Analyses of steady and unsteady thermal behaviour of simulated electronic chips in a liquid cooled vertical rectangular channel

Himangshu Bhowmik

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**Analyses of Steady and Unsteady Thermal Behavior of Simulated
Electronic Chips in a Liquid Cooled Vertical Rectangular Channel**

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School of Mechanical & Aerospace Engineering

A thesis submitted to the Nanyang Technological University
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Doctor of Philosophy

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Nanyang Technological University
School of Mechanical and Aerospace Engineering
PhD Dissertation

Analyses of Steady and Unsteady Thermal Behavior of Simulated Electronic Chips in a Liquid Cooled Vertical Rectangular Channel

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Summary

Steady-state and transient experiments are performed to study the single-phase heat transfer from four in-line simulated electronic chips in a vertical rectangular channel using water and FC-72 as the working fluids. The steady-state experimental data covers a laminar flow regime with Reynolds number based on channel hydraulic diameter and Reynolds number based on heater length ranging from 0 to 2220 and 0 to 2775 respectively. The heat flux ranges from 0.1 W/cm² to 2.6 W/cm² for steady-state mixed and forced convection heat transfer and 0.1 W/cm² to 0.6 W/cm² for steady-state natural convection heat transfer. The transient associated with turning on the heater power, turning off the pump, turning on the pump, turning off the heater power and natural convection heat transfer with power-on transient operations are investigated. The transient experimental data covers the laminar flow regime with Reynolds number, based on heat source length, from 800 to 2625. The heat flux ranges from 1 W/cm² to 7 W/cm² for heater power-on, pump-off, pump-on and heater power-off transient heat transfer and 0.1 W/cm² to 0.6 W/cm² for power-on transient natural convection heat transfer. For flush-mounted ($B = 0$) chips, the effects of heat flux, coolant flow rates and chip numbers are investigated for both steady-state and

time dependent conditions. The effects of protruding height ($B/L = 0, 0.1, 0.2$), and working fluid (water, FC-72) are also investigated.

Flow regimes for steady-state natural, mixed and forced convection are derived from the heat transfer data for flush-mounted chips. The experimental results indicate that the heat transfer coefficient is affected strongly by the heat flux, flow rates and protruding heights. Results from the present study for water-cooling agree well with the results from the open literature for air-cooling of similar conditions. For flush-mounted chips, heat transfer characteristics are studied and empirical correlations are developed for forced, mixed and natural convection heat transfer. For different protrusions, the effects of heat fluxes and flow rates are investigated and found that the protrusion heights lead to increase the heat transfer.

In the experiments of transient heat transfer, the thermal behaviors of flush-mounted electronic chips are studied for water-cooling. The five transient regimes namely power-on, pump-off, pump-on, power-off and power-on transient natural convection are obtained from the experimental data. In all the test cases, temperatures are measured in one-second intervals and the numbers of data are recorded depending on the nature of the curves. From the heat transfer data it can also be seen that chip number independency starts from the third chip. The correlation equations in the transient regimes are reported for individual chip as well as for overall data. The effects of protrusion heights are studied for pump-off and pump-on transient operations and obtained significant effect on heat transfer. The heat transfer coefficient for B/L of 0.2 is greater than that for B/L of 0 and 0.1. The effects of

protrusion diminish with time at the pump-off transient operation and increase with time at the pump-on transient operation. The pump-on transient results elucidate that with increasing time forced convection plays a significant role to increase the heat transfer and the upstream heater (Chip 1) provides greater heat transfer than the downstream chips.

Flow regimes for pump-off, pump-on transient natural convection are derived from the heat transfer data and the correlation equations are reported for FC-72 coolant. It is observed that the power-on transient natural and power-on transient forced convection data are similar for water and FC-72 coolant. By using FC-72, overall increase of 30% and 70% of Nusselt number data are obtained from power-on transient natural and forced convection heat transfer respectively.

The three-dimensional heat transfer through the vertical rectangular channel is also investigated numerically for steady-state and pump-off transient conditions by using **Fluent**. The experimental results are compared with the numerical results and found a favorable agreement with the steady-state and pump-off transient heat transfer results. The numerical correlations are also reported for steady-state and pump-off transient heat transfer.

Key words: Steady and Unsteady Heat Transfer, Laminar Flow, Electronic Cooling, Discrete Heat Sources, Forced, Mixed, Natural Convection, Numerical Modeling.

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Nomenclature

A	channel aspect ratio = $\frac{H}{W}$
A_{ex}	exposed surface area of heat source, m^2
A_f	fluid flow area = WH , m^2
B	protruding height of heat source, m
B_i	Biot number = $\frac{hH_c}{k_c}$
b	annular gap between concentric spheres
C_1, C_2	coefficients in correlations
C_3, C_4	coefficients in correlations
c_p	specific heat at constant pressure, $J/kg \cdot ^\circ C$
D	hydraulic diameter = $\frac{2WH}{W + H}$, m
Fo	non-dimensional time = $\frac{\alpha t}{H_c^2}$
Gr_B	Grashof number based on protruding height = $\frac{g \beta q'' B^4}{k \nu}$
Gr_H	Grashof number based on channel height = $\frac{g \beta q'' H^4}{k \nu}$
Gr_ℓ	Grashof number based on heat source length = $\frac{g \beta q'' \ell^4}{k \nu}$
g	gravitational acceleration, m/s^2
H	height of the channel, m
H_c	height of the chip, m

$\bar{h}$	average heat transfer coefficient, $\text{W/m}^2 \cdot \text{K}$
k	thermal conductivity of fluid, $\text{W/m} \cdot \text{K}$
k_c	thermal conductivity of chip (copper), $\text{W/m} \cdot \text{K}$
L	characteristic wall length of the channel, m
ℓ	length of heat source, m
m, n	exponents
Nu	Nusselt number
$\overline{Nu}$	average Nusselt number
Nu_B	Nusselt number based on protruding height = $\frac{hB}{k}$
$\overline{Nu}_B$	average Nusselt number based on height of the heater = $\frac{\bar{h}B}{k}$
$\overline{Nu}_b$	average Nusselt number based on annular gap = $\frac{\bar{h}b}{k}$
Nu_C	Nusselt number based on flow passage = $\frac{hC}{k} = \frac{h(H-B)}{k}$
Nu_D	Nusselt number based on protruding height = $\frac{hD}{k}$
Nu_H	Nusselt number based on channel height = $\frac{hH}{k}$
$Nu(H/L)$	Nusselt number based on channel height = $\frac{hH}{kL}$
Nu_ℓ	Nusselt number based on heat source length = $\frac{h\ell}{k}$
$\overline{Nu}_L$	average Nusselt number based on channel length = $\frac{\bar{h}L}{k}$
$\overline{Nu}_\ell$	average Nusselt number based on heat source length = $\frac{\bar{h}\ell}{k}$

$\overline{Nu_o}$	average Nusselt number for forced convection
Nu_W	Nusselt number based on channel width = $\frac{hW}{k}$
Nu_x	local Nusselt number = $\frac{hx}{k}$
Pe	Peclet number based on channel height = $\frac{U_i H}{\alpha}$
Pe_ℓ	Peclet number based on heat source length = $Re_\ell \cdot Pr$
Pr	Prandtl number = $\frac{\nu}{\alpha}$
Q	flow rate, m ³ /s
q	heater power, W
q''	heat flux, W/cm ²
q_o	prescribed heat flux, W/m ²
R	resistance, Ω
Ra	Rayleigh number based on channel height = $Gr_H \cdot Pr$
Ra^*	modified Rayleigh number = $\frac{Ra_H}{A}$
Ra'	channel Rayleigh number = $\frac{\rho g \beta c_p (\Delta T) H W^2}{k \nu}$
Ra_B	Rayleigh number based on protruding height = $\frac{g \beta k (\Delta T) B}{\nu \alpha}$
Ra_b^*	modified Rayleigh number = $\frac{g \beta q'' b^4}{k \nu \alpha}$
Ra_H	Rayleigh number based on channel height = $\frac{g \beta k (\Delta T) H}{\nu \alpha}$
$Ra(H/L)$	Rayleigh number based on channel height = $\frac{g \beta k (\Delta T) H}{\nu \alpha L}$

Ra_ℓ	Rayleigh number based on heat source length = $\frac{g\beta q'' \ell^4}{k\nu\alpha}$
Ra_W	Rayleigh number based on heat source length = $\frac{g\beta k(\Delta T)W}{\nu\alpha}$
Re_B	Reynolds number based on protruding height = $\frac{UB}{\nu}$
Re_c	Reynolds number based on flow passage = $\frac{Uc}{\nu} = \frac{U(H-B)}{\nu}$
Re_D	Reynolds number based on hydraulic diameter = $\frac{UD}{\nu}$
Re_H	Reynolds number based on heat source length = $\frac{UH}{\nu}$
Re_L	Reynolds number based on channel length = $\frac{UL}{\nu}$
Re_ℓ	Reynolds number based on heat source length = $\frac{U\ell}{\nu}$
Re_W	Reynolds number based on channel width = $\frac{UW}{\nu}$
Ri_D	Richardson number = $\frac{Gr_D}{Re_D^2}$
S	spacing between the heat source in x and z direction, m
SD	standard deviation
S_ϕ	source term
t^*	dimensionless time = $t/(W/U)$
t	time, s
Δt	time step, s
ΔT	temperature difference between heat source and fluid, K
T_{ad}	adiabatic temperature, K

T_{in}	channel inlet temperature, K
T_m	mixed mean temperature, K
T_{wall}	temperature of the heated wall, K
U	fluid velocity = $\frac{Q}{A_f}$, m/s
u	velocity at the x direction, m/s
V	voltage, volt
v	velocity at the y direction, m/s
W	width of the rectangular channel, m
w	velocity at the z direction, m/s
x	distance along the surface, m
x_o	unheated starting length, m

Greek symbols

α	thermal diffusivity = $\frac{k}{\rho c_p}$, m ² /s
β	thermal expansion co-efficient, 1/K
β_i	inclination angle, deg
μ	dynamic viscosity, N·s/m ²
ν	kinematic viscosity = $\frac{\mu}{\rho}$, m ² /s
ρ	fluid density, kg/m ³
θ	dimensionless temperature = $\frac{T_{wall} - T_i}{\Delta T}$
ϕ	general variable
Γ	diffusion coefficient, N·s/m ²

Subscripts

<i>ad</i>	adiabatic condition
<i>c</i>	flow passage = $(H - B)$
<i>h</i>	for horizontal orientation
<i>f</i>	fluid
<i>h</i>	for horizontal orientation
<i>in</i>	inlet
<i>m</i>	channel entrance condition
<i>s</i>	substrate
<i>v</i>	for vertical orientation
<i>wall</i>	wall of heat source
<i>x</i>	local value
1,2,3	first, second and third electronic component

Chapter 1 Introduction

1.1 Background

Continuing miniaturization of semiconductor electronics has given rise to increasing sophistication in the accompanied development of cooling technology. However, a major driving force in the development of digital computers has been the achievement of ever larger scales of integration for monolithic circuits on a silicon chip. Almost all of the electrical energy consumed by the electronic chips appears as heat. With increasing power density within the chips, as well as the attendant tendency to design chips closely in planar arrays, more stringent requirements in thermal management are needed. This has prompted the consideration of liquid cooling, instead of the prevalent air-cooling, to remove the increasing cooling load from high-power electronic chips (Xu et al. 1996, Chu 1986 and Hannemann et al. 1987).

The problem may be viewed as the one involving convection from small, discrete heat sources mounted on a substrate that may be either a good or a poor thermal conductor. The heat generated per unit volume in the electronic equipment has increased substantially, increasing further the average temperature of the equipment. Studies show that high temperature increases the failure rate and reduces the lifespan of electronic equipment. From the heat transfer viewpoint, electronics cooling has created emphasis on understanding the basic convective fluid flow over discrete heat sources, which have different characteristics from the traditionally studied convection from a heated whole wall.

The circuit densities on a single silicon chip continue to increase and as more chips are packed in closer proximity on multi-chip module, power densities continues to rise at both the chip and module levels. A tremendous effort is made to maintain an acceptable operating temperature below an allowable level of about 85°C. The power density for a single chip is already up to 40 W/cm² and is expected to increase up to 100 W/cm² in the next decade. This has generated great interest and created an impact on the direction of research in heat transfer. Out of the important means of thermal management of electronics cooling is the flow cooling in a channel as shown in Fig. 1.1.

Fig. 1.1 Electronics cooling classification for channel flow.

The remarkably rapid progress in the performances of computers can be directly tied to the cooling processes in semiconductor technology. There are five distinct computer technology generations as shown in Table 1.1. In this table, “VLSI” stands for very large scale integration and “ULSI” for ultra large scale integration respectively. The trend towards higher packaging densities has continued with each new circuit technology generation bringing more and more devices per chip. Increases in circuit density inevitably have resulted in increasing chip power. Such increases have made the role of heat transfer and thermal design more critical than ever, and the development of future large-scale, high-speed circuits may be limited by the inability to maintain effective cooling.

Table 1.1. Computer technology generations (Hannemann 1990)

1945 - 1955	First generation: Vacuum tube electronics
1955 - 1965	Second generation: Transistors (10^0 circuits/chip)
1965 - 1975	Third generation: Small scale integration (SSI) (10^1 circuits/chip)
1975 - 1985	Fourth generation: Large-scale integration (LSI) (10^3 circuits/chip)
1985 -	Fifth generation: LSI – VLSI – ULSI ($10^4 - 10^6$ circuits/chip)

Air-cooling is widely used because of the availability and the ease of handling air. The large air flow rate need more powerful fans, thereby producing system noise and vibration and is limited to chip level heat fluxes of only about $1\text{W}/\text{cm}^2$ (Nakayama and Bergles 1990). As it is becoming increasingly difficult to rely on air cooling to remove the cooling load, liquid cooling considered as it is more effective for removing heat than air cooling and it may be the only practical method for maintaining reasonable component temperatures in the high power chips (Mudawar 1992).

Direct liquid immersion cooling offers a high heat transfer coefficient, which reduces the temperature rise of the chip surface above the liquid coolant temperature. The relative magnitude of the heat transfer coefficient is affected by the coolant and the modes of convection (i.e. forced, mixed and natural convection).

The buoyancy force due to temperature difference acts on the fluid causing it to flow up through the channel is more reliable and low cost. In addition, for vertical channel

orientation, the stream-wise component of the buoyancy force may be predicted to significantly enhance the heat transfer. It is a preferred mode of cooling in many telecommunication devices such as telephone switching units, avionics packages, computers, and many other electronic systems. It allows greater variability in electronic packaging at the board, component and system levels. In many applications, particularly when the electronic devices are operated in non-benign environments, the packages are sealed, and may be modelled as a vertical channel with discrete heat sources on one side wall.

The transient heat transfer considerations are becoming more significant in the design of electronic systems involving heat exchange devices. It is a fundamental problem in thermal science, which has been extensively studied due to its importance in applications such as the heat transfer in double-glazing units, the cooling system of electronic devices, solar thermal engineering and in nuclear reactors.

In recent years, transient problem in electronic systems becoming more important because it is pertinent to the design and performance of electronic devices and the packaging of heat dissipating components. In the analyses of such systems, it is common to focus attention on the heat exchange equipment; for example a single or an array of heated elements and its bounding walls. To cool such systems, two geometries are commonly encountered, that are selected as the circular tube and the parallel plate channels (Siegel 1960).

With the advent of electronics cooling, new studies involving discrete heat sources have been emerged. The average Nusselt number for discrete heating is substantially higher than that of uniform heating (Keyhani et al. 1988). In most of the recent literature, an array of heated square blocks deployed along one wall of a parallel plate duct is used to model the circuit board. The majority of the studies focus on single or multiple strip heat sources and are neglected three-dimensional effects that can be significant for small heat sources arranged in closely spaced arrays. In 1990s, the three-dimensional simulation received a great deal of attention. For three-dimensional simulations, post processing of voluminous numerical data is as important as the process of obtaining solutions, in order to have maximum utilization of the understanding of the complex three-dimensional flow and temperature field. Hence a great deal of work is needed to devise effective data reduction and presentation method.

1.2 Motivations of the Present Research

The motivations toward the present research are

- The increase of heat dissipation within the chips has placed stringent requirement on thermal management and stimulated consideration of liquid cooling. From the standpoint of heat transfer, the problem may be viewed as forced, mixed and natural convection heat transfer from small discrete heat sources. It is worth commenting here that, there is much less works available on discrete heat sources in vertical channel flow under natural and mixed convection conditions and no works are reported on an array of 4×1 heat sources.

- Forced convection has proved to be reliable to remove heat from high-power chips, it necessitates a pump and there may be accompanied undesirable noise and vibration, especially for liquid cooling. Hence mixed convection may offer the significant cooling enhancement with modest penalty of system pressure losses, and such flow condition may be suited to electronics cooling.
- Under low forced convection conditions, with increase in heat fluxes within the device, the buoyancy effect becomes important and leads to a natural or mixed convection situation. Natural convection offers a reliable and low cost cooling process and plays a significant role to increase heat transfer. Increasing power levels within the chips need more coolant flow rates that produce mixed convection flow.
- From both the academic and practical point of view, transient thermal behaviour of electronic systems is important. The transient behaviour can be initiated by the system start-up, shut down or inadvertent loss of coolant flow in the system. At these events, fluid flow and heat flux change with time, leading to temperature changes that affects the system performance and integrity. Thus, there is a need to investigate the transient thermal behaviour of the system, in order to determine the extent of deviation from the normal conditions. It is observed that most of the works in this area focus on the transient associated by changing wall heat fluxes, wall temperatures and flow velocities, no results are produced for transient heat transfer involving discrete heat sources.
- The single-phase heat transfer for protruded chips is higher than that of similar flush-mounted chips. Therefore more attention should be directed towards the

better understanding of the physical mechanism of the effect of protrusions on the thermal behaviour of electronic chips. But in liquid cooling very few studies have been focused on the effects of protrusions and there are hardly any studies on transient heat transfer. All these important issues provide motivations toward the present research.

1.3 Coolant Considerations

The selection of a liquid for direct immersion cooling cannot be made on the basis of heat transfer characteristics alone. Chemical compatibility of the coolant with the chips and other packaging materials exposed to the liquid must be a primary consideration.

There may be several coolants, which can provide adequate cooling, but only a few will be chemically compatible. Water is the most effective coolant and offers greater uniformity of chip temperatures than is provided by air-cooling. But water is unsuitable for direct immersion cooling on account of its chemical characteristics. Fluorocarbon liquids (e.g. FC-72, FC-86, FC-77, etc.) are generally considered to be the most suitable liquids for direct immersion cooling, in spite of their poorer thermo-physical properties. As shown in Table 1.2, the thermal conductivity, specific heat, and heat of vaporization of fluorocarbon coolants are lower than water. These coolants are clear, colourless per-fluorinated liquids with a relatively high density and low viscosity. They also exhibit a high dielectric strength and a high volume resistivity. The boiling points for the commercially available "Fluorinert" liquids manufactured by the 3M Company ranging from 30 to 253 °C.

Table 1.2. Comparison of thermophysical properties of some fluorocarbon coolants and water (Danielson et al. 1987)

Property	FC-87	FC-72	FC-77	H ₂ O
Boiling point, 1atm (°C)	30	56	97	100
Density $\times 10^{-3}$ (kg/m ³)	1.633	1.680	1.780	0.997
Specific heat $\times 10^{-3}$ (W.s/kg.K)	1.088	1.088	1.172	4.179
Thermal conductivity (W/m.K)	0.0551	0.0545	0.057	0.613
Dynamic viscosity $\times 10^4$ (kg/m.s)	4.20	4.50	4.50	8.55
Heat of vaporization $\times 10^{-4}$ (W.s/kg)	8.79	8.79	8.37	243.8
Surface tension $\times 10^3$ (N/m)	8.90	8.50	8.00	58.9
Thermal coefficient of expansion $\times 10^3$ (K ⁻¹)	1.60	1.60	1.40	0.20
Dielectric constant	1.71	1.72	1.75	78.0

These liquids should not be confused with the "Freon" coolants, which are chlorofluorocarbons (CFCs). Although some of the "Freons" (e.g. R-113) exhibit similar cooling characteristics, concern over their environmental effect on the ozone layer preclude their use.

The relative performance and features of cooling modes for gas and liquid are described by Bar-Cohen (1993). Water appears to be the best coolant in direct cooling, as it has large specific heat and high thermal conductivity. The advantages of Fluorinert liquids became apparent. Despite its attractive thermal attributes, direct liquid cooling with or without phase change has not gained widespread popularity in the mainframe computer

industry, largely due to the lack of understanding of the heat transfer mechanisms for dielectric coolant.

1.4 Scopes of the Research

The scopes of the research are

- The steady-state and transient thermal behaviour of a four in-line array of discrete heat sources in a vertical rectangular channel are to be investigated for liquid coolants of water and FC-72.
- The effects of heat flux, coolant flow rates and chip numbers are to be investigated for steady-state as well as transient conditions.
- The steady-state heat convection regimes including single-phase forced, mixed, natural convection heat transfer are to be determined from the experimental data. The empirical correlations are also to be developed in the forced, mixed and natural convection regimes for water cooling.
- Five transient situations, turning on the heater power, turning off the pump, turning on the pump, turning off the heater power and natural convection heat transfer with power-on are to be investigated. The correlation equations are to be produced in the transient regimes for water as well as FC-72 cooling.
- Comparisons of the heat transfer behaviour between water and FC-72 are to be investigated for power-on transient natural and forced convection heat transfer.
- The transient behaviours at three different heat source protrusions ($B = 0, 1, 2$ mm) are to be investigated.

- The steady-state and transient experimental results are to be compared with the three-dimensional numerical simulations using Fluent (1995). The conjugate heat transfer in the solid and fluid regions are also to be investigated.

1.5 Objectives of the Research

The objectives of the research are:

1. To ensure the test rig conditions, the steady-state forced convection results to be compared with the open literatures.
2. To obtain steady-state forced, natural and mixed convective heat transfer regimes from the experimental data for water coolant.
3. To obtain critical heat flux data to develop mixed and natural convection correlations for prediction of CHF from discrete heat sources.
4. To study the effects of heat flux and chip numbers on natural and mixed convection heat transfer.
5. To develop empirical correlations in the mixed and natural convection heat transfer regimes.
7. To obtain transient heat transfer regimes during the heater power-on, pump-off, pump-on, power-off and power-on transient natural convection heat transfer from the experimental data for water coolant.
8. To obtain time dependent transient heat transfer regimes during the pump-off and power-on transient natural convection heat transfer from the experimental data for FC-72 coolant.
9. To develop empirical correlations in the transient regimes.

10. To study the effects of heat flux, coolant flow rates and chip numbers in the transient regimes.
11. To study the effects of protrusions on the steady-state forced convection and time dependent pump-off and pump-on transient heat transfer using water as coolant.
12. To establish the three-dimensional numerical simulations for steady-state and pump-off transient heat transfer solutions to verify the experimental results.
13. To obtain the numerical correlations in the steady-state and pump-off transient heat transfer regimes.

1.6 Original Contributions

The following items are the original contributions stemmed from the thesis.

- Steady-state heat transfer study from discrete heat sources in vertical rectangular channel under natural and mixed convection conditions using water as coolant.
- The transient heat transfer study from discrete heat sources in vertical rectangular channel under power-on, pump-off, pump-on, power-off and power-on transient natural convection conditions using water and FC-72 as coolant.
- The study of the effects of heat flux, coolant flow rates, chip numbers in the steady-state natural and mixed convection and transient heat transfer regimes.
- The empirical correlations for steady-state mixed and natural convection heat transfer for water coolant.
- The empirical correlations for power-on, pump-off, pump-on, power-off and power-on transient natural convection heat transfer for water-cooling.

- The empirical correlations for pump-off and power-on transient natural convection heat transfer for FC-72 cooling.
- The study of the effects of heater protrusions ($B = 1, 2$ mm) in the pump-off and pump-on transient regimes for water-cooling.
- Comparisons of the heat transfer behaviour of water and FC-72, during the power-on and transient natural convection heat transfer.

1.7 Organization of the Thesis

Followings are the out line of the thesis.

- In chapter 1, the introduction of the research is reported. This chapter elucidates the motivations, scopes, objectives and originality of the present research for both steady-state and transient heat transfer.
- In chapter 2, a comprehensive literature survey on steady-state forced, natural and mixed convection heat transfers are reported. The survey also covers the transient heat transfer for various systems. The theory and correlations available in the literature are provided the outline to convey the present research.
- In chapter 3, the detailed study of experimental apparatus, procedures and instrumentations are described. The range of flow conditions, the analyses of experimental data and an overall error analyses are stated for both steady-state and transient heat transfer.
- In chapter 4, the experimental results are investigated for flush-mounted and protruded heat sources for steady-state forced, natural and mixed convection heat transfer

using water as coolant. The effects of heat flux, coolant flow rates, chip numbers are investigated and empirical correlations are reported.

- In chapter 5, the transient experimental results are investigated for flush-mounted and protruded heat sources using water and FC-72 as coolant. The effects of heat flux, coolant flow rates, chip numbers are investigated. The empirical correlations are developed in the power-on, pump-off, pump-on, power-off and power-on transient natural convection heat transfer regimes for water cooling and pump-off and power-on transient natural convection heat transfer for FC-72 cooling. The comparison of the heat transfer behaviour of the four chips between water and FC-72 cooling are also investigated.

- In chapter 6, the three-dimensional numerical analyses are reported to verify the steady-state and pump-off transient experimental results. The conjugate heat transfer that couples the heat transfer in the solid and fluid are also investigated.

- In chapter 7, conclusions and recommendation for future work are presented.

Chapter 2 Literature Review

2.1 Physical Configuration and Notation

The configuration and notation of an array of discrete heaters is illustrated in Fig. 2.1 for the purpose of literature review. The rows are numbered from the upward flow, along the x - direction. The columns are numbered along the z -direction as shown. For protruded heaters, the protruding height is B in y -direction. The heaters are mounted on a substrate or non-conducting wall.

Fig. 2.1 Notation for an array of heated discrete heaters

The U represents the flow velocity of the coolant in x - direction and characteristic lengths are indicated. The channel has width W , height H , length L , and the spacings between heaters are same, S , in the x and z directions. Thermal conditions are influenced by the existence of multiple length scales of the channel. Relevant length scales are associated

with the length, ℓ and the channel hydraulic diameter, D . The heat transfer from one chip may be influenced by its neighbours, as when a chip is located in the thermal boundary layer generated by upstream heat sources. Heat transfer from the chips may also be influenced by the thermal conductivity of the substrate. Descriptions of test setup and flow channel for present experiment is given in Chapter 3.

2.2 Review of Steady-state Heat Transfer

2.2.1 Studies on Forced Convection

Studies based on flush-mounted discrete heat sources have been made by Xu et al. (1996) since 1994. Experimental and theoretical analyses of the boundary layer development and the heat transfer coefficients in various thermally simulated electronic components were studied for forced convection in a channel, having 34×34 square heated elements by Hachohen et al. (1995). The local heat transfer coefficient at a constant flow velocity along a plate with unheated length; x_o is calculated and correlated as (Kays and Crawford 1980):

$$\bar{h} = \frac{1}{\ell} \int_{x_o}^{x_o+\ell} \frac{kNu_x}{x} dx, \quad (2.1)$$

where the local Nu_x at a constant flow velocity along a plate with unheated starting length, x_o is calculated as

$$Nu_x = 0.332 Re_x^{\frac{1}{2}} Pr^{\frac{1}{3}} \left[1 - \left(\frac{x_o}{x} \right)^{\frac{3}{4}} \right]^{\frac{1}{3}}, \text{ for laminar flow,} \quad (2.2)$$

and,
$$Nu_x = 0.0287 Re_x^{0.8} Pr^{0.6} \left[1 - \left(\frac{x_o}{x} \right)^{\frac{9}{10}} \right]^{\frac{1}{9}}, \text{ for turbulent flow. (2.3)}$$

The measurements of the average heat transfer coefficient for the first through the fourth rows are shown in Fig. 2.2. It is observed a little difference between constant heat flux and constant temperature results.

(a)

Fig. 2.2. Measurements of the average heat transfer coefficient in (a) laminar regime, (b) turbulent regime for the first through fourth rows, for the three element heights (Hacohen et al. 1995).

Sugavanam and Ortega (1995) investigated the forced convection heat transfer from a uniform flush heat source located on a two-dimensional conducting substrate. The Re_H ranges from 100 to 1000, corresponding to air velocities of 1 m/s. Numerical solutions were obtained for flow conditions varying from slug flow to hydrodynamically fully-developed flow. It was concluded that for any developing flow domain, the value of $\overline{Nu}_\ell$ was bounded by the slug flow limit and the fully-developed flow limit, at a given value of Re_H . The developed correlation is

$$\overline{Nu}_\ell = 1.833(Pe)^{0.37} \left(\frac{k_s}{k_f} \right)^{-0.0754} \quad (2.4)$$

Olivos and Majumdar (1995) developed a two-dimensional computational model to simulate the fluid flow and heat transfer of electronic blocks in a laminar forced convection cooling channel. The range of Reynolds numbers was 100-1000.

Morris and Garimella (1997) developed an overall empirical correlation for a wide range of fluid flow rates, and geometric configurations in single-phase forced convection heat transfer from an in-line arrays of three-dimensional heated elements in channel. The correlation equation is

$$Nu_\ell = 0.158 Re_\ell^{0.655} Pr^{0.5} \left(\frac{\mu_i}{\mu_H} \right)^{0.13} \left(\frac{S}{H} \right)^{0.125} \left(\frac{H-B}{\ell} \right)^{0.05}, \quad (2.5)$$

which is valid for $610 < Re_\ell < 68580$, $0.71 < Pr < 25.2$, $0.025 < S/H < 3.22$ and $0.06 < [(H-B)/\ell] < 3.87$. A separate correlation proposed for the air data is

$$Nu_\ell = 0.239 Re_\ell^{0.655} Pr^{0.5} \left(\frac{\mu_i}{\mu_H} \right)^{0.13} \left(\frac{S}{H} \right)^{0.18} \left(\frac{H-B}{\ell} \right)^{0.05}, \quad (2.6)$$

which is valid for $1267 < Re_\ell < 31250$, $Pr \approx 0.71$, $0.025 < S/H < 3.22$ and $0.06 < [(H-B)/\ell] < 3.0$. The comparison of the correlation Eqs. (2.5) and (2.6) with predictions and data in the literature are shown in Figs. 2.3(a) and 2.3(b) for liquids and air respectively. The data from the experiments were correlated for a wide range of flow rates, fluids and geometric configurations. All the liquid cooling data were well correlated by the proposed correlation.

Fig. 2.3. Comparison of the correlation Eqs. (2.5) and (2.6), with predictions and experiments (a) for liquids, (b) for air (Morris & Garimella 1997).

In an adiabatic channel, Chen and Wang (1998) studied experimentally and numerically forced convection heat transfer from heated blocks in tandem. The Nusselt number

distributions showed good agreement on all faces of the blocks in the laminar flow and in the turbulent flow cases. The flow pulsation induced at the channel inlet was studied by Kim et al. (1998) to enhance heat transport in thermal system. The transitional characteristics of fluid flow and forced convective heat transfer in the periodic fully-developed region of the flow over a three-dimensional array of heated blocks in a parallel-plate duct were studied numerically and experimentally by Asako et al. (1999). The effect of turbulence on the friction factor and the Nusselt number appears in the range of the value of $Re_H > 700$ and $Re_H > 500$ for the cases of $H/\ell = 1/4$ and $5/8$, respectively. However the transition of Reynolds number based on flow visualization are about 1000 and 6000 for the cases of $H/\ell = 1/4$ and $5/8$, respectively.

Young and Vafai (1999) performed an experimental investigation to determine the thermal effects of an array of heated two-dimensional obstacles in an airflow horizontal channel. The mean heat transfer from individual protruding heat sources in the channel was correlated with $B/\ell = 0.89$, $H/\ell = 1.57$ and $q'' = 8500 \text{ W/m}^2$

$$\overline{Nu_H} = 0.4472 Re_D^{0.5618} \quad (800 \leq Re_D \leq 13000). \quad (2.7)$$

Xu et al. (1998) studied numerically forced convection heat transfer between two parallel plates with flush-mounted discrete heat sources, using the high value of Re_t . Results for water and FC-72 over a range of Re_t from 10^4 to 1.5×10^5 suggested that the heat transfer process near the wall of the heat sources are having similar single-phase heat transfer coefficient.

Experiments were performed by Tso et al. (1999) using water to determine the single-phase forced convection heat transfer from an in-line four simulated flush-mounted electronic chips to one of a vertical rectangular channel. The results were presented for various channel heights, heat fluxes within the range of Re_ℓ from 6×10^2 to 8×10^4 . Three distinct regimes were identified as in Fig. 2.4(a), indicating a transition from laminar to turbulent for flush-mounted discrete heat sources. Results indicated that the heat transfer coefficient is affected strongly by the number of chips and weakly by the channel height. The proposed overall correlation is in the form

$$Nu_\ell = 2.16(Pe_\ell)^{\frac{1}{3}}. \quad (2.8)$$

Several air-cooling studies in the literature are shown in Figs. 2.4(b) and 2.4(c) for laminar and turbulent flow respectively.

Fig. 2.4. (a) The value of Nu_ℓ as function of the value of Re_ℓ , (b) for laminar flow, (c) for turbulent flow (Tso et al. 1999)

The proposed general correlation was proposed as

$$Nu_\ell = C Re_\ell^m Pr^{0.38} \left(\frac{\mu_t}{\mu_{wall}} \right)^{0.11}, \quad (2.9)$$

where the value of C is 1.42 and m is 0.37 for laminar flow, and the value of C is 0.17 and m is 0.65 for turbulent flow. A value of 0.38 was used as the exponent of Pr

(Incropera et al. 1986). The value of Nu_ℓ was also correlated with the values of Re_ℓ and Pr is shown in Fig. 2.5.

Fig. 2.5. Correlation of data for the baseline configuration (Tso et al. 1999)

Incropera (1999) investigated heat transfer behaviour of an array of 3×3 protruding heaters in a rectangular channel for water and air cooling. The data acquired by Gudapati (1993) and Garimella and Schlitz (1995), as well as results obtained by Garimella and Eibeck (1990) for water and other investigators for air, were collected by Morris and Garimella (1996) and correlated as

$$\overline{Nu_{\ell,ad}} = 0.158 Re_\ell^{0.655} Pr^{0.5} \left(\frac{\mu_i}{\mu_{wall}} \right)^{0.13} \left(\frac{S}{H} \right)^{0.125} \left(\frac{H-B}{\ell} \right)^{0.05}, \quad (2.10)$$

where the ranges of the data are $610 < Re_\ell < 69600$, $0.71 < Pr < 25.2$, $0.025 < S/H < 3.29$, and $0.06 < (H - B)/\ell < 3.87$. Here, the Reynolds number is based on the mean velocity and the Nusselt number is defined in terms of an adiabatic heat transfer coefficient,

$$\overline{h_{ad}} = \frac{q}{A_{ex}(T_{wall} - T_{ad})}, \quad (2.11)$$

where T_{ad} is the reference temperature. In the absence of specific data for T_{ad} , it may be approximated by the mixed mean temperature of the upstream fluid. The data for air, water and FC-77 were correlated by Eq. (2.10).

Tou et al. (1998) performed an experiment for FC-72 flowing in a vertical channel of $W = 20$ mm, with four in-line simulated chips of $\ell = 10$ mm, $S = 10$ mm and $H = 10$ mm. Protrusions of the heater, $B = 0$ and 2 mm were considered for Re_ℓ in the range $2200 < Re_\ell < 6.3 \times 10^4$. The mean velocity was determined from the actual cross-sectional area of the flow, $U = Q/(WH - B\ell)$. For thermally developing and hydrodynamically developed flow the correlations are

$$Nu_\ell = 0.695 Re_\ell^{0.544} Pr^{0.33}, \quad (2.12)$$

and
$$Nu_\ell = 0.367 Re_\ell^{0.609} Pr^{0.33}. \quad (2.13)$$

It was found that the value of Nu_ℓ decreased by approximately 40% from the first to the second row. The results for the second through the fourth row were also correlated by Eq. (2.13), that was in good agreement with the results of Gersey and Mudawar (1993). Figure 2.6 shows the value of Nu_ℓ as a function of the value of Re_ℓ for protruding heat sources.

Fig. 2.6 Nu_ℓ variations with Re_ℓ for protruding chips (Tou et al. 1998)

The mean velocity was determined from the actual cross-sectional area of the flow, $U = \dot{m} / \rho(WH - B\ell)$. The data were correlated for the first chip with $B = 0$ as

$$\overline{Nu}_\ell = C Re_\ell^m Pr^{0.33}, \quad (2.14)$$

where $C = 0.397$ and $m = 0.617$. The Nusselt number decreased from the first to the second row and the fully-developed condition being reached after the second row. The value of $\overline{Nu}_\ell$ for the first row exceeded the last row by approximately 22%. For $B = 2$ mm, $\overline{Nu}_\ell$ decreased by approximately 40% from the first to the second row. The results for the second through the fourth row were correlated by Eq. (2.14), with $C = 0.367$ and $m = 0.609$. The correlations were in good agreement with that of Gersey and Mudawar (1993).

Arrays of protruding 4×3 elements with specified heat input was investigated by Gupta and Jaluria (1998). The average of the heat transfer rates from the three elements in each

row was employed and observed the two distinct zones, one for smaller value of Re_H and the other for larger value of Re_H values. For the value of Re_H less than around 4000, the heat transfer increased from the first row to the second one, and then decreased progressively, exhibiting a maximum at the second row. At higher value of Re_H , Nu_H decreased monotonically from the first row to the last. The value of Nu_H for the first row were around 7.6% lower than those for the second row at the lowest value of Re_H , and 7.3% higher at the highest value of Re_H . However, a strong three-dimensional flow interaction due to higher value of Re_H was taken place from the top and the sides of the first row elements. This increases the turbulence and the heat transfer rate. On the other hand, at higher velocities the small stagnation region also led to enhance the heat transfer in the first row. The heat transfer decreases monotonically because of the development of thermal boundary layer. Finally made an attempt to combine the heat transfer results for all the channel heights and spacing between the elements in a single correlation. Figures 2.7(a) and 2.7(b) show the correlation, along with the experimental data. Least square regression techniques (Jaluria 1988) were used to obtain a best fit to the experimental data. The data for the first row and the remaining rows were correlated respectively by Eqs (2.15) and (2.16) where the standard deviations for the value of C and m were 0.98% and 4% respectively. Both the correlations are valid for $2500 \leq Re \leq 9000$, $2 \leq (S/B) \leq 3$ and $3 \leq (H/B) \leq 4$.

$$Nu_H = 0.035 Re_H^{0.98}, \quad (2.15)$$

and

$$Nu_H = 0.15 Re_H^{0.83} \left(\frac{S}{H} \right)^{-0.14} \left(\frac{B}{H} \right)^{0.19}. \quad (2.16)$$

(a)

(b)

Fig. 2.7. (a) Correlations of heat transfer data for the first row of the array, (b) Correlation of data for the rows downstream of the first one for the channel heights, stream-wise spacing, and applied heat input (Gupta & Jaluria 1998).

The results of water-cooling were compared with the results of air-cooling (McEntire and Webb 1990, Sparrow et al. 1982) and the results of water-cooling (Garimella and Eibeck

1990). The factor $Pr^{0.35}$ was used to scale the air-cooled array Nusselt numbers for comparison, based on the recommendations of Kelecy et al. (1987).

An extensive presentation and evaluation of the mean Nusselt numbers along the exposed faces of all obstacles were documented by Young and Vafai (1998a). The mean Nusselt number was found to reach the 5% and 10% difference levels, referenced to the nine obstacles, at the eighth and seventh obstacles, respectively in the array.

Leung et al. (2000) studied numerically the steady-state laminar forced convection air-cooling in a horizontally oriented printed circuit board (PCB). The dependence of temperature fields on Reynolds number, obstacles sizes, and the separation between two obstacles was documented and showed that these parameters have a significant effect on the flow and temperature fields.

2.2.2 Studies on Natural Convection

For natural convection, Hacohen et al. (1995) estimated the increased heat transfer coefficient from the heated elements caused by the protrusion, between two parallel plates horizontal channel. For the flow between two asymmetric isothermal parallel plates, Kraus and Bar-Cohen (1983) observed that the upper bound of heat transfer coefficient was taken as an isolated plate and the lower bound was taken as the heat transfer coefficient for the composite relation. The following equation was also proposed

$$Nu_H = \left[\frac{144}{(Ra_H)^2} + \frac{2.873}{\sqrt{(Ra_H)}} \right]^{-\frac{1}{2}}. \quad (2.17)$$

This composite relation for the vertical channel formed by the heated board, the isothermal plate, and the front wall, which having insulated plate, is formed by two limits: (a) In fully-developed flow, the value of $Nu_H = 1/(12Ra_H)$ for $Ra_H \rightarrow 0$, (b) $Nu_H = 0.59Ra_H$ for $Ra_H \rightarrow \infty$ for an isolated plate. This is shown in Fig. 2.8 with the comparison of measurements with theory. From these results it is seen that for a large channel distance between the plate and the wall ($Ra_H > 100$), the value of Nu_H falls close to the smooth isolated plate case, since for large value of Ra_H and for small velocities no separation of the preceding boundary layers takes place. The measured value of Nu_W was calculated by Eq. (2.17). From these results it is seen that for a large channel distance between the plate and the wall ($Ra_W > 100$), the value of Nu_W falls close to the smooth isolated plate case, since for large value of Ra_W and for small velocities no separation of the preceding boundary layers takes place.

Fig. 2.8. Variation of Nu_H with the lower and upper limits, versus Ra' for an element height of 2 mm (Hacohen et al. 1995)

The steady-state natural convective heat transfer and pressure loss in a horizontal rectangular channel with discrete flush-mounted and protruded heat sources was studied experimentally by Wang and Vafai (1999). Six protruding obstacle heights, which represent the dimensionless protrusion of $0 \leq B/H \leq 0.805$ were studied at a fixed channel height of 39.75 mm. It was observed that at low Reynolds number, for flush-mounted case natural convection, heat transfer was dominant but for protruding heat sources, heat transfer was dominated both by natural and forced convection.

Fig. 2.9. $\overline{Nu}_t$ distribution for all the heaters (Wang & Vafai 1999)

When both natural and forced convection are significant, $\overline{Nu}_t / Re_D^{0.5}$ is usually a function of Gr_H / Re_D^2 . Figure 2.9 shows a plot of $\overline{Nu}_t / Re_D^{0.5}$ versus Gr_H / Re_D^2 , presented by Eq. (2.18) with an average error of 4.7%. It is seen that $\overline{Nu}_t / Re_D^{0.5}$ increases as Gr_H / Re_D^2 is increased. Linear regression of the experimental data yields

$$\frac{\overline{Nu}_\ell}{Re_D^{0.5}} = 0.58 \left(\frac{Gr_H}{Re_D^2} \right)^{0.13} \quad \text{for } \frac{Gr_H}{Re_D^2} = 0.12 - 1.7, \quad (2.18)$$

or

$$\overline{Nu}_\ell = 0.58 (Gr_H)^{0.13} Re_D^{0.24} \quad \text{for } \frac{Gr_H}{Re_D^2} = 0.12 - 1.7. \quad (2.19)$$

An experimental investigation for air-cooling was carried out with the printed circuit board (PCB) in both horizontal and vertical position having different protruding height of the chips by Leung and Kang (1998). For horizontal PCB assembly at $H/B = 3$ and $\ell/B = 4$, the steady-state convective heat transfer was greatest at the first rib, and thereafter decreased for each subsequent rib row to reach row independent and fully-developed condition. However, an increase in heat transfer coefficient occurs at the last two row ribs, which was due to the exit effect. The correlations suggested for the horizontal and vertical assembly, respectively, within the range of $460 \leq Re_C \leq 2300$, $2 \leq H/B \leq 10$, and $3 \leq \ell/B \leq 4$:

$$Nu_C = 0.40 Re_C^{0.24} \left(\frac{H}{B} \right)^{0.68} \left(\frac{\ell}{B} \right)^{0.80}, \quad (2.20)$$

and

$$Nu_C = 0.12 Re_C^{0.29} \left(\frac{H}{B} \right)^{0.78} \left(\frac{\ell}{B} \right)^{1.35}. \quad (2.21)$$

The coefficients and power indicated in Eqs. (2.21) and (2.22) were determined by the least-square method, the maximum deviations from the best fit lines were within -20.3% to 25.7% . The average heat transfer coefficients between the PCB surface and the air flow at different H/B ratios at $\ell/B = 3$ and 4 are shown in Figs. 2.10(a) and 2.10(b), for horizontal and vertical PCB assembly, respectively.

(a)

(b)

Fig. 2.10. Comparison of the average h at $\ell/B = 3$ and 4 (a) for horizontal, (b) for vertical PCB assembly (Leung & Kang 1998).

Wang and Vafai (1999) found that the temperatures on the heater surfaces were smaller than those for the flush-mounted case for the same input heat and with increasing the

protruding height, the temperature decreased due to higher heat transfer coefficients. Forced convection became dominated mode of heat transfer and the natural convection may be neglected at the increase of protruding height of the heat source when $Re_D > 1,500$. The value of $\overline{Nu}_\ell$ increases with increasing the value of Re_D , for each value of B/H ratio are shown in Fig. 2.11 for four heaters. An exception of this trend occurred when the value of Re_D was around 1200. At this Re_D , the value of $\overline{Nu}_\ell$ is larger than that of the value of Re_D of 1500 for all protruding heat source cases. This suggests the critical value of Re_D , at which the transition from laminar flow to turbulent flow occurs in the range from 1200 to 1600 in the rectangular channel with discrete protruding heat source. Finally proposed the following form of a general overall correlation:

$$\overline{Nu}_\ell = C(Re_D)^m \left(\frac{B}{H} \right)^n, \quad (2.22)$$

where the value of C is 1.46, m is 0.53 and n is 0.54 with an average error of about 10%. The value of $\overline{Nu}_\ell$ for forced convection for the protruding cases were obtained from Eq. (2.22) where the value of the exponent m is in good agreement with the results of Roeller et al. (1989), McEntire and Webb (1990), Lehmann and Wirtz (1989) and Young and Vafai (1998b).

Fig. 2.11. Average Nusselt number versus Reynolds number for: (a) heater 1, (b) heater 2, (c) heater 3, (d) heater 4 (Wang & Vafai 1999).

Numerical investigations for laminar natural convection with air in a vertical channel with isothermal obstructions was carried out by Cammarata et al. (2000). The two-dimensional results, valid for low values of the modified Rayleigh number (in fully-developed regime) and moderate values (in developing flow regime), was given as

$$\overline{Nu} = \frac{Ra^*}{24} \left(1 - e^{-\frac{35}{Ra^*}} \right)^{\frac{3}{4}}. \quad (2.23)$$

The results for the unobstructed channel and for different aspect ratios are shown in Fig. 2.12(a), and the variation of the value of $\overline{Nu}$ with the value of Ra^* for unobstructed and obstructed channels are shown in Fig. 2.12(b). For different positions of the obstructions, the value of $\overline{Nu}$ decreases as the distance of the obstructions from the inlet increases for low values of Ra^* .

Fig. 2.12. (a) Comparison between the single plate regime (Elenbaas 1942) and the results of the numerical simulations in unobstructed channels for various aspect ratios, $Pr = 0.71$, (b) Variation of the value of $\overline{Nu}$ versus the value of Ra^* for unobstructed and obstructed channels for three different locations of the obstructions (Cammarata et al. 2000).

MacGregor and Emery (1969) investigated experimentally and two-dimensional numerically the natural convection heat transfer from vertical plane layers, under constant heat flux wall boundary condition. The effects of Prandtl number, Grashof number and Rayleigh number, aspect ratio and variable properties were described and a significant reduction of the Nusselt number with increasing aspect ratio was noted. The Eqs. (2.24) and (2.25) are experimental results for constant wall heat flux condition for turbulent and laminar flow regime, respectively. Equations (2.26) and (2.27) are for numerical results for constant wall heat flux and constant wall temperature conditions, respectively, for laminar flow regime.

$$Nu_H = 0.046 Ra_H^{1/3} \quad \left\{ \begin{array}{l} 10^6 < Ra_H < 10^9 \\ 1 < Pr < 20 \\ 1 < \ell/H < 40 \end{array} \right\}, \quad (2.24)$$

$$Nu_H = 0.42 Ra_H^{0.25} Pr^{0.012} (\ell/H)^{-0.3} \quad \left\{ \begin{array}{l} 10^4 < Ra_H < 10^7 \\ 1 < Pr < 20000 \\ 10 < \ell/H < 40 \end{array} \right\}, \quad (2.25)$$

$$Nu_H = 0.38 (\ell/H)^{-0.25} Ra_H^{0.25} \quad \{10^4 < Ra_H\}, \quad (2.26)$$

and
$$Nu_H = 0.25 (\ell/H)^{-0.25} Ra_H^{0.25} \quad \{10^{4.5} < Ra_H\}. \quad (2.27)$$

Daric et al. (1998) investigated two-dimensional numerical natural convection heat transfer in an asymmetrically heated vertical channel for fluids with $Pr = 7$. The average Nusselt number variation was computed for various aspect ratios at the hot and cold vertical walls and the heat transfer behavior was predicted. Correlations were proposed for a channel with aspect ratio (ℓ/H) of 5 as

$$Nu_\ell = 0.146 Ra_\ell^{0.188}, \text{ for closed channel,} \quad (2.28)$$

and
$$Nu_\ell = 0.132 Ra_\ell^{0.169}, \text{ for open channel.} \quad (2.29)$$

2.2.3 Studies on Mixed Convection

Very limited work has been reported so far on the mixed convection on discrete heat sources in channel flow. The effect of buoyancy on the flow and heat transfer of a shrouded arrays of rectangular blocks was reported by Braaten and Patankar (1985), and showed the combined effect of buoyancy and highly conductive shrouds that lead to enhance the heat transfer as shown in Fig. 2.13.

Fig. 2.13. Effect of buoyancy on $\overline{Nu}_B$ (Braaten & Patankar 1985)

From figure it is seen that Cases 1-4 differed in the orientation of the heated blocks and in the thermal boundary conditions imposed on the shrouds, but they all were for $Pr = 0.7$, and Case 5 was for $Pr = 3.5$. In general, the value of $\overline{Nu}_B$ increases with the value of Ra_B and influenced by the thermal boundary conditions imposed on the shrouds. It is observed that the effect of Pr on the value of $\overline{Nu}_B$ is seen to be quite small, as comparison of the curves for Cases 1 and 5 reveals.

Türkoğlu and Yücel (1995) examined numerically the two-dimensional laminar mixed convection air-cooling in a vertical channel with single discrete heat source. The variations of the mean value of Nu_t with Gr_t at different value of Re_t are shown in Fig.

2.14(a). It is observed that at high Re_ℓ ($Re_\ell \geq 50$), the value of Nu_ℓ is almost independent of Gr_ℓ and at low Re_ℓ ($Re_\ell < 50$), the value of Nu_ℓ increases with increasing Gr_ℓ . At high Gr_ℓ heat transfer is dominated by natural convection and at higher value of Re_ℓ heat transfer is mainly controlled by forced convection. The variations of the value of Nu_ℓ with the width of the channel at different Re_ℓ are shown in Fig. 2.14(b). It is seen that the value of Nu_ℓ does not change with the channel gap and it reaches an asymptotic value with the increasing Re_ℓ . As the value of Re_ℓ increases, the recirculating cell diminishes and the fluid particle flows along the streamlines. However, it implies that at low value of Re_ℓ the natural convection is the dominated flow mode, but at higher value of Re_ℓ , the flow is dominated by the forced convection.

Fig. 2.14. (a) Variation of the mean value of Nu_t with the value of Gr_t for $x/B = 7.5$ at different Re_t , (b) Variation of the mean value of Nu_t with the ratio of H/l for $Gr_t = 5 \times 10^4$ at different value of Re_t and heater centre location of $x/B = 4$ (Türkoğlu & Yücel 1995).

Meinders et al. (1998) investigated experimentally the local convective heat transfer coefficient from a wall-mounted single array (6×1) of cubical protrusion elements along a wall of a horizontal wind tunnel. The average heat transfer coefficient was found for the first three protrusions in the array. After the third element, the flow becomes gradually fully-developed and no notable variation in the convective heat transfer was observed due to the adiabatic heat transfer. The average convective heat transfer coefficient for the mixed convection case, $Re_B = 2086$, differs from the others. In particular, for the cubes further downstream in the array, the heat transfer benefits from the buoyancy-induced vertical motion, which results in an enhancement of the heat transfer coefficient. The heat transfer coefficients averaged over all five cube faces are shown in Fig. 2.15.

Fig. 2.15. Variation of cube averaged h with location of the cube at different Re_B (Meinders et al. 1998).

Busedra and Soliman (2000) investigated experimentally the combined free and forced convection in a semicircular duct with upward and downward inclinations within $\pm 20^\circ$ for laminar water flow and uniform axial heat input. The values of Nusselt number for the horizontal and upward inclinations were found to increase with Grashof number and weakly dependent on Reynolds number.

Du et al. (1998) studied numerically mixed convection heat transfer in an open-ended vertical channel with protruding heaters. Flow and heat transfer rates were examined for ranges of the value of Ra_w from 10^3 to 10^7 , and the value of Re_w from 10 to 200 with the

working fluid of air ($Pr = 0.72$). The variation of dimensionless temperature as a function of the value of Re_W for $Ra_W = 10^5$ and 10^6 is shown in Fig. 2.16(a).

(a)

(b)

Fig. 2.16. Temperature at the electronic components as a function of the value of Re_W and Ra_W for the base case with $A = 4$, (a) Temperature θ as a function of the value of Re_W for $Ra_W = 10^5$, (b) Temperature θ as a function of Ra_W for $Re_W = 50$ (Du et al. 1998).

It is found that cooling is generally more effective with increasing the value of Ra_W . Figure 2.16(b) shows the temperature variation as a function of the value of Ra_W for $Re_W = 50$ and 100. It is observed that at higher value of Re_W , cooling by natural convection has little effect on the temperature of the components.

Wu and Perng (1999) investigated numerically the heat transfer enhancement of mixed convective heat transfer from heated block in a horizontal channel. The variations of time mean Nusselt number along the block surface is presented in Fig. 2.17 for given Gr_ℓ/Re_ℓ^2 and at $Re_\ell = 400$. It is seen that the local Nusselt number increases with an increase in the Gr_ℓ/Re_ℓ^2 values. The maximum local Nusselt number for a given block occurs at the front corner and the minimum value occurs at the groove between two blocks. The influence of mixed convection is prominent along the vertical surfaces of the blocks. Similar observations have been seen in an earlier study (Kim et al. 1992).

Fig. 2.17 Time-mean Nusselt number variations along the block surface (Wu & Perng 1999)

Tsay (1999) studied numerically the unsteady conjugated mixed convective heat transfer in a vertical channel where one wall suddenly subjected to either isothermal or isoflux heated sources. It reveals that the discrete heating can cause the heat transfer direction conversely from fluid (air) to the heated wall during the transient period. This phenomenon cannot be observed for the situation with the plate wall heated throughout.

The effects of buoyancy parameter (Gr_W/Re_W^2) on the flow and temperature fields are investigated numerically for two-dimensional laminar mixed, forced and free convection in a partitioned enclosure by Sathyamurthy et al. (1992).

Lloyd and Sparrow (1970) investigated numerically the combined forced and free convection adjacent to an isothermal vertical plate for liquid metals, gases and ordinary liquids. The buoyancy affected profiles for $Pr = 100$ are similar to those for forced convection whereas for $Pr = 0.72$ the presence of buoyancy results in a relatively sharp rise in the profile near the wall.

In mixed convection condition there is no correlation reported in the literature; only Wang and Vafai (1999) proposed the correlation Eq. (2.18) for predicting the average Nusselt number for mixed convection with discrete flush-mounted heat source. It was observed that natural convection plays an important role in the total heat transfer process that leads to a possibility of mixed convection behavior.

2.3 Review of Transient Heat Transfer

Transient problem is pertinent to the design and performance of the cooling of electronic circuits and the packaging of heat dissipation components. Transient can be initiated by the system start-up, shutdown or inadvertent loss of liquid pumping power in the system. Although many works have been done on the steady-state direct liquid cooling channel flow for electronic systems, the literature on transient direct liquid cooling channel flow for discrete heat sources is still very limited. In fact, there are few related investigations found in this area of electronics cooling. For example, some related studies have been performed on the transient response due to changing wall heat flux, wall surface temperature and flow velocity, but none on discrete heat sources. Nevertheless, the current state-of-art in transient heat transfer is presented below.

Shim and Hyun (1997) analysed numerically to describe the time-dependent adjustment of natural convection in a square cavity with internal heat generation and observed that when the impact of internal heat generation was dominant, the flow adjustment was achieved over the time.

Hyun and Lee (1989) investigated the two-dimensional transient natural convection heat transfer in a square cavity, where the opposing sidewalls were subjected to an arbitrary time variations in temperature. It was observed that the influence of the thermal boundary conditions on the horizontal surfaces appears to have a negligible influence on time. Abu-Hijleh et al. (2003) investigated numerically the transient conjugate forced convection in

channel flow and observed the significant effects of Peclet number and Biot number on Fourier number.

Gebhart (1961) studied numerically the transient natural convection heat transfer for vertical elements and described the general equations. The equations are related to the average surface temperature and time for heating or cooling of vertical elements having arbitrary thermal capacity.

Chen (2003) examined numerically the transient natural convection to show the effects of thermal wall boundary between concentric and vertically eccentric spheres, and proposed the correlation equation in relations with the values of $\overline{Nu}_b$ and Ra_b^* . The value of $\overline{Nu}_b$ for diameter ratio of 2.0 can be presented by the equation

$$\overline{Nu}_b = C(Ra_b^*)^m \quad (2.30)$$

For three different configurations, the values of C and m are determined. The values of C and m are selected as 0.6714 and 0.1814; 0.8033 and 0.1588; 0.9496 and 0.1309 in the Ra_b^* range of 10^3 - 10^5 , 10^3 - 5×10^5 and 10^3 - 10^5 respectively.

Sucec (2002) investigated numerically the transient forced convection in parallel plate duct when there was sinusoidal variation with axial position in the duct wall and observed the transient Nusselt number depended on time. The solutions were found for wall temperature; surface heat flux and fluid bulk mean temperature as a function of position and time.

Seo and Parameswaran (2002) studied numerically to determine the influence of buoyancy on unsteady mixed convection for various Richardson number. Various flow separation patterns as well as vortex shedding were observed and showed vortex shedding enhanced the heat transfer.

Siegel and Sparrow (1959) performed an analysis on transient laminar heat transfer in the thermal entrance region of a flat duct whose bounding surfaces are subjected to an arbitrary time variation of heat flux. The system may be either in an already established steady-state heat transfer situation, or the fluid and the duct walls may be at the same uniform temperature. The velocity distribution in the flow is taken to be fully-developed and unchanging with time. The thermal response to a unit step function change in the wall heat flux was found by using an integral formulation of the energy solved by the method of characteristics. Then, using the linearity of the energy equation, step function results are generalized to the case of arbitrary time-dependent conditions. The final results were expressed in the form of integrals, which may be easily evaluated for a particular application

An analysis was carried out by Siegel (1959) on laminar slug-flow forced convection heat transfer in a parallel-plate channel where the walls were given a step-change in the heat flux and the velocity distribution were assumed to be steady and fully-developed. It was indicated that, at a given place, temperatures change appeared in a pure condition manner until a given time and then the values were stabilized at the steady-state condition. The

abrupt arrival at steady-state is due to the slug flow assumptions. It has been shown that slug-flow assumption did, in fact, reveal the essential physical of the system considered, although the numerical results were somewhat in error.

Perlmutter and Siegel (1961) studied the transient heat transfer with laminar flow between heated (or cooled) parallel plates. The transient process was caused by changing the fluid pumping pressure and the wall heat flux simultaneously. The numerical solution was obtained for both the thermal entrance region and the fully-developed heat transfer region under the assumption of slug flow and negligible wall heat capacity.

Siegel and Perlmutter (1963) analysed numerically the laminar forced convection between two parallel plates for slug flow conditions with wall heating that was a function of axial positions and time. The wall was assumed to be sufficiently thin or highly conducting so that the temperature variation through the wall thickness was neglected. The slug flow assumption was also employed. The method was involved coupling the heat transfer behavior within the fluid to that in the wall solving the governing equations. Results were presented in a closed form and their applications were illustrated by some typical examples.

Kim and Özişik (1987) investigated the transient laminar forced convection in duct with suddenly applied uniform wall heat flux on a parallel plate channel. The flow was thermally developing and hydrodynamically developed with all the physical properties considered constant. It was used a generalized integral transform technique and classical

Laplace transforms to develop a simple low order solution as well as higher order solution. Starting from the inlet, the wall temperature increased monotonically with both time and position along the channel until the location characterizing the beginning of the conduction region was reached. In the conduction region the wall temperature remained invariant with the axial distance, but increased continuously with time. The local Nusselt number, on the contrary, decreased with time, until it reached the steady-state value.

Yang and Zhu (2003) investigated the transient laminar natural convection for water in an inclined parallel plate channel by direct numerical solution. At the bottom wall heated condition, the study reveals that for a plate spacing ratio of less than 1/20 and for the inclined angle ranges from 45° to 90°, the overall Nusselt number can be considered as a monotonic function of the parameter $[Ra(H/L)\sin\beta_i]$. For a spacing ratio of 1/20, the parameter can be used as a suitable monotonic parameter of the average Nusselt number, which is favourably comparable with the experimental correlation of Azevedo and Sparrow (1985), i.e.

$$Nu(H/L) = 0.644 [Ra(H/L)\sin\beta_i]^{0.25} \quad (2.31)$$

The comparison between the calculated results and the experimental results are in good agreement at $t^* = 50$. The behavior of the time oscillation of the overall Nusselt number at different angle and the value of H/L ratio are shown in Figs. 2.18(a) and 2.18(b) when $Ra(H/L)$ at about 2,500 and 20,000 respectively. It is observed that, for larger $Ra(H/L)$, the magnitude of the value of Nu oscillation becomes more significant. However, the overall value of Nu can be expressed as a monotonic function of the parameter $Ra(H/L)\sin\beta_i$.

Fig. 2.18 Variations of overall value of Nu for four different cases when (a) $Ra(H/L) = 2,500$ and (b) $Ra(H/L) = 20,000$ (Yang & Zhu 2003).

2.4 Studies on Protruded Heaters

Many experiments have been carried out with protruding block-like heaters in a duct with air flow, for example, Sparrow et al. (1982), Moffat et al. (1985), Moffat and Ortega (1989), Anderson and Moffat (1990), McEntire and Webb (1990). However, there are

few studies have been extended to liquids. There are hardly any studies on transient heat transfer involving discrete heat sources concerning the effects of heater protrusion.

Experiments were performed to determine the convective heat transfer coefficient for water cooling of in-line and staggered array of 30 heated protruding elements arranged in six rows by Garimella and Eibeck (1990). The channel-height-based Reynolds number ranges from 150 to 5150 with dimensions of the heated element being 25.4 mm $\times$ 25.4 mm $\times$ 10 mm high. The channel height to element height ratio (H/B) was varied from 1.2, 1.9, 2.7 to 3.6. The streamwise spacing to element height ratio (S_x/B) was varied over values of 6.5, 4.3, 2.2, 1.1 and 0. The spanwise spacing to element height ratio (S_y/B) was varied over values of 6.55, 2.2 and 0.5. It was found that the dependence of the heat transfer coefficient on row number decreased, with increasing channel height and decreasing Reynolds number. There was a drop in the heat transfer coefficient as the row number increased, reaching an asymptotic value by the fourth row. However, the heat transfer coefficient can be considered to have reached a row-number-independent, fully-developed value by the fourth row. In air-cooled channels, the attainment of fully-developed conditions in the downstream of three to four rows was also observed by Sparrow et al. (1982) and Moffat et al. (1985). The Nusselt number decreases with increasing H/B and with decreasing Re_H , and approaches an asymptotic value for large H/B . At lower Re_H , the channel height has less influence on the heat transfer coefficient.

Garimella and Eibeck (1991) measured a sharp increase in the turbulent intensity directly above the chip surfaces as a result of the higher shear forces encountered there,

suggesting that increased mixing and enhanced heat transfer may occur above the surface of protruded heat sources compared to flush-mounted heat sources.

Gersey and Mudawar (1993) investigated the effects of chip protrusions on the forced convection heat transfer using FC-72. The multi-chip module used here was a linear array of nine $10\text{ mm} \times 10\text{ mm}$ simulated microelectronic chips that protruded 1 mm into a 20 mm wide side of a rectangular vertical flow channel. The experiments were performed in 5 mm and 2 mm channel gap, and the Reynolds number ranging from 4×10^3 to 2×10^5 . Unlike the flush-mounted single-phase data obtained by Gersey and Mudawar (1992), upstream chips in both the 2 mm and 5 mm channels had slightly higher single-phase heat transfer coefficient than the rest of the chips. They suggested the enhancement in the heat transfer for the upstream chip might be attributed to the rapid acceleration of the fluid. Higher single-phase heat transfer coefficients were measured for the protruded chips as compared to similar flush-mounted chips.

Ramadhyani and Incropera (1988) considered the effect of heat source protrusion for a single, isolated source. Experiments were performed for a single heat source that protruded a distance of 0.5 mm from the substrate, and the results were compared with those for flush-mounted sources. It was observed that the protrusion height enhanced the heat transfer by 30% and 25% for $Re_t \approx 1300$ and $Re_t \approx 3000$ respectively.

Samant and Simon (1989) considered single-phase cooling of a small heater ($0.25\text{ mm} \times 2.0\text{ mm}$) in a rectangular channel with height 2.79 mm and width 10.26 mm. Because of

the short length, the thermal boundary layer was very thin, causing the heat transfer coefficient to be considerably higher than for a computer chip. Results obtained for FC-72 and R-113 in the Reynolds number range $7000 < Re_H < 1.5 \times 10^5$ were correlated by

$$Nu_H = 0.47 Re_H^{0.58} Pr^{0.5}. \quad (2.32)$$

Here it is reported that a heat transfer coefficient of $33000 \text{ W/m}^2\text{K}$ is obtained at a peak heat flux over 200 W/cm^2 and a bulk velocity of 16.5 m/s in R-113 without boiling.

Garimella and Schlitz (1995) focused on the heat transfer from single protruded element with a height of 0.6 mm in an array of 5×5 elements using FC-77 and water. The heat transfer coefficient for a single protrusion was greater than that for flush heat sources. This can be attributed to the perturbation to the flow field caused by the protrusion. The heat transfer from the heated protrusion in the range of $10^4 < Re_\ell < 8 \times 10^4$ was correlated by:

$$Nu_\ell = 0.1495 Re_\ell^{0.661} Pr^{0.5} \left(\frac{\mu_i}{\mu_{wall}} \right)^{0.13} \quad (2.33)$$

However, a combined correlation for the flush and protruding heat sources was also proposed for convenient design predictions:

$$Nu_\ell = 0.0973 Re_\ell^{0.698} Pr^{0.5} \left(\frac{\mu_i}{\mu_{wall}} \right)^{0.13} \quad (2.34)$$

2.5 Studies on Numerical Heat Transfer

Numerical investigations of single-phase forced convection heat transfer of air-cooling of electronic systems have been widely reported. Davalath and Bayazitoglu (1987) studied numerical, the two-dimensional steady laminar heat transfer from a series of parallel plates with three surface mounted heat sources. Culham et al. (1991) conducted a two-dimensional conjugate heat transfer from square flush-mounted heat sources by implementing an iterative scheme in which the thermal boundary conditions were applied on the conduction solution of the boundary layer energy equation. Nigen and Amon (1993) performed two-dimensional numerical modelling to investigate the thermal and fluid behavior for different configurations of surface mounted electronic package. Local heat transfer coefficient were determined and used to calculate the temperature distribution within a surface mounted package. Kim and Anand (1994a) conducted a numerical investigation to analyse two-dimensional laminar heat transfer between a series of parallel plates with surface mounted discrete block heat sources. A numerical investigation was conducted on the laminar heat transfer from single heat source flush-mounted on a two-dimensional conducting substrate by Sugavanam and Ortega (1995).

Much numerical work has been done on turbulent heat transfer as well as laminar heat transfer in air-cooling of electronic system. Agonafer and Moffat (1990), and Kim and Anand (1994b) performed numerical investigations for turbulent flow heat transfer by using two-dimensional, standard form of the k - ϵ turbulence model for Re_H between $5 \times 10^4 \sim 5 \times 10^5$. Asako and Faghri (1991) numerically predicted turbulent flow in channels with three-dimensional heat transfer blocks using the higher Re_H from the form of the k - ϵ

turbulence model for a range of Re_H from $10^4 \sim 10^5$. In addition Asako and Faghri (1995) also performed numerical computation in an array of rectangular heated blocks by using the Lam-Bremhorst low Re_H form of the k - ϵ model, the Re_H ranging from 2000 $\sim$ 5000.

Forced convection heat transfer for hydrodynamically full-developed laminar channel flow with one or two heat sources flush-mounted to one wall of a parallel plate channel had been considered numerically by Ramadhyani et al. (1985). Moffatt et al. (1986) and Incropera et al. (1986) numerically studied the problem of flash-mounted isothermal heat sources embedded in one wall of the horizontal channel with turbulent flow using a relatively simple model: zero equation turbulent model combined with wall function. Mahaney et al. (1990) extended the work of Incropera et al. (1986) to the low Reynolds number regime where mixed convection becomes important. They investigated mixed convection heat transfer for water from a four row in-line array of 12 square heat sources, each 12.7×12.7 mm that are flush-mounted to the lower wall of a horizontal, rectangular channel with 50.8 mm width and 12.7 mm high. Both experiments and three-dimensional computation were performed for Reynolds numbers in range $25 < Re_f < 2500$.

Table 2.1 summarizes the conditions available in the literature, the coolant involves air, and water, FC-72 and fluids having different Pr . The heat transfer results involve the investigations based on three convection modes at different flow and heating conditions.

2.6 Summary and Conclusions

Based on the preceding survey, the following conclusions can be drawn concerning the state of knowledge about single-phase convective modes from discrete heat sources in a flow channel:

1. There are many databases and correlations on single-phase heat transfer in plain heat sources, but they are inconsistent with each other because of different geometric parameters of flow channels, heat sources also their thermophysical properties. Hence, further studies need to be done.
2. Although many experiments have been carried out with protruding heaters in a duct with air flow, there are few studies extended to liquid cooling. It is necessary that effects of protrusion on heat transfer be tested. In general, single-phase heat transfer for protruded chips is expected to be higher than that for similar flush-mounted chips and therefore more attention should be directed towards this aspect. A better understanding on the physical mechanisms for protrusion is required.
3. Hacoheh et al. (1995) provides a simple design tool to evaluate heat transfer coefficients for complicated configurations and to predict the temperature distribution on a PCB by using basic correlation and fundamental physical methods.
4. For any developing flow, Nusselt number was bounded by the slug flow limit and the fully-developed flow limit at a given Reynolds number. The Nusselt number for the slug flow situation was found to be much higher when compared with that of the fully-

developed flow situation at higher Reynolds numbers because of convection dominance, as shown by the numerical result of Sugavanam and Ortega (1995).

5. The average Nusselt number decreases slightly with increasing the ratio of channel height to heater length. The numerical studies for single-phase heat transfer suggest that the wall of the heat source dominates the heat transfer process. Xu et al. (1998) suggested that a narrow channel with small H/ℓ could be used for liquid cooling.

6. For forced convective heat transfer from flush-mounted discrete heat sources, the upstream thermal boundary affects that of downstream, with the decreasing heat transfer coefficients as the chip number increases, but may reach a chip-number-independent, fully-developed value by forth chip, as shown by the result of Tso et al. (1999)

7. Leng et al. (2000) suggested that the flow and temperature fields are strongly depend on Reynolds number, obstacle sizes and the spacing between two obstacles. Under steady-state condition, for the cooling of the both horizontal and vertical simulated printed circuit board (PCB), heat transfer coefficients at the board surface at $\ell/B = 4$ are always greater than those for $\ell/B = 3$ for the same Reynolds number and H/B ratio. It means that a large top surface can be adopted to enhance the heat transfer, as shown by Leng and Kang (1998).

8. For the flush-mounted and protruding discrete heat source case, free convection can play an important role in the total heat transfer process, leading to mixed convection behaviour. Furthermore, the protruding heat source distorts the flow field significantly and cause transition from laminar to turbulent flow at a lower Reynolds number and

enhances the forced convection significantly, as shown by Wang and Vafai (1999), Young and Vafai (1999), Young and Vafai (1998). The differences in heat transfer data between the flush-mounted and protruding chips increases as the Reynolds number increases and the heat transfer by protruding chips always greater than that by flush-mounted chips were suggested by Tou et al. (1998).

9. The use of high Prandtl number fluids is advantageous since the enhancement of heat transfer due to buoyancy is achieved at a smaller penalty in increased pressure drop (Braaten and Patankar 1985)

10. The location of the heater element does not play an important role in heat transfer. For low Reynolds number case heat transfer is mainly controlled by Grashof number and at high Reynolds number heat transfer is mainly controlled by forced convection as suggested by Türkoglu and Yücel (1995). During the transient period discrete heating can cause the heat transfer direction conversely from the fluid to the heated plate (Tsay 1999)

11. The results for natural convection and mixed convection showed that the most effective cooling was always at the component placed at the entrance, followed by that at the middle position and finally least effective cooling was at the top position. This trend seems to be independent of the component size (Du et al. 1998).

For forced convection, the local Nusselt number in various thermally simulated electronic components is determined from Eqs. (2.2) and (2.3) for laminar and turbulent flow respectively on a PCB (Hacohen et al. 1995). For developing flow, the Nusselt number for the slug flow situation is found to be much higher when compared with that of the fully-

developed flow situation at higher Reynolds number (Sugavanam and Ortega 1995) and Eq. (2.4) is proposed to determine the average Nusselt number. For a wide range of fluid flow rates and geometric configurations Eqs. (2.5) and (2.6) are proposed (Morris and Garimella 1997). Correlation Eq. (2.7) is stated to evaluate the average Nusselt number from discrete protruding heat sources in wide range of flow variations (Young and Vafai 1999). For various channel heights, heat fluxes and Reynolds numbers in distinct regimes Eqs. (2.8) and (2.9) are proposed from discrete flush-mounted heat sources (Tso et al. 1999). For thermally developing and hydrodynamically developed flow conditions the correlation Eqs. (2.12) and (2.13) are proposed for flush mounted and protruded heat sources respectively (Tou et al. 1998). However, protruding heat source with specific heat input, Eqs. (2.15) and (2.16) are proposed for the first row and the remaining rows respectively (Jaluria 1988).

For natural convection, Eq. (2.17) is proposed for flow between two asymmetric isothermal parallel plates (Kraus and Bar-Cohen 1983). The dominated mode of heat transfer is observed for flush-mounted discrete heat sources (Wang and Vafai 1999), The Eqs. (2.18) or (2.19) are proposed for protruding heat sources. Equations (2.20) and (2.21) are suggested to determine the Nusselt number for horizontal and vertical PCB assembly, respectively. For low values of modified Rayleigh number Eq. (2.23) is proposed for laminar conditions in a vertical channel with isothermal obstructions (Cammarata et al. 2000). A significant reduction of the Nusselt number is found for increasing aspect ratios (MacGregor and Emery 1969) and the experimental Eqs. (2.24) and (2.25) are proposed for turbulent and laminar flow regime respectively. For laminar flow, the Eqs. (2.26) and (2.27) are proposed for constant heat flux and constant wall

temperature conditions respectively. Finally Eqs. (2.28) and (2.29) are proposed for closed and open channel respectively, for an asymmetrically heated vertical channels (Darie et al. 1998).

For mixed convection there is no correlation reported except Eq. (2.18) for predicting the average Nusselt number for the case of discrete flush-mounted heat source (Wang and Vafai 1999). However, it is found that the location of the heater element does not play a considerable role in the heat transfer processes. For low Reynolds number, Grashof number mainly controls heat transfer and cooling is more effective than forced and natural convection (Türkoğlu and Yücel 1995).

For the flush-mounted and protruded discrete heat source, natural convection plays an important role in the total heat transfer process, leading to a possible mixed convection behavior. Furthermore, the protruding heat source distorts the flow field significantly and causes the transition from laminar to turbulent flow at a lower Reynolds number, and enhances the forced convection significantly (Wang and Vafai 1999, Young and Vafai 1999, Young and Vafai 1998). However, as the Reynolds number increases, the heat transfer by protruding chips is always greater than that by flush chips (Tou et al. 1998).

12. Numerical methods are effective in analysing heat transfer from flush-mounted heaters in single-phase fluid, and their use can provide a useful complimentary study to any experimental works.

13. In the transient forced convection, the Peclet number and time dependent Fourier number have significant effect on total heat transfer. In the transient natural convection,

Rayleigh number and time dependent Fourier number have significant effect on total heat transfer.

14. Although much work have been done on the steady-state direct liquid cooling channel flow or electronic systems, it seems there are very limited studies on the transient thermal response of direct liquid forced and natural convection heat transfer for discrete heat sources in channel flow.

15. Transient behavior of an electronic system is an important subject of electronic cooling. There is a need to launch fundamental study in this area in order to gain basic understanding of the problem.

In view of the above, it is recommended that further studies should be conducted on channel liquid cooling under steady-state and transient heat transfer conditions.

f the literature review.

Source	Convection mode			Flow regime			Discrete heating (array / mesh)		Rectangular channel H = horizontal V = vertical	Working fluid
	Natural	Mixed	Forced	Laminar	Turbulent	Based on heat source length	Flush mounted	Protruded		
E = Experiment N = Numerical										
Cammarata, Desrayaud, Fiche Fischetti (2000); Leng, Chen & Ch (2000); N	✓			$10^2 < Ra_w / A < 10^5$		$5 \times 10^2 < Ra_t / A < 5 \times 10^5$		Heated obstacle	V	$Pr = 0.71$
Wang & Vafai (1999); E	✓	✓	✓	$Re_w = 200$ to 500		$Re_t = 50$ to 125		752×72	H	Air
Tso, Xu & Tou (1999); E			✓	$Re_D = 1500$ to 6300		$Re_t = 392$ to 1644	1×6	1×6	H	Air
Wu & Perng (1999); N		✓		$Re_t = 6 \times 10^2$ to 8×10^4		$Re_t = 6 \times 10^2$ to 8×10^4	1×4		V	Water, FC-72
Tsay (1999); N		✓		$Re_t = 260$ to 530		$Re_t = 260$ to 530		1×5	H	Air, $Pr = 0.7$
Asako, Yamaguchi, Faghri (1999); N			✓	$Re_w = 500$		$Re_t = 5$	1×5		V	$Pr = 0.7$
Young & Vafai (1999); E, N			✓	$Re_c = 200$ to 2000		$Re_t = 800$ to 8000		$16 \times 42 \times 30$	H	$Pr = 0.7$
Meinders, Van D Meer & Hanjalic (1998); E		✓	✓	$800 \leq Re_D \leq 13000$		$1219 \leq Re_t \leq 19809$		1×5	H	Air
Leung & Kang (1998); N, E	✓	✓	✓	$Re_t = 795$, 2086	$Re_t = 3278$, 5066	$Re_t = 795$ to 5066		1×4	H	Air
Du, Bilgen & Vafai (1998); N		✓		$Re_w = 510$ to 2050		$Re_t = 153$ to 615		Printed circuit board	H & V	Air, $Pr = 0.7$
Gupta & Jaluria (1998); E			✓	$0 \leq Re_w \leq 200$ $0 \leq Ra_w \leq 10^7$		$0 \leq Re_t \leq 66.7$ $0 \leq Ra_t \leq 3.7 \times 10^7$		24×26	V	$Pr = 0.72$
					$Re_H = 2500$ to 9000	$Re_t = 1840$ to 6818		4×3	H	Water

Source E = Experimental N = Numerical	Convection mode		Flow regime			Discrete heating (array / mesh)		Rectangular channel H = horizontal V = vertical	Working fluid
	Natural	Mixed	Forced	Laminar	Turbulent	Based on heat source length	Flush mounted		
Tou, Xu & Tso (1998); E			✓		$Re_t = 0.2 \times 10^3$ to 6.3×10^4	$Re_t = 0.2 \times 10^3$ to 6.3×10^4	1×4	V	FC-72
Xu, Tou & Tso (1998); N			✓		$Re_t = 1 \times 10^4$ to 1.5×10^5	$Re_t = 1 \times 10^4$ to 1.5×10^5	1×4	V	Water, FC-72
Kim, Kang & Hy (1998); N			✓	$Re_H = 500$ to 700		$Re_t = 500$ to 700		H	$Pr = 0.7$
Chen & Wang (1998); E & N			✓	$Re_H = 1300$ to 10^4	$Re_H = 1300$ to 10^4	$Re_t = 650$ to 5000		H	Air
Young & Vafai (1998); N			✓	$200 \leq Re_D \leq$ 2000		$12.5 \leq Re_t \leq 125$		H	$Pr = 0.72$
Morris & Garime (1997); E			✓	$Re_t = 610$ to 68580	$Re_t = 610$ to 68580	$Re_t = 610$ to 68580		H	$Pr = 6.97,$ 5.42, 4.34
Olivos & Majum (1995); N			✓	$Re_c = 100$ to 1000		$Re_t = 400$ to 4000		H	Air
Sugavanam & Or (1995); N			✓	$Re_D = 100$ to 1000		$Re_t = 50$ to 500	1×1	H	Air
Türkoğlu & Yüce (1995); N		✓		$Re_t = 1$ to 500		$Re_t = 1$ to 500		V	$Pr = 0.7$
Hacohen, Chiu & Wragg (1995); E	✓		✓	$Ra_D = 1$ to 10^4 $Re_D = 1200$ to 2600		$Ra_t = 0.25$ to 2500 $Re_t = 300$ to 650		H	Air
Braaten & Patank (1985); N		✓		$0 \leq Re_B \leq 10^6$		$0 \leq Re_t \leq 20^6$		H	$Pr = 0.7$

Chapter 3 Experimental Apparatus and Procedures

3.1 Experimental Apparatus

3.1.1 Test Loop

The flow loop of the experimental system is shown in Fig. 3.1. A magnetically coupled centrifugal pump is used to circulate the liquid through the flow loop. Immediately downstream of the pump there is a heat exchanger to dissipate heat added in the test section and to regulate the temperature of fluid entering the channel. The liquid flows then bifurcates; allowing some of the coolant to enter a test section while the remainder is circulate through a by-pass line. Prior to entering the test section, the liquid passes through a 5- μm filter and a rotameter. Both the test section and bypass lines feed coolant into the condenser/reservoir. Pressure in the system is maintained constant by regulating the energy input of the immersion heater with 2 kW in the bottom of condenser/reservoir and by regulating the flow rate of city-water through a solenoid valve.

The system is deaerated before each experimental run by circulating fluid through the flow loop and supplying power to the immersion heater. The mixture of vapour and air is released at the upper section of the outlet reservoir. The immersion heater in the condenser/reservoir is then operated to force air out and the condenser condenses vapour. At the end of deaeration, the relief valve is closed, and power to the immersion heater is adjusted to attain the desired system pressure. Inlet temperature of the working fluid is

achieved by controlling the flow of cold water to the heat exchanger. The pressure level in the inlet of the test section and pressure drop in the test section was measured with two transmitters for differential pressure.

Fig. 3.1. Schematic of the liquid cooling test facility

3.1.2 Test section

The test section is composed of a rectangular flow channel; the longitudinal section of the test section is shown in Fig 3.2. The 840 mm long rectangular channel with 20 mm width (W) and 5 mm height (H) renders the three-dimensional modeling of the flow channel.

Fig. 3.2 Longitudinal section of flow channel

A multi-chip module with an array of four in-line flush-mounted chips is mounted vertically to simulate a cooling channel of electronics equipment. As for the flow channel, Plexiglas is being used as the main material to achieve the desired insulation. The multi-chip module as seen in Fig. 3.3 is machined from high temperature Teflon with low thermal conductivity ($k = 0.4 \text{ W/m}\cdot\text{K}$) to minimize substrate conduction heat loss. The first chip is located 700 mm downstream from the channel inlet. The hydrodynamic entry length is calculated as 360 mm for the value of Re_D as 750. The module is an entity separated from the channel, enabling it to be detached from the channel when required.

The chips are embedded in the center of the channel wall with a 5 mm gap between the edges of the chip and the channel sidewalls. A nomenclature is established to identify a specific chip within the array by referring to 'Chip 1' as the most upstream chip in the array as shown in Fig 3.3. The remaining chips are assigned the following numbers up to four. A film resistive heater is attached to each chip and is controlled by a voltage transformer that is connected in series. As such four similar voltage transformers are used for the four chips in the experiment. Variations in the heater powers between the different heaters are negligible. The resistors are attached to the underside of the chips with thermally conducting epoxy. Teflon is also used to isolate the module from the surroundings to minimize heat loss. A Teflon block is threaded to accommodate four screws to ensure that the resistors are attached well enough to the chips. Contact resistances between the chips and the heaters are found to be very small and assumed to be negligible.

Fig. 3.3 Test section of simulated chips mounted on one side.

3.1.3 Simulated Chips

The orientations and dimensions of a simulated microelectronic chip are shown in Fig. 3.4. Each chip is fabricated from oxygen-free copper with the height, $H_c = 9$ mm, and length, $\ell = 10$ mm. The dimensions of the chip surface in contact with the liquid are $10 \text{ mm} \times 10 \text{ mm}$. Two chromel-alumel thermocouples of the K-type are embedded along the chip centerline in the flow direction at a depth of 0.5 mm under the chip surface and are located 2 mm apart from upstream and downstream edges of the chip respectively. The wall or surface temperature of the chip is the average of the temperatures measured by the two thermocouples.

Fig. 3.4. Construction of the simulated chips

3.1.4 Working Fluids

Immersion cooling of packaged and bare microelectronic chips requires the use chemically stable and inert, non-toxic, nonflammable, low boiling point liquids, possessing a low dielectric constant, high dielectric strength, and high volumetric resistivity. The perfluorinated inert liquids such as 3M's Fluorinerts, have been evaluated extensively for direct contact with electronic packaging materials and appear to be the most suitable fluids for this application (Incropera et al. 1986, Mudawar and Maddox 1989, Garimella and Schlitz 1995). Other liquids, notably the chlorofluorocarbons (CFC)

– such as Dupont’s Freons has also been considered for use in immersion cooling, in particular, has been used frequently in thermofluid studies of direct liquid cooling and for immersion cooling of power devices. Current concern over the environmental effects of the chlorofluorocarbons would appear to diminish the prospects for widespread, commercial use of CFC liquids in immersion cooling. Freon replacement fluids, presently being manufactured, may, however, emerge to challenge the perfluorinated liquids in this market.

The thermal properties of Fluorinerts, R-113 and water (for comparison) are given in Table 1.2. The table reveals that all these candidate immersion-cooling liquids display relatively low thermal conductivities, specific heats, and latent heats of evaporation, but possess relatively high liquid and vapor densities. In addition, these liquids are characterized by low surface tension and high affinity for common atmospheric gases. As may be seen in this tabulation of liquid properties, the primacy of chemical and electrical compatibility in the section of direct contact coolants for electronic components has narrowed the choice to candidate immersion cooling liquids with relatively inferior thermal transport characteristics. The methodology of fluid Figure-of-Merit (FOM’s) defined by Saylor et al. (1988) and Bergles and Bar-Cohen (1990) can be used to evaluate and compare the thermal performance of several candidate immersion cooling fluids for natural and mixed convection.

In the present study, water has been used as a cooling liquid in the experiment. However, water is not an accepted coolant in the cooling of real electronic components due to its

poor electrical insulation. On the other hand, water has been used as a working fluid in several experimental studies of electronic cooling, examined by Garimella and Eibeck (1990, 1991), Garimella and Schlitz (1993, 1995), Incropera (1986, 1990) and Mahaney et al. (1990). It is reasonable to use water as a coolant at the first step of the experiment.

However, 3M's Fluorinerts liquid FC-72 is also used as working fluid. The liquid does not affect the ozone layer. It is extremely non-polar and has essentially no solvent action. It is colorless, odorless, low in toxicity, and non-flammable. It also has high thermal stability, low chemical reactivity, and leave essentially no residue. The thermophysical properties of FC-72 are described in Appendix B in detail.

3.1.5 Instrumentations

In this experiment, data are measured, recorded and reduced by an automatic PC based data acquisition system. These data included the fluid temperatures, wall temperatures in the simulated chips. All temperatures mentioned above are measured with K type thermocouples hooked up to the data acquisition system. They are calibrated prior to use. The voltage and resistance across the each heater are measured to calculate the power supplied to the each heater. The pressure difference between the inlet and the outlet of the test section is measured with one differential pressure transmitter. The flow rates are measured with one of three rotameters depending on the range $2 \times 10^{-3} \text{ m}^3/\text{h} - 45 \times 10^{-3} \text{ m}^3/\text{h}$, $20 \times 10^{-3} \text{ m}^3/\text{h} - 200 \times 10^{-3} \text{ m}^3/\text{h}$, $60 \times 10^{-3} \text{ m}^3/\text{h} - 600 \times 10^{-3} \text{ m}^3/\text{h}$ and $200 \times 10^{-3} \text{ m}^3/\text{h} - 20000 \times 10^{-3} \text{ m}^3/\text{h}$.

3.2 Experimental Procedures

3.2.1 General Experimental Procedures

It is proposed to use water and FC-72 as the working fluid for single-phase heat transfers. The liquid readily absorbs air whose presence in coolant could affect the experimental results. Therefore, prior to test runs, the coolant is thoroughly degassed. The degassing process is conducted in the storage tank where the liquid is boiled by supplying power to the immersion heater, the mixture of vapour and air is released at the upper section of the tank. The immersion heater in this tank is then operated to force air and the water-cooled condenser condenses vapor. Generally, there are two situations under which degassing of the coolant is required; one is the initial filling (or make up) of the liquid in the tank, the other is when the coolant in the loop is exposed to the atmosphere when the test section is changed or removed. After deaerating for twenty minutes, the relief valve is closed. Utilizing a gas chromatographic technique, Danielson et al. (1987) found that the air concentration in similar Fluorinerts was less than 1 percent by volume after only three minutes of deaeration; hence, a deaeration time of twenty minutes is adequate to deaerate a larger system sufficiently. In setting up for a test run to obtain a predetermined flow condition for the single-phase heat transfer experiments, the general sequence of operations are as follows:

1. Apply power to the test section to set the desired heat flux.
2. Preheat the coolant in the tank and precooled the coolant in the plate heat exchanger, set the desired temperature at the test section.
3. Set the required flow rate
4. Take the flowmeter, chip wall temperature and heat flux reading

Prior to performing the experiments, the chip surfaces are polished with a waterproof abrasive silicon carbide paper that has particles of average size $10\text{ }\mu\text{m}$ and should create a uniform surface texture on each chip. Before switching on the pump to circulate the coolant, all the valves in the loop are opened to avoid building up high pressure suddenly. Adjusting the bypass and regulating valves set the desired mass flow rate through the test section. To improve the accuracy of readings, four rotameters with different ranges are used to measure the flow rate.

The system pressure is maintained by adjusting the power to the immersion heater and by regulating the flow rate of city-water through a solenoid valve. The procedure for obtaining data in the single-phase experiments is to start with a given heat flux and to vary the velocity. The increments of heat flux were made of the order of 0.1 W/cm^2 . When the surface temperature of a heater element increased sharply, it was assumed that the critical heat flux had been exceeded, and power to the heaters was immediately cut off.

Before measurements were recorded, the wall temperatures of the simulated chips were allowed to reach steady-state, typically in about 1 hour, after the power level and flow rate were set. Temperature readings were sampled by a PC at 1 Hz and averaged over a period of approximately 15 min. The wall temperatures of the four simulated chips, test section inlet temperature are calculated from the measurements of the thermocouple voltages with the data acquisition system. The experimental data are then reduced and printed for checking. If the reduced data appear good, the raw data of the test run are

recorded. Subsequent test runs used the same procedures. During the experiments, the difference between the temperatures measured two thermocouples in each simulated chip was less than 0.5°C. This suggested that the distribution of surface temperature in the simulated chips is uniform in single-phase liquid cooling. Average of the two surface temperatures yielded the mean surface temperature. The Biot number Bi , in the present study varies from 0.002 to 0.007, hence isothermal condition on the chips is expected (Wang and Vafai 1999, Tso et al. 2004).

3.2.2 Steady-state Natural Convection Heat Transfer

The liquid cooling flow facility with a vertical upflow channel with the test section is shown in Fig. 3.5. The temperature at the test section inlet is maintained constant by means of the heat exchanger and the immersion heater in the reservoir (not shown), and is measured just prior to the test section by a K -type thermocouple. The first chip is located 700 mm downstream of the channel inlet, providing a minimum hydrodynamic entry length of 50 hydraulic diameters.

The experiment begins with a given heat flux and over a zero flow velocity. The simulated chips tested in this study are heated simultaneously at constant heat flux. The chip temperatures always remain less than 70°C.

Fig 3.5. Test facility for natural convection

When the surface temperature of a heater element increases sharply, it is assumed that the critical heat flux has been exceeded, and power to the heaters is immediately cut off. Before measurements are recorded, the wall temperatures of the simulated chips are allowed to reach steady-state, typically in about 1 hour, after setting the power level. The heat flux ranges from 0.1 W/cm^2 to 0.6 W/cm^2 and an inlet temperature of $T_{in} \approx 21^\circ\text{C}$.

3.2.2.1 Range of Flow Conditions

In the steady-state heat transfer tests; the total number of test runs for flush-mounted chips using water is 6 for natural convection. The flow condition ranges for natural convection are tabulated in Table 3.1 as given below:

Table 3.1 Experimental conditions for steady-state natural convection heat transfer

Working fluid	Water
Flow rate, Q (m ³ /h)	0
Velocity, U (m/s)	0
Reynolds number Re_D	0
Heat flux, q'' (W/cm ²)	0.1~0.6
Prandtl number, Pr	6
Protrusion, B (mm)	0

3.2.3 Steady-state Forced and Mixed Convection Heat Transfer

The procedure for obtaining data in the steady-state experiments is to start with a given heat flux and the flow rate ranging from 0.5×10^{-3} m³/h to 20×10^{-3} m³/h. The increments of heat flux were made of the order of 0.1 W/cm². Before measurements were recorded, the wall temperatures of the simulated chips were allowed to reach steady-state, typically in about 1 hour, after the power level and flow rate were set. All experiments with water are performed in a vertical upflow channel over an inlet temperature of $T_{in} \approx 24^\circ\text{C}$, and the heat flux ranging from 0.1 W/cm² to 0.6 W/cm². The simulated chips are also heated simultaneously at constant heat flux and the chip temperatures always remain less than 70°C.

3.2.3.1 Range of Flow Conditions

In the steady-state heat transfer tests, the total number of test runs for flush-mounted chips using water, as working fluid is 209 for forced and mixed convection. The flow

condition ranges for forced and mixed convection are tabulated in Table 3.2 as given below:

Table 3.2. Experimental conditions for steady-state forced and mixed convection heat transfer

Working fluid	Water
Flow rate, Q (m ³ /h)	0.002-0.08
Velocity, U (m/s)	0.0056-0.222
Reynolds number, Re_D	56-2220
Heat flux, q'' (W/cm ²)	0.1~2.6
Fluid inlet temperature (°C)	~20
Prandtl number, Pr	6
Protrusion, B (mm)	0, 1, 2

3.2.4 Transient Heat Transfer

The five transient regimes can be obtained from the experiments namely for power-on, pump-off, pump-on, power-off, and the power-on transient natural convection regimes. In all the cases temperatures are measured in one-seconds intervals and number of data are recorded depending on the nature of the curves. The effects of heat flux and chip numbers are investigated in the transient regime. Finally the correlation equations are developed for four chips as well as for overall data. It is worth mentioning here that the correlation equations are valid in the intermediate period of time elapsed after the change in heating or pumping power. The experimental procedures for these transient operations are discussed below.

Power-on Transient

The temperatures of the simulated chips are allowed to reach steady-state, the electrical powers to the chips are then switched on, and the chip temperature will increase sharply and will reach steady-state again with increase of time. This is the condition for transient heat transfer with constant flow but changing coolant temperature.

1. The required flow rate is set.
2. The heater power is switched on at the required value.
3. The fluid inlet temperature and the chip wall temperatures are measured throughout the power-on transient period until they reach steady-state.

The temperatures of the chips increase sharply with time at the beginning of the period. At the power-on transient period, after the power-on, the data for first 75s are taken for analyses.

Pump-off Transient

To conduct the transient study, the pump power was shut down during steady-state, and instantly the chip wall temperature increased rapidly. When the chip wall temperature reached about 70°C, all the power was shut down to terminate the experiment, ensuring that there was no phase change and the chip temperature was well below the design limit of 85°C. However, this is the condition of transient heat transfer with turning off pump power but changing coolant temperature. At the pump-off transient period, after the pump-off, the data for first 150s are taken for analyses.

Pump-on Transient

The pump power was shut down during steady-state, and instantly the chip wall temperature increased rapidly. When the temperature reached about 70°C, the pump power was turned on. However, the chip temperature decreased sharply and provided the transient operation. This was the condition of transient heat transfer with turning on pump power but changing coolant temperature. In all experiments the chip temperature was well below the design limit of 85°C ensuring that there was no phase change in the flow channel. At the transient period, after the pump-on, the data for first 75s are taken for analyses.

Power-off Transient

At the end of pump-on transient operation the chip wall temperature reached steady-state again. After reaching steady-state the power of the heaters were switched off. Then the chip wall temperature decreased sharply. At the transient period, after the power-off, the data for first 75s are taken for analyses.

Power-on Transient Natural Convection Heat Transfer

All experiments are performed when there is no flow in to the test section and with the temperature at the channel inlet of $T_{in} \approx 21^\circ\text{C}$. The heater power is increased in steps from 0.1 W/cm² to 0.6 W/cm². The temperature of the chips will increase sharply with time at the beginning of the period. This is the condition of transient heat transfer with turning off the pump power but changing coolant temperature. At the transient period, after the power-on, the data for first 70s are taken for analyses.

3.2.4.1 Range of Flow Conditions

In the transient heat transfer, the total number of test runs using water and FC-72, as working fluid is 256 for flush-mounted heaters, whereas for protruding chips the total number of test runs was 512. The flow conditions are tabulated in Table 3.3.

Table 3.3. Experimental conditions for transient heat transfer

Working fluid	Water, FC-72
Flow rate (m ³ /h)	0.036 – 0.091
Velocity (m/s)	0.11 – 0.25
Reynolds number Re_t	800 – 2625
Heat flux (W/cm ²)	1 – 7
Inlet temperature °C	~ 23
Prandtl number Pr	6
Protrusion, B (mm)	0, 1, 2

3.3 Analyses of Experimental Data

All the thermophysical properties (Perry 1963 and Greankoplis 1972) are allowed to vary with temperature and are evaluated based on the bulk fluid temperature T_b , defined here as

$$T_b = \frac{T_{wall} + T_{in}}{2} . \quad (3.1)$$

Equation (3.1) is used to define all the properties (k , ρ , μ , β , c_p) to correlate all the experimental data. It is worth mentioning here that this equation does not contribute to correlate any cited data as well as to carry out numerical computations.

The temperature difference between chip and liquid inlet is defined as:

$$\Delta T = T_{wall} - T_{in}. \quad (3.2)$$

The hydraulic diameter D is as defined conventionally for the effective flow area as shown in Fig. 3.6 for flush-mounted chip, and the velocity U is based on the cross sectional area of the empty space over heat sources, as the following:

$$D = \frac{2(WH - B\ell)}{W + H + B}, \quad (3.3)$$

$$U = \frac{Q}{WH - B\ell}, \quad (3.4)$$

Fig. 3.6. Cross section of flow channel

Reynolds numbers based on heat source length Re_ℓ , and Reynolds number based on hydraulic diameter Re_D , are defined as

$$Re_\ell = \frac{\rho U \ell}{\mu}, \quad (3.5)$$

$$Re_D = \frac{\rho U D}{\mu}. \quad (3.6)$$

To examine the effects of flow conditions on the single-phase heat transfer, the values of Nu_ℓ and heat flux from the test runs are needed for quantitative analysis. Data are

reduced in terms of an average Nusselt number, based on channel height Nu_H , based on heat source length Nu_ℓ , respectively defined by

$$Nu_H = \frac{hH}{k} = \frac{q''H}{k(T_{wall} - T_{in})}, \quad (3.7)$$

and

$$Nu_\ell = \frac{h\ell}{k} = \frac{q''\ell}{k(T_{wall} - T_{in})}, \quad (3.8)$$

where T_{wall} is the average of a chip wall temperature measured by two thermocouples attached on each chip and T_{in} is the inlet fluid temperature in the test section. The chip length is $\ell = 10$ mm, providing exposed chip surface areas 100, 140 and 180 mm² for $B = 0, 1$ and 2 mm respectively.

The Rayleigh number based on height of the channel Ra_H , is defined as

$$Ra_H = \frac{g\beta q'' H^4}{k\nu\alpha}. \quad (3.9)$$

The Rayleigh number based on heat source length Ra_ℓ , is defined as

$$Ra_\ell = \frac{g\beta q'' \ell^4}{k\nu\alpha}. \quad (3.10)$$

The Grashof number based on channel hydraulic diameter Gr_D , height of the channel Gr_H , and heat source length Gr_ℓ , are defined respectively as

$$Gr_D = \frac{g\beta q'' D^4}{k\nu^2}, \quad (3.11)$$

$$Gr_H = \frac{g\beta q'' H^4}{k\nu^2}, \quad (3.12)$$

and

$$Gr_\ell = \frac{g\beta q'' \ell^4}{k\nu^2}. \quad (3.13)$$

The Biot number based on height of the chip Bi , is defined as

$$Bi = \frac{hH_c}{k_c}. \quad (3.14)$$

The Peclet number, based on heat source length Pe_ℓ , is defined as

$$Pe_\ell = Re_\ell \times Pr. \quad (3.15)$$

The Fourier number, based on height of the chip Fo , is defined as

$$Fo = \frac{\alpha t}{H_c^2}. \quad (3.16)$$

3.4 Experimental Error Analyses

The heat loss to the surrounding was estimated by measuring the surface temperature of Teflon exposed to the surroundings and the temperature of ambient air, and based on the assumption that the heat losses by conduction in the multi-chip module are equal to the heat dissipation by natural convection from the surface of multi-chip module to surroundings under the notation of thermal equilibrium. The contact resistance between the copper block and heaters were measured in different test and was found the error of $\pm 0.3 \Omega$ and the percentage of heat loss is estimated to be less than 5 % of the total heat energy dissipated by the heater. Therefore, in the light of the small heat losses, no correction is made for the power dissipated by the thick-film resistor in determining the chip heat flux. In generally, heat losses for liquid cooling of simulated chips are negligible, such as in the studies of Incropera et al. (1986), and Willingham and Mudawar (1992) and Heindel et al. (1992).

The maximum percent uncertainty associated with each experimental reading is given in Table 3.4. An estimation of the overall uncertainties in the experimental parameters is made using standard technique of the propagation of error for single-sample measurements suggested by Kline and McClinlock (1953) and Moffat (1988). The procedures of calculating overall uncertainties are given in Appendix A. Table 3.5 summarizes the overall uncertainties in measuring the important parameters presented in this thesis. These values are based on the assumption of negligible uncertainty in the relevant fluid properties.

Table 3.4. Experimental uncertainties

Experimental readings	Maximum experimental uncertainty ($\pm$)	Method of estimation
Thermocouple	0.3 °C	Calibration
Heat flux	0.1 W/m ²	Manufacturer’s data, Calibration
Flowmeter	0.5 lph	Manufacturer’s data

Table 3.5. Overall uncertainties

Parameter	$\pm$ % of value (max)
U	3.5
ΔT	1.4
q''	5.0
Nu_H	8.8
Nu_ℓ	6.8
Ra_H	3.5
Ra_ℓ	3.5
Re_D	3.5
Re_ℓ	3.8
Gr_D	3.8
Gr_ℓ	3.5
Fo	3.0

Chapter 4 Experimental Results for Steady-state Heat Transfer

4.1 Introduction

In this chapter, experimental results of steady-state laminar forced, mixed and free convective heat transfer from four in-line simulated electronic chips in a rectangular channel using water are described and analysed. The heat flux changes from 0.1 W/cm^2 to 2.6 W/cm^2 . The Reynolds number based on heat source length ranged from 0 to 2775 and based on channel hydraulic diameter ranged from 0 to 2220. The Biot number Bi , in the present study varies from 0.002 to 0.007, hence isothermal condition on the chips is expected (Wang and Vafai 1999). The present results from liquid cooling are compared with other results from air-cooling as well as liquid cooling for flush-mounted chips.

4.2 Studies on Flush-Mounted Chips

Tests are conducted with flush-mounted heat sources with the values of $\ell = 10 \text{ mm}$, $H/\ell = 0.5$ and $S/\ell = 1$, and the results are presented in the following sections. The present results from liquid cooling are also compared with other results from air-cooling.

Before measurements were recorded, after the power level and flow rate were set, the wall temperatures of the simulated chips were allowed to reach steady-state, typically in 3500 s as shown in Fig. 4.1(a) for the conditions of the value of $Re_D = 320$, $q'' = 0.4 \text{ W/cm}^2$.

(a) Wall temperature variations with time

(b) Percentage deviation of temperature at 3500 s versus time

Fig. 4.1. History of temperature readings

Here the % of deviation can be defined as:

$$\% \text{ deviation} = \frac{T_{x+1} - T_x}{T_{x+1}} \times 100$$

where, T_x = temperature at time x second

T_{x+1} = temperature at time $(x+1)$ second

In Fig. 4.1(b) it is seen that this percentage deviation at 3500 s is found to be 0.5% for all the four chips and inlet temperature. However, the temperature at 3500 s is considered as a steady-state temperature for four chips and inlet temperature. During the experiments, the difference between the wall temperatures measured by two thermocouples in each simulated chip is shown in Fig. 4.2.

Fig. 4.2. Distribution of wall temperature

It is noted that the wall temperature distribution in the simulated chips is fairly uniform. The difference between the chip wall temperatures measured by two thermocouples in each simulated chip is less than 0.5 °C during the entire experiment. However, the temperature

difference between the wall temperature and inlet temperature increases as the chip number increases, as was observed by Tso et al. (1999). The variation of the value of Nu_ℓ with chip number is shown in Fig. 4.3 with the Reynolds number based on heat source length changing from 910 to 2380.

Fig. 4.3. Variation of the value of Nu_ℓ with chip number for flush-mounted chips

The value of Nu_ℓ used here is the average of the value of Nu_ℓ at the steady-state period. It can be seen from the Fig. 4.3, there is a noticeable drop of the values of Nu_ℓ as the chip number increases. The upstream thermal boundary affects heat transfer characteristics of the downstream. The value of Nu_ℓ decreases as the chip number increases and reaches chip number independency by the third chip, since the change between the third and fourth chip is small (9% at $Re_\ell = 910$ and this value is almost same for all Re_ℓ). This attainment of chip number independency at downstream of three to four rows was also observed by Incropera

et al. (1986) in turbulent flow, Mahaney et al. (1990) and Garimella and Eibeck (1990) in laminar flow. The value of Nu_ℓ for various chip number at different input powers (0.2 W/cm², 0.3 W/cm², 0.4 W/cm², 0.5 W/cm² and 0.6 W/cm²) are also obtained and are shown in Fig. 4.4.

(a) Chip 1

(b) Chip 2

(c) Chip 3

(d) Chip 4

Fig. 4.4. Effects of heat flux on Nu_t for each chip

It can be seen from this figure that the difference of Nusselt number between 0.2 W/cm^2 and 0.6 W/cm^2 is almost 30% to 36%. This is because of the non-constant properties that had little effect on the heat transfer, as suggested by Sieder and Tate (1936). When the input power increases, the chips wall temperature increases and the viscosity of the liquid decreases (Xu et al. 1998). As the flow rate constant, the inlet temperature increases because the system is closed loop. Thus, we can say that the input power has small effects on the value of Nu_t . Tso et al. (1999) also observed phenomenon in both laminar and turbulent flow regime in forced convection heat transfer.

4.2.1 Laminar Flow Forced Convection

The Nusselt number (Nu_t) versus Peclet number (Pe_t) based on heat sources length is shown in Fig. 4.5, for four chips. For flow in a channel or between parallel plates, with a fully-developed velocity profile, the heat transfer data can be correlated by the equations (Tso et al. 1999, Shah and London 1978, Mahaney et al. 1990) of the following form:

$$Nu_t = C_1 Pe_t^{1/3}, \quad (4.1)$$

where $Pe_t = Re_t Pr$. The experimental results from Tso et al. (1999) are plotted in the same figure, which shows similar results with the present study, especially for the average of the four chips shown in Fig. 4.5(e).

For parallel plate duct, laminar flow and heat transfer have been analysed in detail by Shah and London (1978). The following equations are recommended for thermally developing and hydro-dynamically developed flow for constant heat flux boundary conditions:

$$Nu_\ell = 2.236(Pe_\ell)^{1/3} \quad \text{for } (1/Pe_\ell) \leq 0.001 \quad (4.2)$$

$$Nu_\ell = 2.236(Pe_\ell)^{1/3} + 0.9 \quad \text{for } 0.001 < (1/Pe_\ell) < 0.01 \quad (4.3)$$

$$Nu_\ell = 0.0364 Pe_\ell + 8.235 \quad \text{for } (1/Pe_\ell) \geq 0.01 \quad (4.4)$$

(a) Chip 1

(b) Chip 2

(c) Chip 3

(d) Chip 4

(e) Average of the four chips

Fig. 4.5. Nusselt number (Nu_t) variations with Peclet number (Pe_t) for present experimental data compared to data from two references.

The linear fit method is used to correlate the laminar flow data including the forced convection data with $Nu_t = C_1(Pe_t)^m$ in Fig. 4.5. The value of C_1 and m , standard deviation SD for C_1 and m are listed in Table 4.1. It is found that the value of the exponent m is close to the value of $1/3$. However, it is ensured that the test rig is in good working condition (Tso et al. 2004, Bhowmik and Tou 2005a, 2004a, Bhowmik et al. 2005a).

Table 4.1. Results of linear fit for lines in Fig. 4.5

	Chip 1	Chip 2	Chip 3	Chip 4	Average
C_l	2.35	2.17	1.84	1.76	2.02
Difference of C_l with Shah & London (1978)	4.9%	-3.3%	-21.3%	-26.7%	-10.6%
m	0.361	0.353	0.343	0.324	0.348
Difference of m with Shah & London (1978)	7.6%	5.7%	2.9%	-2.9%	4.2%
SD of C_l	0.38	0.28	0.30	0.25	0.32
SD of m	0.018	0.014	0.018	0.016	0.016

4.2.2 Effects of Reynolds Number

Nusselt number (Nu_t) as a function of Reynolds number (Re_t) based on heat source length is shown in Fig. 4.6, for four chips. The forced convection results for four chips fall along the slope of 1/3 are also shown in Fig. 4.6, for four chips.

Fig 4.6. Variation of $[Nu_e / Pr^{1/3}]$ with Reynolds number (Re_e)

It is seen from this figure, data for the higher value of Re_e lie on this limit. As the value of Re_e is decreased, heat transfer enhancement above the forced convection limit initially occurs for the fourth heater. This behaviour results from the onset of thermal instabilities and the probable development of secondary flow due to heating at the three upstream heaters. However, the value of Nu_e continues to decrease with decreasing Re_e until a minimum is reached, where the reduction in the value of Nu_e of the fourth heater due to forced convection effects is balanced by an increase due to the buoyancy-driven secondary flow. With a further reduction of the value of Re_e , the strength of the buoyancy-induced secondary flow increases due to the reduction of forced flow. Having achieved conditions dominated by mixed convection, a further decrease in Re_e , results in decay of the forced convection, transition to pure natural convection with a gradual decrease of Nu_e . Although

the data appear to be strongly Re_t dependent, the results can be shown to correspond to pure natural convection (Bhowmik et al. 2005a, 2005b, 2004c).

The transition from pure forced convection to mixed convection to pure natural convection occurs in a similar manner for the upstream heaters. Since transition from forced to mixed convection results from the onset of thermal instabilities and from the cumulative development of secondary flows due to heating at all upstream heaters. This is consistent with the experimental and three-dimensional computational results provided by Mahaney et al. (1990) for four rows, in-line array of 12 square flush-mounted heat sources.

4.2.3 Laminar Flow Mixed Convection

The influence of buoyancy can be expressed by the buoyancy parameter or mixed convection parameter, $[Gr_D/Re_D^2]$ is also called the Richardson number, Ri_D (Seo and Parameswaran 2002, Guo and Sharif 2004). The number Ri_D represents the relative magnitude of the natural convection to the forced convection and plays an important role in designating the convection flow regimes. When the value of $Ri_D \approx 1$, both natural and forced convection are equally dominant and the flow regime is designated as mixed convection. If the value of $Ri_D \gg 1$, then natural convection is dominant and the buoyancy effects can be neglected when the value of $Ri_D \ll 1$ (Seo and Parameswaran 2002, Sathyamurthy et al. 1992, Guo and Sharif 2004, Lloyd and Sparrow 1970, Incropera and Dewitt 1998, Bhowmik et al. 2005a, 2005b).

Table 4.2 indicates the value of Re_D and $[Gr_D/Re_D^2]$ for six different heat fluxes. The mixed convection regimes are obtained from the value of $[Gr_D/Re_D^2]$. Here the range of $[Gr_D/Re_D^2]$ has been selected as 0.5 to 1.5 as mixed convection regime. The values of $[Gr_D/Re_D^2]$ below 0.5 can be classified as pure forced convection regime. The values of $[Gr_D/Re_D^2]$ above 1.5 can be classified as pure natural convection regime. However, the range of Re_D can be obtained in the forced, mixed and natural convection regimes.

Table 4.2 Experimental data for different experimental conditions.

Re_D	0.1 W/cm ²	0.2 W/cm ²	0.3 W/cm ²	0.4 W/cm ²	0.5 W/cm ²	0.6 W/cm ²
	Gr_D/Re_D^2	Gr_D/Re_D^2	Gr_D/Re_D^2	Gr_D/Re_D^2	Gr_D/Re_D^2	Gr_D/Re_D^2
46.29	15.91	26.94	25.5	42.5	42.07	47.63
89.54	3.87	6.22	6.12	9.54	10.36	11.29
141.08	1.67	2.36	2.64	3.95	4.21	4.57
189.54	0.97	1.26	1.54	2.01	2.33	2.62
234.95	0.57	0.83	0.93	1.25	1.43	1.67
274.97	0.41	0.57	0.66	0.87	0.96	1.13
327.97	0.30	0.43	0.47	0.61	0.61	0.81
377.12	0.22	0.33	0.36	0.48	0.50	0.56
474.88	0.14	0.21	0.24	0.30	0.31	0.33
565.37	0.10	0.13	0.16	0.21	0.21	0.24
659.21	0.07	0.09	0.11	0.15	0.15	0.17
749.63	0.06	0.07	0.09	0.11	0.11	0.12
851.09	0.04	0.06	0.07	0.08	0.09	0.09
946.81	0.03	0.04	0.06	0.07	0.07	0.07
1054.12	0.02	0.04	0.05	0.05	0.06	0.06
1132.15	0.02	0.03	0.032	0.04	0.05	0.05
1418.82	0.015	0.02	0.02	0.023	0.03	0.027
1719.37	0.01	0.012	0.013	0.016	0.02	0.018
1926.97	0.008	0.01	0.01	0.012	0.013	0.015

Many of the electronic components cooling processes are operated within mixed convection regime. The average value of Nu_t is obtained for a range of the value of Ri_D encompassing the forced, mixed and natural convection regimes. In the present experiment, the significant effect of mixed convection is observed at the value of $Re_t \approx 1000$, as shown in Fig. 4.6.

Figure 4.7 shows the forced and natural convection regimes for the average value of the four chips, for the range of flow from $0.5 \times 10^{-3} \text{ m}^3/\text{h}$ to $20 \times 10^{-3} \text{ m}^3/\text{h}$ and heat flux from $0.1 \text{ W}/\text{cm}^2$ to $0.6 \text{ W}/\text{cm}^2$. It is seen that from $0.3 \text{ W}/\text{cm}^2$ to $0.6 \text{ W}/\text{cm}^2$, there is an indication that around unity for the value of $[Gr_D/Re_D^2]$, there is a change in heat convection mode. However, the mixed convection regime is less clear for $0.1 \text{ W}/\text{cm}^2$ and $0.2 \text{ W}/\text{cm}^2$, with $0.1 \text{ W}/\text{cm}^2$ having a flat characteristic and the results seem to be fairly constant for all three regimes: natural, mixed and forced convection.

Fig. 4.7 Flow regions for natural and forced convection

Figure 4.7 also shows that the total heat transfer is affected by both the forced and natural convection. The experimental data obtained for all test cases are then used for the development of empirical correlations in the mixed convection regime. For mixed convection, the range of $[Gr_D/Re_D^2]$ from 0.5 to 1.5 (1 ± 0.5) has been selected for the development of empirical correlations, as a range encompassing unity. It is found that when both natural and forced convection are significant, $[\overline{Nu}_\ell / Re_D^{0.5}]$ is a function of $[Gr_H/Re_D^2]$ (Wang and Vafai 1999). The Reynolds number and Nusselt number are a function of buoyancy, when the combined forced and natural convections are dominant (Barletta et al. 2003). Therefore the proposed empirical correlation for mixed convection to be the following form (Bhowmik et al. 2005a, 2005b, 2004c):

$$\frac{Nu_\ell}{Re_D^n} = C_2 \left(\frac{Gr_D}{Re_D^2} \right)^m. \quad (4.5)$$

The variation of $[Nu_\ell/Re_D^n]$ with $[Gr_D/Re_D^2]$ for average of the four chips are shown in Figs. 4.8(a), 4.8(b), and 4.8(c) for three values of $n = 0$, $n = 0.5$, and $n = 1.5$ respectively.

(a) At $n = 0$ (same as Fig. 4.7 with range indicated)

(b) At $n = 0.5$

(c) At $n = 1.5$ Fig. 4.8. Variations of $[Nu_t/Re_D^n]$ with $[Gr_D/Re_D^2]$ at different n values

In the mixed convection regime (1 ± 0.5), The exponent n of Re_D on the left side of Eq. (4.5) has been selected from different tests with different exponents ranging from 0.5 to 2.5. The standard deviations (SD) of C_2 and m in Eq. (4.5) at different values of the exponent n of $[Nu_t/Re_D^n]$ are calculated and plotted in Fig. 4.9 and the values are given in Table 4.3. It is seen that the values of SD of C_2 and m reach minimums at the unique value of $n = 1.25$. Therefore the exponent for Re_D is selected as 1.25. Depending on the heater location, the n of Re_D was found to be in the range 0.35-0.44 by McEntire and Webb (1990). Incropera et al. (1986) found 0.6 for the first row heater and 0.78 for the fourth row heater and with the value between 0.6 and 0.78 for the second and third row heaters. For thermally and hydrodynamically developing mixed convection regime, Wang and Vafai (1999) found 0.5. The difference of exponents between the present work

and that of Incropera et al. (1986) is significant. While those of Incropera et al. (1986) is based on fully-developed region, and is conducted for a higher Reynolds number range without any significant mixed convection.

Table 4.3. Exponents for standard deviation SD , of C_2 and m

Exponent, n	SD of m	SD of C_2
0.5	0.1132	0.0360
0.7	0.0845	0.00686
1	0.0438	0.00046
1.1	0.0328	0.00018
1.2	0.0259	0.000069
1.3	0.0266	0.000036
1.6	0.0579	0.000010
2.0	0.116683	0.0000013
2.5	0.1884	0.00000007

Fig. 4.9. Variation of exponent n with standard deviation SD

The variations of $[Nu_e/Re_D^{1.25}]$ with $[Gr_D/Re_D^2]$ for all experimental points averaged together for the four chips at various heat fluxes are shown in Fig. 4.10. Results show that all the data that were represented by separate curves for different heat fluxes in previous plots (Fig. 4.8) collapse into a single line when an appropriate value of n is chosen. Thus, it is possible to represent the thermal behaviour with a single correlation that is independent of heat flux.

Fig. 4.10. Variation of $[Nu_e/Re_D^{1.25}]$ with $[Gr_D/Re_D^2]$

In order to test out the validity of the exponent 1.25, data collected from Busdra and Soliman (2000) where Grashof number is based on radius of the circular part of the channel, Wang and Vafai (1999) where Grashof number is based on height of the channel, Türkoğlu and Yücel (1995) where Grashof number is based on heat source length, Sathyamurthy et al. (1992) where Grashof number is based on width of the inflow

and outflow ports of the enclosure, and Braaten and Patankar (1985) where Grashof number is based on height of the heated block are plotted, using the exponent, 1.25 for Re in the y-axis is shown in Fig. 4.11. They are for different experimental conditions, different geometrical parameters and different definitions for Grashof number, hence not expected to be identical with the present experimental results.

Fig. 4.11. Variation of $[Nu/Re^{1.25}]$ with $[Gr/Re^2]$ for different results

However, from Fig. 4.11 the striking result is found; the slopes are similar for all the curves. Of all published results, only the conditions of the work of Türkoglu and Yücel

(1995) are nearer to our experimental conditions and hence our results are compared with their results (in Fig. 4.12). For comparison, Grashof numbers and Reynolds numbers were re-calculated based on the heat source length. It is seen that even though the curves are not close to each other, they are of similar slope (i.e., a similar exponent m in Eq. (4.5)). In fact, we do not expect good agreement with Türkoğlu and Yücel (1995) because they are two-dimensional numerical results for single heat source with different channel aspect ratio. It is found that the heat transfer coefficient is known to be a strong function of channel aspect ratio and method of heating (MacGregor and Emery 1969). Our Nusselt number is about 35% lower than that of the results of Türkoğlu and Yücel (1995). From Fig. 4.12 it is also seen that even though the form of the correlation is developed for the mixed convection regime, the results seems to be applicable to all three convective regimes: natural, mixed and forced (Bhowmik et al. 2005a, 2005b).

Fig. 4.12. Variation of $[Nu_l/Re_l^{1.25}]$ with $[Gr_l/Re_l^2]$ for average experimental results

Thus the correlation for mixed convection, Eq. (4.5), can be re-written in the following two forms:

$$\frac{Nu_\ell}{Re_D^{1.25}} = C_2 \left(\frac{Gr_D}{Re_D^2} \right)^m \quad (\text{based on hydraulic diameter, } D), \quad (4.6)$$

$$\frac{Nu_\ell}{Re_\ell^{1.25}} = C_3 \left(\frac{Gr_\ell}{Re_\ell^2} \right)^m \quad (\text{based on heat source length, } \ell). \quad (4.7)$$

This is done because in the literature both D and ℓ are commonly used to define the characteristic length, where m is same for both the two cases because only the constant ratio of D and ℓ is used in this experiment to convert D to ℓ . The variation of $[Nu_\ell/Re_D^{1.25}]$ with $[Gr_D/Re_D^2]$ at different heat fluxes for average of the four chips with the enlarge view of mixed convection region is shown in Fig. 4.13.

Fig. 4.13. Mixed convection region

The values of C_2 and m in Eq. (4.6) for the four chips are determined by linear fit method (Jaluria 1988) to correlate the laminar flow data, the results shown in Fig. 4.14 and the data for SD of C_2 and m are given in Table 4.4. The standard deviations for both C_2 and m are within 10%.

(a) Chip 1

(b) Chip 2

(c) Chip 3

(d) Chip 4

Fig.4.14. $[Nu_t/Re_D^{1.25}]$ versus $[Gr_D/Re_D^2]$ and linear fit results for each chip

Table 4.4. The linear fit results based on D for each chip

	C_2	SD of C_2	m	SD of m
Chip 1	0.00536	0.00005	0.5201	0.0207
Chip 2	0.00491	0.00007	0.5036	0.0337
Chip 3	0.00407	0.00006	0.4691	0.0319
Chip 4	0.00353	0.00005	0.4082	0.0313
Overall recommended	0.00441	0.00008	0.4766	0.0413

On the other hand, the values of C_3 and m of Eq. (4.7) for the four chips are also determined by using linear fit method in the mixed convection regime is shown in Fig. 4.15. The data SD of C_3 and m are given in Table 4.5. The average standard deviation of C_3 and m are also within 10%.

(a) Chip 1

(b) Chip 2

(c) Chip 3

Fig.4.15. $[Nu_{\ell}/Re_{\ell}^{1.25}]$ versus $[Gr_{\ell}/Re_{\ell}^2]$ and linear fit results for each chip

Table 4.5. The linear fit results based on ℓ for each chip

	C_3	SD of C_3	m	SD of m
Chip 1	0.00321	0.00003	0.5201	0.0207
Chip 2	0.00296	0.00005	0.5036	0.0336
Chip 3	0.00249	0.00004	0.4691	0.0319
Chip 4	0.00219	0.00005	0.4082	0.0431
Overall	0.00268	0.00006	0.4766	0.0429
Türkoğlu & Yücel (1995)	0.0214	0.0039	0.4878	0.0339

The results obtained from the numerical results of Türkoğlu and Yücel (1995) are compared with the present form of equations and found that the average deviation of C_3

and m are about 18%. The correlation equations for mixed convection are given in Eqs. (4.8) and (4.9)

$$\frac{Nu_\ell}{Re_D^{1.25}} = 0.0044 \left(\frac{Gr_D}{Re_D^2} \right)^{0.5} \quad (\text{based on hydraulic diameter, } D), \quad (4.8)$$

$$\frac{Nu_\ell}{Re_\ell^{1.25}} = 0.0027 \left(\frac{Gr_\ell}{Re_\ell^2} \right)^{0.5} \quad (\text{based on heat source length, } \ell), \quad (4.9)$$

$$\frac{Nu_\ell}{Re_\ell^{1.25}} = 0.0214 \left(\frac{Gr_\ell}{Re_\ell^2} \right)^{0.5} \quad (\text{Türkoğlu \& Yücel 1995}) \quad (4.10)$$

The present analyses also been extended to cover the general convection modes (forced, natural and mixed). However, the exponent n of $[Nu_\ell / Re_D^n]$ of Eq. (4.5) has also been selected from different tests with different n , from 0.5 to 2.5. The values of SD of C_2 and m of Eq. (4.5) at different values of n of $[Nu_\ell / Re_D^n]$ are plotted in Fig. 4.16 and the values are given in Table 4.6. It is seen that the values of SD of C_2 and m are minimum at the unique value, $n = 1.15$. Therefore, the exponent for Re_D of Eq. (4.5) is selected as 1.15 (Bhowmik et al. 2005a, 2005b, 2004c).

Table 4.6. Exponents for standard deviation SD , of C_2 and m

Exponent, n	SD of m	SD of C_2
0.5	0.0090	0.01580
0.7	0.0065	0.00295
1	0.0035	0.00021
1.1	0.0031	0.00009
1.2	0.0033	0.00005
1.3	0.0040	0.00003
1.6	0.0073	0.000007
2.0	0.0125	0.0000008
2.5	0.0192	0.00000004

Fig. 4.16. Variations standard deviation with exponent

The variations of $[Nu_t/Re_D^{1.15}]$ with $[Gr_D/Re_D^2]$ for all the present experimental points for average of the four chips are shown in Fig. 4.17. Results show that all the data that were

represented by separate curves for different heat fluxes in previous plots (Fig. 4.8) collapse into a single line when an appropriate value of n is chosen. Thus, it is possible to represent the thermal behaviour with a single correlation that is independent of both heat flux and convection mode (forced or natural).

Fig. 4.17. Variation of $[Nu/Re_D^{1.15}]$ with $[Gr_D/Re_D^2]$

For comparison, Grashof number and Reynolds number are also re-calculated based on the heat source length as shown in Fig. 4.18. It is seen that even though the curves are not close to each other, they are of similar slope. Our Nu_t data are about 35% lower than that of the results of Türkoğlu and Yücel.

Fig. 4.18. Variation of $[Nu_\ell / Re_\ell^{1.15}]$ with $[Gr_\ell / Re_\ell^2]$ for average experimental results

Thus, to cover the three convection modes (forced, natural and mixed), the Eq. (4.5) can be re-written as the following form:

$$\frac{Nu_\ell}{Re_D^{1.15}} = C_4 \left(\frac{Gr_D}{Re_D^2} \right)^m \quad (4.11)$$

The values of C_4 and m in Eq. (4.11) for the four chips are also determined by linear fit method to correlate the laminar flow data. The results are shown in Fig. 4.19 and Table 4.7. The average SD of C_4 and m are about 20% (Bhowmik et al. 2005a).

(a) Chip 1

(b) Chip 2

(c) Chip 3

(d) Chip 4

Fig. 4.19. Linear fit results for each chip.

Table 4.7. Linear fit results for four chips.

	C_4	SD of C_4	m	SD of m
Chip 1	0.0106	0.0022	0.5264	0.1007
Chip 2	0.0095	0.0020	0.5009	0.1030
Chip 3	0.0080	0.0017	0.4948	0.1062
Chip 4	0.0070	0.0015	0.4930	0.1022
Overall recommended	0.0088	0.0018	0.5038	0.1030

From the analysis it is found that there is a higher value of Nu_ℓ for the first chip and a drop in value as the chip number increases. The values of Nu_ℓ of the second, third and fourth chips are about 98%, 82% and 78% of the first chip respectively. Thus, the development of the thermal boundary layer decreases the values of Nu_ℓ with increasing chip number. This observation is similar to the forced convection condition (Tso et al. 1999). Thus, the overall recommended correlation for the four chips can be written as (Bhowmik et al. 2005a, 2005b, 2004c)

$$\frac{Nu_\ell}{Re_D^{1.15}} = 0.0088 \left(\frac{Gr_D}{Re_D^2} \right)^{0.5} \tag{4.12}$$

4.2.4 Natural Convection Heat Transfer

Experiments are performed when there is no flow into the test section. The heater power is increased in steps from 0.1 W/cm² to 0.6 W/cm², and the steady-state conditions are obtained at each condition. The chip temperatures always remain less than 70°C. The

variation of the value of Nu_ℓ that reflects the change of heat transfer coefficient with chip number is shown in Fig. 4.20 at six different heat fluxes.

Fig. 4.20. Variation of Nusselt number with chip number

As seen from this figure, there is a higher value of Nu_ℓ for the first chip and a drop in Nu_ℓ as the chip number increases. It is also found that the Nu_ℓ of the second, third and fourth chips are about 89%, 84% and 83% of the first chip. This conveys the fact that as the chip number increases, the difference between Nu_ℓ for the neighbouring chips become smaller. However, it is deduced that the value of Nu_ℓ decreases as the chip number increases and may reach chip number independency by the third chip, since there is a little change of 1% in values between third and fourth chips. This observation is consistent with the results of Incropera et al. (1986) in turbulent flow, Mahaney et al. (1990) in laminar flow,

and Tso et al. (1999) in both laminar and turbulent flow. The differences in Nu_t between the neighbouring chips are small, because in natural convection, the buoyancy force plays a significant role to increase heat transfer.

The experimental data obtained for all the test cases are then used for the development of empirical correlations. The form of the correlation proposed in Eq. (4.13) based on the correlations provided by Ortega and Lall (1998), MacGregor and Emery (1969), Vliet (1969), Andreozzi et al. (2002), Bhowmik et al. (2005c).

$$Nu_H = C_5 Ra_H^m, \quad (4.13)$$

where $Ra_H = Gr_H Pr$. The values of C_5 and m in Eq. (4.13) for the four chips, where Nusselt number and Rayleigh number are based on channel height, are determined by linear fit method to correlate the natural convection heat transfer data, as shown in Fig. 4.21.

(a) Chip 1

(b) Chip 2

(c) Chip 3

(d) Chip 4

Fig. 4.21. Plot of Nu_H versus Ra_H and linear fit results for four chips

The values of the SD of C_5 and m are tabulated in Table 4.8. It is found that the overall SD of C_5 and m are less than 35% and 8%, respectively.

Table 4.8. Correlation coefficients and exponents and standard deviations

	C_5	SD of C_5	m	SD of m
Chip 1	0.061	0.016	0.307	0.019
Chip 2	0.059	0.025	0.301	0.030
Chip 3	0.059	0.020	0.299	0.025
Chip 4	0.058	0.023	0.296	0.028
Overall recommended	0.060	0.021	0.300	0.025

As there are no other discrete heat sources results available for comparison, the present results are compared with natural convection results of MacGregor and Emery (1969) for heated whole wall. The Eqs. (4.14) and (4.15) are experimental results for constant wall heat flux condition, in the turbulent and laminar flow regime, respectively. Equations (4.16) and (4.17) are for numerical results for constant wall heat flux and constant wall temperature conditions, respectively for laminar flow regime.

$$Nu_H = 0.046 Ra_H^{1/3} \quad \left\{ \begin{array}{l} 10^6 < Ra_H < 10^9 \\ 1 < Pr < 20 \\ 1 < \ell/H < 40 \end{array} \right\}, \quad (4.14)$$

$$Nu_H = 0.42 Ra_H^{0.25} Pr^{0.012} (\ell/H)^{-0.3} \quad \left\{ \begin{array}{l} 10^4 < Ra_H < 10^7 \\ 1 < Pr < 20000 \\ 10 < \ell/H < 40 \end{array} \right\}, \quad (4.15)$$

$$Nu_H = 0.38(\ell/H)^{-0.25} Ra_H^{0.25} \quad \{10^4 < Ra_H\}, \quad (4.16)$$

and
$$Nu_H = 0.25(\ell/H)^{-0.25} Ra_H^{0.25} \quad \{10^{4.5} < Ra_H\}. \quad (4.17)$$

In Eqs. (4.14) - (4.17), the length parameters have been modified to suit our notation. The comparison is shown in Figure 4.22 and it is seen that the experimental results for turbulent regime is closer to our experimental conditions, since the ratio ℓ/H for our experiments is 2 and falls in the range 1 to 40 of Eq. (4.14). The average SD of C_5 and m of our results compared with Eq. (4.14) are given in Table 4.9. On the other hand, the Eq. (4.15) is having a higher range of ℓ/H ratio, 10 to 40, even though the other conditions are similar.

Fig. 4.22. Results for average of four chips and comparison with MacGregor & Emery

From Fig. 4.22, it is seen that all the lines, for both constant heat flux and constant wall temperature conditions are having a higher value of Nu_H than our results that may be due to the different flow and heating conditions.

Table 4.9. Comparison of present results with results of MacGregor & Emery

	Chip 1	Chip 2	Chip 3	Chip 4	Overall
C_5	0.061	0.059	0.059	0.058	0.06
Comparison in C_5	+ 24%	+ 22%	+ 21%	+ 21%	+ 20%
m	0.307	0.301	0.299	0.296	0.3
Comparison in m	- 8.5%	- 10.8%	- 11.2%	- 12.4%	- 11%

As a further comparison, the results for average of the four chips are also compared with constant wall temperature results of Darie et al. (1998). Their correlations, Eq. (4.18) and Eq. (4.19), are proposed for closed and open channel respectively, for a channel with aspect ratio (ℓ/H) of 5, and the comparison with the present study is shown in Fig. 4.23. They have used parameters based on heater wall length, as defined in Eq. (3.8) and Eq. (3.10), and hence we have converted our results to that based on heater length also.

$$Nu_\ell = 0.146Ra_\ell^{0.188} \quad (\text{closed channel}), \quad (4.18)$$

$$Nu_\ell = 0.132Ra_\ell^{0.169} \quad (\text{open channel}) \quad (4.19)$$

Fig. 4.23 Results for average of four chips and comparison with results of Darie et al.

We do not expect good agreement with Darie et al. because they are numerical results for constant wall temperature conditions and based on laminar model. The channel aspect ratio of Darie et al. is 5, but our is 2. Figure 4.23 shows that our results are nearer to the results for closed channel, even though the slope of the curves are similar. Our Nusselt numbers are about 35% and 50% higher than that of the results for closed and open channel, respectively. The heat transfer coefficient is known to be a strong function of aspect ratio and method of heating (MacGregor and Emery 1969).

Considering all the comparisons, it is found that our condition is closer to the condition of Eq. (4.14) and the results are tabulated in Table 4.9 (Bhowmik and Tou 2005b). It is found that the values of the correlation coefficient C_5 are higher, and the value of the

exponent m is slightly lower for all the chips. The overall value of C_5 is about 20% higher and the value of m is about 11% lower than that of the results of heated wall. Finally, it is noted that, even though the results are not close to the reported results that are for different conditions, the slope of our curve is similar to that of the others. The overall recommended correlation for the four chips is (Bhowmik et al. 2005c)

$$Nu_H = 0.06 Ra_H^{1/3}, \quad (4.20)$$

4.3 Studies on Protruding Chips

4.3.1 Effects of Number of Chips

The tests are conducted for protruding heat sources with $\ell=10$ mm, $H/\ell=0.5$ and $B/\ell=0.1$ and $B/\ell=0.2$. The variation of Nusselt number with chip number is shown in Fig. 4.24 for two different heater protrusions, for the experimental condition of 0.3 W/cm^2 . As seen from the figure, there is a drop in Nu_ℓ as the chip number increases and reached a chip-number independency at the third chip, since there is little change in values between the third and fourth chips. These results are similar to those from flush-mounted chips. The percent of Nu_ℓ with number of chips is also listed in Table 4.10.

Fig. 4.24. Variation of Nusselt number with chip number for protruding chips

Table 4.10. Percent of Nu_ℓ with number of chips for protruding chips

	Re_ℓ	$Nu_{\ell 1}$	$Nu_{\ell 2}/Nu_{\ell 1}$	$Nu_{\ell 3}/Nu_{\ell 1}$	$Nu_{\ell 4}/Nu_{\ell 1}$
$B = 1$ mm	910	26.61	84.04%	79.21%	77.52%
	1170	29.52	83.09%	77.44%	74.66%
$B = 2$ mm	910	32.99	90.23%	79.14%	77.42%
	1170	36.14	87.85%	77.12%	74.87%

4.3.2 Effects of Reynolds Number

The Nusselt number as a function of Peclet number based on heat source length for different protrusions is shown in Fig. 4.25. The results of Tso et al. (1999) for vertical channel and Shah and London (1978) for parallel plate ducts are also plotted in the same figure. It is found higher average Nu_ℓ values of 12% and 26% with respect to flush-mounted heater for 1 mm and 2 mm protrusions respectively.

Fig. 4.25. Nusselt number as a function of Peclet number for protruding chips

Due to the heater protrusions, the flow is obstructed by the protrusion heights, causes the flow separation and vortex shedding that leads to increase the average Nu_t . In addition to this due to the heat source protrusion, the flow is obstructed and the thermal boundary layer becomes thin leading to higher heat transfer. This observation is similar to the effects of protrusions on pump-off transient heat transfer studies (Bhowmik and Tou 2005c). The average experimental data of the four chips are agreed with the analytical solution of constant heat flux boundary condition but larger than that of the values of constant wall temperature boundary condition. These observations are also confirmed by the study of Seo and Parameswaran 2002, Garimella and Schlitz 1995.

4.4 Summary and Conclusions

The following conclusions can be achieved based on the results of experimental studies of forced, mixed and natural convection heat transfer from an in-line four flush-mounted and protruded heat sources for water cooling.

1. Upstream thermal boundary affects heat transfer characteristics of downstream. The value of Nu_t decreases as the chip number increases and reaches chip number independency by the third chip.
2. The experimental results indicate that the value of Nu_t is strongly affected by the value of Re_t .
3. Flow regimes for forced, mixed and natural convection can be derived from the heat transfer data.
4. The steady-state mixed convection results having similar slope with Türkoglu and Yücel (1995), the present data are about 35% lower. The results seem to be applicable for general convection regimes: natural, mixed and forced convection.
5. For flush-mounted chips, the value of Nu_t for all four chips fall along 1/3 of the theoretical slope of Pe_t for laminar forced convection. The values of $[Nu_t/Re_D^{1.25}]$ fall along 1/2 of the theoretical slope of $[Gr_D/Re_D^2]$ and $[Gr_t/Re_t^2]$ for laminar mixed convection, and the Nu_H fall along 1/3 of the theoretical slope of Ra_H for natural convection.

6. Results from the present study agree well with the results from the literature for air-cooling as well as liquid cooling of heat sources of similar geometry for flush-mounted chips.

7. The protruding height is found to have a more significant effect on Nu_ℓ . The value of Nu_ℓ for B/ℓ of 0.2 is greater than that for B/ℓ of 0 and 0.1.

Chapter 5 Experimental Results for Transient Heat Transfer

5.1 Introduction

In this chapter, experimental results of time dependent transient heat transfer from four in-line simulated electronic chips in a vertical rectangular channel using water and FC-72 are described and analysed. For water cooling the transient created by turning on the power, turning off the pump, turning on the pump, turning off the power and natural convection with power-on operations are investigated. For FC-72 cooling the transient created by turning off the pump and natural convection with power-on transient operations are investigated. The results cover the range of laminar flow regime. The Reynolds number, based on heat source length ranges from 800 to 2625 and the inlet temperature is about $T_{in} \approx 23^{\circ}\text{C}$. The range of heat flux supplied to chips varies from 1 W/cm^2 to 7 W/cm^2 , and all the four chips are heated simultaneously at constant heat flux. The effects of heat flux, coolant flow rates and geometric parameters such as chip numbers are investigated in the transient regimes. The correlation equations are presented for individual chips as well as for overall data in the transient regime. Finally, comparisons of heat transfer behaviour of water and FC-72 cooling during power-on transient forced and natural convection are investigated.

5.2 Studies on Flush-mounted Chips

5.2.1 Pump-off Transient Heat Transfer

5.2.1.1 Analyses for Water Cooling

To conduct the pump-off transient study, the pump power was shut down during steady-state, and instantly the chip wall temperature increased rapidly. When the chip wall temperature reached about 70°C, all the power was shut down to terminate the experiments, ensuring that there was no phase change in the liquid. However, the chip temperature was well below the design limit of 85°C. A typical experimental chip wall temperature history is shown in Fig. 5.1 for the experimental condition of the values of $Re_t = 1313$, $q'' = 3 \text{ W/cm}^2$. The steady-state and transient regimes are indicated in the figure.

Fig. 5.1. Pump-off transient chip wall temperature history

For analyses the transient heat transfer regime for 150 s are taken after the pump-off, while keeping the heater power is on. The variations of the values of Nu_ℓ and Ra_ℓ as a function of time during the pump-off transient period are displayed in Figs. 5.2(a) and 5.2(b) respectively, for the case of $q'' = 3 \text{ W/cm}^2$ and initial value of $Re_\ell = 1313$. It can be seen that Nu_ℓ decreases continuously with time while the value of Ra_ℓ increases during the entire pump-off transient period. It can be seen that during the transient period, the chip wall temperatures increase rapidly (in Fig. 5.1). After some time, the temperatures begin to increase slowly. This phenomenon indicates that the transition from forced convection to natural convection takes some time because of the inertia of the forced flow. As the natural convection mode comes in, the buoyancy force plays a significant role to increase the average value of Nu_ℓ . The same phenomenon occurs for the test case of q'' from 1 W/cm^2 to 7 W/cm^2 and initial value of Re_ℓ from 800 to 2625.

(a) Nusselt number variation

(b) Rayleigh number variation

Fig. 5.2. Nusselt number and Rayleigh number variations with time

It is found that at the beginning of the pump-off, the values of Nu_t of the second, third and fourth chips are about 97%, 84% and 80% of the first chip respectively, and at the end of experiments (at 150th s) they are about 93%, 83% and 79% of the first chip respectively. This conveys the fact that as the chip number increases, the difference between the neighbouring chips becomes smaller and have reached chip number independency, since a little change of 4% in values between third and fourth chip is obtained (Tso et al. 2004). This observation is similar to the forced convection condition (Tso et al. 1999).

The experimental data obtained for all the test cases are than used for the development of empirical correlations. Chen (2003) examined numerically the thermal boundary effects

for transient natural convection between concentric and vertically eccentric spheres and proposed the form of the correlation as $\overline{Nu_b} = C(Ra_b^*)^m$. Sucec 2002 reported that the transient Nusselt number also depend on time. However, in the present analyses, the effects of non-dimensional time Fo and Rayleigh number Ra_t on Nusselt number Nu_t has been considered. Thus, the proposed correlation is in the following form (Tso et al. 2004, Bhowmik and Tou 2005b)

$$Nu_t = CFo^m(Ra_t)^n \quad (5.1)$$

The exponent n of the value of Ra_t has been determined as described below. The results in Fig. 5.2 are re-plotted in Fig. 5.3(a) by showing the variations of $Nu_t/(Ra_t)^n$ with Fo for different n and for the overall data of the four chips. For each of the six values of n (from 0 to 1.0), the linear fit line (Jaluria 1988) is drawn and the corresponding R-squared values (Andreozzi et al. 2002) (described in Appendix C) are obtained and plotted in Fig. 5.3(b). It is found that maximum R-squared value occurs at about 1/4 that is the generally accepted exponent for the steady laminar natural convection correlations in open-ended channels (Azevedo and Sparrow 1985).

(a) Variation of $Nu_t/(Ra_t)^n$ with Fo for different n

(b) Variation of R-squared value with n

Fig. 5.3. Selection of exponent n for correlation

Having selected the value of n as $1/4$, simplifies the Eq. (5.1) as the following form (Tso et al. 2004)

$$Nu_{\ell} = CFo^m(Ra_{\ell})^{1/4} \tag{5.2}$$

The Fo versus $[Nu_{\ell}/(Ra_{\ell})^{1/4}]$ during transient operations is plotted in Fig. 5.4 for each chip, where curve lines are the experimental results and the straight lines are linear fits. The R-squared values for four chips are tabulated in Table 5.1. It is seen that the values are close to 1 that ensures a good linear fit.

Table 5.1. R-squared values for four chips

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.982	0.991	0.993	0.992

Fig. 5.4. Fo versus $[Nu_{\ell}/(Ra_{\ell})^{1/4}]$ during pump-off transient operation

The above procedures are followed to determine the values of C and m in Eq. (5.2) for each experimental condition and for each chip. The average of these results are taken as the value for each chip. Finally, the relations between the values of C and m with respect to the values of q'' and Re_t for four chips are also determined by the linear fit method and the results are presented in Table 5.2 and Table 5.3. It is seen that the absolute values of b and m are always less than 0.02, it can be concluded that the values of C and m are independent on the q'' and initial Re_t .

Table 5.2. Linear fit results [$C = (a + b \times q'')$, $m = (a + b \times q'')$] for four chips

	$C = (a + b \times q'')$					$m = (a + b \times q'')$				
	a	$SD(a)$	b	$SD(b)$	$R\text{-sq.}$	a	$SD(a)$	b	$SD(b)$	$R\text{-sq.}$
Chip 1	0.702	0.018	-0.774	0.019	0.997	-0.262	0.015	0.004	0.001	0.812
Chip 2	0.635	0.015	-0.783	0.018	0.998	-0.263	0.028	0.004	0.001	0.801
Chip 3	0.574	0.018	-0.784	0.022	0.997	-0.264	0.019	0.005	0.002	0.827
Chip 4	0.562	0.018	-0.802	0.010	0.999	-0.271	0.022	0.005	0.002	0.853

Table 5.3. Linear fit results [$C = (a + b \times Re_t)$, $m = (a + b \times Re_t)$] for four chips

	$C = (a + b \times Re_t)$					$m = (a + b \times Re_t)$				
	a	$SD(a)$	b	$SD(b)$	$R\text{-sq.}$	a	$SD(a)$	b	$SD(b)$	$R\text{-sq.}$
Chip 1	0.406	0.128	-0.035	0.004	0.882	-0.118	0.011	-5E-5	2E-6	0.871
Chip 2	0.395	0.260	-0.041	0.008	0.873	-0.132	0.022	-6E-5	2E-6	0.892
Chip 3	0.329	0.116	-0.042	0.005	0.891	-0.135	0.019	-6E-5	2E-6	0.913
Chip 4	0.324	0.082	-0.052	0.003	0.905	-0.159	0.021	-6E-5	2E-6	0.936

The mean values of C and m for the four chips as well as for overall are given in Table 5.4. The overall recommended correlation for the present conditions is given in Eq. (5.3) where the average standard deviation of C and m are less than 2% (Tso et al. 2004).

$$Nu_t = 0.31(fo)^{-0.23}(Ra_t)^{1/4}. \quad (5.3)$$

Table 5.4. Mean values of C and m for four chips

	Mean (C)	SD of C	Mean (m)	SD of m
Chip 1	0.3493	0.0029	-0.2164	0.0034
Chip 2	0.3257	0.0034	-0.2229	0.0043
Chip 3	0.2896	0.0025	-0.2248	0.0035
Chip 4	0.2772	0.0029	-0.2384	0.0041
Overall recommended	0.3105	0.0030	-0.2262	0.0035

5.2.1.2 Analyses for FC-72 Cooling

All the thermophysical properties (3M Company, 1986) were allowed to vary with temperature as described in Appendix B. To conduct the pump-off transient study, the pump power was shut down during steady-state, and instantly the chip wall temperature increased rapidly. When the chip wall temperature reached about 70°C, all the power was shut down to terminate the experiment, ensuring that there was no phase change in the liquid. A typical experimental chip wall temperature history is shown in Fig. 5.5 for the experimental condition of $Re_\ell = 1313$, $q'' = 3 \text{ W/cm}^2$. The steady-state and transient operations are indicated in the figure.

Fig. 5.5. Transient temperature history

For analyses, the data up to 150 s are taken as the transient heat transfer regime, while heater power is on. The variations of the values of Nu_t and Ra_t as a function of time in the transient regime are displayed in Figs. 5.6(a) and 5.6(b) respectively for the case of $q'' = 3 \text{ W/cm}^2$ and initial value of $Re_t = 1313$. It can be seen that Nu_t decreases continuously with time while the Ra_t increases during the entire pump-off transient operation. The same phenomenon occurs for the test case of q'' from 1 W/cm^2 to 7 W/cm^2 and initial Reynolds number Re_t from 800 to 2625.

(a) Nusselt number variations

(b) Rayleigh number variations

Fig. 5.6. Nusselt number and Rayleigh number variations with time

It is found that at the start of pump-off and at the end of experiments (150th s) the values of Nu_t of the second, third and fourth chips are about 99%, 97% and 96% of the first chip respectively.

The experimental data obtained for all the test cases are then used for the development of empirical correlations. For transient natural convection Chen (2003) and Sucec (2002) observed the influence of Rayleigh number and time dependent Fourier number on Nusselt number. However, the proposed correlation can be written as the following form (Tso et al. 2004, Bhowmik and Tou 2005b)

$$Nu_t = CFo^m(Ra_t)^n \tag{5.4}$$

The exponent n of the Ra_t of Eq. (5.4) is taken to be 1/4, which has generally been accepted for transient natural convection heat transfer for water cooling in (Tso et al. 2004, Bhowmik and Tou 2005b). Thus, the Eq. (5.4) can be simplifies as the following form

$$Nu_t = C(Fo)^m(Ra_t)^{1/4}, \tag{5.5}$$

The Fo versus $[Nu_t/(Ra_t)^{1/4}]$ in the transient regime is plotted in Fig. 5.7 for four chips, where curve lines are the experimental data and the straight lines are linear fits. The corresponding R-squared (Andreozzi et al. 2002) values for four chips are tabulated in Table 5.5. It is seen that the values are close to unity that ensures a good linear fit.

Table 5.5. R-squared values for four chips

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.953	0.953	0.952	0.952

Fig. 5.7. Fo versus $[Nu_l/(Ra_l)^{1/4}]$ during the transient regime

The above procedures are also help to determine the values of C and m in Eq. (5.5) for each experimental condition and for each chip. The average of these results are taken as the value for each chip. Finally, the relations between the values of C and m with respect to q'' and Re_l are also determined by the linear fit method. It is found that in all cases the slopes are less than 0.02. Thus, it can be concluded that the values of C and m are independent on the q'' and initial Re_l . The overall recommended correlation is given in Eq. (5.6), where the standard deviations of C and m are about 8%. The summary of the results is also given in Table 5.6.

$$Nu_l = 0.28(Ra_l)^{1/4}(Fo)^{-0.05}. \tag{5.6}$$

Table 5.6. Mean values of C and m for four chips

	Mean, C	SD of C	Mean, m	SD of m
Chip 1	0.282	0.022	-0.052	0.001
Chip 2	0.280	0.021	-0.053	0.001
Chip 3	0.279	0.032	-0.054	0.002
Chip 4	0.276	0.023	-0.055	0.004
Overall recommended	0.279	0.025	-0.054	0.003

It can be seen from Table 5.4 and Table 5.6 the values of C and m are not same for water and FC-72. The values of C and m values for water cooling are higher than that of the values for FC-72 cooling. This can be elucidated that after the pump-off the chip temperatures will increase faster in FC-72 than water. In other way the transient heat transfer in FC-72 starts immediately after the pump-off.

5.2.2 Power-on Transient Heat Transfer

To conduct the power-on transient experiments, the electrical power to the chips was turned on after reaching the steady-state temperatures. Then the chip temperature will increase sharply and will reach steady-state with increase of time. The working fluid used here was water. Fig. 5.8 shows the typical temperature history for the four chips with indicating power-on transient operation for the condition of $Re_t = 1575$ and $q'' = 5 \text{ W/cm}^2$.

Fig. 5.8. Power-on transient chip wall temperature history

For analyses, the transient heat transfer data for 75 s are taken after the power-on, while keeping the pump power is on. The variations of values of Nu_t and Pe_t as a function of time during the power-on transient period are displayed in Figs. 5.9(a) and 5.9(b) for the case of $Re_t = 1575$ and $q'' = 5 \text{ W/cm}^2$. It can be seen that the values of Nu_t and Pe_t decrease continuously with time during the entire power-on transient period. The same phenomenon occurs for the other test cases. It can be seen from Fig. 5.9(a), at the beginning of the heater power-on, the values of Nu_t of the second, third and fourth chips are about 95%, 81% and 77% of the first chip respectively, and at the end of experiments (75th s) they are about 75%, 64% and 60% of the first chip respectively. This conveys the fact that as the chip number increases, the difference between the neighbouring chips become smaller and has reached chip number independency, since a little difference of 4% in values between the third and fourth chip.

From Fig. 5.9(b) it is observed that Pe_ℓ decreases continuously with time during the entire transient operation. Analysing this situation, it can be seen that the value of Pe_ℓ decreases with time when the initial value of $Re_\ell = 1575$, is constant. This may be due to fact of changing fluid properties (Perry 1963 and Greankoplis 1972) with temperature. It is worth mentioning here that all the fluid properties are allowed to vary with temperature and are evaluated based on bulk fluid temperature T_b , defined in Eq. (3.1). At the heater power-on, condition the chip temperature increases and the density ρ of the liquid decreases. It can be seen that in Eq. (3.5), Re_ℓ is the function of density ρ and in Eq. (3.15), Pe_ℓ is the function of Re_ℓ . However, the fluid density can play significant role to decrease the Pe_ℓ with time.

(a) Nusselt number variation

(b) Peclet number variation

Fig. 5.9. Nusselt number and Peclet number variations with time

Abu-Hijleh et al. (2003) presented an analysis of transient conjugate forced convection in horizontal channel and observed the Peclet number has significant effect on heat transfer. Zueco et al. (2004) also observed that heat transfer increases as the Peclet number decreases. Sucec (2002) found that transient Nusselt number depends on time. However, in the present analyses, the effect of Peclet number (Pe_t) and non-dimensional time (Fo) on Nusselt number (Nu_t) during the transient operation is considered. Thus, the correlation equation can be proposed as the following form (Bhowmik and Tou 2005a, 2005d, Bhowmik et al. 2003, 2004b):

$$Nu_t = C(Pe_t)^n Fo^m. \quad (5.7)$$

The exponent n of the Peclet number of Eq. (5.7) is taken to be $1/3$ that has generally been accepted for steady-state forced convection laminar flow by Tso et al. (1999) and Shah and London (1978). Therefore the Eq. (5.7) can be simplified to

$$Nu_\ell = C(Pe_\ell)^{1/3} Fo^m, \tag{5.8}$$

The coefficient C , and exponent m , of the correlation Eq. (5.8) for the four chips are determined by using the linear fit method and checking independency of the values of C and m on the initial heat flux and Re_ℓ to correlate the transient heat transfer data for the four chips. To observe the linear fit lines, the R-squared values for four chips are determined and tabulated in Table 5.7. It is found that the R-squared values are near to 1 that ensures the good linear fit. The Fo versus $[Nu_\ell/(Pe_\ell)^{1/3}]$ plot is shown in Fig. 5.10 for the conditions of $Re_\ell = 1575$ and $q'' = 5 \text{ W/cm}^2$. Where the curve lines are for experimental data and the straight lines are for linear fits.

Table 5.7. R-squared Values for Four Chips

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.963	0.982	0.961	0.964

Fig. 5.10. Fo versus $[Nu_t/Pe_t]^{1/3}$ during the power-on transient operation

Finally, the relations between C and m with respect to q'' and Re_t for each chip are also determined by the linear fit method. It is found that in all cases the slopes are always less than 0.02.

The mean values of C and m for the four chips as well as for overall data are given in Table 5.8. The overall recommended correlation for the present conditions is given in Eq. (5.9) where the average standard deviation of C and m are less than 4% (Bhowmik and Tou 2005d, Bhowmik et al. 2004b)

$$Nu_t = 0.945(Pe_t^{1/3})Fo^{-1/2}. \quad (5.9)$$

Table 5.8. Mean values of C and m for four chips

	Mean (C)	SD of C	Mean (m)	SD of m
Chip 1	1.148	0.011	-0.496	0.008
Chip 2	0.977	0.010	-0.523	0.008
Chip 3	0.896	0.010	-0.531	0.007
Chip 4	0.759	0.012	-0.546	0.008
Overall recommended	0.945	0.013	-0.524	0.008

5.2.3 Pump-on Transient Heat Transfer

The pump-on transient experiments for water cooling are conducted initially at the pump-off and the heater power-on conditions. After reaching the temperature at design limit (70°C) the pump power is turned on. Then the temperature of the chips decrease sharply.

Fig. 5.11. Pump-on transient chip wall temperature history

Figure 5.11 shows the temperature history for the four chips, indicating the pump-on transient operation for the experimental condition of the values of $Re_\ell = 1260$ and $q'' = 5 \text{ W/cm}^2$. For analyses, the transient heat transfer data for 75 s are taken after the pump-on, while keeping the heater power-on. The variations of the values of Nu_ℓ and Pe_ℓ as a function of time during the pump-on transient period are displayed in Figs. 5.12(a) and 5.12(b) for the conditions of $Re_\ell = 1260$ and $q'' = 5 \text{ W/cm}^2$.

From Fig. 5.12 it can be seen that the values of Nu_ℓ and Pe_ℓ increases continuously with time during the entire pump-on transient period. The same phenomenon occurs for all the test case of q'' from 1 W/cm^2 to 7 W/cm^2 and Re_ℓ from 800 to 2625.

(a) Nusselt number variations

(b) Peclet number variations

Fig. 5.12. Nusselt number and Peclet number variations with time

It is observed that, the fluid starts to flow when the pump is turned on. The value of Pe_t increases with time and the forced convection becomes the dominant mode of heat transfer. As the flow velocity increases the value of Nu_t also increases with time that may be due to the decrease of chip wall temperature. At the beginning of the pump-on, the values of Nu_t of the second, third and fourth chips are about 91%, 85% and 82% of the first chip respectively, and at the end of experiments (75th s) they are about 93%, 83% and 80% of the first chip respectively. This also conveys the fact that as the chip number increases, the difference between the neighbouring chips become smaller and have reached a chip number independency, since there is a little change of 3% in values between third and fourth chip at 1st and 75th s.

The experimental data obtained for all test cases are then used for the development of empirical correlations. It is found that during the entire pump-on transient operation, the Pe_ℓ and time dependent Fo have influence on Nu_ℓ (Abu-Hijleh et al. 2003, Zueco et al. 2004, Sucec 2002). However, the proposed form of the correlation can be written as (Bhowmik and Tou 2005a, 2005d, Bhowmik et al. 2003, 2004b)

$$Nu_\ell = C(Pe_\ell)^n Fo^m. \tag{5.10}$$

The value of the exponent n of the Peclet number of Eq. (5.10) is taken to be 1/3, which has generally been accepted for steady-state forced convection for laminar flow (Tso et al. 1999, 2004 and Shah and London 1978). Selection of the value of n as 1/3 simplifies Eq. (5.10) as the following form (Bhowmik and Tou 2005a)

$$Nu_\ell = C(Pe_\ell)^{1/3} Fo^m. \tag{5.11}$$

The Fo versus $[Nu_\ell/(Pe_\ell)^{1/3}]$ in the transient regime is plotted in Fig. 5.13 for each chip, where curve lines are for the experimental data and the straight lines are for linear fits. The corresponding R-squared values for four chips are tabulated in Table 5.9. It is seen that the values are close to 1 ensuring a good linear fit.

Table 5.9. R-squared values for four chips

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.981	0.983	0.974	0.973

Fig. 5.13. Fo versus $[Nu_t/(Pe_t)]^{1/3}$ during the transient operation

The above procedures are followed to determine the values of C and m in Eq. (5.11) for each experimental condition as well as for each chip. The average of these results are taken as the value for each chip. Finally, the relations between C and m with respect to q'' and Re_t for each of the four chips are also determined by the linear fit method. It is found that in all cases the slopes are always less than 0.02. Thus, it can be concluded that the C and m are independent on the values of q'' and Re_t . The mean values of C and m for the four chips as well as for overall data are given in Table 5.10. The overall recommended correlation for the present conditions is given in Eq. (5.12) where the average standard deviation of C and m are about 13% and 3% respectively (Bhowmik and Tou 2005a).

$$Nu_t = 3.5(Pe_t^{1/3})(Fo)^{1/4} \quad (5.12)$$

Table 5.10. Mean values of C and m for four chips

	Mean (C)	SD of C	Mean (m)	SD of m
Chip 1	4.408	0.519	0.274	0.006
Chip 2	3.407	0.520	0.251	0.006
Chip 3	3.195	0.519	0.247	0.005
Chip 4	2.954	0.519	0.235	0.006
Overall recommended	3.491	0.519	0.252	0.006

5.2.4 Power-off Transient Heat Transfer

The wall temperature of the simulated chips were allowed to reach steady-state. The power of the chips was turned off while keeping the pump-on. The temperature of the chips will decrease sharply and all the readings are taken at this level. The working fluid used here was water. Figure 5.14 shows the temperature history, indicating the steady-state and the power-off transient operations for the experimental condition of $Re_t = 1575$, $q'' = 6 \text{ W/cm}^2$.

Fig. 5.14. Typical temperature history

For analyses, after the power-off, the data for 75 s are taken as the transient heat transfer regime. The variations of the values of Nu_ℓ and Pe_ℓ as a function of time during the power-off transient period are displayed in Figs. 5.15(a) and 5.15(b) for the experimental condition of $Re_\ell = 1575$, $q'' = 6 \text{ W/cm}^2$.

From Fig. 5.15 it can be seen that the values of Nu_ℓ and Pe_ℓ increases continuously with time during the entire power-off transient period. The same phenomenon occurs for all the test case of q'' from 1 W/cm^2 to 7 W/cm^2 and Re_ℓ from 800 to 2625.

(a) Nusselt number variations

(b) Peclet number variations

Fig. 5.15. Nusselt number and Peclet number variations with time

It can be seen that the values of Nu_t of the four chips at the beginning of power-off are close to each other and they diverge with time. However, the values of Nu_t increase with

time, which is due to the chip wall temperature decreasing. It is found that at the beginning of the power-off, the values of Nu_ℓ of the second, third and fourth chips are about 95%, 81% and 77% of the first chip respectively, and at the end of experiments (at 75th second) they are about 75%, 64% and 60% of the first chip respectively. This also conveys the fact of chip number independency, as the difference between the neighbouring chips is small.

To develop the empirical correlations, a simple form is proposed as in Eq. (5.13), where the Nusselt number Nu_ℓ is a function of Peclet number Pe_ℓ and non-dimensional time Fo during the transient operations (Bhowmik and Tou 2005a, 2005d, Bhowmik et al. 2003, 2004b)

$$Nu_\ell = C(Pe_\ell)^n Fo^m. \quad (5.13)$$

The exponent n of the Pe_ℓ of Eq. (5.13) is taken to be 1/3, which has generally been accepted for steady-state forced convection for laminar flow (Tso et al. 1999 and Shah and London 1978). Selection of n as 1/3 simplifies Eq. (5.13) as the following form:

$$Nu_\ell = C(Pe_\ell)^{1/3} Fo^m, \quad (5.14)$$

The coefficient C and exponent m of Eq. (5.14) for the four chips are determined by using the linear fit method and checking independence of C and m on the initial heat flux and Re_ℓ to correlate the transient heat transfer data for the four chips. The Fo versus $[Nu_\ell/(Pe_\ell)^{1/3}]$ plot is shown in Fig. 5.16 for the conditions of $Re_\ell = 1575$, $q'' = 6 \text{ W/cm}^2$. To observe the linear fit lines, the R-squared values for four chips are determined and

tabulated on Table 5.11. It is found that the R-squared values are near to 1; this means that good linear fits are obtained.

Table 5.11. R-squared values for four chips.

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.954	0.953	0.945	0.963

Fig. 5.16. Fo versus $[Nu_t/(Pe_t)]^{1/3}$ during the power-off transient period

The relation of C and m of Eq. (5.14) with respect to q'' and Re_t for the four chips are also determined and found that the slopes are less than 0.02 in all the cases, showing that the values of C and m are independent on the initial values of q'' and Re_t . The overall recommended correlation is given in Eq. (5.15) and the values of SD of C and m are about 6% and 8% respectively. The summary of the results are given in Table 5.12 (Bhowmik et al. 2003).

$$Nu_{\ell} = 25.44(Pe_{\ell})^{1/3}(Fo)^{0.41} \tag{5.15}$$

Table 5.12. Mean values of C and m for four chips

	Mean (C)	SD of C	Mean (m)	SD of m
Chip 1	35.687	2.325	0.453	0.021
Chip 2	24.594	1.473	0.423	0.019
Chip 3	23.308	1.341	0.415	0.018
Chip 4	21.154	1.267	0.362	0.023
Overall recommended	25.436	1.702	0.413	0.031

5.2.5 Power-on Transient Natural Convection Heat Transfer

5.2.5.1 Analyses for Water Cooling

All experiments are performed when there is no flow into the test section. The temperature at the channel inlet is about 21°C. The heater power is increased in steps from 0.1 W/cm² to 0.6 W/cm². The configuration of simulated chips tested in this study is always flush-mounted and all the four chips are heated simultaneously at constant heat flux. Figure 5.17 shows the temperature history of the four chips, indicating the power-on transient operation.

Fig. 5.17. Transient temperature history

For analyses, after the power-on, the data for 70 s are taken as the transient heat transfer regime. The variations of the values of Nu_t and Ra_t as a function of time in the transient regime are displayed in Figs. 5.18(a) and 5.18(b) respectively for the experimental condition of $q'' = 0.2 \text{ W/cm}^2$.

Fig. 5.18. Nusselt number and Rayleigh number variations with time

From Fig. 5.18 it can be seen that the value of Nu_ℓ decreases continuously with time while the value of Ra_ℓ increases during the entire transient operation. The same phenomenon occurs for the test case of q'' from 0.1 W/cm^2 to 0.6 W/cm^2 . It is also found

that at the beginning of power-on transient, the values of Nu_t of the second, third and fourth chips are about 71%, 67% and 65% of the first chip respectively, and at the end of experiments (at 70th s) they are about 88%, 83% and 81% of the first chip respectively. This also ensures the chip number independency as the difference between the neighboring chips is small (2%) (Bhowmik and Tou 2005b).

The experimental data obtained for all the test cases are then used for the development of empirical correlations. Since Rayleigh number Ra_t and non-dimensional time Fo have an influence on Nusselt number Nu_t (Chen 2003 and Sucec 2002). The correlation equation can be proposed as the following form (Bhowmik and Tou 2005b, Bhowmik et al. 2004a, Tso et al. 2004)

$$Nu_t = C Fo^m (Ra_t)^n \quad (5.16)$$

The exponent n of the Ra_t of Eq. (5.16) is taken to be 1/4 (Bhowmik and Tou 2005b, Tso et al. 2004), which has generally been accepted for the steady laminar natural convection correlations in open-ended channels (Azevedo and Sparrow 1985). Selection of $n = 1/4$ simplifies the Eq. (5.16) as the following form:

$$Nu_t = C (Fo)^m (Ra_t)^{1/4}, \quad (5.17)$$

The Fo versus $[Nu_t/(Ra_t)^{1/4}]$ in the transient regime is plotted in Fig. 5.19 for each chip, where curve lines are the experimental results and the straight lines are linear fits. The corresponding R-squared values for four chips are tabulated in Table 5.13. It is seen that the values are close to 1 that ensures a good linear fit.

Table 5.13. R-squared values for four chips

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.972	0.981	0.974	0.973

Fig. 5.19. Fo versus $[Nu_t/(Ra_t)^{1/4}]$ during the transient regime

The above procedures are followed to determine the values of C and m in Eq. (5.17) for each experimental condition and for each chip. The average of these results are taken as the value for each chip. Finally, the relations between C and m with respect to q'' are also determined by the linear fit method. It is found that in all cases the slopes are less than 0.04. Thus, it can be concluded that the values of C and m are independent on the q'' . The overall recommended correlation is shown in Eq. (5.18) where the value of SD of C and m about 2% and 3% respectively. The summary of the results is given in Table 5.14 (Bhowmik and Tou 2005b).

$$Nu_\ell = 1.06(Ra_\ell)^{1/4}(Fo)^{-0.47}. \tag{5.18}$$

Table 5.14. Mean values of C and m for four chips

	Mean, C	SD of C	Mean, m	SD of m
Chip 1	1.265	0.013	-0.438	0.009
Chip 2	1.055	0.015	-0.448	0.011
Chip 3	0.991	0.013	-0.456	0.009
Chip 4	0.912	0.016	-0.521	0.012
Overall recommended	1.060	0.014	-0.470	0.014

5.2.5.2 Analyses for FC-72 Cooling

All experiments are performed when the pump is off and heater is on. The temperature at the channel inlet is about 21°C. The heater power is increased in steps from 0.1 W/cm² to 0.6 W/cm². The configuration of simulated chips tested in this study is always flush-mounted and all the four chips are heated simultaneously at constant heat flux. Figure 5.20 shows the transient temperature history of the four chips.

Fig. 5.20. Transient temperature history

For analyses, the data for 70 s are taken after the power-on, as the transient heat transfer regime. The variations of the values of Nu_t and Ra_t as a function of time in the transient regime are displayed in Figs. 5.21(a) and 5.21(b) respectively for the experimental condition of $q'' = 0.3 \text{ W/cm}^2$.

(a) Nusselt number variations

(b) Rayleigh number variations

Fig. 5.21. Nusselt number and Rayleigh number variations with time

From Fig. 5.21(a) it can be seen that the value of Nu_ℓ decreases continuously with time while the value of Ra_ℓ increases during the entire transient operation. The same phenomenon occurs for the test case of q'' from 0.1 W/cm^2 to 0.6 W/cm^2 . It is also found that at the start of transient, the values of Nu_ℓ of the second, third and fourth chips are about 67%, 52% and 48% of the first chip respectively, and at the end of experiments (at 70^{th} s) they are about 90%, 82% and 78% of the first chip respectively. This also conveys the fact of chip number independency as the difference between the neighboring chips is small (4%).

The experimental data obtained for all the test cases are then used for the development of empirical correlations. For transient natural convection Chen (2003) and Succi (2002)

found the influence of Rayleigh number and time dependent Fourier number on Nusselt number. The proposed form of correlation can be written as

$$Nu_t = CFo^m(Ra_t)^n \tag{5.19}$$

The exponent n of the Ra_t of Eq. (5.19) is taken to be 1/4 (Tso et al. 2004, and Bhowmik and Tou 2005b). Selection of $n = 1/4$ simplifies the Eq. (5.19) as the following form

$$Nu_t = C(Fo)^m(Ra_t)^{1/4}, \tag{5.20}$$

The Fo versus $[Nu_t/(Ra_t)^{1/4}]$ in the transient period is plotted in Fig. 5.22 for four chips, where curve lines are the experimental data and the straight lines are linear fits. The corresponding R-squared values for four chips are tabulated in Table 5.15. It is seen that the values are close to 1 that ensures a good linear fit.

Table 5.15. R-squared values for four chips

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.968	0.957	0.950	0.946

Fig. 5.22. Fo versus $[Nu_t / (Ra_t)^{1/4}]$ during the transient regime

The above procedures are also help to determine the values of C and m in Eq. (5.20) for each experimental condition and for each chip. The average of these results are taken as the value for each chip. Finally, the relations between C and m with respect to q'' are also determined by the linear fit method. It is found that in all cases the slopes are less than 0.02. Thus, it can be concluded that the values of C and m are independent on the q'' . The overall recommended correlation is shown in Eq. (5.21) where the standard deviations for C and m are about 5%. The summary of the results is given in Table 5.16.

$$Nu_t = 0.47(Ra_t)^{1/4}(Fo)^{-0.35}. \quad (5.21)$$

Table 5.16. Mean values of C and m for four chips

	Mean, C	SD of C	Mean, m	SD of m
Chip 1	0.492	0.015	-0.283	0.013
Chip 2	0.489	0.013	-0.311	0.014
Chip 3	0.463	0.011	-0.365	0.016
Chip 4	0.441	0.017	-0.451	0.018
Overall recommended	0.471	0.015	-0.352	0.015

It can be seen from Table 5.14 and Table 5.16 the value of C and m are not same for four chips. The value of C for water-cooling are higher than that of the values for FC-72 cooling. This can be elucidated that after the power-on the chip temperatures will increase faster in FC-72 than water. In other way, the transient heat transfer in FC-72 starts immediately after the turning on the heater power.

5.3 Comparison of Water and FC-72 Results

5.3.1 Power-on Transient Natural Convection Heat Transfer

In this section, the results for FC-72 are compared with the results of water for four flush-mounted chips. Figure 5.23 shows the experimental results of water and FC-72 in the transient natural convection regime for the experimental condition of $q'' = 0.3 \text{ W/cm}^2$.

It can be seen from Fig. 5.23, that at the start of power-on transient the value of Nu_t for FC-72 of the first chip is about 5% higher and the second, third and fourth chips are about

14%, 11% and 15% lower than those of water data respectively. At the end of experiments (at 70th s), the first, second, third and fourth chips are about 36%, 33%, 33% and 34% higher than those of water data, respectively. However, an overall increase of 30% in values is obtained by using FC-72. Samant and Simon (1989), Garimella and Schlitz (1995) also obtained this difference between water and FC-72.

Fig. 5.23. Comparison of FC-72 and water in transient natural convection regime

5.3.2 Power-on Transient Heat Transfer

Figure 5.24 shows the experimental results for water and FC-72 in the power-on transient regime for the experimental condition of $Re_t = 1575$ and $q'' = 5 \text{ W/cm}^2$. It can be seen that at the beginning of transient heat transfer the value of Nu_t of FC-72 of the first, second, third and fourth chips are about 34%, 43%, 50% and at the end of experiments (75th s),

74%, 75%, 76% and 78% higher than those of water data respectively. However, an overall increase of 70% in values is obtained by using FC-72 (Bhowmik and Tou, 2005e).

Fig. 5.24. Comparison of FC-72 and water in power-on transient regime

5.4 Studies on Protruding Chips

5.4.1 Pump-off Transient Heat Transfer

The tests are conducted for protruding heat sources with $\ell=10$ mm, $H/\ell=0.5$ and $B/\ell=0.1$ and $B/\ell=0.2$. The variations of the value of Nu_t with time during the pump-off transient period are displayed in Fig. 5.25 for four chips and for different B/ℓ ratio, for the conditions of $q'' = 3$ W/cm² and initial $Re_t = 1313$.

Fig. 5.25. Nusselt number variations during pump-off transient operations

It can be seen from Fig. 5.25 that the value of Nu_t decreases continuously with time. The same phenomenon occurs for all experimental conditions and for four chips. The percentage values of Nu_t of the four chips are also listed in Tables 5.17 and 5.18. It is observed that the difference between the neighbouring chips is small and reached chip number independency, since a little change of 4% in values between third and fourth chip is obtained.

Table 5.17. The percentage value of Nu_ℓ with respect to chip 1 at 1st s

	$Nu_{\ell 1}$	$Nu_{\ell 2}/Nu_{\ell 1}$	$Nu_{\ell 3}/Nu_{\ell 1}$	$Nu_{\ell 4}/Nu_{\ell 1}$
$B = 0$ mm	35.92	97.43%	83.69%	79.89%
$B = 1$ mm	85.15	88.52%	75.35%	71.46%
$B = 2$ mm	125.89	81.07%	74.47%	70.30%

Table 5.18. Table 5.17. The percentage value of Nu_ℓ with respect to chip 1 at 150th s.

	$Nu_{\ell 1}$	$Nu_{\ell 2}/Nu_{\ell 1}$	$Nu_{\ell 3}/Nu_{\ell 1}$	$Nu_{\ell 4}/Nu_{\ell 1}$
$B = 0$ mm	25.86	93.04%	82.99%	78.72%
$B = 1$ mm	29.07	82.94%	75.71%	71.11%
$B = 2$ mm	31.51	81.85%	74.58%	70.18%

It is also found that at the beginning of pump-off, a higher average Nu_ℓ values of 54% and 70% are achieved with respect to flush-mounted heater for 1 mm and 2 mm protrusions respectively and at the end of experiments (150th s) they are about 5% and 12% respectively (Bhowmik and Tou 2005c).

The experimental data obtained for all test cases are then used for the development of empirical correlations. The form of the equation is recommended as follows:

$$\frac{Nu_\ell}{C(Fo)^m (Ra_\ell^*)^{1/4}} = m_1 \left(\frac{B}{\ell} \right) + C_1 \quad (5.22)$$

The C and m of equation (5.22) are obtained from the results of flush mounted case in Table 5.4 (Tso et al. 2004) and are tabulated in Table 5.19. For the protruding case the C_1 and m_1 are also determined by the linear fit method and the mean values are given in

Table 5.19. Figure 5.26 shows the linear fit results for four chips (Bhowmik and Tou, 2005c).

Fig. 5.26. Linear fit results for four chips.

Table 5.19. Mean Values of C , C_1 and m , m_1 for Four Chips

	C	m	C_1	m_1
Chip1	0.3493	-0.2164	0.8322	2.1535
Chip2	0.3257	-0.2229	0.8159	1.8645
Chip3	0.2896	-0.2248	0.7988	1.8630
Chip4	0.2772	-0.2384	0.7380	1.3690
Overall recommended	0.3105	-0.2262	0.7962	1.8125

To observe the linear fit lines, the R-squared values of Fig. 5.26 are obtained and tabulated in Table 5.20. It is seen that the R-squared vales are close to 1 ensuring a good linear fit. Thus the overall recommended correlation for the present experimental conditions is given as (Bhowmik and Tou, 2005c)

$$\frac{Nu_\ell}{0.31(Fo)^{-0.23}(Ra_\ell^*)^{1/4}} = 1.81\left(\frac{B}{\ell}\right) + 0.80 \tag{5.23}$$

Table 5.20. R-squared Values for Four Chips

	Chip 1	Chip 2	Chip 3	Chip 4
R-squared value	0.996	0.984	0.998	0.996

5.4.2 Pump-on Transient Heat Transfer

In the transient regime, the data obtained for four in-line protruding chips are converted to $[Nu_\ell/(Pe_\ell)^{1/3}]$ and plotted against Fo as shown in Fig. 5.27. For 1 mm protrusion, the average values of Nu_ℓ of the second, third and fourth chips are about 84%, 77% and 73%

of the first chip respectively, and at the 2 mm protrusion they are about 77%, 68% and 64% of the first chip respectively. However, the differences between the neighbouring chips become small and reached chip number independency. The higher average Nu_t values of 36% and 51% are achieved with respect to flush-mounted heaters for 1 mm and 2 mm protrusions respectively (Bhowmik and Tou 2005c).

Fig. 5.27. Transient results at different protrusion

Since the upstream and downstream chips have the similar heat transfer data at different experimental conditions, the average data of all the four chips for different protrusions are also correlated by Eq. (5.11) and shown in Fig. 5.27. For different protruding heights ($B = 1, 2$ mm), the C and m in Eq. (5.11) are determined by linear fit method, for each experimental condition and for each chip. The average of these results are taken as the value for each chip and are given in Table 5.21. The R-squared values for all cases are about 0.95 and the average standard deviation (SD) of experimental data are about 15% (Bhowmik and Tou 2005c).

Table 5.21. Mean values of C and m in Eq. (5.11) for protruding chips

	1 mm protrusion		2 mm protrusion	
	C	m	C	m
Chip 1	7.219	0.383	10.216	0.415
Chip 2	5.871	0.348	9.968	0.389
Chip 3	5.666	0.341	9.263	0.384
Chip 4	5.477	0.309	8.777	0.328

5.5 Effects of Protrusions

From Figs. 5.25 and 5.27, it is seen that the heat transfer data for protruding chips are higher than those of flush-mounted heaters. It is worth mentioning here that the difference between the values of Nu_t of the flush-mounted and protruding chips increase with time for pump-on but decrease with time for pump-off transient operations. In the pump-on transient regime, with increasing time forced convection plays a significant role

to increase the average Nu_t . Nevertheless, in the pump-off transient regime, with increasing time natural convection plays a significant role to increase the average Nu_t . In the two cases, it can be observed that the value of Nu_t increases with the increase of protrusion heights. This may be due to the combined effects of fluid impingement upon the leading edge and the reattachment of the boundary layer upon the top surfaces of the chips. Moreover, the flow is obstructed by the protrusion heights, causes the flow separation and vortex shedding that also leads to increase the average Nu_t . These results are consistent with the results of Seo and Parameswaran (2002), Garimella and Schlitz (1995), Ramadhyani and Incropera (1988).

5.6 Summary and Conclusions

The following conclusions are achieved based on the results of experimental studies of transient heat transfer from an in-line four flush-mounted and protruded heat sources with water and FC-72 cooling in a rectangular channel.

1. For water and FC-72 cooling, the upstream thermal boundary affects the heat transfer characteristics of downstream, with the value of Nu_t decreases as the chip number increases.
2. At the initial period, the thermal boundary layer thickness is developed and after a long period, the thermal boundary layer thickness is governed by velocity or temperature rather than non-dimensional time, Fo . This can be seen in Figures 5.4, 5.7, 5.10, 5.13, 5.16, 5.19, and 5.22.

3. The experimental results indicate that the value of Nusselt number is strongly affected by the value of Reynolds number.
4. Transient heat transfer regimes for pump-off, pump-on, power-on, power-off and power-on transient natural convection heat transfer for water cooling are deduced. The pump-off and power-on transient heat transfer regimes are also deduced for FC-72 cooling.
5. For flush-mounted chips, the values of Nu_t for all four chips fall along 1/3 of the theoretical slope of Pe_t for transient forced convection, and 1/4 of the theoretical slope of Ra_t for transient natural convection.
6. The correlation equations for water and FC-72 cooling are reported for four chips as well as for overall data in the transient regimes.
7. It is observed from the water and FC-72 cooling data, the power-on transient natural convection and power-on transient forced convection having similar heat transfer behaviour and FC-72 data are higher than those of water data.
8. The data obtained from this study for FC-72 cooling are having a fair agreement with the data obtained by Samant and Simon (1989), Garimella and Schlitz (1995). The present overall Nusselt number data are about 30% and 70% higher than those obtained by transient natural convection and power-on transient operations respectively.
9. The protruding height is found to have significant effect on heat transfer. The value of Nu_t for B/ℓ of 0.2 is greater than that for B/ℓ of 0 and 0.1 in the transient regime for four

chips. The Nusselt number data at different protrusions diminishes with time at the pump-off transient operation and increases with time at the pump-on transient operation.

10. The correlation Eq. (5.3) of flush-mounted heaters is modified to represent the heat transfer behaviour during the different protrusions of the chips. The overall recommended correlation is given in Eq. (5.23) and the summary of the results is given in Table 5.19.

Chapter 6 Numerical Analyses

6.1 Introduction

In order to simulate the three-dimensional laminar flow heat transfer in a rectangular channel with four flush-mounted discrete heat sources mounted on one wall, a numerical model is constructed using Gambit. After specifying the geometry, boundary conditions and the non-uniform grid spacings in Gambit, the meshes are then transferred to the software Fluent 5.6 to carry out the numerical simulation.

The three-dimensional steady-state and transient heat transfers are investigated numerically using Fluent Inc. (1995). The finite volume technique and the semi-implicit method for pressure-linked equation (SIMPLE) are used to solve the basic conservation equations (Patankar 1980). This algorithm provides a remarkably successful implicit method for simulating incompressible flow including those involved in the cooling of electronic equipments (Asako and Faghri 1991, Davalath and Bayazitoglu 1987, Kim and Anand 1994a, 1994b).

6.2 Geometry

The rectangular channel simulated is shown in Figs. 6.1(a) and 6.1(b). Four square heat sources that are flush-mounted on one wall are shown in figures.

(a) Three-dimensional view of the physical model

(b) Cross sectional view of the flow channel

Fig. 6.1. Schematic diagram of the physical model

The rectangular channel used in the numerical simulation is shown in Fig. 6.1(b), i.e., it is the same as the experimental setup.

6.3 Governing Equations

The numerical model is constructed with the assumptions of incompressible, three-dimensional laminar flow heat transfer. The flow field and temperature field in the simulation satisfy the following governing equations.

The continuity equation:

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \tag{6.1}$$

The convection-diffusions equation:

$$\frac{\partial(\rho\phi)}{\partial t} + \frac{\partial(\rho u\phi)}{\partial x} + \frac{\partial(\rho v\phi)}{\partial y} + \frac{\partial(\rho w\phi)}{\partial z} = \frac{\partial}{\partial x}\left(\Gamma \frac{\partial\phi}{\partial x}\right) + \frac{\partial}{\partial y}\left(\Gamma \frac{\partial\phi}{\partial y}\right) + \frac{\partial}{\partial z}\left(\Gamma \frac{\partial\phi}{\partial z}\right) + S_\phi. \tag{6.2}$$

In the above equation, S_ϕ is the source term; the general variable ϕ represents u , v , w and T , where Γ represents the appropriate transport coefficients respectively as shown in Table 6.1.

Table 6.1. General variables and corresponding diffusion coefficients and source terms

ϕ	Γ	S_ϕ
u	μ	$-(\partial P/\partial x)$
v	μ	$-(\partial P/\partial y) - \rho g$
w	μ	$-(\partial P/\partial z)$
T	μ/Pr	0

6.4 Steady-state Simulation

The general form of the convection diffusion Eq. (6.2) can be simplified for the steady-state simulations. The first term, $\frac{\partial(\rho\phi)}{\partial t}$ of Eq. (6.2) is removed to simplify the general equation for the steady-state simulations. However, Eq. (6.2) can be simplified as the following form:

$$\frac{\partial(\rho u\phi)}{\partial x} + \frac{\partial(\rho v\phi)}{\partial y} + \frac{\partial(\rho w\phi)}{\partial z} = \frac{\partial}{\partial x} \left(\Gamma \frac{\partial\phi}{\partial x} \right) + \frac{\partial}{\partial y} \left(\Gamma \frac{\partial\phi}{\partial y} \right) + \frac{\partial}{\partial z} \left(\Gamma \frac{\partial\phi}{\partial z} \right) + S_\phi \quad (6.3)$$

6.4.1 Boundary Conditions for Without Conjugate Modelling

All the surfaces except the embedded heaters are assumed adiabatic and heated with constant heat flux q'' . It is assumed that the surface outside the heat source is adiabatic.

The boundary conditions are referred to Fig. 6.1(b) as follows

$$\text{At } x = 0.5W, \quad u = v = w = 0. \quad (6.4a)$$

$$\text{At } x = 0 \text{ (symmetry plane)}, \quad u = 0, \partial v / \partial x = 0, w = 0. \quad (6.4b)$$

$$\text{At } y = 0, \quad u = w = 0. \quad (6.4c)$$

$$\text{At } y = 0, \quad v = v_{in}. \quad (6.4d)$$

$$\text{At } z = 0, z = H, \quad u = v = w = 0. \quad (6.4e)$$

$$\text{At } x = 0.5W \text{ (insulated)}, \quad \partial T / \partial x = 0. \quad (6.4f)$$

$$\text{At } x = 0 \text{ (symmetry plane)}, \quad \partial T / \partial x = 0. \quad (6.4g)$$

$$\text{At } y = 0, \quad T = T_{in}. \quad (6.4h)$$

$$\text{At the outlet, } y = L, \quad \partial T / \partial y = 0. \quad (6.4i)$$

$$\text{At the outlet, } y = L, \quad \partial v / \partial x = \partial v / \partial y = \partial v / \partial z = 0. \quad (6.4j)$$

$$\text{At } z = H \text{ (insulated),} \quad \partial T / \partial z = 0. \quad (6.4k)$$

$$\text{At } z = 0 \text{ (at the substrate, insulated),} \quad \partial T / \partial z = 0. \quad (6.4l)$$

$$\text{At } z = 0 \text{ (at the heat sources),} \quad q'' = q_o. \quad (6.4m)$$

Equations. (6.4d) and (6.4h) prescribe a uniform velocity and uniform temperature at the channel entrance, and Eq. (6.4i) is the streamwise energy diffusion flux to be negligible at the outlet.

6.4.2 Boundary Conditions for With Conjugate Modelling

Conjugate heat transfer couples the heat transfer in the solid and fluid regions. The heat transfers through water and side wall of thickness $S = 9$ mm are also considered. The heat convection through Teflon substrate is accounted by introducing natural convection boundary conditions with the value of $h = 2$ W/m²K (Incropera and Dewitt, 2002). The energy equation in the substrate region reduces to

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0. \quad (6.5)$$

The interface conditions are also derived in the substrate-liquid interface. Using these equations, energy transport in the entire region, including substrate and liquid regions, are solved simultaneously. The temperature dependent fluid density is employed in the simulations. The Eqs. (6.4a) to (6.4j) can be considered as the boundary conditions for conjugate analyses. The following conditions are also considered for the heat transfer in substrate region:

$$\text{At } z = -S \text{ (at the substrate)} \quad q'' = h(T_{Sub} - T_\infty). \quad (6.6a)$$

$$\text{At } z = -S \text{ (at the heat source)} \quad q'' = q_o. \quad (6.6b)$$

6.4.3 Grid Independent Solution and Convergence Criterion

For the purpose of the grid independence study, four different $x \times y \times z$ non-uniformly spaced grid sizes are considered: $15 \times 350 \times 15$ (with 10 and 20 grid points in the x and y directions respectively on the heat source), $25 \times 350 \times 15$ (with 5 and 10 grid points in the x and y directions respectively on the heat source), $25 \times 350 \times 20$ (with 5 and 10 grid points in the x and y directions respectively on the heat source), $25 \times 400 \times 15$ (with 10 and 20 grid points in the x and y directions respectively on the heat source). For the purpose of the grid independence study, the value of Re_e is chosen to be 10 and constant heat flux is 0.2 W/cm^2 . Grid independence is established by examining the temperature and velocity distributions at different points in the channel. The values are tabulated in Table 6.2(a) and Table 6.2(b) at $z = 1.25 \text{ mm}$ and $z = 2.5 \text{ mm}$ respectively. The corresponding temperature and velocity profiles are shown in Figs. 6.2(a) and 6.2(b) respectively.

Table 6.2(a). The temperature and velocity at $x = 2.5 \text{ mm}$, $y = 0 \sim 840 \text{ mm}$, $z = 1.25 \text{ mm}$.

Channel length, m	Chip wall temperature (K)				Flow velocity (m/s)			
	$15 \times 350 \times 15$	$25 \times 350 \times 15$	$25 \times 350 \times 20$	$25 \times 400 \times 15$	$15 \times 350 \times 15$	$25 \times 350 \times 15$	$25 \times 350 \times 20$	$25 \times 400 \times 15$
0.68	293	293	293	293	0.000651	0.000657	0.000658	0.000653
0.705	295.145	295.213	295.211	295.135	0.00065	0.000657	0.000658	0.000652
0.735	300.367	300.3601	300.366	300.353	0.000651	0.000657	0.000658	0.000655
0.765	302.652	302.698	302.692	302.636	0.00065	0.000657	0.000658	0.000653
0.80	301.562	301.541	301.539	301.544	0.00065	0.000657	0.000658	0.000653

Table 6.2(b). The temperature and velocity at $x = 2.5\text{ mm}$, $y = 0\sim 840\text{ mm}$, $z = 2.5\text{ mm}$.

Channel length, m	Chip wall temperature (K)				Flow velocity (m/s)			
	15×350×15	25×350×16	25×350×20	25×400×15	15×350×15	25×350×15	25×350×20	25×400×15
0.68	293	293	293	293	0.000866	0.000874	0.000876	0.000869
0.705	293.286	293.351	293.346	293.288	0.000867	0.000874	0.000876	0.000869
0.735	296.868	296.834	296.835	296.868	0.000866	0.000874	0.000876	0.000869
0.765	300.254	300.290	300.285	300.255	0.000866	0.000874	0.000876	0.000869
0.80	301.481	301.481	301.48	301.481	0.000866	0.000874	0.000876	0.000869

(i) Temperature profile

(ii) Velocity profile

(a) At $x=2.5\text{ mm}$, $y=0\sim 840\text{ mm}$ and $z=1.25\text{ mm}$

(b) At $x = 2.5$ mm, $y = 0 \sim 840$ mm and $z = 2.5$ mm.

Fig. 6.2. Temperature and velocity profile at different points

From the temperature and velocity profiles at different points we can see that there is no remarkable differences found in the temperature profiles, but a little differences are found in the velocity profiles. From the velocity profile we can select any one of the first two grid sizes. Considering the speed of the simulation calculation, the second grid is employed in the numerical simulation. The computation domain consists of 25 grid lines in the x -direction, 350 grid lines in the y -direction and 15 grid lines in the z -direction where 5 grid lines in the x -direction and 10 grid lines in the y -direction on the heat source (Tso et al. 2004, Bhowmik et al. 2005a). The image of the grid allocation inside the flow domain is shown in Fig. 6.3. Here the grid cells are going from coarser to fine toward the heat source to have uniform wall temperature on the chip.

Fig. 6.3. Grid allocation inside the flow domain

A sufficient number of iterations were performed to obtain a converged solution as shown in Fig 6.4. From the figure, it is seen that all the residuals (continuity, x -velocity, y -velocity, z -velocity and energy equations) are satisfied with the convergent criteria 10^{-6} , which was selected at the start of the numerical simulation.

Fig. 6.4. Convergent criteria of the numerical simulation ($Re_\ell = 10$)

6.4.4 Steady-state Results and Analyses

The relative static pressure distribution during the steady-state period along the channel is shown in Fig. 6.5. It is seen that the static pressure at the entrance of the channel is not linear but after some length becomes linear. This may be due to the inlet boundary condition of the fluid. Similar to the experimental condition, the fluid enters to the channel at constant velocity. It should be noticed that the relative static pressure at the inlet is set zero as the reference pressure (Patankar 1980). For laminar flow, the accepted correlation of hydrodynamic entrance length L_e is:

$$\frac{L_e}{D} = 0.06 Re_D \quad (6.7)$$

For this model, the hydrodynamic entrance length is about 360 mm for the value of $Re_D = 750$ as calculated from Eq. (6.7). It means that the laminar flow is already fully-developed at the chips. This is similar with the assumption made in the experiment and numerical simulation.

Fig. 6.5. Relative static pressure profile plot

Figure 6.6 shows a typical wall temperature distribution of the four in-line heat sources at the values of $Re_t = 1313$ and $q'' = 0.6 \text{ W/cm}^2$ for with and without conjugate effects. It is seen; with conjugate effects the chip wall temperatures are sharply peaked at the leading edge of the heat source and maintain a uniform temperature on the chip (Tso et al. 2004, Bhowmik et al. 2005a). These results are confirmed by the numerical study of Bessaih and Kadja (2000), and the experimental study of Wang and Vafai (1999).

Fig. 6.6. The wall temperature distributions at $x = 0$, $y = 0 \sim 840$ mm and $z = 0$

From Fig. 6.6 it is also observed that the chip temperature increases as the chip number increases from upstream to downstream. It is also found that with considering the conjugate effects, the temperature variation on each heater surface is quite small. The maximum temperature difference appeared between Chip 3 and Chip 4, and the value is less than 0.4 K. However, for horizontal channel flow with discrete heat sources Wang and Vafai (1999) found a maximum difference of 0.1 K at the fourth heater. Figure 6.7 shows the thermocouple positions for the four in-line heat sources for the experimental condition of the values of $Re_i = 1313$ and $q'' = 0.6$ W/cm². The figure also shows a favourable comparison of the numerical results and the experimental results. The difference here is about 10% for all chips (Tso et al. 2004, Bhowmik et al. 2005a).

Fig. 6.7. Wall temperature distributions at $x = 0$ and $z = 0$

The results for with or without conjugate effects are tabulated in Table 6.3, where the last column shows the effects of conjugate. It is seen that with considering the conjugate effect, the temperature of the heater surface reduces by approximately 5%. This may be due to the wall heat conduction effects, where the substantial amount of heat is penetrated from the heated sections to the unheated regions. On the other hand, due to the features of discrete heating, the wall heat conduction can be caused an axial dissipation of input heat energy from the heated sections to the unheated sections. These observations are consistent with the study of Tsay (1999).

Table 6.3. Comparison of present experimental results with numerical results

Chip no.	Experimental results	Numerical results				
		Without conjugate effects		With conjugate effects		Conjugate effects (°C)
		Wall temp. (°C)	% deviation	Wall temp. (°C)	% deviation	
1	25.91	26.92	+ 6.19	26.59	+ 6.09	- 0.33
2	26.07	29.26	+ 10.88	28.56	+ 8.69	- 0.70
3	26.78	30.32	+ 11.68	29.39	+ 8.91	- 0.92
4	26.54	31.25	+ 11.88	30.20	+ 8.82	- 1.05

The span-wise temperature and axial velocity variations at various channel elevations with conjugate effects are shown in Figs. 6.8(a) and 6.8(b) respectively.

(a) Temperature distributions

(b) Velocity distributions

Fig. 6.8. Temperature and velocity distributions at $x = 0$ and $z = 0 \sim 5$ mm.

From Fig. 6.8(a) it can be seen that before the first chip there is no variation of temperature. The elevated temperature near the chips observed in the temperature profiles. Similar flow phenomena are observed at Chip 2 and Chip 3 with upward flow being strengthened. After Chip 4, the fluid flows up across the entire channel. The elevated temperature appears near the chip and it decreases to minimum at the opposite wall. Figure 6.8(b) shows the velocity variations at different channel elevations, and observed similar parabolic profile at different elevations.

The effects of heat flux on the values of Nu_ℓ are shown in Fig. 6.9 for four chips. There is no remarkable variation of Nu_ℓ with q'' is observed. However, it can be obtained that the input power has little effect on the heat transfer. However, in the experimental analyses, between 0.2 W/cm^2 and 0.6 W/cm^2 , the variations of Nu_ℓ are about 30% to 36% (in Fig. 4.4). The reasons of this discrepancy are described in Section 4.2.

(a) Chip 1

(b) Chip 2

(c) Chip 3

(d) Chip 4

Fig. 6.9. Effect of heat flux on Nusselt number

Figure 6.10 shows favourable comparison of the numerical results with the experimental results, for four chips, for the experimental conditions of $Re_i = 1170$, $q'' = 0.4 W/cm^2$. It is

observed that the experimental results agree well with the numerical results for conjugate heat transfer with the differences of Nu_e within 10%.

Fig. 6.10. Comparison of numerical results with experimental results

It is also found that the Nu_e of the second, third and fourth chips are about 80%, 69% and 62% of the first chip respectively (Bhowmik et al. 2005a). This conveys the fact that as the chip number increases, the difference between the neighbouring chips become smaller and have reached a chip number independent, since there is a little difference of 5% in values between third and fourth chip.

In the mixed convection regime the experimental results are compared with the numerical results and the numerical correlations are also reported with considering the two effects. Figure 6.11(a) shows the comparison of experimental results with the numerical results

for four chips and overall data without considering the conjugate effects. The values of SD of C_2 and m of Eq. (4.6) are given in Table 6.4(a). The correlation equation based on overall data can be written as

$$\frac{Nu_\ell}{Re_D^{1.25}} = 0.0069 \left(\frac{Gr_D}{Re_D^2} \right)^{0.41}. \quad (6.8)$$

Figure 6.11(b) shows the comparison of experimental results with numerical results with conjugate effects, for four chips and for overall data. The data for the standard deviations for the values of C_2 and m of Eq. (4.6) are given in Table 6.4(b). The correlation equation based on numerical results can be given as

$$\frac{Nu_\ell}{Re_D^{1.25}} = 0.0069 \left(\frac{Gr_D}{Re_D^2} \right)^{0.5}. \quad (6.9)$$

Table 6.4(a). The results for linear regression for without conjugate effects

	C_2	SD of C_2	m	SD of m
Chip 1	0.01053	0.00017	0.4157	0.0057
Chip 2	0.00756	0.00007	0.4129	0.0032
Chip 3	0.00599	0.00005	0.4113	0.0029
Chip 4	0.00495	0.00006	0.3997	0.0042
Overall recommended	0.00694	0.00037	0.4068	0.0187

Table 6.4(b). The results for linear regression for with conjugate effects

	C_2	SD of C_2	m	SD of m
Chip 1	0.01005	0.00039	0.5053	0.0158
Chip 2	0.00745	0.00032	0.5044	0.0176
Chip 3	0.00609	0.00031	0.5033	0.0207
Chip 4	0.00504	0.00026	0.5029	0.0212
Overall	0.00691	0.00036	0.5021	0.0213

(a) Results for without conjugate effects

(b) Results for with conjugate effects

Fig. 6.11. Comparison of present experimental results with numerical results

In the general convection regime (forced, mixed, natural), the experimental results are compared favourably with the numerical results with conjugate effects, where the difference is about 10% for all chips, shown in Table 6.5. It is observed that the difference increases as the chip number increases, because of decreasing Nusselt number from the first to the fourth chip, since heat losses are greater for the fourth chip compared to the first. Figure 6.12 shows a fair comparison between experimental and numerical results. The average *SD* of the values of C_4 and m of Eq. (4.11) are also given in Table 6.6. It is seen that the overall deviation of the values of C_4 and m are about 18%. Thus, the correlation based on the numerical results can be written as (Bhowmik et al. 2005a)

$$\frac{Nu_t}{Re_D^{1.15}} = 0.0072 \left(\frac{Gr_D}{Re_D^2} \right)^{0.5} \tag{6.10}$$

Table 6.5. The difference of numerical results with experimental results

Chip no	Wall temperature (°C)		Difference (%)
	Experimental results	Numerical results	
1	25.91	27.59	6.09
2	26.07	28.56	8.69
3	26.77	29.37	8.82
4	27.54	30.23	8.91

Fig. 6.12. Comparison of experimental results with numerical results for four chips

Table 6.6. Comparison of experimental results with numerical results

	Chip 1	Chip 2	Chip 3	Chip 4	Overall
C_4	0.0101	0.0074	0.0061	0.0050	0.0072
Difference of C_4 with experimental results	+ 5%	+ 22%	+ 23%	+ 28%	+ 18.1%
m	0.5053	0.5044	0.5033	0.5029	0.5039
Difference of m with experimental results	+ 4%	- 1%	- 2%	- 2%	- 0.0%

In the general convection regime (forced, mixed, natural), the experimental correlation is given in Eq. (4.12) and numerical correlation is given in Eq. (6.10). However, the correlation equation can be written as (Bhowmik et al. 2005a):

$$\frac{Nu_f}{Re_D^{1.15}} = 0.008 \left(\frac{Gr_D}{Re_D^2} \right)^{0.5} . \quad (6.11)$$

6.5 Time Dependent Transient Simulations

6.5.1 Governing Equations

The numerical model as described for steady-state analyses is also considered for the transient analyses. Here the effects of wall heat conduction in the substrates region is considered. The flow field and temperature field in the simulation satisfy the governing equations as given in Eqs. (6.1) and (6.2).

The present analysis also considers heat transfer through the water and the side wall of thickness $S = 9$ mm, as shown in Fig. 6.1(b). The heat convection through Teflon substrate is accounted by introducing natural convection boundary conditions with the value of $h = 2$ W/m²K (Incropera and Dewitt, 2002). The energy equation for substrate region reduces to

$$\frac{\partial T}{\partial t} - \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) = 0. \quad (6.12)$$

The interface conditions are derived from an energy balance across the substrate-liquid interface. Using these equations, energy transport in the entire region, including substrate and liquid regions, are solved simultaneously.

6.5.2 Boundary and Initial Conditions

All the surfaces are assumed to be adiabatic, except the embedded heaters that are heated with constant heat flux q'' . The boundary conditions, referring to Fig. 6.1(b), are given in Eqs. 6.4(a) to 6.4(m) and Eqs. 6.6(a) to 6.6(b).

In the transient simulation, the time dependent equations are discretized in both space and time, and it involves the integration of every term in the differential equations over a time step. The spatial discretization for the time dependent equations is identical to the steady case. Therefore, in the transient simulation the steady-state boundary conditions are used as an initial condition. Only the transient behaviour of the inlet velocity is supposed to simulate the pump-off transient operation. The inlet velocity in the y -direction (v) is specified to vary linearly from inlet velocity (v_{in}) to zero in 0.1 s.

6.5.3 Grid Independent Solution and Convergence Criterion

Since the steady-state conditions are used as the initial condition for the transient simulation. The grid sizes obtained from the steady-state grid independence study are taken for the transient simulations. In the transient simulation the inlet velocity v_{in} is assumed to decrease from v_{in} to 0 linearly in $\Delta t = 0.1$ s, and remains at 0. This assumption is made based on many Δt trial runs (1s, 0.5s, 0.1s, 0.05s). It is found that $\Delta t = 0.1$ s is consistent with the experimental results. However, the computational domain eventually selected, consists of 25 grid lines in the x -direction, 350 grid lines in the y -direction and 15 grid lines in the z -direction where 5 grid lines in the x -direction and 10 grid lines in the y -direction on the heat source (Tso et al. 2004, Bhowmik et al. 2005a).

6.5.4 Pump-off Transient Results and Analyses

Figure 6.13 shows experimental and numerical results of the four chips, for the conditions of values of $q'' = 5$ W/cm² and initial $Re_\ell = 1575$. It can be seen from numerical results that during the transient period, at the pump-off, the chip wall temperatures increase rapidly in the first 20 s. After this period, the temperatures begin to increase slowly. This phenomenon indicates that the transition from forced convection to natural convection takes about 20 s because of the inertia of the forced flow. As the natural convection mode comes in, the buoyancy force plays a significant role in the heat transfer process. The differences between the experimental and numerical results are found to be within 0.5 °C (Tso et al. 2004).

Fig. 6.13. Chip wall temperature history of the numerical simulation

Figure 6.14 shows fair agreement between numerical and experimental results. The average standard deviations of the values of C and m in Eq. (5.2) from the numerical results are given in Table 6.7, where the differences of the values of C and m are within 10%. Thus, the correlation equation based on numerical results can be written as (Tso et al. 2004)

$$Nu_\ell = 0.31(Fo)^{-0.21}(Ra_\ell)^{1/4}. \quad (6.13)$$

In the pump-off transient regime, the experimental correlation is given in Eq. (5.3) and numerical correlation is given in Eq. (6.13). Thus overall recommended correlation for the present conditions can be written as the following form (Tso et al. 2004):

$$Nu_\ell = 0.3(Fo)^{-0.2}(Ra_\ell)^{1/4}. \quad (6.14)$$

Table 6.7. Comparison of C and m with experimental results

	Chip 1	Chip 2	Chip 3	Chip 4	Overall
C	0.3408	0.3236	0.3079	0.2654	0.3094
Difference of C with experimental results	-3%	-1%	+6%	-4%	+1%
m	-0.2056	-0.2067	-0.2073	-0.2094	-0.2073
Difference of m with experimental results	-1%	-7%	-8%	-13%	-9%

Table 6.8 shows the comparison of the experimental and numerical m for steady-state mixed convection and transient heat transfer. It is seen that the experimental value of m are closer to the numerical value. For steady-state heat transfer, the numerical value of m is 5% higher than that of experimental values. For transient heat transfer the numerical value of m is 9% lower than that of experimental values. It is seen that the differences are small and may be due to the losses in experiments.

Table 6.8. Comparison of m for steady-state mixed convection and transient heat transfer

	Steady-state heat transfer		Transient heat transfer	
	Experimental	Numerical	Experimental	Numerical
Chip 1	0.5201	0.5053	-0.2164	-0.2056
Chip 2	0.5036	0.5044	-0.2229	-0.2067
Chip 3	0.4691	0.5033	-0.2248	-0.2073
Chip 4	0.4082	0.5029	-0.2384	-0.2094
Overall recommended	0.4766	0.5021	-0.2262	-0.2073

Fig. 6.14. Comparison of numerical results with experimental results

6.6 Summary and Conclusions

In this chapter, three-dimensional heat transfer in a rectangular channel with four flush-mounted discrete heat sources mounted on one wall is investigated numerically using Fluent.

The steady-state results are produced with considering the conjugate effects of the model. The effects of heat flux and chip number on the values of Nu_ℓ of the four chips are investigated. The experimental results for the four chips are compared with the numerical results for water-cooling. The correlation equations are reported based on the numerical results.

In the general convection regime (forced, mixed, natural), the experimental results are evaluated by comparing them with three-dimension numerical simulations. Experimental correlations are also compared with the numerical correlations. Finally, the overall recommended correlation for the present conditions is provided in Eq. (6.11).

The numerical simulations are also conducted to study the thermal behaviour of the four chips during the pump-off transient operation. In the transient regime, the experimental results are compared with the numerical results. Finally, the correlations are reported for four chips and the overall correlation is given in Eq. (6.14).

Chapter 7 Conclusions and Recommendations

7.1 Conclusions

The steady-state and transient heat transfer studies have been performed to study the thermal behaviour of four in-line simulated electronic chips that are mounted on one wall of a vertical rectangular channel. The effects of heat flux, coolant flow rates and chip numbers are investigated, and the following conclusions can be drawn:

1. The steady-state forced convection experimental results for flush-mounted heat sources are compared with the results of Tso et al. (1999) and Shah and London (1978) of similar geometry. It is obtained a fair agreement of 5% discrepancy with the present results. This ensures the test rig is in good working conditions.
2. For both steady-state and transient conditions, upstream wall thermal boundary affects the heat transfer behaviour of downstream. The value of Nu_i decreases as the chip number increases and reaches chip number independency by the third chip.
3. The steady-state and transient experimental results indicate that the value of Nu_i is strongly affected by Reynolds number and the number of chips. The forced, mixed and natural convection regimes are derived from the experimental data.

4. For flush-mounted chips and water cooling analyses, it is found that the slope of Pe_t is 1/3 for laminar forced convection, the slope of $[Gr_D/Re_D^2]$ and $[Gr_t/Re_t^2]$ are 1/2 for laminar mixed convection, and the slope of Ra_H is 1/3 for natural convection.
5. The slope of the steady-state mixed convection correlation is similar to that of Türkoglu and Yücel (1995) and the present heat transfer data are about 35% lower. The results seem to be applicable in the general convection regimes: natural, mixed and forced.
6. The empirical correlations for overall data in the mixed convection regime, based on channel hydraulic diameter and heat source length are given in Eqs. (4.8) and (4.9) respectively. In the general convection regime, based on channel hydraulic diameter is given in Eq. (4.12).
7. The steady-state natural convection results are qualitatively similar to the correlation of MacGregor and Emery (1969) for turbulent natural convection heat transfer and the present heat transfer data are about 20% lower. The empirical correlation for natural convection is given in Eq. (4.20).
8. The protruding height is found to have a more significant effect on the Nu_t for steady and transient heat transfer. The value of $Nu_t B/\ell$ of 0.2 is greater than that for B/ℓ of 0 and 0.1.

9. Due to the protrusion heights the flow is obstructed causing the flow separation and vortex shedding that leads to increase the average value of Nu_ℓ . In addition to this, due to protrusions, the flow is obstructed and the thermal boundary layer becomes thin that also leads to increase the heat transfer.

10. The pump-off, pump-on, power-on, power-off and power-on transient natural convection heat transfer regime for water cooling are obtained from the experimental data. The correlation equations are reported for four chips as well as for overall data in the transient regimes. The flush-mounted correlations for pump-off transient heat transfer are modified by introducing the chip's protrusion height (B) in the dimensionless (B/ℓ) form as given in Eq. (5.23).

11. For transient forced convection heat transfer, the values of Nu_ℓ fall along 1/3 of the theoretical slope of Pe_ℓ , and 1/4 of the theoretical slope of Ra_ℓ for transient natural convection heat transfer.

12. For FC-72 cooling, flow regimes for pump-off and power-on transient natural convection heat transfers are derived from the experimental data. The correlation equations are reported for four chips as well as for overall data in the transient regimes. It is found that the Nu_ℓ data of FC-72 are similar to the data of water; and the Nu_ℓ data of FC-72 are higher than that of water data.

13. The transient heat transfer data for FC-72 cooling are consistent with the data of Samant and Simon (1989) and Garimella and Schlitz (1995). The present Nu_ℓ data are

about 30% and 70% higher than those are obtained by power-on transient natural convection and power-on transient forced convection respectively.

14. The steady-state and transient experimental results are compared with the three-dimensional numerical simulations using Fluent. The numerical correlations are also reported for steady-state mixed convection and pump-off transient heat transfer. In the mixed convection regime, the numerical results are agreed well with the experimental results and the differences of the value of Nu_ℓ are about 10%.

15. The steady-state experimental results in the general convection regime are also agreed well with the numerical results for conjugate heat transfer. The numerical correlation is given in Eq. (6.10).

16. The numerical pump-off transient heat transfer results indicate that the transition from forced convection to natural convection takes about 20 s because of the inertia of the forced flow. As the natural convection mode comes in, the buoyancy force plays a significant role in the total heat transfer processes.

17. In the pump-off transient regime, the differences between the experimental and numerical results are found to be less than 0.5 °C. The overall correlation based on numerical results is given in Eq. (6.14) and the summary of the results is tabulated in Table 6.7.

7.2 Recommendations for Further Research

The experimental and numerical studies can be considered a major step in the thermal management. Direct liquid cooling can cool the high heat flux level electronic components. The following recommendations are made for further studies:

1. The heat transfer data that have obtained for water and FC-72 cooling are not enough to obtain a general correlation with considering the geometric parameters and thermophysical properties.
2. To understand the heat transfer mechanism for the protruding chips, flow visualisations should be done to study the effect of protrusion on the flow structure and thermal wake. The surface temperature distributions in the various faces (i.e. top face, front face, rear face and side faces) should be measured in future.
3. The range of test conditions needs to be extended to have more data in the mixed convection regime.
4. For protruding heat sources the three-dimensional numerical analyses should be done to study the flow and temperature field on the heat sources.
5. The present laminar flow heat transfer data from flush-mounted chips are compared with the analytical solutions of Shah and London (1978). It should be noted that one infinite plate is adiabatic and another infinite plate is heated by constant wall heat flux or temperature, so their boundary conditions are different from discrete heat sources. It is suggested that an approximate method for predicting laminar heat transfer

between parallel plates having discrete heat sources to be developed for simulating the liquid cooling of electronic components. These methods have also been introduced by Figliola and Thomas (1995) to predict air-cooling of electronic components.

6. The transient thermal behaviour of the system during power-on, pump-on, power-off and power-on transient natural convection heat transfer should be analysed numerically.

7. The steady-state natural and mixed convection studies should be done with FC-72, to study the effects of coolant on the heat transfer behaviour of the chips. In this connection, the effects of protrusions on FC-72 cooling also should be studied.

List of Publications

International Journal Papers

1. **H. Bhowmik** and K. W. Tou, 2005, "Experimental study of transient natural convection heat transfer from simulated electronics chips," *Experimental Thermal and Fluid Science*, Vol. 29(4), pp. 485-492.
2. **H. Bhowmik** and K. W. Tou, 2005, "An experimental study of transient heat transfer from discrete heat sources in water cooled vertical rectangular channel," *ASME Trans, Journal of Electronic Packaging*, Vol. 127(3), (in press).
3. **H. Bhowmik** C. P. Tso and K. W. Tou, 2005, "Analyses of convection heat transfer from discrete heat sources in a vertical rectangular channel," *ASME Trans, Journal of Electronic Packaging*, Vol. 127(3), (in press).
4. **H. Bhowmik** and K. W. Tou, 2005, "Effect of protrusion on thermal transient behaviour of chips in a liquid channel during loss of pumping power," *International Communications in Heat and Mass Transfer*, Vol. 32(1-2), pp. 166-174.
5. **H. Bhowmik**, C. P. Tso, K. W. Tou and F. L. Tan, 2005, "Convection heat transfer from discrete heat sources in a liquid cooled rectangular channel," *Applied Thermal Engineering*, (Corrected proof available).
6. C. P. Tso, K. W. Tou and **H. Bhowmik**, 2004, "Experimental and numerical thermal transient behaviour of chips in a liquid channel during loss of pumping power," *ASME Trans, Journal of Electronic Packaging*, Vol. 126(4), pp. 546-553.
7. **H. Bhowmik** and K. W. Tou, 2005, "Experimental study of transient forced convection heat transfer from simulated electronic chips," *Heat and Mass Transfer*, Vol. 41(7), pp. 599-605.

8. **H. Bhowmik** and K. W. Tou, 2005, "Study of transient forced convection heat transfer from discrete heat sources in a FC -72 cooled vertical channel," *International Journal of Thermal Sciences*, Vol. 44(5), pp. 499-505.
9. **H. Bhowmik**, C. P. Tso and K. W. Tou, 2005, "Analyses of natural convection heat transfer in a liquid cooled vertical channel," *International Journal of Heat and Technology*, (in press).

International Conference Papers

10. **H. Bhowmik**, C. P. Tso and K. W. Tou, 2004, "Study of thermal behaviour of electronics chips during power-on transient period," *Proc. 2nd BSME-ASME International Conference on Thermal Engineering*, pp.1059-1064.
11. **H. Bhowmik**, C. P. Tso and K. W. Tou, 2003, "Thermal behaviour of simulated chips during power-off transient period," *Proc. 5th Electronic Packaging Technology Conference*, pp. 497-500.
12. **H. Bhowmik**, K. W. Tou and C. P. Tso, 2003, "Unsteady thermal behaviour of electronic chips inside FC-72 cooled channel," *Proc. 6th Electronic Packaging Technology Conference*, pp. 581-585.
13. **H. Bhowmik**, K. W. Tou and C. P. Tso, 2004, "Water cooling study of natural convection heat transfer from simulated electronic chips during power-on transient period," *Proc. IEEE Semiconductor Thermal Measurement and Management Symposium*, pp. 130-134.
14. **H. Bhowmik**, C. P. Tso, K. W. Tou and F. L. Tan, 2004, "General convection heat transfer from discrete heat sources in a rectangular channel," *Proc. 4th European Thermal Science Conference*, FCH 3, pp. 1-10.

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Appendix A Overall Uncertainties

An estimation of the overall uncertainties in the experimental parameters is made using standard technique of the propagation of error for single-sample measurements suggested by Kline and McClintock (1953) and Moffat (1988).

The uncertainties interval in each results as:

$$d_R = \sqrt{\left(\frac{\partial R}{\partial v_1} d_1\right)^2 + \left(\frac{\partial R}{\partial v_2} d_2\right)^2 + \left(\frac{\partial R}{\partial v_3} d_3\right)^2 + \dots \left(\frac{\partial R}{\partial v_n} d_n\right)^2}, \quad (\text{A.1})$$

where R be a function of n independent variables, $v_1, v_2, v_3, \dots, v_n$,

$$R = R(v_1, v_2, v_3, \dots, v_n). \quad (\text{A.2})$$

For example:

$$\text{Inlet velocity, } U = \frac{Q}{WH} \quad (\text{A.3})$$

The maximum experimental uncertainties for U (m/s) is given,

$$d_U = \sqrt{\left(\frac{\partial U}{\partial Q} dQ\right)^2 + \left(\frac{\partial U}{\partial W} dW\right)^2 + \left(\frac{\partial U}{\partial H} dH\right)^2}, \quad (\text{A.4})$$

$$\text{Overall uncertainties for } U \text{ is } (\pm\%) = \frac{d_U}{U} \times 100 \quad (\text{A.5})$$

Appendix B Thermophysical Properties of FC- 72

The following equations are used to calculate the thermophysical properties of FC- 72. Property data are obtained from the manufacturer (3M Company, 1986). Each equation is given as a function of temperature in Kelvin. The functional relations between thermal conductivity k , density ρ , specific heat c_p , and dynamic viscosity μ with temperature are

Thermal conductivity (W/m.K):

$$k = 0.001 \times (90 - 0.11T_{wall}) . \quad (B.1)$$

Density of liquid (kg/m³):

$$\rho = \frac{340}{0.1619 + 3.406 \times 10^{-5} T_{wall} + 3.406 \times 10^{-7} T_{wall}^2} . \quad (B.2)$$

Specific heat of fluid (J/kg.K):

$$c_p = 561.3 + 1.704T_{wall} - 2.381 \times 10^{-4} T_{wall}^2 . \quad (B.3)$$

Dynamic viscosity of liquid (N.s/m²):

$$\mu = 2.702 \times 10^{-2} - 2.1734 \times 10^{-4} T_{wall} + 5.9832 \times 10^{-7} T_{wall}^2 - 5.5694 \times 10^{-10} T_{wall}^3 . \quad (B.4)$$

Appendix C Calculation of R-squared Value

R-squared value is a measure of goodness of the linear fit. This value is a fraction between 0.0 and 1.0 and has no units. It represents a quantitative measure of the linear relation between the x and y values. When R-squared value equals to 1.0 all points lie exactly on a straight line and 0 means x does not help to predict y . This value can be calculated as

$$\text{R-squared value} = \left[1 - \frac{S_{yx}^2}{S_y^2} \right]^{1/2} \quad (\text{C.1})$$

where
$$S_{yx}^2 = \frac{\sum (y_i - y_{ci})^2}{N - 2}$$

$$S_y^2 = \frac{1}{(N - 1) \sum (y_i - \bar{y})^2}$$

- S_{yx} is the standard error of the fit.
- S_y is the sample variance.
- $\bar{y}$ is the mean value of y_i .

y_{ci} is the value of polynomial evaluated at the data point (x_i, y_i) .