

EFFECT OF NOZZLE ANGLE ON JET IMPINGEMENT
COOLING SYSTEM

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SUPERVISOR'S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering.

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I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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Dedicate to my beloved dad, lovely mom and my honour siblings

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ABSTRACT

Cooling system using jet impingement is already widely used in industries nowadays. There were various approaches that have been investigated in order to produce more efficient jet impingement cooling system. This thesis is study about the effect of the nozzle angle on jet impingement in order to identify the relationship in heat transfer. Besides, investigation on spacing distance between nozzle's edge to the impingement surface and Reynolds number at certain angle also identified in this study. Those studies are needed parallel to the current researchers endeavor for future development of cooling system in global industries. The experiment were perform by vary 3 major parameters such as angle of the nozzle (30° , 45° , 60° , and 90°), distance between nozzle's edge to the impinge surface ($H/d= 2, 4, 6, \text{ and } 8$) and also Reynolds number ($Re= 2300, 1960, 930 \text{ and } 500$). The heat source are heated at 100°C and cooled down by the flow of air from the nozzle. The heat source temperature after cooling are measured and collected. The result discovers about the relationship nozzle angle for jet impingement cooling system which is heat transfer are more efficient when the angle of nozzle approaching to the normal line as the Nusselt Number are more higher at 90° in range $31.5 \text{ w/m}^2\text{K}$ of heat transfer coefficient compare to the lower angles of the nozzle. Furthermore, higher Reynolds number and close range of distance between nozzle's edges to impingement surface will also gives high Nusselt number which means both also effective cooling effects for the systems.

ABSTRAK

Sistem penyejukan dengan menggunakan system penyejukan hentaman jet merupakan bidang yang luas dalam perindustrian pada masa kini. Pelbagai pendekatan dan eksplorasi untuk menghasilkan sistem penyejukan jet hentaman ini. Tesis mengkaji kesan sudut bagi muncung jet hentaman dalam proses pemindahan haba pada sesebuah sistem. Selain dari itu, penelitian terhadap perubahan pada halaju udara yang digunakan sebagai medium penyejukan dan juga jarak antara hujung muncung jet hentaman ke atas permukaan hentaman pada sudut tertentu terhadap sistem penyejukan juga dikenal pasti dalam kajian ini. Kajian tentang hubungan kait tersebut pada system penyejukan ini sebenarnya amat diperlukan selaras dengan usaha para pengkaji seluruh dunia giat berusaha dalam memajukan system penyejukan untuk industri global masa hadapan. Eksperimen yang dijalankan akan mengubah 3 parameter utama iaitu sudut muncung jet hentaman (30° , 45° , 60° , and 90°), nombor Reynold ($Re = 2300, 1960, 930$ and 500), dan juga jarak antara muncung jet ke atas permukaan hentaman ($H/d = 2, 4, 6, \text{ and } 8$). Spesimen dipanaskan pada suhu 100°C dan disejukan oleh pengaliran udara yg dibawa oleh muncung jet. Suhu spesimen selepas penyejukan tersebut akan diambil dan dikumpulkan. Apa yang diperolehi dalam eksperimen tersebut adalah hubungkait pemindahan haba apabila sudut bagi muncung jet hentaman berubah di mana kadar pemindahan haba dijangkakan tinggi apabila sudut muncung jet hentaman mendekati garisan normal dan ini jelas ditunjukkan pada nilai nombor Nusselt yang tinggi pada sudut 90° iaitu pada julat $31.5 \text{ w/m}^2\text{K}$ pekali kadar pemindahan haba berbanding dengan sudut yang jauh lebih rendah. Selain dari itu, nombor Reynold yang tinggi dan juga jarak antara hujung muncung jet hentaman ke atas permukaan hentaman yang rendah akan menghasilkan nombor Nusselt yang tinggi dan ini membuktikan kedua-dua faktor itu juga faktor yang berkesan dalam sistem pemindahan haba.

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LIST OF SYMBOLS

Re	Reynolds Number
\dot{Q}	Rate of net heat transfer
H/d	Dimensionless jet to heat source spacing to the
r/d	Dimensionless radius of the heat source area
h	Heat transfer coefficient
Nu	Nusselt Number
$^{\circ}C$	Degree Celcius
$^{\circ}$	Degree
D	Diameter of the wall
d	Diameter of the nozzle
r_c	Radius of the nozzle
L_c	Characteristic Length
T_s	Temperature surface
T_{∞}	Ambient temperature
ΔT	Temperature Different
k	Thermal conductivity
c_p	Specific heat
μ	Dynamic viscosity
V	Fluid velocity
ρ	Density
ν	Kinematic viscosity
m	Molar mass

\dot{m}	Mass flow rate
A_c	Area of the nozzle
A_s	Area of the wall

LIST OF ABBREVIATIONS

FYP	Final year project
vs	Versus
3-D	Three Dimension

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND OF STUDY

Air jets have been widely used in many industrial applications in order to achieve enhanced coefficients for convective heating, cooling or drying. A single air jet or arrays of air jet, impinging normally on a surface are an effective method to enhance heat and mass transfer. High convective heat transfer coefficient is a very important factor that leads to the many usage of impingement jets in industrial for heating and cooling purposes. Jet impingement is an attractive cooling mechanism due to the capability of achieving large heat transfer rates. This cooling method has been used in industrial applications such as annealing of metals, tempering of glass, cooling of gas turbine blades, cooling in grinding processes and cooling of photovoltaic cells. Jet impingement has also used for high-powered electronic and photonic thermal management solutions and numerous jet impingement studies have been aimed directly at electronics cooling.

Due to the many industrial applications of impinging jets research has been conducted to understand their flow and heat transfer characteristics. The main variables for jet impingement heat transfer are the angle of impingement, the jet Reynolds number and the height of the nozzle above impingement surface. Convective heat transfer to an impinging air jet is known to yield high local and area averaged heat transfer coefficients. The research is concerned with the measurement of heat transfer to impinging air jets over a wide range of test parameters. These include Reynolds numbers, Re , nozzle to impingement surface distance, H/d , and angle of impingement, from 30° to 90° .

1.2 PROBLEM STATEMENT

There are problems statements that have been defined in conjunction of this project. For convenience, the problems are determined and discover. Firstly, some application will use an effective method for their product such as for electronic cooling system. The study of effect of nozzle angle for jet impingement cooling system needed. The current method of heat removal, involving extended surfaces and fan arrays, frequently used in electronics is clearly insufficient, especially as the electronic components become more and more powerful, dissipating more heat, whereas the space around these components continues to be reduced due to miniaturization trends. Furthermore, there are some relevant results we need to compare in order to state the best effects for cooling system for jet impingement cooling system.

1.3 OBJECTIVES OF STUDY

In order to assure this project run smoothly, there are objectives that have been defined and which have to be fulfilled. The first objective of this project is to study about the effect of nozzle angle on the jet impingement cooling system. Besides that, the objective also to determine the expected cooling effect of the nozzle at a certain angle by vary the other factor such as Reynolds Numbers and spacing between nozzle's edge to impingement surface.

1.4 SCOPES OF STUDY

The scope of this project will comprise the boundaries of project study. The jet impingement cooling systems are wide range of study. Many characteristic should be bound in order to make this project achieve the objectives. First of all, the study of this project are using laminar flow which is Reynolds number are determine to be at 500, 930, 1960, and 2300. The nozzle angle studied also already set at 30°, 45°, 60° and 90°. Any angle values other than that rounded to the closest angle.

Further more, this project only study using one material of nozzle and diameter of nozzle that have been studied are constant at 10 mm. In this project, it is only using an air as working fluids for cooling system. The boundaries in this project are exist in order to state that the priority of project analysis of the effect of nozzle angle on the jet impingement cooling system to be more clear and understand. In addition, temperatures of the heated plate are constant at 100°C. It assumes that the system is in steady state condition.

CHAPTER 2

LITERATURE REVIEW

2.1 JET IMPINGEMENT

An impinging jet can be characterized in many different ways, such as being submerged or free-surface, or being confined or unconfined. A fluid jet issuing into a region containing the same fluid is characterized as a submerged jet while a fluid jet issuing into a different, less dense, fluid is characterized as a free-surface jet.

Confined impinging jet the radial spread is confined in a narrow channel, usually between the impingement surface and the orifice plate. This has also been described in the literature as semi-confinement due to the diminishing effect of confinement with increasing jet-to- target spacing. The presence of a confining top wall in jet impingement causes lower heat transfer coefficients, thought to be caused by the recirculation of fluid heated by the target plat. Confinement promotes a more uniform heat transfer distribution for the area enclosed by a non-dimensional radial distance from the stagnation point. The key parameters determining the heat transfer characteristics of a single impinging jet are the Reynolds number, jet diameter and jet-to-target spacing.

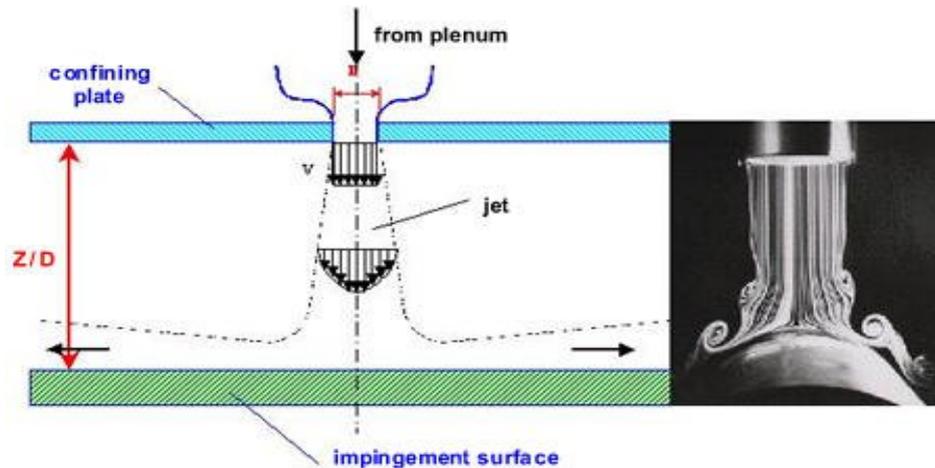


Figure 2.1: Example one of the jet impingement

Source: Wikipedia (2009)

2.2 NOZZLE

A nozzle is a mechanical device or orifice designed to control the characteristics of a fluid flow as it exits (or enters) an enclosed chamber or pipe. A nozzle is often a pipe or tube of varying cross sectional area and it can be used to direct or modify the flow of a fluid (liquid or gas). Nozzles are frequently used to control the rate of flow, speed, direction, mass, shape, and/or the pressure of the stream that emerges from them. Types of nozzles:

Table 2.1: List of types of nozzle

Types of Nozzle	Characteristics
Jets Nozzle	Gas jet, fluid jet, or hydro jet is a nozzle intended to eject gas or fluid in a coherent stream into a surrounding medium.
High velocity Nozzles	Frequently the goal is to increase the kinetic energy of the flowing medium at the expense of its pressure energy and/or internal energy.
Propelling Nozzles	Hot air is passed through a high speed nozzle, a propelling nozzle greatly increasing its kinetic energy

Magnetic nozzles	Magnetic nozzles have also types of propulsion which the flow of plasma is directed by magnetic fields
Spray nozzles	Nozzles produce a very fine spray of liquids.
Vacuum nozzles	Vacuum cleaner nozzles come in several different shapes.
Shaping nozzles	Nozzles are shaped to produce a stream that is of a particular shape.

2.3 CONVECTION OF HEAT TRANSFER

Mechanism that involve in heat transfer is convection. Convection is classified as natural (or free) and forced convection, depending on how the fluid motion is initiated. In forced convection, the fluid is forced to flow over a surface or in a pipe by external means such as a pump or a fan. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect, which manifests itself as the rise of warmer fluid and fall of the cooler fluid. Convection involves fluid motion as well as conduction. The fluids motion enhances heat transfer, since it brings warmer and cooler chunks of fluids into contact, initiating higher rates of conduction at greater number of sites in a fluid. Convection heat transfer strongly depends on the fluid properties dynamic viscosity, μ , thermal conductivity, k , density and specific heat c_p as well as the fluids velocity V . It also depends on the geometry and the roughness of the solid surface, in addition to the type of fluid flow (such as being streamlined or turbulent). Thus, convection heat transfer relations to be rather complex because of the dependence of convection on so many variable

The rate of convection heat transfer is observed to be proportional to the temperature difference and is conveniently expressed by Newton's Law of cooling:

$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) \quad (2.1)$$

Important factor that effecting heat transfer is due to h , convection heat transfer coefficient, $\text{W/m}^2 \cdot ^\circ\text{C}$ Heat transfer will increase by increasing h . Value h also define

as the rate of heat transfer between a solid surface and a fluid per unit surface area per unit temperature difference. Newton's Law of Cooling states that the rate of change of the temperature of an object is proportional to the difference between its own temperature and the ambient temperature (i.e. the temperature of its surroundings). Newton's Law makes a statement about an instantaneous rate of change of the temperature.

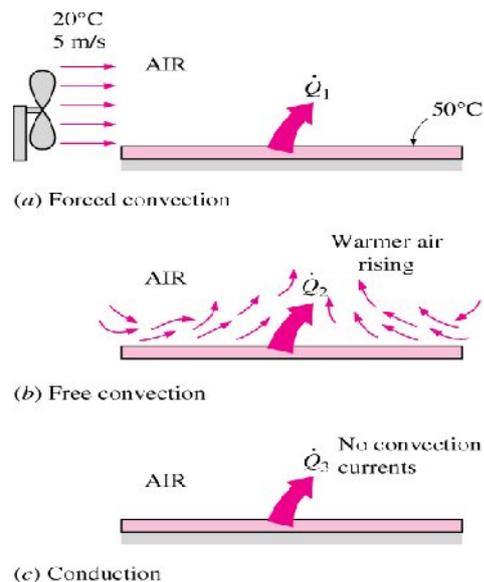


Figure 2.2: Comparison on Forced convection, natural convection and conduction

Source: Lienhard J.H.(2001)

2.4 LAMINAR FLOW

Some flows are smooth and orderly while other are rather chaotic. The highly ordered fluid motion characterized by smooth layers of fluid is called laminar. The word laminar comes from the movement of adjacent fluids particles together in laminates. The flow of high viscosity fluids such as oils at low velocities is typical laminar. The highly disordered fluid motion that typically occurs at high velocities and is characterized by velocity fluctuations is called turbulent. The flow of low viscosity fluids such as air at high velocities is typically turbulent. The flow regime greatly influences the required power for pumping .A flow that alternates between being laminar and turbulent is called transitional.

The flow regime for laminar characterized by smooth streamlines and highly-ordered motion. Turbulent flow characterized by velocity fluctuations and highly-disordered motion. Laminar flow is encountered when highly viscous fluids such as oil flow in a small pipes or narrow passages.

2.5 REYNOLDS NUMBER

The existence of these laminar, transitional and turbulence flow regimes by injecting some dye streak into the flow of glass tube as British scientist Osborn Reynolds did over a century ago. The forms of a straight and smooth line at low velocities are known as laminar flow. This laminar flow depends on the surface geometry, surface roughness, flow velocity, surface temperature and type of fluids among the other things. Osborn Reynolds (1842-1912) discovered that the flow regime depend mainly on the ratio of the inertia forces to viscous forces in the fluids. The ratio is called the Reynolds number, which is a dimensionless quantity and expressed as

$$\text{Re} = \frac{\text{Inertia forces}}{\text{Viscous forces}} = \frac{VL_c}{\nu} = \frac{\rho VL_c}{\mu} \quad (2.2)$$

Where

- ρ = Density
- V_∞ = Free stream velocity
- L_c = Characteristic Length sale
- μ = dynamic viscosity
- ν = kinematic viscosity = μ/ρ

For small Re numbers, viscous forces are dominant. Fluctuations in flow are damped out therefore it is a laminar flow. For large Re numbers, inertial forces are important, fluctuations in flow become amplified which means that the flow is turbulent. For every geometry there is a critical Re at which transition to turbulence occur. For example for:

- a) Flat plate, $Re_{critical} = 5 \times 10^5$
- b) Pipes and Spheres External flow, $Re_{critical} = \sim 2 \times 10^5$
- c) Pipes Internal flow, $Re_{critical} = 2 \times 10^3$
- d) Different geometries have different critical values

2.6 NUSSELT NUMBER

In convection studies, it is common practice to nondimensionalize the equation and combines the variables, which group together into dimensionless numbers in order to reduce the number of variables. Nusselt number defined as

$$Nu = hL_c / k \quad (2.3)$$

Where k is the thermal conductivity of the fluid and L_c is the characteristic length. The Nusselt number is named after Wilhelm Nusselt, who made significant contribution to convective heat transfer in the first half of twenty century, and it is viewed as the dimensionless convective heat transfer coefficient.

To understand the physical significance of the Nusselt number, consider a fluid layer of thickness L and temperature difference $\Delta T = T_2 - T_1$. Heat transfer through the fluid layer is by convection when the fluid involves some motion and by conduction when the fluid layer is motionless. The Nusselt number represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer. The larger the Nusselt number, the more effective the convection. A Nusselt number of $Nu=1$ for a fluid layer represent heat transfer across the layer by pure conduction.

2.7 STEADY STATE

General formula for steady state system:

$$\dot{Q} = \dot{m} c_p \Delta T \quad \text{or} \quad \dot{Q} = \rho V A_c \quad (2.4)$$

Which,

- \dot{Q} = Rate of net heat transfer, (kJ/s)
 \dot{m} = Mass flow rate, (kg/s)
 C_p = Constant pressure specific heat, (kJ/kg.K)
 ρ = Fluid density, (kg/m³)
 ΔT = Temperature different, (K)
 A_c = Cross section area, (m²)

With the mass conservation equation, the physical idea is that any rate of change of energy in the control volume must be caused by the rates of energy flow into or out of the volume. The heat transfer and the work are already included and the only other contribution must be associated with the mass flow in and out, which carries energy with it. Note that the steady flow energy does not contain a term for friction, nor does it require any such term. It is a total energy balance. Any frictional effects will reduce the mechanical energy terms and increase the internal energy but will have no influence on the overall energy balance. It follows that equation that is applicable both to ideal (frictionless) processes and also to real processes that involve viscous resistance and turbulence. In airflow systems, mechanical energy is expended against frictional resistance only but many not normally use the airflow to produce a work output through a turbine or any other device.

2.8 HUMIDITY

Humidity can be measured in several ways, but relative humidity is the most common. Absolute humidity is the mass of water vapor divided by the mass of dry air in a volume of air at a given temperature. The hotter the air is, the more water it can contain. Relative humidity is the ratio of the current absolute humidity to the highest possible absolute humidity (which depends on the current air temperature). A reading of 100 percent relative humidity means that the air is totally saturated with water vapor and cannot hold any more, creating the possibility of rain. This does not mean that the relative humidity must be 100 percent in order for it to rain it must be 100 percent where the clouds are forming, but the relative humidity near the ground could be much

less. Humans are very sensitive to humidity, as the skin relies on the air to get rid of moisture.

Humid air is less dense than dry air because a molecule of water ($m = 18$) is less dense than a molecule of nitrogen ($m = 28$) and a molecule of oxygen ($m = 32$). About 78% of the molecules in dry air are nitrogen (N_2). Another 21% of the molecules in dry air are oxygen (O_2). The final 1% of dry air is a mixture of other gases. For any gas, at a given temperature and pressure, the number of molecules present is constant for a particular volume. So when water molecules (vapor) are introduced to the dry air, the number of air molecules must reduce by the same number in a given volume, without the pressure or temperature increasing. Hence the mass per unit volume of the gas (its density) decreases. The humidity in Malaysia is very high. The mean monthly relative humidity falls within 70 to 90%, varying from place to place and from month to month. The monthly relative humidity varies from a minimum of about 3% to a maximum of about 15% from any specific area in Malaysia. The mean relative humidity varies from a low 84% in February to a high of only 88% in November in Peninsular Malaysia. The maximum variation can be found in the northwest area of the Peninsula.

LOCATION	MONTH	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEPT	OCT	NOV	DEC
Alor Star	A	26.30	27.30	27.70	27.90	27.60	27.20	26.90	26.80	26.50	26.30	26.10	26.20
	B	32.70	34.30	34.50	33.90	32.60	31.80	31.60	31.60	31.10	31.30	31.30	31.40
	C	74.40	71.60	75.10	80.30	84.20	85.10	85.10	85.00	85.40	86.90	85.80	80.50
	D	95.30	94.30	95.00	96.90	97.50	97.60	97.90	97.00	99.10	98.00	99.00	96.20
Ipoh	A	26.40	26.80	27.20	27.20	27.20	27.10	26.80	26.70	26.40	26.20	26.00	26.00
	B	32.70	33.70	33.90	33.50	33.20	32.90	32.70	32.60	32.30	65.00	31.70	31.80
	C	77.20	79.80	78.20	82.30	82.40	80.10	80.80	79.60	79.80	81.20	85.50	83.20
	D	96.60	96.20	96.40	97.20	97.00	96.90	96.70	97.00	97.20	97.80	98.20	97.70
Sitiawan	A	26.00	26.50	26.90	27.10	27.10	26.90	26.60	26.50	26.30	26.30	26.00	25.90
	B	31.50	32.20	32.60	32.50	32.50	32.30	31.90	31.90	31.60	31.50	31.10	31.20
	C	84.90	83.60	84.20	85.50	85.50	84.80	84.30	84.00	83.00	87.10	88.10	87.40
	D	99.20	98.90	99.10	99.10	99.00	98.80	98.80	98.90	99.30	99.40	99.40	99.30
Kuala Lumpur	A	25.90	26.30	26.60	26.80	27.00	26.70	26.40	26.40	26.20	26.20	25.90	25.80
	B	31.90	32.80	33.00	32.80	32.70	32.30	31.90	32.00	31.80	31.70	31.20	31.20
	C	82.60	81.20	82.40	84.90	84.30	83.90	83.40	83.10	84.60	85.50	86.90	85.70
	D	98.60	98.20	98.40	98.40	98.20	98.10	97.80	97.90	98.30	98.70	98.90	98.90
Malacca	A	26.10	26.70	26.80	26.90	27.00	26.70	26.30	26.10	26.30	26.40	26.00	25.90
	B	31.50	32.70	32.80	32.20	31.90	31.20	30.90	30.70	30.80	31.20	30.80	30.90
	C	79.50	78.20	81.60	85.20	85.80	86.50	86.30	87.00	86.50	86.20	87.50	84.70
	D	95.60	95.40	97.50	99.10	99.40	99.60	99.60	99.70	99.60	99.60	99.50	98.00
Johore Baru	A	25.10	25.30	25.70	25.90	26.10	25.70	25.40	25.50	25.50	25.50	25.20	25.10
	B	31.10	31.30	32.30	32.30	32.10	31.50	30.90	31.00	31.20	31.40	30.80	30.50
	C	84.30	84.10	85.20	88.00	87.50	87.90	87.90	87.90	87.60	87.70	87.80	89.50
	D	99.00	99.00	99.10	99.20	99.20	99.30	99.50	99.30	99.30	99.20	99.30	99.20

COUNTRY: MALAYSIA

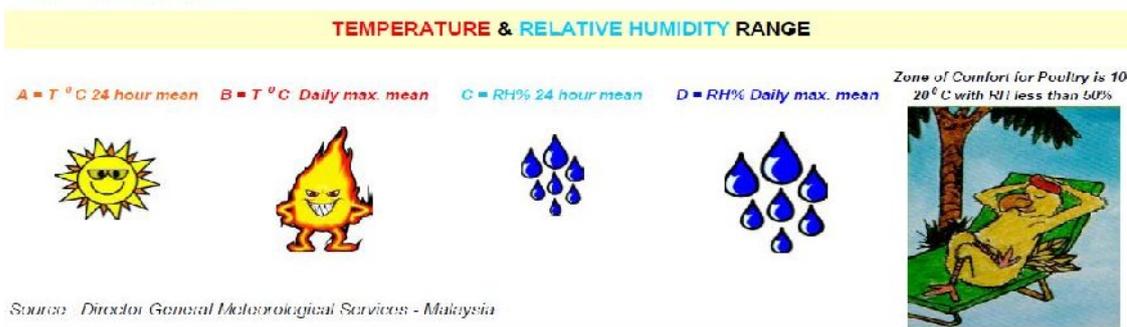


Figure 2.3: Temperature and Relative Humidity Range in Malaysia

Source: Director General Meteorological Malaysia (2006)

2.9 PREVIOUS RESEARCH

Some research have been done by Donovan and Murray (2005) which was using turbulent flow that consist of Reynolds number between 10000 to 30000. They have been conclude that, the magnitude of the fluctuations has an influence on the mean heat transfer from the impingement surface. Besides, velocity fluctuations normal to the impingement surface are more significant influence on the heat transfer. For oblique angles of the vertical characteristic of the flow and consequently heat transfer different in the uphill and downhill direction. Which mean, at the downhill direction, the vortices have time to develop and pass as large scale low frequency vortices. Meanwhile, for the uphill direction, the vortices have not completed a merging process and pass intermittently as large as scale low frequency vortices and as small scale high frequency vortices.

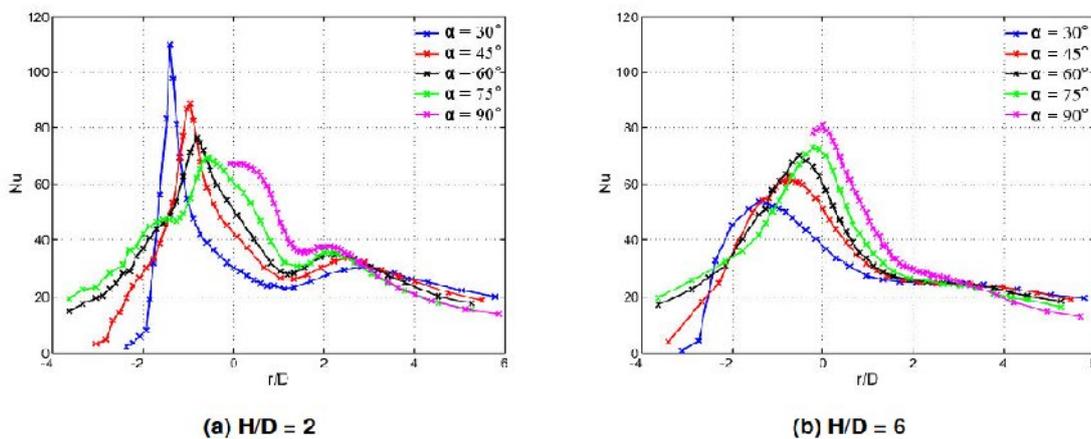


Figure 2.4: Graphs of obliquely Impinging Jet Mean Nusselt Number Distributions;
Re = 10000

Source: Donovan (2005)

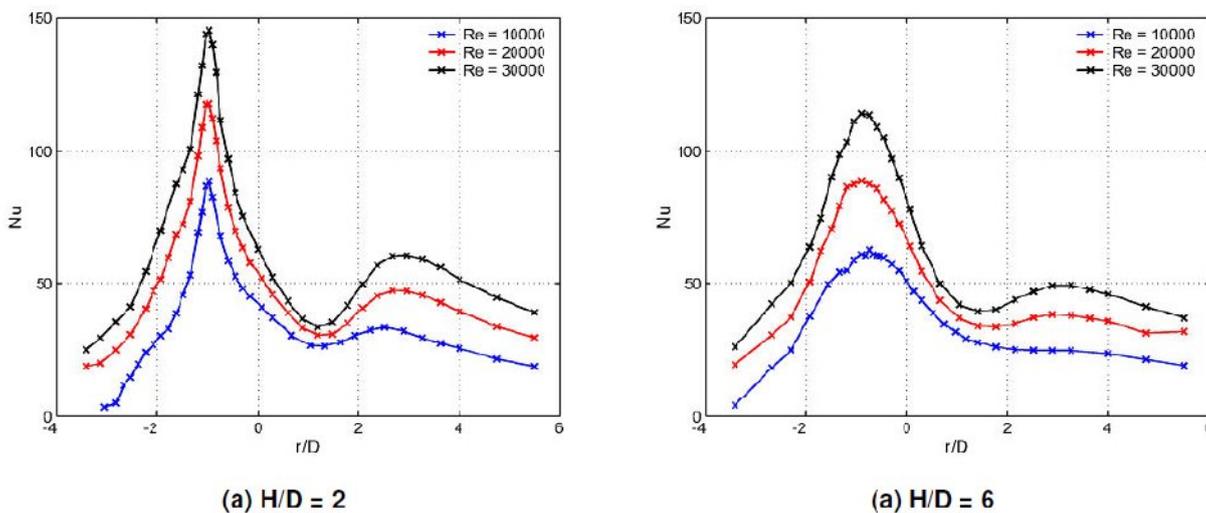


Figure 2.5: Graphs of obliquely Impinging Jet Mean Nusselt Number Distributions
Angle= 45°

Source: Donovan (2005)

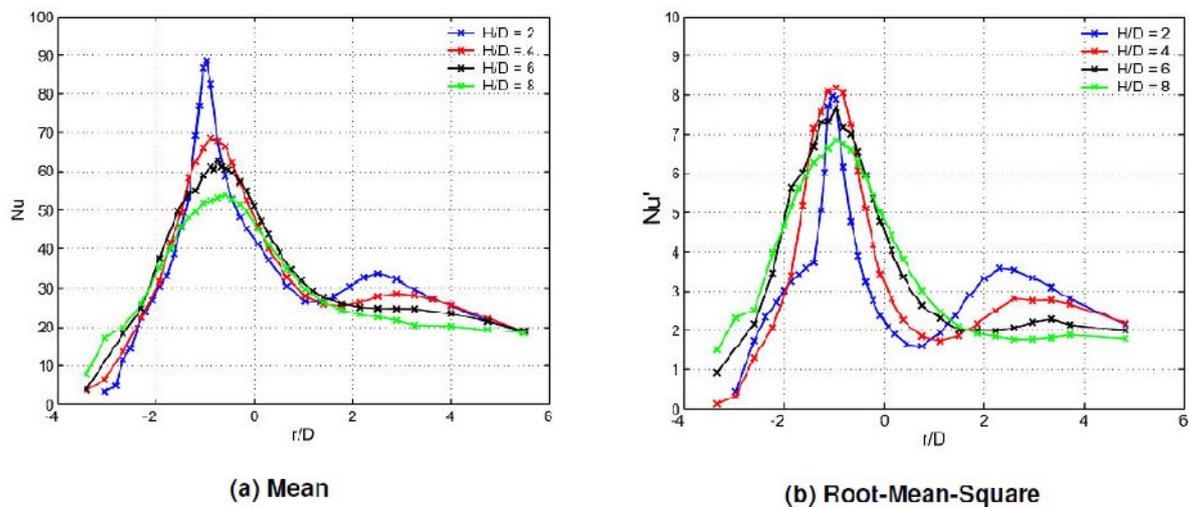


Figure 2.6: Graphs of obliquely Impinging Jet Nusselt Number Distributions; $Re = 10000$; Angle = 45°

Source: Donovan (2005)

Aharon and Womac (1991) have been shown that the jet-to-target spacing has a much greater influence on heat transfer for submerged jets than for free-surface jets. Many studies have shown little change in stagnation and average heat transfer for $H/d < 4$, then a decrease in heat transfer as H/d increases beyond this point. The relative consistency of heat transfer for $H/d < 4$ in the above studies can be explained by the jet impingement taking place within the potential core with its nearly uniform velocity, while the decrease in heat transfer at higher H/d values is attributed to complete degradation of the potential core prior to impingement. Baughn and Shimizu (1990) tested for H/d between 2 and 14 for a large air jet diameter (25mm) and Reynolds number of 23000 and found the maximum stagnation heat transfer coefficient to be at $H/d = 6$. However, Lytle and Webb show increasing heat transfer with decreasing H/d from 6 to 0.1.

Several researchers have shown the presence of secondary peaks for low non-dimensional jet-to-target spacing, with some studies showing that the secondary peaks increase in magnitude relative to stagnation heat transfer with increasing Reynolds number and decreasing H/d , eventually becoming global maxima. Lytle and Webb (1995) show that the position of the secondary peak moves toward the stagnation point

with decreasing H/d . The minimum position of the secondary peak measured by Lytle and Webb (1995) was at a non-dimensional radial distance of $r/d \approx 1.2$. All secondary peaks in the other above studies occurred at $1.5 \leq r/d \leq 2$. Secondary peaks in this region can be explained by transition to turbulence within the wall jet by Fitzgerald and Garimella (1998). Decreasing H/d increases the turbulence levels close to the impingement surface.

Royne and Dey (2006) investigated the effect of nozzle geometry on the heat transfer and pressure drop to confined-submerged jet arrays over a Reynolds number range of $1000 \leq Re \leq 7700$. Four different geometries were investigated with sharp-edged nozzles experiencing the largest heat transfer coefficient and countersunk nozzles encountering the lowest. In terms of pressure drop, the same was again true with sharp-edged nozzles experiencing the greatest pressure drop and countersunk nozzles the lowest. These results were attributed to their respective favorable and unfavorable discharge coefficients. Two conventional straight nozzles were also examined and the array of greater plate thickness, and therefore longer development length, experienced a smaller pressure drop than its thinner counterpart. This was accredited to the formation of a separation bubble at the inlet, which reattached for the nozzle of greater thickness thus increasing the discharge coefficient. With regards to heat transfer for a given pumping power, it was reported that the sharp-edged and contoured nozzles both experienced enhanced performances in comparison to the conventional straight nozzle arrays. The sharp-edged nozzle array decreased the flow rate required for a given heat transfer, whilst the countersunk nozzle required an increased flow rate at a reduced pressure drop.

CHAPTER 3

RESEARCH METHODOLOGY

3.0 INTRODUCTION

In this project, experiment will be conducted by varying the angles of the nozzle, Reynolds number (by changing air velocities), and H/d (by changing the distance of nozzle's end from specimen). Angle of the nozzle will vary at 30°, 45°, 60°, and 90°. Reynolds Number varies at 2300, 1960, 930, and 500. Meanwhile, H/d will vary at 2, 4, 6, and 8. Research design and approach will be described clearly in the flow chart, apparatus, experiment setup and procedures. Flow chart will show the overall flow of the project's process. In other hand, the research methodology for this project will need equivalent apparatus and experimental setup in order to achieve the determination of the project. The collected data from the experiment will be used for further analysis.

3.1 FLOW CHART

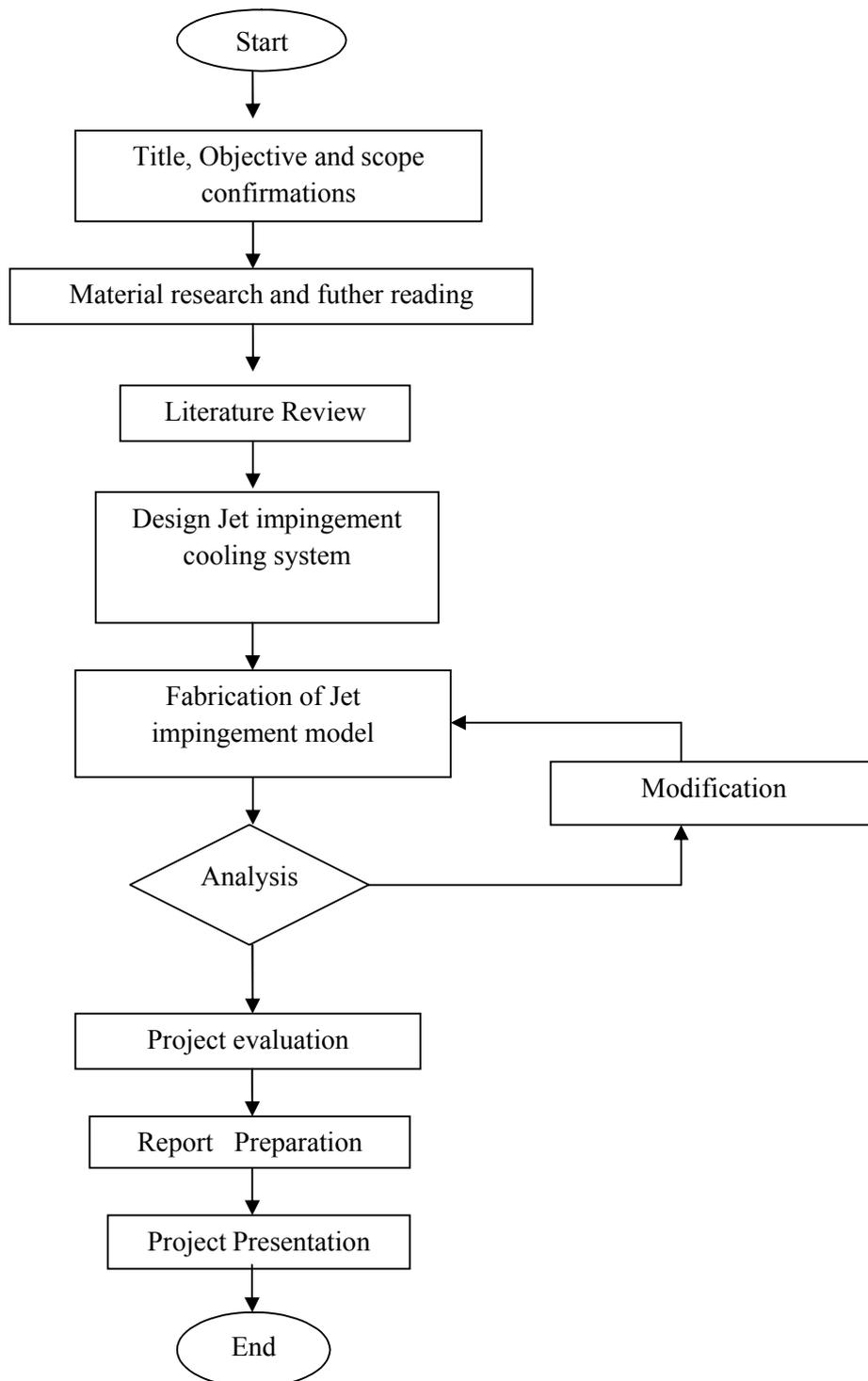


Figure 3.1: The flow diagram of the project

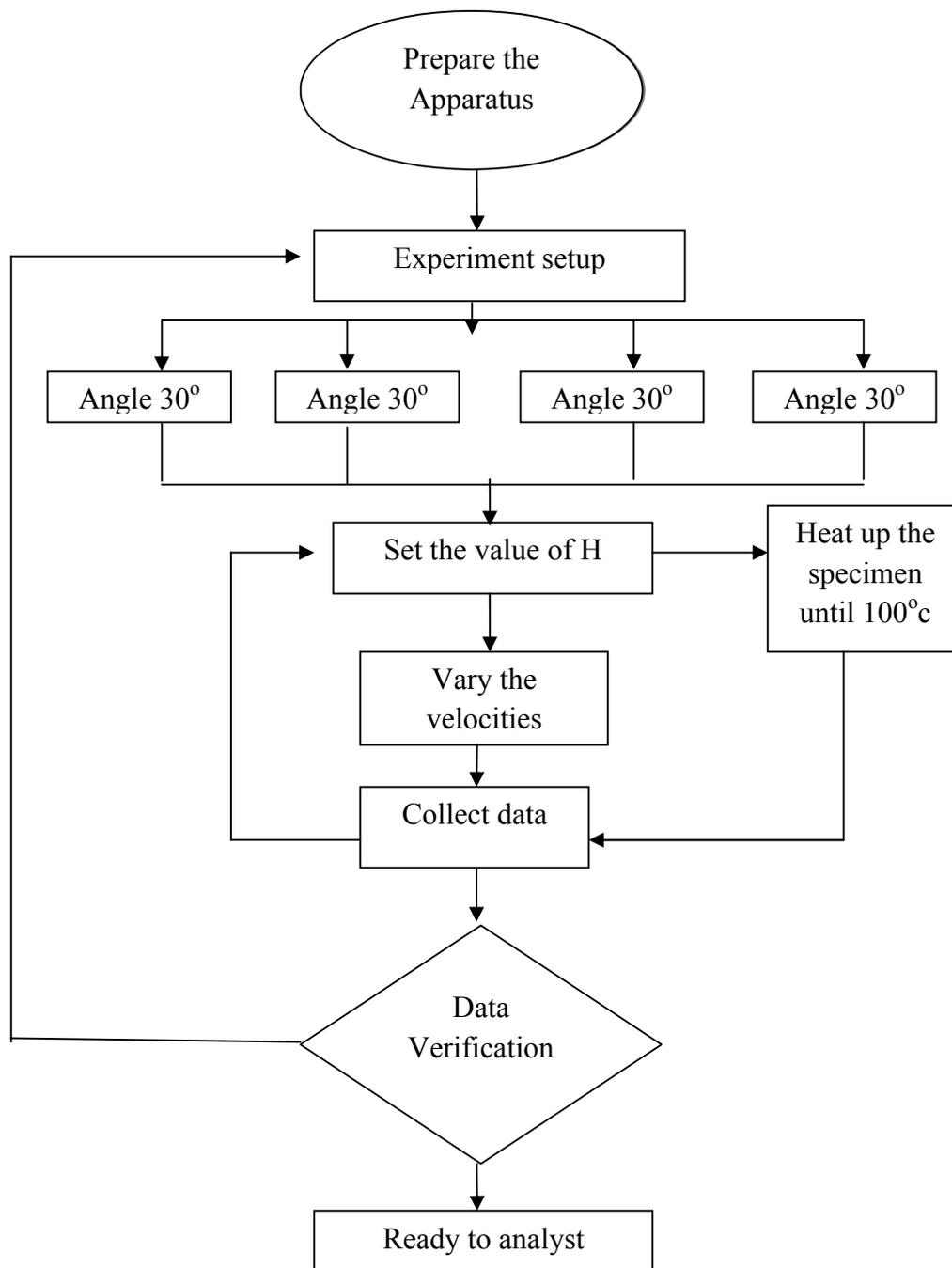


Figure 3.2 : The flow diagram of the experiment

3.2 APPARATUS

Table 3.1 : List of the Equipments typically required for the experiment

Equipment	Description
Steel plate (12cm x 12cm x 0.8 cm)	Steel plate is use as specimens that will be heat at 100°C. Then this specimen will be cool by the air flow through the nozzle. At the surface of the plate, aluminum wall at diameter, $D= 6.5$ cm are attached
Aluminum Nozzle	Nozzle hose with constant diameter, d of 10 mm will be use as air flow carrier.
Retort Stand	Retort stand used to support and holding the nozzle. It also used to vary the nozzle's angle displacement.
Anemometer	Anemometer is a device to measure the air flow velocity. It also can measure the room temperature.
Laser thermocouple	Laser thermocouple is a device to measure the temperature.
Air Regulator	Control the air velocity that flow to the nozzle from the air pipe.
Heater	Heater is use to heat the specimen to 100°C temperature. The temperature is set constant.

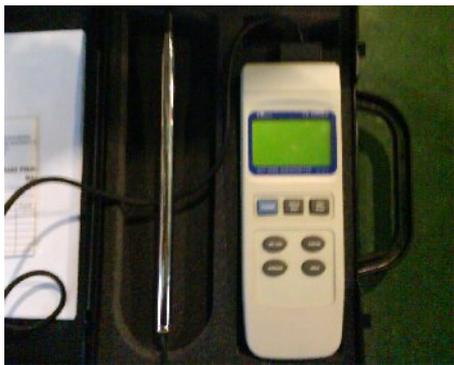


Figure 3.3: Anemometer



Figure 3.4: Laser Thermometer



Figure 3.5: Air Regulator



Figure 3.6: Air Pipe

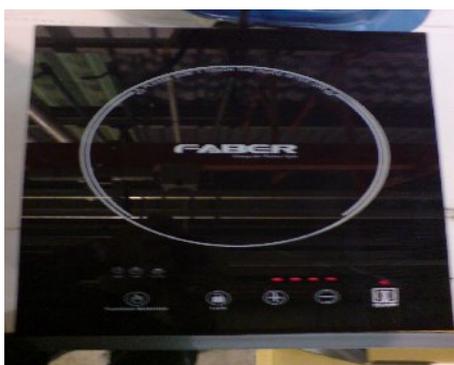


Figure 3.7: Heater

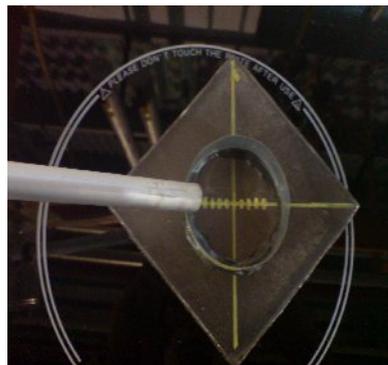


Figure 3.8: Steel plate

3.3 EXPERIMENTAL SETUP

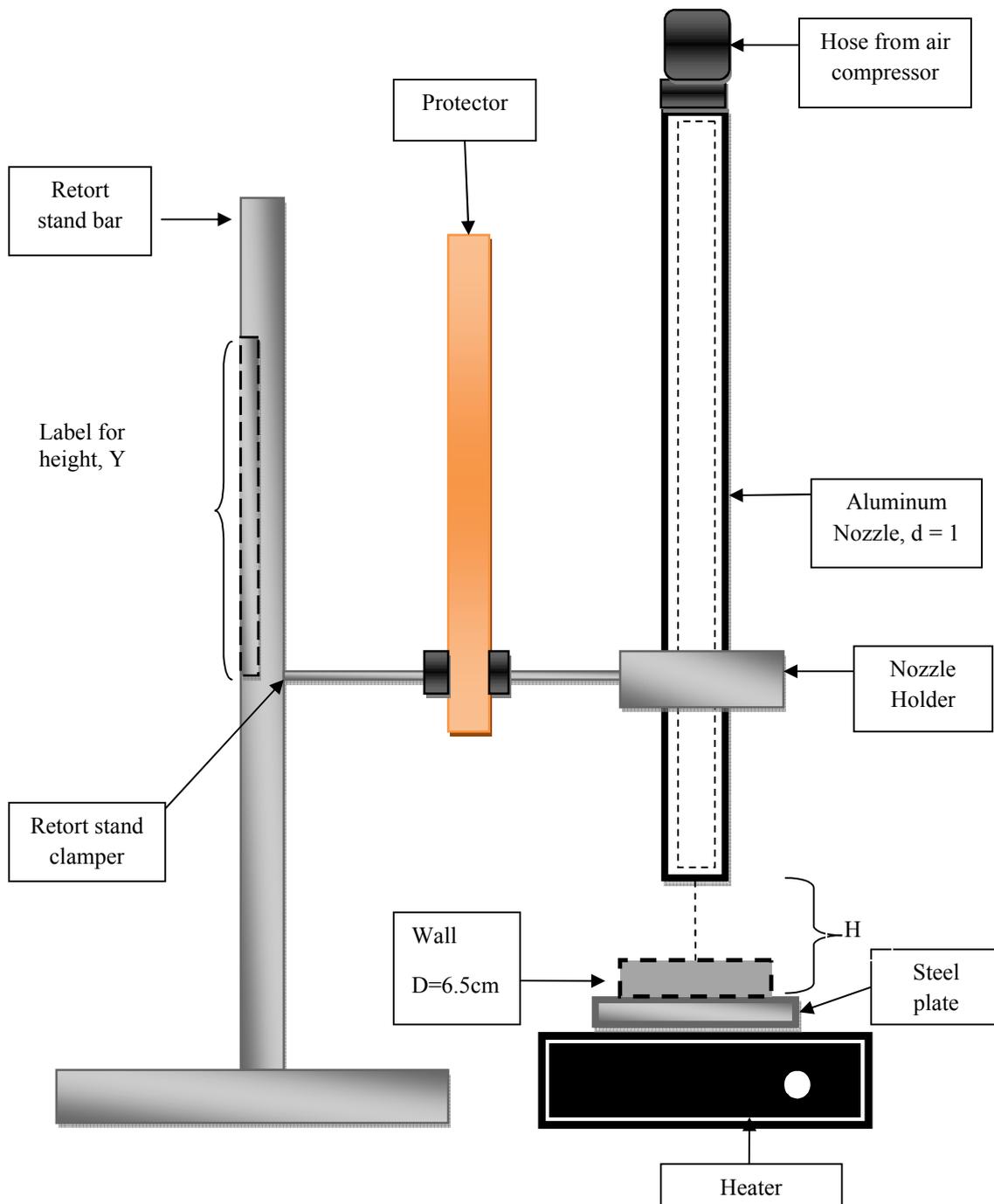


Figure 3.9: Experimental setup

3.3.1 Retort Stands Scale Labeling

1. Attach the nozzle onto retort stand and setup all the apparatus as in figure.
2. Loose up the clamps and let the end of nozzle touch onto the specimen.
3. Tighten the clamps and mark at that level as an initial point for height scale.
4. Start scaling at initial point using ruler.

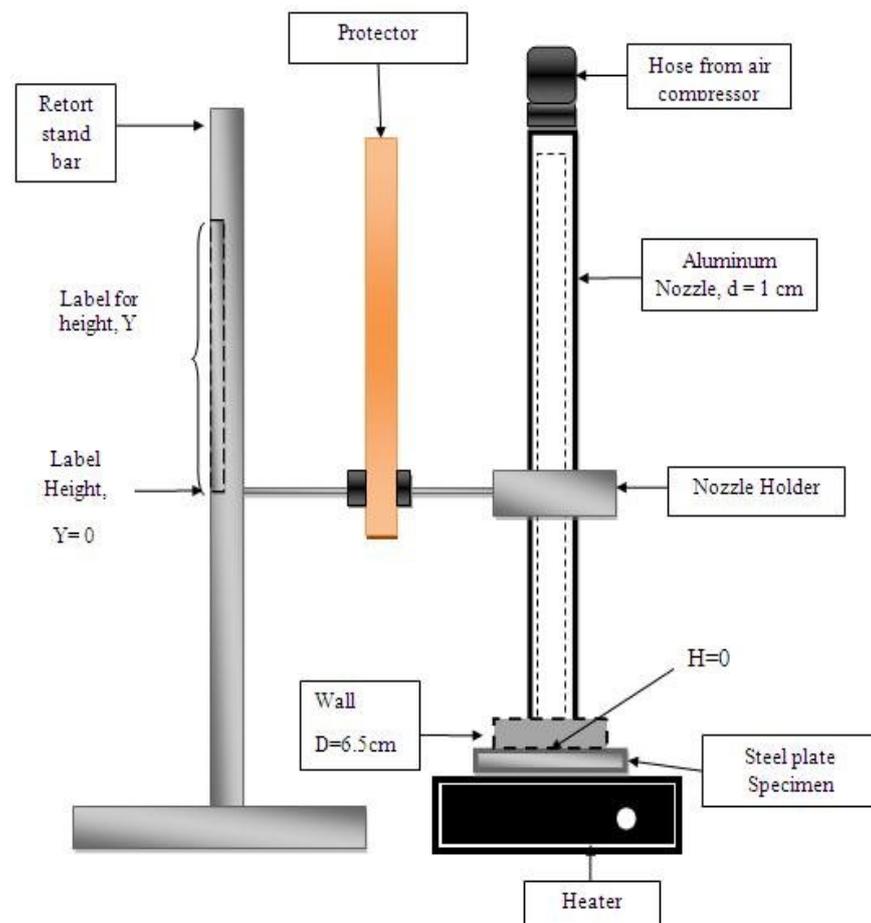


Figure 3.10: Experiment setup for labeling height, y scale at retort stand bar (side view).

3.3.2 Labeling Specimen

1. Make sure specimen have 12 cm long, 12cm wide and 0.8 cm thickness.
2. Mark the center point of specimen, denote as stagnation point
3. At the left side of the specimen surface label the scale and denote as x scale.
4. Besides that, marks 5 point to the right side of the specimen surface start from stagnation point. The length of each point is 0.5 cm.

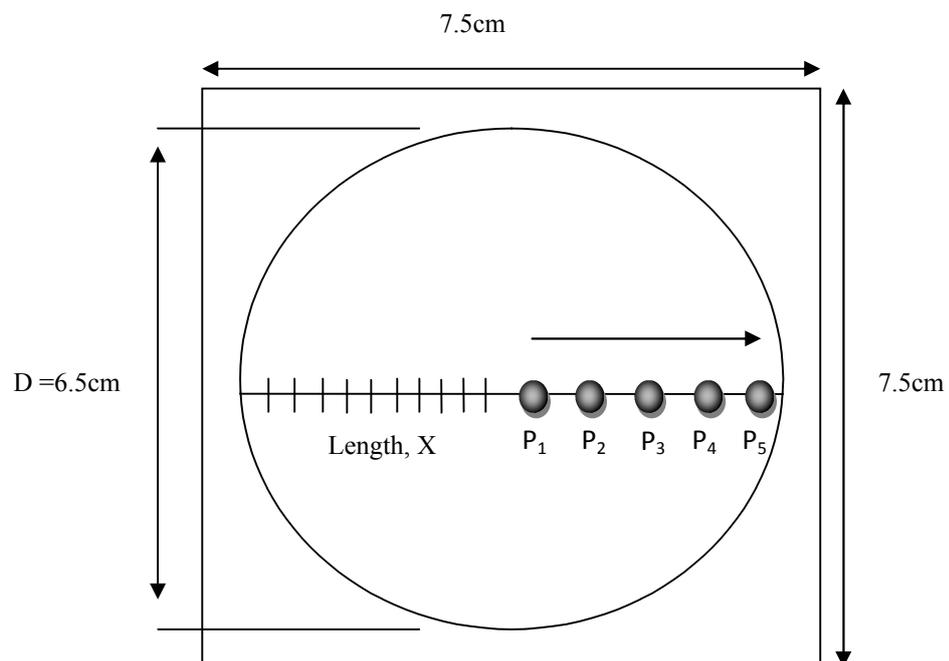


Figure 3.11: Labeling specimen

3.3.3 Angle Setup

For angles setup theorem Pythagoras will be use to determine the angle of nozzle. That's why labels height, Y scale and length, X scale are used to accomplish it. To make sure the angle of nozzle are correct the protector are used to verify. Furthermore, measuring tape are used to verify the length of the height of Y and the length of X.

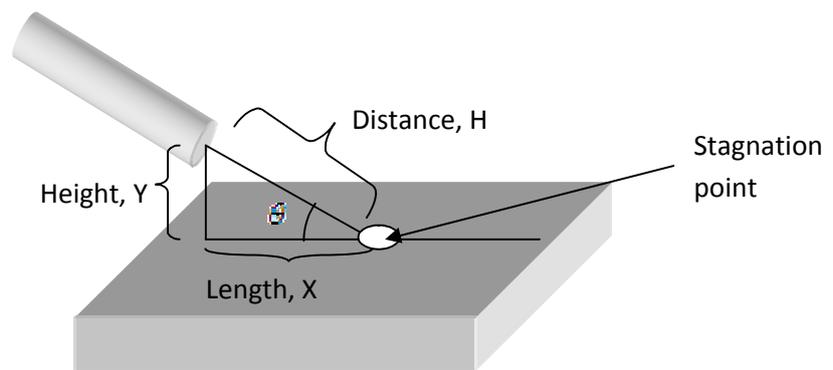


Figure 3.12: 3-D view for angles setup by using Theorem Pythagoras

Which,

$$Y = H \cos \Theta \text{ cm}$$

$$X = H \sin \Theta \text{ cm}$$

3.4 PROCEDURES

1. Prepare all the apparatus needed and make sure experiment setup already done.
2. Before get start, measure the room temperature by using the anemometer.
3. For the first angle at 30° , as mention in experiment setup we set the nozzle at 30° . At the same time start heat up the specimen using heater.
4. Set the distance between end of nozzle and specimen, H at 2 cm.
5. When the specimen already at 100°C , open the valve of air regulator to let the air flow through the nozzle. By using anemometer, air velocity can be measure and set to 0.8 m/s at the end of the nozzle.
6. Measure the temperature at 5 point start at stagnation point at P1 until P5, by using laser thermometer.
7. Repeat procedure 4 and 5 by using air velocity at 1.528 m/s, 3.152 m/s, and 3.698 m/s.
8. Take the data measured and fill into the table.
9. Then, repeat procedure 3 until procedure 6 by setting the distance, H at 4 cm, 6 cm, and 8 cm.
10. Take the data measured and fill into the table.
11. Repeat procedure 2 until procedure 8 at by setup the angles at 45° , 60° , and 90° .
12. Take the data measured and fill into the table.
13. Measure room temperature at the end of the experiment.

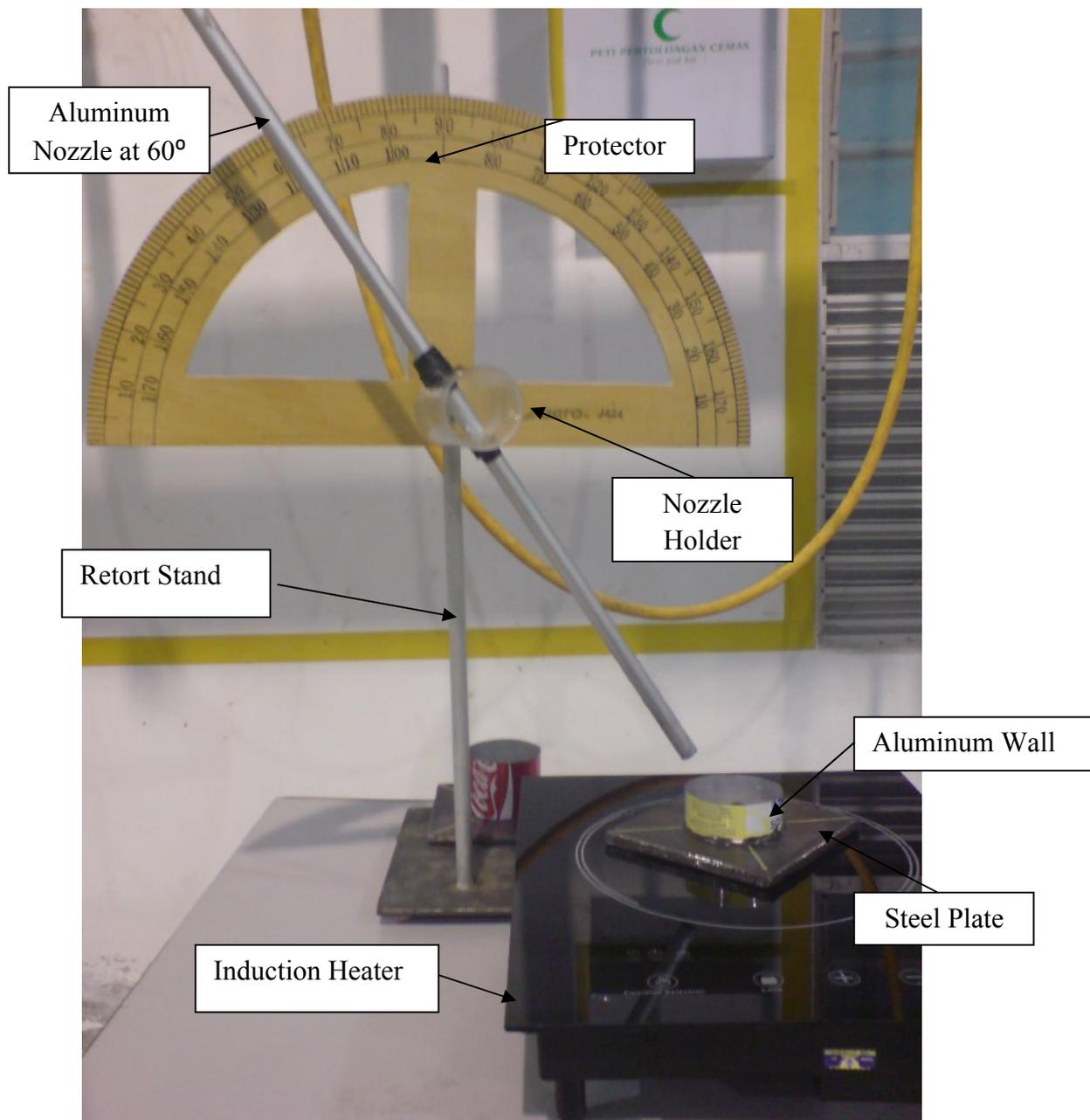


Figure 3.13: Actual Experimental Setup



Figure 3.14: Steel Plate heated at 100°C

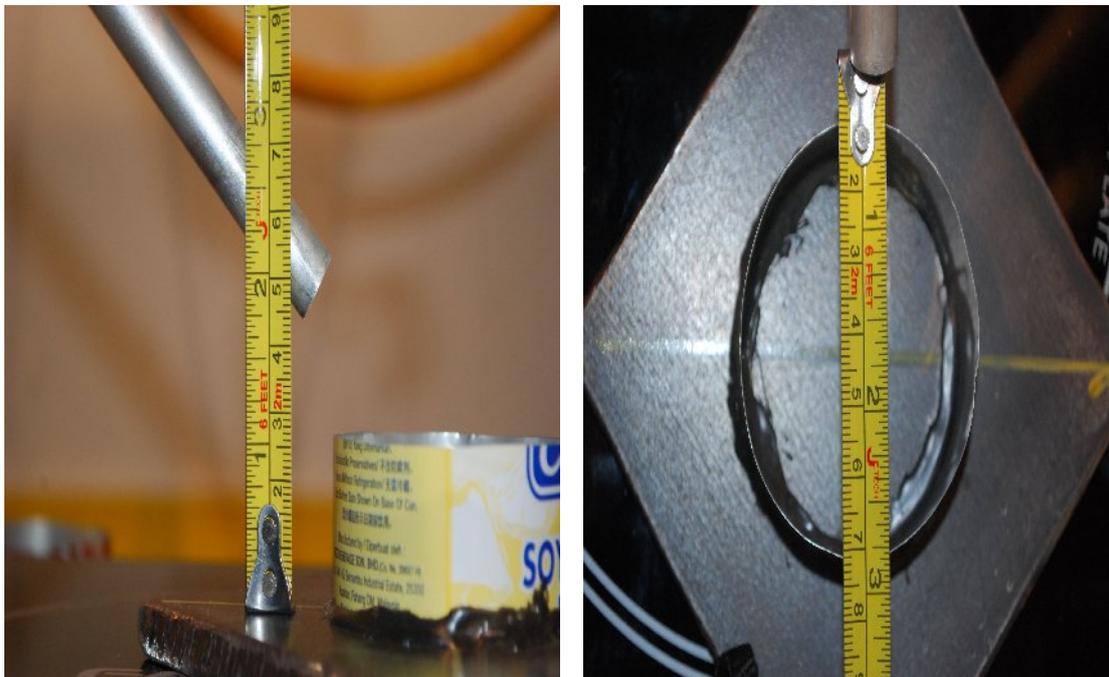


Figure 3.15: Angle Setup for 45°, $H/d=6$, Height $Y=4.3\text{cm}$ and $X=4.3\text{cm}$

3.5 DATA DISTRIBUTION

Table 3.2: Data distribution for experiment

Angle of nozzle	Y (cm) $Y = H \cos \Theta$	X (cm) $X = H \sin \Theta$	H (cm)	H/d	Re	V (m/s)
30°	1.732	1	2	2	500	0.8
	3.464	2	4	4		
	5.196	3	6	6		
	6.928	4	8	8		
	1.732	1	2	2	930	1.528
	3.464	2	4	4		
	5.196	3	6	6		
	6.928	4	8	8		
	1.732	1	2	2	1960	3.152
	3.464	2	4	4		
	5.196	3	6	6		
	6.928	4	8	8		
	1.732	1	2	2	2300	3.698
	3.464	2	4	4		
	5.196	3	6	6		
	6.928	4	8	8		
45°	1.414	1.414	2	2	500	0.8
	2.828	2.828	4	4		
	4.243	4.243	6	6		
	5.659	5.659	8	8		
	1.414	1.414	2	2	930	1.528
	2.828	2.828	4	4		
	4.243	4.243	6	6		
	5.659	5.659	8	8		
	1.414	1.414	2	2	1960	3.152
	2.828	2.828	4	4		
	4.243	4.243	6	6		
	5.659	5.659	8	8		
	1.414	1.414	2	2	2300	3.698
	2.828	2.828	4	4		
	4.243	4.243	6	6		
	5.659	5.659	8	8		

Table 3.2: Continued

Angle of nozzle	Y (cm) $Y = H \cos \Theta$	X(cm) $X = H \sin \Theta$	H (cm)	H/d	Re	V (m/s)
60°	1	1.732	2	2	500	0.8
	2	3.464	4	4		
	3	5.196	6	6		
	4	6.928	8	8		
	1	1.732	2	2	930	1.528
	2	3.464	4	4		
	3	5.196	6	6		
	4	6.928	8	8		
	1	1.732	2	2	1960	3.152
	2	3.464	4	4		
	3	5.196	6	6		
	4	6.928	8	8		
	1	1.732	2	2	2300	3.698
	2	3.464	4	4		
	3	5.196	6	6		
	4	6.928	8	8		
90°	-	-	2	2	500	0.8
	-	-	4	4		
	-	-	6	6		
	-	-	8	8		
	-	-	2	2	930	1.528
	-	-	4	4		
	-	-	6	6		
	-	-	8	8		
	-	-	2	2	1960	3.152
	-	-	4	4		
	-	-	6	6		
	-	-	8	8		
	-	-	2	2	2300	3.698
	-	-	4	4		
	-	-	6	6		
	-	-	8	8		

For Velocity, V: Varies velocity(0.8 m/s, 1.528 m/s, 3.152 m/s, and 3.698 m/s)

Height, H: Varies distance, H (4 cm, 6 cm, and 8 cm)

Table 3.3: Sample of results table

No	Plate surface Temperature, T_s (°C)																			
	$T_1=$					$T_2=$					$T_{\infty}=$									
	30°					45°					60°					90°				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
1																				
2																				
3																				
4																				
5																				

3.6 EXPERIMENT PROCESS FLOW

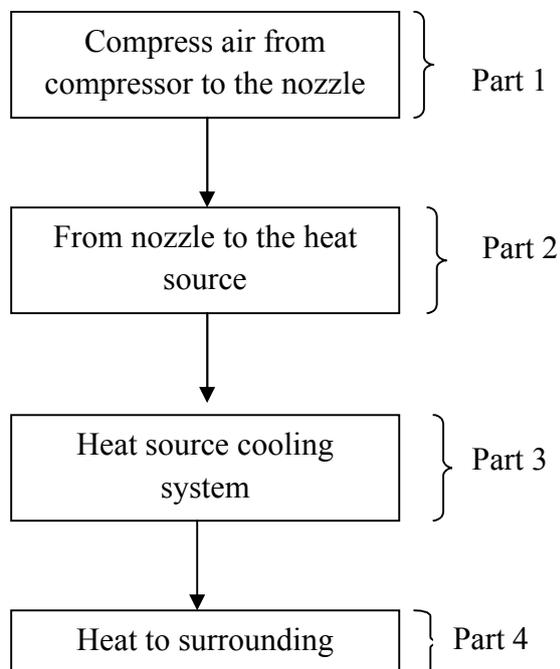


Figure 3.16: Experiment process flow

- i. Part 1
 - The compressor is attached to the nozzle that carry the air flow to the nozzle exit and the velocity of air is set by controlling the pressure value according to the Reynolds Number. The air flow velocity measured by using anemometer.
- ii. Part 2
 - The nozzle carries the air flow to the specimen and cooled down the specimen in about 15 seconds.
- iii. Part 3
 - Specimen temperature will be decreasing as heat transfer occurs in the system. Temperature after impingement on the specimen taken at five (5) different points on the specimen.
- iv. Part 4
 - The heat in the plate released to the surrounding.

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

In this chapter, all of the raw results will be rearranged and the selected finding will be discussed briefly to give the proper explanation about the analysis and the important point of the result. All evaluated data calculated and some in order to get the value of heat transfer coefficient and Nusselt number for each case. The relationship of different angles on the jet impingement cooling system will be discussed.

4.2 CALCULATION

Conservation Energy:

$$\dot{Q}_{\text{conv}} = hAs(T_{\text{impinge surface}} - T_{\infty}) = \dot{m}c_p\Delta T$$

Heat Transfer Coefficient, h;

$$h = \frac{\dot{m} c_p (T_s - T_{\text{impinge surface}})}{As(T_s - T_{\infty})} \quad (4.1)$$

Nusselt Number, Nu;

$$Nu = \frac{hL_c}{k} = \frac{hr_c}{k}$$

Those are parameters;

i. Average surrounding temperature, $T_{\infty} = \frac{30.1^{\circ}\text{C} + 32.2^{\circ}\text{C}}{2}$
 $= 31.15^{\circ}\text{C}$

ii. From Table-15 (APPENDIX A), at 31.15°C ,

$$\rho_{\text{air}} = 1.1596 \text{ Kg/m}^3$$

$$C_p = 1007 \text{ J/Kg.K}$$

$$k = 0.02597 \text{ w/m.k}$$

iii. Area for wall, A_s

$$A_s = \frac{\pi (0.065)^2}{4}$$

$$= 0.003318 \text{ m}^2$$

iv. Area of Nozzle, A_c

$$A_c = \frac{\pi (0.01)^2}{4}$$

$$= 7.854 \times 10^{-5} \text{ m}^2$$

v. Mass flow rate, \dot{m}

- For Reynolds Number 2300, $V = 3.698 \text{ m/s}^2$

$$\dot{m} = \rho_{\text{air}} V A_c$$

$$= 3.3607 \times 10^{-4} \text{ kg/s}$$

- For Reynolds Number 1960, $V = 3.152 \text{ m/s}^2$

$$\dot{m} = \rho_{\text{air}} V A_c$$

$$=2.8708 \times 10^{-4} \text{ kg/s}$$

- For Reynolds Number 930, $V=1.528 \text{ m/s}^2$

$$\begin{aligned} \dot{m} &= \rho_{\text{air}} V A_c \\ &= 1.3917 \times 10^{-4} \text{ kg/s} \end{aligned}$$

- For Reynolds Number 500, $V=0.8 \text{ m/s}^2$

$$\begin{aligned} \dot{m} &= \rho_{\text{air}} V A_c \\ &= 7.2862 \times 10^{-5} \text{ kg/s} \end{aligned}$$

The temperature after impingement at the specimen are taken based on point prepared on the heat source plate. The readings for every point are taken repeated three (3) times and average or the best temperatures are taken. All the calculated data for heat transfer coefficient, h and Nusselt number, Nu are arranged in the table and graph are plotted (Refer APPENDIX B, APPENDIX C and APPENDIX D).

4.3 TABLES AND GRAPHS

There are affecting tables and graph that is really important in this project to achieve the objectives.

4.3.1 At constant Reynolds Number ($Re=2300$) at various $H/d= (2, 4, 6, \text{ and } 8)$ cm.

Table 4.1: Data Configurations for Re=2300, H/d=2

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^{\circ}\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{-K}$)	Nusselt Number, Nu
$\theta = 30^{\circ}$	0.0	89.2	19.798	24.776
	0.5	89.7	18.720	23.427
	1.0	89.8	18.507	23.160
	1.5	90.1	17.871	22.365
	2.0	90.3	17.451	21.839
$\theta = 45^{\circ}$	0.0	87.3	24.069	30.121
	0.5	88.9	20.454	25.597
	1.0	89.4	19.365	24.234
	1.5	88.5	21.339	26.704
	2.0	88.9	20.454	25.597
$\theta = 60^{\circ}$	0.0	85.4	28.639	35.840
	0.5	86	27.162	33.991
	1.0	86.7	25.478	31.885
	1.5	87.7	23.146	28.966
	2.0	88.9	20.454	25.597
$\theta = 90^{\circ}$	0.0	84.3	31.434	39.338
	0.5	85.3	28.888	36.152
	1.0	86.7	25.478	31.885
	1.5	87.2	24.302	30.412
	2.0	88.4	21.562	26.983

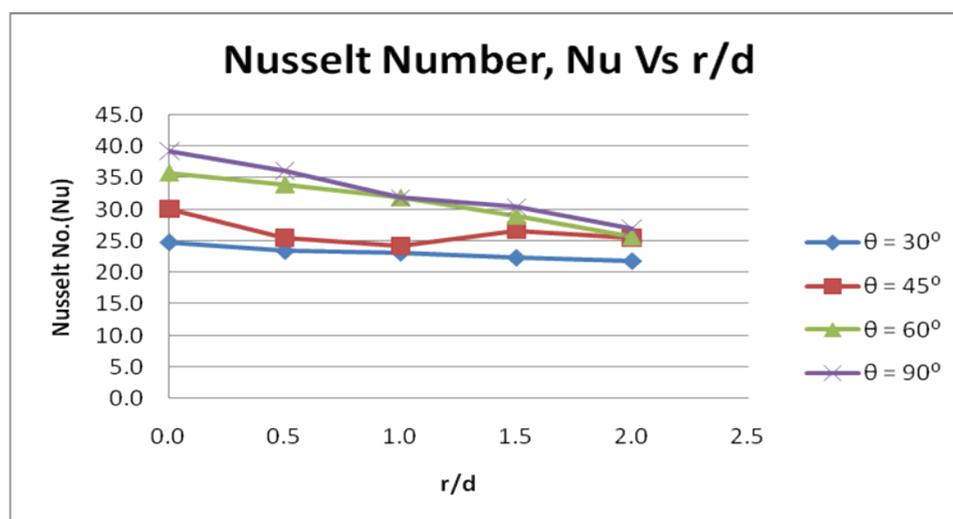
**Figure 4.1:** Graph Nusselt Number, Nu vs. r/d at Re=2300, H/d=2

Table 4.2: Data Configurations for Re=2300, H/d=4

Nozzle Angle	r/d	T _{plate Impinge surface} (°C)	Heat Transfer Coefficient, h (W/m ² ·K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	90.2	17.661	22.101
	0.5	90.3	17.451	21.839
	1.0	90.8	16.413	20.540
	1.5	91.4	15.190	19.009
	2.0	91.9	14.189	17.756
$\theta = 45^\circ$	0.0	89.7	18.720	23.427
	0.5	90.9	16.207	20.282
	1.0	91.8	14.388	18.005
	1.5	91.3	15.392	19.262
	2.0	90.2	17.661	22.101
$\theta = 60^\circ$	0.0	87.8	22.917	28.680
	0.5	88.2	22.010	27.545
	1.0	89.3	19.581	24.505
	1.5	90.5	17.034	21.317
	2.0	91.4	15.190	19.009
$\theta = 90^\circ$	0.0	86.7	25.478	31.885
	0.5	87.1	24.535	30.705
	1.0	88.8	20.674	25.872
	1.5	90.3	17.451	21.839
	2.0	92.4	13.204	16.524

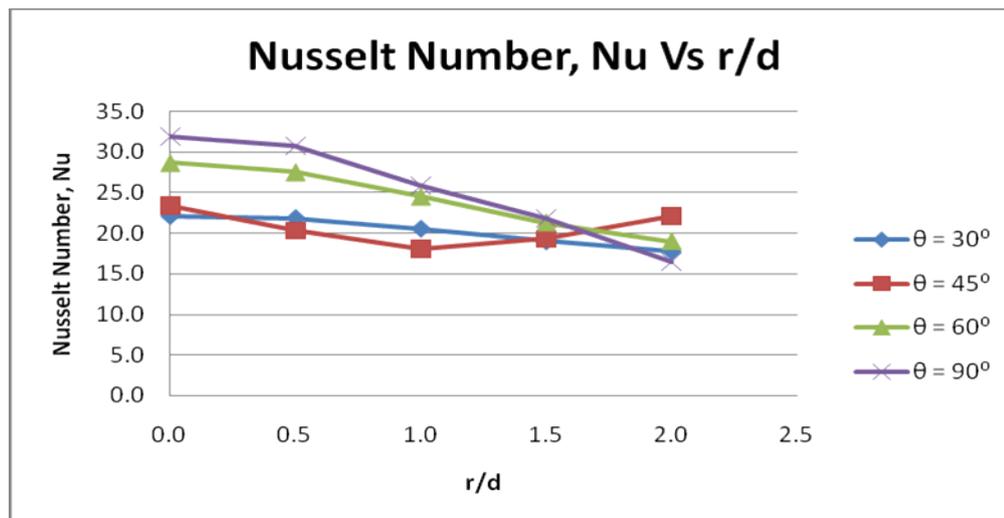
**Figure 4.2:** Graph Nusselt Number, Nu vs. r/d at Re=2300, H/d=4

Table 4.3: Data Configurations for Re=2300, H/d=6

Nozzle Angle	r/d	T _{plate Impinge surface} (°C)	Heat Transfer Coefficient, h (W/m ² K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	93.3	11.472	14.356
	0.5	93.6	10.906	13.648
	1.0	93.9	10.345	12.946
	1.5	94.5	9.239	11.562
	2.0	95.7	7.089	8.871
$\theta = 45^\circ$	0.0	91.5	14.988	18.757
	0.5	92.9	12.236	15.312
	1.0	93.4	11.283	14.119
	1.5	92.8	12.428	15.553
	2.0	92.7	12.621	15.795
$\theta = 60^\circ$	0.0	89.7	18.720	23.427
	0.5	90.3	17.451	21.839
	1.0	92	13.990	17.508
	1.5	93.4	11.283	14.119
	2.0	94.3	9.605	12.020
$\theta = 90^\circ$	0.0	88.3	21.786	27.264
	0.5	90.5	17.034	21.317
	1.0	91.4	15.190	19.009
	1.5	93.8	10.531	13.179
	2.0	94.1	9.974	12.482

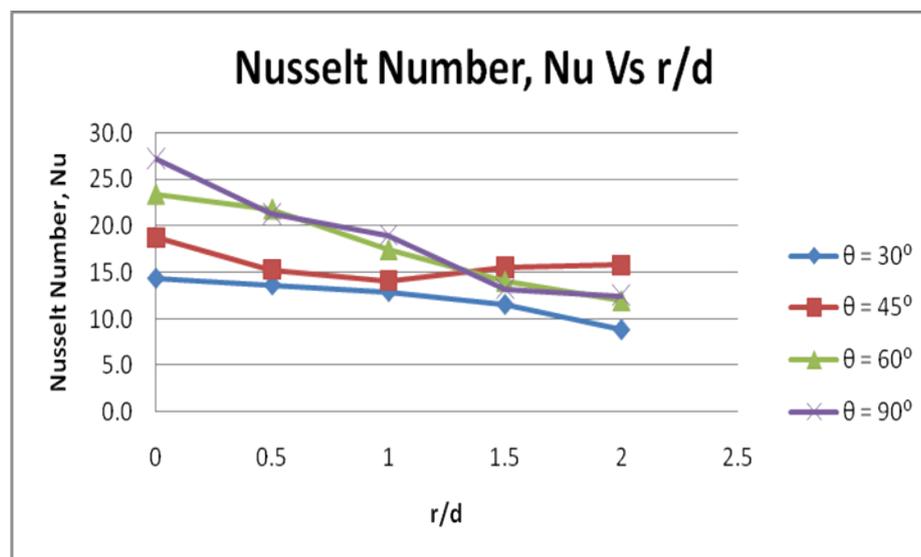
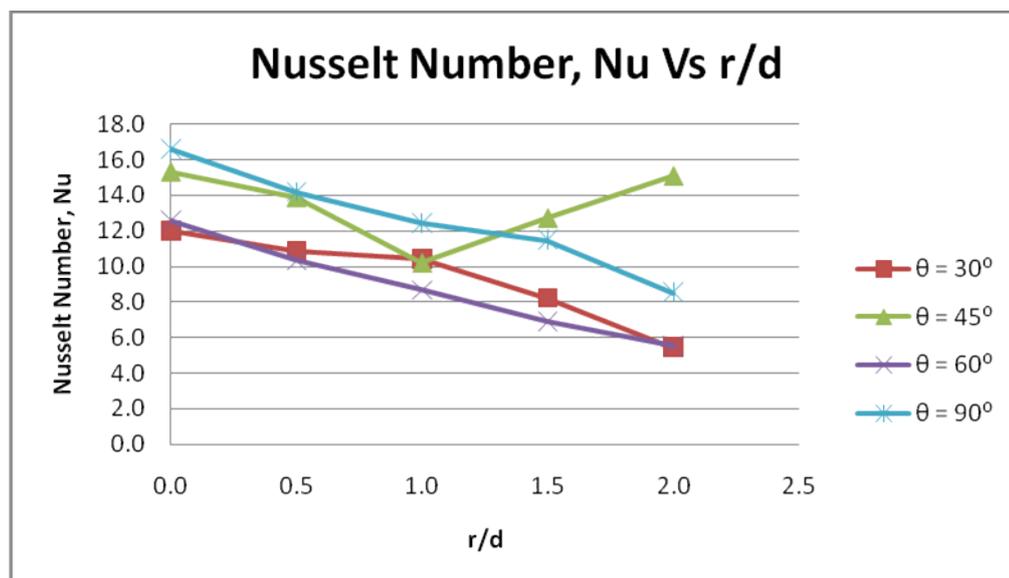
**Figure 4.3:** Graph Nusselt Number, Nu vs. r/d at Re=2300, H/d=6

Table 4.4: Data Configurations for Re=2300, H/d= 8

Nozzle Angle	r/d	T _{plate Impinge surface} (°C)	Heat Transfer Coefficient, h (W/m ² ·K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	94.3	9.605	12.020
	0.5	94.8	8.694	10.880
	1.0	95.0	8.333	10.429
	1.5	96.0	6.564	8.214
	2.0	97.3	4.343	5.436
$\theta = 45^\circ$	0.0	92.9	12.236	15.312
	0.5	93.5	11.094	13.883
	1.0	95.1	8.154	10.204
	1.5	94.0	10.159	12.713
	2.0	93.0	12.044	15.072
$\theta = 60^\circ$	0.0	92.7	12.621	15.795
	0.5	93.9	10.345	12.946
	1.0	94.8	8.694	10.880
	1.5	95.8	6.913	8.652
	2.0	96.6	5.528	6.918
$\theta = 90^\circ$	0.0	90.7	16.619	20.798
	0.5	91.9	14.189	17.756
	1.0	92.8	12.428	15.553
	1.5	93.3	11.472	14.356
	2.0	94.9	8.513	10.654

**Figure 4.4:** Graph Nusselt Number, Nu vs. r/d at Re=2300, H/d=8

4.3.2 At constant H/d= 2 cm and at various Reynolds Number .

Table 4.5: Data Configurations for Re=2300, H/d= 2

Nozzle Angle	r/d	T _{plate Impinge surface} (°C)	Heat Transfer Coefficient, h (W/m ² ·K)	Nusselt Number, Nu
θ = 30°	0.0	89.2	19.798	24.776
	0.5	89.7	18.720	23.427
	1.0	89.8	18.507	23.160
	1.5	90.1	17.871	22.365
	2.0	90.3	17.451	21.839
θ = 45°	0.0	87.3	24.069	30.121
	0.5	88.9	20.454	25.597
	1.0	89.4	19.365	24.234
	1.5	88.5	21.339	26.704
	2.0	88.9	20.454	25.597
θ = 60°	0.0	85.4	28.639	35.840
	0.5	86	27.162	33.991
	1.0	86.7	25.478	31.885
	1.5	87.7	23.146	28.966
	2.0	88.9	20.454	25.597
θ = 90°	0.0	84.3	31.434	39.338
	0.5	85.3	28.888	36.152
	1.0	86.7	25.478	31.885
	1.5	87.2	24.302	30.412
	2.0	88.4	21.562	26.983

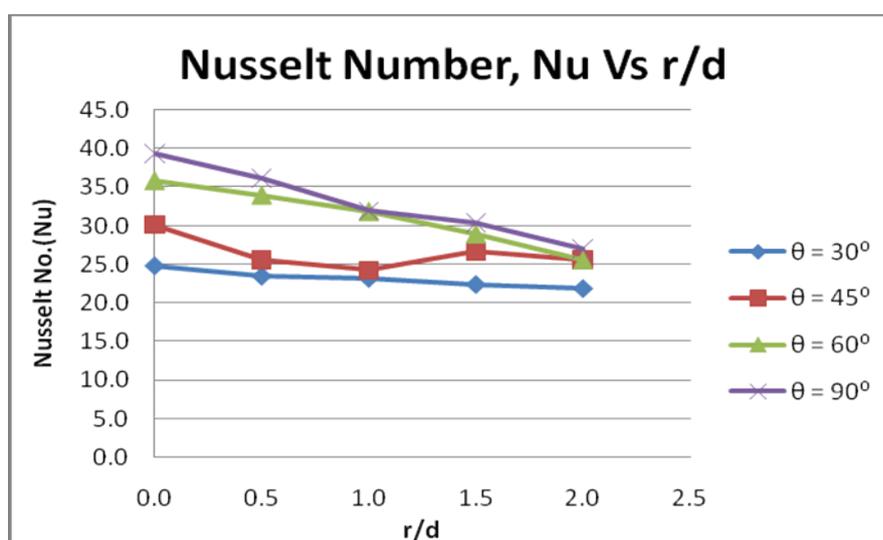


Figure 4.5: Graph Nusselt Number, Nu vs. r/d at Re=2300, H/d=2

Table 4.6: Data Configurations for Re=1960, H/d= 2

Nozzle Angle	r/d	T _{plate Impinge surface} (°C)	Heat Transfer Coefficient, h (W/m ² K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	91.9	12.122	15.169
	0.5	92.3	11.448	14.326
	1.0	92.9	10.453	13.081
	1.5	94.3	8.206	10.269
	2.0	93.7	9.157	11.459
$\theta = 45^\circ$	0.0	88.9	17.474	21.868
	0.5	89.7	15.993	20.014
	1.0	90.7	14.198	17.768
	1.5	88.8	17.662	22.103
	2.0	89.9	15.629	19.559
$\theta = 60^\circ$	0.0	86.4	22.378	28.005
	0.5	87.7	19.774	24.746
	1.0	88.9	17.474	21.868
	1.5	89.9	15.629	19.559
	2.0	90.2	15.088	18.882
$\theta = 90^\circ$	0.0	85.0	25.324	31.691
	0.5	86.6	21.970	27.494
	1.0	87.4	20.364	25.485
	1.5	88.0	19.190	24.015
	2.0	89.1	17.100	21.400

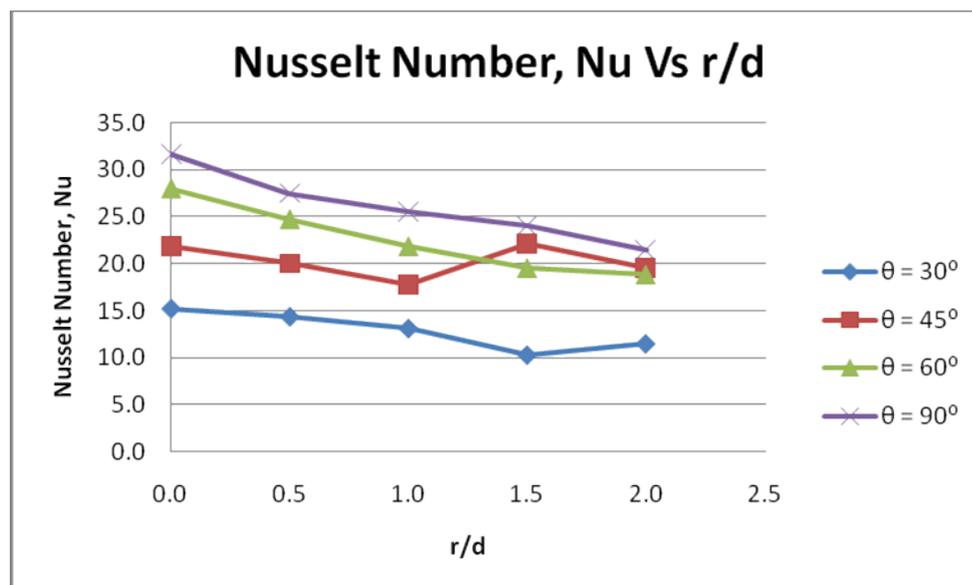
**Figure 4.6:** Graph Nusselt Number, Nu vs. r/d at Re=1960, H/d=2

Table 4.7: Data Configurations for $Re=930$, $H/d=2$

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}} (^{\circ}\text{C})$	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
$\theta = 30^{\circ}$	0.0	92.4	5.467	6.841
	0.5	92.6	5.305	6.639
	1.0	93.5	4.593	5.748
	1.5	93.7	4.437	5.553
	2.0	93.4	4.671	5.846
$\theta = 45^{\circ}$	0.0	88.3	9.019	11.287
	0.5	87.0	10.255	12.833
	1.0	90.7	6.880	8.610
	1.5	88.6	8.742	10.941
	2.0	89.8	7.662	9.589
$\theta = 60^{\circ}$	0.0	86.9	10.352	12.955
	0.5	88.2	9.112	11.404
	1.0	88.4	8.927	11.171
	1.5	89.2	8.197	10.258
	2.0	90.0	7.486	9.369
$\theta = 90^{\circ}$	0.0	85.4	11.857	14.838
	0.5	86.2	11.044	13.821
	1.0	87.6	9.678	12.111
	1.5	88.8	8.559	10.711
	2.0	89.2	8.197	10.258

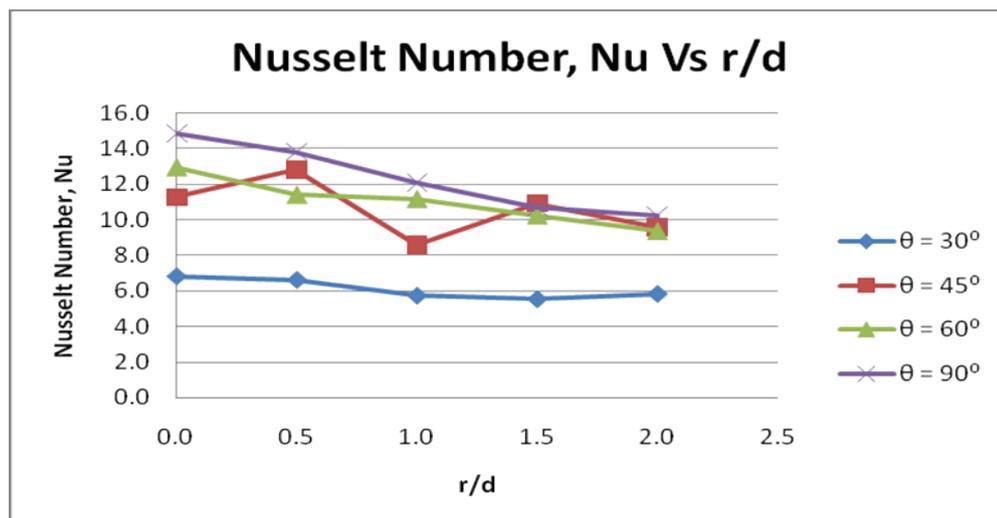
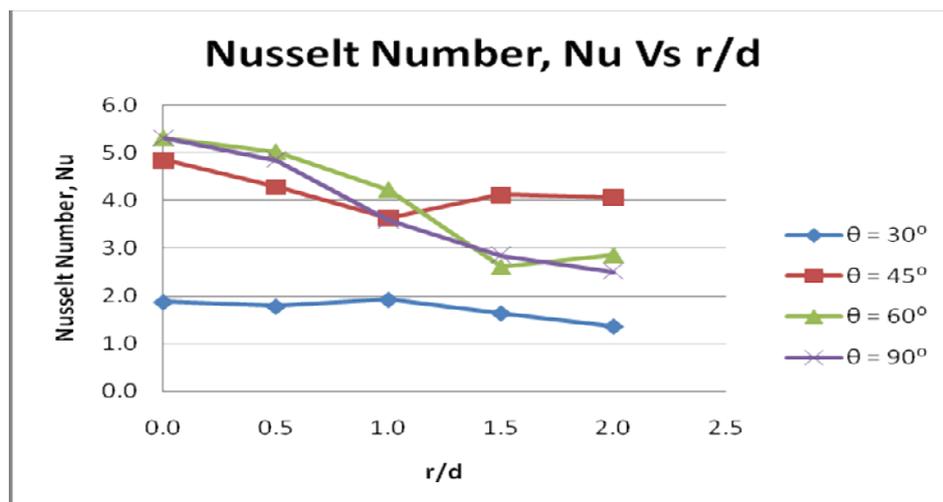
**Figure 4.7:** Graph Nusselt Number, Nu vs. r/d at $Re=930$, $H/d=2$

Table 4.8: Data Configurations for $Re=500$, $H/d=2$

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^{\circ}\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
$\theta = 30^{\circ}$	0.0	95.8	1.500	1.877
	0.5	96.0	1.424	1.782
	1.0	95.7	1.538	1.924
	1.5	96.3	1.311	1.640
	2.0	96.9	1.088	1.362
$\theta = 45^{\circ}$	0.0	90.1	3.876	4.851
	0.5	91.1	3.427	4.288
	1.0	92.3	2.906	3.637
	1.5	91.4	3.295	4.123
	2.0	91.5	3.251	4.068
$\theta = 60^{\circ}$	0.0	89.3	4.247	5.315
	0.5	89.8	4.014	5.024
	1.0	91.2	3.383	4.233
	1.5	94.3	2.083	2.607
	2.0	93.8	2.284	2.859
$\theta = 90^{\circ}$	0.0	89.3	4.247	5.315
	0.5	90.1	3.876	4.851
	1.0	92.4	2.864	3.584
	1.5	93.8	2.284	2.859
	2.0	94.5	2.004	2.508

**Figure 4.8:** Graph Nusselt Number, Nu vs. r/d at $Re=500$, $H/d=2$

4.4 ANGLE DISTRIBUTION

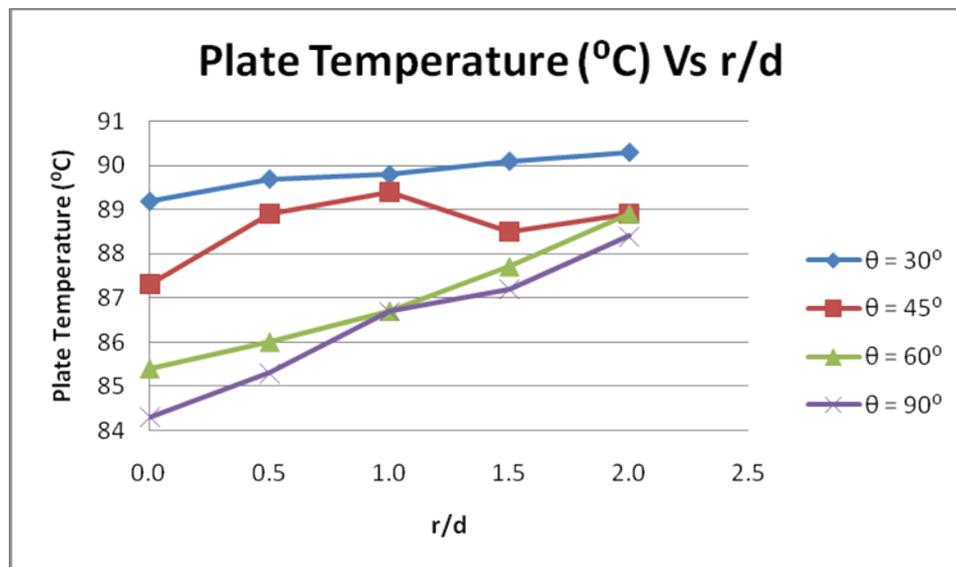


Figure 4.9: Graph Plate Temperature ($^\circ\text{C}$) vs. r/d at $\text{Re}=2300$, $H/d=2$

Figure 4.9, at Reynolds Number 2300 with $V=3.698$ m/s and $H/d=2$ show for each angles ($30^\circ, 45^\circ, 60^\circ$, and 90°) the temperature after cooling at stagnation point (P1), which $r/d=0.0$ is lower than other points. Secondly, at stagnation point for angle 90° gives the lowest temperature and the temperature at stagnation point for angle 30° gives the highest value. Cooling system decreasing as pointing points increase to the side of impinge plate surface. This is because the air flow at nozzle angle 90° is focusing to the stagnation point symmetrically and the flows are less spreading to the other points

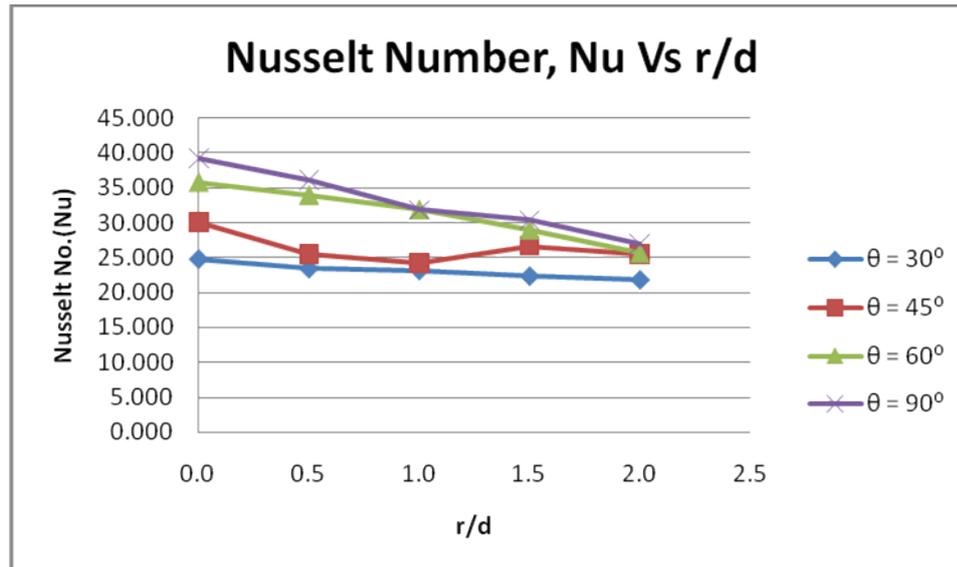


Figure 4.10: Graph Nusselt Number, Nu vs. r/d at Re=2300, H/d=2

Figure 4.10 present the Nusselt number distributions for Reynolds Number of 2300 and $H/d = 2$ at various angle. Nozzle angle at 90° gives highest Nusselt number compare to the other nozzle angle at stagnation points. This is shown as Nusselt number increase when nozzle angles are increasing to the normal line. The air flow are pointing to the stagnation point which contribute to a peak in the Nusselt number at stagnation points, while the air flow at 30° the air flow are widely spread to the plate surface. The figure 4.10 has been shown that at nozzle angle 30° graph line are more shelving compare to 90° which is more steep even the heat transfer at 30° is more lower than 90° . This make at nozzle angle 30° cooling systems more uniform at wide area even though nozzle angle at 90° the cooling system of jet impingement are more effective at stagnation points compare to the other angles.

4.5 REYNOLDS NUMBER DISTRIBUTIONS

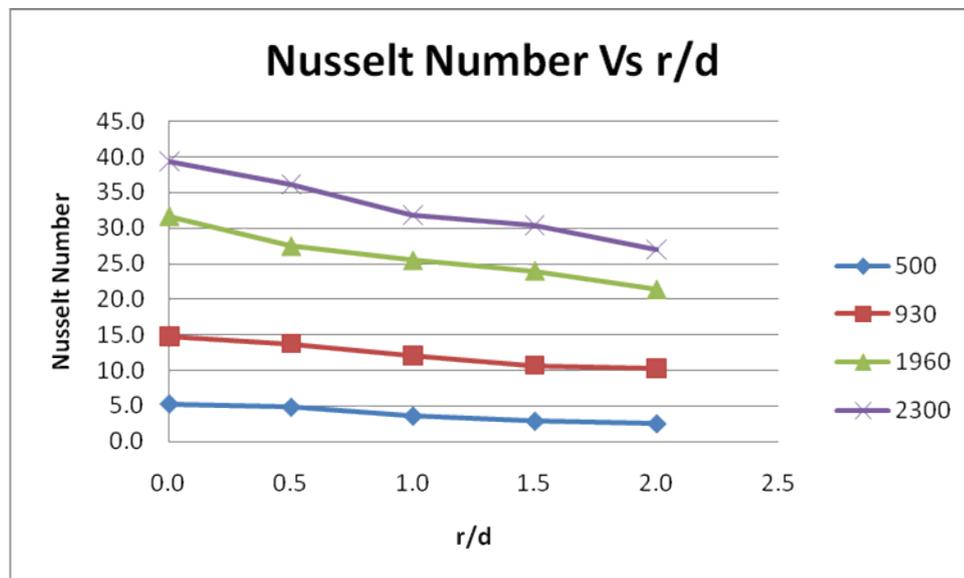


Figure 4.11:Graph Nusselt No. Vs r/d at angle 90° and $H/d=2$

Figure 4.11 present the Reynolds number distributions for $H/d = 2$ at angle 90° . Graph line for Reynolds number 2300 are higher than Reynolds number graph line at 500 which is the lowest. At Reynolds number 2300, air velocities are higher compare to the others. The heat transfers are increase as Reynolds number increase. The air velocities seem give more influence in cooling system compare to the change of nozzle angle factor because the increasing of the velocity flow effect the types of flow regime occur at heat source plate whether the air flows were laminar or turbulent in cooling system. At low Reynolds Number the jet impingement cooling system are not effective as Figure 4.11 show that Nusselt number are closer to the zero value. At zero Nusselt number, there are no heat transfer occur in the system during the experiment.

4.6 H/d DISTRIBUTION

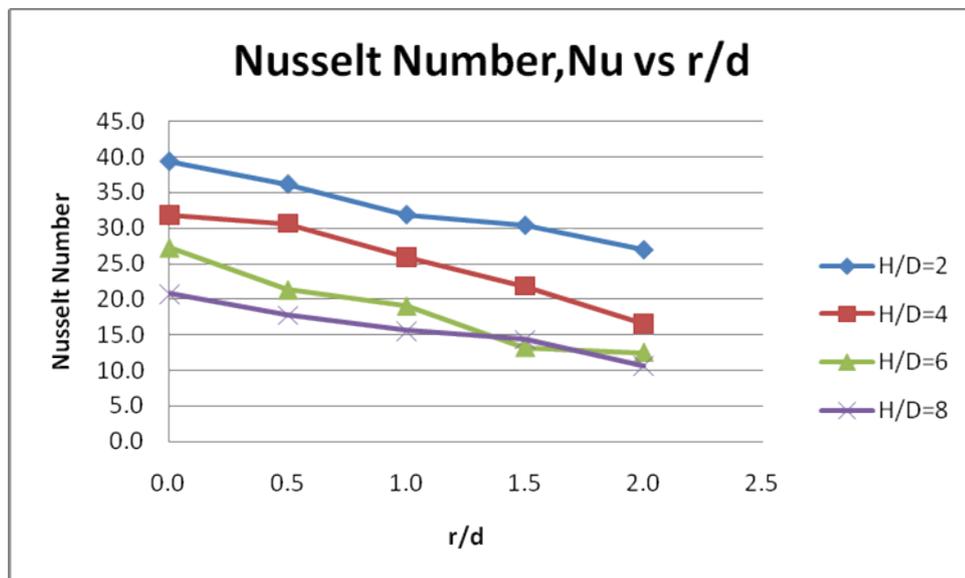


Figure 4.12:Graph Nusselt No. Vs r/d at Reynolds No 2300,Angle 90°

Figure 4.12 shown the dimensionless jet to heat source spacing, H/d distribution for Reynolds Number 2300 at 90°. Graph line for H/d= 2 give more higher value of Nusselt number compare to others. As the diameter of the nozzle are constant, the different height of the nozzle to the impingement surface are effecting the heat transfer. From the graph line shown at the Figure 4.12, the heat transfer increase as the H/d are decrease. As the height of the nozzle more closer to the impinge surface, the impinge surface more expose to the air flow and when the spacing of jet to the heat source increasing, the cooling system are not effective.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

For this project the effect of the nozzle angle for jet impingement cooling system are determine after experimental and data analyzed. From the graph and discussion it can conclude that heat transfer are more effective when the nozzle of the angle closer to the normal line (horizontal line at 90°) and the effective for cooling system is nozzle angle at 90° due to the highest Nusselt Number at that range.

In this project, other properties are determined that affected the heat transfer in cooling system. Higher Reynolds Number will give more efficient in heat transfer compare to the low Reynolds number. As Reynolds number increase (in the range of laminar flow), Nusselt number are proportionally increasing. Furthermore, decreasing of the distance between nozzle edge and impingement surface will increase the heat transfer. This mean at lower H/d , the heat transfer of impingement surface will be higher. At the surface of impingement plate, there were determined that heat transfer are more higher at stagnation point as the nozzle edge pointing and focusing the impingement at that point compare to the others.

5.2 RECOMMENDATIONS

At some point in this project, there are some recommendations in order to overcome the constraints during the experiment. This recommendation can be used for the future in order to improve this project more successful and achieve more quality finding.

For this study, it is important that heater can heat the plate constantly at 100°C temperature. So, the isothermal heater is more equivalent for this project which can give more efficient result. This heater can reduce the error or uncertainty of the result which the plate temperature after impingement are originally affected during the cooling process.

Besides that, this study also needs to improve the method for varying the angle to make the project experiment easier. This also can vary more parameters of the study which more angles can be performed in this project.

In this study also needs to improve the method to measure the temperature at impingement surface. Wire thermocouples are more accurate compared to the laser thermometer since laser thermocouples occur more errors due to the pointing skills during the temperature for every point.

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This project is prepared based on the following references;

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APPENDIX A

Table A-15

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TABLE A-15 Properties of air at 1 atm pressure							
Temp. $T, ^\circ\text{C}$	Density $\rho, \text{kg/m}^3$	Specific Heat $c_p, \text{J/kg} \cdot \text{K}$	Thermal Conductivity $k, \text{W/m} \cdot \text{K}$	Thermal Diffusivity $\alpha, \text{m}^2/\text{s}$	Dynamic Viscosity $\mu, \text{kg/m} \cdot \text{s}$	Kinematic Viscosity $\nu, \text{m}^2/\text{s}$	Prandtl Number Pr
-150	2.866	983	0.01171	4.158×10^{-5}	8.636×10^{-6}	3.013×10^{-5}	0.7246
-100	2.038	966	0.01582	8.036×10^{-5}	1.189×10^{-5}	5.837×10^{-5}	0.7263
-50	1.582	999	0.01979	1.252×10^{-4}	1.474×10^{-5}	9.319×10^{-5}	0.7440
-40	1.514	1002	0.02057	1.356×10^{-4}	1.527×10^{-5}	1.008×10^{-4}	0.7436
-30	1.451	1004	0.02134	1.465×10^{-4}	1.579×10^{-5}	1.087×10^{-4}	0.7425
-20	1.394	1005	0.02211	1.578×10^{-4}	1.630×10^{-5}	1.169×10^{-4}	0.7408
-10	1.341	1006	0.02288	1.696×10^{-4}	1.680×10^{-5}	1.252×10^{-4}	0.7387
0	1.292	1006	0.02364	1.818×10^{-4}	1.729×10^{-5}	1.338×10^{-4}	0.7362
5	1.269	1006	0.02401	1.880×10^{-4}	1.754×10^{-5}	1.382×10^{-4}	0.7350
10	1.246	1006	0.02439	1.944×10^{-4}	1.778×10^{-5}	1.426×10^{-4}	0.7336
15	1.225	1007	0.02476	2.009×10^{-4}	1.802×10^{-5}	1.470×10^{-4}	0.7323
20	1.204	1007	0.02514	2.074×10^{-4}	1.825×10^{-5}	1.516×10^{-4}	0.7309
25	1.184	1007	0.02551	2.141×10^{-4}	1.849×10^{-5}	1.562×10^{-4}	0.7296
30	1.164	1007	0.02588	2.208×10^{-4}	1.872×10^{-5}	1.608×10^{-4}	0.7282
35	1.145	1007	0.02625	2.277×10^{-4}	1.895×10^{-5}	1.655×10^{-4}	0.7268
40	1.127	1007	0.02662	2.346×10^{-4}	1.918×10^{-5}	1.702×10^{-4}	0.7255
45	1.109	1007	0.02699	2.416×10^{-4}	1.941×10^{-5}	1.750×10^{-4}	0.7241
50	1.092	1007	0.02735	2.487×10^{-4}	1.963×10^{-5}	1.798×10^{-4}	0.7228
60	1.059	1007	0.02808	2.632×10^{-4}	2.008×10^{-5}	1.896×10^{-4}	0.7202
70	1.028	1007	0.02881	2.780×10^{-4}	2.052×10^{-5}	1.995×10^{-4}	0.7177
80	0.9994	1008	0.02953	2.931×10^{-4}	2.096×10^{-5}	2.097×10^{-4}	0.7154
90	0.9718	1008	0.03024	3.086×10^{-4}	2.139×10^{-5}	2.201×10^{-4}	0.7132
100	0.9458	1009	0.03095	3.243×10^{-4}	2.181×10^{-5}	2.306×10^{-4}	0.7111
120	0.8977	1011	0.03235	3.565×10^{-4}	2.264×10^{-5}	2.522×10^{-4}	0.7073
140	0.8542	1013	0.03374	3.898×10^{-4}	2.345×10^{-5}	2.745×10^{-4}	0.7041
160	0.8148	1016	0.03511	4.241×10^{-4}	2.420×10^{-5}	2.975×10^{-4}	0.7014
180	0.7788	1019	0.03646	4.593×10^{-4}	2.504×10^{-5}	3.212×10^{-4}	0.6992
200	0.7459	1023	0.03779	4.954×10^{-4}	2.577×10^{-5}	3.455×10^{-4}	0.6974
250	0.6746	1033	0.04104	5.890×10^{-4}	2.760×10^{-5}	4.091×10^{-4}	0.6946
300	0.6158	1044	0.04418	6.871×10^{-4}	2.934×10^{-5}	4.765×10^{-4}	0.6935
350	0.5664	1056	0.04721	7.892×10^{-4}	3.101×10^{-5}	5.475×10^{-4}	0.6937
400	0.5243	1069	0.05015	8.951×10^{-4}	3.261×10^{-5}	6.219×10^{-4}	0.6948
450	0.4880	1081	0.05298	1.004×10^{-3}	3.415×10^{-5}	6.997×10^{-4}	0.6965
500	0.4565	1093	0.05572	1.117×10^{-3}	3.563×10^{-5}	7.806×10^{-4}	0.6986
600	0.4042	1115	0.06093	1.352×10^{-3}	3.846×10^{-5}	9.515×10^{-4}	0.7037
700	0.3627	1135	0.06581	1.598×10^{-3}	4.111×10^{-5}	1.133×10^{-3}	0.7092
800	0.3289	1153	0.07037	1.855×10^{-3}	4.362×10^{-5}	1.326×10^{-3}	0.7149
900	0.3008	1169	0.07465	2.122×10^{-3}	4.600×10^{-5}	1.529×10^{-3}	0.7206
1000	0.2772	1184	0.07868	2.398×10^{-3}	4.826×10^{-5}	1.741×10^{-3}	0.7260
1500	0.1990	1234	0.09599	3.908×10^{-3}	5.817×10^{-5}	2.922×10^{-3}	0.7478
2000	0.1553	1264	0.11113	5.664×10^{-3}	6.630×10^{-5}	4.270×10^{-3}	0.7539

Note: For ideal gases, the properties c_p , k , μ , and Pr are independent of pressure. The properties ρ , ν , and α at a pressure P (in atm) other than 1 atm are determined by multiplying the values of ρ at the given temperature by P and by dividing ν and α by P .

Source: Data generated from the EES software developed by S. A. Klein and F. L. Alvarado. Original sources: Keenan, Chao, Keyes, Gas Tables, Wiley, 198; and Thermophysical Properties of Matter, Vol. 3: Thermal Conductivity, Y. S. Touloukian, P. E. Liley, S. C. Saxena, Vol. 11: Viscosity, Y. S. Touloukian, S. C. Saxena, and P. Hestermanns, IFI/Plenum, NY, 1970, ISBN 0-306067020-8.

Figure:Table A-15

APPENDIX B

Data From Experiment

Table For Reynolds No= 2300 and H/D=2

Nozzle Angle	r/d	T _{plate Impinge surface} (°C)	Heat Transfer Coefficient, h (W/m ² ·K)	Nusselt Number, Nu
θ = 30°	0.0	89.2	19.798	24.776
	0.5	89.7	18.720	23.427
	1.0	89.8	18.507	23.160
	1.5	90.1	17.871	22.365
	2.0	90.3	17.451	21.839
θ = 45°	0.0	87.3	24.069	30.121
	0.5	88.9	20.454	25.597
	1.0	89.4	19.365	24.234
	1.5	88.5	21.339	26.704
	2.0	88.9	20.454	25.597
θ = 60°	0.0	85.4	28.639	35.840
	0.5	86	27.162	33.991
	1.0	86.7	25.478	31.885
	1.5	87.7	23.146	28.966
	2.0	88.9	20.454	25.597
θ = 90°	0.0	84.3	31.434	39.338
	0.5	85.3	28.888	36.152
	1.0	86.7	25.478	31.885
	1.5	87.2	24.302	30.412
	2.0	88.4	21.562	26.983

Table For Reynolds No= 2300 and H/D=4

Nozzle Angle	r/d	T _{plate Impinge surface} (°C)	Heat Transfer Coefficient, h (W/m ² ·K)	Nusselt Number, Nu
θ = 30°	0.0	90.2	17.661	22.101
	0.5	90.3	17.451	21.839
	1.0	90.8	16.413	20.540
	1.5	91.4	15.190	19.009
	2.0	91.9	14.189	17.756
θ = 45°	0.0	89.7	18.720	23.427
	0.5	90.9	16.207	20.282
	1.0	91.8	14.388	18.005
	1.5	91.3	15.392	19.262
	2.0	90.2	17.661	22.101
	0.0	87.8	22.917	28.680

$\theta = 60^\circ$	0.5	88.2	22.010	27.545
	1.0	89.3	19.581	24.505
	1.5	90.5	17.034	21.317
	2.0	91.4	15.190	19.009
$\theta = 90^\circ$	0.0	86.7	25.478	31.885
	0.5	87.1	24.535	30.705
	1.0	88.8	20.674	25.872
	1.5	90.3	17.451	21.839
	2.0	92.4	13.204	16.524

Table For Reynolds No= 2300 and H/D=6

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ (°C)	Heat Transfer Coefficient, h (W/m ² K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	93.3	11.472	14.356
	0.5	93.6	10.906	13.648
	1.0	93.9	10.345	12.946
	1.5	94.5	9.239	11.562
	2.0	95.7	7.089	8.871
$\theta = 45^\circ$	0.0	91.5	14.988	18.757
	0.5	92.9	12.236	15.312
	1.0	93.4	11.283	14.119
	1.5	92.8	12.428	15.553
	2.0	92.7	12.621	15.795
$\theta = 60^\circ$	0.0	89.7	18.720	23.427
	0.5	90.3	17.451	21.839
	1.0	92	13.990	17.508
	1.5	93.4	11.283	14.119
	2.0	94.3	9.605	12.020
$\theta = 90^\circ$	0.0	88.3	21.786	27.264
	0.5	90.5	17.034	21.317
	1.0	91.4	15.190	19.009
	1.5	93.8	10.531	13.179
	2.0	94.1	9.974	12.482

Table For Reynolds No= 2300 and H/D=8

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ (°C)	Heat Transfer Coefficient, h (W/m ² K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	94.3	9.605	12.020
	0.5	94.8	8.694	10.880
	1.0	95.0	8.333	10.429
	1.5	96.0	6.564	8.214
	2.0	97.3	4.343	5.436

$\theta = 45^\circ$	0.0	92.9	12.236	15.312
	0.5	93.5	11.094	13.883
	1.0	95.1	8.154	10.204
	1.5	94.0	10.159	12.713
	2.0	93.0	12.044	15.072
$\theta = 60^\circ$	0.0	92.7	12.621	15.795
	0.5	93.9	10.345	12.946
	1.0	94.8	8.694	10.880
	1.5	95.8	6.913	8.652
	2.0	96.6	5.528	6.918
$\theta = 90^\circ$	0.0	90.7	16.619	20.798
	0.5	91.9	14.189	17.756
	1.0	92.8	12.428	15.553
	1.5	93.3	11.472	14.356
	2.0	94.9	8.513	10.654

Table For Reynolds No= 1960 and H/D=2

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ (°C)	Heat Transfer Coefficient, h (W/m ² K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	91.9	12.122	15.169
	0.5	92.3	11.448	14.326
	1.0	92.9	10.453	13.081
	1.5	94.3	8.206	10.269
	2.0	93.7	9.157	11.459
$\theta = 45^\circ$	0.0	88.9	17.474	21.868
	0.5	89.7	15.993	20.014
	1.0	90.7	14.198	17.768
	1.5	88.8	17.662	22.103
	2.0	89.9	15.629	19.559
$\theta = 60^\circ$	0.0	86.4	22.378	28.005
	0.5	87.7	19.774	24.746
	1.0	88.9	17.474	21.868
	1.5	89.9	15.629	19.559
	2.0	90.2	15.088	18.882
$\theta = 90^\circ$	0.0	85.0	25.324	31.691
	0.5	86.6	21.970	27.494
	1.0	87.4	20.364	25.485
	1.5	88.0	19.190	24.015
	2.0	89.1	17.100	21.400

Table For Reynolds No= 1960 and H/D=4

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ (°C)	Heat Transfer Coefficient, h (W/m ² K)	Nusselt Number, Nu
	0.0	91.1	13.497	16.890
	0.5	91.8	12.291	15.382

$\theta = 30^\circ$	1.0	92.2	11.615	14.536
	1.5	93.1	10.126	12.672
	2.0	93.8	8.997	11.259
$\theta = 45^\circ$	0.0	89.9	15.629	19.559
	0.5	91.8	12.291	15.382
	1.0	92.0	11.952	14.958
	1.5	92.5	11.114	13.908
	2.0	91.3	13.149	16.456
$\theta = 60^\circ$	0.0	88.9	17.474	21.868
	0.5	90.6	14.375	17.989
	1.0	91.4	12.977	16.240
	1.5	92.6	10.948	13.701
	2.0	93.6	9.317	11.659
$\theta = 90^\circ$	0.0	87.4	20.364	25.485
	0.5	87.9	19.384	24.258
	1.0	89.3	16.728	20.935
	1.5	91.2	13.323	16.673
	2.0	92.9	10.453	13.081

Table For Reynolds No= 1960 and H/D=6

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ (°C)	Heat Transfer Coefficient, h (W/m ² ·K)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	93.9	8.838	11.060
	0.5	93.5	9.478	11.861
	1.0	94.2	8.363	10.466
	1.5	94.9	7.273	9.102
	2.0	95.8	5.906	7.391
$\theta = 45^\circ$	0.0	92.5	11.114	13.908
	0.5	93.5	9.478	11.861
	1.0	94.3	8.206	10.269
	1.5	94.0	8.679	10.861
	2.0	93.5	9.478	11.861
$\theta = 60^\circ$	0.0	90.2	15.088	18.882
	0.5	92.0	11.952	14.958
	1.0	93.7	9.157	11.459
	1.5	94.8	7.427	9.295
	2.0	95.6	6.207	7.767
$\theta = 90^\circ$	0.0	90.1	15.268	19.107
	0.5	91.1	13.497	16.890
	1.0	92.7	10.782	13.494
	1.5	93.9	8.838	11.060
	2.0	95.3	6.661	8.336

Table For Reynolds No= 1960 and H/D=8

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ (°C)	Heat Transfer Coefficient, h (W/m ² ·K)	Nusselt Number, Nu
	0.0	95.2	6.813	8.526
	0.5	95.2	6.813	8.526

$\theta = 30^\circ$	1.0	95.3	6.661	8.336
	1.5	96.0	5.608	7.017
	2.0	97.6	3.284	4.109
$\theta = 45^\circ$	0.0	94.3	8.206	10.269
	0.5	95.9	5.757	7.204
	1.0	96.9	4.286	5.364
	1.5	95.8	5.906	7.391
	2.0	95.3	6.661	8.336
$\theta = 60^\circ$	0.0	93.6	9.317	11.659
	0.5	94.7	7.582	9.488
	1.0	95.5	6.357	7.956
	1.5	95.8	5.906	7.391
	2.0	96.6	4.723	5.910
$\theta = 90^\circ$	0.0	92.5	11.114	13.908
	0.5	93.0	10.289	12.876
	1.0	94.6	7.737	9.683
	1.5	95.7	6.056	7.579
	2.0	96.8	4.431	5.546

Table For Reynolds No= 930 and H/D=2

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	92.4	5.467	6.841
	0.5	92.6	5.305	6.639
	1.0	93.5	4.593	5.748
	1.5	93.7	4.437	5.553
	2.0	93.4	4.671	5.846
$\theta = 45^\circ$	0.0	88.3	9.019	11.287
	0.5	87.0	10.255	12.833
	1.0	90.7	6.880	8.610
	1.5	88.6	8.742	10.941
	2.0	89.8	7.662	9.589
$\theta = 60^\circ$	0.0	86.9	10.352	12.955
	0.5	88.2	9.112	11.404
	1.0	88.4	8.927	11.171
	1.5	89.2	8.197	10.258
	2.0	90.0	7.486	9.369
$\theta = 90^\circ$	0.0	85.4	11.857	14.838
	0.5	86.2	11.044	13.821
	1.0	87.6	9.678	12.111
	1.5	88.8	8.559	10.711
	2.0	89.2	8.197	10.258

Table For Reynolds No= 930 and H/D=4

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
	0.0	92.8	5.145	6.439
	0.5	92.3	5.548	6.943

$\theta = 30^\circ$	1.0	93.3	4.749	5.944
	1.5	91.6	6.122	7.661
	2.0	93.0	4.986	6.240
$\theta = 45^\circ$	0.0	89.9	7.574	9.478
	0.5	91.1	6.541	8.185
	1.0	92.4	5.467	6.841
	1.5	92.3	5.548	6.943
	2.0	91.0	6.625	8.291
$\theta = 60^\circ$	0.0	88.7	8.651	10.826
	0.5	89.7	7.750	9.699
	1.0	91.4	6.289	7.870
	1.5	93.7	4.437	5.553
	2.0	93.2	4.828	6.042
$\theta = 90^\circ$	0.0	87.0	10.255	12.833
	0.5	88.2	9.112	11.404
	1.0	89.8	7.662	9.589
	1.5	91.0	6.625	8.291
	2.0	92.9	5.066	6.339

Table For Reynolds No= 930 and H/D=6

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	93.9	4.283	5.360
	0.5	93.8	4.360	5.456
	1.0	94.0	4.206	5.263
	1.5	94.7	3.674	4.598
	2.0	95.6	3.008	3.764
$\theta = 45^\circ$	0.0	92.9	5.066	6.339
	0.5	91.5	6.205	7.765
	1.0	94.0	4.206	5.263
	1.5	93.7	4.437	5.553
	2.0	93.0	4.986	6.240
$\theta = 60^\circ$	0.0	90.2	7.312	9.150
	0.5	92.3	5.548	6.943
	1.0	93.5	4.593	5.748
	1.5	94.4	3.901	4.881
	2.0	95.8	2.862	3.582
$\theta = 90^\circ$	0.0	90.9	6.710	8.397
	0.5	91.6	6.122	7.661
	1.0	92.4	5.467	6.841
	1.5	94.6	3.749	4.692
	2.0	95.9	2.790	3.491

Table For Reynolds No= 930 and H/D=8

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
	0.0	93.4	4.671	5.846
	0.5	92.6	5.305	6.639

$\theta = 30^\circ$	1.0	89.9	7.574	9.478
	1.5	96.4	2.431	3.042
	2.0	98.0	1.318	1.649
$\theta = 45^\circ$	0.0	93.2	4.828	6.042
	0.5	95.8	2.862	3.582
	1.0	97.0	2.007	2.512
	1.5	96.8	2.147	2.687
	2.0	94.9	3.525	4.411
$\theta = 60^\circ$	0.0	91.0	6.625	8.291
	0.5	93.7	4.437	5.553
	1.0	94.9	3.525	4.411
	1.5	96.8	2.147	2.687
	2.0	90.1	7.399	9.259
$\theta = 90^\circ$	0.0	90.1	7.399	9.259
	0.5	92.1	5.710	7.146
	1.0	92.9	5.066	6.339
	1.5	94.4	3.901	4.881
	2.0	94.3	3.977	4.976

Table For Reynolds No= 500 and H/D=2

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	95.8	1.500	1.877
	0.5	96.0	1.424	1.782
	1.0	95.7	1.538	1.924
	1.5	96.3	1.311	1.640
	2.0	96.9	1.088	1.362
$\theta = 45^\circ$	0.0	90.1	3.876	4.851
	0.5	91.1	3.427	4.288
	1.0	92.3	2.906	3.637
	1.5	91.4	3.295	4.123
	2.0	91.5	3.251	4.068
$\theta = 60^\circ$	0.0	89.3	4.247	5.315
	0.5	89.8	4.014	5.024
	1.0	91.2	3.383	4.233
	1.5	94.3	2.083	2.607
	2.0	93.8	2.284	2.859
$\theta = 90^\circ$	0.0	89.3	4.247	5.315
	0.5	90.1	3.876	4.851
	1.0	92.4	2.864	3.584
	1.5	93.8	2.284	2.859
	2.0	94.5	2.004	2.508

Table For Reynolds No= 500 and H/D=4

Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
	0.0	95.4	1.653	2.068
	0.5	96.5	1.236	1.547

$\theta = 30^\circ$	1.0	96.9	1.088	1.362
	1.5	97.3	0.942	1.179
	2.0	97.8	0.762	0.953
$\theta = 45^\circ$	0.0	94.3	2.083	2.607
	0.5	95.6	1.576	1.972
	1.0	96.5	1.236	1.547
	1.5	94.3	2.083	2.607
	2.0	95.9	1.462	1.829
$\theta = 60^\circ$	0.0	92.7	2.738	3.426
	0.5	92.8	2.696	3.373
	1.0	94.3	2.083	2.607
	1.5	95.3	1.691	2.116
	2.0	96.7	1.162	1.454
$\theta = 90^\circ$	0.0	91.2	3.383	4.233
	0.5	92.8	2.696	3.373
	1.0	93.3	2.488	3.114
	1.5	94.6	1.964	2.458
	2.0	95.8	1.500	1.877

Table For Reynolds No= 500 and H/D=6

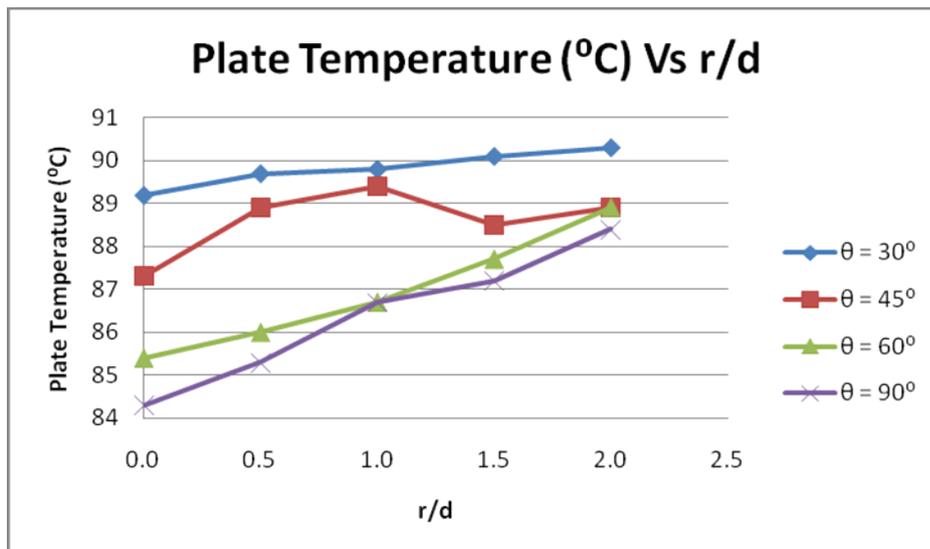
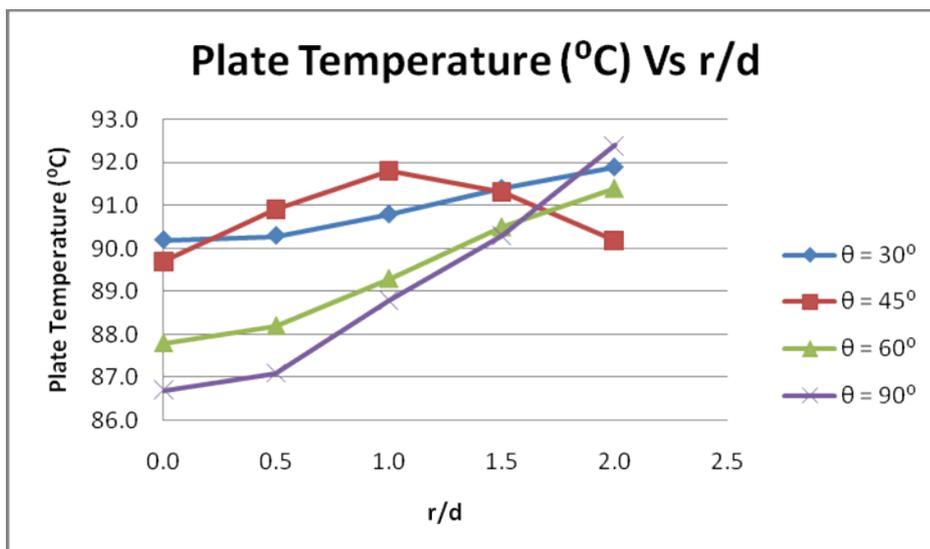
Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
$\theta = 30^\circ$	0.0	94.9	1.847	2.311
	0.5	95.0	1.807	2.262
	1.0	95.4	1.653	2.068
	1.5	95.9	1.462	1.829
	2.0	96.4	1.273	1.594
$\theta = 45^\circ$	0.0	94.3	2.083	2.607
	0.5	95.6	1.576	1.972
	1.0	96.5	1.236	1.547
	1.5	95.5	1.614	2.020
	2.0	95.2	1.730	2.165
$\theta = 60^\circ$	0.0	93.2	2.530	3.166
	0.5	94.3	2.083	2.607
	1.0	94.2	2.123	2.657
	1.5	95.4	1.653	2.068
	2.0	96.5	1.236	1.547
$\theta = 90^\circ$	0.0	93.0	2.612	3.269
	0.5	94.5	2.004	2.508
	1.0	95.8	1.500	1.877
	1.5	96.1	1.386	1.734
	2.0	96.3	1.311	1.640

Table For Reynolds No= 500 and H/D=8

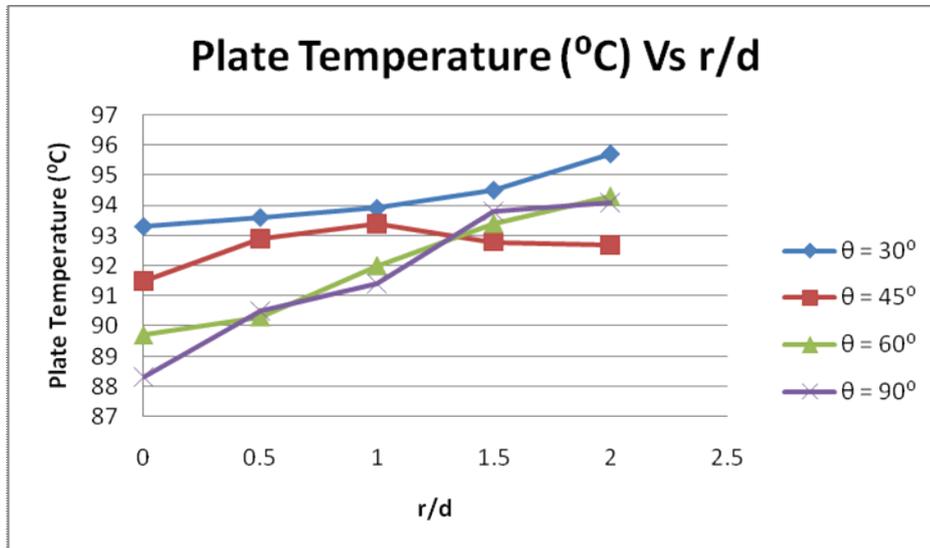
Nozzle Angle	r/d	$T_{\text{plate Impinge surface}}$ ($^\circ\text{C}$)	Heat Transfer Coefficient, h ($\text{W}/\text{m}^2\text{K}$)	Nusselt Number, Nu
	0.0	97.2	0.978	1.225
	0.5	97.6	0.834	1.043

$\theta = 30^\circ$	1.0	97.6	0.834	1.043
	1.5	98.9	0.375	0.469
	2.0	98.6	0.479	0.600
$\theta = 45^\circ$	0.0	95.3	1.691	2.116
	0.5	96.8	1.125	1.408
	1.0	97	1.052	1.316
	1.5	97.4	0.906	1.134
	2.0	97.9	0.726	0.909
$\theta = 60^\circ$	0.0	94.3	2.083	2.607
	0.5	94.8	1.886	2.360
	1.0	95.7	1.538	1.924
	1.5	96.5	1.236	1.547
	2.0	97.3	0.942	1.179
$\theta = 90^\circ$	0.0	93.8	2.284	2.859
	0.5	94.8	1.886	2.360
	1.0	95.6	1.576	1.972
	1.5	96	1.424	1.782
	2.0	96.4	1.273	1.594

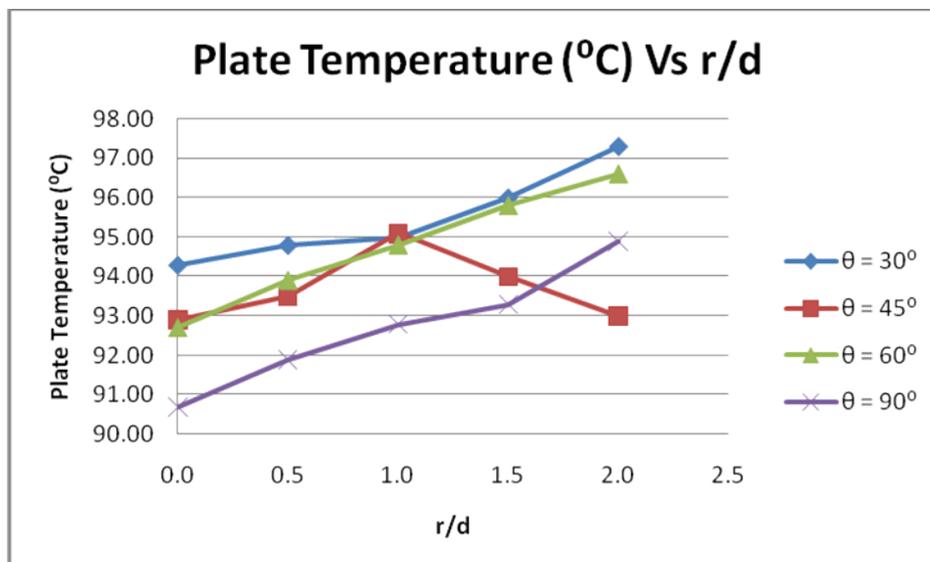
APPENDIX C

Graph Temperature ($^{\circ}\text{C}$) vs r/d Graph For Reynolds No= 2300 and $H/D=2$ Graph For Reynolds No= 2300 and $H/D=4$ 

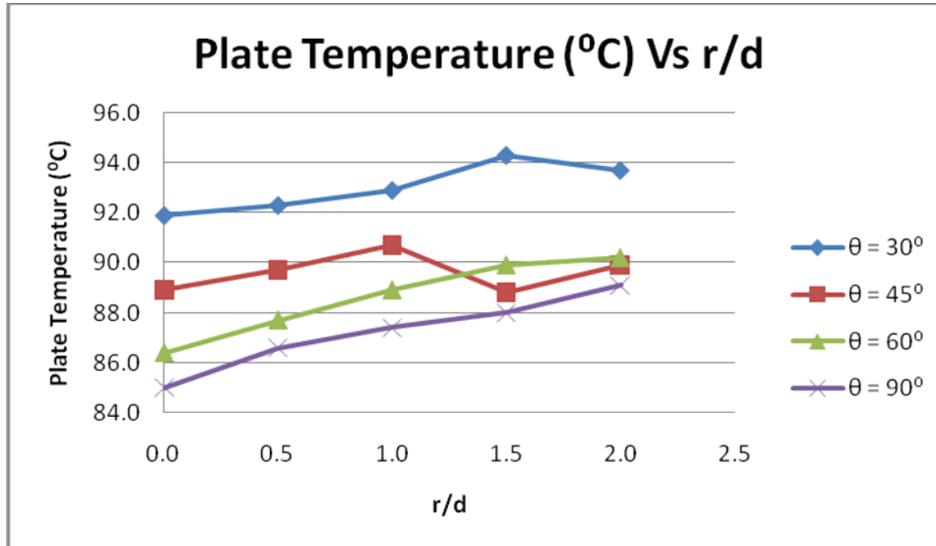
Graph For Reynolds No= 2300 and H/D=6



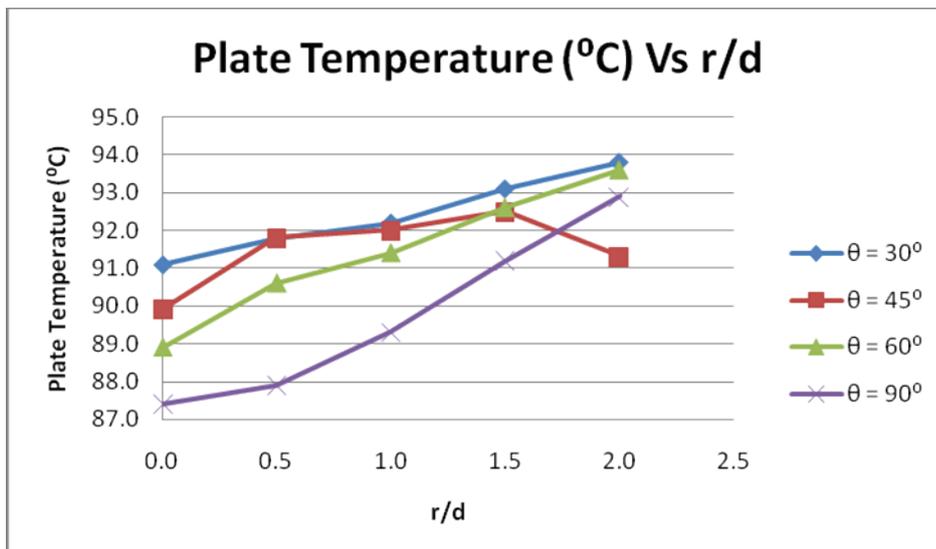
Graph For Reynolds No= 2300 and H/D=8



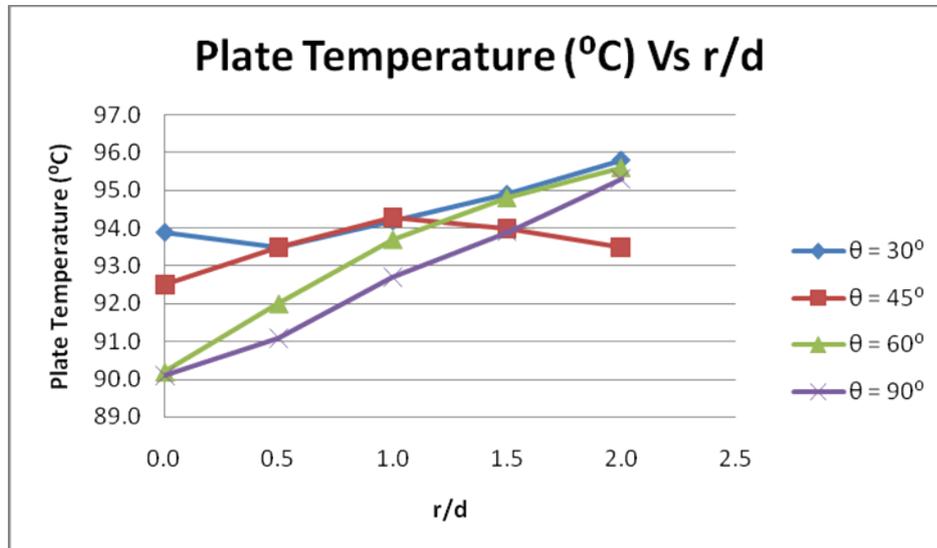
Graph For Reynolds No= 1960 and H/D=2



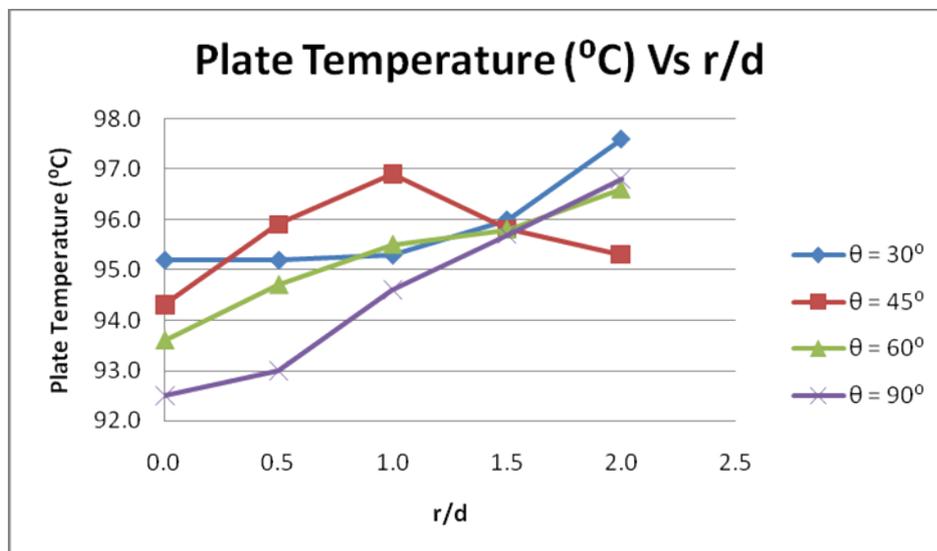
Graph For Reynolds No= 1960 and H/D=4



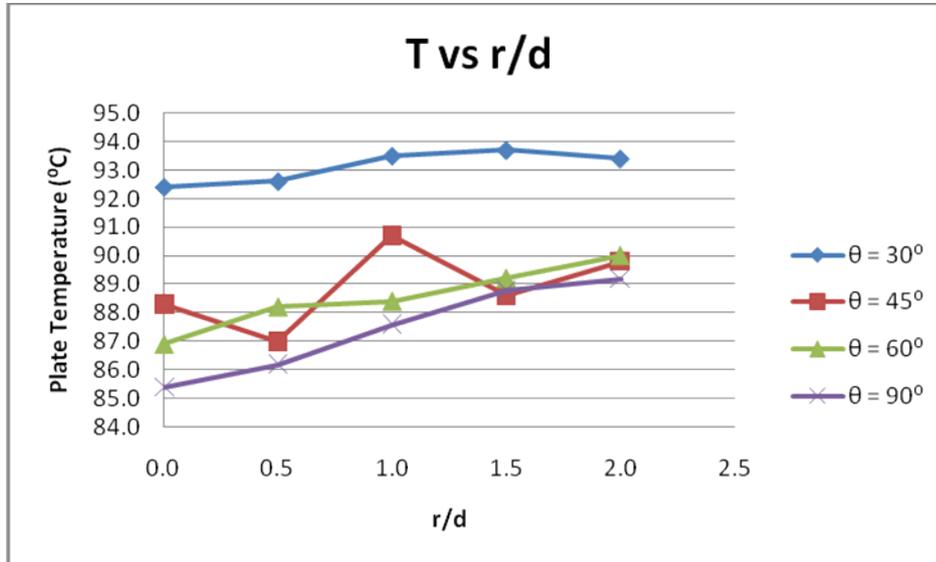
Graph For Reynolds No= 1960 and H/D=6



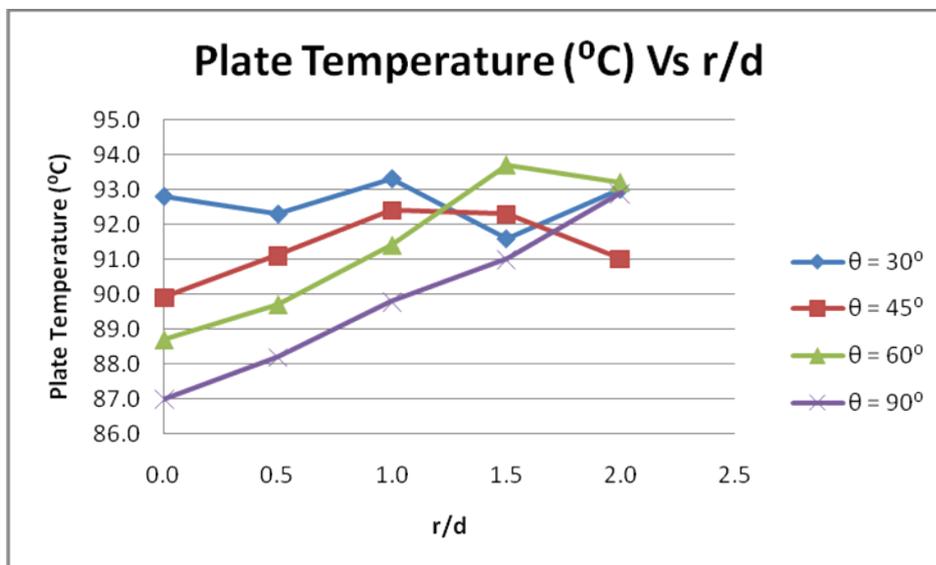
Graph For Reynolds No= 1960 and H/D=8



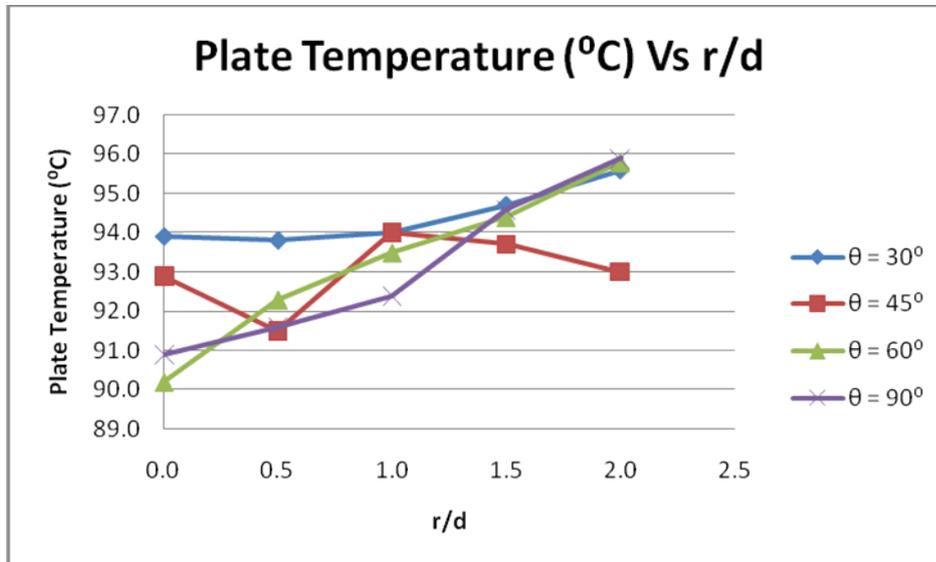
Graph For Reynolds No= 930 and H/D=2



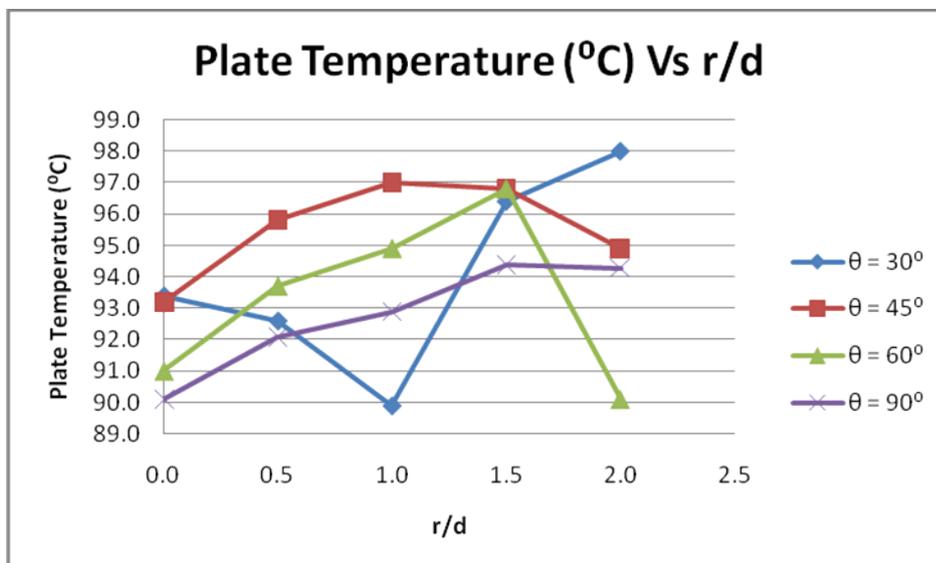
Graph For Reynolds No= 930 and H/D=4



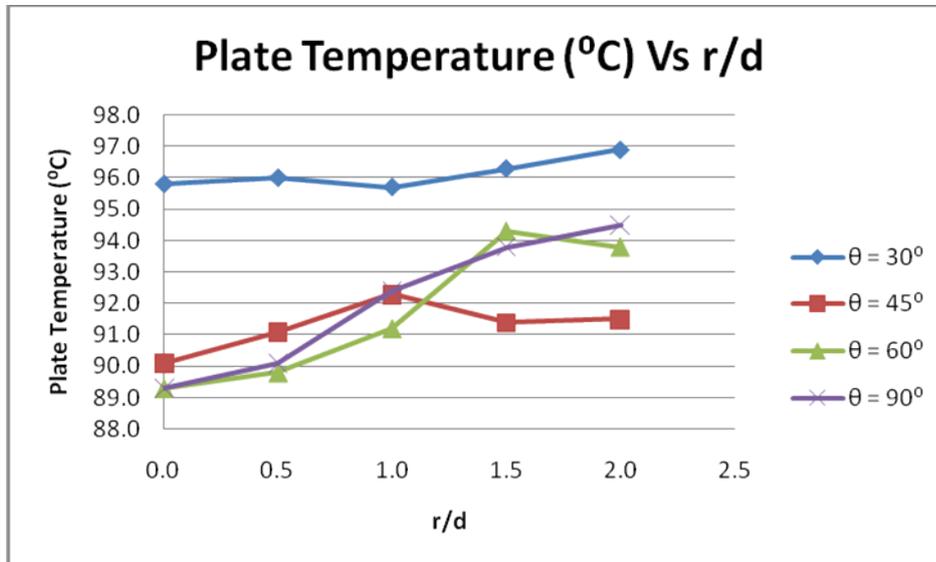
Graph For Reynolds No= 930 and H/D=6



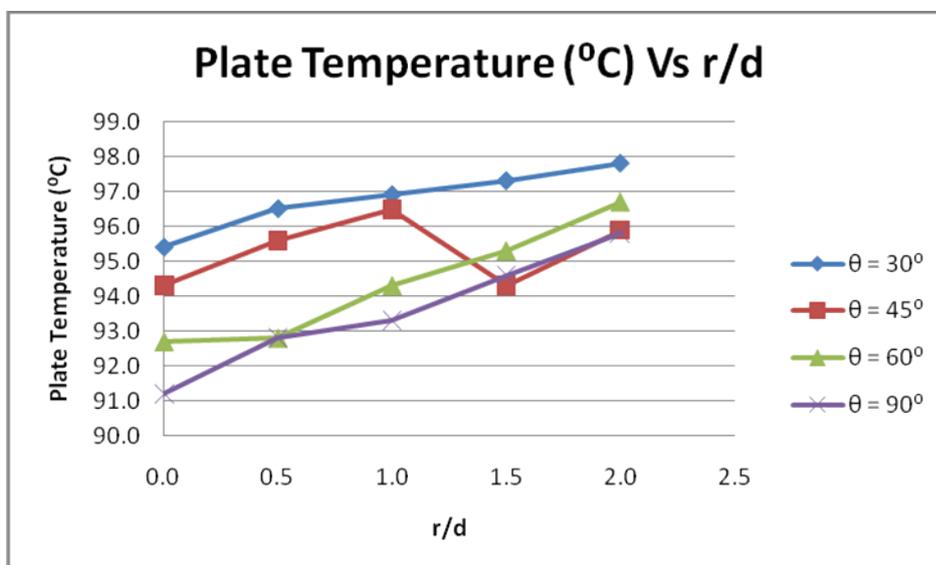
Graph For Reynolds No= 930 and H/D=8



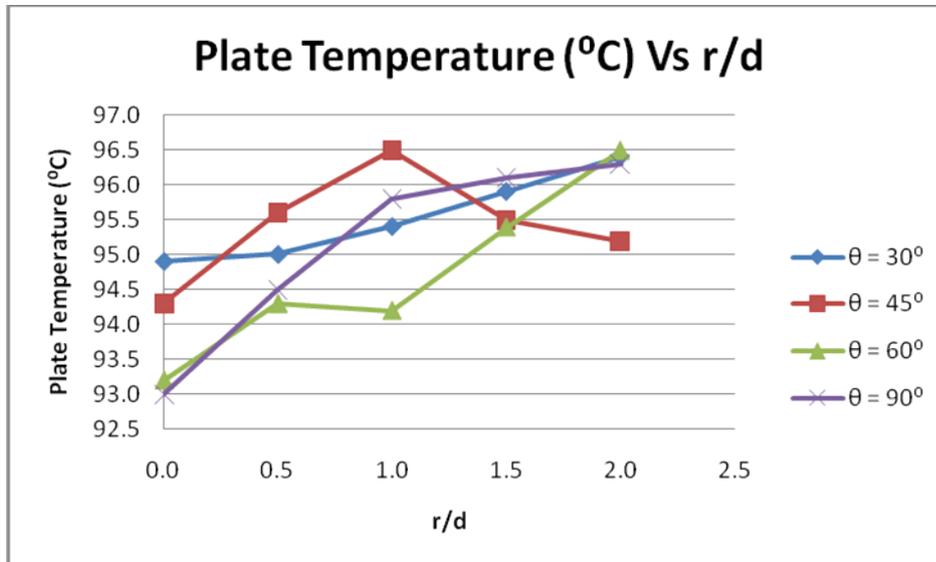
Graph For Reynolds No= 500 and H/D=2



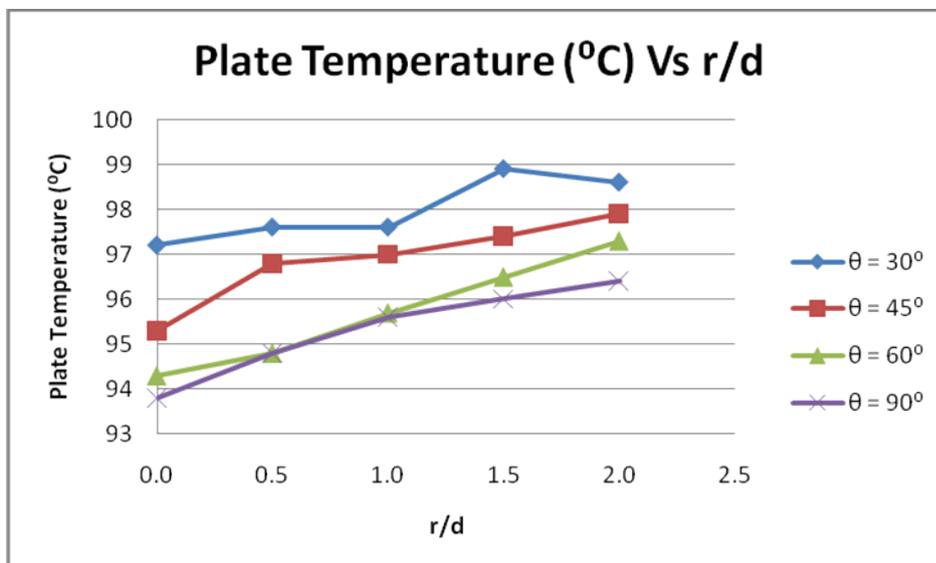
Graph For Reynolds No= 500 and H/D=4



Graph For Reynolds No= 500 and H/D=6



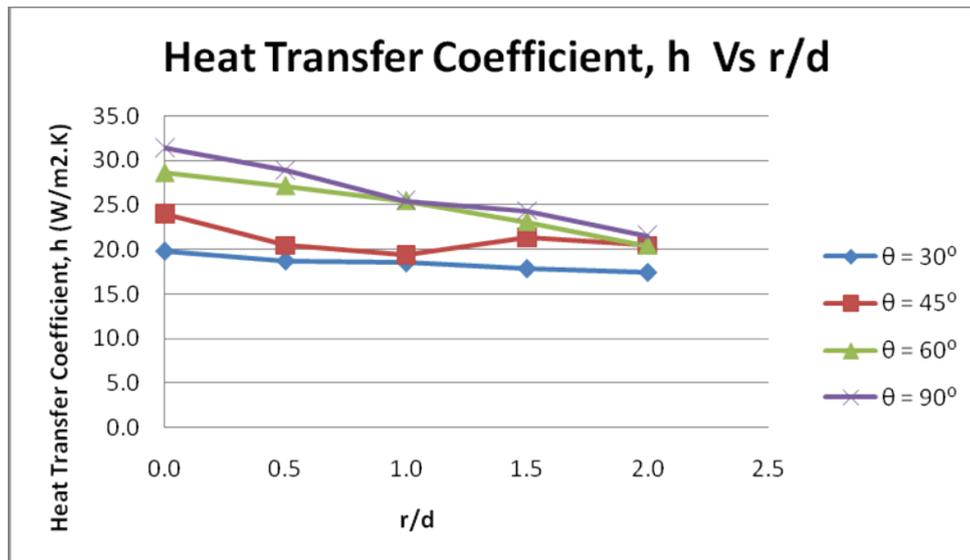
Graph For Reynolds No= 500 and H/D=8



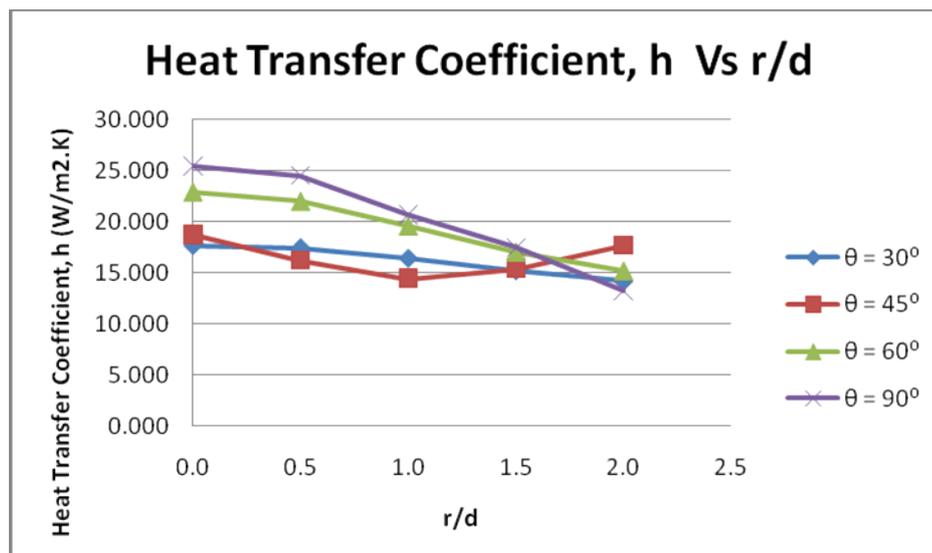
APPENDIX C

Graph heat transfer coefficient vs r/d

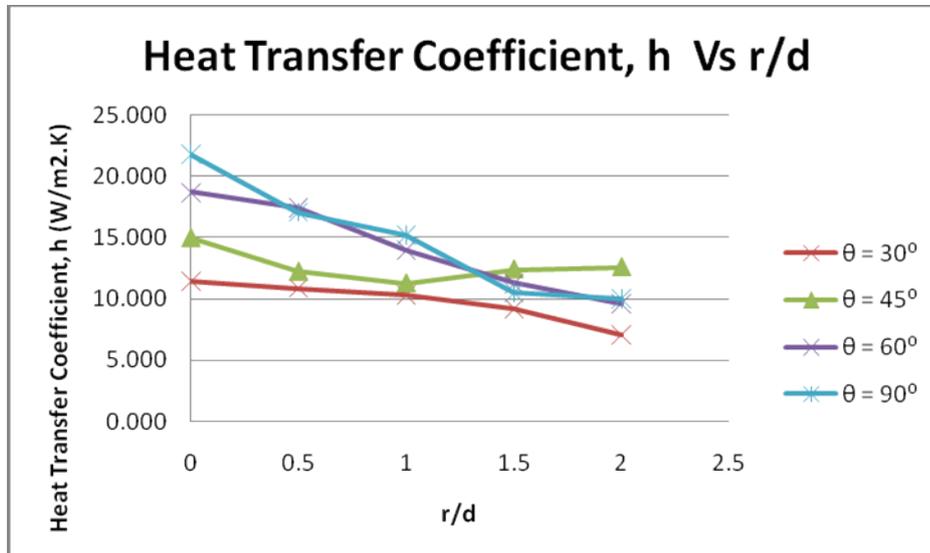
Graph For Reynolds No= 2300 and H/D=2



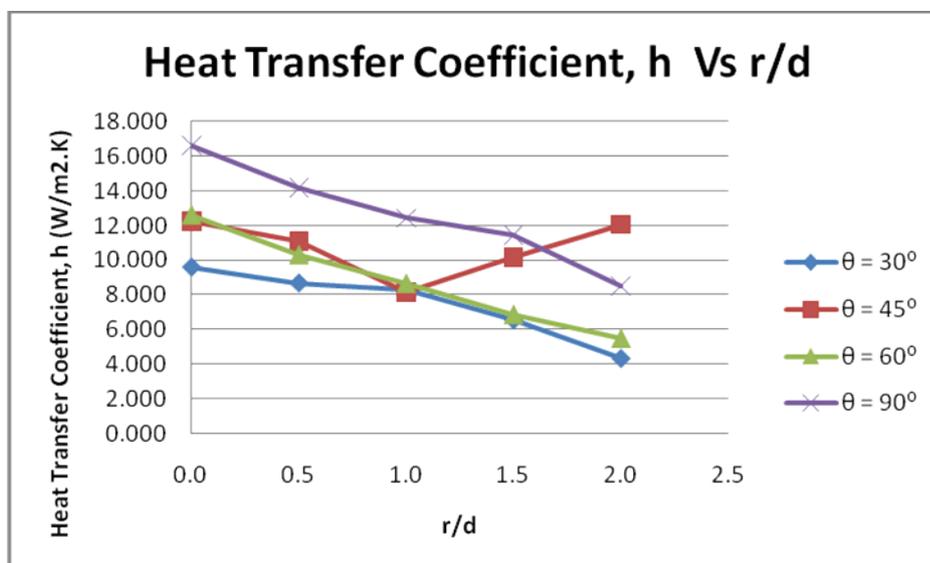
Graph For Reynolds No= 2300 and H/D=4



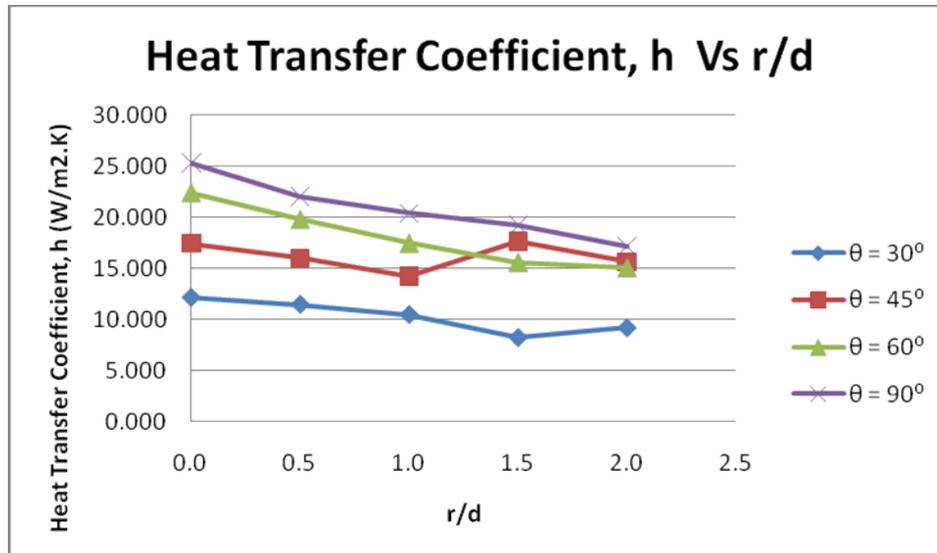
Graph For Reynolds No= 2300 and H/D=6



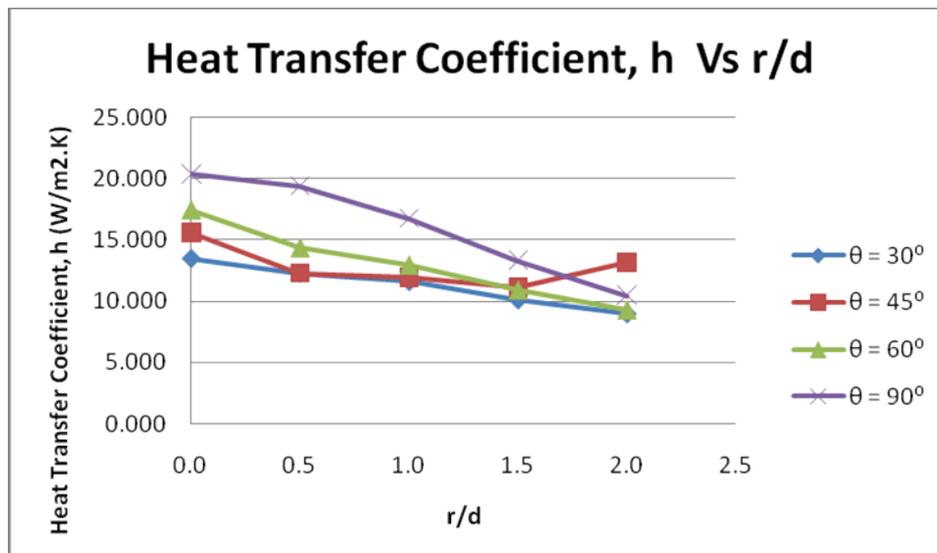
Graph For Reynolds No= 2300 and H/D=8



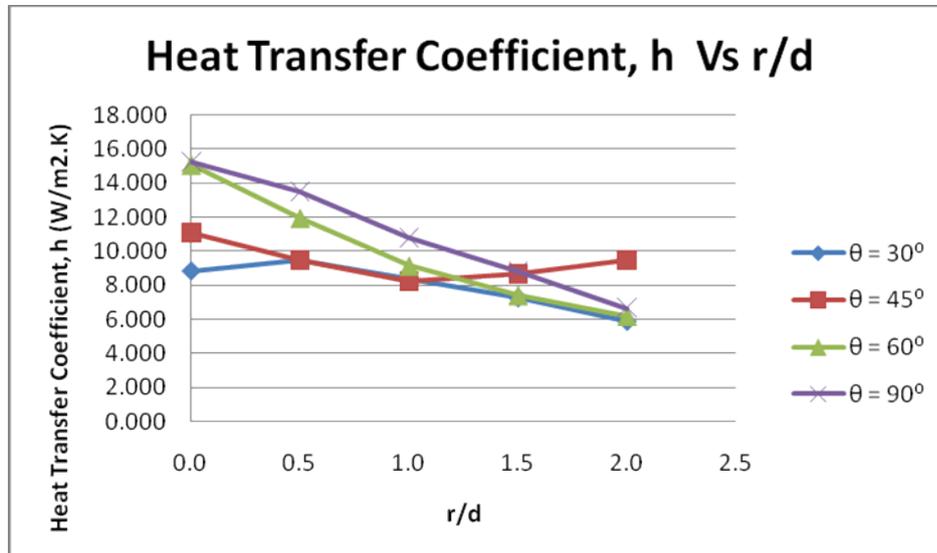
Graph For Reynolds No= 1960 and H/D=2



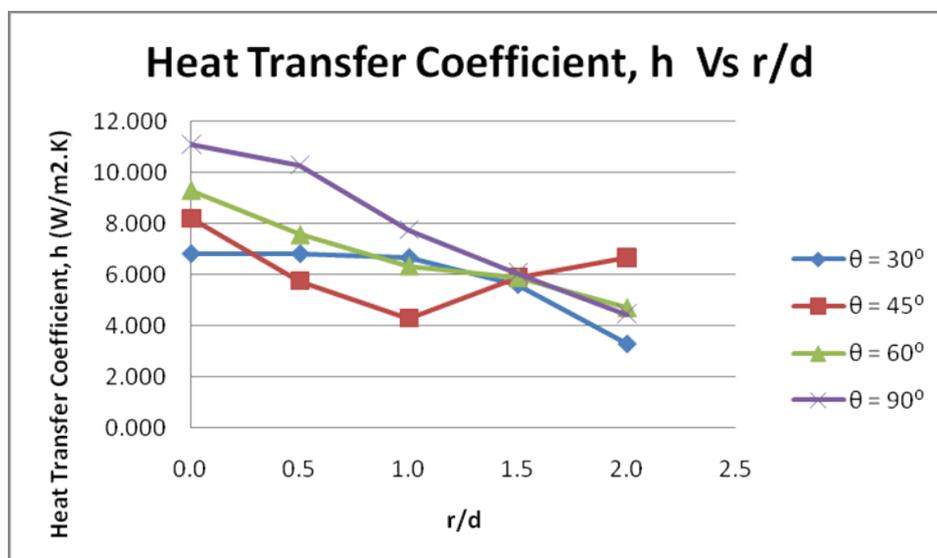
Graph For Reynolds No= 1960 and H/D=4



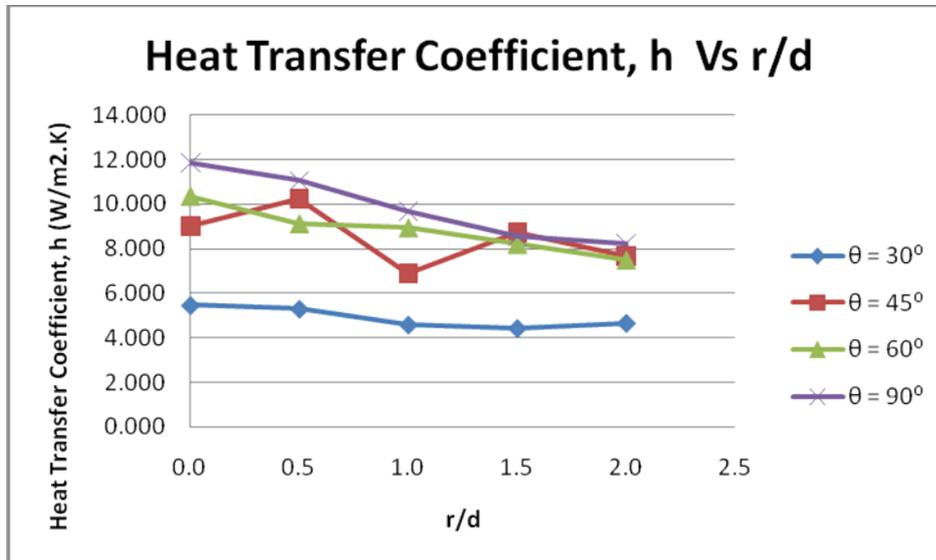
Graph For Reynolds No= 1960 and H/D=6



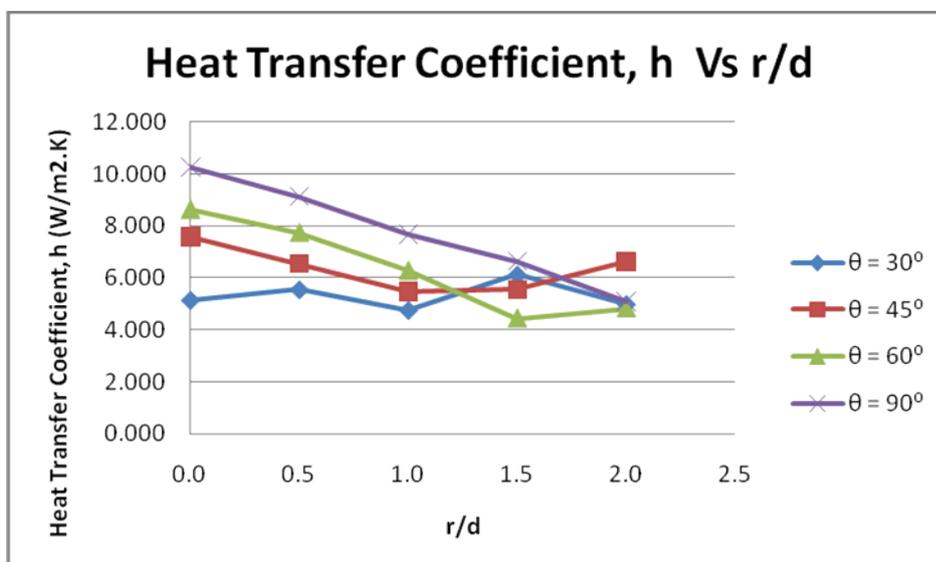
Graph For Reynolds No= 1960 and H/D=8



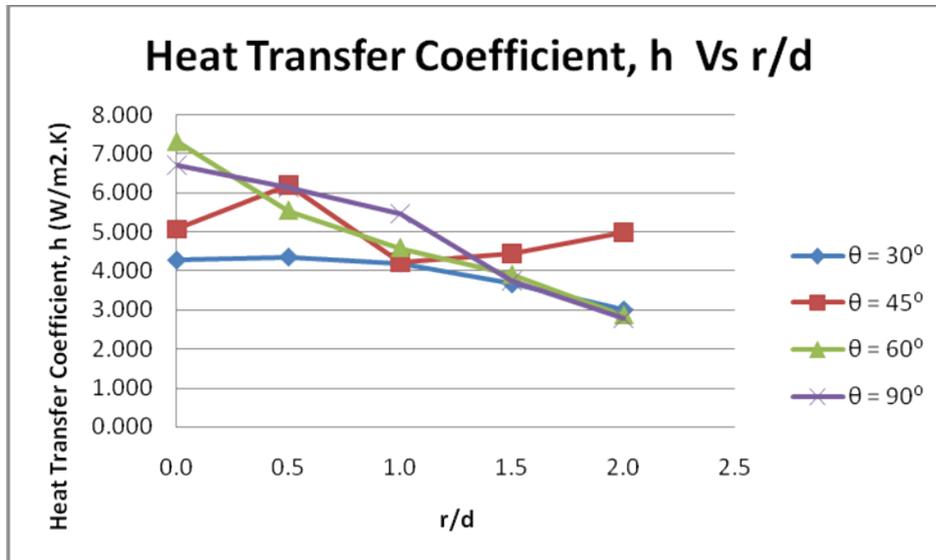
Graph For Reynolds No= 930 and H/D=2



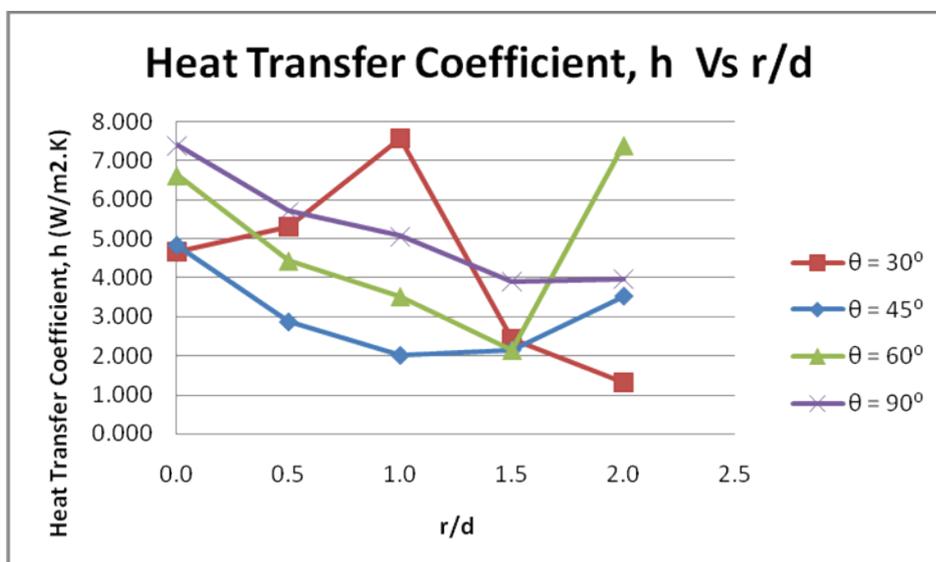
Graph For Reynolds No= 930 and H/D=4



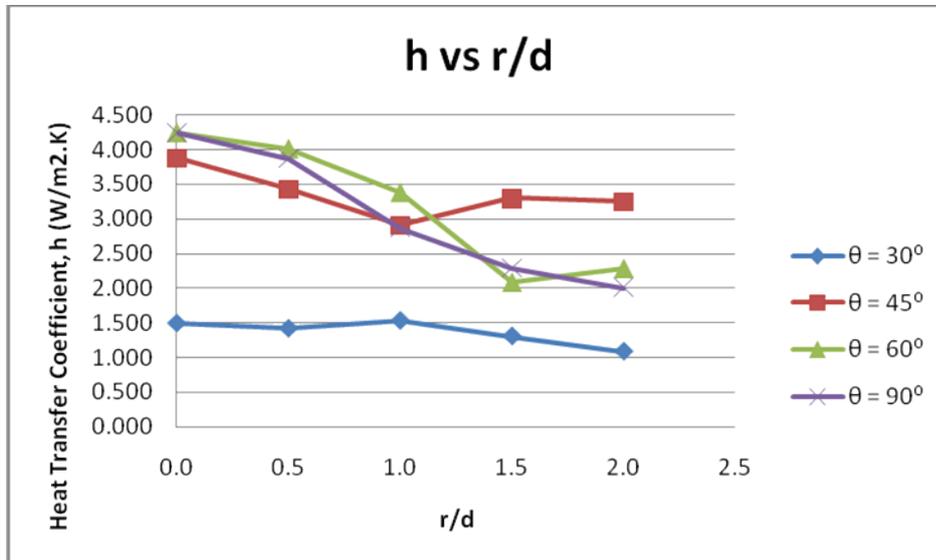
Graph For Reynolds No= 930 and H/D=6



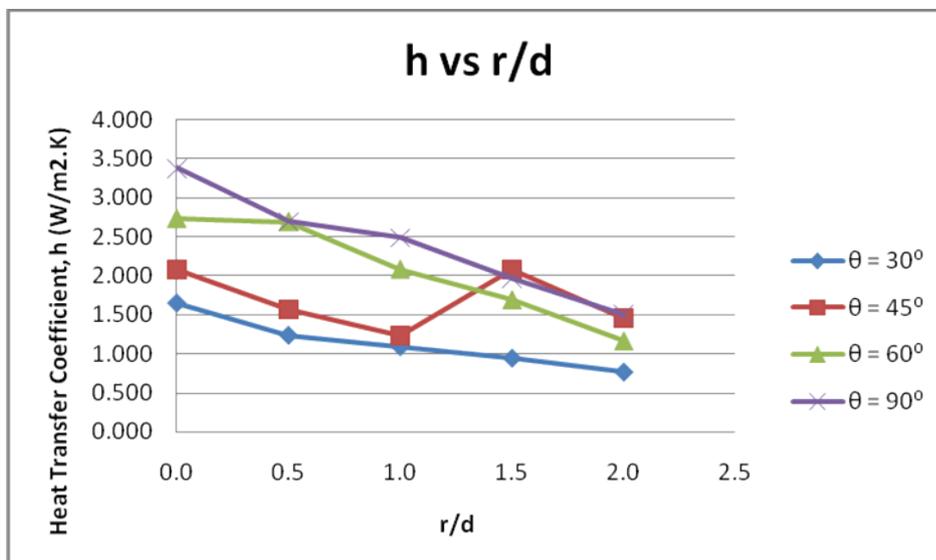
Graph For Reynolds No= 930 and H/D=8



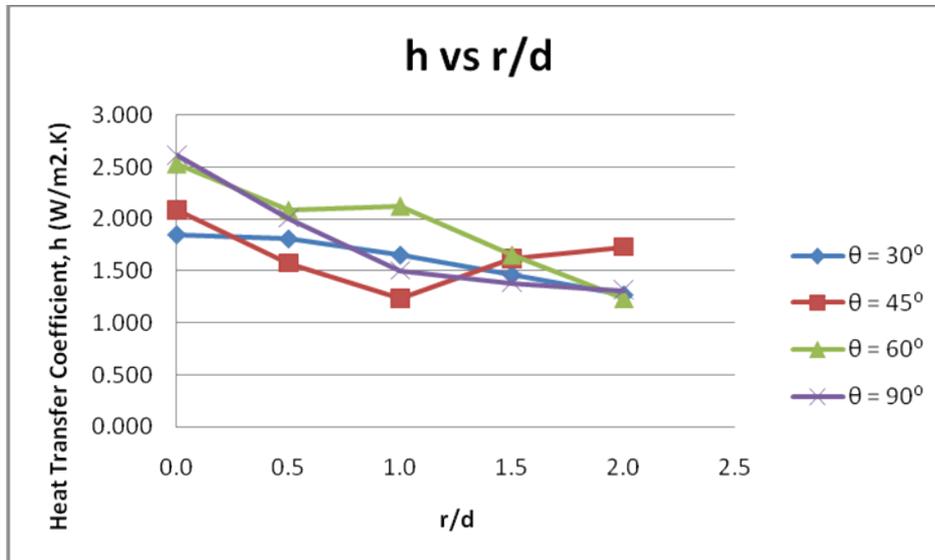
Graph For Reynolds No= 500 and H/D=2



Graph For Reynolds No= 500 and H/D=4



Graph For Reynolds No= 500 and H/D=6



Graph For Reynolds No= 500 and H/D=8

