EFFECT OF NOZZLE ANGLE ON JET IMPINGEMENT COOLING SYSTEM

KHAIDER BIN ABU BAKAR

Report submitted in fulfilment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering

Faculty of Mechanical Engineering
UNIVERSITI MALAYSIA PAHANG

NOVEMBER 2009

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.

Signature

Name of Supervisor: MR. WAN AZMI BIN WAN HAMZAH

Position:LECTURER

Date: 23 NOVEMBER 2009

STUDENT'S DECLARATION

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 concurently submitted for award of other degree.

Signature
Name: KHAIDER BIN ABU BAKAR
ID Number: MA 06053
Date: 23 NOVEMBER 2009

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

ACKNOWLEDGEMENTS

First and foremost, grateful to Allah SWT for making it possible for me to complete this project on time. It is pleasure to acknowledge the help and support of everyone concerned with this project. To my father, mother and brothers who have been so keen in helping and encouraging me during the several months it took me to complete this thesis.

I would like to thank and express my deepest appreciation to my supervisor, Mr. Wan Azmi bin Wan Hamzah for providing me opportunity for his helpful reviews, comments and suggestion during the progression of this project. . I am truly grateful for his progressive vision about my training in science, his tolerance of my naive mistakes, and his commitment to my future career. I also sincerely thanks for the time spent proof reading and correcting my many mistakes.

Finally, my sincere thanks go to all my labmates Mr. Mohammad Khyru bin Aris, Mr.Dulfharah Nizam bin Memth Ali, members of the staff of the Mechanical Engineering Department, UMP : Mr. Mohammad Khalid bin Wahid, Mr. Nizam Bin Abdullah, Mdm. Noor Sa'adah binti Nor Azmi, and members of the laboratory staff Chemical Engineering Deparment,UMP, for the cooperation.

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^o, 45^o, 60^o, and 90^o), distance between nozzle's edge to the impinge surface (H/d= 2, 4, 6, and 8) and also Reynolds number (Re= 2300, 1960,930 and 500).The heat source are heated at 100 ^oC 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^o in range 31.5 w/m^2K 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 hubung 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, 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 w/m^2K 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.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

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
$°C$	Degree Celcius
^{o}	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
υ	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^{o}, 45^{o}, 60^{o} and 90^{o}.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^{o}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.

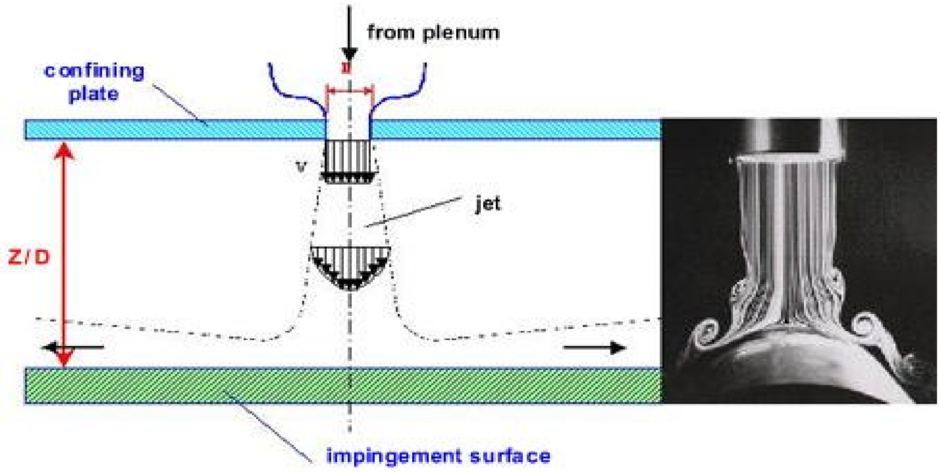


Figure 2.1: Example one of the jet impingement

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:

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

Important factor that effecting heat transfer is due to h, convection heat transfer coefficient, W/m^2.^{o} 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.

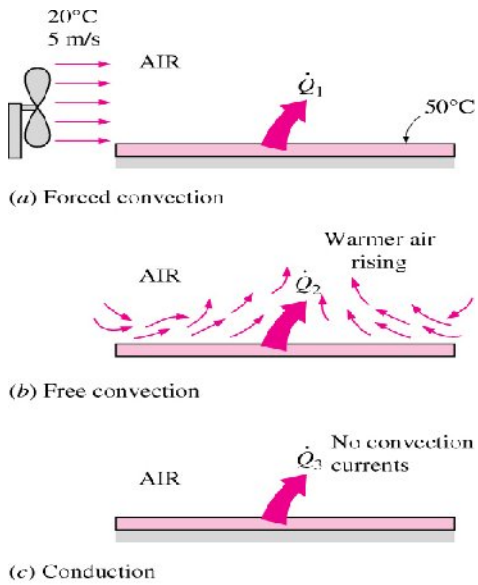


Figure 2.2: Comparison on Forced convection, natural convection and conduction

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

$$\mathrm{Re} = \frac{\text{Inertia forces}}{\text{Viscous forces}} = \frac{VL_c}{\nu} = \frac{\rho VL_c}{\mu} \tag{2.2}$$

Where

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

$$Nu = hL_c / k \tag{2.3}$$

Where k is the thermal conductivity of the fluid and Lc 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:

$$\dot{Q} = \dot{m} c_p \Delta T \qquad \text{or} \qquad \dot{Q} = \rho V A_c \tag{2.4}$$

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/m^3)

ΔT = Temperature different, (K)

A_c = Cross section area, (m^2)

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 70to 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	86.40	86.90	85.80	80.50
	D	95.30	94.30	95.80	96.90	97.50	97.60	97.90	97.80	98.10	98.00	98.00	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	86.00	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	87.60	87.70	87.80	89.50	87.60
	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

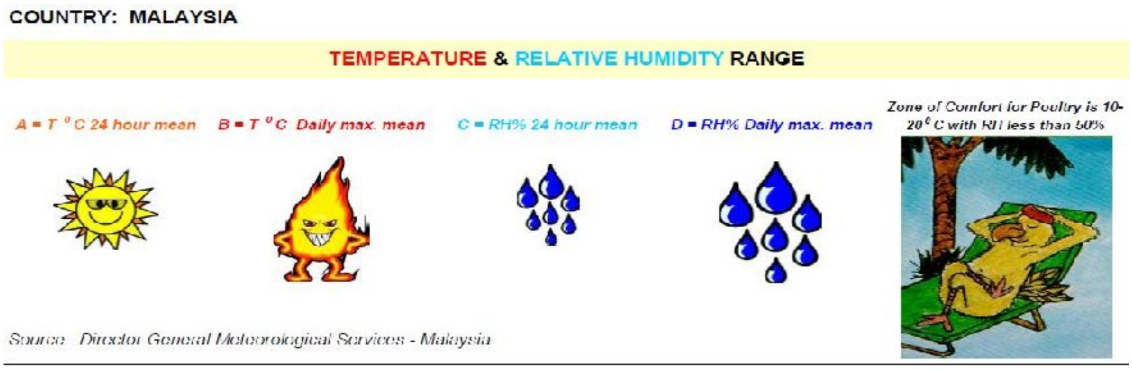


Figure 2.3: Temperature and Relative Humidity Range in Malaysia

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.

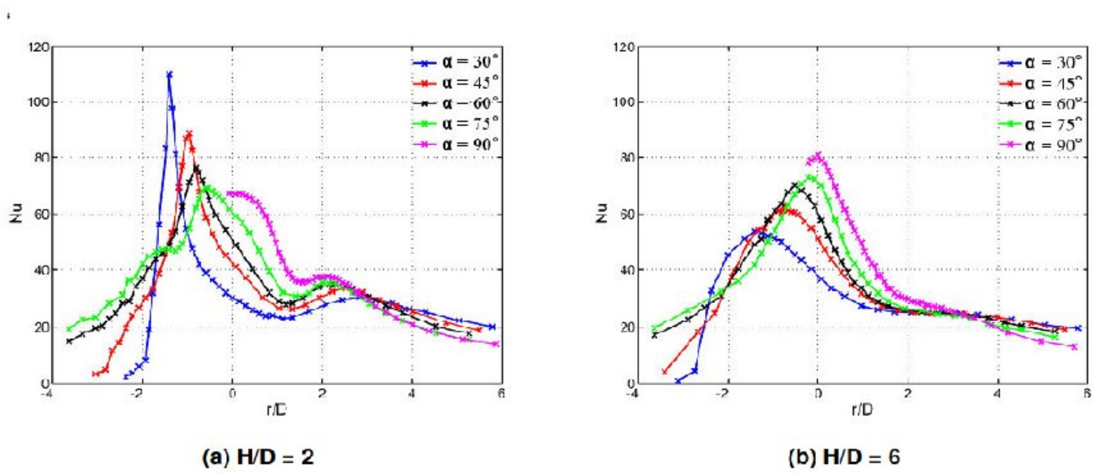


Figure 2.4: Graphs of obliquely Impinging Jet Mean Nusselt Number Distributions; Re = 10000

Source: Donovan (2005)

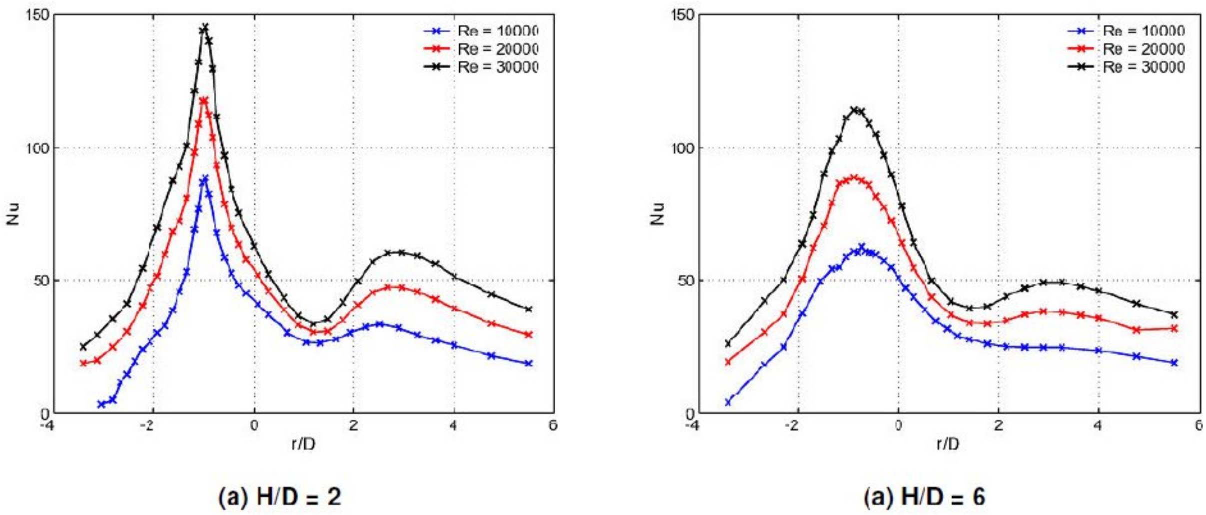
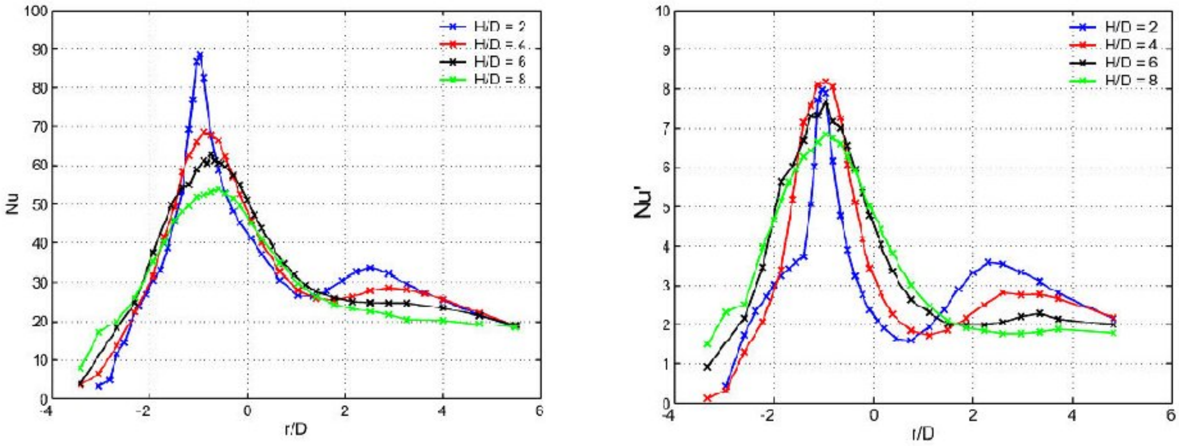


Figure 2.5: Graphs of obliquely Impinging Jet Mean Nusselt Number Distributions Angle= 45°

Source: Donovan (2005)



(a) Mean

(b) Root-Mean-Square

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.