sin(\theta)</td>
</tr>
<tr>
<td>3.6b</td>
<td>$v$</td>
<td>Gr</td>
<td>1</td>
<td>$-\frac{3}{\rho} \frac{\partial}{\partial y} (\xi - \bar{\xi})$ cos(\theta)</td>
</tr>
<tr>
<td>3.6c</td>
<td>$w$</td>
<td>Gr</td>
<td>1</td>
<td>$-\frac{3}{\rho} \frac{\partial}{\partial z} \xi$</td>
</tr>
<tr>
<td>3.7</td>
<td>$\theta$</td>
<td>Gr</td>
<td>1/Pr</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 3.3: The values of the different terms in the general differential equation.

$N, S, U, D,$ are the East, West, North, South, Up and Down neighbours of $P$ in the three-dimensional presentation. The boundaries of the control volume are situated halfway between two grid points. It should be noted that it is not necessary to define a uniform grid, but for the sake of clarity equal grid distances have been used in fig. 3.13.

It is convenient to combine the convection and the diffusion fluxes which appear in eq. (3.19). Let $J_i$ denote the total flux (i.e. convection plus diffusion) in the $i$-direction:
The general differential equation reads now:

\[ \frac{\partial J}{\partial x} = S \]  

(3.20)

The integration of this equation over the control volume shown in fig. 3.13 leads to:

\[ \sum_{j} J_{j} A_{j} - J_{n} A_{n} + J_{e} A_{e} - J_{w} A_{w} + J_{u} A_{u} - J_{d} A_{d} = \bar{S}, V \]  

(3.22)

\( \bar{S} \) is the averaged source term over the control volume and \( V \) represents the volume of the control volume. Although according to fig. 3.13 all the areas of the faces of the control volume are equal for generality separate symbols have been used.

The \( J \)'s represent total fluxes at the control volume faces.

To obtain a finite difference representation of eq. (3.22) consider for example the total flux at the south face of the control volume (see fig. 3.13).

\[ J_{s} = Fu_{s} \left( \xi - \frac{G}{Fu_{s}} \frac{\partial \xi}{\partial x} \right) \]  

(3.23)

TEACH uses a so called hybrid difference scheme to obtain a finite difference representation of \( J_{s} \). The total flux \( J_{s} \) as defined by the hybrid scheme is

\[ J_{s} = Fu_{s} \xi_{s} \]  

(3.24)

in which \( \xi_{s} = Fu_{s} \xi_{s} / G \)

\( \xi_{s} \) can be considered as the ratio of transport by convection and transport by diffusion. As \( |\xi| > 2 \) convection dominates and the total flux is approximated by an upwind difference scheme. For \( |\xi| < 2 \) diffusion is important and the total flux is approximated by a central difference scheme. Applying the hybrid scheme for all the fluxes at the faces of the control volume leads to

\[ \sum_{j} A_{j}^{s} \xi_{j}^{s} = \bar{S}_{s} V \]  

\( j = N,S,E,W,U,D \)  

(3.25)

in which \( A_{j}^{s} = \sum_{j} A_{j}^{s} \)

Equation (3.25) is the general finite difference equation of eq. (3.19). The particular form of the coefficients of eq. (3.25) are dependent on \( \xi_{s} \), according to the hybrid difference scheme.

The source term has to be defined. The only equations which contain a source term are the components of the momentum equation eq. (3.6a - 3.6c).

\[ S_{u} = - \frac{\partial P}{\partial x} \sin(\theta) \cdot (\theta - \theta_{0}) \]  

(3.26a)

\[ S_{v} = - \frac{\partial P}{\partial y} \cos(\theta) \cdot (\theta - \theta_{0}) \]  

(3.26b)

\[ S_{w} = - \frac{\partial P}{\partial z} \]  

(3.26c)
In TEACH the source terms have been made linear around grid point P. For an example the source term for the x-direction becomes:

\[
\frac{\partial p}{\partial x} = \frac{p_P - p_S}{\gamma p_S} \sin(\theta) \cdot (\theta - \theta_S) = \sin(\theta) \cdot \left(\frac{\theta_P - \theta_S}{2}\right)
\]

The discretization of the source terms for the y- and z-direction have been derived by an analogous procedure. The only variable which still has to be determined is the pressure. The TEACH programme links the pressure field and the velocity field in the continuity equation by a method called SIMPLE (semi-implicit method for pressure-linked equations). Starting from an estimated pressure field SIMPLE calculates the velocities u, v and w from the momentum equations. In general the velocity field obtained does not satisfy the continuity equation, therefore the pressure has to be corrected. This leads to corrections in the velocities, so that the momentum equations are satisfied. After solving the equation for the other variables the corrected pressure, as obtained, is regarded as the estimated pressure in the next iteration step. Details of the SIMPLE algorithm can be found in Spalding (1976).

In the three dimensional version of TEACH, four different grids have been defined for the five variables. The pressure and temperature have been calculated at the points of the same grid but for the velocity components, staggered grids have been used. The location of the different grid points are shown in Fig. 3.14.

The grid points for the velocity components lie halfway between the grid points for the pressure and temperature. This means that the normal velocities are known at the faces of the control volume for p and \( \theta \). Furthermore the pressure is known at the faces of the different control volumes for the velocity components and can be used directly to calculate the velocity component located between them.

As a consequence of this arrangement is that interpolation of the convective terms in the momentum equations is needed.

To solve equation (3.25) for each of the variables an iterative procedure has been used. The procedure used in the three dimensional TEACH programme is basically the same as for the two dimensional programme. This method uses the TDMA (tridiagonal matrix algorithm) as its basic unit. All the equations along one grid line have been considered simultaneously for one variable and solved. The next line is then considered. In the three dimensional programme one starts at the first line in the first z-plane. After calculating all the lines in the first plane, the next z-plane has to be considered. When all the lines in the different z-planes have been calculated, one iteration of the line-by-line method is completed for a single variable. In Appendix A a description has been given of the solution procedure.

3.2.3.2 The non-uniform grid.

The choice of a grid is essential for the convergence and accuracy as well as for the economic use of the computer programme. Too few grid points (especially in areas where steep gradients can be expected) result in a not correctly defined representation of the physical problem and leads to an inaccurate solution. Too much grid points need excessive computer time to reach a converged situation. In our case, near the hot and cold wall, the fluid is accelerated, due to buoyancy forces. At the four side walls the fluid is
slowed as a result of the no-slip condition. Therefore near all the walls temperature and large velocity gradients may be expected and at these places a fine grid is required. In the center of the enclosure a relative tranquil core region exists and a coarser grid is permitted. Moreover to incorporate the two symmetry properties it is convenient if the grid possesses the same symmetries as that of the problem. To accommodate the above mentioned requirements a non-uniform grid has been used. This grid is defined by:

\[ x(i+1) = 0.5 A_x (i) \quad k \quad i = 1, k \]

\[ x(i) = - x(2) \]

With \( N_x \), the total of grid points in the \( x \)-direction, \( k = (N_x - 1)/2 \) and \( A_x \) is the non-linearity parameter. The grid is symmetrical around the point \( 0.5 A_x \), thus

\[ x(N_x - i) = A_x - x(i + 1) \]

\[ x(N_x) = A_x + x(2) \]

For \( A_x = 1 \) the grid is uniform. A typical value for \( A_x \), as we used in our calculations is 1.5. For the \( y \)- and \( z \)-direction the grids have been defined in a similar way. The points of intersection of the \( x \)-, \( y \)- and \( z \)-lines, defined above, are the grid points for the pressure and the temperature. The staggered grids for the velocities have been derived by placing the velocity grid points halfway at the upstream faces of the control volume for \( p \) and \( \theta \), as already described in chapter 3.2.3.1.

3.2.3.3 The optimum grid.

The grid chosen depends on two variables. First the number of grid points, secondly the value of the non-linearity parameter. Both parameters can be used to optimize the grid, with regard to accuracy and economic use of computer time. In our case most calculations have been performed with equal non-linearity parameters \( A_x = A_y = A_z = 1.5 \) for the three dimensional case and \( A_x = A_y = 1.5 \) for the two dimensional case. For the different aspect ratios \( A_x \) the number of grid points in the \( x \)-direction has been varied. The optimum number of grid points obtained from this procedure for the two and three dimensional case is shown in table 3.4. For the \( y \)-direction 21 grid points have been used for all calculations (\( N_y = 21 \)). In the \( z \)-direction 11 points have been taken for \( A_z = 1 \) (\( N_z = 11 \)). For an enclosure with \( A_x = 0.25, A_z = 1 \), the symmetry properties incorporated, the total number of grid points, which have to be considered in our calculation amounts to \((8 \times 19 \times 7) = 1064\). With this number of grid points the calculations for small aspect ratio \( A_x = 0.25 \) have been within 0.1% of calculations performed with a finer grid \((21 \times 21 \times 21)\) for \( Ra = 10^6 \) and within 3% for \( Ra = 10^8 \).

For larger \( A_x \), in which we are not primarily interested, larger deviations occur. For smaller values of \( A_x \) the accuracy at high Rayleigh numbers is better (0.1% deviation for \( A_x = 0.125, Ra = 10^8 \)).

A typical grid as used for the two dimensional convection calculations is

\[ A_x = 0.25, N_x = 13, N_y = 21, A_x = A_y = 1.5. \]
shown in fig. 3.15.

3.2.3.4 The boundary conditions.

With the non-uniform grid as defined, it is necessary to incorporate the boundary conditions. The boundary conditions have been given by eq. (3.11) and (3.12) for the velocity and temperature respectively. Consider as an example the first grid point near the wall as shown in fig. 3.16. The south face of the control volume of P lies exactly on the wall. The south neighbour (S) of P lies, as defined by eq. (3.28b) south of the boundary line ("in the wall"). The incorporation of the velocity boundary condition becomes very simple by setting \( u_s \) to zero. At the other walls the incorporation of the no-slip condition is analogous.

As boundary condition for the temperature we have that on the walls the temperatures are known \( \Theta = 1, \Theta = 0 \) and \( \Theta = 1 - y \) for the hot, cold and side walls respectively. The total heat flux through the south face of the control volume at the wall is given by eq. (3.29) and table 3.3:

\[
J_s = - \frac{1}{\Pr} \frac{\partial \Theta}{\partial x}
\]

(3.29)

In the finite difference equation the heat flux is approximated by

\[
J_s = - \frac{1}{\Pr} \frac{\partial \Theta}{\partial x}
\]

(3.30)

Incorporation of the heat flux at the south face of the control volume at the wall in the general finite difference equation is done by setting \( a_s = 0 \) while \( J_s \) as given by eq. (3.30) is added as a source term.

\[
S = b \Theta_P + c
\]

(3.31a)

where

\[
b = \frac{2}{\Pr x_{PS}}
\]

(3.31b)

and

\[
c = \frac{2}{\Pr x_{PS}}
\]

(3.31c)

3.2.3.5 Heat transfer calculation.

To characterize the heat transfer by convection a Nusselt number has been defined as:

\[
Nu_{x,z} = \frac{1}{A_{x,z}} \int_0^A \int_0^z Nu_{x} \, dx \, dz
\]

(3.32)

in which \( Nu_{x} \) represents the local Nusselt number at the hot wall \( (y = 0) \). In non-dimensional form the local Nusselt number is
To calculate the local Nusselt number at the hot wall a cubic polynomial has been fitted through values of 0 at the wall and the first two grid points near the wall. The average Nusselt number has been obtained by numerical integration of \( \text{Nu}_\theta(x,z) \).

3.2.3.6 The convergence criterion.

Since the determination of the heat transfer is the particular goal of the calculations, the convergence criterion is based on the change of the averaged Nusselt number between successive iterations. A converged situation is reached when the following criterion has been satisfied five times:

\[
\frac{\text{Nu}_\theta(i) - \text{Nu}_\theta(i-1)}{\text{Nu}_\theta(i-1)} < \delta \tag{3.34}
\]

where \( i \) denotes the iteration number and \( \delta \) is \( 10^{-5} \).

For each variable eq. (3.25) has to be solved for all the grid points. The numerical calculation has been programmed so that each variable is calculated for all the grid points before switching to the next variable. One complete iteration is defined by solving all the variables for all the grid points once. Due to the coupled character of the equations and the line-by-line method, which is used to solve the equations, the set of variables obtained shall normally not satisfy eq. (3.25) and a residual sum \( R^5 \) can be defined for each variable at each grid point.

\[
R^5 = R^5_p - \sum_j a^5_{ij} R^5_j - \bar{S}^5_{ij} U \\
= N,S,E,W,U,D \tag{3.35}
\]

Summing up all the residues at each point for the five variables results in five residual sources \( R^5_p \). After successive iterations each residual source should decrease to a small value. \( R^5_p(\%) \) is dependent on the number of grid points. The residual source terms have been used for setting each underrelaxation factor for each variable. Underrelaxation is needed to obtain the solution in as few iterations as possible. Equation (3.25) can be written as

\[
\xi_p(i) = \xi_p(i-1) + \sum_j a^5_{ij} R^5_j - \bar{S}^5_{ij} U - \xi_p(i-1) \tag{3.36}
\]

where \( i \) denotes again the iteration number. The term within the square brackets represents the change of \( \xi_p \) for the current iteration. To reduce this change, an underrelaxation factor \( \gamma(\%) \) (\( 0 \leq \gamma(\%) \leq 1 \)) can be introduced so that

\[
\xi_p(i) = \xi_p(i-1) + \gamma(\%) \left( \sum_j a^5_{ij} R^5_j - \bar{S}^5_{ij} U - \xi_p(i-1) \right) \tag{3.37}
\]

Table 3.5: Typical values of \( R^5_p(\%) \) and \( \gamma(\%) \) for \( Ra = 18^5 \), \( \phi = 45^\circ \).

<table>
<thead>
<tr>
<th>( R^5_p(%) )</th>
<th>( \gamma(%) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( u )</td>
<td>( 10^{-3} )</td>
</tr>
<tr>
<td>( v )</td>
<td>( 10^{-3} )</td>
</tr>
<tr>
<td>( w )</td>
<td>( 10^{-5} )</td>
</tr>
<tr>
<td>( p )</td>
<td>( 10^{-6} )</td>
</tr>
<tr>
<td>( \theta )</td>
<td>( 10^{-3} )</td>
</tr>
</tbody>
</table>

For \( \gamma(\%) = 1 \) no underrelaxation is affected; \( \gamma(\%) \) close to zero means that the value of \( \xi_p \) changes very slowly. Effective underrelaxation highly reduces computer time. In table 3.5 characteristic values of \( R^5_p(\%) \) and \( \gamma(\%) \) have been given for an enclosure with \( A_x = 0.25 \) and \( A_z = 1 \), as used in this investigation.

Using the optimum underrelaxation factor for each variable a converged situa-
tion was reached within 396 iterations (at high Rayleigh numbers), which took about 158 CPU seconds on an Amdahl 470/V7B computer.

3.2.4 The two dimensional test problem.

In order to test the numerical method an accurate solution of a well defined problem should be known. In 1979 Jones (1979) introduced an international comparison exercise for the calculation of buoyancy driven flows in two dimensional enclosures. Participants on this comparison exercise were invited to produce solutions for the problem of natural convection of air in a square adiabatic cavity. In fig. 3.17 the problem is given.

We participated in this comparison exercise with solutions of the two dimensional TEACH programme. For the three dimensional programme no such comparison exercise has been done. However, knowing the results of the two dimensional comparison exercise and the fact that the three dimensional programme is an extended version of the two dimensional programme without essential differences, justifies its use. To test the three dimensional computer programme a number of test runs have been made and the results have been compared with the two dimensional solutions.

Figure 3.17: The comparison problem as proposed by Jones.

3.2.4.1 Results of the test problem.

For the comparison exercise stated above a benchmark numerical solution has been provided by deMahl Davis in 1981 (latest revised version of the benchmark solution by the same author has been published in 1992). The benchmark solution is obtained by solving the governing equations in streamfunction-vorticity formulation with a finite difference method. Second order central difference approximations have been used for all space derivatives and forward differences for the time derivatives. Extreme mesh refinement and extrapolation leads to solutions for $10^3 \leq Ra \leq 10^8$, which are assumed to be accurate to better than 1% for $Ra = 10^6$ and down to 0.1% for $Ra = 10^3$. For Rayleigh values of $10^3$ and $10^4$, solutions have been obtained using uniform meshes from 11x11 to 41x41. At the higher values of the Rayleigh number even finer meshes up to 81x81 have been used.

In table 3.6a the following quantities, describing the benchmark solution, have been given.

- $\phi_{\text{mid}}$: the absolute streamfunction value at the midpoint of the cavity.
- $\phi_{\text{max}}$: the absolute maximum value of the streamfunction, with its location.
- $v_{\text{max}}$: the maximum horizontal velocity on the vertical midplane of the cavity, with its location.
- $u_{\text{max}}$: the maximum vertical velocity on the horizontal midplane of the cavity, with its location.
- $\langle Nu \rangle$: the average Nusselt number throughout the cavity.
- $Nu_{y \parallel}$: the average Nusselt number on the vertical midplane of the cavity.
- $Nu_{y \perp}$: the average Nusselt number at the hot wall ($y = 0$).
- $Nu_{\text{max}}$: the maximum value and its location of the local Nusselt number at the hot wall.
- $Nu_{\text{min}}$: the minimum value and its location of the local Nusselt number at the hot wall.

Our results are shown in table 3.6b. These results have been obtained with our non-uniform 21x21 mesh and the calculations have been performed with the application of the centro-symmetry property. To correct the results for the influence of the grid the non-linearity parameter $\alpha_y$ has been varied. Four different $\alpha_y$ values have been taken, ranging from 1.5 to 1.8. Using the
extrapolation procedure, as described by Schinkel (1980), produced the for the grid corrected results. The percentage differences of our solution with the bench mark solution has been given also in table 3.6b.

It should be noted that our values for Nu\(\text{y}\) have been averaged for the four different \(A_z\) cases. No values for the average Nusselt number in the cavity have been derived, since it was not asked for at the initial participation form. Very good agreement has been found for low Rayleigh numbers, \(10^3\) and \(10^4\). At higher Rayleigh number (\(10^5\)) the agreement is still good. Only large deviations occur in the local heat transfer, Nu\(\text{y}\), for \(Ra = 10^6\). In our opinion a finer grid should give the same accuracy for \(Ra = 10^6\) as is new obtained for the lower Rayleigh values. However as our main interest is in the average heat transfer, given by Nu\(\text{y}\), we did not try to improve our programme. Even at \(Ra = 10^6\) the deviation for this value is only 3.2 %.

We compared the results of the test runs of our three dimensional programme with our two dimensional results. All test runs for the three dimensional programme have been performed for a vertical square enclosure \((\ell = 98; A_z = 1)\) and perfectly conducting side walls (pc). In the vertical orientation an unicellular flow might be expected in the enclosure, as long as the length to depth ratio \((A_z)\) is much larger than 1. The influence of the end walls at \(z = 0\) and \(z = A_z\) is small for large \(A_z\). For increasing \(A_z\) the flow becomes more and more two dimensional. Experiments done by Schinkel (1980) showed that the flow at the midplane of a vertical enclosure \((z = 0.5 A_z)\) can be considered two dimensional if \(A_z\) is larger than four.

To compare results of the three dimensional programme with the two dimensional one, we increased the aspect ratio \(A_z\) for \(Ra = 10^4\), \(10^5\) and \(10^6\). The results for the average Nusselt number as a function of \(A_z\) is shown in fig. 3.18.

For example the percentage differences with the two dimensional solution are -13.1 %, -7.1 % and -2.2 % for \(A_z = 1\) at \(Ra = 10^4\), \(10^5\) and \(10^6\), respectively. Suppression of the flow and a decrease in heat transfer can only be expected for very small values of \(A_z\) \((A_z < 1)\). This agrees with the results of Mallinson and deVahl Davis (1977).

To compare the flow structure of the two and three dimensional cavity flow, it is convenient to prepare streamline contour plots. However in the three

**Table 3.6a:** The benchmark solution.

<table>
<thead>
<tr>
<th>(Ra)</th>
<th>(10^3)</th>
<th>(10^4)</th>
<th>(10^5)</th>
<th>(10^6)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{mid}})</td>
<td>1.174</td>
<td>5.071</td>
<td>9.111</td>
<td>16.32</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{max}})</td>
<td>-</td>
<td>-</td>
<td>9.612</td>
<td>16.75</td>
</tr>
<tr>
<td>(x)</td>
<td>-</td>
<td>-</td>
<td>0.881</td>
<td>0.829</td>
</tr>
<tr>
<td>(y)</td>
<td>-</td>
<td>-</td>
<td>0.681</td>
<td>0.829</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;)_{\text{max}})</td>
<td>3.649</td>
<td>16.178</td>
<td>34.73</td>
<td>64.63</td>
</tr>
<tr>
<td>(x)</td>
<td>0.813</td>
<td>0.823</td>
<td>0.855</td>
<td>0.850</td>
</tr>
<tr>
<td>(y)</td>
<td>0.178</td>
<td>0.119</td>
<td>0.866</td>
<td>0.8379</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{max}})</td>
<td>3.597</td>
<td>19.617</td>
<td>34.95</td>
<td>64.63</td>
</tr>
<tr>
<td>(x)</td>
<td>0.813</td>
<td>0.823</td>
<td>0.855</td>
<td>0.850</td>
</tr>
<tr>
<td>(y)</td>
<td>0.178</td>
<td>0.119</td>
<td>0.866</td>
<td>0.8379</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;)_{\text{max}})</td>
<td>3.649</td>
<td>16.178</td>
<td>34.73</td>
<td>64.63</td>
</tr>
<tr>
<td>(x)</td>
<td>0.813</td>
<td>0.823</td>
<td>0.855</td>
<td>0.850</td>
</tr>
<tr>
<td>(y)</td>
<td>0.178</td>
<td>0.119</td>
<td>0.866</td>
<td>0.8379</td>
</tr>
</tbody>
</table>

**Table 3.6b:** Our results for the comparison exercise and the percentage deviation with the benchmark solution.

<table>
<thead>
<tr>
<th>(Ra)</th>
<th>(10^3)</th>
<th>(10^4)</th>
<th>(10^5)</th>
<th>(10^6)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{mid}})</td>
<td>1.144</td>
<td>4.948</td>
<td>9.155</td>
<td>16.342</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{max}})</td>
<td>-</td>
<td>-</td>
<td>9.522</td>
<td>16.591</td>
</tr>
<tr>
<td>(x)</td>
<td>-</td>
<td>-</td>
<td>8.488</td>
<td>8.537</td>
</tr>
<tr>
<td>(y)</td>
<td>-</td>
<td>-</td>
<td>0.503</td>
<td>0.537</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;)_{\text{max}})</td>
<td>3.633</td>
<td>16.033</td>
<td>34.033</td>
<td>64.566</td>
</tr>
<tr>
<td>(x)</td>
<td>0.823</td>
<td>0.823</td>
<td>0.874</td>
<td>0.885</td>
</tr>
<tr>
<td>(y)</td>
<td>0.1748</td>
<td>0.111</td>
<td>0.865</td>
<td>0.836</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{max}})</td>
<td>3.633</td>
<td>16.033</td>
<td>34.033</td>
<td>64.566</td>
</tr>
<tr>
<td>(x)</td>
<td>0.823</td>
<td>0.823</td>
<td>0.874</td>
<td>0.885</td>
</tr>
<tr>
<td>(y)</td>
<td>0.1748</td>
<td>0.111</td>
<td>0.865</td>
<td>0.836</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;)</td>
<td>1.102</td>
<td>2.153</td>
<td>4.519</td>
<td>9.880</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{max}})</td>
<td>1.120</td>
<td>2.257</td>
<td>4.535</td>
<td>9.516</td>
</tr>
<tr>
<td>(x)</td>
<td>0.822</td>
<td>0.822</td>
<td>0.882</td>
<td>0.8158</td>
</tr>
<tr>
<td>(&lt;\text{Nu}&gt;_{\text{max}})</td>
<td>0.699</td>
<td>0.589</td>
<td>0.798</td>
<td>1.674</td>
</tr>
<tr>
<td>(x)</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>
dimensional case the streamfunction is usually constant along closed planes, forming streamtubes. In our case we are mainly interested in the flow structure at the midplane, where the flow resembles the two dimensional flow the most. Due to the z-symmetry the u-velocity component at the midplane is zero and the flow is therefore strictly two dimensional and a two dimensional streamfunction can be defined. In fig. 3.19 streamline contour plots are shown for three different Rayleigh numbers ($10^5$, $10^6$, $10^7$).

For each Rayleigh number three cases have been given: (1) the two dimensional contour plot ($\phi = 90^\circ$); (2) the three dimensional contour plot at the midplane ($z = 0.5 A_x$) for $A_x = 10$ and (3) the three dimensional contour plot at the midplane for $A_x = 0.5$. The contour plots for the two dimensional case and the three dimensional case with $A_x = 10$ show the same flow structure at equal Rayleigh numbers. This agrees with the obtained heat transfer results. For $A_x = 0.5$ the contour plots show a somewhat different flow pattern, due to the increased end wall effect. At $Ra = 10^5$, $A_x = 0.5$ secondary flow does not exist, as has been obtained for $A_x = 10$ and for the two dimensional case. The development of the flow is suppressed by the strong influence of the side walls at $z = 0$ and $z = A_x$. This results in a smaller heat transfer, as was obtained. For $Ra = 10^5$ the influence of the end walls affects the flow pattern at the midplane and a different flow pattern has been obtained. We obtained with the three dimensional programme, as expected, for small values of $A_x$ a decrease of natural convective heat transfer in comparison with the solutions obtained with the two dimensional programme. For large $A_x$ ($A_x \geq 10$) the differences in heat transfer as well as in flow structure between the two and three dimensional cases are negligible.

In addition the natural convection has been calculated for a vertical cubic enclosure ($A_y = A_z = 1$) with in one case perfectly conducting and in another case adiabatic side walls. The results for the average Nusselt numbers are shown in fig. 3.20.

Two cases (pc and ad) have been calculated with a 21x21x11 mesh and with $1.5$ as value for all three non-linearity parameters of the grid. As shown in the figure the power in the $Nu_A - Ra$ relation is equal (0.33) for both cases, but the convective heat transfer is larger for the ad-case than for the pc-case. This is in agreement with two dimensional results obtained by Schinkel (1980). He also found equal $Nu_A - Ra$ dependence for the two cases and larger heat transfer for the ad-case.

We compared our results also with those obtained by Mallinson and deVahl Davis (1977). They calculated the heat transfer by natural convection in three dimensional vertical enclosures mainly for enclosures with $A_y = 2$, adiabatic side and end walls and $Pr$ in the range of 0.2 to 100. Only one single case for a cubic enclosure has been calculated by them for air at $Ra = 5 \times 10^5$. However from their result ($Nu = 7.37$) for a cubic enclosure in comparison with our result for $A_x = 2$ ($Nu = 7.43$) and our results for dif-
ferent \( A_z \) as described earlier, no great differences may be expected in heat transfer due to this difference in aspect ratio \( A_z \). In fig. 3.28 their result for \( A_z = 2 \) is given by a dashed line together with their single result for \( A_z = 1 \) (\( \text{Ra} = 5 \times 10^5 \)). Only small differences have been found in comparison with our results. These differences have two causes. At high Rayleigh numbers the deviation is due to the difference in the number of grid nodes. They used 15 points for the \( y \)-direction and therefore underestimated the heat transfer. At low Rayleigh numbers the difference in heat transfer is presumably due to the aspect ratio effect. Larger heat transfer is expected for \( A_z = 2 \). In general the agreement is good.

In fig. 3.21 the \( x \)-integrated Nusselt numbers on the hot wall have been shown. The \( x \)-average Nusselt number is defined by

\[
\langle \text{Nu}_\text{B}(z) \rangle_x = \frac{1}{A_x} \int_0^{A_x} \text{Nu}_\text{B}(x,z) \, dx 
\]

for \( y = 0 \) \hspace{1cm} (3.36)

This equation represents the local Nusselt number at the hot wall for a specific value of \( z \), integrated over \( x \) along the hot wall. The results have been obtained with non-linearity parameters equal to 1.5. The influence of the end walls on the flow for the pc-case is larger than for the ad-case.

For the pc-case reduced heat transfer has been obtained near the end walls and for high Rayleigh values (\( \text{Ra} = 10^6 \)) at the midplane of the enclosure. The same end wall effects have been found by Mallinson and deWahl Davis (1977) and qualitative good agreement has been found.

From these results the conclusion is drawn that the three dimensional programme for calculating the laminar and stationary natural convection in enclosures is correct.

3.2.5 Results.

The natural convection calculations have been performed for enclosures with perfectly conducting side and end walls (pc) for the Rayleigh number range of interest: \( 10^4 \leq \text{Ra} \leq 10^6 \) (see chapter 1.2). The angle of inclination, measured from the horizontal, has been taken at 90, 60 and 45 degrees.

For the two dimensional case the aspect ratio \( A_x \) has been varied between 0.5 and 8.1, whereas for the three dimensional calculations \( A_x \) has been varied between 0.1 \( \leq A_x \leq 0.5 \) and \( A_z \) has been taken equal to \( A_x = 1 \) and 10. The results have been obtained for values of the non-linearity parameters \( \alpha_x = \alpha_y = \alpha_z = 1.5 \). Equal grids have been used for the same aspect ratio \( A_x \) in the two and three dimensional calculations. No corrections have been made for the grid used. Since we have been interested primarily in enclosures with small aspect ratio, the grids used, as defined in chapter 3.2.3.2, have been found sufficiently fine. At higher Rayleigh numbers (\( \text{Ra} \geq 10^6 \))
deviations from the correct solutions might occur. As mentioned before these deviations are within 3% of $N_u$ for small aspect ratio $A_x$ ($A_x < 0.25$).

3.2.5.1 Two dimensional convection calculations.

With the two dimensional programme natural convective heat transfer has been calculated for enclosures with aspect ratio $A_x$ equal to 0.5, 0.25, 0.125 and 0.1. The result for the averaged heat transfer, given by the aver-

age Nusselt number $N_u$, as a function of Rayleigh, is shown in fig. 3.22. For all angles of inclination and Rayleigh numbers considered the heat transfer is largest for $A_x = 0.5$, but decreases rapidly with decreasing aspect ratio $A_x$. For $A_x = 0.125$ and 0.1 the average Nusselt number is hardly greater than 1 at the Rayleigh range considered. This indicates that no significant convective heat transfer occurs.

For aspect ratio $A_x = 0.25$ the Nusselt number remains very near 1 up to $Ra = 10^8$, but for higher Rayleigh numbers natural convective heat transfer is no longer suppressed and $N_u$ increases rapidly with Rayleigh.

For $A_x = 0.5$ natural convection is already significant at $Ra = 10^4$ and after an initial strong increase of the Nusselt number for Rayleigh up to $10^6$ a more gradual increase has been obtained for larger Rayleigh. No direct power law relations between $N_u$ and $Ra$ have been derived. Only for $A_x = 0.5$ and $Ra > 10^5$ such a power law relation has been found, as obtained by Wirtz (1979) for $A_x = 0.5$.

For the angles of inclination considered we obtained the following relations:

\[
\begin{align*}
N_u &= 0.020 \, Ra^{0.419} \quad \theta = 90; \ Ra > 10^5 \\
N_u &= 0.036 \, Ra^{0.376} \quad \theta = 60; \ Ra > 10^5 \\
N_u &= 0.037 \, Ra^{0.369} \quad \theta = 45; \ Ra > 10^5 \\
\end{align*}
\] (3.39)

Reasonable agreement has been found with the results of Wirtz and Tseng (1979) as can be seen in fig. 3.22. Wirtz obtained larger Nusselt numbers. This is mainly due to the difference in temperature boundary condition. As explained by Schinkel (1980) adiabatic side walls result in larger $N_u$ than perfectly conducting side walls.

It might be expected that for $A_x = 0.25$ and smaller the natural convection at even larger Rayleigh number ($Ra > 10^6$) becomes significant and that for those Rayleigh numbers a power law relation can be used for the $N_u - Ra$ dependence also.

For the adiabatic case Lee (1980) found the strong increase of the heat transfer with the Rayleigh number followed by a more gradual increasing heat transfer with $Ra$ (see fig. 3.5). For larger $A_x$ ($A_x = 0.4$) Lee found a single power law relation. Although a quantitative comparison between Lee's results and our results is not possible, due to the different side wall boundary condition, good qualitative agreement has been found.

The dependence of the Nusselt number on the angle of inclination for the two dimensional case is small as can be seen in fig. 3.22. Only for $A_x = 0.25$ and 0.5, when natural convective heat transfer occurs, maximum heat transfer has been found for $\theta = 60$ degrees. For smaller values of $A_x$ the angle of inclination ($\theta < 45$) has no significant effect on the heat transfer at the Rayleigh range considered. Most important is the fact that decreasing the angle of inclination does not immediately results in a decrease of heat transfer.

If a critical Rayleigh number ($Ra_c$) is defined as the Rayleigh number where convective heat transfer becomes significant (here arbitrarily taken as $N_u = 1.2$) our Rayleigh range considered includes the critical Rayleigh numbers for $A_x = 0.5$ and $A_x = 0.25$. In table 3.7 the critical Rayleigh numbers, obtained from fig. 3.22 have been given.

The effect of the angle of inclination on $Ra_c$ is small for the aspect ratios and angles considered.
Table 3.7: Critical Rayleigh numbers ($Ra_c$) for different angles of inclination.

<table>
<thead>
<tr>
<th>$A_x$</th>
<th>0</th>
<th>90</th>
<th>60</th>
<th>45</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>$1.8 \times 10^4$</td>
<td>$1.5 \times 10^4$</td>
<td>$1.6 \times 10^4$</td>
<td></td>
</tr>
<tr>
<td>0.25</td>
<td>$2.2 \times 10^5$</td>
<td>$2.1 \times 10^5$</td>
<td>$2.1 \times 10^5$</td>
<td></td>
</tr>
<tr>
<td>0.125</td>
<td>&gt; $10^6$</td>
<td>&gt; $10^6$</td>
<td>&gt; $10^6$</td>
<td></td>
</tr>
<tr>
<td>0.1</td>
<td>&gt; $10^6$</td>
<td>&gt; $10^6$</td>
<td>&gt; $10^6$</td>
<td></td>
</tr>
</tbody>
</table>

In fig. 3.23 the Nusselt dependence on $A_x$ has been given for $Ra$ equal to 

![Figure 3.23: Dependence of $Nu_{ho}$ on the aspect ratio $A_x$.](image)

Figure 3.23: Dependence of $Nu_{ho}$ on the aspect ratio $A_x$.

An interesting conclusion from this figure is that for $A_x \geq 0.125$ and all angles of inclination between 45 and 90 degrees natural convective heat transfer has been suppressed totally. At high Rayleigh number ($Ra > 5 \times 10^5$) Nusselt increases very strongly with increasing $A_x$ for $A_x$ larger than 8.125. A comparison of our results with the theory of Bejan (1980) has been given in fig. 3.24.

The quantitative large differences in Nusselt values are mainly due to the difference in the boundary condition for the temperature at the side walls. The theory of Bejan has been developed for adiabatic side walls. Our calculations have been done for perfectly conducting side walls. All our results show smaller Nusselt values than the corresponding results from Bejan's theory. This is in agreement with the results of Schinkel (1980), as referred earlier. We calculated insufficient different aspect ratio enclosures to obtain a smooth function for the $Nu_{ho} - A_x$ dependence, but our results agree qualitatively very well with those of the theory of Bejan. Moreover also Bejan obtained suppression of natural convective heat transfer for $A_x < 0.125$ at the Rayleigh range of interest.

Finally in fig. 3.25 a,b,c streamline contour plots have been given for

![Figure 3.25: Streamline contour plots for $Ra = 5 \times 10^5$.](image)

Figure 3.25: Streamline contour plots for $Ra = 5 \times 10^5$. 

Ra = 5 \times 10^5 \text{ and for enclosures with aspect ratio 0.5, 0.25 and 0.125.} \\

The relative high Rayleigh number of 5 \times 10^3 has been chosen because at this Rayleigh number flow development can be seen even at small aspect ratios. For \( \theta = 90 \) degrees (see fig. 3.25a) suppression of the flow due to the decrease in aspect ratio can be seen. A significant boundary layer flow has been obtained for \( A_x = 0.5 \) and in the centre of the enclosure the start of secondary flow can be observed. For \( A_x = 0.25 \) a single cellular circulating base flow has been obtained. Small flow constrictions can be seen in the centre of the enclosure. This indicates the tendency of the flow to break into two separate flow regions. For \( A_x = 0.125 \) the flow is divided into separate regions. A circulating base flow along the four walls and near the the hot and cold wall a circulating roll has been obtained. The rolls are separated by a very tranquil centre in which hardly any flow has developed. At an angle of inclination of 60 degrees the flow structure is similar for aspect ratio 0.5 in comparison with the vertical situation. Only secondary flow in the center of the enclosure has been suppressed completely. For aspect ratio 0.25 and 0.125 the tendency exist to form a base flow in the whole enclosure. At \( \theta = 45 \) degrees again no great differences of flow structure has been obtained for \( A_x = 0.5 \). For smaller aspect ratio \( A_x = 0.25 \) and 0.125 the base flow is stronger again as can be seen for \( A_x = 0.125 \). Here the rolls at the hot and cold wall have disappeared and the centre of the enclosure is a part of the base flow.

From the two dimensional calculations we conclude that suppression of the flow and thereby a minimum in natural convective heat transfer shall be obtained at all angles of inclination of interest for \( A_x = 0.125 \), at the Rayleigh range of interest \( (10^4 \leq Ra \leq 10^6) \).

3.2.5.2 Three dimensional convection calculations.

To consider suppression of natural convection in cell structures three dimensional calculations have been performed for rectangular enclosures. Contrary to the two dimensional calculations for slit structures, the three dimensional programme possesses the feasibility to account for the influence of the end walls. The viscous forces, acting on the natural convection flow, shall suppress the flow even more. Moreover the velocity becomes three dimensional in the numerical presentation.

Our calculations concerned three types of enclosures:

(a): enclosures with large \( A_z \), \( A_z = 10 \). This enclosure resembles the two dimensional case. However due to the three dimensional velocity vector the flow structure is not restricted to the xy-plane of the cavity. Calculations have been performed for height to width aspect ratio \( A_z \) of 0.5, 0.25, 0.125 and 0.1.

(b): enclosures with medium \( A_z \), \( A_z = 1 \). For this depth to width aspect ratio the natural convection flow becomes significantly influenced by the end walls, according to the results of the test problem. Aspect ratio \( A_z \) has been taken 0.5, 0.25, 0.125 and 0.1.

(c): enclosures with small \( A_z \), \( A_z = A_x \). This is called a honeycomb enclosure. The hot and cold wall are squares and the largest suppression of the flow might be expected in such enclosures. The aspect ratios have been chosen as \( (A_z = A_x = 0.5, 0.25, 0.125 \) and 0.1.

For all types of enclosures natural convection has been calculated for angles of inclination of 90, 60 and 45 degrees and the side and end walls have been considered perfectly conducting (pc). In fig. 3.26 the three different types

![Figure 3.26: Example of the different A_x enclosures, A_x = 0.25.](image)
been given by the dotted lines. Fig. 3.27 shows the result for large $A_z$ enclosures. For $\phi = 90$ degrees complete suppression of natural convective heat transfer has been obtained for $A_x = 0.1$ and 0.125. For $A_x = 0.25$ and 0.5 a gradual increase of the Nusselt number with Rayleigh has been calculated. For the vertical situation and $A_x = 10$ the influence of the end walls is small and a two dimensional flow might be expected. The heat transfer results for $\phi = 90$, $A_z = 10$ as given in fig. 3.27a for various $A_x$ and obtained with the two dimensional programme are equal to results of the comparable two dimensional calculations.

For angles of 60 and 45 degrees (fig. 3.27b and c) complete suppression of natural convective heat transfer has no longer be accomplished for the Rayleigh range considered. Even for $A_x = 0.125$ and 0.1 natural convection develops and therefore increased natural convective heat transfer has been obtained. Especially for $A_x = 0.5$ and 0.25 the increase in heat transfer is much stronger than the increase as found for the two dimensional calculations. Moreover, contrary to the two dimensional calculations, where maximum heat transfer has been obtained for $\phi = 60$ degrees, the heat transfer still increases for decreasing $\phi$ in the three dimensional case. This change of heat transfer behaviour can not be due to the end wall effects, when considering the negligible effect the end walls show in the vertical situation and by comparison with the two dimensional case. Therefore the differences in heat transfer obtained for comparable cases in inclined situations with the two and three dimensional programme must be due to three dimensional flow effects. Obviously the flow becomes three dimensional in inclined situations. Even for $A_z = 10$ these three dimensional effects are important. This indicates that an important change in flow structure has taken place.

A considerable decrease in heat transfer has still been found by decreasing $A_x$ from 0.125 to 0.1. For complete suppression of natural convection at the parameter range considered, $Nu_B : 1.2$, $A_x$ should be chosen smaller than 0.1.

The heat transfer results for medium $A_z$ enclosures ($A_z = 1$) have been given in fig. 3.28.

For angles of 60 and 45 degrees (fig. 3.27b and c) complete suppression of natural convective heat transfer has no longer be accomplished for the Rayleigh range considered. Even for $A_x = 0.125$ and 0.1 natural convection develops and therefore increased natural convective heat transfer has been obtained. Especially for $A_x = 0.5$ and 0.25 the increase in heat transfer is much stronger than the increase as found for the two dimensional calculations. Moreover, contrary to the two dimensional calculations, where maximum heat transfer has been obtained for $\phi = 60$ degrees, the heat transfer still increases for decreasing $\phi$ in the three dimensional case. This change of heat transfer behaviour can not be due to the end wall effects, when considering the negligible effect the end walls show in the vertical situation and by comparison with the two dimensional case. Therefore the differences in heat transfer obtained for comparable cases in inclined situations with the two and three dimensional programme must be due to three dimensional flow effects. Obviously the flow becomes three dimensional in inclined situations. Even for $A_z = 10$ these three dimensional effects are important. This indicates that an important change in flow structure has taken place.

A considerable decrease in heat transfer has still been found by decreasing $A_x$ from 0.125 to 0.1. For complete suppression of natural convection at the parameter range considered, $Nu_B : 1.2$, $A_x$ should be chosen smaller than 0.1.

The heat transfer results for medium $A_z$ enclosures ($A_z = 1$) have been given in fig. 3.28.

Again for 90 degrees an almost two dimensional situation has been obtained. Only a slight decrease in heat transfer can be observed for all $A_x$ and $\phi$ in comparison with the large $A_z$ enclosures and the two dimensional calculations. This reduction in heat transfer is due to the increased effect of the end walls on the flow. For an inclination of 60 and 45 degrees a slight reduction of heat transfer has been obtained in comparison with comparable calculations for large $A_z$ enclosures. This is due to the increased end wall effect. In comparison with the two dimensional calculations, the increased heat transfer in the three dimensional case is once more a result of a change...
in flow structure. Since no large differences have been found with regard to \( A_z = 10 \) calculations, it must be concluded that with reduction of \( A_z \) to 1 no important suppression of the flow has been obtained. As has been obtained for \( A_z = 10 \) enclosures, for complete suppression of the natural convective heat transfer at the Rayleigh range of interest \( A_x \) should be chosen smaller than 0.1.

To determine the effect of the change in flow structure, we investigated the flow structure of the natural convection flow for the medium \( A_z \) enclosures. As already mentioned in chapter 3.2.4.1 three dimensional flow is characterized by streamtubes. Several methods have been applied to give a visual image of the flow structure of the three dimensional flow in the enclosure. Mallinson and de Vale Davis (1977) applied particle tracking to visualize the three dimensional calculated natural convection flow in a box. Chan and Banerjee (1979) made velocity diagrams at several \( xy \)-planes in the \( z \)-direction. In our case due to the \( z \)-symmetry the velocity field at the midplane \( z = 0.5 A_z \) is strictly two dimensional and can therefore be depicted with a streamline contour plot. As long as the main flow is in the \( xy \)-plane the obtained streamline contour plot gives a good qualitative image of the flow structure. However, when the main flow is no longer in the \( xy \)-plane other methods should be used. To visualize these flow patterns we made streamline contour plots at \( x = 0.5 A_x \) also. Interpreting these contour plots one should always remember that a two dimensional representation is given from a three dimensional flow and that flow perpendicular on the plane exists.

Based on the heat transfer results we expect that for \( \theta : 90 \) and small \( A_x \) the flow structure has been transformed from a transversal base flow (mainly in the \( xy \)-plane) to a flow in the \( yz \)-plane of the enclosure. We determined the streamfunctions for \( A_x = 0.5, 0.25 \) and 0.125 at the midplanes \( z = 0.5 A_z \) (results in \( \psi_{xy} \)) and at \( x = 0.5 A_x \) (\( \psi_{yz} \)) for the three angles of inclination. The maximum values of the streamfunctions, which indicate the flow intensity, have been given in table 3.3.

In addition the ratio between the maximum values of the streamfunctions have been given in the last column of the table. For the vertical situation the maximum value of the streamfunction in the \( xy \)-plane is much larger than for the \( yz \)-plane for all aspect ratios considered. For the inclined situations the streamfunction ratio is still larger than 1 and increases with decreasing \( \theta \) for \( A_x = 0.5 \). This means that the flow in the \( xy \)-plane remains relatively stronger in comparison with the flow in the \( yz \)-plane. Moreover \( \psi_{xy} \) increases with decreasing inclination angle. For \( A_x = 0.25 \) and

<table>
<thead>
<tr>
<th>( A_x )</th>
<th>( \theta )</th>
<th>( \psi_{xy} )</th>
<th>( \psi_{yz} )</th>
<th>( \psi_{xy} / \psi_{yz} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>90</td>
<td>2.02</td>
<td>0.58</td>
<td>3.512</td>
</tr>
<tr>
<td>60</td>
<td>9.78</td>
<td>2.08</td>
<td>4.527</td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>12.10</td>
<td>1.86</td>
<td>6.492</td>
<td></td>
</tr>
<tr>
<td>0.25</td>
<td>90</td>
<td>0.75</td>
<td>0.33</td>
<td>2.388</td>
</tr>
<tr>
<td>60</td>
<td>3.35</td>
<td>0.33</td>
<td>9.918</td>
<td></td>
</tr>
<tr>
<td>45</td>
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<td>2.77</td>
<td>1.51</td>
<td></td>
</tr>
<tr>
<td>0.125</td>
<td>90</td>
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<td>2.77</td>
<td>19.118</td>
</tr>
<tr>
<td>60</td>
<td>0.41</td>
<td>1.34</td>
<td>0.388</td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>0.62</td>
<td>1.89</td>
<td>0.33</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.3: Maximum values of the streamfunctions in the \( xy \) and \( yz \) planes for medium \( A_z \) enclosures.

0.125 a decrease of the inclination angle results in a stronger development of the flow in the \( yz \)-plane than in the \( xy \)-plane. For \( A_x = 0.25 \) the streamfunction ratio is smaller than 1. This indicates that the strongest flow occurs in the \( yz \)-plane. For \( A_x = 0.125 \) the same effect occurs for \( \theta = 45 \) and 60 degrees.

The \( yz \)-streamlines obtained for \( A_x = 0.25, A_z = 1 \) have been given in fig. 3.29. The hot wall is the lower horizontal wall of the enclosure. In the vertical situation only downwards directed flow near the side walls can be seen. At \( \theta = 60 \) degrees flow development in the center of the enclosure can be seen. The streamfunction ratio for this case is 0.91, what means that the flow in the \( yz \)-plane is approximately as strong as the flow in the \( xy \)-plane. For \( \theta = 45 \) degrees and 60 degrees the flow in the \( yz \)-plane is formed by a longitudinal flow pattern. Rolls with their axis in the \( z \)-direction of the enclosure, upslope the hot wall occur.

In conclusion decreasing the inclination angle results in a stronger base flow in the \( xy \)-plane of the enclosure and in addition also in flow development in the \( yz \)-plane of the enclosure in the form of longitudinal rolls. Especially for small aspect ratio enclosures (\( A_x : 0.25 \)) this superposed flow is as strong or even stronger than the base flow and contributes considerably to the natural convective heat transfer.

Finally the heat transfer results for small \( A_z \) enclosures \( (A_x = A_z) \) can be seen in fig. 3.30.

For \( \theta = 90 \) degrees and all \( A_x = A_z \) the heat transfer has been reduced in
Figure 3.29: Streamline contour plots for the yz-plane for \( A_x = 0.25 \) and \( A_z = 1 \).

Figure 3.30: Three dimensional heat transfer calculations \( \text{Nu}^* \) as function of the Rayleigh number, \( A_x^2 = A_x \) (b) \( A_z = A_x \), (c) \( A_x = A_z \).

Figure 3.31: Dependence of \( \text{Nu}^* \) on the angle of inclination \( \theta \) for Ra = 5 \( 10^5 \).

Comparison with the medium \( A_z \) enclosure. This reduction in heat transfer is due to the increased effect of the end walls. This effect becomes stronger with decreasing aspect ratios \( (A_x = A_z) \). The effect can be seen very clearly if one compares the \( A_x = A_z \) three dimensional calculations with the two dimensional calculations for the vertical situation (see fig. 3.38a). For an aspect ratio of 0.5 the effect is still small. For high Rayleigh values this effect is even negligible. However for \( A_x = A_z = 0.25 \) the Nusselt number for the three dimensional calculation for Ra = \( 10^6 \) amounts to 1.320, whereas for the comparable two dimensional calculation \( \text{Nu}^* = 2.896 \). For smaller aspect ratios no notifiable effect has been obtained for Ra less than \( 10^5 \). For these aspect ratios the convection has been suppressed totally for both the two dimensional and the three dimensional calculations.

At angles of inclination of 60 and 45 degrees important heat transfer reduction has been found by decreasing the aspect ratio from 0.25 to 0.125. For \( A_x = A_z = 0.5 \) and 0.25 the influence of the aspect ratio on the heat transfer is still negligible in comparison with the medium \( A_z \) enclosures. It might be stated that for all aspect ratios considered a decrease of \( A_z \) delays the onset of natural convection and decreases the natural convective heat transfer. However to reduce the effect of the three dimensional natural convection on the heat transfer at the chosen Rayleigh range \( (10^4 \leq \text{Ra} \leq 10^6) \), honeycombs should be chosen with aspect ratios of at least 0.1.

Contrary to the two dimensional calculations, the influence of the angle of inclination on the heat transfer has been found to be very significant for all enclosures considered.

The dependence of \( \text{Nu}^* \) on \( \theta \) is shown in fig. 3.31 for Ra = 5 \( 10^5 \). For three dimensional enclosures a decrease of \( \theta \) from the vertical situation gives rise to a significant increase of natural convective heat transfer. Since this effect has not been found for two dimensional flow and according to fig. 3.31a, b, c no great influence of \( A_z \) has been obtained, this effect...
must arise from a change in flow structure as explained before. For a certain angle of inclination, the critical angle \( \theta_c \), the almost two dimensional transversal base flow, which is present for an angle of inclination of 90 degrees, changes into a three dimensional flow structure. Decreasing the angle of inclination to values smaller than \( \theta_c \) should intensify only the development of this flow. This results in an increase of the Nusselt number.

If one considers the critical Rayleigh number (Ra at which natural convective heat transfer becomes important, i.e. \( \mathrm{Nu}_B > 1.2 \)) as function of the aspect ratio \( A_x \), several deviations from the two dimensional calculations have been found. Due to the strong \( \mathrm{Nu}_B \) dependence, separate figures have been made for the different angles of inclination.

In fig. 3.32 a,b,c the \( \mathrm{Ra}_c - A_x \) dependence is shown for \( \theta = 90, 60 \) and 45 degrees, respectively.

Further in each figure the different types of enclosures have been distinguished. For comparison purposes the two dimensional obtained values for \( \mathrm{Ra}_c \) have been also noted in the figure. For \( \theta = 60 \) and 45 degrees critical Rayleigh numbers have been found for three aspect ratios at the Rayleigh range considered for each enclosure. Despite the inaccurate way for determination of \( \mathrm{Ra}_c \), a single power law dependence of \( \mathrm{Ra}_c \) on \( A_x \) seems justified.

The \( \mathrm{Ra}_c - A_x \) relations obtained for \( \theta = 60 \) and 45 degrees, have been given in table 3.9.

<table>
<thead>
<tr>
<th>( \theta )</th>
<th>( \mathrm{Ra}_c = a A_x^b )</th>
</tr>
</thead>
<tbody>
<tr>
<td>60</td>
<td>( A_x = 10 )</td>
</tr>
<tr>
<td>60</td>
<td>( A_x = 1 )</td>
</tr>
<tr>
<td>45</td>
<td>( A_x = A_{x'} )</td>
</tr>
</tbody>
</table>

Table 3.9: Critical Rayleigh number relations with \( A_x \).

Since the relations given in table 3.9 have been based only on three points, the coefficients "a" and "b" shall be very sensitive for small deviations from the method used to determine \( \mathrm{Ra}_c \). However some conclusions can be made from fig. 3.32 and table 3.9.

First of all it has been found, as expected, that for each \( A_x \) at any angle of inclination the critical Rayleigh number decreases with increasing \( A_x \). The differences between the obtained \( \mathrm{Ra}_c \) for \( A_x = 1 \) and \( A_x = 10 \) are small. This indicates once more that suppression of natural convection by cell structures shall be only efficient for honeycombs with \( A_x < 1 \). Secondly for \( \theta = 90 \) degrees the obtained \( \mathrm{Ra}_c \) for the two and three dimensional cases become equal for large \( A_x \). For inclined situations the two dimensional programme overestimates the \( \mathrm{Ra}_c \), due to the development of the three dimensional flow pattern, which can not be represented with a two dimensional programme.

3.2.5.3 Comparison and conclusions of flow calculations.

If we want to compare the two and three dimensional calculated results, we have to distinguish between the vertical (\( \theta = 90 \)) and the inclined situation (\( \theta = 60, 45 \) degrees). The flow in the vertical situation is mainly in the xy-plane of the enclosure, but for inclined enclosures of small aspect ratio \( A_x \) (\( A_x < 0.25 \)) the flow orientation changes and forms also longitudinal rolls in the yz-plane of the enclosure. The longitudinal rolls shall only be suppressed sufficiently by decreasing \( A_x \) to values smaller than one.
Due to this change in flow pattern, the critical Rayleigh numbers obtained with the three dimensional programme are smaller than the two dimensional calculated $Ra_2$ for inclined situations. With the two dimensional computer programme only flow in the xy-plane can be calculated. Therefore our results obtained with the two dimensional programme do not give correct solutions for the inclined situations and small $A_x$. For the vertical situation a comparison of two and three dimensional heat transfer results have been given.

Figure 3.33: Comparison between two and three dimensional results.

Figure 3.33a shows the heat transfer results for $A_x = 0.5$ and for different $A_z$ values. The two dimensional results ($A_z = m$) and the results for $A_z = 10$ lie so close to each other that it is impossible to distinguish. For smaller $A_z$ larger suppression of natural convection has been obtained, especially in the transition regime from base flow to boundary layer flow. At higher Rayleigh values the differences between the different $A_z$ enclosures are much smaller. For instance at $Ra = 5 \times 10^4$, $A_x = 0.5$, $\theta = 90$ the obtained Nusselt values amount to 1.788, 1.761, 1.598 and 1.378 for $A_z = m$, 10, 1, 0.5 respectively, whereas at $Ra = 5 \times 10^5$ $Nu = 4.926$, 4.904, 4.775 and 4.584 for the same aspect ratios, respectively.

In fig. 3.33b the results for $A_x = 0.25$ are shown. What applies for $A_x = 0.5$ is valid also for $A_x = 0.25$. Again two dimensional and $A_z = 10$ results are indistinguishable and decreasing $A_z$ results in decreased natural convective heat transfer. For enclosures with $A_x \leq 0.125$ complete suppression has been found for all $A_z$ enclosures in the vertical situation. Since we are interested mainly in suppression of natural convection in order to minimize the heat transfer, this goal has been satisfied for all enclosures with $A_x \leq 0.125$, $\theta = 90$ degrees. In this case the influence of the depth to width aspect ratio $A_z$ has been found very small. For angles of inclination of 45 and 60 degrees the heat transfer increases significantly for each enclosure considered. This increment of heat transfer is due to the change of flow structure. This flow structure can be represented only by a three dimensional convection model. Total suppression of natural convective heat transfer, at the Rayleigh range considered, can be obtained only for cell structures with $A_z = A_2 \leq 0.10$.

However for large $A_2$ enclosures the natural convective heat transfer for $A_2 = 0.10$, $\theta = 45$ degrees has been suppressed up to $5.4 \times 10^5$ and at $Ra = 10^6$ the Nusselt number amounts 1.486 only. Therefore slit structures with aspect ratio $A_x = 0.10$ and $A_2 \geq 10$ inclined at angles to 45 degrees shall be a good alternative for honeycombs at the Rayleigh range considered.

3.3 Computation of radiation.

3.3.1 Formulation of the problem.

As described before (chapter 3.1) the total heat transfer between the absorber and cover plate of a solar collector, in which a convection suppression device is used, is due to convective, radiative and conductive heat transfer simultaneously.

In this chapter the radiative heat transfer shall be described. We consider radiative heat transfer in slit structures only, based on the conclusion, made in chapter 3.2.5.3, that slit structures of aspect ratio $A_x = 0.1$ suppress the natural convection at the parameter range considered efficiently.

If slit structures are used a two dimensional analysis of radiation is sufficient to calculate the radiative heat transfer. The air in the enclosure is considered to be a non-participating medium, i.e. a medium that does not absorb, emit or scatter radiation and hence has no effect on the radiation traveling through it. Radiative exchange occurs only between the boundary surfaces. The properties ($\alpha$, $\epsilon_p$ and $T$) of the walls of the enclosure have been taken as total values at the wavelength interval of the black radiation spectrum for the average operating temperature ($T_0 = 318$ K).
3.3.2 The simplified zone analysis.

In general, the radiative properties of the surfaces of the enclosure may vary and also the temperature may vary from point to point over the surfaces. This makes the calculation of the radiative heat transfer in the enclosure very complicated. The analysis can be simplified by dividing each surface of the enclosure into a finite number of zones. The following conditions are assumed to be satisfied at the surface of each zone: (1) The radiative properties are uniform; the surfaces are gray. (2) A uniform temperature is prescribed for each zone. (3) The surfaces are diffuse emitters and diffuse reflectors. (4) The radiant energy leaving the zone (the radiosity) is uniform over each zone. (5) The surfaces are opaque, i.e., \( P_d = 1 - \alpha \).

We used the absorption factor method as developed by Gebhart (1958) to calculate the radiative heat flux at each zone of the enclosure. In our case two surfaces of the enclosure have been assumed isothermal at temperatures \( T_H \) and \( T_C \). The two other surfaces (the slit structure walls) have a temperature distribution \( T_W \) with \( T_C \leq T_W \leq T_H \). The temperature distribution is determined by conduction, convection, and radiation. To calculate this slit wall temperature profile it is necessary to calculate the radiative heat flux absorbed by each zone. With the absorption factor method the net radiative heat flux at each zone can be determined as a function of the temperature.

### 3.3.2.1 Gebhart's method.

The absorption factor method, as developed by Gebhart (1958; 1971), calculates the net radiant energy loss from a surface. For our case, due to cell symmetry, radiative heat exchange shall be assumed to be equal in every cell. This means that one cell has to be considered only. Consider the enclosure as shown in Fig. 3.34.

Two isothermal walls (temperatures \( T_H \) and \( T_C \)) have been connected by two walls of the slit structure material with unknown temperature distribution. Although the slit structure walls possess a finite thickness, the surfaces of the slit structure material, which form the enclosure, have been drawn only. The radiative heat transfer is determined by the wall temperatures. The surfaces of the enclosure have been divided into zones. Consider the \( j \)-th zone with surface area \( A_j \). If all the surfaces are black (\( \varepsilon_i = 1 \)) the rate \( q_j \) of radiant energy loss from surface \( A_j \) is given by:

\[
q_j = \varepsilon_j \sigma T_j^4 A_j - \sum_{i=1}^{N} F_{ij} \varepsilon_i \sigma T_i^4 A_i
\]  

(3.48)

where the summation is taken over all the zones \( N \) of the enclosure; \( \sigma \) is the constant of Stefan-Boltzmann and \( F_{ij} \) is the angle or view factor from zone \( i \) to zone \( j \), defined as the fraction of the radiant energy emitted by a black surface \( i \) which falls directly upon a black surface \( j \).

For gray surfaces the amount of radiative energy emitted by the \( i \)-th zone and absorbed by \( A_j \) is not equal to \( F_{ij} \), due to reflections at all the zones of the enclosure. Gebhart defines an absorption factor \( G_{ij} \) as the total fraction of the emitted energy by \( A_j \) that is absorbed by \( A_i \), taking into account all possible reflections whereby this energy may reach \( A_i \). According to this definition eq. (3.48) reads for gray enclosures:

\[
q_j = \varepsilon_j \sigma T_j^4 A_j - \sum_{i=1}^{N} G_{ij} \varepsilon_i \sigma T_i^4 A_i
\]  

(3.49)

For any zone \( A_j \) of the enclosure we need to determine \( N \) absorption factors \( G_{ij} \) to calculate the net radiant heat loss from eq. (3.41). Moreover, there are \( N \) zones in the enclosure. This results in \( N \times N \) absorption factors for the enclosure, given by the matrix \( G \).
Since all the surfaces have been assumed to be opaque, the energy emitted by each zone \( A_i \) shall, for a stationary case, be absorbed by the \( N \) zones (including \( A_i \) itself). This means that the sum of each row of the matrix \( G \) is equal to one.

\[
\sum_{j=1}^{N} G_{ij} = 1
\]  

(3.43)

If we know the temperatures \( T_i \) of each zone, the net radiative flux at zone \( A_j \) can be calculated if the absorption factors \( G_{ij} \) have been determined for each zone. For example let us consider \( G_{1j} \), i.e. the energy emitted by \( A_1 \) that reaches \( A_j \) by any path and is absorbed. The emitted energy of \( A_1 \) shall be absorbed directly by \( A_j \) for the fraction \( F_{1j} \). In general at the \( i \)-th zone the fraction \( F_{ij} \) shall be reflected. Now considering the assumptions of the simplified zone analysis the fraction of the energy of \( A_1 \) which is diffusely reflected by \( A_i \), and is absorbed by \( A_j \), is given by \( F_{1i} G_{ij} \). A summation of all the fractions of reflected energy, emitted by \( A_1 \) which are absorbed by \( A_j \), determine \( G_{ij} \):

\[
G_{ij} = F_{1j} + F_{1i} P_{di} G_{ij} + F_{12} P_{d2} G_{2j} + \cdots + F_{IN} P_{dN} G_{Nj}
\]  

(3.44)

A similar relation can be written for each zone, resulting in \( N \) linear equations with \( N \) unknowns; \( G_{1j} \), \( G_{2j} \), \( \ldots \), \( G_{Nj} \).

\[
\begin{bmatrix}
G_{11} & G_{12} & \cdots & G_{1N} \\
G_{21} & G_{22} & \cdots & G_{2N} \\
\vdots & \vdots & \ddots & \vdots \\
G_{N1} & G_{N2} & \cdots & G_{NN}
\end{bmatrix}
= \begin{bmatrix}
G_{1j} \\
G_{2j} \\
\vdots \\
G_{Nj}
\end{bmatrix}
\]  

(3.42)

This set of equations is valid for each zone. Therefore a total number of \( N \times N \) equations determine the absorption factors.

For the determination of the absorption factors \( G_{ij} \) it is convenient to write the \( N \) sets of \( N \) linear equations in matrix form.

Equation 3.45 applied on all the zones then becomes:

\[
\begin{bmatrix}
F_{11} P_{d1} & F_{12} P_{d2} & \cdots & F_{1N} P_{dN} \\
F_{21} P_{d1} & F_{22} P_{d2} & \cdots & F_{2N} P_{dN} \\
\vdots & \vdots & \ddots & \vdots \\
F_{N1} P_{d1} & F_{N2} P_{d2} & \cdots & F_{NN} P_{dN}
\end{bmatrix}
= \begin{bmatrix}
G_{11} \\
G_{21} \\
\vdots \\
G_{N1}
\end{bmatrix}
\]  

(3.45)

This equation can be written shortly as:

\[
(\vec{P}_d F \cdot \vec{G}) = (\vec{F} \cdot \vec{G})
\]  

(3.46b)

In this equation matrix \( \vec{G} \) is the unknown. The view factors \( F_{ij} \) are determined by the configuration of the enclosure and \( P_d \) and \( \varepsilon \) are properties of the surface material. Solving matrix \( \vec{G} \) from equation (3.46b) has been done.
in the usual way by inversion of matrix \( P_d F \) by means of a LU decomposition method. This results in

\[
G = (P_d F)^{-1} \epsilon \in F
\]

(3.47)

A reciprocity relation between the absorption factors has been found by Gebhart (1971). This reciprocity relation can be found straightforward if one considers the net radiative heat exchange between two surfaces \( A_i \) and \( A_j \). The net radiative heat flux \( q_{ij} \) is equal to zero for equal temperatures \( T_i = T_j \). This means that

\[
G_{ij} \epsilon_i A_i \epsilon_T T_i^4 = G_{ji} \epsilon_j A_j \epsilon_T T_j^4
\]

(3.48a)

with \( T_i = T_j \) the obtained reciprocity relation reads:

\[
G_{ij} \epsilon_i A_i = G_{ji} \epsilon_j A_j
\]

(3.48b)

This reciprocity relation reduces the number of unknown absorption factors and has been used to simplify the problem. By combining equation (3.48b) with eq. (3.41) the net energy loss of zone \( A_j \) can be written as,

\[
q_j = \epsilon_j \sigma A_j (T_j^4 - \sum_{i} G_{ij} T_i^4)
\]

(3.49)

Using the summation rule of eq. (3.43) the determination of the net energy loss of each zone can be written as:

\[
q_j = \epsilon_j \sigma A_j (\sum_{i} G_{ij} T_i^4 - \sum_{i} \sum_{j} G_{ji} T_j^4)
\]

(3.50)

By taking the two summations together, then linearizing the temperature difference and again using the summation rule, the net energy loss \( q_j \) will be finally determined by:

\[
q_j = 4 \epsilon_j \sigma A_j (T_j - \sum_{i} G_{ji} T_i)
\]

(3.51)

with the mean temperature equal to \( T_\theta = (T_i + T_j) / 2 \). The mean temperature has to be determined actually between the \( i \)-th and \( j \)-th zone temperature \( = (T_i + T_j) / 2 \) and is not a constant under the summation sign. We used in our analysis a constant mean temperature approximated by the average of the extreme temperatures \( T_i \) and \( T_j \). It is only possible to use the linear form of the radiation fluxes as long as the temperature differences between the zones are relatively small. In our case extreme temperature differences of 70 K might be expected and therefore large deviations from the 4-th power determined radiative fluxes shall not occur (at averaged temperatures of 318 K the difference is less than 2%). When the absorption factors have been determined the net radiative heat loss for each zone has been reduced to a linear function of the temperatures of all the zones.

3.3.2.2 The view factor.

According to eq. (3.47) the absorption factors are determined by radiative properties of the zones and the view factors. The view factors themselves are only functions of the configuration of the enclosure. For surfaces which are diffuse emitters and diffuse reflectors the view factor has been defined as the fraction of radiant energy leaving zone \( i \) that falls directly on zone \( j \) and is given by \( F_{ij} \). For the case of black zones the absorption factors and the view factors are equal. A reciprocity relation exists for view factors given by

\[
A_{i} F_{12} = A_{2} F_{21}
\]

(3.52)

Moreover in the case of opaque enclosures a summation rule for view factors can be also defined. For an enclosure of \( N \) zones, all the energy leaving any zone inside the enclosure, for example \( A_i \), shall be incident on all the zones of the enclosure.
The determination of the view factor results for the case of two finite areas in the integration of the fraction of radiative energy of area 1 that reaches directly area 2 for the zones \( A_1 \) and \( A_2 \). For a two dimensional situation, as in our case, Hottel (1954) developed a simplified method to determine the view factor. This method is known as the "crossed string" method. The view factor \( F_{12} \) is determined by

\[
F_{12} = \frac{(ad+bc)-(ac+bd)}{2ab}
\]

Figure 3.35: Hottel's crossed string method.

3.3.2.3 Specular reflections.

The simplified zone analysis, as described above, is only applicable for surfaces which emit and reflect radiant energy diffusely. Real surfaces, however, possess both diffuse and specular reflectivity components \( P_d \) and \( P_s \), respectively. Moreover, in our case a significant transmissivity component \( T \) might be expected, due to the kind of material and its thickness we use as slit structure walls. Due to cell symmetry the transmissivity component can be equated to a specular reflectivity component as can be seen in Fig. 3.36a.

This results in a total reflectivity coefficient \( P_t \) for the slit structure walls, which can be separated in an effective specular and diffuse component.

\[
P_t = P_d + P'_s
\]

\[
P'_s = P_s + T
\]

To modify the absorption factor method for use in enclosures having specular reflective walls, it is necessary to account also for the radiant energy, which is leaving zone \( A_j \) and reaches \( A_i \) by one or more specular reflections. Eckert (1961) and Sparrow (1962) were the first to apply the so-called 'mirror image' method for the determination of the total view factor, which accounts for both diffuse and specular reflections. To comprehend the mirror image method let us consider the radiant energy emitted by \( A_i \) which reaches \( A_j \), as shown in Fig. 3.36b. The fraction of radiant energy leaving zone \( A_1 \), which falls directly on \( A_j \) is determined by \( F_{ij} \). The fraction of radiant energy leaving zone \( A_1 \), which falls on \( A_j \) after one specular reflection on surface \( A_2 \) can be regarded as energy originating from zone \( A_1(2) \). This is the mirror image of zone \( A_1 \) through mirror \( A_2 \). The fraction is determined by \( P'_{s2} F_{i(2)j} \), which is the black view factor from...
zone \( A_1(2) \) to \( A_j \) (seen through mirror \( A_2 \)) multiplied by the effective specular reflectivity of the mirror \( A_2 \). Likewise is the fraction of radiant energy that reaches \( A_j \) after successive specular reflections on surfaces \( A_4 \) and \( A_2 \) coming from zone \( A_2(4) \). This is the mirror image of \( A_2(4) \) through mirror \( A_2 \) (where \( A_2(4) \) is the mirror image of \( A_4 \) through mirror \( A_2 \) ) and this fraction is given by \( P'S_2 P'S_2 F_{1(4)} \). More successive specular reflections can be taken into account by adding more mirror image cells. The fraction of radiant energy leaving zone \( A_j \) which falls on \( A_j \) directly or after specular reflections, can now be represented in a total view factor given by

\[
F_{ij} = F_{ij} + P'S_2 F_{1(2)} + P'S_4 P'S_2 F_{1(4)} + \cdots \quad (3.56)
\]

Replacement of the black view factors \( F \) by the total view factors \( F' \) in the equation for the absorption factor (eq. (3.47)) gives the absorption factors, which have been corrected for specular reflections (see Ozisik (1973)).

The calculation of the net radiative heat flux for each zone follows according to the method described in chapter (3.3.2.1).

In the above given analysis concerning specular reflectivity the specular reflective components of the hot and cold wall have been neglected. The main reason for this simplification is that the area of the hot and cold wall together amounts to less than 5% of the total enclosure area for a slit of aspect ratio 6.1. Moreover the specular reflectivity of the cold wall (glass cover of the solar collector) shall normally be less than 0.05.

Without this simplification the mirror image concept as given in fig. 3.36b must be extended with mirror image cells not only neighbouring surfaces \( A_2 \) and \( A_4 \) but also neighbouring \( A_1 \) and \( A_3 \) and their mirror images \( A_1(2) \), \( A_3(2) \), \( A_1(3) \), \( A_3(3) \) .......

As is obtained straightforward from eq. (3.56) if all effective specular reflectivity coefficients \( P'S \) are equal to zero the total view factor is equal to the black view factor. The number of specular reflections that has to be considered is dependent on the value of the effective spectral reflectivity \( P'S \) of the side walls. For high specular reflective side walls \( P'S = 1 \) a number of at least 200 successive specular reflections have to be considered to obtain a solution within 0.1% accuracy for \( A_2 = 0.1 \). For low effective specular reflectivity \( P'S \leq 0.1 \) ten specular reflections shall be sufficient to reach this accuracy.

3.4 Coupled heat transfer.

In the foregoing chapters the different modes of heat transfer, which occur in a slit structure cell, have been discussed separately; i.e.

natural convective heat transfer by air and the radiative heat transfer at the walls of the enclosure. Neither heat transfer modes have taken into account the finite thickness of the slit structure material and the conductive heat transfer through this material. However, as discussed earlier in chapter 3.1, the temperature distribution in the side walls of the enclosure influences directly the radiative heat transfer. As the side wall boundary condition influences also the natural convection in the slit, a coupled heat transfer analysis is necessary to calculate the total heat transfer in the slit. We considered coupled heat transfer in slit structures of aspect ratio 0.1. As noted before a three dimensional calculation of natural convection would be essential for inclined situations at \( A_2 = 0.1 \). However we are mainly interested in cases where natural convective heat transfer is suppressed sufficiently. If the critical Rayleigh number is not affected by the coupled boundary condition for the two dimensional programme it might be expected that the radiative heat transfer does not affect \( R \) as obtained with the three dimensional programme either. As long as \( R \leq R_c \) suppression of natural convective heat transfer can be assumed.

In the following chapters two different cases of convective heat transfer by the air in the cell shall be distinguished:

(1) stagnant air; the natural convection of air in the cell has been suppressed completely by the slit structure and conduction only has been taken into account for the heat transfer by the air.

(2) natural convection of air; flow development and natural convective heat transfer have been considered.

With the second model the influence of the radiation on the onset of convection can be determined.

Calculations for both cases have been restricted to two dimensions only.
3.4.1 The boundary condition at the side walls.

To calculate the temperature distribution of the side walls we consider a wall element (see fig. 3.37).

The temperature of the element is determined by heat conduction of the slit structure material through the boundaries $A_s$ and $A_n$ and by the radiative and convective (or conductive) heat transfer at $A_s$ and $A_n$, which are the outer surfaces of the side wall. As a consequence of the small thickness of the slit structure material ($h < 10^{-4}$ m) an uniform temperature across the wall has been assumed. For stationary conditions the net heat transfer is zero, resulting in the equation:

$$
\lambda_w \frac{\partial}{\partial y} A_s (\frac{\partial q''}{\partial y} + \frac{\partial q''}{\partial s}) = q''_w + q''_{cd,s} n + q''_{cd,s} w + q''_{cv,s} e + q''_{cv,w} e
$$

For our problem $A_h = A_s$ and $A_e = A_n$. In principle the temperature of the wall element influences all the heat fluxes. Due to the complex interaction of the temperature distribution of the side wall with the radiative and convective heat fluxes a direct solution is complicated. For the numerical programme an iterative procedure has been chosen to determine the wall temperature distribution. For an iteration the radiative and convective heat fluxes have been calculated using the temperature distribution of the previous iteration. With this assumption the new dimensionless wall temperature ($\Theta_w$) appears only in the wall conduction fluxes.

$$
\Theta_w = \Theta_{w-1} + \gamma_w (\Theta_w - \Theta_{w-1})
$$

(3.59)

Here $i$ denotes the iteration number and $\gamma_w$ is the underrelaxation factor. Very strong underrelaxation was needed to reach a solution. Underrelaxation factors of 0.1 for $h_w = 10^{-4}$ m down to 0.81 for $h_w = 25 \times 10^{-6}$ m have been used. By using optimum underrelaxation solutions have been obtained normally within 30 iterations. The calculation procedure that we used is given more in detail in Appendix B.

It should be noted that with the introduction of the coupled convective-radiative heat transfer no use can be made any longer of the centro-symmetry property to calculate the natural convection. To use symmetry properties it is essential that besides the equations the boundary conditions possess the symmetry properties. With the introduction of $\xi_h$ and $\xi_c$ this requirement shall not be fulfilled generally. Only when $\xi_h = \xi_c$ and radiation is calculated with a linearized model use can be made of the centro-symmetry property.

Using the calculation procedure described above a solution has been reached normally within 280 CPU seconds on an Amdahl 478/7 underrelaxation. However when convection was totally suppressed and only conduction through air occurs a solution was reached within 20 CPU seconds by solving the implicit Laplace
3.4.2 Test of the calculation method.

To test the calculation procedure for coupled heat transfer in the enclosure we considered first the radiative heat transfer. For the case of diffuse and specular reflective walls the summation rules for the absorption and view factors as given in eq. (3.43) and eq. (3.53) have been checked. For completely specular reflective side walls a number of 206 successive specular reflections have been taken into account before the summation rules were satisfied within 0.1%. For diffuse side walls such deviations obviously did not occur. According to Gebhart (1961) any opaque adiabatic surface, which is in radiant balance in an enclosure, achieves a steady state temperature independent of the emissivity of the surface. We checked this by calculating the side wall temperature distribution with the diffuse model taking into account radiative heat transfer only. Indeed the side wall temperature distribution was independent of the emissivity of the side wall. The obtained temperature distributions for BB, SB, and SS cases have been given in fig. 3.38 for different $A_x$ values.

The discontinuities for the temperature at $y = 0$ and $y = 1$ are a consequence of the fact that only radiant energy exchange has been considered. In comparison with the linear conduction temperature profile radiative heat exchange tends to steep temperature gradients near the hot and cold wall. For the SB-cases a strong non-symmetrical temperature profile has been obtained due to the non-symmetrical boundary condition for the radiative heat transfer at the hot and cold wall ($\varepsilon_h = 0.41; \varepsilon_c = 1.0$). For the BB-case increasing aspect ratio $A_x$ tends to a more and more isothermal side wall. For the SB- and SS-cases the side wall temperature distribution is almost independent on $A_x$ in our range of aspect ratio.

In fig. 3.39 the influence of $A_x$ on the radiative heat transfer between the hot and cold wall have been depicted.

In the figure the fraction of radiative heat transfer has been given in comparison with the theoretical infinite parallel plates model ($\eta_{rd,\infty}$) as a function of $A_x$. As shown in the figure large radiative heat transfer reduction might be expected for diffuse reflective side walls for the BB-case, especially for $A_x < 1$. For the SB- and SS-cases the influence of the aspect ratio is much smaller and negligible for $A_x > 1$. Large reduction might be expected for $A_x < 0.1$.

For purely specular reflective side walls ($P_s^r = 1$) the radiative heat transfer should be equal to the radiative heat transfer as calculated with the infinite parallel plates model and independent on the aspect ratio $A_x$. We checked this by increasing the number of specular reflections, which had
to be taken into account to reach solutions within 0.1% accuracy (for example 289 specular reflections have been considered for $A_\tau = 0.1$, $P'_s = 1$). Moreover for each case the diffuse and specular model should give equal solutions for $\varepsilon_{hc} = 1$, whereas for $\varepsilon_{hc} = 0$ ($P'_d = 1$ and $P'_s = 1$ for the diffuse and specular model respectively) non-coupled heat transfer should be obtained. For this particular case the specular model should give a solution equal to the one given by the independent mode analysis.

Our programme satisfied all tests.

The optimum number of grid points for radiation calculations has been determined before starting the parameter variation investigation. Grids for radiative and convective (or conductive) heat transfer have been determined in a likewise manner as for the non-coupled convection calculations (see chapter 3.2.3.3). Grid parameter variation has been performed for the SS-case. The non-linearity parameters $\alpha_x$ and $\alpha_y$ have been chosen both equal to 1.5. Mesh refinements in both $x$- and $y$-direction have been carried out until the total heat transfer coefficient varied less than 0.1% in comparison with a coarser grid. In this manner the optimum grid for the coupled conduction-radiation calculations has been determined as $N_l = 7; N_l = 21$, and for the coupled convection-radiation calculations a $13x21$ grid has been used for $A_\tau = 0.1$. The radiative heat transfer has been calculated with equal grids as for the convection or conduction calculations, although coarser grids were sufficiently fine.

3.4.3 Coupled conduction-radiation.

Calculations of the coupled conductive-radiative heat transfer have been basically concerned with parameter variation of the optical quantities $\varepsilon_{hc}$ and $P'_s$, for the slit structure walls of the BB-, SB- and SS-cases. For the calculations a standard slit has been chosen with $A_\tau = 0.10$, $D = 0.04$ m, $T_0 = 318$ K, side wall thickness $h_w = 25 \times 10^{-6}$ m and a thermal conductivity of the side walls equal to that of FEP teflon, $\lambda_w = 0.245$ W/mK.

By introducing a constant average temperature $T_0$ the heat transfer coefficient $h_t$ becomes independent of the temperature difference between the hot and cold wall. This is due to the linear form of the radiation model. We considered the total heat transfer coefficient ($h_t$) as function of the emissivity of the side walls for the BB-, SB- and SS-case. Results have been given for specular reflective ($P'_s = 1 - \varepsilon_{hc}$, $P'_d = 0$) and diffuse reflective side walls ($P'_d = 1 - \varepsilon_{hc}$, $P'_s = 0$). Moreover distinction must be made between total specular reflective CSW walls ($P'_s = 1$), which results in a $h_t$ value equal to the one calculated according to the IMA (non-coupled heat transfer and radiation calculated according to the infinite parallel plate model) and total diffuse reflective CSW walls ($P'_d = 1$) non-coupled heat transfer but radiation calculated with the absorption factor method with $P'_s = 1$). This case has been called the non-coupled case.

The results of the calculations have been given in fig. 3.40. The total heat transfer coefficients calculated for the BB-case with the specular model decrease with increasing $\varepsilon_{hc}$ and are always smaller than the value calculated according to the independent mode analysis. For increasing $\varepsilon_{hc}$ the slit structure acts more and more as a thermal radiation shield. Only for $\varepsilon_{hc} = 0$ $h_t$ is equal to the value calculated with the independent mode analysis (IMA).

For diffuse reflective side walls the total heat transfer coefficients are much smaller than for the independent mode analysis even for $\varepsilon_{hc} = 0$. Moreover for the diffuse case the $h_t$ value is almost independent of $\varepsilon_{hc}$. Both effects are due to the decrease of radiative heat transfer. As a consequence of the diffuse reflecting side walls radiation is reflected.
Maximum heat transfer occurs for \( \xi = 9.3 \) and \( \xi = 9.5 \) for the SB- and SS-case. Also given in fig. 3.40 is the dependence of \( h_t \) for the SB- and SS-case. The obtained results are quite different in comparison with the BB-case. For the specular reflectivity model a significant increase of \( h_t \) occurs initially for increasing \( \xi_{hc} \). The heat transfer is even larger than calculated with the independent mode analysis (\( \xi_{hc} = 0; P_s = 1 \)). Maximum heat transfer occurs for \( \xi_{hc} \approx 0.8 \) and \( \approx 0.5 \) for the SB- and SS-case respectively. A further increase of \( \xi_{hc} \) decreases \( h_t \).

The results for the SB- and SS-case obtained with the diffuse model show after an initial increase of \( h_t \) for \( \xi_{hc} \leq 0.2 \) a more gradual increase of \( h_t \) with \( \xi_{hc} \) takes place. As shown in fig. 3.40 the influence of \( \xi_{hc} \) on \( h_t \) is most significant for the specular reflection model. For the BB-case an increase of \( \xi_{hc} \) decreases \( h_t \) significantly. For the SB- and SS-case results an increase of \( \xi_{hc} \) in an increase of \( h_t \) at small values of \( \xi_{hc} \) and in a decrease of \( h_t \) at larger \( \xi_{hc} \). For these cases and \( \xi_{hc} = 0 \) the total heat transfer is always larger than predicted according to the independent mode analysis.

The values of \( h_t \) obtained with the diffuse model are always smaller than the specularly predicted \( h_t \) values (at fixed \( \xi_{hc} \)), except for \( \xi_{hc} = 1 \), where they equal because all reflectivity components are zero. The specularly predicted \( h_t \) values for the SB-case are even larger than the diffusely predicted values for the BB-case over a large range of \( \xi_{hc} \) (0.07 \( \leq \xi_{hc} \leq 0.72 \), see fig. 3.48). Likewise the \( h_t \) values predicted with the specular model for the SS-case are larger than the diffusely predicted \( h_t \) values for the SB-case (0.15 \( \leq \xi_{hc} \leq 0.67 \)).

For a given slit structure material, with a specific \( \xi_{hc} \) and \( P_s \), the smallest total heat transfer shall occur in the situation with the smallest radiative heat exchange in the enclosure. So the SS-case gives a lower \( h_t \) value than the SB-case, which shall be lower than the \( h_t \) value for the BB-case. These differences decrease for \( \xi_{hc} \) near 1.

Although the slit structure reduces the heat transfer for the BB-case in comparison with the independent mode analysis, the heat transfer for the SB- and SS-case is enlarged for stagnant air conditions. However in comparison with non-honeycomb collectors, where natural convection has not been suppressed, reduction of heat total transfer has still been obtained. These results are qualitatively in good agreement with the results of the analytical model of Hollands (1979), see chapter 3.2.2 fig. 3.12.

To comprehend the coupled heat transfer process in the slit structure let us consider the radiative, air conductive and wall conductive heat fluxes separately as obtained for the coupled heat transfer calculation. In fig. 3.41a-h the heat transfer coefficients have been given once again. However in each figure we have separated the wall conductive, air conductive and radiative heat transfer at the isothermal walls.

For the BB- and SS-cases the heat transfer coefficients are equal at the hot and cold wall, due to symmetrical radiative properties at the isothermal walls (\( \xi_h = \xi_c = 1.0 \) for the BB-case and \( \xi_h = \xi_c = 0.1 \) for the SS-case). For the SB-case the heat transfer distributions at the hot and cold wall have been given separately.

In fig. 3.41a the heat transfer distribution for the specular reflective BB-case has been given (the figure accounts for the hot and cold wall distribution). The reduction of the total heat transfer (solid line) with increasing \( \xi_{hc} \) is due to reduction of the radiative heat losses of the hot and cold wall. In comparison with the heat losses predicted by the independent mode analysis (\( \xi_{hc} = 0 \)), the wall and air conductive heat losses at the isothermal walls have been enlarged. The net heat transfer result gives a significant decrease of total heat transfer for increasing \( \xi_{hc} \). In this case the slit structure acts mainly as a radiation shield.

For the specular SB-case the heat transfer distributions for the hot and cold wall have been given in fig. 3.41b and c. At the hot wall (\( \xi_h = 0.1 \)) only a small amount of the total heat transfer is due to radiation, but large conductive heat fluxes occur (in comparison with IMA). At the cold wall the total heat transfer is mainly due to radiation and the conductive heat fluxes are not significantly increased. At small \( \xi_{hc} \leq 0.1 \) even a reduction of conductive heat transfer at the cold wall has been obtained together with an increase of radiative heat transfer. This means that heat, which leaves the hot wall by conduction, has been converted by the side walls for a large part into radiative heat transfer to the cold wall. Near the hot wall the emitted radiative energy of the side walls is large compared with the absorbed radiation, due to the low emissivity of the spectral selective hot wall. To balance the heat fluxes at the side walls increased wall and air conduction takes place and large temperature gradients occur in the side walls and the air in the region near the hot wall. Up to the cold wall radiative exchange becomes more and more in balance and less conductive heat transfer is needed to balance the heat fluxes at the side walls. The net result is that the
increase of total heat transfer at the hot wall is largely due to increased conductive heat transfer, whereas at the cold wall enlarged radiative heat transfer has been obtained.

For the specular SS-case (fig. 3.41d) a symmetrical situation has been obtained with equal radiative and conductive heat transfer at the hot and cold wall. The amount of radiative heat transfer is small due to the low emissivities of the hot and cold wall. With increasing $\tau_{hc}$ both air and wall conduction increase, but the amount of radiative heat transfer is not significantly affected by variation of $\tau_{hc}$. The net result gives enlarged total heat transfer in comparison with the independent heat analysis (given in the figure by $\tau_{hc} = 0$, $\rho_{S} = 1$).

For the diffuse reflectivity model (fig. 3.41e - h) the BB- and SS-case show again symmetrical situations at the hot and cold wall. For the BB-case only a slight reduction of radiative heat transfer with increasing $\tau_{hc}$ has been obtained. The wall and air conduction increase with increasing $\tau_{hc}$ resulting in an almost constant total heat transfer coefficient $h_{t}$.

The heat transfer for the diffuse SB-case at the hot wall (see fig. 3.41f) is again largely due to air conduction, the radiative heat transfer is small as a result of the low emissivity of the hot wall. At the cold wall the radiative heat transfer is the largest heat flux and the conduction is less important. The net result on $h_{t}$ is that, after an initial strong increase of $h_{t}$ for $\tau_{hc} < 0.15$, the total heat transfer becomes practically independent of $\tau_{hc}$.

For the diffuse SS-case (fig. 3.41h) the radiative heat transfer is practically independent of $\tau_{hc}$. The air and wall conductive heat fluxes increase with increasing $\tau_{hc}$, resulting in an increase of total heat transfer. In general the coupled heat transfer effects are less pronounced for the diffuse model in comparison with the specular model. This is due to the more uniform distribution of radiative heat transfer in the enclosure, thanks to the diffuse reflective side walls.

The coupled heat transfer effects for both reflectivity models explain the disappointing results which have been obtained (Buchberg (1977)) for honeycombs in combination with spectral selective solar collectors. The coupled heat transfer results at the hot wall in enlarged conduction heat fluxes. This heat is transported up to the cold wall by conduction and by reradiation of the slit structure walls.

From the results of the diffuse and specular model it is clear that an infra-
red diffuse reflective side wall material for slit structures is preferable to infrared specular or transparent material. Interaction of conduction and radiation results in increased total heat transfer for the SB- and SS-cases. Moreover the $h_t$ values for the diffuse model are smaller than those for the specular model for any $\epsilon_{hc}$ (except for $\epsilon_{hc} = 1$).

The side wall and air temperature distribution shall deviate from the uniform temperature distribution, due to the interaction of radiative and conductive heat transfer at the side walls.

Figure 3.42: The temperature distribution of the slit structure as obtained with the coupled heat transfer programme.

In fig. 3.42a the temperature distributions of the side walls have been depicted as obtained with the specular reflectivity model for the BB-, SB- and SS-case ($\epsilon_{hc} = 0.5$, $P_r = 0.5$). For comparison the uniform temperature distribution as used for the independent mode analysis has been given by the dashed line. In fig. 3.42b the temperature distributions obtained with the diffuse reflectivity model have been given ($\epsilon_{hc} = 0.5$, $P_d = 0.5$).

Firstly it is noted that the temperature distributions for the BB- and SS-cases are symmetrical with respect to the midplane ($y = 8.5$). Near the hot wall ($y = 0$) and cold wall ($y = 1$) large temperature gradients have been obtained. This explains the enlarged conductive heat fluxes at both walls through the side walls and the air. Larger gradients have been found for the SS-cases than for the BB-cases, due to the fact that the amount of emitted radiative energy is smaller for the SS-cases and therefore less radiation shall be absorbed by the side walls in comparison with the BB-cases. To balance the heat fluxes at the side walls more conductive heat transfer is needed, resulting in larger gradients.

For the SB-cases non-symmetrical temperature profiles have been calculated. The largest temperature gradient is located near the hot wall, which induces large conductive heat fluxes.

The temperature distributions obtained with the specular reflectivity model are similar to the ones calculated for the diffuse model except that larger temperature gradients occur. For the diffuse model the radiative energy in the enclosure is more uniform. Distinction cannot be made between diffusely reflected or (diffusely) emitted energy. As a consequence radiative heat exchange at the side walls is more balanced for the diffuse than for the specular model. This results in less conductive heat transfer to the side walls.

Real materials possess normally both a diffuse and a specular reflectivity component. Moreover a transmissivity component might be expected, which enlarges the effective specular reflectivity considerably. Given the large differences in $h_t$ values as obtained with the specular and diffuse model as shown in fig. 3.40 mixed reflectivity calculations are necessary.

Figure 3.43: Results of mixed reflectivity calculations.

(a) BB-case, (b) SB-case.

In fig. 3.43 results have been given of mixed reflectivity calculations for the BB- and SB-case. The $h_t$ values have been given as function of $\epsilon_{hc}$. The
solid lines indicate the results for the specular and diffuse BB-case. The dots represent the results for the mixed reflectivity calculations. From the diffuse BB-case to the specular BB-case each dot represents an increment of 0.1 for $P'_s$ and a decrement of 0.1 for $P_d$ ($\xi_{hc}$ has been kept constant, for example $\xi_{hc} = 0.2$, $P_d = 0.5$ gives for $P'_s = 0.3$). For the SB-case the increment of $P'_s$ has been 0.2.

In fig. 3.43a the dashed line marked (1) indicates for $\xi_{hc} = 0.2$ the variation of the reflectivity from the diffuse to the specular reflective model. Large differences in heat transfer have been obtained. The dashed line marked (2) gives the $h_t$ values for constant $P'_s$ ($P'_s = 0.6$). This means that $P_d$ has been varied between 0, and 1 - $P'_s$ and $\xi_{hc}$ has been varied between 1 - $P'_s$ and 0. For constant $P'_s$ only minor differences in $h_t$ values occur.

In fig. 3.43b the dashed lines indicate lines of constant $P'_s$. These results indicate that it is extremely important to use a material having an effective specular reflectivity as small as possible, meaning a small specular reflectivity and a small transmissivity coefficient for infrared radiation. To predict the heat loss in a black solar collector, in which a slit structure has been applied, it is very important to know the effective specular reflection of the slit structure material used. Varying $\xi_{hc}$ and $P_d$ for a certain value of $P'_s$ has negligible no influence on the heat transfer coefficient $h_t$ and knowledge of the exact value of $\xi_{hc}$ and $P_d$ is therefore less important for the BB-case. For spectral selective collectors knowledge of $P'_s$ and $\xi_{hc}$ (especially at small $\xi_{hc}$) is necessary to predict the total heat transfer in the slit structure accurately.

The materials used for slit structures shall be chosen primarily for a large solar transmissivity. However a considerable transmissivity might be expected also for radiative heat transfer at the infrared spectrum considered ($2 \mu m \approx 25 \mu m$), especially if one bears in mind that the thickness of the side wall material is of the order $\approx 10^{-5}$ m.

Since the slit structure also reduces radiative heat transfer (except for the case $P'_s = 1$) the variation of $h_t$ with the emissivity of the hot plate has been investigated for a SB-collector in combination with a FEP teflon slit structure.

The result has been given in fig. 3.44.

Again the heat transfer has been given for the hot and cold wall separately. For increasing $\xi_{hc}$ the total heat transfer increases monotonically. However the increase of $h_t$ is much smaller than calculated according to the parallel plate model. At the hot wall the amount of radiative heat transfer increases with $\xi_{hc}$, but the wall and air conductive heat transfer decreases. At the cold wall the three heat transfer modes increase gradually. For the theoretical value $\xi_{hc} = 0$, the heat transfer at the hot wall is due totally to wall and air conduction. At the cold wall radiative heat transfer is the largest of the three heat transfer modes.

Considering the results of the $\xi_{hc}$ variation it is clear that for materials, which possess a large emissivity for thermal radiation, the spectral selectivity of the absorber of the solar collector becomes less important. But a decrease of $\xi_{hc}$ shall always result in a decrease of $h_t$.

### 3.4.3.1 The influence of a gap.

The large effect that the mechanism of the coupled heat transfer has on the total heat transfer can be reduced by the introduction of a gap between the slit structure and the absorber (more in general at the place where the largest temperature gradients are expected). For a spectral selective solar collector large temperature gradients exist in the air and the slit structure walls near the absorber. This induces large conductive heat transfer. Application of a gap shall decrease the coupling between the heat transfer mechanisms locally and reduce the total heat transfer. Moreover the gap introduces an air layer with a smaller thermal conductivity than the thermal conductivity of the slit structure walls. It is obvious that a gap might reduce the
convection suppressing function of the slit structure and that convective heat transfer might increase. Therefore, we expect that an optimum gap width exists where minimum heat transfer occurs. To calculate the total heat transfer for slit structures (height $d$) with a gap between the absorber and the slit structure, our numerical program on coupled heat transfer has been adjusted. At the place of the gap, the effective specular reflectivity coefficient ($P_s'$) has been set equal to 1 and the conductivity of air has been used. This resulted in a more extensive determination of the total view factor ($\phi^t$) since locally varying specular reflectivity has to be considered.

To show the effect of the introduction of a gap on the total heat transfer in a slit structure, calculations have been performed for the SB-case with a Teflon slit structure. The relative gap width $s$ (defined as $s = (D-d)/D \times 100\%$) has been chosen 0, 5 and 10% of the plate distance $D$.

\[
h = 5 \times 10^{-9} \text{ m}
\]

\[
S = 5\%, 10\%
\]

\[
NHC
\]

![Figure 3.45: The dependence of $h_t$ on $D$ for varying gap width $s$.](image)

Stratified air conditions.

In Fig. 3.45 the results of the calculations have been given for varying plate distance $D$. For comparison, the NHC case has been given also (no slit-structure; radiative heat transfer calculated according to the infinite parallel plates model). As shown in the figure, the application of a relatively small gap results in a relatively much larger reduction of the total heat transfer. For instance, for a gap width of 5% the reduction in heat transfer amounts 13% for $D = 0.03$ m up to 29% for $D = 0.08$ m. For a gap width of 10% these values are 27% up to 41%.

The effect of the gap on the temperature distribution has been given in Fig. 3.46 where the temperature distribution in the slit structure has been given for $d = 0.05$ m and $s = 0\%, 5\%$ and 10%. Due to the gap, the temperature gradients near the absorber have been reduced strongly and the temperature distribution becomes more alike the linear non-coupled distribution.

Finally, we determined the dependence of the total heat transfer on the side wall thickness $h_w$ for a Teflon slit structure ($\varepsilon_{HC} = 0.6, P_s' = 0.4$) of $A_x = 0.1$ and $D = 0.04$ m. The results for a BB-, SB-, and a SS-case have been given in Fig. 3.47. The thickness of the side walls has been varied from $h_w = 10^{-5}$ m to $2 \times 10^{-4}$ m. The emissivity and total reflectivity of the material have been kept constant. For the BB-case, the $h_t$ values become almost independent of $h_w$ for $h_w > 5 \times 10^{-5}$ m. For the SB-case, this occurs for a value of approximately $2 \times 10^{-5}$ m, while for the SS-case the $h_t$ value still decreases for decreasing $h_w$ at the range considered. The three cases show qualitatively the same dependence on $h_w$, and the conclusion is that total heat transfer shall be lower for smaller side wall thickness. However, the overall effect on the total heat transfer is small. Moreover, the wall emissivity shall increase significantly with this large variation of $D$ given and this has not been taken into account.

3.4.4 Coupled convection-radiation.

To find the influence of the coupled heat transfer mechanisms on the onset of natural convection, we calculated numerically the total heat transfer.
as a result of coupled natural convective, wall conductive and radiative heat transfer simultaneously. We considered a Teflon slit structure of aspect ratio \( A_s = 0.25 \) and wall thickness \( 25 \times 10^{-5} \) m. The aspect ratio \( A_s = 0.25 \) has been chosen because natural convection development occurs at the Rayleigh range of interest, as has been found for the two dimensional non-coupled convection calculations. The problem has been considered only for two dimensions and an angle of inclination of 45 degrees. Although it is clear from our three dimensional convection calculations that the flow in the slit is three dimensional only a two dimensional convection model has been used to determine the effect of the coupled heat transfer on the onset of natural convection. With the results predictions can be given for the influence of the coupled heat transfer in the three dimensional situation. We varied the Rayleigh number by varying the plate distance \( D \) (from 0.015 - 0.055 m) for a constant temperature difference of 78 K \( (T_0 = 45^\circ C) \) for the BB-, SB- and SS-case. The results of the total heat transfer, as a function of the plate distance \( D \), have been given in fig.3.48.

Figure 3.47: Variation of the slit honeycomb wall thickness.

As shown in the figure, minimum heat transfer has been found for each case for \( D = 0.035 \) m.

If we consider the amount of natural convective heat transfer in the slit as

\[
\frac{\text{Nu}}{\text{Nu}_{\text{H}}/1} = \frac{\text{Nu}_{\text{conv,air}}}{\text{Nu}_{\text{cdr,air}}}
\]

The obtained \( \text{Nu}^\prime \) numbers for the BB- and the SS-case have been given in fig.3.49a as function of the Rayleigh number. The Nusselt values for the non-coupled two dimensional \( \text{pc} \)-case have been given for comparison also. For the non-coupled \( \text{pc} \)-case the modified Nusselt number at the hot wall \( \text{Nu}_{\text{H}}/1 \) is equal to the previously defined Nusselt number \( \text{Nu}_{\text{H}} \). Evaluation of this figure results in the conclusion that \( \text{Nu}^\prime \) is equal to 1 for \( \text{Ra} < 10^{15} \) and that for smaller Rayleigh numbers no convective heat transfer occurs. In this case we speak of a stagnant air situation. For larger Rayleigh numbers \( \text{Ra} > 10^{15} \) \( \text{Nu}^\prime \) is larger than 1 and convection participates to the total heat transfer process. Although the onset of convective heat transfer occurs at approximately the same Rayleigh number as for the non-cou-
The development of convective heat transfer is less pronounced for the BB-case and even more so for the SS-case. Equal results have been found for the hot and cold wall for these cases, due to symmetry. For the SB-case distinction must be made between the heat transfer at the hot wall \((y = 0)\) and at the cold wall \((y = 1)\), due to non-symmetrical boundary conditions. The results of the calculations have been given in fig. 3.49b. The \(Nu'\) values, given in fig. 3.49b, show again that the critical Rayleigh number is not significantly influenced. However, the development of natural convective heat transfer is much stronger at the cold wall than at the hot wall.

The main conclusion from these coupled convection-radiation calculations is that the critical Rayleigh number is not significantly influenced by the interaction of convection, wall conduction and radiation. That we might expect that for the three dimensional situation the critical Rayleigh numbers shall be approximately equal to those as derived with the non-coupled three dimensional natural convection program. However, the development of natural convective heat transfer is quite different. In the cases considered, the heat transfer development is more gradual, except for the SB-case at the cold wall.

Figure 3.49: \(Nu' - \text{Ra}\) results for coupled convection-radiation calculations.

4.1 Introduction.

Our experimental investigations considered both the flow structure and the natural convective and total heat transfer in slit structures. Flow visualization has been applied to investigate qualitatively the natural convection flow of air in slit structures of \(A_x < 0.5\) and angles of inclination from \(0\) (horizontal) to 90 degrees (vertical).

Heat transfer measurements have been performed with a heat transfer apparatus using a caloric method. Measurements on natural convective heat transfer concerned slit structures of \(0.125 < A_x < 0.5\) for angles of inclination between \(45 < \phi < 90\), while the total heat transfer measurements have been performed for \(A_x = 0.10\) and \(\phi = 45\) degrees. The aim of the total heat transfer measurements was to find a configuration with minimum total heat transfer. To find this optimum the plate distance \(D\) and the gap width between the slit structure and the hot plate have been varied.

Experiments have been done with a small solar simulator to determine the solar transmissivity of the slit structure. For these measurements non-CSD situations have been compared with situations with a slit structure applied. Aspect ratios of 0.25 and 0.1 have been chosen, \(D\) has been taken 0.85 m and 0.89 m and the angle of insolation has been 98 and 78 degrees, taken relative to the absorber surface.

Finally, collector test procedures, as recommended by the Commission of European Communities (CEC) (1988) shall be described. Total heat loss measurements on flat plate solar collectors equipped with and without slit structures have been performed.
4.2 Review of literature; experimental investigations.

In recent years several investigations concerning suppression of natural convection in flat plate solar collectors have been performed. Some of these investigations examined the influence of different CSD's on the onset of convection and the natural convective heat transfer (Hollands (1978); Edwards et al. (1978)). Others, for instance Buchberg et al. (1977) and Marshall et al. (1976) applied CSD's in flat plate collectors and determined the overall effect of the CSD on the efficiency of the solar collector. A third topic of investigation has been the solar transmittance of CSD's.

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Table 4.1: Summary experimental investigations concerning CSD's.

In table 4.1 a list of investigations concerning CSD's has been given. The experimental investigation performed by Hollands et al. (1979) concerned both honeycombs and slit structures. Details of these studies have been described by Hollands et al. (1976),(1979). Their honeycombs were made of Mylar, while the slit structures were made of polyethylene film. As already mentioned in chapter 3.2.2 the influence of the infrared reradiation on the convective heat transfer has been investigated by using different emissivities for the hot and cold wall (BB, SB, SS). Also the emissivity of the slit structure has been varied. As CSD material have been used transparent polyethylene \( (h_w = 40 \times 10^{-6} \text{ m with } \epsilon_{hc} \approx 0.13) \) and black opaque polyethylene \( (h_w = 150 \times 10^{-6} \text{ m with } \epsilon_{hc} \approx 0.9) \).

Hollands compared the results for the slit structures with those obtained for the square-celled honeycombs (Hollands (1976), Smart et al. (1980)). Both CSD's delayed the onset of natural convection in comparison with a non-honeycomb situation. The Nusselt-Rayleigh dependence, derived with a caloric method, can be seen in fig. 4.1. A comparison of slit structures and honeycombs, for the same amount of material used, showed that the honeycombs give better suppression of natural convection, for angles of inclination \( \phi \leq 75 \), while for \( \phi > 75 \) the slit structures are superior.

We can explain this effect by referring to our three dimensional natural convection calculations described in chapter 3.2.5.2. The effect is a result of a change in flow structure. Below a critical angle of inclination the flow is a combination of a transversal base flow (in the xy-plane) and a series of longitudinal rolls (in the yz-plane). For \( \phi < \phi_c \) it is quite clear that a slit structure with \( A_x = 0.1, A_z = \# \) gives better suppression of the base flow than a cell structure with \( A_x = A_z = 0.2 \). While for \( \phi > \phi_c \) the cell
structure with $A_z = 0.2$ gives better suppression of the longitudinal rolls than the slits with $A_z \geq 1$.

According to Hollands et al. (1978) the aspect ratios $A_x$ and $A_z$ and the emissivities of the hot and cold wall and the CSD material influence the onset of convection. They measured for different cases, i.e. BB ($\psi = 9.0$, $\sigma = 0.865$, $\epsilon_w = 0.9$) and SS ($\psi = 9.0$, $\sigma = 0.865$, $\epsilon_w = 0.865$) and different $\epsilon_{hc}$ ($\epsilon_{hc} = 0.13$ or 0.9) the critical Rayleigh numbers for slit structures of aspect ratio $A_x = 0.1$, 0.33 and 0.5.

They gave, based on analytical methods developed and tabulated by Sun (1970) and Buchberg (1976b), relations for $Ra_c$ and the averaged Nusselt number, $Nu_g$. For $\theta \leq 75$ these equations read:

$$Ra_c = \frac{1}{A_x^2 \cos \theta} \left( \frac{1536}{\sigma A_x} + 48 \frac{\epsilon_{hc}}{A_x} \right) \frac{N}{1 - S} \left( 1 - \epsilon_{hc} \right)$$

with $C = \frac{2 k_w \theta}{k_w h_w}$

$$N = \frac{40 T^3 D}{k_B}$$

$$S = 1.0182 \cdot 1.4389 A_x - 9.4653 A_x^2 + 31.448 A_x^3 - 27.515 A_x^4$$

The obtained $Nu_g - Ra$ dependence for $0 \leq \theta \leq 75$ is:

$$Nu_g = \frac{1 + c \left( Ra \cos \theta \right)^{1/3} \left( 1 - \exp \left( -a \theta \right) \right) \left( \frac{Ra}{Ra_c} \right)^{n} \left( 1 - \frac{Ra}{Ra_c} \right) \left( 1 - S \right) \left( 1 - \epsilon_{hc} \right)}{Ra \cos \theta}$$

with $a$, $b$, $c$ and $n$ assumed constants, chosen so that the sums of the square departure of the measured $Nu_g$ from the values given by eq. (4.2) over all the data points have been minimized. The values obtained for the constants are $a = 0.18$, $b = 1.2 \times 10^{-3}$ (degree$^{-1}$), $c = 0.131$, $n = 0.513$. The averaged rms error in $Nu_g$ amounts 18%. However these relations can not be fully predictive, since they do not incorporate the effect of the hot and cold wall emissivity, as remarked by Hollands (1978). Smaller values have been found for the BB-case and the SB-case. From these measurements Hollands concluded that decreasing hot and cold wall emissivity results in increasing $Ra_c$ for all angles of inclination. For example the critical Rayleigh numbers for $\psi = 0$ degrees and $A_z = 0.2$ are $4.7 \times 10^4$, $5.2 \times 10^4$ and $6.5 \times 10^4$ for the BB-, SB- and SS-case, respectively. A more detailed investigation considering the coupled radiative-conductive coupled heat transfer, has been described by Hollands (1979) as discussed earlier in chapter 3.2.2.

The experimental work performed by Arnold et al. (1977) and Edwards et al. (1978) concerned rectangular honeycombs cells. More specific they studied the effect of tilt on the onset of natural convection. Critical Rayleigh numbers have been determined for $0 \leq \theta \leq 98$ degrees for silicon oil as the working fluid. Moreover the effect of the aspect ratio $A_z$ on the onset of convection has been studied. Results showed no remarkable influence of $A_z$ on $Ra_c$ in the vertical situation. For $\theta \geq 98$ larger suppression of convection has been obtained for decreasing $A_z$ (see Arnold (1977)). However this influence of $A_z$ on the onset of natural convection in inclined situations has been obtained only for $A_z < 1$ (Edwards (1978)). Their results show strictly the effect of the suppressing influence of rectangular honeycombs on natural convection. The results of the investigations of Edwards and Hollands show qualitative good agreement.

Several investigators considered the effect of CSD’s on the overall performance of solar collectors.

Marshall et al. (1976) performed an experimental and analytical investigation concerning the use of hexagonal shaped honeycombs in solar collectors. Honeycombs have been constructed using the so-called expansion method. Different materials have been used for the fabrication of the honeycombs. Aspect ratios varied between 0.1 and 1. The aspect ratio has been varied by varying both the equivalent honeycomb cell diameter as well as the honeycomb cell height. The selected honeycomb materials have been Lexan ($h_w = 76 \times 10^{-6}$ m), Mylar ($h_w = 76 \times 10^{-6}$ m), Kapton ($h_w = 25 \times 10^{-6}$ m), Tedlar ($h_w = 192 \times 10^{-6}$ m) and FEP Teflon ($h_w = 76 \times 10^{-6}$ m). With a solar transmittance apparatus the solar transmittance has been measured for various angles of incidence.

The results for the different honeycombs have been given in fig. 4.2.

As shown in the figure the honeycombs made from Teflon, Lexan and Tedlar had transmissivities all greater than 0.99 for all cell diameters, aspect ratios
and incident angles of solar radiation ($\theta_0$). The Mylar and Kapton honeycombs have a considerably smaller solar transmission, what makes these materials less suitable for CSD applications.

Collector efficiency tests have been performed with selective black (SB) and black (BB) absorbers. The honeycombs were simply inserted between the glass cover plate and the absorber. Because the Lexan honeycomb was found to be the least difficult to fabricate with the expansion method, parameter variation has been done most thoroughly with Lexan honeycombs. In fig. 4.3a the effect of decreasing aspect ratio $A_x$ on the collector efficiency ($\eta$) has been given and compared with a black non-honeycomb collector (BB).

The collector efficiency increases as the honeycomb aspect ratio decreases, due to a reduction in convection and reradiation heat losses.

A comparison between honeycomb and non-honeycomb (NHC) collectors has been given in fig. 4.3b. Although considerable increases of efficiency have been obtained by applying honeycombs to solar collectors the results are lower than the analytical predicted performance. Largest deviations between measurements and predictions occurred for SB-collectors. These differences have been attributed to problems with selective coatings and failures of the analytical model.

Marshall concluded that Lexan was the best performer for honeycomb applications fabricated by the expansion method. However, next to Lexan FEP Teflon has been recommended for honeycomb collector applications, considering its fine optical properties, ultraviolet stability and high temperature stability. Due to difficulties with the fabrication of Teflon honeycombs with the expansion method no efficiency tests have been performed for Teflon honeycombs.

Buchberg, Edwards and Mackenzie (1977) used sheets of thin glass for the fabrication of CSD's. Several glass CSD's have been made and applied in flat plate solar collectors.

With a specially designed "hot roller" the glass sheets have been corrugated to sinusoidal formed sheets. The corrugated sheets were stacked and glued together to keep the corrugated pieces in position, thus forming a sinusoidal honeycomb (see McMurrin (1977). Heat transfer measurements showed that for temperature differences larger than 38 K natural convection was no longer suppressed by the sinusoidal honeycomb.

In addition CSD's have been fabricated from thin glass tubings. The glass tubings have been packed in a hexagonal pattern.

Efficiency measurements have been performed for solar collectors equipped
Figure 4.4: Example of a sinusoidal honeycomb.

with the cylindrical CSD. Comparisons with other conventional flat plate collectors showed that, due to suppression of natural convection by the CSD, collector efficiencies might be expected to increase from 10% for a selective black collector up to 15 - 30% for a non-selective black collector.

Edwards, Arnold and Catton (1976) investigated a practical problem, which arises when CSD’s must be designed for implementation in solar collectors. The question was whether or not it is important to minimize the gaps between the CSD and the absorber and between the CSD and the cover plate. They measured for a collector model, using a caloric method the Nusselt vs Rayleigh dependence for different top and bottom gap widths and for inclination angles between 0 and 30 degrees. No distinction has been made between different emissivities of the absorber and/or the cover plate. Results, obtained with a rectangular honeycomb ($A_\chi = 0.25, A_\Sigma = 2$), showed that small top and bottom gaps up to $1.5 \times 10^{-3}$ m did have no significant influence on the critical Rayleigh number. Moreover a bottom gap of $1.5 \times 10^{-3}$ m in combination with a relatively large top gap of $4.6 \times 10^{-3}$ m can be tolerated. Larger gaps resulted in intercellular convection and increased heat transfer.

During the same period of our investigation a project on advanced flat plate solar collectors has been performed at the CSIRO Division of Energy Technology, Victoria Australia. Based on the results of Hollands (1979) they directed their experimental programme on square honeycombs, slit structures and V-slit structures. An overview of the work as presented by Symons (1983), will be discussed now.

Honeycombs, slit structures and V-slit structures have been fabricated from FEP Teflon ($h_u = 13 \times 10^{-6}$ m). Solar transmittance measurements (Symons 1982) showed that all CSD’s made off Teflon had superior solar transmittance above glass tubular CSD’s ($h_u = 5 \times 10^{-4}$ m), $\tau_{hc} = 0.93$ and 0.87 respectively.

Natural convection heat transfer measurements performed for the Teflon CSD’s showed that the honeycomb with $A_\chi = A_\Sigma = 1/6$ suppressed the natural convection better than the slit structure with $A_\chi = 1/12$ for angles of inclination smaller than 75 degrees. This is in agreement with the results of Hollands et al. (1978). Between the slit and the U-slit no large differences have been measured. Both slit structures have been found only slightly inferior to the honeycomb.

Moreover it has been found that upslope oriented slit structures (upward directed; N-S direction) suppress natural convection better than cross slope oriented slit structures (horizontal, E-W direction, see fig. 4.5) for angles of inclination between 25 and 75 degrees.

Comparison of the yearly absorptance-transmittance product, i.e. the fraction of the solar radiation absorbed by a stationary collector over a year operating time, between collectors with and without CSD, showed that the reduction of absorbed energy by the absorber, due to the CSD, was approximately 3% for the square honeycomb and 6 - 8% for the cross slope slit structure and 7 - 10% for the upslope slit structure.

A prototype high temperature flat plate solar collector with CSD has been manufactured at CSIRO and has been tested. This prototype consisted of a single cover of low-iron glass with an infrared reflective coating, an absorber panel with chrome-black selective surface and 8.1 m of back insulation. The absorber to cover spacing has been chosen equal to 0.87 m, comprising a 0.86 m high CSD and a 0.81 m gap between the CSD and the absorber. The instantaneous collector efficiency has been measured for the honeycomb and upslope slit...
structure to temperatures up to 158° C, which resulted in $q = 0.97$ and $0.98$ and $F_R U_L = 2.77$ and 2.92 W/m²K for the honeycomb and upslope slit structure, respectively ($F_R$ is the heat removal factor of the collector).

A cost study has been performed of the prototype by the Australian Mineral Development Laboratories. The study revealed that the slit CSD prototype was more cost effective than the conventional flat plate collectors at temperatures above 88° C. Above 120° C a linear concentrating collector showed to be more cost effective.

4.3 Flow visualization.

To investigate the structure of the natural convection flow and to examine the assumptions, which have been made for the numerical modelling of the problem in chapter 3.2.1, flow visualization experiments have been performed in slit structures of aspect ratio $A_x : A_z = 8.125 : 5$. Special attention has been given to the transition of the transversal base flow to the flow pattern with longitudinal rolls and the interaction between slits of the slit structure, which might occur when a gap between the slit structure and the hot plate exists.

4.3.1 The experimental set-up and method.

The enclosure used for the flow visualization experiments is shown in fig. 4.6.

Two parallel isothermal plates ($0.280 \times 0.386$ m²) have been adjusted by four transparent perspex walls (D = 0.84 m) thus forming the enclosure ($A_z = 5$). The isothermal plates, one made of copper, one made of aluminum in our investigation respectively, the hot and cold wall, have been insulated from the environment by 0.81 m thick perspex. The hot wall has been kept at a constant temperature by means of a thermostat bath filled with water. The cold wall has been kept at a uniform lower temperature by cooling with silicone oil. In each plate four calibrated copper-constantan thermocouples have been placed ($3 \times 10^{-3}$ m from the surface) to measure its temperature. Slit structures have been made of Hostaphan BN 58 (polyethylene) foil of thickness $50 \times 10^{-6}$ m, which has been wound along pillars in the for this purpose build up and down side walls. By varying the distance between the pillars the aspect ratio has been varied. The smallest aspect ratio which we could create has been $A_x = 0.125$. To visualize the flow pattern a narrow parallel beam of light has been used to illuminate a cross section of the enclosure. Since the four side walls have been made of transparent perspex cross sections of the $xy$-plane or the $yz$-plane could be illuminated. A parallel beam of light has been obtained by focusing the light of a 500 W Hg source on a rotate slit diaphragm. An additional positive lens has been used to convert the slit-shaped light beam into a parallel beam, see fig. 4.7.

Figure 4.6: The experimental model for flow visualization.

Figure 4.7: Optical configuration of the flow visualization set-up.
By means of a notable flat mirror the light beam could illuminate the enclosure at all angles of inclination considered.

The enclosure was mounted on a adjusting table, which made it possible to vary the angle of inclination from 0 to 90 degrees with increments of 10 degrees. The experimental set-up has been placed in a black painted cupboard to prevent inconvenient scattering and to obtain an optimal contrast. For all our experiments the distance between the isothermal plates has been kept at 0.04 m, which meant that the Rayleigh number has been varied by varying the temperature difference between the hot and cold plate ($T_h - T_c$). The Rayleigh number varied between $10^4 \leq Ra \leq 2.5 \times 10^5$. At low Rayleigh number ($\leq 10^4$) the accuracy of the Rayleigh number was 12 %, due to the small temperature difference between the hot and cold plate. For $Ra = 2 \times 10^5$ the accuracy of $Ra$ was 5 %. The accuracy of $A_x$ was 3 %, 2 %, 1.5 % for $A_x = 0.125, 0.25$ and 0.5, respectively. The adjusting accuracy of the inclination angle $\phi$ has been 2 degrees.

Experiments have been performed by injecting cigar smoke with a syringe in the slit viewed. Shortly after injection ($\leq 10$ s) the flow pattern could be observed. Observations have been made visually as by photography. Photographs have been made with an Asahi Pentax ES2 automatic camera with an 100 mm, F4 macro lens. The exposure time varied between 1/8 to 8 seconds with a film speed of 400 ASA.

4.3.2 Results of flow visualization.

For the validity of the assumptions made for the numerical modelling of the natural convection flow we examined first if non-stationary flow exists at the parameter range considered.

For $A_x = 0.125, 0.25$ and 0.5 only stationary flow has been observed for all angles of inclination for $Ra$ up to $2.5 \times 10^5$. This is in good agreement with an extrapolation of the results of Schinkel (1980), who determined the transition from stationary to non-stationary flow for $A_x = 1, 2, 4$ and 7.

His results for $\phi = 0$ degrees have been given in fig. 4.8.

Extrapolation of his results for smaller $A_x$ shows that for $A_x = 0.5$ non-stationary flow may be expected in the horizontal orientation for $Ra \geq 2 \times 10^5$. For $A_x = 0.25$ ($A_z = 5$) this Rayleigh number will be approximately $5 \times 10^5$. For other angles of inclination ($\phi > 0$ degrees) the flow behaviour is more stable and transition to non-stationary flow will occur at even higher Rayleigh numbers. Although the highest Rayleigh number during the flow visualization experiments has been $2.5 \times 10^5$ only, it might be expected that for all Rayleigh numbers considered in the numerical part of this investigation ($10^4 \leq Ra \leq 10^6$), for $\phi > 45$ degrees and $A_x \leq 0.25$ stationary flow exists. Only for $A_x = 0.5$ and $\phi \approx 90$ degrees non-stationary flow behaviour is possible at high Rayleigh number ($Ra \geq 5 \times 10^5$). For these situations our numerical model might be no longer sufficient.

We have been interested also in the flow behaviour at inclined situations, based upon the large differences in heat transfer we obtained between the two and three dimensional calculations (chapter 3.2.5.3). Special attention has been given to the determination of the transition of transversal base flow (mainly in the xy-plane) to flow with longitudinal rolls in the yz-plane. We expect for the vertical situation and for all $A_x$ two dimensional transversal base flow in the xy-plane.

In fig. 4.9 photographs of observed flow patterns in the xy-plane for the vertical situation has been given for the aspect ratios considered. For $A_x = 0.5, Ra = 1.3 \times 10^5$ (fig. 4.9a) strong natural convection has been observed. In the center of the enclosure the tendency to form two secondary rolls can be seen. For $A_x = 0.25, Ra = 1.9 \times 10^5$ suppression of natural
convection has been obtained (fig. 4.9b) and for $A_x = 0.125$, $Ra = 2.5 \times 10^5$ (fig. 4.9c) the enclosure has been divided into different regions: two end regions near the hot and cold wall where the flow circulates and a very tranquil center region where air flow can hardly be observed.

By decreasing the angle of inclination, with increments of 10 degrees the transitions from base flow to longitudinal rolls have been determined as a function of the Rayleigh number. For $A_x = 0.5$ and 0.25 the inclination angle has been varied from 98 to 0 degrees while for $A_x = 0.125$ it varied between 98 and 45 degrees. The obtained transition curves have been given in fig. 4.10.

For $A_x = 0.5$ and 0.25 the angles of transition from base flow to longitudinal rolls have been found to be almost independent of the Rayleigh number.

Above the transition curve base flow in the xy-plane has been observed, while below the transition curve the flow appears predominantly as longitudinal rolls in the yz-plane.

A photograph of longitudinal rolls in an yz-plane (at $x = 0.5 A_x$) is shown in fig. 4.11a for $\theta = 0$, $Ra = 10^5$ and $A_x = 0.25$. The diameter of the rolls is approximately the plate distance $D$ and the number of rolls in this situation is six. According to the analysis of Stork and Müller (1972) the number of rolls should be equal to the aspect ratio $A_x$. In our case both six and eight rolls (for $Ra = 6 \times 10^4$) have been observed (fig. 4.11b).

For our situation the number of rolls is influenced by the side wall temperature boundary condition. Because $T_B$, the average temperature of the enclosure, has been higher than the ambient temperature $T$, heat losses from the enclosure to the surroundings occur. This forces the flow downwards at the side walls and results for $A_x = 5$ in an even number of rolls. The transition for $A_x = 0.25$ takes place at a higher angle of inclination than for $A_x = 0.5$. For $Ra > 10^5$ longitudinal rolls have been obtained for $\theta < 60$ degrees.

Regarding the tendency that the transition occurs at a higher inclination angle for decreasing $A_x$, we may expect a transition curve for $A_x = 0.125$ at $\theta > 60$ degrees. The triangles given in fig. 4.10 are data points obtained for $A_x = 0.125$. It should be noted that flow visualization readings have been almost impossible for $A_x = 0.125$ with our experimental set-up at this Rayleigh range. This was due to the large suppression of flow. Only for $Ra > 10^5$ reliable observations could be made. The triangles give therefore
more an indication of the start of flow development than of an accurate transition point. For increasing Ra number longitudinal rolls develop. At lower Ra numbers no specific flow structure has been observed. The tendency to earlier flow development at a lower inclination angle is in qualitative good agreement with our three dimensional numerical results. As an example, the critical Rayleigh numbers obtained in chapter 3.2.5.2 for $A_x = 0.125$, $A_z = 10$ have been $5 \times 10^5$ and $3.2 \times 10^5$ for $\theta = 60$ and 45 degrees respectively. According to our flow visualization results the first flow development has been observed for $Ra \approx 2 \times 10^5$ and $9.4 \times 10^4$ and appeared as longitudinal rolls in the yz-plane. An explanation for the differences between the experimental and numerical obtained critical Rayleigh numbers is that the numerically derived Ra $C$ is based upon the onset of convective heat transfer, while for the flow visualization the critical Rayleigh number is determined visually for the first flow development.

For $\theta > 75$ degrees and $Ra = 2.5 \times 10^5$ base flow develops. Therefore it might be expected that the transition to longitudinal rolls for $A_x = 0.125$ occurs for $60 < \theta < 70$. For $Ra \approx 10^5$ complete suppression of the flow has been obtained.

4.3.2.1 Flow structure after gap application.

Application of a gap between the hot wall and the slit structure has been accomplished by reducing the foil width to $35 \times 10^{-3}$ m, thus creating a gap of $5 \times 10^{-3}$ m. For an inclination angle of 98 degrees the flow for $A_x = 0.5$ showed no large differences with the no-gap case (see fig. 4.12a in comparison with fig. 4.9a).

The natural convection flow remains in one slit and does not interact with neighbouring slits. Also for $\theta = 45$ degrees (fig. 4.12b) interactions between slits have not been found and transversal base flow has been obtained only. For $A_x = 0.25$ the flow in the slits does not extend to the gap (see fig. 4.12c). The flow is "forced" to stay within the slit by an upwards directed laminar flow in the gap at the hot plate. This upwards directed flow is due to boundary effects at the left and right enclosure walls of our model. In this respect our model is not ideal. Starting at the bottom of the gap near the hot plate a significant laminar boundary flow develops at the hot wall. At the top the flow is directed to the side walls and from here downwards back to the bottom of the enclosure from where it started. This circulating flow has an important effect on the flow in the slit structure. Not only the
flow is kept between the slit structure walls, but also the circulating flow interacts with the slit structure flow and connects neighbouring slits with each other. Attempts to reduce this effect by preventing the circulating flow to stream downwards at the side walls have not been fully successful. Although large reduction of the circulating flow has been established, the remaining flow was still strong enough to maintain the same flow pattern. This effect might cause a significant heat loss in a solar collector as well, when a gap has been introduced between the CSD and the absorber plate. The gap flow might transfer heat to the sides of the collector. Insulating the sides of the collector and a close fit of the CSD at the sides will decrease this effect.

For $A = 0.125$ only a circulating roll has been observed near the gap between the slits. Near the cold wall stagnant air occurred.

For $\phi = 45$ degrees longitudinal rolls have been observed for $A_x = 0.25$ and $0.125$ between the slit structure walls and the upwards directed laminar boundary flow at the hot wall.

Main conclusions from the flow visualization experiments are:

1. Stationary flow has been observed only for all $A_x$ at the Rayleigh range considered: $10^4 \leq Ra \leq 2.5 \times 10^5$.
2. For $45^\circ \leq \phi \leq 90^\circ$ the main flow in a slit of $A_x = 0.5$ remains in the xy-plane of the enclosure.
3. For $A_x = 0.25$ and $0.125$ a transition of transversal base flow to longitudinal rolls occurs almost independent of the Rayleigh number at $\theta \approx 40$ and $\theta \approx 70$ degrees, respectively.
4. Introduction of a gap near the hot plate results in a similar flow pattern within the slits. Interaction between slits through the gaps at the hot wall is small. However enclosure wall effects increase, a circulating flow up the hot wall and down the enclosure walls has been observed.

This effect might be reduced in a solar collector by insulating the sides and by a close fit of the CSD at the sides.

4.4 Caloric measurements.

Heat transfer measurements have been performed with a specially purpose-built heat transfer apparatus according to a caloric method. A caloric method has been chosen because this measuring technique enables us to measure the total heat transfer, including radiative heat transfer. However local heat transfer measurements are not possible with our set-up and all our results obtained with the heat transfer apparatus have been average values. This means that for a NHC situation the obtained Nusselt numbers and total heat transfer coefficients $h_t$ have been averaged over the hot plate. For situations with a slit structure $Nu_h$ and $h_t$ have been averaged over a number of slits ($\approx 100$, for $D = 0.05 \text{ m}$, $A_x = 0.1$).

4.4.1 The experimental set-up.

The heat transfer apparatus consists of a philitex box where three aluminium plates ($0.549 \times 0.530 \text{ m}^2$) of thickness $5 \times 10^{-3} \text{ m}$ have been placed parallel to each other (see fig. 4.13). The single aluminum plate (notated (1) in fig. 4.13) represents the cold wall. The two other plates form the hot plate system. The front plate (2) of this system acts as the isothermal hot wall for the enclosure. The back plate (3) is the guard heater with a temperature equal to that of the front plate (2). The function of the guard heater is to minimize the heat losses to the back.

In the enclosure slit structures can be winded along pillars in the perspex side walls of the box (4). The plates can be moved parallel to each other by means of spacers (5) to vary the distance $D$ between the hot and cold plate.
Additional roofmate insulation (0.95 m thick, \(\lambda \approx 0.03 \text{ mK}^{-1}\)) at all the sides of the box decreases the heat loss to the surroundings. The box can be placed in tilted situations with fixed angles of 40, 45, 50, 60, 90 and 100 degrees.

The cold plate has been kept at an uniform temperature \(T_C\) by cooling with a waterloop from a thermostat bath. The temperature has been measured with nine calibrated copper-constantan thermocouples, placed just below the plate surface.

![Hot plate system diagram](image)

**Figure 4.14**: The hot plate system.

The hot plate system is schematically shown in fig. 4.14. The front plate consists of two aluminum plates of thickness \(5 \times 10^{-3}\) m. At the back of the first plate heating foils (eight Minco HK 6870-11 B119 and four Minco HK 6858-11 B169) have been glued on the surface. The heating foils have been placed parallel in an electrical circuit and have been regulated so that the front plate was isothermal within 1% of the temperature difference \(T_h - T_c\). The power level for the heating foils has been regulated by the temperature of the hot plate with respect to a set reference temperature \(T_{\text{ref}}\), where \(T_{\text{ref}}\) is the temperature of the hot plate at which we like to measure. If \(T_{\text{ref}} - T_h > 5\) K maximum heating occurs. For \(T_{\text{ref}} - T_h < 5\) K the power supply is proportional to the temperature difference \(T_{\text{ref}} - T_h\). The value of the effective voltage supplied has been measured by an effective voltmeter. The effective voltage has been used to determine the dissipated energy by the heating foils.

Just below the surface of the hot plate 19 Cu-Co thermocouples have been placed to measure the hot plate temperature. To prevent damaging of the heating foils and to establish an uniform temperature at the back of the first plate a second (also \(5 \times 10^{-3}\) m thick) aluminum plate covered the heating foils. The guard heater has been placed at \(5 \times 10^{-2}\) m behind the hot plate. Glasswool insulation filled the enclosure between the main and guard heater. The temperature difference between the main and guard heater has been measured with 9 double temperature difference thermocouples. A signal from one of the temperature difference thermocouples has been used to regulate the power of 13 power transistors (2N4292), which heat the guard heater. This way the temperature difference between the main and guard heater has been kept within 8.7 K for all measurements.

The hot plate system has been placed in a perspex box. Behind the cold plate and the hot plate system rockwool insulation filled the space between the phillipex box and the plates. The whole system has been controlled by a HP 85A microcomputer combined with a HP 3854A datalogger.

During the heating process all the signals of the thermocouples have been registered periodically. RMS errors of the temperatures of the hot and cold plate have been determined. Whether stationary conditions have been obtained, has been tested by two criteria:

1. \(T_h (i) - T_h (i-1) < 0.01\) K, where \(i\) denotes the time step (\(\Delta t = 3\) minutes).
2. Changes in the temperature difference between the hot plate and the guard heater: \(\frac{\Delta T}{\Delta t} < 2 \times 10^{-5}\) K/s, considered for 18 consecutive measurements at time intervals of three minutes.

It was found empirically that \(h_t\) was constant within measurement accuracy (4%) when both criteria had been fulfilled. After the hot plate and the cold plate had reached their equilibrium temperature within 0.1 K, an additional time of 1 up to 3 hours was necessary to reach stationary conditions according to criteria 1 and 2. When stationary conditions have been reached the effective voltage has been recorded and supplied to the microcomputer, which calculates the different heat transfer coefficients and desired parameters (\(N_d, R_a, h_t, T_h - T_c\)).

### 4.4.2 Determination of the heat transfer.

For stationary conditions the dissipated energy by the heating foils is equal to the total heat transfer in the enclosure plus the heat losses to the surroundings, in formula:

\[
E = h_L A (T_h - T_c) + U_{\text{ins}} (T_h - T_a) \tag{4.3}
\]

Here \(h_t\) represents the total heat transfer coefficient between the hot and...
cold plate (W/m²K), \( E \) the dissipated energy by the heating foils (W), \( U_{\text{ins}} \) the heat loss factor for the heat losses to the surroundings (W/K), \( T_a \) the ambient temperature and \( A_p \) the area of hot plate \( (A_p = 0.291 \text{ m}^2) \). Our measurements concern the determination of the dissipated energy \( E \) and the temperature differences \( T_h - T_c \) and \( T_h - T_a \).

From eq. (4.3) the total heat transfer coefficient \( h_t \) can be determined if \( U_{\text{ins}} \) is known. The determination of \( U_{\text{ins}} \) has been described in Appendix C and it's value has been found to be 0.092 + 0.28D W/K. Moreover we like to distinguish for the total heat transfer the amount due to convective heat transfer. For enclosures without a slit structure this distinction can be made straightforward. For this case the total heat transfer between the hot and cold plate can be calculated from a separate convective and radiative part:

\[
h_t(T_h - T_c) = N_u d \frac{\lambda}{D} (T_h - T_c) + \frac{\sigma (T_h^4 - T_c^4)}{(\xi_h^{-1} + \xi_c^{-1} - 1)}
\]

where \( N_u d \) represents the average Nusselt number for the convective heat transfer between the plates. The radiative heat transfer has been calculated as for radiative heat transfer between two isothermal infinite parallel plates. When \( \xi_h \) and \( \xi_c \) have been determined the average Nusselt number can be obtained from eq. (4.4). The determination of \( \xi_h \) and \( \xi_c \) has been described in Appendix C.

When a slit structure has been applied eq. (4.4) is no longer appropriate to determine the total heat transfer, due to the interaction of radiative, convective and conductive heat transfer. The application of a slit structure has an effect on all three heat transfer mechanisms in the slit. Due to radiative interchange with the slit structure walls the radiative heat transfer from the hot to the cold wall shall in general decrease. The slit structure acts as a radiation shield for thermal radiation. The air and wall conductive heat transfer shall increase as a result of the coupling between the heat transfer modes. We used the following method to determine the convective heat transfer. First the non-coupled total heat transfer coefficient has to be measured. When the heat transfer apparatus is tilted 100 degrees (hot plate above the cold plate) the air in the enclosure may be assumed stratified and the total heat transfer is given by coupled radiative, air conductive and side wall conductive heat transfer:

\[
E_{100} = h_{t,100} A_p (T_h - T_c) + U_{\text{ins}} h_c (T_h - T_c)
\]

\( E_{100} \) can be determined by measuring the dissipated energy in the heating foils and represents the total coupled radiative, air conductive and side wall conductive heat transfer, for stratified air conditions.

When the angle of inclination is \( \phi \), the dissipated energy of the heating foils in stationary conditions is given by:

\[
E_{\phi} = h_{t,\phi} A_p (T_h - T_c) + U_{\text{ins}} h_c (T_h - T_c)
\]

Here \( h_{t,\phi} \) represents the total heat transfer coefficient for inclination angle \( \phi \). In chapter 4.4.4 measurements concerning the total heat transfer coefficients \( h_{t,100} \) and \( h_{t,\phi} \) shall be described.

To determine the convective heat transfer we assume that the radiative and wall conductive heat transfer and the heat losses to the surroundings have not been changed by the change of inclination angle. The difference in total heat transfer now obtained, in comparison with the stratified air situation, must be due to natural convection. The Nusselt number should be defined based on the coupled air conductive heat transfer \( (Q'_{cd}) \), as we did in chapter 3.4.4. However it is not possible with our measurements to distinguish between air conductive and radiative heat transfer. So it is not possible to determine the coupled conductive heat transfer \( Q'_{cd} \). Therefore we define \( Q_{cd} \) as \( Q_{cd} = \lambda/\omega(T_h - T_c) A_p \), which is the theoretical conductive heat transfer through the air if no radiation would occur and the Nusselt number can now be defined by:

\[
N_u' = 1 + \frac{E_{\phi} - E_{100}}{Q_{cd}}
\]

The prime on the Nusselt number denotes that the Nusselt number has been obtained for coupled convective-radiative heat transfer. According to this definition \( N_u' \) gives not the correct value for the total air convective heat transfer, since due to the coupled heat transfer modes the temperature distribution in the air for \( \phi = 180 \) degrees is not equal to the non-coupled
linear temperature distribution. In this case the coupled air conductive heat transfer is not equal to the non-coupled conductive heat transfer either: \( Q'_{\text{cd}} \neq Q_{\text{cd}} \). However \( Nu'_B \) equal to 1 means that the coupled convective heat transfer is nil and the heat transfer by the air is conductive-radiative heat transfer. \( Nu'_B \approx 1 \) indicates that natural convection occurs. The obtained critical Rayleigh number is the correct value, but the \( Nu'_B \) number does not determine the coupled convective heat transfer since \( Q'_{\text{cd}} \) is unknown.

It is clear that the assumption that radiative, conductive and convective heat losses have not been influenced by the change of the inclination angle \( \phi \), can not be true. All heat transfer mechanisms are coupled and a change of the convective heat transfer, due to a change of the inclination angle, results automatically in a change of the radiative and conductive heat transfer. Only when a change in the convective heat transfer remains small the assumptions made (eq. 4.7) shall be approximated. The results for the convective heat transfer have been given in chapter 4.4.3.

For evaluation of the experiments we will also use the total heat transfer coefficient \( h_t \) which can be written as:

\[
h_t = \frac{U_{\text{eff}}^2 / R_h - U_{\text{ins}}^2 (T_h - T_C)}{U_{\text{ins}} (T_h - T_C) / R_h}
\]

(4.8)

We will consider the systematic errors and the experimental inaccuracy. Systematic errors are due to the inaccuracy of the value of \( R_p, U_{\text{ins}} \) and \( R_h \) (the effective resistance of the heating foils). The relative value of the systematic error amounts to 7% (mainly caused by the error in \( R_h \)). This produces for our measured \( h_t \) values an absolute error of \( \pm 0.2 \text{ W/m}^2\text{K} \). This error should be taken into account when measured and numerically calculated \( h_t \) values are compared.

Secondly we have to consider the error due to inaccurate measurements of \( T_h, T_C, T_a \) and \( U_{\text{eff}} \). The maximum relative error amounts to 4% for \( h_t \approx 2 \text{ W/m}^2\text{K} \) and \( \approx 2 \% \) for \( h_t \approx 4 \text{ W/m}^2\text{K} \) and has been caused predominantly by the inaccuracy of \( U_{\text{eff}} \) (\( \pm 0.1 \text{ V} \)).

For the \( Nu'_B \) values larger errors occur for \( Nu'_B \) near 1. Because the heating power \( (E_g) \) in the inclined situation has been measured only marginally larger than for the stratified air situation \( (E_{180}) \). For \( Nu'_B \approx 1 \) the relative error is of the order of 10%.

The error in the Rayleigh number, mainly due to the inaccuracy of the plate distance \( D \) and temperature difference \( (T_h - T_C) \), amounts \( \approx 3 \% \) for \( Ra = 5 \times 10^4 \) and \( \approx 2 \% \) for \( Ra = 10^6 \).

Another aspect which we have to consider is the quality of the slit structures. All slit structures have been made by hand and although the same procedure has been applied for all slit structures, deviations might occur (for instance the tightness at the isothermal walls might differ). This reproducibility has been tested for two slit structures of \( A_x = 0.1 \). One slit structure has been made in the usual way, the second (equally sized) slit structure has been badly winded on purpose. In this way it was possible to compare the effects of the slit structure quality on the suppression of natural convection. The results of the \( h_t \) measurements have been given in fig. 4.15.

![Figure 4.15: Comparison between a neatly and an extremely badly winded slit structure.](image)

The heat transfer coefficients for stratified air conditions \( (h_{180}) \) have been measured equal for both slit structures, within our measurement accuracy. Differences have been obtained in the critical temperature difference where the convection starts and the development of the convection. Bad reproducibility accounts especially for the measurement of the temperature difference at the onset of natural convection \( (\Delta T_{cr}) \). This value has been measured significantly worse than the values for the carefully winded slit structure.

This means that differences in slit structures influence primarily the onset of natural convection and secondly the development of natural convection.
For our measurements the differences have been smaller than for the extreme case shown in fig. 4.15. The duplication of measurements has been very good. Repetition of the measurements with the same set-up for the same slit structure resulted in values within the experimental relative error of 4%.

4.4.3 Results convective heat transfer measurements.

To measure the convective heat transfer for one situation a series of three measurements are done. First a measurement of the total heat transfer under stratified air conditions \( E_{100,a} \). Secondly the total heat transfer has been measured for the inclined situation \( E_{b} \) and the third measurement done gives again a value for the total heat transfer under stratified air conditions \( E_{100,b} \). The average \( E_{100} \) has been corrected for the small change in temperature difference in \( T_{h} - T_{c} \), which is due to experimental reproducibility. The average corrected heat transfer for the stratified air condition is then given by:

\[
E'_{100} = \frac{E_{100,a} + E_{100,b}}{2} \left[ T_{h} - T_{c} \right]_{100} \tag{4.9}
\]

Using this value for the dissipated energy under stratified air conditions in eq. (4.7) gives the contribution of convective heat transfer (denoted by \( Nu'_{g} \), and defined by eq. (4.7)) to the total heat transfer.

The dependence of \( Nu'_{g} \) on \( Ra \) has been measured for slit structures of \( A_{x} = 0.5, 0.25 \) and 0.125 and for inclination angles of 40, 50 and 60 degrees. To vary the Rayleigh number the plate distance \( D \) as well as the temperature difference \( T_{h} - T_{c} \) has been varied. In this way it was possible to perform measurements for \( 10^{3} < Ra < 10^{6} \). The \( Nu'_{g} \) measurements have been performed without any coating on the aluminum plates, which meant that \( \varepsilon_{h} = \varepsilon_{c} = 0.2 \) (see Appendix C for determination of the emissivities).

For each angle of inclination considered the non-CSD situation has been measured also, which has been denoted in the following figures by "NHC" (\( Nu'_{g} = Nu'_{g} \) in this case). The results of our measurements for all aspect ratios considered has been given in fig. 4.16. The average \( Nu'_{g} \) numbers have been correlated with \( Racos\theta \) as parameter.

![Figure 4.16: Convective heat transfer results, \( Nu'_{g} \) as function of \( Racos\theta \).](image)

The relations obtained for each aspect ratio and derived for \( \theta = 0, 40, 50 \) and 60 degrees have been given in table 4.2. The suppression of natural convective heat transfer by the slit structures can be seen clearly in fig. 4.16. For a certain Rayleigh number, \( Nu'_{g} \) has the lowest value for \( A_{x} = 0.125 \), a larger one for \( A_{x} = 0.25 \) and an even larger value for \( A_{x} = 0.5 \). For the Rayleigh range considered \( (10^{3} < Ra < 10^{6}) \) the NHC-case gives the largest convective heat transfer.

Moreover the critical Rayleigh numbers for \( \theta = 0 \) as found from our measurements (see table 4.2) show that onset of convection has been delayed longest for \( A_{x} = 0.125 \). However even for \( A_{x} = 0.125 \) natural convection develops at the Rayleigh range of interest. For complete suppression the aspect ratio \( A_{x} \) should be chosen smaller than 0.125. The strongest development of natu-
ral convection, for \( Ra > Ra_c \), has been obtained for the slit structures with the smallest aspect ratio.

![Graph](image)

**Figure 4.17:** \( Nu'_{\theta} - Ra \) dependence for various \( A_x \) and different inclination angles.

In fig. 4.17 the \( Nu'_{\theta} - Ra \) dependence has been given for \( A_x = 0.125 \) and \( 0.25 \) with \( \theta = 40 \), 50, and 60 degrees together with the \( Nu'_{\theta} - Ra \) dependence of the NHC-case. A comparison of the results for the different inclination angles show that the slit structure suppresses the natural convection better at larger inclination angle. This is in agreement with the three dimensional numerical results and the flow visualization experiments.

A comparison between our calorically derived NHC results with experiments of Hollands (1976) and numerical work of Schinkel (1980) has been given in fig. 4.18 for \( \theta = 45 \) degrees. Our data points have been given for the different plate distances used. Our results appear to be somewhat higher than those obtained by Hollands and Schinkel, but in general the agreement is good.

4.4.4 Results total heat transfer measurements.

Evaluation of the results of the convective heat transfer measurements show that effective suppression of natural convection in slit structures requires slit structures of at least \( A_x = 0.1 \). Moreover since the different heat transfer modes influence each other total heat transfer measurements are necessary. To simulate a spectral selective solar collector the cold plate of the model has been given an emissivity of \( \approx 0.95 \) by using black paint. The hot plate has still an emissivity of \( \approx 0.2 \). In Appendix C the determination of the plate emissivities has been described.

Although most of our total heat transfer measurements have been performed for slit structures of \( A_x = 0.1 \) a few measurements have been done for \( A_x = 1 \). The fact that coupled heat transfer occurs is clearly demonstrated by the results of the measurements for the stratified air conditions \( (\theta = 180 \) degrees). In fig. 4.19 the \( h_{t,180} \) values have been given for \( A_x = 0.1, 1 \), and for the NHC case. For the stratified air condition the NHC measurements give the lowest heat transfer for all plate distances \( D \) considered. Larger heat transfer occurs for \( A_x = 1 \) and the largest \( h_{t,180} \) values have been measured for \( A_x = 0.1 \).

According to the analysis in which the radiative heat transfer is assumed not to be affected by the application of the slit structure and equal to the value according to the infinite parallel plates model, the \( h_{t,180} \) value should increase only with the conductive heat transfer through the slit structure material. Actually the slit structure would reduce the radiative heat transfer due to the shielding effect the slit structure has on the thermal radiation. For the NHC case and \( D = 0.04 \) m a value of 2.49 W/m²K has been found. The theoretical increase for the wall conduction amounts for \( A_x = 0.1 \) and 1 \( \approx 0.02 \) W/m²K and 0.002 W/m²K, respectively. So instead of approximately equal or reduced heat transfer we found an increase of the total heat transfer coefficient. The measured values are much larger than the theoretical values, \( 4 h_{t,180} = 0.53 \) W/m²K and 0.24 W/m²K, respectively.
Figure 4.19: Demonstration coupled heat transfer effects.

The large differences cannot be explained with this analysis and coupled heat transfer must occur.

The measured heat transfer coefficients for the stratified air conditions have been compared with numerically derived $h_{\text{t,180}}$ values. We calculated with the coupled conduction-radiation programme (described in chapter 3.4) the experimental stratified air cases. Several parameters of the heat transfer apparatus have been measured. This resulted in the following values: $\epsilon_h = 0.2$, $\epsilon_c = 0.95$, $\epsilon_{hc} = 0.6$, $\rho'_t, h_c = 0.4$, $\kappa_w = 50 \times 10^{-6}$ m, $\kappa_m = 0.13$ W/mK and $\alpha_c = 0.1$.

The gap width $s$, the plate distance $D$ and the foil width $d$ have been varied. The comparison has been given in fig. 4.20. Good agreement has been obtained between experimental and numerical results within the experimental accuracy (systematic and measurement accuracy $\pm 10\%$). This result validates the numerical model with all its assumptions as used by us. It shows again the coupling of the heat transfer modes.

The main purpose of our tests has been to find an optimum plate distance $D$ in combination with an optimum gap width $s$, where minimum heat transfer occurs. We varied therefore the plate distance $D$ between $35.5 \times 10^{-3}$ m and $91 \times 10^{-3}$ m. By using different foil widths and variation of $D$ the relative gap width $s$ could be varied.

In fig. 4.21 results have been given for $\Phi = 45$ degrees and $A_x = 0.1$. The plate distance $D$ has been varied. The slit structure fitted as close as possible to the isothermal walls, still leaving a gap of $\approx 4\%$.

Figure 4.21: Total heat transfer results for different plate distances $D$. Gap width $s \approx 4\%$. 
The $h_t$ value has been given as function of the temperature difference between the hot and cold wall and different curves have been given for the different plate distances. We found for each $D$ (except for $D = 0.88$ m, due to experimental limitations) a temperature difference range where no deviations with the measured $h_{t,108}$ value occurred (the so-called stagnant air situation). The $h_{t,108}$ values given by the dashed lines in the figure have been an average value of at least 5 measurements under stratified air conditions. At a critical temperature difference natural convection develops and the total heat transfer increases. The $h_t$ values obtained for the temperature differences larger than this critical temperature difference have been fitted as a linear function of $\Delta T$ ($h_t = a + b \Delta T$ for $\Delta T > \Delta T_{crit}$).

As shown in the figure the smallest heat transfer occurs for $D < 50 \times 10^{-3}$ m at the temperature range of interest. For larger $D$ natural convection will be initiated at a smaller temperature difference.

It should be noted that for each plate distance a different slit structure has been winded and as discussed earlier in chapter 4.4.2 this may lead to deviations due to the quality of the slit structures. Especially the critical temperature differences might be influenced. However for $\theta = 180$ degrees the influence is small. Therefore a reliable conclusion that can be made is that for increasing plate distance the total heat transfer in stratified air conditions decreases.

To examine the effect of the gap width we must consider the measurements performed for each slit structure separately. In fig. 4.22a-d the results have been given for four foil widths ($d = 35, 40, 50$ and $88 \times 10^{-3}$ m). We varied now the gap width $s$ between the slit structure and the absorber. In this case there is no problem of winding a new slit structure when we compare the results. The data points have been omitted for clearness.

The gap width $s$ has been varied by increasing the plate distance $D$. In fig. 4.22a the results have been given for $d = 35 \times 10^{-3}$ m ($A_x = 0.1$). For increasing gap width three effects occur. The heat transfer for stratified air conditions decreases. This is partly due to the increase of $D$ and also due to the increase of the gap width $s$ which results in a less stronger coupling between the heat transfer modes. The third effect concerns the development of the natural convection. Increasing gap width results in a smaller critical temperature difference and a more progressive development of natural convection. In fig. 4.22b the results have been given for foil width $48 \times 10^{-3}$ m and the same effects can be seen. For larger foil widths (fig. 4.22b) the increase of $D$ results for all gap widths in a stronger development of natural convection. Moreover convection has been initiated at smaller temperature differences ($\Delta T < 28$ K). The critical temperature difference for $d = 0.88$ m could not be measured with our experimental set-up. We may expect these for $\Delta T_{crit} < 10$ K. For $d = 0.88$ m the total heat transfer coefficient $h_t$ can be fitted as a linear function of the temperature difference for $\Delta T > 28$ K. The gradient $b$ as found for these cases ($d = 0.88$ m) are more or less equal. This indicates that natural convection has been developed completely.

In table 4.3 the relations obtained for the different slit structures have been given.

In fig. 4.23 the $h_t$ results have been given for stratified air conditions as function of the plate distance $D$. The dashed lines show the decrease of $h_{t,108}$ with increasing plate distance and increasing gap width $s$. Largest heat transfer has been measured for the slit structures with the smallest gap width.

The effects can be explained by the less stronger coupling between the heat transfer modes for increasing $D$ and $s$. For comparison the $h_{t,108}$ values for...
Table 4.3: Results for cross slope slit structures.

<table>
<thead>
<tr>
<th>d (m)</th>
<th>s (%)</th>
<th>$h_{t,180}$ (W/m²K)</th>
<th>$h_t$ (W/m²K)</th>
<th>$\Delta T_{cr}$ (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.035</td>
<td>4</td>
<td>3.14</td>
<td>2.30 + 0.015 $\Delta T$</td>
<td>49.8</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>2.95</td>
<td>2.30 + 0.013 $\Delta T$</td>
<td>42.2</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>2.43</td>
<td>2.31 + 0.020 $\Delta T$</td>
<td>16.8</td>
</tr>
<tr>
<td>0.04</td>
<td>4</td>
<td>3.82</td>
<td>2.22 + 0.020 $\Delta T$</td>
<td>48.8</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>2.78</td>
<td>2.14 + 0.022 $\Delta T$</td>
<td>28.9</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>2.43</td>
<td>2.06 + 0.027 $\Delta T$</td>
<td>13.6</td>
</tr>
<tr>
<td>0.05</td>
<td>4</td>
<td>2.28</td>
<td>2.16 + 0.021 $\Delta T$</td>
<td>22.3</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>2.30</td>
<td>1.81 + 0.020 $\Delta T$</td>
<td>19.4</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>2.13</td>
<td>1.74 + 0.030 $\Delta T$</td>
<td>13.2</td>
</tr>
<tr>
<td>0.06</td>
<td>4</td>
<td>2.28</td>
<td>2.45 + 0.022 $\Delta T$</td>
<td>&lt; 10</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>1.97</td>
<td>2.58 + 0.023 $\Delta T$</td>
<td>&lt; 10</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>1.77</td>
<td>2.73 + 0.021 $\Delta T$</td>
<td>&lt; 10</td>
</tr>
</tbody>
</table>

Figure 4.23: The dependence of $h_{t,180}$ on the relative gap width $s$.

The NHC case have been given in fig. 4.23 also. For $D = 0.091$ m, $s = 12\%$ the obtained total heat transfer coefficient is almost equal to the NHC case. This must be explained by the fact that in this special case the radiative heat transfer reduction by the slit structure balances the increase of coupled conductive heat transfer. For larger $D$ the dashed curves approach the NHC curve. It might be expected that for much larger $D$ the combined effect of decreasing coupled heat transfer and increasing radiation shielding results in $h_{t,180}$ values even smaller than the NHC values.

Application of a gap width $s$ results also in a smaller $\Delta T_{cr}$ . Whether the decrease of $\Delta T_{cr}$ is a result of the increase of $D$ only or also from a loss of suppression quality of the slit honeycomb, can be determined if we convert the $D$, $\Delta T_{cr}$ combinations into critical Rayleigh numbers. For the onset of natural convection a critical Rayleigh number can be defined as

$$ Ra_{c,ex} = \frac{K(T_0)}{h_n c} D^3 (T - T_e) $$

(4.10)

where $Ra_{c,ex}$ represents the experimentally obtained critical Rayleigh number and $K(T_0)$ is a constant dependent on fluid properties evaluated at $T_0$ (for our case $K(T_0) = 0.724 \times 10^8$ m$^{-3}$ K$^{-1}$).

As long as the gap width has no significant influence on the convection suppressing quality of the slit structure, the critical Rayleigh number should be equal for our measurements. The experimentally obtained critical Rayleigh numbers for all our measurements have been given in fig. 4.24.

In the figure the line represents the average critical Rayleigh number for the measurements, where the slit structure has the smallest possible gap with
the hot plate ($s = 4\%$). This Rayleigh value amounts $2 \times 10^5$. The slit structures with a gap of $7\%$ still show this critical Rayleigh number, within the experimental error (shaded area). This means that these slit structures suppress the natural convection as well as the $4\%$ slit structures. The critical temperature difference for a foil width of $d = 0.08\text{ m}$ has not been measured but has been taken $\approx 4\text{ K}$. This results also in a critical Rayleigh number of $\approx 2 \times 10^5$. It must be concluded that a small gap ($s < 7\%$) does not influence the convection suppressing quality of the cross slope slit structure. However a decrease of $Ra_{cex}$ has been obtained for the gaps of $12\%$, indicating the loss of suppressing quality of the slit honeycomb.

The critical Rayleigh number as predicted according to eq. (4.1) of Hollands (1978) for our case lies between $2.3 \times 10^5$ and $2.8 \times 10^5$ for plate distances of $0.0365\text{ m}$ up to $0.854\text{ m}$. Equation (4.1) matches experimental SS-results of Hollands rather well, but for the SB-case ($A_x = 0.2$) a $28\%$ smaller critical Rayleigh number has been found. Extrapolating this for our measurements should result in critical Rayleigh numbers from $1.8 \times 10^5$ up to $2.2 \times 10^5$. This is in excellent agreement with our results.

A comparison between our measurements for $A_x = 0.1$ (including $4\%$ and $7\%$ gaps only) and predictions of Hollands according to eq. (4.2) is shown in fig. 4.25. The results are in excellent agreement with Hollands. Since eq. (4.2) has been derived for no-gap cases the results show once more that the convection suppressing quality of the slit structure has not yet been influenced by the introduction of the gaps.

In addition we performed measurements for upslope slit structures ($A_z = 0.1$ and $A_x \approx m$) with foil of width $d = 0.05$ and $d = 0.80\text{ m}$. For these cases the gap width $s$ has been varied also. The results of these measurements have been given in fig. 4.26.

In fig. 4.26 the results have been given for $d = 0.05\text{ m}$. First it is noted that a critical temperature difference, as found for the cross slope slit structures, has not been measured. This indicates that natural convective heat transfer occurs even for smaller temperature differences than the range measured here. Secondly the heat transfer coefficients for stratified air conditions have been found equal to the corresponding cross slope values (within our experimental accuracy of $4\%$). This is obvious since for $\theta = 180$ degrees cross slope and upslope slit structures are indistinguishable. However it shows the accuracy of our measurements.

The quantitative results for the upslope slit structures have been given in table 4.4.

<table>
<thead>
<tr>
<th>$d$ (m)</th>
<th>$s$ (%)</th>
<th>$h_t,100$ (W/m²K)</th>
<th>$h_t$ (W/m²K)</th>
<th>$d\Delta T_{cr}$ (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05</td>
<td>4</td>
<td>2.63</td>
<td>2.65 + 0.810 $d\Delta T$</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>2.36</td>
<td>2.39 + 0.816 $d\Delta T$</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>2.11</td>
<td>2.34 + 0.821 $d\Delta T$</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>0.80</td>
<td>4</td>
<td>2.20</td>
<td>2.44 + 0.819 $d\Delta T$</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>2.04</td>
<td>2.28 + 0.824 $d\Delta T$</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>1.77</td>
<td>2.29 + 0.825 $d\Delta T$</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

Table 4.4: Results for upslope slit structures.

fig. 4.25. The results for the upslope slit structures have been given in table 4.4.
We correlated our $h_t$ values for $\Delta T > 26$ K and obtained increasing values for $b$ for increasing gap width, although the values are smaller than the corresponding cross slope values. In fig. 4.26b the results have been given for $d = 0.08$ m. Again the $b_{188}$ values agree extremely well with the corresponding cross slope values. Stagnant air situations are not found in the inclined situations and increasing $b$ values have been obtained for increasing gap width. However, in comparison with the cross slope slit structures the $b$ values for the cross slope structures are equal or larger, indicating stronger development of natural convection.

The fact that critical temperature differences have not been obtained for upslope slit structures is due to the fact that in this geometry a base flow exist for every Rayleigh number not equal to 0. The flow is for small Rayleigh numbers ($Ra \approx 2000$) in the so-called conduction regime ($Nu_b \approx 1$). Increasing Ra leads to a continuous development of the base flow and increasing convective heat transfer, whereas for cross slope slit structures natural convection has been suppressed totally up to a critical Rayleigh number. Application of a large gap between the absorber and the slit structure is according to our measurements not recommended for upslope slit structures.

For both temperature differences ($\Delta T = 40$ K and $40$ K) the results obtained for upslope slit structures differ not significantly for $D > 0.05$ m. The obtained $h_t$ values are equal within the experimental error. For $D \geq 0.08$ m the upslope slit structure is superior to the cross slope slit structure for both temperature differences, but the obtained values are larger than the values for $D \geq 0.05$ m.

In conclusion slit honeycomb of $A_x = 0.1$ should be made for $D \geq 0.05$ m in combination with a gap of $7\%$ for cross slope structures and preferable less than $4\%$ for upslope structures. As discussed before the difference between upslope and cross slope slits is small. However cross slope slits should be preferred if one considers the manufacturing tolerance, which is larger for cross slope slit structures. Even a gap of $7\%$ has no influence on $h_t$. For upslope slit structures application of such a large gap is not recommended. Furthermore since at very large temperature differences natural convective heat transfer shall be larger for cross slope slit structures the stagnation temperature of the solar collector shall be lower in this case than for the collector with an upslope slit structure. This results in less severe requirements for the CSD material.

4.5 Solar transmissivity of the slit structures.

Application of convection suppressing devices in solar collectors results not only in decreased top heat losses of the collector but also in a loss of solar absorption on the absorber plate. This is due to the absorption of solar radiation by the CSD. In general the maximum absorption of solar radiation in a conventional solar collector is determined by the transmissivity of the cover, $\tau_c$, and the absorptivity of the absorber $\alpha_p$. Here $\alpha_p$ and $\tau_c$ represent total values for the solar spectrum. The product of $\alpha_p$ and $\tau_c$ determines the maximum efficiency of the solar collector $\eta_{max}$. For a CSD collector this value will be decreased by a factor $\tau_{hc}$, where $\tau_{hc}$ represents the total solar transmissivity of the CSD.

4.5.1 The small solar simulator.

To find the solar transmissivities of the slit structures applied we build a small solar simulator. A scale model of a solar collector area
Figure 4.23: Sketch of the small solar simulator.

Figure 4.29: A picture of the solar simulator.

The artificial sun has been composed of nine quartz halogen lamps (Philips, type 13177). The lamps have been positioned as shown in fig. 4.30a to obtain an as good as possible uniform parallel beam of light.

The spectral distribution of the lamps is dependent on the voltage applied.
and has been given for a single lamp in fig. 4.36b for the voltage advised (17 V - 9 A). At this operating level the lifetime of the lamps is about 1000 hours. The lamps have been placed in series in an electrical circuit with a stabilized DC-source (248 V - 38 A).

The absorber has been placed in a perspex box. The box has been insulated at the bottom by 8.025 m thick roofmate insulation. This resulted in a heat loss coefficient to the back of ≈ 1.2 W/m²K. The copper absorber has been fitted in the insulation as shown in fig. 4.31.

The absorber consists of two copper plates with in the bottom plate a channel structure for leading the heat transporting fluid (water). At the surface of the upper plate a spectral selective layer has been used. The measurements for the Hostaphan structures have been done with a black nickel foil, as spectral selective layer, marketed under the name Maxorb, manufacturer MPD ltd., England ($\alpha_p = 0.95$, $\epsilon_p = 0.14$). This spectral selective layer has been replaced later by a layer of cobalt–ferro-oxide (CoFe$_2$O$_4$). This layer has spectral selective values of $\alpha_p = 0.94$ - 0.98 and $\epsilon_p = 0.18$. The cover plate of the model has been a plate of low iron glass ($\alpha_c \approx 0.9$). The distance between the cover and the absorber has been varied by placing the absorber (with insulation) at different depths in the box. At the sides of the box precautions have been taken for winding the slit structures.

Figure 4.31: Schematic view of the collector model.

4.5.2 Determination of $\eta_{\text{HC}}$

To determine the transmissivity of a slit structure two series of collector efficiency measurements are needed. The first series concerned the determination of the maximum efficiency of the collector model without slit structure ($\eta_{\text{B,HC}}$). The second series concerned the maximum efficiency with slit structure ($\eta_{\text{B,HC}}$). The solar transmissivity of the slit structure is then defined by

$$\eta_{\text{HC}} = \frac{\eta_{\text{B,HC}}}{\eta_{\text{B,HC}})}$$

(4.11)

The values of the maximum efficiencies have been derived by calculation of the efficiency at $T^* = 0$ ($T^* = (T_p - T_a)/(E_\text{p})$ [W/m²]) by fitting a second order polynomial to the collector efficiency data. The data have been found for a series of at least 7 collector efficiency measurements at different $T^*$.

For steady state conditions the efficiency of the collector model is given by

$$\eta = \frac{P_C}{\rho_v \theta_v A T_f} = \frac{Q_u}{A_p \langle E_s \rangle}$$

(4.12)

Here $P$ and $C_p$ represent the density and heat capacity of the heat removal fluid through the collector at the average collector temperature, $\theta_v$ the volume flow through the collector and $\Delta T_f$ the temperature difference between the outlet and inlet fluid temperature. $A_p$ is the absorber area and $\langle E_s \rangle$ is the average insolation at the outer surface of the cover plate of the artificial sun. The difference of the fluid temperature between the outlet and inlet of the collector has been measured with a temperature difference meter, specially designed and developed by the Institute of Applied Physics, TPD-TNO/TH. The temperature difference meter has been calibrated in combination with the flow meter. The calibration procedure has been described in Appendix D. The calibration resulted in a relation between the useful heat transported by the fluid ($Q_u$) and the signal of the temperature difference meter $\Delta V$ (Volt).
with \( \alpha_u \) a calibration constant, which is dependent on the flow rate and the mean temperature of the fluid. The value of \( \alpha_u \) has been given dependent on the mean fluid temperature \( \langle T_f \rangle \) for different read outs of the flow meter (\( \theta_{ro} \) non-linear scale \( 0 - 20 \)). The result of the calibration has been given in fig. 4.32.

Correlation of the calibration measurements resulted in

\[
\alpha_u = \alpha_0 + a_1 \langle T_f \rangle + a_2 \theta_{ro} + a_3 \theta_{ro}^2
\]  

(4.14)

with \( a_0 \), \( a_1 \) and \( a_2 \) constants, which are dependent on the mean fluid temperature (see Appendix D).

The mean fluid temperature \( \langle T_f \rangle \) has been determined by measuring the outlet fluid temperature \( T_{fo} \) and the flow rate \( \theta_{ro} \). Using the constant sensitivity of the temperature difference meter \( S_{TDM} \) for mass flows larger than \( 7 \times 10^{-3} \) Kg/s (which for our measurements mean \( \theta_{ro} \) 15) results in

\[
\langle T_f \rangle = T_{fo} - \frac{S_{TDM} \cdot \Delta V}{2}
\]  

(4.15)

After substitution of \( \langle T_f \rangle \) in eq. (4.14) and eq. (4.13) the useful heat gain \( Q_u \) can be determined. The outlet fluid temperature and the ambient temperature have been measured with copper-constantan thermocouples. The relative error in the determination of \( Q_u \) amounts 2% (see Appendix D).

Since we make use of nine small solar lamps the local intensity of the insolation at the absorber surface varies. This is due to the fact that the beam is not uniform and parallel. Therefore the solar insolation \( \langle E_s \rangle \) has been determined by averaging the local insolation over the absorber area. We measured the local insolation with a Kipp solarimeter, type CM 18, sensitivity \( S_{sol} = 5.76 \times 10^{-6} \text{ Vm}^2/\text{W} \), at 78 positions at about 0.85 m above the absorber area. The average insolation needed for the efficiency calculations has been determined by

\[
\langle E_s \rangle = \frac{1}{78_{sol}} \sum_{i=1}^{70} E_{s,i}
\]  

(4.16)

The insolation at the absorber is less than 5% than the value given by eq. (4.16). This is due to the difference in distance to the artificial sun and the divergence of the beam. However it is not necessary to correct the average insolation values as long as the obtained efficiency values are compared with the results of measurements performed in the same situation.

Each time a parameter has been varied (for instance angle of insolation \( \theta_s \) or the lamp voltage) the average insolation \( \langle E_s \rangle \) has been determined.

An example of the local line average insolation distribution over the absorber has been given in fig. 4.33 for \( \theta_s = 90 \) degrees.

In figure 4.33a the average insolation along the height and in fig. 4.33b along the width of the absorber has been given. As shown in the figure the largest deviations occur at the edges of the absorber. Moreover the variation over the height is much larger than over the width of the absorber (from -28 % to +10 % and -6 % to +5 %, respectively).

The inaccuracy of the measured insolation have been derived by repetition of
Figure 4.33: The average insolation distribution at the absorber.
(a) along the height, (b) along the width.

These measurements. The estimated value of the inaccuracy is 3%. Together with the 2% relative error in the determination of $Q_u$ the maximum relative error in determination of $\eta_0$ 5%. For an insolation angle of 70 degrees the relative error amounts 5% also. However the insolation at the absorber surface is less uniform, due to increased distances differences to the artificial sun.

4.5.3 Results solar simulator.

In fig. 4.34 the results for the efficiency measurements have been given for $D = 8.83 \text{m}$ and $\theta_s = 90$ degrees. The data points have been noted also.

In addition the NHC case has been given.

A second order curve fit of the experimental data resulted in an efficiency curve as given by

$$\eta = \eta_0 - a_1 T^8 - a_2 \left(E_s^\circ\right) T^2$$

(4.17)

In this formula $\eta_0$ represents the maximum efficiency. The efficiency values have been given for $\left(E_s^\circ\right) = 899.2 \text{ W/m}^2$. By using eq. (4.17) it is possible to evaluate the efficiency curves for other insolation values. The efficiency curves given in the following figures have been depicted for insolation values $\left(E_s^\circ\right)$ of $880 \text{ W/m}^2$. This is a common insolation value for collector tests.

Figure 4.34: Obtained datapoints for determination of $\eta_0$
for $D = 8.03 \text{ m}$, using Hostaphan BN 50 foil.

In fig. 4.35 the efficiency curves have been given for $D = 8.85 \text{ m}$ and $8.93 \text{ m}$ and $\theta_s = 98$ and 78 degrees.

Consider first the results for $\theta_s = 98$ degrees (fig. 4.35a). As mentioned before our quantitative results can be compared only within one series of measurements (see chapter 4.5.2). Therefore the measurements as given by
curve 1 (NHC, D = 0.05 m) should be compared with curve 3 and for the same reasons curve 2 with curve 4. For both series the \( \eta_b \) - value for the NHC situation is larger than for the slit structure collector with aspect ratio 0.1. Moreover the two \( \eta_b \) - values for NHC situations have been found equal, within the experimental error. The \( \eta_b \) - value for \( A_x = 0.1 \) and \( D = 0.03 \) m has been found considerably smaller than for \( A_x = 0.1 \) m and \( D = 0.05 \) m. This is due to the increased number of slits per collector area.

<table>
<thead>
<tr>
<th>material</th>
<th>( D )</th>
<th>( \eta_b )</th>
<th>( \tau_{hc} )</th>
<th>( \theta_s = 90 )</th>
<th>( \theta_s = 70 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>NHC</td>
<td>BN 50</td>
<td>0.63</td>
<td>0.874</td>
<td>0.883</td>
<td></td>
</tr>
<tr>
<td>0.1</td>
<td>&quot;</td>
<td>0.63</td>
<td>0.834</td>
<td>0.955</td>
<td>0.768</td>
</tr>
<tr>
<td>NHC</td>
<td>&quot;</td>
<td>0.85</td>
<td>0.874</td>
<td>0.788</td>
<td></td>
</tr>
<tr>
<td>0.1</td>
<td>&quot;</td>
<td>0.85</td>
<td>0.866</td>
<td>0.991</td>
<td>0.766</td>
</tr>
<tr>
<td>NHC</td>
<td>FEP Teflon</td>
<td>0.84</td>
<td>0.888</td>
<td></td>
<td>0.888</td>
</tr>
<tr>
<td>0.1</td>
<td>&quot;</td>
<td>0.84</td>
<td>0.871</td>
<td>0.986</td>
<td>0.776</td>
</tr>
</tbody>
</table>

Table 4.5: The measured \( \eta_b \) - values and slit structure transmissivities for the solar simulator experiments.

In table 4.5 the obtained \( \eta_b \) and \( \tau_{hc} \) for the two series have been given. Also given in the table 4.5 are the results for the FEP Teflon slit structure (\( D = 0.04 \) m), which have been derived with CoFeO as spectral selective layer. This resulted in a larger \( \eta_b \) - value. For all series extremely high \( \eta_b \) - values have been measured. This can be explained by the high glass transmissivity (\( \tau_c \approx 0.9 \)) and high absorptance of the spectral selective layer (for both layers applied: \( \alpha_p \approx 0.95 \)). The collector efficiency factor and the heat removal factor of the collector model are approximately 1. This should result in an \( \eta_b \approx (\alpha_p \tau_c) \approx 0.85 \). However we measured higher values for all cases. This can be explained by the fact that no correction has been made for the infrared radiation absorbed by the absorber. Such a correction on the maximum efficiency would decrease the \( \eta_b \) - value several percents, depending on the amount of infrared radiation of the lamps. However the solarimeter measures the infrared contribution also. More likely is the systematically too high measured \( \eta_b \) due to a not correct (too high) calibration value of the solarimeter. A new calibration of the solarimeter is therefore necessary. Nevertheless the \( \eta_b \) - values measured and the expected values are equal within the experimental accuracy.

An overall conclusion from the efficiency measurements with the solar simulator is that the transmissivity of the slit structures is large (\( \tau_{hc} > 0.95 \)) and the deviation with the NHC cases is within the estimated error. Secondly the influence of the angle of insolation on the slit structure transmissivity is small for the angles considered (70° < \( \theta \) < 90 degrees). An estimated value for \( \tau_{hc} \) for all measurements is

\[
\tau_{hc} \approx 0.95
\]  

(4.18)

4.6 Total heat loss measurements on flat plate solar collectors.

So far we considered the influence of the slit structures on the heat loss from the absorber to the cover (chapter 4.4) and the effect of the slit structures on the solar absorptance of the absorber (chapter 4.5). In this chapter measurements shall be described concerning the influence of the slit honeycombs on the total heat losses of flat plate solar collectors.

For this purpose slit structures have been applied in the spacing of commercial flat plate collectors. We used for our experiments three different flat plate solar collectors. The three collectors chosen have been: (1) a black collector manufactured by EBS BV, Grave in The Netherlands, (2) a spectral selective collector, \( \theta_h \approx 0.25 \) (EBS BV), (3) a spectral selective collector, \( \theta_h \approx 0.15 \), manufactured by DRU BV, Oss, The Netherlands. The main differences between the original collectors and the collectors after slit honeycomb application, is the increased absorber to cover spacing (from \( \approx 0.015 \) m to \( \approx 0.06 \) m) and secondly a slightly improved back insulation.

4.6.1 Collector testing.

To compare different collectors with each other an uniform collector testing method for the determination of collector efficiency has been developed. The test method has been described by the Commission of the European Communities in a report titled "Recommendations for European Solar Collector
test methods, liquid heating collectors* (CEC (1989)). This report describes both indoor and outdoor collector tests.

A schematic representation of a closed loop test installation for outdoor

![Schematic representation of a closed loop test installation](image)

Figure 4.36: Schematic representation of a closed loop test installation.

...instantaneous efficiency measurements have been given in fig. 4.36. The most important measurements, which have to be performed for determination of the instantaneous efficiency of a solar collector, are:

1. Measurement of the in- and outlet temperature of the heat transfer fluid.
2. The ambient temperature.
3. The flow rate of the collector heat transfer fluid.
4. The solar radiation.
5. The surrounding wind speed.

In addition have to be determined
6. The angle of incidence of the direct solar radiation.
7. The pressure drop across the collector.

Severe guidelines have been given for the accuracy of the different measurements. For instance the temperature difference over the collector has to be measured to an accuracy of ± 0.1 K and thermopiles or a matched pair of platinum resistance thermometers should be used for this purpose. Moreover mixing elements should be used to ensure an uniform fluid temperature. The fluid flow rate should be measured within 1% and for this purpose magnetic flow meters have been recommended. For collector efficiency determination indoors the solar radiation of the artificial sun should be averaged over the aperture area.

Main test conditions for the measurements are

1. The heat transfer fluid should flow from the bottom of the collector to the top.
2. The fluid flow rate should be 0.02 kg/s ± 10% per square metre collector area.
3. The temperature rise of the fluid, traveling through the collector, should be between 1.5 and 15 K.

The instantaneous efficiency \( \eta \) may be calculated from

\[
\eta = \frac{Q_u}{A \cdot \langle E_s \rangle} \tag{4.19}
\]

with \( Q_u \) is the useful heat gain derived from

\[
Q_u = \rho \cdot C_v \cdot \langle T_m - T_i \rangle \tag{4.20}
\]

Measurements for several inlet temperatures lead to a number of efficiency values, which are usually presented in a figure containing the efficiency \( \eta \) dependent on \( T^\# \). Here the variable \( T^\# \) is defined by:

\[
T^\# = \frac{T_m - T_a}{\langle E_s \rangle} \tag{4.21}
\]

where \( T_m = 0.5 (T_i + T_o) \). Curve fitting of the \( \eta, T^\# \) combinations results in a second order polynomial

\[
\eta = \eta_0 + a_1 T^\# - a_2 \langle E_s \rangle T^\#^2 \tag{4.22}
\]

A second important parameter which has to be determined is the total heat loss factor of the collector. Using eq. (4.22) and eq. (4.19) the total heat loss factor is defined by
This way indoor heat loss measurements give an indication of the performance of the solar collector, when the \( T_0 \) point is known. Moreover this method is in use to determine the instantaneous efficiency curve for solar collectors by combining indoor and outdoor tests. The main difference between the efficiency measurement procedure and the heat loss measurements method lies in the fact that the flow direction through the collector should be reversed for the heat loss test. This is in order to assure that the temperature profile of the absorber should be similar to the profile at the efficiency measurements.

The heat loss rate of the collector can than be derived by

\[
Q_L = \frac{C_P}{\rho v L} (T_m - T_o)
\]  

(4.24)

Measurements at several inlet temperatures result in a number of data points, which are presented graphically with \( Q_L \) as a function of \( (T_m - T_a) \). The collector heat loss coefficient can than be calculated from

\[
U_L = \frac{Q_L}{A_p (T_m - T_a)}
\]

(4.25)

The heat loss coefficient can be fitted as a linear function of the temperature difference \( T_m - T_a \) as given by eq. (4.23).

4.6.2 The experimental set-up for total heat loss measurements.

The experimental set-up used for the total heat loss measurements of flat plate solar collectors has been given in fig. 4.37. The collector has been placed in an angle of 45 degrees with the horizontal. The fluid flow through the collector is driven by a thermostat bath, which regulates the temperature level and the mass flow of the fluid. The flow direction and the mass flow have been chosen as suggested by the Commission of European Communities (1980, see chapter 4.6.1). According to this scheme the fluid shall be cooled by flowing down through the collector.

A fan has been used to simulate an outdoor wind on the cover plate of approximately 5 m/s. The spacing between the glass cover and the absorber of the solar collectors used has been increased to 0.86 m. The slit structures have been made by winding Hostaphan BN 58 foil cross slope between two metal combs, attached at the outer ends of the absorber. The modification of the collectors have been given in fig. 4.38. The back and side heat losses of the collectors have been calculated according to a one dimensional model. All three absorber plates have been made by spot welding of two separate stainless steel plates. This resulted in a not perfectly flat absorber surface. However the heat removal factor \( F_R \) of the collectors is approximately
For the EBS absorbers these gap heights are about \( \approx 2 \times 10^{-3} \) m, while for the DRU absorber the gap heights are approximately \( \approx 5 \times 10^{-3} \) m. Due to this effect a close fit of the slit honeycomb with the absorber could not be achieved. As already explained in chapter 4.4.4 and chapter 3.4.3, these small gaps have a positive influence on the total heat loss factor.

Measurement of the temperature difference of the fluid between the in- and outlet of the collector and the flow rate through the collector results in the total heat loss of the collector. The total heat loss factor can then be defined by:

\[
U_L = \frac{m \cdot C_p \cdot (T_i - T_o)}{(T_p - T_a) \cdot A_p}
\]  

with \( m \) the mass flow, \( C_p \) the heat capacity of the fluid, \( (T_i - T_o) \) the temperature difference between in- and outlet, \( T_p \) the average absorber temperature, \( T_a \) the ambient temperature and \( A_p \) the aperture area of the collector.

The mass flow, through the collector, has been measured by means of a calibrated flowmeter. All the temperatures have been measured with calibrated Fe-Co thermocouples. The temperature difference of the fluid has been measured additionally with a thermopile of four Fe-Co thermocouples. Mixing elements in the pipes assures premixing of the fluid before measurement of the temperatures took place. The inaccuracy in the measurements is large, especially at low temperature level and for small heat loss factors. The total relative error in the total heat loss measurements is determined mainly by the error in the temperature difference measurements. For small temperature differences this relative error exceeds 10%.

The relative errors in the total heat loss factor have been given in table 4.6 for several heat loss factors at and temperature levels.

A significant systematic error occurs in the heat loss measurements. By performing the experiments indoors, an important heat loss mechanism has been reduced. The radiative heat loss from the cover to the surroundings is determined by the cover and ambient temperature for indoor measurements and by the cover and sky temperature for outdoor measurements. This systematic error results in a too low measured heat loss factor. However we are primarily interested in the influence of the slit structure on the total heat loss factors, and less interested in the absolute values of the heat loss factors.

<table>
<thead>
<tr>
<th>( T_p - T_a )</th>
<th>10 K</th>
<th>20 K</th>
<th>30 K</th>
<th>40 K</th>
<th>50 K</th>
<th>60 K</th>
<th>70 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>( U_L = 2 )</td>
<td>27.8</td>
<td>14.8</td>
<td>9.7</td>
<td>7.6</td>
<td>6.3</td>
<td>5.5</td>
<td>4.9</td>
</tr>
<tr>
<td>( U_L = 3 )</td>
<td>19.9</td>
<td>10.6</td>
<td>7.4</td>
<td>5.9</td>
<td>4.9</td>
<td>4.3</td>
<td>3.9</td>
</tr>
<tr>
<td>( U_L = 4 )</td>
<td>16.5</td>
<td>8.8</td>
<td>5.6</td>
<td>4.3</td>
<td>3.8</td>
<td>3.7</td>
<td>3.4</td>
</tr>
<tr>
<td>( U_L = 5 )</td>
<td>14.4</td>
<td>7.9</td>
<td>4.5</td>
<td>3.8</td>
<td>3.4</td>
<td>3.4</td>
<td>3.1</td>
</tr>
<tr>
<td>( U_L = 6 )</td>
<td>13.1</td>
<td>7.1</td>
<td>4.2</td>
<td>3.6</td>
<td>3.2</td>
<td>2.9</td>
<td></td>
</tr>
<tr>
<td>( U_L = 7 )</td>
<td>12.1</td>
<td>6.6</td>
<td>4.8</td>
<td>3.9</td>
<td>3.4</td>
<td>3.8</td>
<td>2.8</td>
</tr>
</tbody>
</table>

Table 4.6: The relative error in the total heat loss factor.

Therefore the heat loss factors obtained have not been corrected for this systematic error.

4.6.3 Results of the total heat loss measurements.

In fig. 4.39 the results have been given for the three flat plate collectors. The figures 4.39 a-c show the results for each collector without forced ventilation at the cover. The figures d-f show the measured \( U_L \) values, which have been obtained with a simulated wind velocity of approximately 5 m/s.

For each case data points have been given for the collector with (noted HO) and without (NHC) slit honeycomb. The dashed lines represent the calculated side and back insulation heat losses. Comparison of the NHC results for the three collectors, show that the largest total heat loss factor has been measured for the EBS/BB-collector with wind simulation. It is evident that this is due to the large emissivity of the absorber. Both spectral selective collectors have a considerably lower heat loss factor. The DRU/SB (\( \varepsilon_h \approx 0.15 \)) collector has the smallest heat loss factor for all cases considered.

Without wind simulation the heat loss factors are lower.

Application of a slit honeycomb results in a reduction of the total heat loss factor. For the EBS/BB-collector the reduction is considerably larger than for the two SB-collectors. This large reduction (\( \approx 2.3 \) W/mK) is due to the suppression of the natural convective heat transfer and the reduced radiative heat transfer of the absorber.
Figure 4.39: Measured indoor total heat loss factors.

EBS/BB collector, with a slit structure, is only slightly higher than the DRU-SB/NHC collector. For the SB-collectors the reduction is less but still considerably (≈ 1 W/m²K). The smallest heat loss factor has been found for the collector with the smallest emissivity of the absorber. Despite the large relative errors for small temperature differences (indicated in fig. 4.39a) a first order fit for the total heat loss factor as function of the temperature difference seems justified for all measurements. The relations derived are of the form:

$$U_L = a_1 + a_2 (T_p - T_a)$$  \hspace{1cm} (4.27)

In table 4.7 the obtained relation constants have been given.

For the collectors concerned the application of a slit structure reduces predominantly the coefficient "a1". This coefficient represents the total heat loss factor for the collector at ambient temperature ($U_{L_B}$). For the EBS/BB-collector a reduction of "a2" has also been achieved, indicating the reduced influence of the radiative heat loss of the absorber. For the SB-collectors the coefficient "a2" is more or less constant.

It should be noted that in no case the total heat loss factor for the HC-collectors remained constant up to a critical temperature difference, as obtained for the caloric heat transfer measurements in chapter 4.4.4, but increased monotonically for increasing temperature difference. This can be explained by the fact that due to the increase of the average absorber temperature ($T_p$), the radiative heat transfer increases from absorber to cover and from cover to the surroundings also. For the caloric measurements of chapter 4.4.4 we kept the average temperature constant. Moreover due to the spacing of 0.04 m and the less tightly winded slit structures in comparison with the slit structures in the experimental model, used for the caloric measurements, the onset of natural convection may be expected at smaller temperature differences. Furthermore the effect of a constant $h_L$ value has only a partial effect on the total heat loss factor of the collector.

The quality of different flat plate solar collectors are compared usually by comparing $U_L$ and $U_L$ at $T^* = 0$ ($a_1 = 0$ in eq. (4.27)). Hereby one neglects the temperature dependence of $U_L$, considering "a2" equal for the different collectors. That this is no longer true can be seen in table 4.7.

It is advantageous to use a slit honeycomb in the EBS/BB-collector results in a reduction of "a2" also. Furthermore a considerably reduction of the total heat loss factors can be gained if one improves the back and side insulation of the collectors. This should decrease $U_{ins}$ to approximately 0.5 W/m²K.
CHAPTER 5

UTILIZATION AND PROTOTYPE

5.1 Introduction and parameter determination.

The main purpose of our investigation has been the improvement of flat plate solar collectors, by reduction of natural convective heat transfer in the spacing between the absorber and cover plate of the collector, using slit honeycomb structures. Application of these CSD's should result in a significant increase of performance of the solar collector against a marginal increase of costs.

The material used for fabricating the CSD and the production process are important for a cost effectiveness analysis. However these aspects have not been considered in our investigation.

From a physical point of view requirements can be defined for the material used.

(a): Large solar transmissivity. This should result in a total CSD solar transmissivity of at least 0.95.

(b): Low conduction heat losses through the material.

(c): The CSD should withstand the stagnation temperatures, which occur in the solar collector. Due the decreased total heat loss factor the stagnation temperatures shall be higher than for the conventional collectors.

(d): UV durability for the life time of the collector.

Moreover the flat plate CSD solar collector should compete vacuum tube collectors at solar engineering fields, as for example domestic hot water supply, heating and cooling of buildings, seasonal heat storage etc.

In this chapter we describe the design and development of a prototype flat plate slit honeycomb solar collector, based on our experimental and numerical results. The results and experiences with two collectors shall be discussed.

The first collector has been the commercially available EBS Sunstrip collector in which we applied a slit honeycomb. The second collector has been a specially for our purpose designed and build solar collector.
5.2 Evaluation of numerical and experimental results.

From our numerical natural convection calculations, we concluded that an aspect ratio of at least 0.1 for slit structures should be necessary to suppress the natural convection sufficiently. Therefore our experiments and also our slit structures, which we applied in solar collectors, have been made of this aspect ratio. This aspect ratio has been used also for the prototype slit honeycomb solar collectors.

The interaction between radiative heat exchange and convective/conductive heat transfer has been explained. Moreover it has been shown that a gap, between the slit structure and the absorber reduces this interaction significantly, particularly for spectral selective collectors. Furthermore it has been shown (although for the two dimensional case only) that the critical Rayleigh number for the onset of natural convection does not change significantly, despite this interaction.

The experimental results showed once more very clearly the interaction between radiative, convective and conductive heat transfer in slit honeycombs. For cross slope slit structures the optimum cover to absorber spacing is about 0.83-0.85 m. An optimum gap width of about 7% of the spacing D has been obtained. For larger gaps the strong development of natural convection results in an increase of total heat transfer. For smaller relative gap widths the coupled heat transfer will be enlarged.

In addition total heat loss measurements on flat plate solar collectors showed that improvement of the back and side insulation of the collector can decrease the total heat loss factor of the collector with approximately 0.5 W/m²K. The choice of the CSD material shall be based primarily on the solar transmissivity, but considering the coupled heat transfer calculations a very small emissivity (< 0.10) in the infrared wavelength region is desirable.

5.2.1 Design of the high efficiency flat plate solar collector.

We have used two solar collectors for application of the slit honeycomb structures. The first collector, EBS Sunstrip, is commercially available on the Dutch market. The absorber consists of a series of 6 fin absorbers, which have been connected by two header pipes (see fig. 5.1).

Each fin consists of a 5 \times 10^{-4} m thick aluminum plate with in the center a flow channel of 8 \times 10^{-3} m diameter. At the ends of the flow channel connecting pipes with the headers have been sealed. The cover of the collector is a 3 \times 10^{-3} m thick roughened tempered low-iron glass. The cover plate has a measured solar transmissivity of \approx 0.92. At the absorber surface a spectral selective layer has been applied according to a chemical process. The solar absorptivity of the selective layer amounts approximately 0.95 and the emissivity amounts \approx 0.10 . The aperture area of the collector is 1.41 m².

The absorber has been placed in an aluminum casing on top of a 0.1 m thick layer of insulation material (0.06 m rockwool, \lambda = 0.035 W/mK and 0.04 m roofmate, \lambda = 0.028 W/mK). The slit honeycomb has been wound along glass pillars (diameter \approx 4.5 \times 10^{-3} m) placed in an aluminum frame. The glass cover has been placed in such a way on the casing that a close fit has been realized between the glass cover and the slit structure.

In fig. 5.2 the construction has been sketched for the two longer sides of the collector. The two shorter sides have been insulated up to the cover. The aluminum frame has been made of two U-balks connected by two iron bars at the top and bottom of the frame (hidden in the insulation of the sides of the collector). The frame has been placed on two aluminum holders along the long sides of the casing. The inner width of the U-balks has been 48 \times 10^{-3} m, the thickness has been 4 \times 10^{-3} m. This way a gap width of approximately 7% of
the absorber to cover spacing has been realized.

The second collector has been developed after the test results of the EBS Sunstrip were known. Therefore this collector has been changed on two essential points. First, and most important change, concerned the absorber plate. Instead of a fin absorber the absorber has been made by spot welding two 5 $10^{-4}$ m thick stainless steel plates. The wet area of the absorber has been significantly enlarged and this way the heat transfer from the absorber to the fluid has been improved ($F_R = 1$). Four strips of a spectral selective material, Cusorb foil $10^{-4}$ m thick, manufacturer MPD, England, has been glued on the surface of the absorber with a special adhesive. The solar absorptivity of the Cusorb foil is 0.98 and the emissivity amounts to 0.09. Again the cover has been made of tempered low-iron glass.

The second change has been the reduction of the size of the collector. The absorber has been reduced to an almost square absorber with aperture area $0.97 \text{ m}^2$. This has been done to reduce the weight of the collector. As a consequence the sides heat losses increase relatively. The collector casing and slit honeycomb frame were equal to the casing and frame as described for the EBS Sunstrip collector.

An overall view on the prototype slit honeycomb collector has been given in fig. 5.3.

According to the CEC recommendations (1980) the efficiency curve for a
d solar collector should be presented as a second order polynomial of the variable $T^*$ ($= (T_p - T_a) / \langle E_s \rangle$).

$$
\eta = \eta_0 - a_1 T^* - a_2 \langle E_s \rangle T^*^2
$$

(5.1)

where $\eta_0$ represents the maximum efficiency, $a_1$ and $a_2$ are polynomial constants and $\langle E_s \rangle$ is the average solar insolation at the aperture area. Duffie and Beckmann (1989) give an equation (equation (6.7.6)) for the efficiency of a flat plate solar collector as

$$
\eta = \frac{Q_u}{A_c \langle E_s \rangle} = F_R \left[ \frac{a_1 T_c - U_L \langle E_s \rangle}{\langle E_s \rangle} \right]
$$

(5.2)

In this equation $T_i$ is the inlet fluid temperature of the collector, $F_R$ the heat removal factor of the solar collector and $U_L$ the total heat loss factor. $U_L$ is a function of the temperature difference ($U_L = U_g + b x (T_i - T_a)$). The heat removal factor relates the actual useful energy gain $Q_u$ of the collector to the useful energy gain which will be derived when the whole collector is at the fluid inlet temperature and is determined by the collector efficiency factor $F'$ and the collector flow factor $F^*$. 

---

**Figure 5.2**: Sketch of the side wall of the casing of the slit honeycomb collector.

**Figure 5.3**: The prototype slit honeycomb collector.
Comparison of eq. (5.1) with eq. (5.2) shows that: $\eta_0$ equals $F_R = F' F''$ (5.3)

In eq. (5.1) $U_L$ is related to $(T_p - T_a)$. By defining $T^\# = (T_p - T_a)/(E_s)$ the correction for the heating of the fluid along the absorber is no longer needed. This means that the value of $F''$ can be taken equal to unity. So $a_1$ represents $U_L(T^\# = 0) = U_{L0}$ and $a_2$ is the gradient in the relation between $U_L$ and $(T_p - T_a)$. Knowing this, an efficiency curve can be given for both collectors based on experimental and numerical derived results.

For the EBS Sunstrip collector the $(a_1, T_c)$-value amounts approximately $0.087$. However the heat removal factor $F_R$ for this type of absorber results in an important reduction of $\eta_0$. This is due to the collector efficiency factor $F'$. For the EBS Sunstrip collector $F' = 0.88$, according to Duffie and Beckman (1980). This means that for the EBS Sunstrip collector applied with a slit honeycomb a maximum efficiency might be expected of approximately $0.77 \eta_{HC}$.

The heat loss coefficient in the enclosure between the absorber and the cover has been calculated with the coupled conduction-radiation programme for this situation (Hostaphan BN 58 as slit honeycomb material), and amounts 2.22 W/m²K. In combination with the heat loss coefficient from the cover ($\approx 25$ W/m²K) and the insulation heat losses ($\approx 0.4$ W/m²K) the total heat loss factor of the EBS-HC collector is approximately 2.4 W/m²K. This is the minimum value of the total heat loss factor, it assumes complete suppression of natural convection and represents therefore $U_{L0}$. When natural convection occurs the temperature dependency of $U_L$ on the temperature difference should be found also in the efficiency curve. Therefore $a_2$ is expected to be $\approx 0.02$ W/m²K.

For the Cusorb collector the heat removal factor is approximately 1, as explained before. This should result in a $\eta_0$ -value of $\approx 0.08$. For the heat loss coefficients the calculation is similar to that of the EBS-HC collector, resulting in $a_1 = U_{L0} = 2.4$ W/m²K and $a_2 = 0.02$ W/m²K also. The expected efficiency curve for the Cusorb collector is

$$\eta = 0.88 - 2.4 T^\# - 0.02 (E_s) T^2$$ (5.4)

5.3 The prototype characteristics.

Both collectors went through an indoor solar simulator test, according to CEC recommendations, at the Solar Pilot Test Facility of the Institute of Applied Physics, TNO-TH.

In total four tests have been performed, one with and one without a slit honeycomb for each collector. The experimental results have been given in

![Figure 5.4: Experimental efficiency results.](image)

(a) EBS Sunstrip, (b) Cusorb collector.

Both HC-collectors have a slightly less $\eta_0$ - value in comparison with the NHC $\eta_0$ - values, but significantly larger efficiency at larger $T^\#$ values.

The efficiency curves obtained from the solar simulator tests should be corrected for the amount of infrared radiation of the artificial sun and also for the reduced radiative heat loss of the cover. In the solar simulator the collectors are directed to the artificial sun, which has temperatures locally of $\approx 1000^\circ$C. Therefore the radiative heat loss from the cover is smaller than would be obtained outdoors. Applying these standard corrections for the PTF artificial sun on our results gives a corrected $\eta_0$ - value for each collector.
lector. For the EBS Sunstrip collector the $\eta_0$ values become 0.751 and 0.763 for the HC and NHC collector. For the Cusorb collector the corrected $\eta_0$ values amount 0.805 and 0.814, respectively.

Evaluation of these values give for the solar transmissivity of the two slit honeycombs 0.984 and 0.989 for the EBS and Cusorb collector, respectively. The derived solar transmissivity is extremely large. This has been expected since the efficiency measurements have been done for perpendicular insolation. A comparison between the obtained and predicted efficiency coefficients have been given in Table 5.1 and Fig. 5.5.

Good agreement has been obtained between the predicted and measured efficiency curves for the EBS-HC collector, especially for $\eta_0$. The experimentally derived $U_0$ value ($a_1$) is larger than the theoretically evaluated $U_0$ value. This difference can be explained since we used three additional iron bars over the height of the honeycomb frame to stretch the slits. The heat leak caused by the bars increases $U_0$. The coefficient $a_2$ is in agreement with our experimental value, especially if one bears in mind that the experimental derived coefficients are extremely sensitive for the curve fitting.

For the Cusorb collector the agreement between predicted and measured values is worse. The large difference found in the $\eta_0$ values suggest a $F_R$ value of $\approx 0.92$. This is due to the contact between the Cusorb foil and the stainless steel absorber. During the tests the edges of the Cusorb strips came off the absorber at several places. It is evident that this influenced the thermal contact between the spectral selective layer and the stainless steel plate. The difference in the $U_0$ values must be due to heat leaks from the absorber to the surroundings. These heat leaks can be caused for instance by the frame, the in- and outlet pipes of the absorber etc. The coefficient $a_2$, which indicates the temperature dependence of the total heat loss factor, agrees well with the expected value. In Fig. 5.5 the efficiency curves have been compared with the predicted curves.

The effect of the total suppression of natural convection up to a critical temperature difference can not be seen in the correlation between $\eta$ and $T^*$. However a more intensive testing of the collector at a larger number of $T^*$ values should answer whether a critical $T^*$ value is present or not. For both collectors the first four datapoints (see Fig. 5.4) could be fitted on a straight line (regression coefficients of -0.999 and -1.000 for the EBS-HC and Cusorb-HC respectively). This indicates a constant $U_0$ and therefore complete suppression of natural convection.

In conclusion it may be stated that application of a slit honeycomb shall increase the efficiency of a spectral selective flat plate collector significantly for $T^*$ values larger than 8.08 Km²/W. The efficiency increase shall be approximately 25% at $T^* = 0.08$ Km²/W up to 58% or more for $T^* = 0.12$ Km²/W.

5.4 Comparison with high efficiency solar collectors.

To evaluate the results obtained with the flat plate slit CSD solar collectors the results should be compared with other high efficiency solar collectors. We compared the Cusorb/HC collector with three collectors, which participated in two international Round Robin tests for solar collectors.
One has been initiated by the International Energy Agency (abbrev. IEA) and one by the Commission of European Communities (CEC). The Institute of Applied Physics/TNO-TH participates in both. We used for our comparison the results obtained at the Pilot Test Facility of the Institute of Applied Physics. So the results given below, have been derived with the same experimental set-up.

The three collectors selected for the comparison have been:

(a) the DRU-HI/F collector: a spectral selective flat plate solar collector. This collector has been the subject of the CEC-4 Round Robin test and is in principle equal to the collector as used by us for the total heat loss experiments described in chapter 4.6. Main features of the collector are \( a_p = 0.9 \); \( e_p = 0.15 \), aperture area = 1.18 m².

(b) the Philips UTR 151/120 collector used for CEC-5. This collector is an evacuated tubular collector. It uses a heat pipe system to transport the absorbed energy to a waterloop circuit. Heat losses of the collector are very small due to the vacuum (no natural convection heat losses) and the low emissivity (\( e_p \approx 0.85 \), Cobalt oxide) of the spectral selective layer. An array of these vacuum tube collectors has been tested with a white background to reflect the solar radiation to the absorbers. The working fluid in the heat pipe has been isobutane, which has a critical temperature of 135°C. The total collector system consisted of 12 tubes resulting in a total absorber area of 1.15 m².

(c) The Sanyo vacuum tube collector used for IEA-3. This spectral selective tubular collector (\( a_p \approx 0.9, \ e_p \approx 0.12 \)) has been tested with water as heat transporting fluid and by using a white background for the reflection of solar radiation. The total aperture area of the collector has been 2.44 m².

The results of the efficiency measurements have been given in fig. 5.6. The obtained efficiency relations have been given in table 5.2.

Comparison of the Cusorb/HC results with the other solar collectors show that the Cusorb/HC collector has much higher efficiency than the conventional flat plate DRU-HI/F collector. The maximum efficiency \( \eta \) is considerably larger due to the larger absorptivity of the spectral selective layer and the use of tempered low-iron glass as cover plate. In addition the total heat loss factor of the Cusorb/HC collector has been reduced significantly by the implementation of the slit honeycomb.

In comparison with the Philips vacuum tube, the total heat loss factor of the Cusorb/HC collector is larger. However due to the larger \( \eta \) — value the performance of the Cusorb/HC collector is superior to the vacuum tube for \( T^* \) values up to \( T^* = 8.10 \text{ K/m/W} \). For larger \( T^* \) values the heat pipe collector is superior.

Moreover Philips developed a new heat pipe vacuum tube collector (VTR 361 water-filled and VTR 261 with a working fluid of neopentane). The collector tube of the VTR 361 is about 30 % longer than the VTR 151/120. This collector has a slightly higher maximum efficiency and therefore better efficiency than the VTR 151/120. A complete Round Robin test has not been performed yet.

The Sanyo vacuum tube collector has a smaller total heat loss factor than the Cusorb/HC collector and a relatively high \( \eta \) value. The point of intersection of the efficiency curves lies at \( T^* = 8.85 \text{ K/m/W} \) and above this value the Sanyo collector is superior to all collectors considered.

It is clear that the effectiveness of a slit honeycomb collector (or in gen-
eral a CSD collector) is dependent on the working temperature of the collector. For instance if heat storage demands a minimum working temperature of 88°C, the working point of the collector for $T_a = 20°$ C and $<E_\text{s}> = 758 \text{ W/m}^2\text{K}$ amounts approximately $T_s = 0.88 \text{ K/mW}$. At this range the performance of the Cusorb/HC is competitive with the two vacuum tube collectors described above and highly superior to conventional flat plate collectors.

Which collector is the most economic to use depends on the application of the solar system (hot water supply, heating or cooling of buildings), the price of the solar system etc. This can only be determined after a thorough cost analysis considering all these aspects. However since the evacuated tubular collectors are considerably more expensive than conventional flat plate collectors (factor 3.3) the application of a relatively cheap fabricated convection suppressing device might be very promising. For the CSIRO project in Australia (Symons, 1984) an upslope slit CSD collector has been developed for which the retail costs have been assessed to be 215,- Australian dollar per square metre. A costs analysis showed that a slit CSD collector was the most cost effectiveness collector for temperatures between 88° and 120° C. The Dutch firm EBS aims at a cost price of approximately Dfl. 580,- per square metre for a slit CSD high performance collector. Such a collector is now under development.

6.1 Evaluation numerical and experimental research.

Considering the two and three dimensional non-coupled natural convection calculations several conclusions can be made. Our three dimensional natural convection model shows clearly that a two dimensional convection model can not be used for the calculation of natural convection in inclined enclosures of aspect ratios less than one ($A < 1$). A three dimensional model is necessary for inclined situations since the natural convection flow changes its main orientation from a base flow in the xy-plane of the enclosure for $\phi = 90$ degrees, to a combined base and longitudinal flow in the yz-plane of the enclosure. Application of a two dimensional instead of a three dimensional model underestimates the natural convective heat transfer and results in too large critical Rayleigh numbers for the onset of natural convection.

Furthermore it is shown that natural convective heat transfer decreases with a decrease of the aspect ratio $A_x$ for $A_x < 1$. In addition a decrease of aspect ratio $A_z$ results only for $A_z < 1$ in a significant decrease of natural convective heat transfer. For CSD applications in solar collectors slits of $A_z \approx 0.1$ or cells with $A_z = A_x \approx 0.2$ will be sufficient for suppressing natural convective heat transfer for Rayleigh numbers up to $\approx 10^5$.

The coupled conduction-radiation calculations show that application of a slit CSD, in the spacing between the absorber and the cover of a solar collector, results not only in suppression of natural convection, but influences the thermal radiative heat transfer and wall conductive heat transfer as well. The coupling between the three heat transfer modes is so significant that an independent mode analysis can not be used for predicting the total heat transfer in the slit CSD. The effects of the coupled heat transfer mechanisms on the total heat transfer are dependent on the collector type and the emissivity and (effective) reflectivity of the CSD material. For non-
spectral selective collectors (SB) the slit CSD acts as a radiation shield as well, and the total heat transfer decreases with increasing CSD emissivity ($\varepsilon_{hc}$), for the case of specular reflection of thermal radiation by the CSD material. For diffusely reflecting material the total heat transfer is almost independent of the emissivity of the slit CSD walls, but considerably lower than predicted according to the independent mode analysis (IMA), where radiative heat transfer has been calculated according to the infinite parallel plates model.

For a spectral selective collector (SB) the application of a slit CSD (or in general a CSD) results in the suppression of natural convective heat transfer, provided the CSD has been designed properly. However the total heat transfer is considerably larger than expected according to the IMA. Inspection of the heat balance shows that at the hot wall, the air and wall conduction occurs also for the S3-collector, although an equal heat transfer occurs also for the SS-collector, although an equal heat transfer distribution is obtained at the hot and cold wall. For diffusely reflecting CSD material the effects are less pronounced for these three cases. Furthermore it is noted that the smallest effects occur for very small CSD emissivity ($\varepsilon_{hc} < 0.1$).

A considerable decrease of coupled heat transfer can be obtained by the introduction of a gap between the absorber and the CSD. This way the interaction between the heat transfer mechanisms becomes weaker and this results in a strong reduction of total heat transfer. Good agreement has been found between experimental and numerical work concerning the gap width. The developed programme for calculating the coupled conductive-radiative heat transfer predicted the experimentally derived total heat transfer coefficients within the experimental accuracy and is therefore a reliable tool to predict the minimum value of the heat loss factor of a slit CSD solar collector.

The coupled convection-radiation calculations have been performed for two dimensional flow only. This is not correct according to an earlier conclusion. However we considered only the effect of the coupled heat transfer on the critical Rayleigh number for the onset of convection. The results showed no important influence on the onset of natural convective heat transfer for the three cases considered. Since no important influence has been found, it might be expected that the influence on the onset of natural convection for a three dimensional situation will be small too. Therefore the critical Rayleigh numbers obtained with the three dimensional programme can be considered to be a good approximation for actual CSD situations.

The experimental results for the cross slope slit structures showed that the total heat transfer from the absorber to the cover for a particular situation is governed primarily by three parameters (1) the temperature difference $T_h - T_c$, and (2) the plate distance $D$ (usually combined in the Rayleigh number) and (3) the gap width $s$. For cross slope slit structures the total heat transfer will be equal to the total heat transfer in stratified air conditions up to a critical $(\Delta T, D)$ combination, or the critical Rayleigh number for this configuration. Increasing the gap width $s$ decreases the total heat transfer for the stratified air situation. However too large gap widths ($s >> D$) result in a decrease of the critical Rayleigh number also. Once convection has been initiated the total heat transfer can be approximated by a linear function of the temperature difference for $\Delta T \approx \Delta T_c$.

We found experimentally a critical Rayleigh number of $2 \times 10^5$ for the $A_x = 0.1$ slit structure, while we derived numerically $R_{y_0} \approx 5.4 \times 10^5$. The difference must be due to experimental circumstances in particular the quality of the slit structures used. For example we never succeeded in creating a slit structure that perfectly fitted between the hot and cold plate in the heat transfer apparatus. The smallest gap that we achieved in our experiments was still a gap of $4 \%$. Such small gaps can lead to circulating natural convection flows to the sides of the collector, as we have seen qualitatively with our flow visualization experiments. This effect might increase the total heat transfer.

For application purposes the optimum plate distance/gap width combination for cross slope slit structures has been found for $D \approx 0.08 m$, $s = 7 \%$. Once natural convection has been initiated in a cross slope slit structure, the flow development is stronger than that for the upslope slit structure. This results in a smaller stagnation temperature for the solar collector. The gap width $s$ and the smaller stagnation temperature for cross slope slit structures, are large advantages concerning the choice of the CSD material and the allowed tolerance for the manufacturing process.

6.2 Final conclusions.

Our experimental work showed that the application of slit CSD in flat
plate solar collectors reduces the total heat loss factor of a solar collector significantly. Moreover the results of the small solar simulator experiments showed that the solar transmittance of a slit CSD made of a high transparent material (FEP Teflon or Hostaphan BN 50) exceeds 0.95, even for non-perpendicular insolation angles. The experimental work, considering the flat plate slit CSD solar collectors, showed that the application of a slit CSD leads to a considerable increase in collector performance at the high temperature range. Especially for $T_s \geq 800$ K/m/W the increase amounts up to 50 %, taken relatively to the NHC collector efficiency. Moreover with the Cusorb slit CSD collector we reached efficiency values which are competitive with vacuum collectors up to $T_s \approx 1200$ K/m/W.

As a result of our experimental work the following recommendations can be given:

1. The solar absorptivity of the absorber should be as high as possible. It should be noted that for a spectral selective CSD collector it might be more rewarding to increase the solar absorptivity than to minimize the emissivity of the spectral selective layer.

2. The solar transmissivity of the cover should be as large as possible, as for example tempered low-iron glass with a transmittance of $\approx 90$.

3. The heat removal factor $F_R$ should be near 1. This means that the well known fin absorbers are less suitable, due to their lower fin efficiency. These three recommendations result in a large $U_L$ value of the collector. Application of the following recommendations shall minimize the total heat loss factor $U_L$ of the collector.

4. A cross slope slit CSD, or for large temperature differences an upslope slit CSD, of $A_x \approx 0.1$ with dimensions $D \approx 0.05$ m and $s = 7 %$ (4 % or less for upslope) should be used to minimize the heat losses from the absorber to the cover of the solar collector.

5. To minimize the back and side heat losses of the collector an insulation layer of at least 0.10 m thickness at the back, and 0.05 m thickness at the sides of the collector should be applied. This is approximately twice as thick as used normally in conventional solar collectors. The side insulation is recommended since the distance between the absorber and the cover of the collector has been increased.

6. The slit CSD material should have a large solar transmittance, so that the overall CSD transmissivity remains high also for non-perpendicular insolation. The thickness should be smaller than $50 \times 10^{-5}$ m and the material should have outstandingly high temperature (up to $180^\circ$ C) and ultra violet durability. Considering these demands FEP Teflon is an excellent candidate for CSD applications.
APPENDICES
Appendix A: The three dimensional solution procedure.

The algebraic equation which has to be solved for the different variables at each point reads

\[(a_P^6 - b_P^6) \xi_P = \sum_j a_j^6 \xi_j + c_j^6 \quad j = N, S, E, W, U, D \quad (A1)\]

with \(b_P^6\) and \(c_P^6\) components of the linearized source term of eq. (3.46). The equations have been solved line-by-line using TDMA. Therefore the equations have been written as follows:

\[(a_P^6 - b_P^6) \xi_P = a_N^6 \xi_N + a_S^6 \xi_S + c_1^6 \quad (A2)\]

with

\[c_1^6 = a_N^6 \xi_N + a_E^6 \xi_E + a_W^6 \xi_W + a_D^6 \xi_D + c_1^6 \quad (A3)\]

or

\[x_i = a_{i+1}^i x_{i+1} + b_{i+1}^i x_{i+1-1} + c_i^6 \quad (A4)\]

where \(x(i), x(i+1)\) and \(x(i-1)\) denote the value of the variable at grid point \(i, i+1\) and \(i-1\), respectively.

Combining eq. (A4) for the \(i\)-th and \((i+1)\)-th grid point results in the following recurrent formula

\[x_i = a_i^i x_{i+1} + b_i^i \quad (A5)\]

with

\[a_i = a_N^i + a_S^i + a_E^i + a_W^i + a_D^i + c_i^6 \quad \text{and} \quad b_i = a_P^i \]

The components of the coefficients in the recurrent formula are:

\[a_i = a_N^i + a_S^i + a_E^i + a_W^i + a_D^i + c_i^6 \quad (A6)\]

\[b_i = a_P^i \quad (A7)\]

All defined at grid point \(i\).
Appendix B: Calculation procedure for coupled heat transfer.

Two basically equal programmes have been developed for the coupled heat transfer calculation. For stagnant air conditions programme CDR calculates the heat transfer by air and wall conduction in interaction with radiation. Programme CYR calculates the coupled convection-radiation heat transfer. The calculation procedure used for both programmes is schematically shown in fig. B.1.

![Diagram of calculation procedure](image)

Figure B.1: Calculation procedure for coupled calculations.

Different blocks can be distinguished in the diagram:

- **Block I**: Input
  - Sets all parameters describing the problem: $T_h$, $T_c$, $D$, $h_w$, $A_k$, $\eta_w$, $\xi_h$, $\xi_c$, $\xi_{hc}$, $\rho'$

- **Block II**: Initial operations
  - Determines grids, views and absorption factors, inverse side wall matrix; gives $u$, $v$, $p$, $\theta$, $\theta_w$ start values

- **Block III**: Iteration loop
  - **IIIa**: Natural convective heat transfer is determined by using the two dimensional TEACH programme. This iterative procedure is terminated when the natural convective heat transfer is constant within the desired accuracy.
  - **IIIb**: Radiative heat fluxes on the walls are determined by using the absorption factor method.
  - **IIIc**: The side wall temperature distribution $\theta_w$ is determined according to the procedure described in chapter 3.4.1. The temperature distribution obtained is set as a temperature boundary condition for the convection (or conduction) and radiation calculation.
  - **IIId**: The total heat transfer coefficients at the hot and cold wall are determined and depending on the difference with the previous iteration the iterative procedure continues.

- **Block IV**: Output
  - When a solution has been reached "output" prints the velocity, pressure and temperature fields and the calculated radiative, convective and conductive heat fluxes.
Appendix C: Determination of $U_{\text{ins}}$ and plate emissivities.

Determination of the total heat transfer between the isothermal plates is possible when the heat losses to the surroundings have been determined (given by $U_{\text{ins}}$).

Moreover to determine the convective heat transfer in NHC situations the radiative heat transfer between the plates has to be known. Therefore we need to know the emissivity values of the hot and cold plate. In addition knowledge of the emissivity values of the plates is indispensable for the comparison between numerical and experimental results.

According to chapter 4.4.2 the insulation value $U_{\text{ins}}$ of the heat transfer apparatus is given by

$$U_{\text{ins}} = \frac{E - h A (T_h - T_c)}{T_h - T_a}$$  \hspace{1cm} (C.1)

To measure the insulation value $U_{\text{ins}}$ we heated the isothermal plates up to a value $T_h > T_c > T_a$ with the heat transfer apparatus placed in the stratified air situation ($\phi = 180$ degrees). The second term in the numerator on the right-hand side of eq. (C.1) becomes very small in comparison with the first term and the heating power $E$ is approximately equal to the heat losses through the insulation. This way the insulation value $U_{\text{ins}}$ has been determined with an accuracy of $\pm 5\%$.

During our measurements we measured $U_{\text{ins}}$ each time when a change in the experimental set-up was established. The value of $U_{\text{ins}}$ has been given as a function of the plate distance $D$ for several temperature differences $T_h - T_a$ in fig. C.1. During our measurements the value of $U_{\text{ins}}$ was $0.092 + 0.28D$ W/K.

Our measurements have been performed with different hot and cold wall emissivities. Three situations have been distinguished:

(1): two aluminum plates without a special coating (representing SS-case).

(2): two aluminum plates painted black (BB).

(3): the cold plate painted black and the hot plate without special coating (SB).

For each of these situations the wall emissivities have been determined. The

**Figure C.1: The dependence of $U_{\text{ins}}$ on the plate distance.**

heat transfer apparatus has been placed in the stratified air situation (hot plate above cold plate) without a slit structure (NHC). The plate distance between the hot and cold plate has been made small $(D \approx 0.02 \text{ m})$ and a small temperature difference $(\Delta T \approx 20 \text{ K})$ has been applied with $T_h = 45^\circ \text{C}$. The stationary heating power in this situation has been given by:

$$E = Q_{\text{cond}} + Q_{\text{rad}} + Q_{\text{ins}}$$  \hspace{1cm} (C.2)

with

$$Q_{\text{cond}} = \lambda B A_p (T_h - T_c) / D$$

$$Q_{\text{rad}} = \frac{d A_p (T_h^4 - T_c^4)}{(e_h)^{-1} + (e_c)^{-1} - 1}$$

$$Q_{\text{ins}} = U_{\text{ins}} (T_h - T_a)$$

With the known values of $U_{\text{ins}}$, the constant of Stefan-Boltzmann ($\sigma = 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4$), $A_p = 0.291 \text{ m}^2$ and $\lambda = 0.0274 \text{ W/K}$ the plate emissivities have been determined.
For the three cases considered the derived emissivity values were

1: SS-case $\varepsilon_h = \varepsilon_c = 0.20 \pm 0.05$
2: BB-case $\varepsilon_h = \varepsilon_c = 0.95 \pm 0.05$
3: SB-case $\varepsilon_h = 0.20 \pm 0.05; \varepsilon_c = 0.95 \pm 0.05$

Appendix D: Calibration of the temperature difference meter.

The maximum useful heat gain of the fluid flow through the collector is given by $A_E$. For the collector model area of 0.86 m$^2$ and a mean solar insolation of 968 W/m$^2$ the useful heat gain amounts approximately 50 Watt. For an average fluid mass flow of $1 \times 10^{-2}$ kg/s the expected temperature rise of the fluid travelling through the collector is approximately 1 K. It is difficult to measure such small temperature differences accurately. We used a so-called temperature difference meter (TDM), which has been specially designed for our application by the Institute of Applied Physics (TPD/TNO-TH, Delft). The TDM consists of a thermopile of $\approx 1000$ thermocouples, which has been placed between two copper channels. To establish an uniform temperature of the fluid the flow channels have been placed perpendicular on the flow direction (see fig. D.1).

We calibrated the TDM in combination with the flowmeter in the experimental set-up. We replaced the collector model by an electrical heater, which heats the fluid with a known heating power. By placing the TDM and the electrical heater in a good thermal insulated box (0.05 m roofmate insulation, $\lambda = 0.027$ W/mK) the heat losses to the surroundings have been minimized. An estimated value of the heat loss at a temperature difference of 48 K is 0.2 W. This amount is less than 0.5 % of the heat transported by the fluid. The heat transfer between the channels of the TDM amounts approximately 0.2 W also.

The calibration of the TDM at different temperatures and different flows resulted in a relation between $\varphi_M$ and $\varphi_{RO}$ (where $\varphi_{RO}$ is the "read off" value of the flowmeter (sc)).
\[ a_0 \langle T_f \rangle, \phi \rangle = a_{u0} + a_{u1} \phi + a_{u2} \phi^2 \]  
\[ \text{(D.1)} \]

with:
\[ a_{u0} = b_{u0} + b_{u1} \langle T_f \rangle + b_{u2} \langle T_f^2 \rangle \]
\[ a_{u1} = c_{u0} + c_{u1} \langle T_f \rangle + c_{u2} \langle T_f^2 \rangle \]
\[ a_{u2} = d_{u0} + d_{u1} \langle T_f \rangle + d_{u2} \langle T_f^2 \rangle \]

and the calibration constants found are:

\[ b_{u0} = -2.6849 \times 10^2 \text{ W/°C} \]
\[ b_{u1} = 1.1072 \times 10^1 \text{ W/°C}^2 \]
\[ b_{u2} = -8.2975 \times 10^{-2} \text{ W/°C}^2 \]
\[ c_{u0} = 1.4889 \times 10^2 \text{ W/°C} \]
\[ c_{u1} = -3.2038 \text{ W/°C}^2 \]
\[ c_{u2} = 3.6958 \times 10^{-2} \text{ W/°C}^2 \]
\[ d_{u0} = -1.2661 \text{ W/°C} \]
\[ d_{u1} = 1.0781 \times 10^{-1} \text{ W/°C}^2 \]
\[ d_{u2} = -1.1499 \times 10^{-3} \text{ W/°C}^2 \]

The read off inaccuracy of the flowmeter is 0.5 %, the inaccuracy in the signal of the TDM and the electrical heating power is 0.5 % and 0.1 % respectively. The total relative error in measuring \( a_u \) therefore amounts 1.6 %. Using this calibration relation for the prediction of the useful heat gain by the fluid as produced by the electrical heater, resulted in values of \( Q_u \) within 2 % of the measured electrical heating power. The overall inaccuracy for predicting the useful heat gain is therefore 2 %. Additional calibration of the flowmeter at the temperature range between 20 °C and 50 °C resulted in a relation between the mass flow \( \phi_m \) and the read off value of the flowmeter.

\[ \phi_m = a_{m0} + a_{m1} \phi + a_{m2} \phi^2 \]  
\[ \text{(D.2)} \]

Here \( a_m \) is the mass calibration constant for the flowmeter at a certain average fluid temperature and flowmeter read off. As for \( a_u \) the calibration resulted in a relation between \( a_m \) and the calibration results \( \phi_0 \) with \( \langle T_f \rangle \) as parameter. The relation is given by

\[ a_m \langle T_f \rangle, \phi_0 \rangle = b_{m0} + b_{m1} \langle T_f \rangle + b_{m2} \langle T_f^2 \rangle \]  
\[ \text{(D.3)} \]

with:
\[ b_{m0} = b_{m0} + b_{m1} \langle T_f \rangle + b_{m2} \langle T_f^2 \rangle \]
\[ a_{m1} = c_{m0} + c_{m1} \langle T_f \rangle + c_{m2} \langle T_f^2 \rangle \]
\[ a_{m2} = d_{m0} + d_{m1} \langle T_f \rangle + d_{m2} \langle T_f^2 \rangle \]

and the calibration constants found are:

\[ b_{m0} = 1.714 \times 10^{-1} \text{ kg/s°C} \]
\[ b_{m1} = 5.456 \times 10^{-3} \text{ kg/s°C}^2 \]
\[ b_{m2} = -3.905 \times 10^{-5} \text{ kg/s°C}^2 \]
\[ c_{m0} = 1.929 \times 10^{-2} \text{ kg/s°C}^2 \]
\[ c_{m1} = -1.592 \times 10^{-4} \text{ kg/s°C}^2 \]
\[ c_{m2} = 1.450 \times 10^{-7} \text{ kg/s°C}^2 \]
\[ d_{m0} = -4.525 \times 10^{-4} \text{ kg/s°C}^2 \]
\[ d_{m1} = 7.799 \times 10^{-6} \text{ kg/s°C}^2 \]
\[ d_{m2} = -2.345 \times 10^{-8} \text{ kg/s°C}^2 \]

The calibration results have been given in fig. 4.32, where \( a_m \) has been given as function of the mean fluid temperature for different flow rates. The inaccuracy for the calibration of the flowmeter is 2 %. The sensitivity of the TDM \( (CT_\phi) \) has been obtained by combining the calibration results of \( a_m \) and \( a_u \). According the definitions of \( a_m \) and \( a_u \) the
Sensitivity can be calculated by:

\[ S_T = \frac{\alpha_u}{\alpha_m \cdot \phi_{ro} \cdot C_p} \quad (0.4) \]

The sensitivity of the TDM has been found constant for mass flow \( \phi_m > 7 \cdot 10^{-3} \) kg/s for all fluid temperatures considered. The average sensitivity determined for this parameter range is

\[ (S_T) = 52.6 \, K/U \quad (0.5) \]

At lower flow rates the sensitivity increased. The inaccuracy of \( (S_T) \) is the sum of the errors in \( \alpha_m, \alpha_u, \phi_{ro} \) and \( C_p \) and amounts 5 %. At all temperature levels the obtained constant value of the sensitivity was within this accuracy.

REFERENCES


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Symons,J.G; Peck,M.K; 1983, "An overview of the CSIRO project on advanced flat plate solar collectors", presented at ISES Solar World Congress, Perth, Australia, August 1983

Tabor, H; 1969, "Cellular insulation (Honeycombs)", Solar Energy, 12, pp 549-552.

**LIST OF SYMBOLS**

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>SI Unit</th>
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<tbody>
<tr>
<td>(a_g)</td>
<td>thermal diffusivity</td>
<td>((m^2/s))</td>
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<tr>
<td>(A)</td>
<td>area</td>
<td>((m^2))</td>
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<td>(A_h)</td>
<td>aspect ratio (h/D)</td>
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</tr>
<tr>
<td>(A_L)</td>
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<td>(C_p)</td>
<td>specific heat</td>
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<td>(d)</td>
<td>foil width</td>
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<tr>
<td>(D)</td>
<td>distance between hot and cold wall</td>
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<tr>
<td>(E)</td>
<td>dissipated energy</td>
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<td>(g)</td>
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<tr>
<td>(G_{1,j})</td>
<td>absorption factor</td>
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<tr>
<td>(h)</td>
<td>distance between two slits (N-S direction)</td>
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<td>(h_t)</td>
<td>total heat transfer coefficient between hot/absorber plate and cold/cover plate</td>
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<tr>
<td>(h_w)</td>
<td>wall thickness</td>
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<td>(J)</td>
<td>dimensionless flux in finite difference scheme</td>
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<tr>
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<td>(Pr)</td>
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<td>(q')</td>
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<td>(Ra)</td>
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<td>critical Rayleigh number for the onset of convective heat transfer</td>
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<td>source</td>
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<td>(S_T)</td>
<td>sensitivity temperature difference meter</td>
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<tr>
<td>(T)</td>
<td>temperature</td>
<td>((K))</td>
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<tr>
<td>(T^*)</td>
<td>modified temperature parameter</td>
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<td>(v)</td>
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<tr>
<td>(v')</td>
<td>velocity vector</td>
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<tr>
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<tr>
<td>(y)</td>
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<tr>
<td>(z)</td>
<td>dimensionless z-coordinate</td>
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**Greek Symbols**

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<th>Symbol</th>
<th>Description</th>
<th>SI Unit</th>
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<tr>
<td>(\alpha_p)</td>
<td>absorptivity absorber plate</td>
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</tr>
<tr>
<td>(\alpha_{x})</td>
<td>non-linearity parameter x-direction</td>
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<tr>
<td>(\alpha_{y})</td>
<td>non-linearity parameter y-direction</td>
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<tr>
<td>(\alpha_{z})</td>
<td>non-linearity parameter z-direction</td>
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<td>(\beta)</td>
<td>coefficient of thermal expansion</td>
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<tr>
<td>(\gamma)</td>
<td>under-relaxation factor</td>
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<td>(\epsilon)</td>
<td>emissivity for thermal radiation</td>
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<tr>
<td>(\eta)</td>
<td>dynamic viscosity/efficiency</td>
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<td>(\theta)</td>
<td>dimensionless temperature</td>
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<tr>
<td>(\theta_s)</td>
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<td>(\lambda)</td>
<td>thermal conductivity</td>
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<td>(\nu)</td>
<td>kinematic viscosity</td>
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<tr>
<td>(\rho)</td>
<td>density</td>
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<td>(\rho_d)</td>
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<tr>
<td>(\rho_s)</td>
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<td>(\rho_t)</td>
<td>total reflectivity</td>
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SUMMARY

The major heat loss of spectral selective flat plate solar collectors is due to natural convective heat transfer from the absorber to the cover plate of the solar collector. If one could prevent the development of natural convection of air in the enclosure between the absorber and the cover plate for collector operating conditions, the efficiency of the collector can be improved significantly. Heat transfer by the air to the cover is then possible by conduction only. A method to achieve this goal is to insert convection suppressing devices (CSD) in the enclosure. This way the enclosure is divided into a large number of individual cells. In such a cell the development of natural convection is delayed by the increased amount of viscous forces. A correct design of a CSD should suppress the natural convection at collector working conditions. This investigation concerns the application of rectangular slit and cell structures as CSD in flat plate solar collectors.

The theoretical, numerical part of the investigation is described in chapter 3. The first part of this chapter considers the laminar and stationary natural convection flow in two and three dimensional rectangular enclosures. The governing equations have been solved according to a finite difference method. The results give an indication of the dimensions which are needed for an effective slit or cell CSD application. From the results it appeared that slit structures of (at least) aspect ratio 6.1, or cell structures with aspect ratios 0.2, are necessary to suppress natural convection sufficiently for Rayleigh values up to $10^6$.

The second part of chapter 3 includes the radiative and wall conductive heat transfer. We directed us at slit structures of aspect ratio 0.1 and calculated the coupled heat transfer by natural convection, radiation and wall conduction in a slit CSD. We considered non-spectral selective (BB), spectral selective (SB) and collectors with a spectral selective absorber and a thermal radiation reflective cover as well (SS). The results show that the application of a slit CSD in a BB-collector leads to both suppression of natural convection as radiative heat transfer. For SB- and SS-collectors the heat transfer is reduced less than expected by calculations based on a non-coupled heat transfer mode analysis. The total heat transfer in the slit for...
the three collector types is very dependent on the emissivity and (effective) reflectivity coefficient of the CSD material in the infrared wavelength region.

Chapter 4 describes our experiments performed for slit structures. The flow visualization experiments agree and confirm qualitatively the numerical two and three dimensional flow calculations.

Caloric measurements of the natural convective and total heat transfer show that initiation of natural convection in cross slope slit structures of $A_e \approx 0.1$ can be delayed up to $Ra \approx 2 \times 10^5$. Moreover the natural convective heat transfer results are in good agreement with literature. The total heat transfer measurements for stratified air conditions match the coupled heat transfer calculations. In addition it has been shown that a gap of approximately 7% of the plate distance, between the absorber and the slit CSD, effectively decreases the coupling between the heat transfer modes, without affecting the convection suppressing quality of the slit CSD. As a result of the partial decoupling the total heat transfer decreases.

Application of slit CSD's in commercial solar collectors showed that the total heat loss for a SB-collector can be reduced by $\approx 30\%$. Experiments performed with a small solar simulator showed that the maximum efficiency $\eta_b$ of the slit CSD solar collector is at least 95% of the $\eta_b$ value of the conventional collector even for non-perpendicular insolation.

In chapter 5 we describe and discuss the actual implementation of a properly designed slit CSD in two solar collectors. One is a commercially available spectral selective collector, which after application of the slit CSD has an improved performance of $\approx 30\%$ at high temperatures, relative to its original performance. The second collector is a by us designed and built flat plate slit CSD solar collector showing that the performance of a slit CSD collector can compete with vacuum collectors. Despite of adhesive problems with the spectral selective coating, the efficiency results were much better than other flat plate collectors (inclusive the commercial collector with the slit CSD application) and competitive with vacuum collectors up to $T_a \approx 612 \text{Km}^2$.W.

As general conclusion, it may be stated that the use of a slit CSD improves the efficiency of flat plate solar collectors significantly, especially at the high temperature range. Moreover further optimizations, as, for instance, $\eta_b$ value and improved insulation at the back and sides of the collector leads to a thermal performance which is competitive with vacuum collectors up to large temperature differences.

---

**SAMENVATTING**

Het grootste warmteverlies van spectraal selectieve vlakke plaat zonne-collectoren wordt veroorzaakt door de warmteoverdracht door natuurlijke convectie van de absorber naar de afdekruit in de spouw van de collector. Het collectorrendement zal aanzienlijk verbeteren als men de ontwikkeling van de natuurlijke convectiestroming in de spouw tijdens collector-werkcondities kan voorkomen. Warmteoverdracht door de lucht van de absorber naar de afdekruit is dan slechts mogelijk door geleiding. Het inbrengen van convectie-onderdrukkende structuren in de spouw is een methode om dit doel te bereiken. Hierbij wordt de spouw in een groot aantal individuele cellen verdeeld. Door de toename van de wrijvingskrachten per cel wordt de stromingsontwikkeling onderdrukt. Een goed ontworpen structuur zal de convectie tijdens collector-werkomstandigheden nog onderdrukken. Dit onderzoek beschouwt de toepassing van recht hoekige spleet- en celstructuren als convectie-onderdrukkers in vlakke plaat zonne-collectoren.

Het theoretisch-numerieke deel van dit onderzoek is beschreven in hoofdstuk 3. Het eerste deel van dit hoofdstuk beschouwt de laminair, stationaire natuurlijke convectiestroming in twee- en drie-dimensionale recht hoekige ruimten. De problemabeschrijvende vergelijkingen zijn volgens een eindige differentie methode opgelost en deze resultaten geven een indicatie hoe groot een effectieve spleet- of celstructuur minimaal moet zijn. Uit de resultaten blijkt dat spleetstructuren met aspect ratio 8.1 en celstructuren met aspect ratio 8.2 minimaal nodig zijn om de natuurlijke convectie afdoende te onderdrukken. Het tweede deel van hoofdstuk 3 beschouwt tevens de stralings- en wandgeleidingswarmteoverdracht. Bij beperkt ons tot spleetstructuren en berekenden de gekoppelde warmteoverdracht door natuurlijke convectie, warmtestraling en wandgeleiding. Bij hebben zwarte (BB), spectraal selectieve (SB) en collectoren met een infra-ruim reflecterende laag op de afdekruit beschouwd (SS). De resultaten tonen aan dat door toepassing van een spleetstructuur in een zwarte collector zowel de natuurlijke convectie als de warmtestraling door de spleetstructuur wordt onderdrukt. Toepassing in SB- en SS-collectoren doet de totale warmteoverdracht minder sterk afnemen dan men
op grond van een ongekoppelde berekeningsmethode zou verwachten. De totale warmteoverdracht in de spleet is voor de drie typen collectoren sterk afhankelijk van de emissiecoëfficiënt en de (effectieve) reflectiecoëfficiënt van het spleetstructuur materiaal in het infra-rode spectrum.

Onze experimenten met spleetstructuren staan beschreven in hoofdstuk 4. De "flow visualization" experimenten komen kwalitatief goed overeen met en bevestigen de numerieke resultaten.

De calorische metingen van de warmteoverdracht door natuurlijke convectie en de totale warmteoverdracht laten zien dat in horizontaal gespannen spleetstructuren met aspect ratio 0.1 de natuurlijke convectiestroming kan worden onderdrukt tot $Ra \approx 2 \times 10^5$. Bovendien zijn de resultaten van de convectieve warmteoverdracht in overeenstemming met literatuurgegevens. De metingen van de totale warmteoverdracht, met het model onder 100 graden (de zgn. "stratified air case"), zijn goed in overeenstemming met onze numerieke resultaten aangaande de gekoppelde warmteoverdrachtsberekeningen. Tevens is aangetoond dat een ruimte van ongeveer 7% van de spouwbreedte, aangebracht tussen de spleetstructuur en de absorber, het koppelingseffect vermindert zonder dat de stromingsonderdrukkende werking van de spleetstructuur wordt aangetast.

Toepassingen van spleetstructuren in commerciële zonnecollectoren tonen aan dat de totale warmteoverdrachtsfactor van de collector met ongeveer 30% vermindert. Experimenten met een kleine zonnesimulator laten zien dat het maximale rendement $\eta$ van de spleetstructuur-collector zelfs voor niet loodrecht invallende zonnestraling nog minstens 95% bedraagt t.o.v. het maximale rendement van de conventionele collector.

In hoofdstuk 5 wordt de fabricage en inpassing van een spleetstructuur in twee collectoren besproken. De eerste collector, een commercieel verkrijgbare spectraal selectieve collector, gaf na toepassing van de spleetstructuur een rendementverbetering van ± 30% bij hogere temperatuursverschillen (relatief t.o.v. het rendement van de originele collector). De tweede collector is door ons ontworpen en gebouwd om aan te tonen dat het rendement van een spleetstructuur-collector kan wedijveren met vacuum-collectoren. Ondanks problemen met de bevestiging van de spectraal selectieve laag bleek het rendement aanzienlijk hoger dan van andere vlakke plaat collectoren (inclusief de eerder genoemde spleetstructuur-collector) en was concurrerend met vacuum-collectoren tot $T \approx 8.12 \text{ Km}^2/\text{W}$. Als algemene conclusie kan worden gesteld dat het gebruik van spleetstructuren (of meer algemeen convectie-onderdrukkende structuren) het rendement van vlakke plaat zonnecollectoren aanzienlijk verbetert vooral bij hoge temperatuursverschillen. Bovendien kunnen optimalisaties van bijvoorbeeld $\eta$ en verbeterde warmteisolatie aan de zij- en achterkant van de collectoren leiden tot een rendementscurve die concurrerend is met vacuum-collectoren zelfs bij hoge gebruikstemperaturen.
CURRICULUM VITAE

De schrijver van dit proefschrift is op 26 februari 1955 geboren te Hoek van Holland.


De inschrijving voor de studie technische natuurkunde aan de Technische Hogeschool te Delft volgde in september van dat jaar.


Vanaf 15 februari 1979 tot 15 mei 1984 was hij als wetenschappelijk medewerker in dienst van de Stichting FOM gestationeerd bij de groep Warmtentransport van de Technische Hogeschool te Delft. Dit proefschrift beschrijft het onderzoek dat gedurende deze periode is verricht.

Momenteel is hij werkzaam als wetenschappelijk ambtenaar bij de vakgroep Petroleumwinning en Technische Geofysica, afdeling der Mijnbouwkunde van de Technische Hogeschool te Delft.
NAWOORD

Bij het schrijven van dit nawoord realiseer ik me pas goed hoeveel personen er bij dit onderzoek betrokken zijn geweest.

Ten eerste is dat mijn promotor prof. ir. C.J. Hoogendoorn die ik bijzonder erkentelijk ben voor de manier waarop hij mij deze jaren heeft begeleid.

De studenten Els Boogh, Pim Bos, Jan Glimmerveen, Jan de Heer, Chris Kleijn, Joris Kuin, Adriaan Lankhorst, Ronald de Putter, Paul Schaareman, Henk de Vlugt, Reinier Warschauer en Fokke Zwaan hebben in het kader van hun 4e of 5e jaars onderzoek meegewerkt aan dit project.

Tijdens de utilisatiefase van dit onderzoek was Rimmert Hoekstra van onschatbare waarde. Rimmert, zonder jouw grenzeloos optimisme en (soms te) fanthemierlijke ideeën zou de realisatie van de collector binnen de tijdsduur van het project niet mogelijk zijn geweest. Ook dank ik je voor het tekenen van de figuren die dit proefschrift bevat. Het was een plezier om met jou samen te werken.

De werkplaats en de glasblazerij van de afdeling der Technische Natuurkunde dank ik voor het uitvoeren van de vele "haastklussen".

Een speciaal woord van dank ben ik verschuldigd aan de heren Baardmans en Thomassen van de firma EBS, die mij de nodige specifieke informatie en materiaal verschaften.

De medewerkers van de groep Warmteinstrumentatie van de Technisch Physische Dienst TNO-TH, in het bijzonder diegenen die betrokken zijn geweest bij het testen van de verschillende prototypen, dank ik voor hun medewerking.

Annette, Ineke en Dave dank ik voor hun hulp bij het corrigeren van dit proefschrift.

Tenslotte ben ik alle medewerkers en studenten van de groep Warmtetransport bijzonder erkentelijk voor de gezelligheid en vriendschap die ik de afgelopen jaren heb mogen ondervinden.
Stellingen
behorende bij het proefschrift van Steef Linthorst

1: De relatie welke Glaser geeft, ter berekening van de K-waarde van dubbel glas, is niet in overeenstemming met zijn metingen.

2: Het twee dimensionaal numeriek berekenen van stromingsproblemen is vaak onjuist door optredende drie dimensionale effecten.
(dit proefschrift)

3: De door Choudry in zijn "reconstructive algorithm", t.b.v. de bepaling van de convectieve warmteoverdracht tussen twee vlakke platen, gebruikte aanname, dat het temperatuurverloop recht evenredig is met de plaats, is in tegenspraak met zijn eerder geformuleerde "structural algorithm".
(A. Choudry, "Automated digital processing of interferograms", proefschrift TH Delft 1982)

4: De invloed van warmtestraling op warmteoverdrachtsprocessen wordt vaak ten onrechte onderschat.

5: De door Hollands e.a. beschreven resultaten aangaande convectieve warmteoverdracht in honingraatstructuren, geven slechts een indicatie wanneer convectieve warmteoverdracht optreedt; de gevonden Nusselt waarden zijn echter niet representatief voor de warmteoverdracht.
(K.G.T Hollands, G.D. Raithby, T.E. Unny, "Methods for reducing heat losses from flat plate solar collectors", Univ. of Waterloo Research Institute)

6: Gelet op de warmtestraling afschermende werking van honingraatstructuren verdient het aanbeveling te streven naar een verhoging van de absorptiecoëfficiënt van de toegepaste spectrale selectieve laag in de CSD collector, i.p.v. een verdere reductie van de emissiecoëfficiënt.
(dit proefschrift)
Het berekenen van collector dagrendementen voor enkele standaarddagen, dient te worden opgenomen in collectortest procedures.

Om de resultaten van de diverse numerieke programma's te vergelijken, is het noodzakelijk de oplossingsalgoritmen te toetsen aan zgn. "benchmark" problemen.

Gezien de grote verschillen, welke voorkomen in de resultaten van de CEC "Round Robin" collectortests, is het noodzakelijk de internationale collectortest procedure verder te verbeteren.

Het op financiële gronden beperken van het aantal open hartoperaties is een vorm van passieve euthanasie.

Het huidige systeem voor het vervullen van vervangende dienstplicht door gewetensbezwaarden, schept voor hen een bevoorrechte uitgangspositie op de arbeidsmarkt.