Natural convection heat transfer from a trombe wall geometry to a rectangular enclosure
by Peng-Cheng Lin

A thesis submitted in partial fulfillment of the requirements for the degree of MASTERS OF SCIENCE in Mechanical Engineering
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Abstract:
A 1/18th scale experimental model was developed to characterize the natural convection heat transfer in passive solar heating with a Trombe wall geometry. Steady natural convection heat transfer was measured from an isothermal heated Trombe wall (inner body) with various gap sizes, 2.54 cm gap, 0.635 cm gap, and zero gap, to a rectangular enclosure (outer body). Temperature profiles within the test space were obtained and fluid flow patterns were observed. Dow Corning 20 cs fluid was utilized as the working fluid with Prandtl numbers which ranged from 125 to 277. The inner body with 2.54 cm gap appears to give a highest heat transfer from the Trombe wall, especially in the low temperature range, due to the exchange of mass between the Trombe wall space and living space. However, when the gap size was increased the temperature stratification in the living space was also increased. A circulation cell around the upper portion of the living space was observed. The recommended empirical equation describing the heat transfer using two independent parameters was $Nu_h = 1.6106\ Ra_H^{0.1760} (H/H_i)^{-0.2159}$ for $6.2 \times 10^8 < Ra_H < 1.5 \times 10^9$ with an average percent deviation of 3.01.

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NATURAL CONVECTION HEAT TRANSFER FROM A TROMBE WALL GEOMETRY TO A RECTANGULAR ENCLOSURE

by

PENG-CHENG LIN

A thesis submitted in partial fulfillment of the requirements for the degree

of

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in

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Figure | Page

NOMENCLATURE

Symbol	Description
A	Aspect ratio, H/L
A_i	Heat transfer area on the inner body with zero gap, 929 cm^2 (1.0 ft^2) (3-7)
A_p	Aperture ratio, the ratio of central opening to the enclosure height
C_{1-4}	Empirically determined constants
C_ℓ	Universal function of Pr for laminar flow (2-10)
C_p	Specific heat at constant pressure
C_t	Universal function of Pr for turbulent flow(2-11)
f	Denotes arbitrary function
g	Acceleration of gravity, 9.81 m/sec^2 (32.17ft/sec^2)
Gr	Grashof number, $g\beta\rho^2X^3\Delta T/\mu^2$
H_i	Height of the inner body, Figure 1.1
H	Height of the outer body, Figure 1.1
h	Heat transfer coefficient
k	Thermal conductivity
L	The test rectangular enclosure total length
L_1	Left hand side space length, Figure 1.1
L_2	Right hand side space length, Figure 1.1
Nu_s	Nusselt number, hs/k
Pr	Prandtl number, $\mu C_p/k$

Symbol	Description
Q_{cond}	Heat transfer by conduction
Q_{conv}	Heat transfer by convection
Q_{rad}	Heat transfer by radiation
Q_{tot}	Total amount of heat transfer, $Q_{tot} = Q_{conv} + Q_{condv} + Q_{rad}$
Ra_s	Rayleigh number, $C_p g \beta \rho^2 s^3 \Delta T/\mu k$
s	Some characteristic length
T_c	Average cold surface temperature
T_h	Average hot suface temperature
T_f	Film temperature, $(T_h + T_c)/2$
T^*	Dimensionless temperature, $(T - T_c)/(T_h - T_c)$
ΔT	Temperature difference, $T_h - T_c$
x	Horizontal distance from the south facing wall of the enclosure
x^*	Dimensionless length, x/L_1
X	Distance along vertical flat plate from leading edge
y	Vertical distance from the enclosure top
y^*	Dimensionless length, y/H
β	Thermal expansion coefficient
μ	Dynamic viscosity
ρ	Density of the fluid

ABSTRACT

A 1/18th scale experimental model was developed to characterize the natural convection heat transfer in passive solar heating with a Trombe wall geometry. Steady natural convection heat transfer was measured from an isothermal heated Trombe wall (inner body) with various gap sizes, 2.54 cm gap, 0.635 cm gap, and zero gap, to a rectangular enclosure (outer body). Temperature profiles within the test space were obtained and fluid flow patterns were observed. Dow Corning 20 cs fluid was utilized as the working fluid with Prandtl numbers which ranged from 125 to 277. The inner body with 2.54 cm gap appears to give a highest heat transfer from the Trombe wall, especially in the low temperature range, due to the exchange of mass between the Trombe wall space and living space. However, when the gap size was increased the temperature stratification in the living space was also increased. A circulation cell around the upper portion of the living space was observed. The recommended empirical equation describing the heat transfer using two independent parameters was

$$Nu_H = 1.6106\ Ra_H^{0.1760}\ (H/H_i)^{-0.2159}$$

for $6.2\text{x}10^8 < Ra_H < 1.5\text{x}10^9$ with an average percent deviation of 3.01.

CHAPTER I

INTRODUCTION

The study of natural convection heat transfer within enclosures has been receiving increasing attention in recent years. Despite this, there is still a dearth of knowledge in this field. The study of natural convection within enclosures has not advanced sufficiently to the point where an analytical or a numerical investigation can be developed without simplifying the governing equations, idealizing the boundary conditions, or applying the boundary layer assumptions. As a result, experimental studies are needed to tackle the problem of natural convection heat transfer within enclosures. This is particularly true for predicting fluid flow behavior from a body to an enclosure. Interest has been concerned with solar heating design, nuclear reactor technology, cooling electronic devices, aircraft cabin design, and numerous other applications. Most recently applications in passive solar heating have required a better understand of natural convection within enclosures.

Since 1967, when the prototype "Trombe Wall" house was built and monitored in Odeillo, France, several experimental and analytical investigations have been presented pertinent to this geometry in passive solar heating. The passive solar heating with a Trombe wall geometry is illustrated in

Figure 1.1. Two disadvantages of full-scale building studies were observed: (1) the thick concrete Trombe wall has a long conductive time constant resulting in a time dependent heat transfer problem, (2) solar irradiation changes too rapidly making full-scale data difficult to evaluate. Nevertheless, the convective time constant is only a few seconds, which is calculated by dividing Trombe wall height by airflow velocity [30]. Thus, steady convective data are appropriate for application to the time dependent heat transfer behavior of a Trombe wall system. Since passive solar heating designs, including the Trombe wall itself, are becoming economically competitive with conventional heating systems, there is a need to obtain steady convective data for application to a Trombe wall system. These considerations led Mussulman and Warrington [1] to propose to build a 1/18 scaled experimental model to eliminate the above two disadvantages. The criteria for designing this experimental model will be described in chapter III.

The objective of the present study was to develop a 1/18th scale experimental model to characterize the natural convection heat transfer in passive solar heating with a Trombe wall geomety. In order to accomplish this the heat

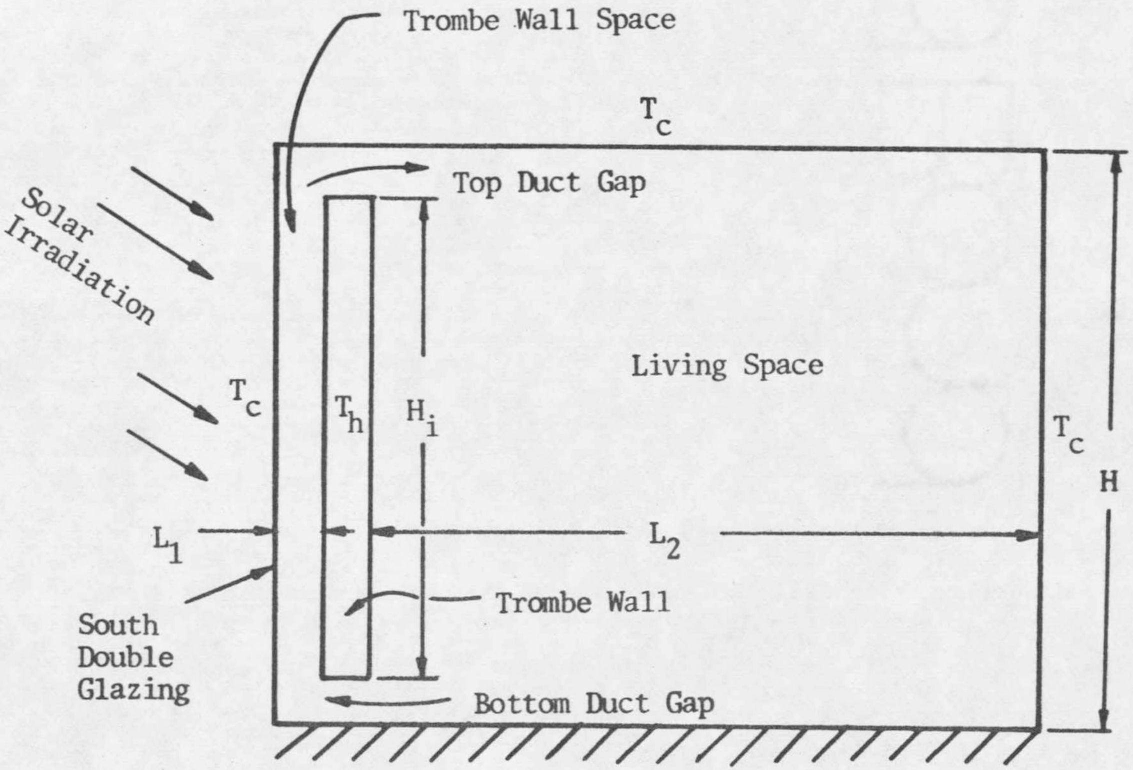

Figure 1.1 Solar Passive Heating with a Trombe Wall Geometry

transfer from an isothermal heated Trombe wall geometry with three different gap sizes to a rectangular enclosure was measured. Temperature profiles within the Trombe wall space and living space were obtained and the fluid flow patterns in the living space were observed. Dow Corning 200 (silicone) fluid with a kinematic viscosity of 20 centistokes at 32°C (90°F) was utilized as the test fluid. The Rayleigh number based on the enclosure height varied from 6.2×10^8 to 1.5×10^{10}. Moreover, empirical equations to determine the heat transfer within the living space were determined for several different combinations of parameters.

Chapter II

LITERATURE REVIEW

The phenomenon of natural convection is a buoyancy-driven effect, that is, the fluid motion results from density differences caused by temperature gradients in the fluid. A considerable number of analytic and experimental studies of natural convection heat transfer have been made over the last 70 years. The most throughly studied case is that of external natural convection from a body to an infinite fluid medium [2-8]. Some of the more recent studies [9-20] have been focused on natural convection within simple rectangular vertical enclosures or rectangular enclosures inclined from the vertical. Cylindrical and spherical enclosure geometries have also been investigated [21-23]. The study of natural convection heat transfer from several different geometric bodies such as spheres, cubes, and cylinders to both cubical and spherical enclosures has also been presented [24]. Moreover, several recent studies of natural convection within rectangular enclosures have investigated passive solar heating geometries [19-20,24-32].

The focus of the following discussion is directed toward published articles on natural convection. The discussion is organized into two sections: vertical flat plates and rectangular enclosures. In addition, studies in passive solar heating with a Trombe wall geometry are presented.

VERTICAL FLAT FLATES

Numerous early works dealt with natural convection heat transfer from vertical flat plates [2-8]. Researchers have developed theoretical solutions to heat transfer problems by simplifying the governing equations, idealizing the boundary conditions, and applying the boundary layer assumption. These solutions were in a good agreement with experimental results.

Eckert and Jackson [4] analyzed turbulent natural convection on an isothermal vertical plate by applying momentum integral equations to boundary layer theory. A heat transfer equation was derived,

$$Nu = 0.0246(Gr)^{2/5}(Pr)^{7/15}[1+0.494(Pr)^{2/3}]^{-2/5} \qquad (2\text{-}1)$$

for Grashof numbers greater than 10^{10}. Nu, Gr, and Ra without a subscript are based on the height of the vertical flat plate. A comparision with the experimental heat transfer results of Jakob [2] and McAdams [3] showed good agreement. They also pointed out that the transition region was between Rayleigh numbers of 10^8 to 10^{10} with air as the working fluid.

Five years after Eckert and Jackson's work, Bayley [5] proposed a theoretical analysis of turbulent natural convection from an isothermal vertical plate. Two equations were presented

$$Nu_X = 0.10(Gr_X Pr)^{0.33}, \text{ for } 2\times10^9 < (Gr_X Pr) < 10^{12} \quad (2\text{-}2)$$

and

$$Nu_X = 0.183(Gr_X Pr)^{0.31}, \text{ for } 10^{12} < (Gr_X Pr) < 10^{15} \;. \quad (2\text{-}3)$$

In addition, an approximation of the heat transfer for mercury ($Pr = 0.01$) was

$$Nu_X = 0.06(Gr_X)^{0.25}, \text{ for } 10^{10} < Gr_X < 10^{15} \;. \quad (2\text{-}4)$$

With the same geometry, Warner and Arpaci [6] performed an experimental investigation of turbulent natural convection along a vertical isothermal heated flat plate with air as the test medium. Several plots of temperature profile data near the hot wall were obtained. The heat transfer results of their study have shown good agreement with the analytical correlation of Bayley [5] for Rayleigh numbers up to 10^{12}. The one third power of this correlation showed that when turbulent free convection is encountered, the local heat transfer coefficient is essentially constant with X.

Vliet and Liu [7] experimentally studied turbulent natural convection boundary layers for a constant heat flux surface condition using water as the working fluid. The isothermal surface condition data of Eckert and Jackson [4] and Bayley [5] have revealed quite good agreement with their heat transfer data, but exhibited a much earlier transition point due to the different heating modes.

A general correlation of laminar and turbulent free natural convection from an isothermal vertical surface was developed by Churchill and Chu [8]. This correlation is

$$Nu^{1/2} = 0.825+0.387Ra^{1/6}/[1+(0.437/Pr)^{9/16}]^{8/27} \quad (2\text{-}5)$$

which may be applied to the entire range of Rayleigh numbers. Although this equation is suitable for most engineering calculations, slightly better accuracy can be obtained for laminar flow by using

$$Nu = 0.68+0.67Ra^{1/4}/[1+(0.492/Pr)^{9/16}]^{4/9} \quad (2\text{-}6)$$

for $Ra < 10^9$. The two resulting equations may be applied for constant heat flux conditions as well as for constant surface temperature conditions.

RECTANGULAR ENCLOSURES

Natural convection within enclosures has been investigated analytically from the middle of this century. However, the interactions of the boundary layers of the enclosure with its core region cause complex flow patterns making analytical solutions difficult to obtain. Theoretically, at large Rayleigh numbers the heat transfer across each of the turbulent boundary layers might approach the heat transfer from a vertical plate in an infinite medium; this was not found experimentally. The existing analytical solutions for natural convection within rectangular enclosures dealt only with the steady two-dimensional case. Such solutions have compared poorly with the experimental results for high Rayleigh numbers with aspect ratios less than one because three-dimensional effects are important. The aspect ratio is defined as the ratio of the enclosure height to length normal to the hot wall. In general, rectangular enclosures with two opposite vertical walls at different temperatures and insulated horizontal surfaces have been extensively examined. The depth (the distance between the remaining walls) of this rectangular enclosure was made sufficiently large to assure two-dimensional flow in the central region of the cavity.

Natural convection in a rectangular cavity with adiabatic top and bottom surfaces and two opposing isothermal vertical walls at different temperatures was first investigated analytically by Batchelor [9]. He expanded temperature and stream functions in power series of the Rayleigh number and defined various flow regimes. The resulting equation was

$$Nu_L = 0.48\ Ra_L^{1/4}(A)^{3/4} \text{ for } (Ra_L/500) > A. \qquad (2\text{-}7)$$

Eckert and Carlson [9] made an experimental study of natural convection in a rectangular enclosure with three different aspect ratios (2.5, 10, and 20) using water as the working fluid. Temperature distributions within the enclosed space were obtained using an interferometer to describe conduction, transition, and boundary regimes. Grashof numbers based on the enclosure height were on the order of 10^6 in this investigation. Flow fluctuation and wave motions were observed in some cases.

Elder [11-12] performed an extensive experimental investigation in a rectangular cavity with aspect ratios ranging from 1 to 60 for laminar and turbulent flow regimes. Medicinal paraffine, silicone oil ($Pr \sim 1000$) and water were utilized as test fluids. The flow was made visible by using aluminum powder suspended in the fluid. The experiments were conducted especially to gain further insight into the fluid flow behavior. Travelling wavelike motions growing up the hot wall of the slot and down the cold wall were observed for Rayleigh numbers above approximately $8 \times 10^8 (Pr^{1/2}/A^3)$. These waves developed most readily midway between the two endwalls. Near $Ra = 10^{10}/A^3$ an intense entrainment and mixing process between the region near the wall and the interior was initiated. As the Rayleigh number increased, the turbulent middle portion of the flow extended further toward the endwalls.

Numerical and experimental studies of natural convection between vertical planes with moderate and high Prandtl number fluids were carried out by MacGregor and Emery [13]. The vorticity and stream function were substituted into the governing equation and a numerical solution was obtained by using a finite difference technique for isothermal and constant heat flux boundary conditions. They concluded that the net heat transfer was a strong function of the aspect ratio when correlated with Ra_L. The correlations for both the isothermal and constant heat flux conditions obtained from the experiments were:

$$Nu_L = 0.42(A)^{-0.30}Pr^{0.012}Ra_L^{0.25} \text{ for } 10^4<Ra_L<10^7 \qquad (2\text{-}8)$$

with aspect ratios from 1 to 40. However, the following one parameter correlation could be used:

$$Nu_L = 0.046\ Ra_L^{1/3} \quad \text{for } Ra_L > 10^6 \qquad (2\text{-}9)$$

Ostrach and Raghaven [14] conducted an experimental study which described the effect of stabilizing thermal gradients on natural convection in rectangular enclosures with aspect ratios of 1 and 3. In order to track the streamlines and measure the approximate velocity of the fluid, Pliolite plastic particles were mixed into silicone oils with kinematic viscosities of 10,000 and 2,000 centistokes at 25 C. Thermal boundary conditions were established by heating, cooling, or insulating on opposite walls of the enclosure so that simulataneous horizontal and vertical heat flows were achieved. Different streamline patterns were observed by varying the ratio of vertical to horizontal Grashof numbers and the aspect ratio. The results showed that a stabilizing effect on the flow was established by heating the upper surface and cooling the lower surface.

An approximate analysis of natural convection within the rectangular enclosures was proposed by Raithby et al. [15]. The resulting solutions were

$$Nu_L = 0.75C_{\ell}(Ra_L/A)^{1/4} \tag{2-10}$$

and for the laminar regime

$$Nu_L = 0.29C_t(Ra_L)^{1/3} \tag{2-11}$$

for the turbulent regime where $C_{\ell} = 0.50/(1+(0.49/Pr)^{9/16})^{4/9}$ and $C_t = 0.14Pr^{0.084}$. Good agreement was found in comparing these equations to the data available up to 1975. There is a lack of data at high Rayleigh numbers ($Ra > 10^{10}$) and low aspect ratios ($A < 5$). The validation of predictions covering this range of parameters has to rely on further experiments.

Berkovsky and Polevikov [16] developed a numerical solution for natural convection heat transfer within vertical slots. The heat transfer solution was

$$Nu_L = 0.22(A)^{-0.25}(Ra_L Pr/(0.2+Pr))^{0.28} \qquad (2\text{-}12)$$

for $2 < A < 10$, $Pr < 10^5$ and $Ra_L < 10^{10}$.

Catton, Ayyaswamy, and Clever [17] studied convection in a rectangular cavity at various angles of tilt. Of interest in this analytical investigation was the comparison of their results for adiabatic and perfectly conducting horizontal surfaces. The results for a vertical slot showed that the overall heat transfer across the gap is lower for perfectly conducting horizontal surfaces than for adiabatic horizontal surfaces. Examination of the local heat transfer indicates that at high aspect ratios, there is very little difference except very near the horizontal surface. A significant difference can be seen at the lower aspect ratios ($A = 1$ and $A = 0.2$). This is due to the more pronounced thermal interaction at the perfectly conducting boundaries. With a similar geometry, Elsherbiny, Raithby, and Holland [18] obtained an empirical correlation for a vertical air slot with perfectly conducting horizontal surfaces:

$$Nu_1 = 0.0605\ Ra_L^{1/3} \tag{2-13}$$

$$Nu_2 = [1+\{0.104Ra_L^{0.293}/(1+(6310/Ra))^{1.36}\}^3]^{1/3} \tag{2-14}$$

$$Nu_3 = 0.242(Ra_L/A)^{0.272} \tag{2-15}$$

and the maximum of Nu_1, Nu_2, and Nu_3 was recommended for $5 < A < 110$ and $10^2 < Ra_L < 2x10^7$.

More recently the works of Bauman et al. [19] and Nansteel et al. [20] were motivated by studies of natural convection heat transfer within buildings. Bauman et al. [19] conducted an experimental and numerical study of the natural convection heat transfer in a rectangular enclosure which simulated a full scale room with an aspect ratio of 0.5. The enclosure consisted of two vertical copper endwalls at different temperatures and adiabatic plexiglas horizontal surfaces. The Rayleigh numbers achieved, based on the enclosure length, were about 10^{10} using water as the working fluid. The experimental heat transfer result in the enclosure appeared somewhat higher than the approximation of Raithby et al. [15]. This may have occured, because the vertical boundary conditions were not perfectly isothermal and the horizontal surfaces were not well insulated. With a similar geometry and the same working fluid, Nansteel and Greif [20] focused on an experimental study of natural convection in undivided and partially divided rectangular enclosures. Rayleigh numbers based on the enclosure length in the range of 2.3×10^{10} to 1.1×10^{11} were obtained. It seemed that no fully developed turbulent flow was observed within the enclosure, even for Ra_L as high as 10^{11}. The recommended heat transfer correlations were

$$Nu_L = 0.748\ A_p^{0.256}\ Ra_L^{0.226} \tag{2-16}$$

for a conducting partition and

$$Nu_L = 0.726\ A_p^{0.473}\ Ra_L^{0.226} \tag{2-17}$$

for an adiabatic partition, where A_p is the ratio of the central opening to the height of the enclosure, and is called the aperture ratio.

From this review of the existing work on heat transfer by natural convection within a rectangular enclosure, the important dimensionless parameters are

$Nu_s = hs/k$ (Nusselt Number),

$Gr_s = (g\beta\rho^2 s^3 \Delta T)/\mu^2$ (Grashof Number),

$Pr = (\mu C_p)/k$ (Prandtl Number),

and $A = H/L$ (Aspect Ratio),

where s is some characteristic length. The dimensionless grouping (Gr_s Pr) is widely used as the Rayleigh number, which is

$$Ra_s = Gr_s\ Pr = (C_p g \beta\rho^2 s^3 \Delta T)/\mu k\ .$$

In general, the heat transfer by natural convection can be determined in functional form as

$$Nu_s = f\ (A,\ Ra_s,\ Pr)\ .$$

Since 1967, when the prototype Trombe wall house was built and monitored in Odeillo, France, several analytical and experimental investigations have been presented pertinent to this geometry in passive solar heating.

Balcomb et al. [27-29] conducted several full scale experimental studies in passive solar heating with either unvented or vented Trombe-type thermal storage walls. Temperature variations of the building walls and the ambient room temperature on a daily or monthly basis were obtained. The effect of storage capacity on annual energy delivery for a Trombe-type passive system was also examined. However, no convective heat transfer data were obtained, nor could it be calculated from their experimental data. It is probably difficult to reach the steady state condition for the full scale building, because the solar irradiation changes too rapidly and the Trombe wall is rather thick providing a long conductive time constant (10 to 15 hours) to transfer heat from one side to another.

Free convective laminar flow within the Trombe wall space was assumed similar to the flow between two parallel, infinitely wide vertical plates with an adiabatic bottom surface by Akbari and Borgers [30]. By this assumption, they solved nondimensional boundary layer equations by using a forward-marching, line by line implicit finite difference technique. Several velocity and temperature profiles were obtained for different wall temperatures.

An experimental investigation of the Trombe wall passive solar energy system was carried out by Casperson and Hocevar [31]. The wood-frame test room was constructed with outside dimensions of 3.66 m x 4.27 m x 3.05 m (12 ft x 14 ft x 10 ft). The Trombe wall consisted of a 30.48 cm (12 in.) thick solid concrete block wall with a movable double-glazed Kalwall cover unit, which allowed the wall gap (Trombe wall space) to be varied from approximately 2.54 cm (1 in.) to 25.4 cm (10 in.). Velocity and temperature profiles obtained in the Trombe wall space with room temperature at 14.8 °C (55 °F) and 15.6 °C (60 °F) indicated that the flow is likely to be turbulent. No heat transfer data were obtained in this study.

Stotts, Warrington, and Mussulman [25] developed a computer model to simulate direct gain, indirect gain, isolated gain, and Trombe wall passive solar heating systems. The Trombe wall model performance was verified using data from passive test cells at NCAT (National Center for Appropriate Technology) [32], located in Butte, Montana. Radiative interactions between room's interior surfaces were considered. Convection from the walls to the air was also considered. The simulated globe temperature showed good agreement with the measured values.

The study of natural convection within enclosures has increased rapidly in the last decade. Results describing the heat transfer within rectangular enclosures are not applicable for the full-scale building in passive solar heating due to aspect ratio mismatches. Although several studies have been made pertinent to the passive solar heating with a Trombe wall geometry, no natural convection heat transfer data have been published to date. There is a definite need to study the phenomena of natural convection in passive solar heating with a Trombe wall geometry with either scaled models or full-scale structures. These studies will help the development of computer simulations and the design of Trombe wall systems. The intent of this investigation was to develop a scale model for convective heat transfer which simulated a full-scale structure with Trombe wall geometries.

CHAPTER III

EXPERIMENTAL APPARATUS AND PROCEDURE

EXPERIMENTAL APPARATUS

A 1/18th scale parametric study of passive solar heating with a Trombe wall geometry was carried out as part of the present experimental investigation. The present experiment was to simulate a two story high rectangular room with inside dimensions of 5.5 m in height, 8.2 m in length, and 5.5 m in depth. Typical values of the dimensionless parameters characterizing the convection process for this room, filled with air at 21 °C, and with a maximum of 9 °C temperature difference [26] between a Trombe-type thermal storage wall and the inside building are:

$$A = H / L = 0.67,$$

$$Pr = 0.71,$$

and $$Ra_H = Gr_H\, Pr < 1.4\times10^{11}.$$

The range of Ra_H, based on the height of room, suggests that natural convective flow within this building can be either laminar or turbulent, governed by the specific temperature distributions on the boundary surfaces and the configuration of the enclosure. In order to characterize buoyancy-driven convection in this two story room, the design of a heat transfer apparatus has to meet the above requirements. The experiment covered the following range of parameters

$$A = 0.67$$

$$124.7 < Pr < 276.9$$

$$6.2x10^8 < Ra_H < 1.5x10^{10}$$

using 20 cs fluid as a working fluid was appropriate because it permitted typical Ra_H to be approached in a 1/18th scale apparatus; Raithby et al. [15] have pointed out that natural convection processes are insensitive to Pr > 5. Although the range of present Prandtl numbers is higher than the Prandtl number of air, the Nusselt numbers attained in this experiment, the general flow pattern, and heat transfer can be expected to be similar to the full-scale building.

A heat transfer apparatus was then designed to provide the capability to study the heat transfer from an isothermal heated Trombe wall geometry to a rectangular enclosure. Moreover, the temperature profiles and the natural convective flow around this Trombe wall geometry were investigated. A photograph of the assembled apparatus and peripheral components is shown in Figure 3.1. A schematic of the entire experimental system, consisting of a rectangular test enclosure, a water jacket, a vertical heated wall, a water cooling system, and temperature controlling and monitoring instruments, is shown in Figure 3.2. These will be described in detail below.