EXPERIMENTAL STUDY OF NATURAL CONVECTION HEAT TRANSFER IN A VERTICAL INTERNALLY FINNED TUBE

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for the award of the degree of
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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 with Automotive Engineering.

Signature

Name of Supervisor:

Position:

Date:

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 concurrently submitted for award of other degree.

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"To my beloved mother, father, my sister, family, and my beloved friends who give me support towards this study"

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بسم الله الرحمن الرحيم

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ABSTRACT

This paper present an experimental study of a vertical internally finned tube subjected to natural convection heat transfer. All the main parameters that can significantly influence the heat transfer performance of finned tube have been examined. An experimental set-up was designed to study the heat transfer performance in the entrance region as well as in the fully-developed region. I using material called mild steel and its fins are assembly with the body of the cylinder tube internally. The length of tube was 100mm. The inner diameter of tube was 80mm and the outer diameter was 90mm. The tube contains four radial, straight, and equally distributed around the circumference of the tube. The height of the fins are 100mm and the length of the fins are 25mm. Air was used as a working fluid in the experiment. It is found that, unlike expected, the value of Nu for vertical cylinder under variables time varies with the temperature is increasing. Also it is found that the value of Nu increases as the time increases. In this experiment we I using a mild steel as a material for cylinder tube and fins. When mild steel are used and under the conditions of the current experiment no improvement were introduced by using mild steel material. To be conclusive on this result a further experimentations are needed by using aluminum for the cylinder tube material and different length of cylinder tube or its diameter and also using different variables length of fins.

ABSTRAK

Projek ini menjalankan eksperimen perolakan semula jadi peminadahan haba bagi kedudukan menegak untuk tiub yang mempunyai sirip di dalam nya. Kesemua parameter utama mempengaruhi perlaksanaan pemindahan haba yang amat nyata sekali telah di selidik. Eksperimen telah di bentuk untuk mengkaji perlaksaan pemindahan haba. Saya telah menggunakan bahan jenis aloi keras untuk tiub sirip. Tiub dan sirp digabungkan bersama iaitu di dalam tiub. Panjang tiub adalah 100mm. Diameter dalam adalah 80mm dan diameter luar adalah 100mm. Tiub mengandungi empat sirip yang radial, sejajar dan dibahagikan sama sepanjang lilitan tiub. Tinggi sirip adalah 100mm dan panjang adalah 25mm. Udara digunakan sebagai bendalir dalam eksperimen ini Didapati, tidak seperti yang di jangka, nilai Nu untuk silinder tiub yang menegak di bawah pemboleh ubah masa berubah dengan suhu semakin meningkat. Didapati nilai Nu meningkat dengan kenaikan masa. Dalam eksperimen ini saya menggunakan alaoi keras untuk tiub dan sirip. Apabila menggunakan aloi keras untuk eksperimen ini tiada peningkatan diperkenalkan. Secara kesimpulan, untuk keputusan eksperimen yang akan datang memerlukan menggunakan aluminum sebagai bahan untuk tiub dan sirip. Dan juga panjang yang berlainan untuk tiub silinder atau berlainan diameter. Juga menggunakan panjang sirip yang berlainan sebagai pemboleh ubah.

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CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Natural mode of heat transfer employed in many applications as a higher pressure drop penalty. The pitch was seen to play an important role in the entrance region. The-core region insert was shown to enhance the heat transfer much more than the wall region insert, contrary to the earlier assumption that the core may not play as important a role. However, the combined effect of the two regions could be different from the individual influence of each region. The need for compact and efficient heat transfer convection heat transfer is important equipment has stimulated research efforts in the area of heat transfer augmentation. In order to enhance the rate of heat transfer, finned surfaces have been applied to cooling devices for electronic equipment and compact heat exchangers for many years.

Convection involves the transfer of heat by the motion and mixing of “macroscopic” portions of a fluid (that is, the flow of a fluid past a solid boundary). The term forced convection is used if this motion and mixing is caused by an outside force, such a pump. The transfer of heat from a hot water radiator to a room is an example of heat transfer by natural convection. The transfer of heat from the surface of a heat exchanger is an example of forced convection.

Natural or “Buoyant” or “Free” convection is a very important mechanism that is operative in a variety of environments from cooling electronic circuit boards in computers to causing large scale circulation in the atmosphere as well as in lakes and oceans that influences the weather. It is caused by the action of density gradients in conjunction with a gravitational field. This is a brief introduction that will help you understand the qualitative features of a variety of situations you might encounter.

There are two basic scenarios in the context of natural convection. In one, a density gradient exists in a fluid in a direction that is parallel to the gravity vector or opposite to it. Such situations can lead to “stable” or “unstable” density stratification of the fluid. In a stable stratification, less dense fluid is at the top and denser fluid at the bottom. In the absence of other effects, convection will be absent, and we can treat the heat transfer problem as one of conduction. In an unstable stratification, in which less dense fluid is at the bottom, and more dense fluid at the top, provided the density gradient is sufficiently large, convection will start spontaneously and significant mixing of the fluid will occur.

Figure 1.1: Natural convection scenarios

The example is the flow of air at the tip of a lit cigarette; in this case, the smoke from the cigarette actually traces that flow for us. In a common technique used for home heating, the baseboard heater consists of a tube through which hot water flows, and the heater is placed close to the floor. The tube is outfitted with fins to provide additional heat transfer surface. The neighboring air is heated, and the hot air rises, with cooler air moving in toward the baseboard at floor level. This natural convection circulation set up by the hot baseboard provides a simple mixing mechanism for the air in the room and helps us maintain a relatively uniform temperature everywhere. Clearly, the convection helps the heat transfer process here.

The used of heat transfer enhancement has become widespread during the last 50 years. The goal of heat transfer enhancement is to reduce the size and cost of heat exchanger equipment, or increase the heat duty for a give size heat exchanger. This goal can be achieve in two ways: active and passive enhancement. Of the two, active enhancement is less common because it requires addition of external power (e.g., an electromagnetic field) to cause a desired flow modification. On the other hand, passive enhancement consists of alteration to the heat transfer surface or in corporation of a device whose presence results in a flow field modification. The most popular enhancement is the fin.

Heat transfer by convection is more difficult to analyze than heat transfer by conduction because no single property of the heat transfer medium, such as thermal conductivity, can be defined to describe the mechanism. Heat transfer by convection varies from situation to situation (upon the fluid flow conductions), and it is frequently coupled with the mode of fluid flow. In practice, analysis of heat transfer by convection is treated empirically (by direct observation).

Convection heat transfer is treated empirically because of the factors that affect the stagnant film thickness:

- Fluid velocity
- Fluid viscosity
- Heat flux
- Surface roughness
- Type of flow (single-phase/two-phase)

1.2 PROJECT BACKGROUND

Internally finned tubes have received considerable attention because of the fact that they have been used widely in industrial applications. Internally finned tube has found extensive use in heat exchangers. When improvement in the process of heating or cooling is required, then better design of fin compactness and spatial geometry is very essential. Several studies have been conducted to investigate the effect of fin characteristics on heat transfer. Most of the relevant previous works have focused on limited cases of the number and length of the internal fin.

The apparent advantages of fins are that they increase the heat transfer rate by providing additional surface area. However, fins placed in a tube cause complex flow patterns and increase flow resistance. As the number or the height of fins increases, flow friction increases, thus requiring greater pumping power to sustain a given mass flow rate. Therefore, to design a compact heat exchanger with internally finned tubes, we should optimize the fin geometry by accounting for both flow friction and heat flux.

The present study was undertaken to develop new experimental data on Nusselt number and heat transfer during laminar flow an air as a medium in internally finned tube with vertical position. Even though there have been several numerical studies on fluid flow and heat transfer performance of internally finned tubes, the experimental data for thermally developing flow in internally tubes has been scarce. Therefore, the new data expected to provide very valuable and timely addition to the finned tube.

The analysis is base on the vertical internally finned tube. During the experiment the data taken was temperature at the surface, fins, and inside the tube.

1.3 PROBLEM STATEMENT

The problem to be considered is that of Natural Convection heat transfer for fully developed flow in an internally finned tube. The fins are radial, straight and equally distributed around the circumference of the tube as shown in Figure 1.2. The length of the fin is fixed. The flow is subjected to a uniform heat input flux unit axial length. Because of the symmetry the calculation domain is performed over a half sector (the complete sector is the area between the two consecutive fins).

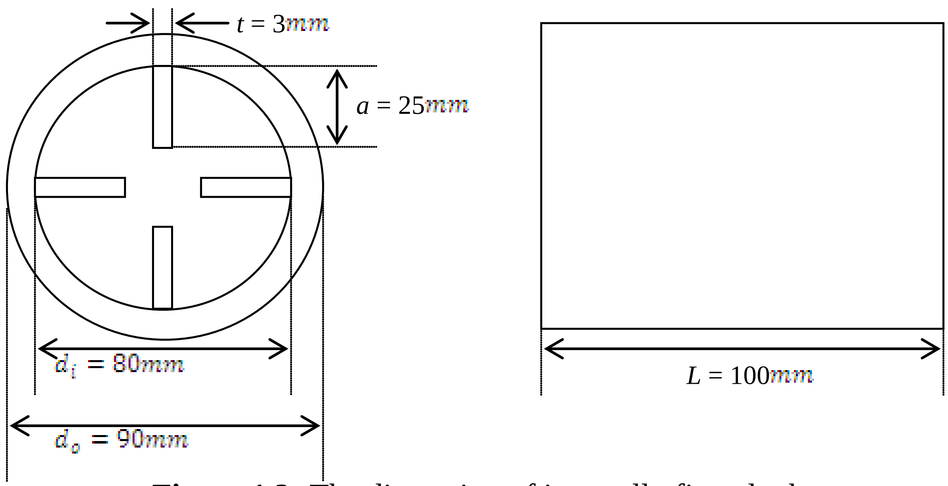


Figure 1.2: The dimension of internally finned tube

1.4 OBJECTIVES

The objectives of the project are to determine the effect of flow in the tube with fins. Besides, in this project I also have to determine the heat transfer from surrounding in the vertical position of the cylinder tube. Then I also need to find the value of the coefficient with the given formula:

$$Nu = C_1 . {R_a}^{C_2} . a^{C_3}$$

Where, I need to find ad calculate the coefficient of $C_1, C_2\ and\ C_3$.

1.5 PROJECT SCOPES

i. Used air as a medium
ii. Used laminar flow
iii. Measure the temperature
iv. Measure the time to show the behavior surface temperature with time.
v. Calculate the heat transfer, Nusselt number and find the coefficient.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

Internally longitudinal finned tubes are widely used in many engineering fields to enhance heat transfer, such as power plants, chemical process and petroleum industries. Many researchers have investigated the problem of optimizing the shape of the finned surfaces in order to reduce the weight and the size of heat exchangers and increase heat transfer.

2.2 NATURAL CONVECTION FROM EMBEDDED VERTICAL CYLINDERS

Minkowycz and Cheng [1] gave an approximate solution for heat transfer in the case of vertical cylinders embedded in porous media. The following equations describe the problem of steady natural convection about a vertical cylinder of radius R embedded in a saturated porous medium with a prescribed wall temperature.

$$\frac{\partial}{\partial r}(rv) + \frac{\partial}{\partial x}(ru) = 0 \qquad (1)$$

$$u = \frac{K}{\mu}\left(\frac{\partial p}{\partial x} + g\right) \qquad (2)$$

$$v = -\frac{K}{\mu}\frac{\partial P}{\partial r} \qquad (3)$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial r} = \alpha\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{\partial^2 T}{\partial x^2}\right] \tag{4}$$

The boundary condition for this problem is:

$$v = 0, T = T_w \;\; at \;\; r = R \tag{5}$$

$$u \to 0, T \to T_\infty \;\; at \;\; r = \infty \tag{6}$$

For the case of constant wall temperature, the ratio of the local surface heat flux for a cylinder $(q_c")$ to that of a vertical plate $(q")$ is given by:

$$\frac{q_c"}{q"} = \frac{[-\theta'(\xi,0,\lambda)]}{[-\theta'(0,\lambda)]} \tag{7}$$

Where the curvature parameter $q_c"$ and $q"$ are defines as:

$$\xi = \frac{2x}{R}\frac{1}{(R_{a_x})^{1/2}} \tag{8}$$

Huang and Chen [2] studied the effects of surface suction or blowing in a case of a cylinder subject to uniform heat flux density. Two models were used: two-temperature model and one-temperature model. Kimura [3] studied the transient problem and showed that the transient heat transfer for both forced and natural convection can be expressed in a unified manner with a single parameter representing the curvature effect of the cylinder surface. Yucel and Lai et al. [4, 5] studied combined heat and mass transfer problems in vertical cylinders. The mixed convection along a slender cylinder with variable surface heat flux considering Darcian flow problems were studied by Pop et al. [6]. This study also analyzed the effects of surface curvature and buoyancy in surface heat flux. The numerical solution of the transformed equations was obtained using the Keller box method to study the significant influence of these factors on the flow and heat transfer characteristics. The mixed convection along a cylinder with variable surface heat flux was analyzed by Aldoss et al [7]. When non-Darcy model was applied and no similarity solutions were obtained for the case of variable wall temperature. Kumari and Nath [8] and Aldoss [9] analyzed the mixed convection problem for conduction fluids under magnetic fluid.

2.3 NATURAL CONVECTION FOR EMBEDDED HORIZONTAL CYLINDERS

Merkin [10] studied the problem of a horizontal cylinder embedded in porous media. This is a special case of a general two-dimensional heated object that he analyzed. For the large Rayleigh number, the heat transfer coefficient can by calculated using:

$$Nu = 0.565 {R_{a_D}}^{1/2} \tag{9}$$

Where:

$Nu = \frac{hD}{k}$

$R_{a_D} = \frac{gdK\,(T_W - T_\infty)}{\nu\,\alpha_m}$

α_m= thermal diffusivity for porous media

h = average heat transfer coefficient

The results were later generalized by Chen and Chen [11] for fluids with non-Newtonian viscosities. They extended it to power law fluids. Fand et al. [12] studied heat transfer problems and conducted experiments in porous media packed with random glass spheres using different fluids. This study suggested the following relationships for different ranges of Reynolds numbers:

For $0.001 < Re_{max} = a$

$Nu = Pr^{0.0877} = 0.618 {R_{a_D}}^{0.698} + 8.54 \times 10^6 GeSech\left(R_{a_D}\right)$ for $0.001 < Re_{max} 3$ (10)

$Nu = Pr^{0.0877} = 0.766 {R_{a_D}}^{0.37} (\frac{C_1 D}{C_2})^{0.173}$ for $3 < Re_{max} \leq 100$ (11)

Where:

$Nu = \frac{hD}{k}$

$Ge = \frac{gD}{c_p}$

C_1 and C_2 are constants.

The problem of horizontal cylinders embedded in a semi-infinite porous media was analyzed by Farouk and Shayer [13]. The heat transfer experiments of this problem were done by Fernandez and Schrock [14]. Cheng [15] studied the problem of mixed convection about a horizontal cylinder. Others, like Sano [16], Ingham et al. [17], Pop et al. [18, 19], Tyvand [20], Sundfor and Tyvand [21] and Bradean et al. [22] used detailed analytical and numerical analysis of transient natural convection from embedded horizontal cylinders.

2.4 HYDRAULIC AND HEAT TRANSFER PERFORMANCE IN INTERNALLY FINNED TUBE

The earliest works on straight internally finned tubes were experimentally done by Vasil'chenko and Barbaritskaya [23,24]. Transformer oil was used as the test fluid with operating conditions of $200 < Re < 10{,}000$ and $70 < pr < 140$. Both laminar and turbulent heat transfer were observed in this region. Transition Reynolds number for tubes with different number of fins of fin height were different. Five finned tubes with fin geometries of $4 \leq N \leq 8$ and $0.26 \leq H \leq 0.63$ were included in this study which showed 30 to 70% increases in heat transfer with the use of finned tubes compared to smooth tubes done by Xiaoyue [25].

Watkinson et al [26, 27] tested 18 finned tubes; five had straight fins and 13 had spiral fins. Water and air were used as test fluids with tube geometry of $6 \leq N \leq 50,\ 0.05 \leq H \leq 0.30$ and $0° \leq \gamma \leq 15°$ with $1000 \leq Re \leq 100{,}000$ and $1.0 \leq Pr \leq 3.4$ for water and with $10{,}000 \leq Re \leq 100{,}000$ and $Pr = 0.71$ for air. At $Re = 50{,}000$ the experimental results indicated that a 17 to 95% heat transfer enhancement (Nusselt number increase) over the smooth tube for air and 15 to 85% was gained for water, depending on different fin geometries. Spirally finned tubes showed better heat transfer enhancement performance than straight finned tubes. Analysis confirmed that fin efficiencies were close to 100%. Correlations developed for friction factor and Nusselt numbers were strongly dependent on specific fin geometries and fluid properties used in their experiments' and limited to rather small helix angles.

Obot et al [26] and Obot [28,29] also has developed a method to correlate pressure drop and heat transfer in smooth and rough passages (including finned tubes) through the use of the critical Reynolds number for transition. Smooth tube correlations gave results satisfactory for design calculations when the usual Reynolds number was replaced by a reduced Reynolds number. The critical parameters at the onset transition to turbulent flow are required to calculate this number, so a priori knowledge of transition is needed to estimate performance using this approach. The transition Reynolds number is another global flow characteristic variable, similar to the friction factor, dominated by the particular tube geometries, which is connected closely with the mechanism of turbulent flow. Therefore, it may be as difficult to predict as friction factor itself.

More recently, Vlakanic [30] tested fifteen spirally finned tubes for obtaining the pressure drop and heat transfer performance. The fin geometries covered $8 \leq N \leq 54,\ 0.015 \leq Pr \leq 0.17$ and $0^\circ \leq \gamma \leq 15^\circ$. Friction factors and Nusselt numbers were measured in the range of $7{,}000 \leq Re \leq 70{,}000, 0.62 \leq \mu_b \mu_w \leq 1.35$ and $3.5 \leq Pr \leq 5.0$ for both heating and cooling conditions. According to the experimental data obtained with water, it was concluded that the heat transfer performance was augmented by about 20 to 200% compared to a smooth tube depending on the specific fin geometry. The corresponding pressure drop penalty increase ranged from 40% to about 170%.

Compared with previous studies on internally finned tubes, Vlakancic's data may be the largest and most complete database in overall friction factors and heat transfer performance. Because varieties of fin geometries and wide range of Reynolds numbers were covered in the experiments, the effect of fin heights, fin width and helix angle were extensively investigated including both “tall fins” and “micro-fins”.

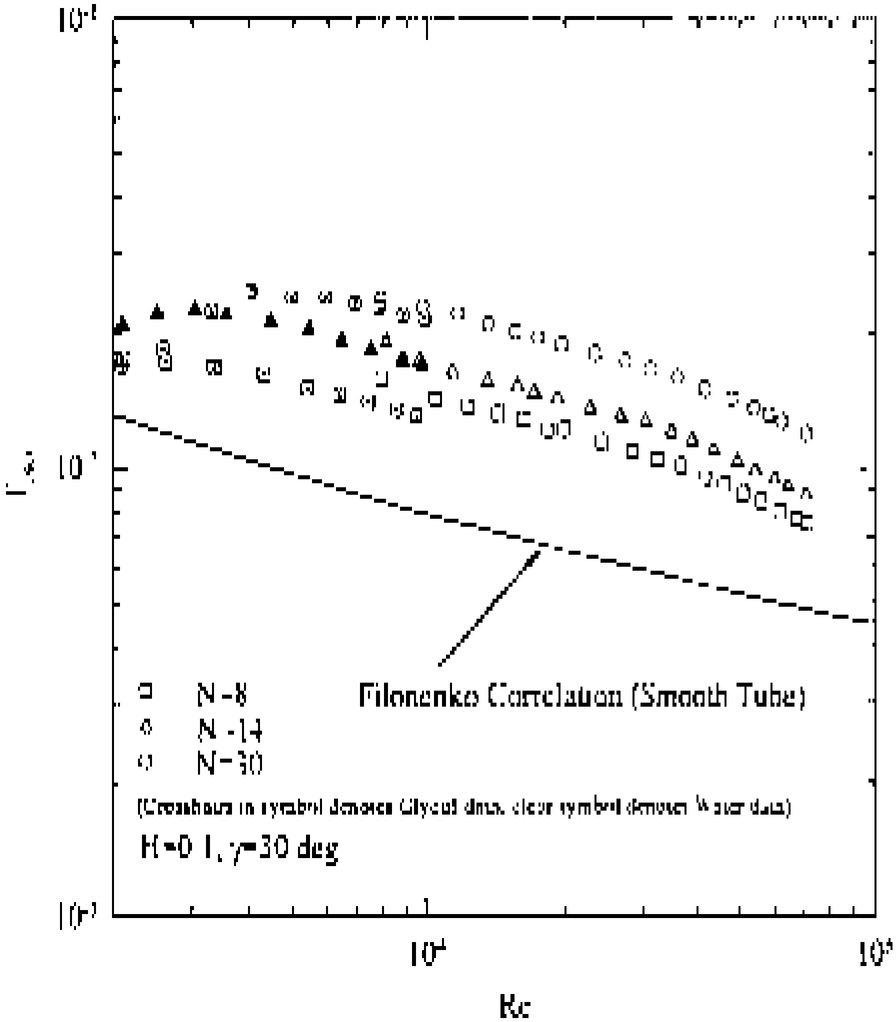


Figure 2.1: The Number of Fin Effect on Fraction Factor, Vlakancic [30]

The fin number effect on friction factors is shown in Figure 3 for tubes with relatively high fin height. As expected, the friction factors increased with more fin numbers. For N=8, the friction factors increase compared to a smooth tube was around 70%, for N=14 it was about 100% and 140% for N=30. It is obvious that the increase in fraction factor with the increase of fin numbers is not linear. The friction factor increase from N=14 to 30 is much larger than N=8 to 14. Figure 2.1 showed a similar friction factor comparison between different fin numbers but they had a much shorter fin height (H=0.03). The striking feature was that the friction factor profiles were very different

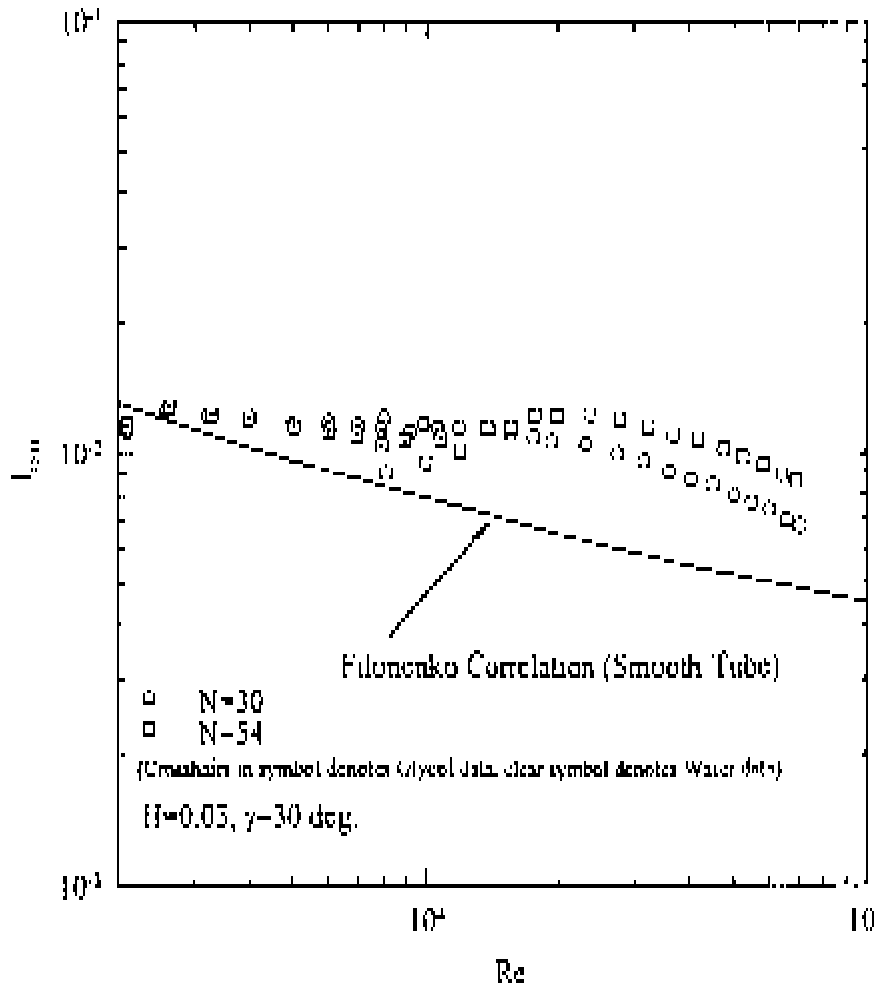


Figure 2.2: The Effect of Number of Fin on Friction for Shorter Fins, Vlakancic[30]

from that of tubes with higher fin height. At low Reynolds number between 4,000 to 15,000, the friction factors remained almost unchanged with Reynolds number and there was no appreciable difference for the friction factors between N=30 to N=54. The tubes with very short fin height and relatively more fin numbers are referred as “micro-fin” tubes in industry. This peculiar phenomenon of friction factors at low Reynolds number was also observed by Al-Fahed et al [31] and Chiou et al [32] and still remains unexplained.

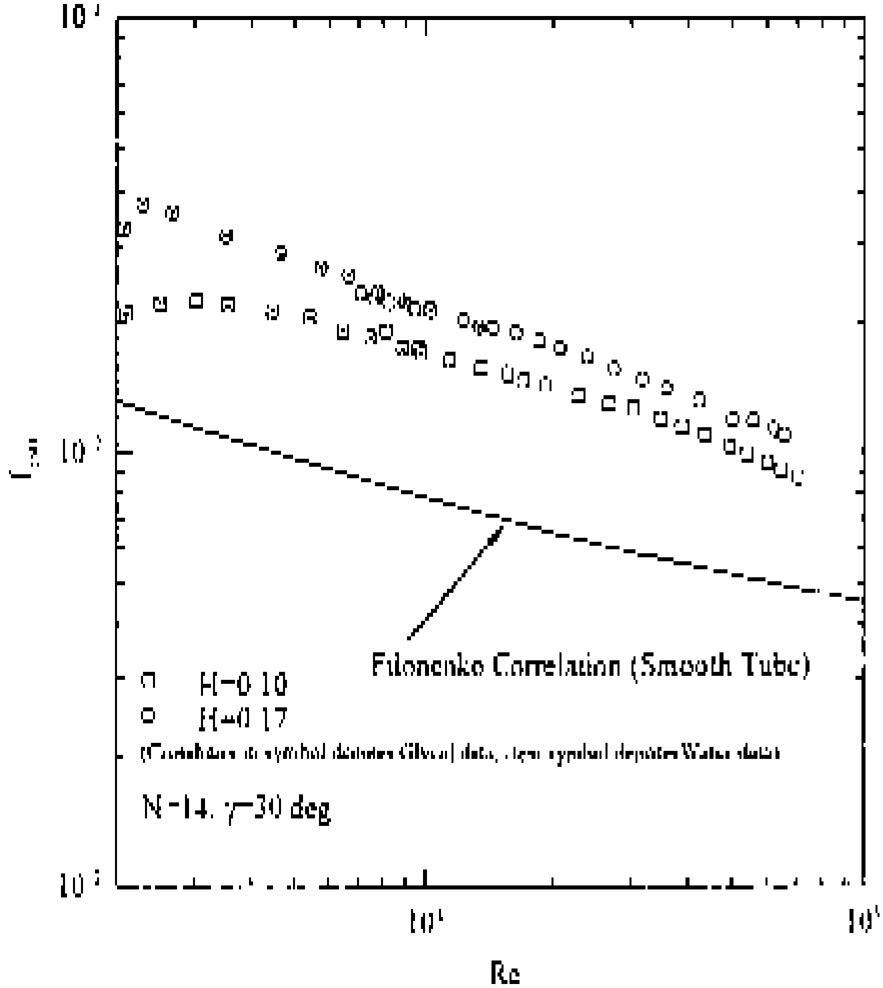


Figure 2.3: The Effect of Fin Height on Friction Factor, Vlakancic [30]

The fin height effects on friction factors are shown in Figure 2.3. The friction factors increased with the increased fin height. But the increase compared to a smooth tube depends on fin number and helix angle. An exceptional case can be seen from Figure 2.4, where the tube with H=0.02 possessed higher friction factors than that of the tube with H=0.03. It was stated by the author that this tube with H=0.02 had the smaller inter-fin region area compared with other tubes, so that the flow field in the inter-fin region was strongly affected by the side walls of the fins. In this case, the fin shape may also play an important role in the fraction factor performance. Only length-averaged data were measured during experiments. No detailed flow field and heat transfer data were available for looking into the governing mechanism.

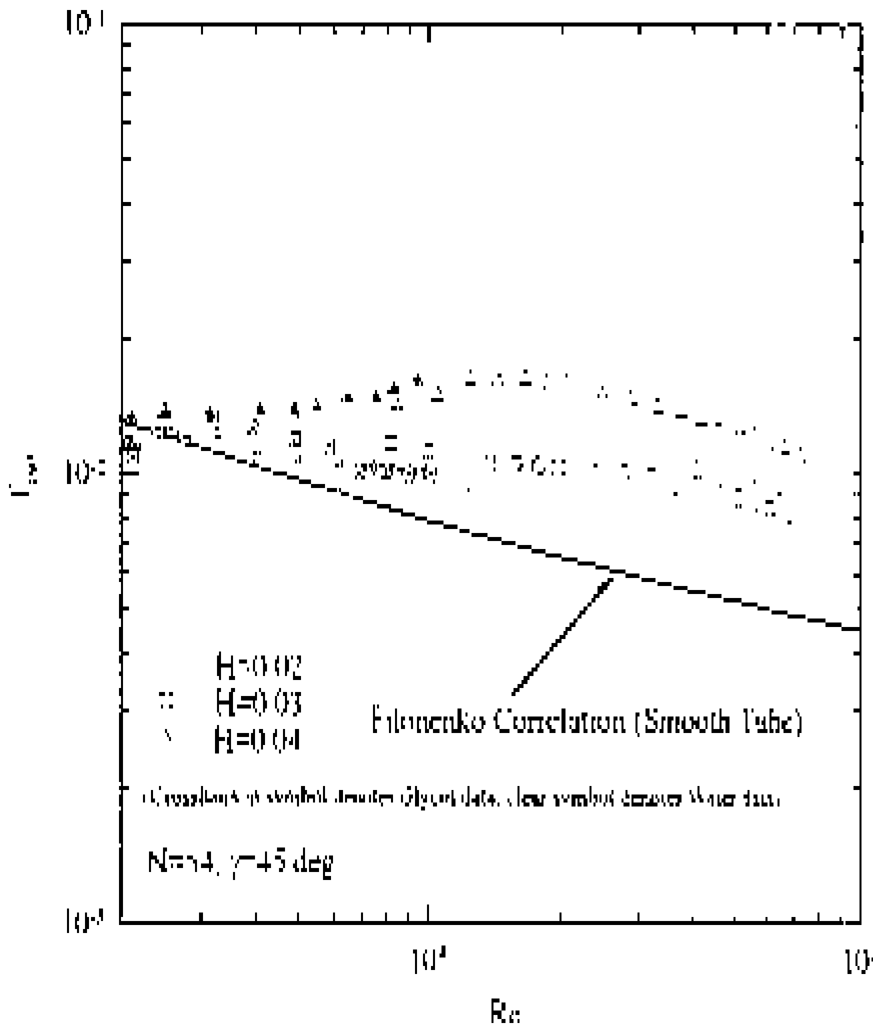


Figure 2.4: Inter-fin Region Effect on Friction Factors, Vlakancic [30]

The helix angle effects on friction factors were also investigated. He concluded that friction factors usually increased with increasing helix angle for all fins or small fin numbers. For micro-fin tubes, the inter-fin area or fin shapes might play more important roles in certain situations when their scale is comparable to fin height.

The impact of fin numbers, fin height and helix angle on heat transfer performance were well addressed in Vlakancic's work. Generally speaking, increasing the fin number, fin height and helix increased the Nusselt number as result of the increase of heat transfer area. However, there are exceptions to this. As shown in Figure 2.5, the Nusselt numbers actually decreased when the fin height increased from 0.1 to 0.17. On the contrary, friction factors increased with fin height for this particular geometry.

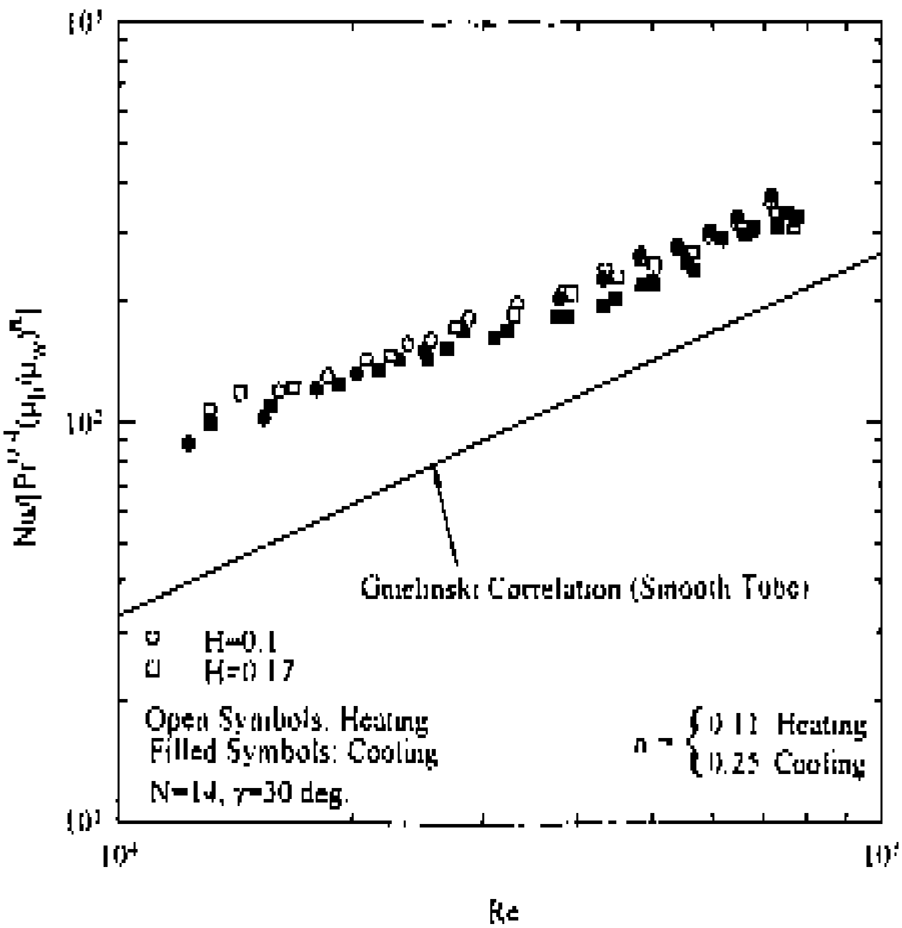


Figure 2.5: The Inverse Effect of Fin Height on Nusselt Number, Vlakancic [30]

2.5 CONVECTION HEAT TRANSFER FROM A CYLINDER WITHOUT FINS

On the other hand, a considerable work was carried out by several researchers regarding of convection heat transfer from a cylinder (vertical and horizontal) without fins. Cebeci [33] studied the convection heat transfer from vertical cylinder under constant heat flux condition. He suggested that an outer surface of a vertical cylinder can be treated as a vertical plate when the diameter of the cylinder is sufficiently large so that the curvature effects are negligible. This condition is satisfied if $D/L \geq 35{G_L}^{-0.25}$. When this criterion is satisfied, the relations for vertical plate can also be used for vertical cylinder. Churchill and Chu [34] have compiled the data of a number of workers in the filed and recommended a correlation as:

$$Ra_D \leq 10^{12} \; Nu = \{0.6 + \frac{0.387 {Ra_D}^{1/6}}{\left[1+(0.559/Pr)^{9/16}\right]^{8/27}}\}^2 \tag{12}$$

Fuji, et al. [35] studied experimentally the natural convection heat transfer from the outer surface of vertical cylinder to liquid and developed correlations. Tetsu and Uehara [36] study laminar natural convection along the outer surface of a vertical cylinder and compared it with laminar natural convection along vertical plate and they found temperature distribution and heat flux at cylinder surface.

2.6 LAMINAR AND TUBULENT FLOW

For both laminar and turbulent flow regimes, the finned tubes exhibited substantially higher heat transfer coefficients when compared with corresponding smooth (un-finned) tubes. Hu and Chang [37] analyzed fully developed laminar flow in internally finned tubes by assuming constant and uniform heat flux in tube and fin surfaces. By using 22 fins extended to about 80% of the tube radius, they showed that an enhancement as high as 20 times that of un-finned tube could be realized. Soliman et al. [38] also analyzed fully-developed laminar flow, but accounting for conduction in the tube wall and fins and keeping the outer surface of the tube at a constant temperature. In a later study, Prakash and Patankar [39] investigated the influence of buoyancy on heat transfer in vertical internally finned tube under fully-developed laminar flow condition. Analyses of thermally-developing laminar flow in internally finned tubes were carried out by Prakash and Liu [40] and Rustum and Soliman [41] by numerically solving the governing conservation equations. They concluded that internal finning influences the thermal development in a complicated way, which makes it inappropriate to extend the smooth tube results to internally finned tubes on a hydraulic diameter basis. Patankar et al. [42] analyzed the fully-developed turbulent flow and heat transfer characteristics for tubes and annuli with longitudinal internal fins using the mixing length model. The local heat transfer coefficients exhibited a substantial variation along the fin height, with smallest value at the base and largest value at the tip. Experimental data for pressure drop and heat transfer coefficient in internally finned tubes has been reported, among others, by Hiding and Coogan [43], Bergies et al.[44], Watkinson [26], and Rustum and Soliman [45]. Bulk of these data characterized the laminar flow regime using air, water, and oil as working fluids.

2.7 INTERNALLY FINNED TUBE

Masliyah and Nandakumar [46] have studied the heat transfer in internally finned tube. The internal fins were of triangular shape and the number of fins was changed up to 24 fins and the length up to 0.8 of the tube radius. Finite element method was used to analyze a laminar fully developed flow in an internally finned circular tube with uniform axial heat flux around the wall. They conclude that the Nusselt number based on the inside diameter was higher than that for a smooth tube without fins and also they found that for maximum heat transfer there exists an optimum fin number for a given fin configuration. The influence of the buoyancy force in combined free and forced convection in vertical tubes with internal fins was studied by Patankar and Prakash [39]. A laminar fully developed flow was solved for the velocity and temperature using finite difference technique. Straight radial fin configurations were analyzed for a range of Rayleigh number and for various values of fin height up to 0.8 of the tube radius and number of fins up to 25 fins. In their result they found that the buoyancy force increases the friction and heat transfer. The effect of buoyancy is more significant when the number of fins is small and the fins are short. For smooth finless circular tube, the fully developed combined forced and free convection has been solved analytically by Morton [47] and also by Tao [48] and investigated numerically by Kemeny and Somers [49]. Numerous studies have been focused on studying different shapes and arrangement of longitudinal shrouded fin array

The problem of laminar fully developed flow in circular tube with internal radial fins has been studied numerically by Masiliyah and Nandakumar [46] and analytically by Hu and Chang [37]. However, detailed studies for a wide range of fin number practically encountered in the industry for a wide range of fin length are not available in literature.

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION:

Natural convection is the fluid flow originated by gravity forces acting on non-uniform-density fluids; the density changes may be due to thermal or to solutal gradients. Many different natural-convection configurations are of interest, from the simplest hot/cold vertical plate in a fluid medium, to external convection around hot/cold bodies, or internal convection within hot/cold enclosures (non-isothermal).

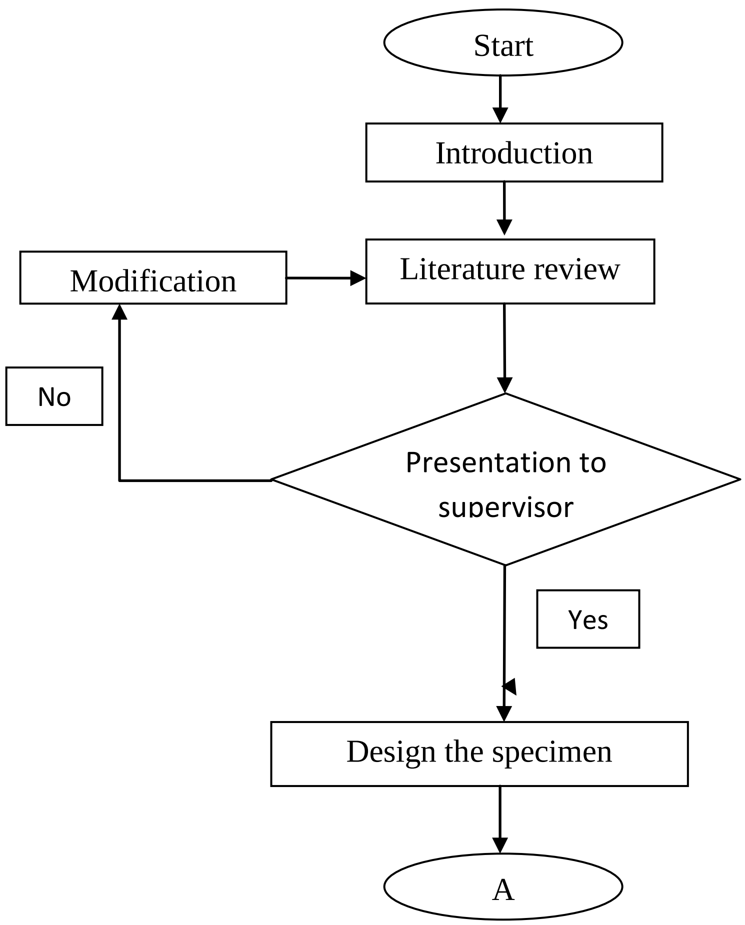
A small tilting in the hot vertical plate analysed before, already causes great changes in the flow, since the boundary layer detaches at several places along the upper side of the plate (if hot; the lower side in a cold plate), forming three-dimensional patches due to flow instabilities. That is why most heat and mass natural-convection correlations are empirical fittings from experimental data.

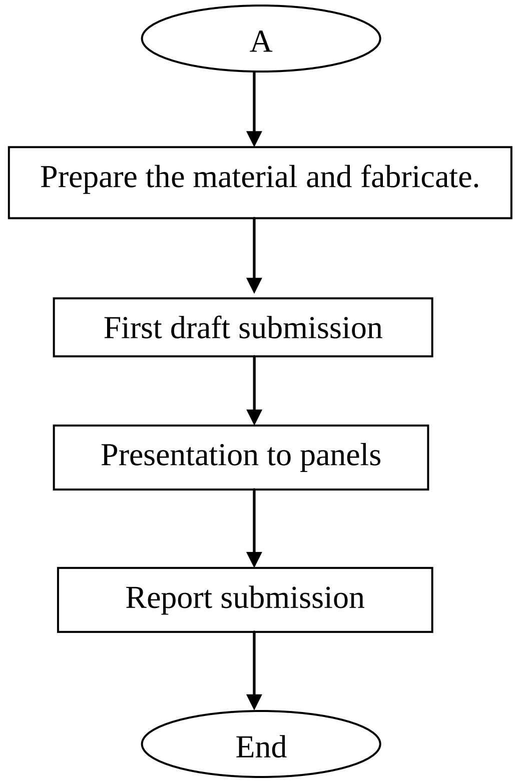
Ambient air is the omnipresent convection fluid in all terrestrial applications, either by natural convection due to buoyancy, or by forced convection with a fan. Any electrical appliance serves to illustrate the point.

Air is a free commodity, clean, non-flammable, non-corrosive, and it does not boil or freeze (well, frosting may be a problem with humid air, and any fluid would condense at a low-enough temperature). However, it has a very low thermal conductivity and density. Hydrogen or helium is used when higher conductivity gases are needed.

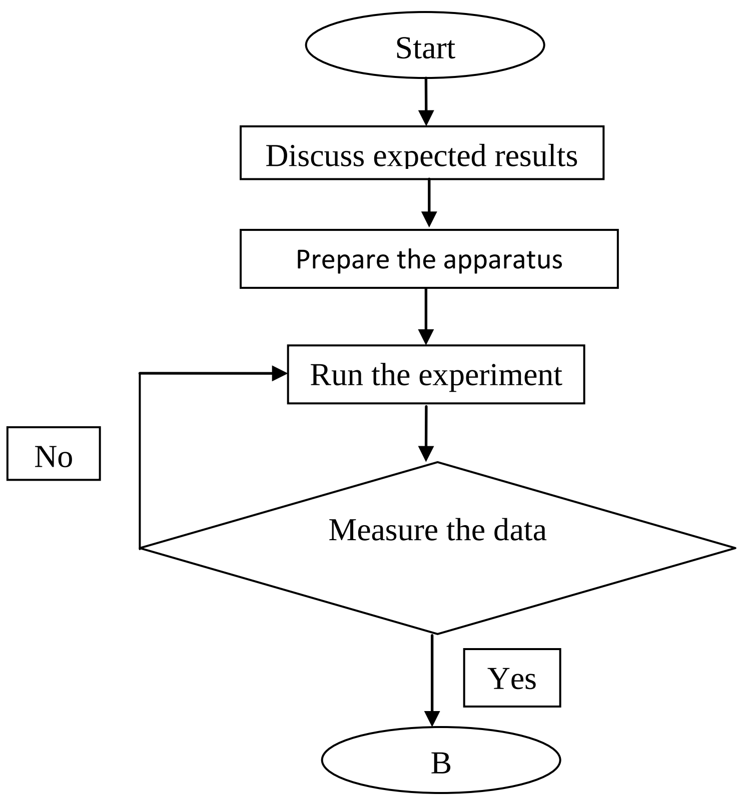
3.2 FLOW CHART:

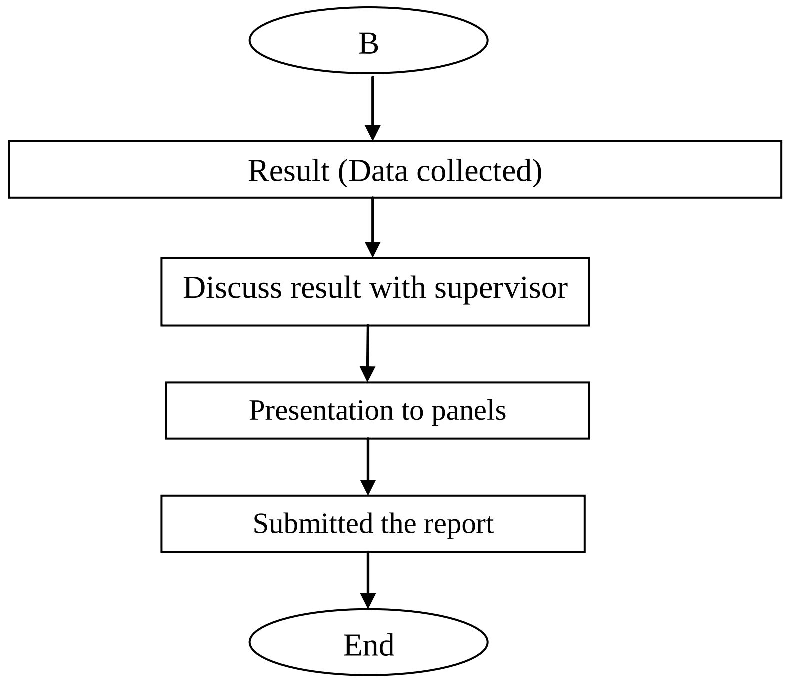
3.2.1 Flow Chart for FYP 1



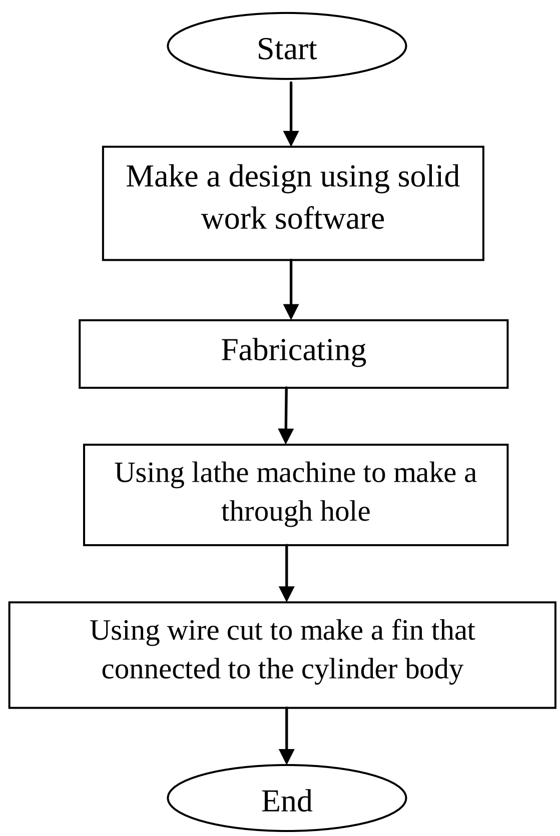


3.2.2 Flow Chart for FYP 2

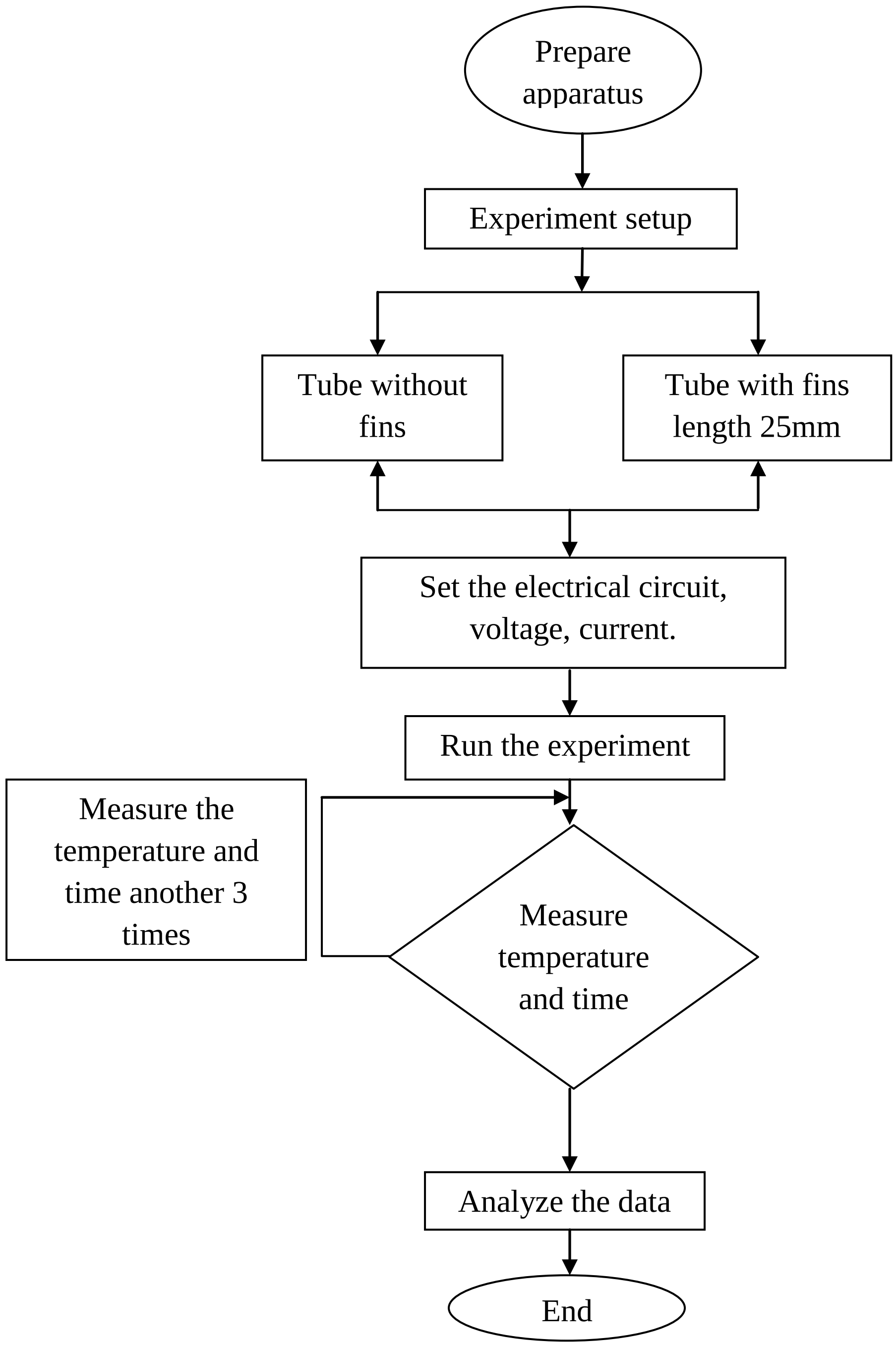




3.2.3 Flow Chart for Design and Fabricate the Specimen



3.2.4 Flow Chart for Experiment



3.3 APPARATUS

Table 3.1: The apparatus and its descriptions

Apparatus	Descriptions
Thermocouple	To measure the point that experienced heat transfer in the cylinder tube.
Laser Thermometer	To measure temperature at the surface of cylinder tube and fins surface.
Digital Thermocouple	To measure the temperature inside the cylinder tube which between fins and the middle of the tube.
Hair dryer	Used to heat the cylinder tube.

3.4 MATERIAL

The material that will apply for cylinder tube is aluminum. This material has low density and has ability to resist corrosion. This material also has low thermal conductivity and it can be good thermal conductor. But in this experiment mild steel material has been used for cylinder finned tube.

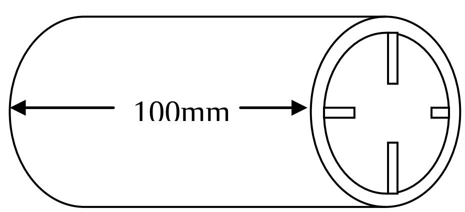


Figure 3.1: Dimension of material

3.5 SCHEMATIC DIAGRAM

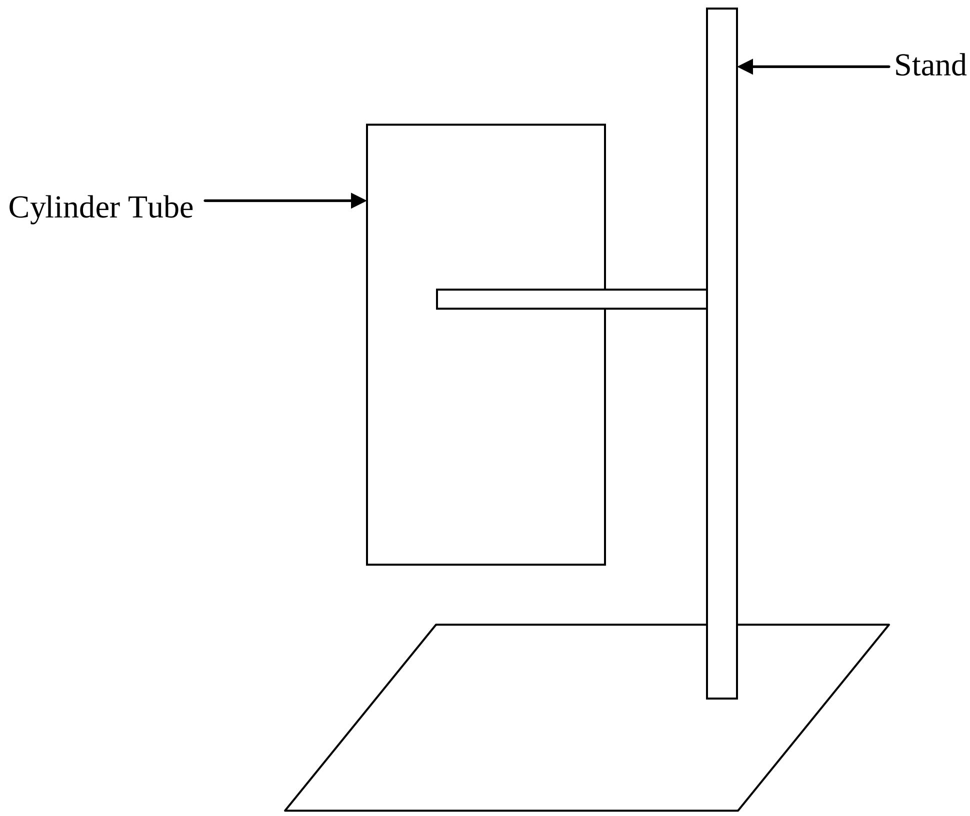


Figure 3.2: Schematic Diagram

3.6 EXPERIMENT SETUP AND PROCEDURE

The finned tube used for experiments were manufactured from mild steel. Finned tube was designed using Solid-work before fabricate it. The schematic diagram of the experimental facility is depicted in Figure 9. The procedure to run experiment was:

(i) First, have to setup the cylinder vertically and hold it with a stand so that it far from ground.
(ii) Before put the cylinder tube with the stand have to cover the cylinder with aluminum strips. Because it's easy for heater to heat the mild steel cylinder tube.
(iii) After that, heat the cylinder finned tube with hair dryer. Used four hair dryer so that cylinder tube can heat constantly around the cylinder.
(iv) Then, set the timer. After 30 seconds, stop the hair dryer and measure the temperature ambient, which is temperature inside the tube using digital thermocouple.
(v) Also measure the temperature at the surface of the tube. Take about four reading of temperature at the surface using laser thermometer.
(vi) Measure the fin surface temperature too with four reading.

Since, the experiment setup that has been run in this project is too simple and its not follow the exact experiment for convection. Suhil [50] has conducted the exact experiment for natural convection heat transfer with the vertical position finned tube. In his experiment he used a porous fin but the procedure for his experiment can be used for the experiment for this project. The procedure for his experiment is:

(i) The cylinder heated internally over its entire length with uniform heat flux generated by a Kapton heater (33.4Ω), and thermally insulated from both ends with Bakelite insulating material of thickness 20 mm each end.
(ii) The magnitude of the heat flux was adjusted by varying the intensity of the current measured with the ammeter and supplied by the direct current power supply. Temperatures of the surface at different axial positions were recorded using twelve K-type omega thermocouples.

(iii)The procedure followed during each experiment is as follows. An initial power input was supplied and then adjusted such that the heat flux gives Ra_L in the required range. The temperatures along the cylinder were continuously measured and monitored with a scanning each 0.617 sec. Usually an initial period of approximately 2–3 hrs was required before reaching steady-state conditions (considered to be attained, when the temperatures indicated by the thermocouples did not vary with more than ±1 C within a period of about 2 min). To reduce the noise specific to each sensor as well as the noise induced in the electric wires by the surrounding electromagnetic fields, each data point was obtained by averaging 40 discrete values acquired with the above mentioned rate. After collecting a set of data at steady-state conditions, the supplied mass current to the heater was increased so that the next value of q" is obtained.

(iv) A new set of data was collected when steady-state conditions were reached again, usually within a period of approximately 30-45 minutes. Heat losses from the heated section from cylinder ends through the insulation material by conduction, to the atmosphere by radiation and natural convection were accounted for by applying the Equation (17 and 18) below.

(v) The porosity of each porous medium was determined by measuring the mass of all screens present in its structure. In order to compare the numerical predictions with the experimental results, permeability K and inertia coefficient F of the porous media were experimentally determined by using the Forcheimer equation:

$$\Delta p = \frac{\mu L_0}{K} v_D + \frac{\rho F L_0}{\sqrt{K}} v^2{}_D \tag{13}$$

Equation (13) can be rewritten in the following form

$$\Delta p = \alpha_o v_D + \beta_o v^2{}_D \tag{14}$$

Where coefficient α_o and β_o are defined as:

$$\alpha_o = \frac{\mu L_0}{K} \text{ and } \beta_o = \frac{\rho F L_0}{\sqrt{K}} \tag{15}$$

The procedure followed relies on measuring the pressure drop Δp over a porous medium of thickness Lo, in a wind tunnel whose test section is completely filled with porous material, at different Darcian velocities vD. By fitting a second order polynomial through these points (see Figure 10) the coefficient α_o and β_o given in equation (15) can be determined.

Pressure drops measured at different flow velocities over the porous media was presented in Figure 3.3. The dashed line in this figure indicates the curve fitted through the experimental points. By plugging the values of α_o and β_o from these figures into Eq. (15) one can obtain the corresponding values for permeability K and inertia coefficient F. Then, using the values for permeability K, one can also determine the corresponding the numbers. The values of these parameters are summarized in Table 2.

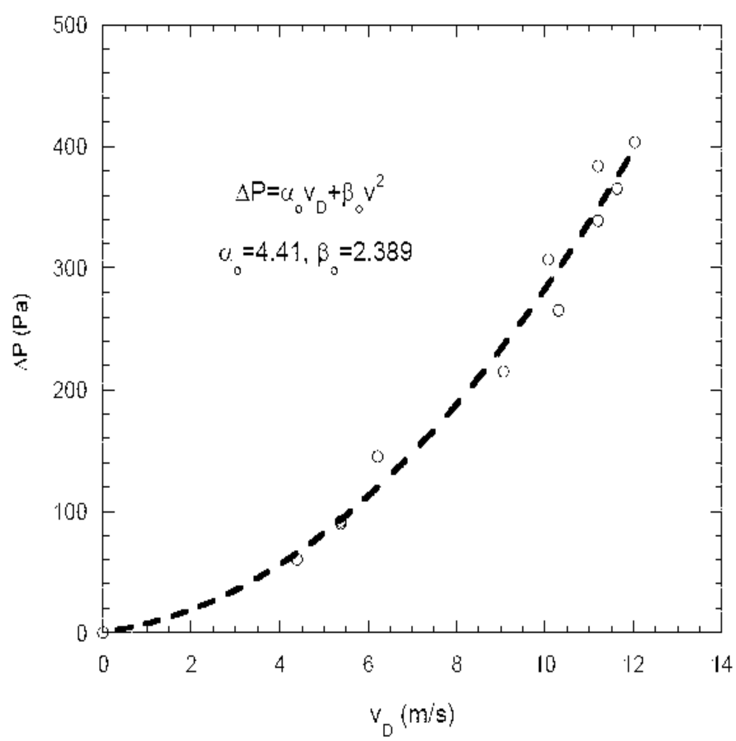


Figure 3.3: Experimental Determination of K and F, Suhil [50]

Table 3.2: The Characteristics of the Porous Material used, Suhil [50]

Material	α_o	β_o	Permeability, K	Forchheimer, F
Porous A	0.223	0.2617	3.52×10^{6}	0.008
Porous B	0.441	0.238	1.78×10^{6}	0.006
Porous C	1.234	2.156	6.53×10^{6}	0.029

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 INTRODUCTION

This chapter will discuss about the result that has been taken during the experiment. The data will be analyzed. The parameter from this experiment is the temperature.

This research aimed at experimentally investigating the effect of using fins on the natural convection heat transfer from a finned vertical cylinder. This should lead to obtaining an experimental data for the convection heat transfer from a vertical cylinder subjected to constant and uniform heat flux simulating a fin-tube heat exchanger, and, hence, a design parameter finned heat exchangers. This work is also aimed at studying the effect of parameters such as fin spacing and Rayleigh number on the heat transfer for the problem under consideration.

4.2 RESULTS

4.2.1 Analysis Results

Table below show the entire results for both cylinders with fins and without fins. Result below has been get from the experiment for this project.

Table 4.2.1: Data for Cylinder Tube without Fins

Time, s	**Tsurface, $^{\circ}C$**	**Tambient, $^{\circ}C$**	**Heat Flux ($\dot{q}$), W/m^2**	**Heat Transfer ($\dot{Q}$), W**
30	41.95	30.7	86.9178	2.1485
70	44.43	31.0	107.852	2.7106
120	49.78	32.0	150.896	3.7920
180	53.30	33.1	174.780	4.3927

Table 4.2.2: Data for Cylinder Tube with Fins

Time, s	**Tsurface, $^{\circ}C$**	**Tambient, $^{\circ}C$**	**Heat Flux ($\dot{q}$), W/m^2**	**Heat Transfer ($\dot{Q}$), W**
30	37.88	30.7	49.740	0.9948
70	41.45	31.0	79.135	1.5827
120	47.30	32.0	125.550	2.5109
180	50.80	33.1	148.797	2.9759

Besides calculate the heat flux and heat transfer, Rayleigh number and Nusselt number also need to calculate. By calculating Nusselt number can find the coefficient of heat transfer by the given formula which is:

$Nu = C_1 . Ra^{C_2} . a^{C_3}$

Table 4.2.3: Nusselt Number and Rayleigh Number for Cylinder Tube without fins

Nusselt Number	Rayleigh Number
29.32	7.93838×10^6
30.36	8.9813×10^6
31.798	10.5744×10^6
32.213	11.071×10^6

Table 4.2.4: Nusselt Number and Rayleigh Number for Cylinder Tube with fins

Nusselt Number	Rayleigh Number
26.44	5.49595×10^6
28.75	7.4029×10^6
30.86	9.5188×10^6
31.41	10.1347×10^6

Table 4.2.5: Coefficient for Cylinder Tube with fins

Time, s	Nusselt Number	Coefficient
30	26.44	1.013
70	28.75	1.033
120	30.86	1.051
180	31.41	1.055

Since, the experiment for this project is not follow the exact experiment that has been conducted from previous study, here is the analysis result from previous experiment that has been conducted by Suhil [50].

4.2.2 Analysis Result by Suhil

In the Suhil [50] experiments' he said that the heat generated by the electrical heater inside the cylinder dissipates from the cylinder surface by radiation and convection. Part of this heat will transfer to the outside from cylinder ends by conduction through the thermal insulation material. Therefore, an energy balance to the cylinder leads

$$P_{elec} = A_s(q''_c + q''_r) + A_{insul}q''_{insul} \tag{16}$$

Where the value of *Pelec* is calculated form the measured values of the voltage drop and the current passing in the electrical heater. The part of the energy lost by radiation is calculated from the equation:

$$q''_r = \varepsilon\sigma(T_s{}^4 - T_\infty{}^4) \tag{17}$$

And the energy lost by axial conduction through the insulation material is calculated from the equation:

$$q''_{insul} = k_{insul}\frac{\Delta T}{\lambda} \tag{18}$$

He also concluded it should be mentioned that temperatures used for radiation losses is the measured surface temperature and the temperature drop across the thermal insulation material is measured experimentally. In the experiment that has been conducted by Suhil, the value of the emissivity, ε, is estimated to be 0.04 for polished aluminum and the value of kin for Bakelite is taken to be 0.2 W/m. K. Its worth mentioning that measurements indicated that the heat lost by radiation and the axial conduction heat are about 5% each. Based on the above procedure, the convection heat flux at any location along the cylinder is estimated by

$$q''_c = \frac{P_{elec} - A_c q''_{insul}}{A_s} - q''_r \tag{19}$$

He concluded that in the case of natural convection heat transfer from vertical cylinder when no porous material used, the obtained results are analyzed in order to compare it with the available results in the literature. The local heat transfer coefficient is calculated from the equation

$$h_x = \frac{q''_c}{T_x - T_\infty} \tag{20}$$

And, hence, Nusselt numbers and the modified Rayleigh number are obtained using

$$Nu_{x\infty} = \left.\frac{h_x X}{k}\right|_{T_f} \text{ and } Ra_{x\infty} = \left.\frac{g\beta q''_c x^4}{vk\alpha}\right|_{T_f} \tag{21}$$

Where the properties are evaluated at T_f (film temperature) which is defined as

$$T_f = \frac{T_s + T_\infty}{2} \tag{22}$$

Finally the average Nusselt number for each cylinder at certain heat flux is estimated using the equation

$$\overline{Nu}_L = \frac{\bar{h}_i L}{k} \tag{23}$$

Where the value of average heat transfer coefficient is obtained from either:

$$h_1 = \frac{q''_c}{\bar{T} - T_\infty} \tag{24}$$

Where $\bar{T}$ is the arithmetic mean temperature along the cylinder calculated by using:

$$\bar{T} = \sum_1^{12} \frac{T_x}{12} \tag{25}$$

Or, $\bar{h}_2 = \frac{1}{L}\int_0^L h_x \, dx$ (26)

Suhil has study the performance of using porous material required additional analysis. He introduced and used two concepts which is surface effectiveness and fin efficiency.

First concept is effectiveness efficiency, ε_o. Effectiveness surface is a measure of the heat transferred from the cylinder when porous material is used compared to that when no porous material is used:

$$\varepsilon_o = \frac{q_{w/porous}}{q_{w/o}} \tag{27}$$

q_{porous}= heat transferred from cylinder covered by porous material.

$q_{w/o}$= heat transferred from cylinder without using porous material

Where, it was calculated using:

$$q = \bar{h}A_s(T_w - T_\infty) \tag{28}$$

And can find h from the correlation:

$$\bar{h} = \frac{\overline{Nu_L}k}{L} = \frac{k}{L}\int_0^L \frac{CRa_x{}^m}{x}dx = \frac{kCRa_L{}^m}{4mL} \tag{29}$$

The constant m and C can be obtained from the correlations for each case. By substituting (29) into (27):

$$\varepsilon_o \frac{[(C/4m)Ra_L{}^m]_{w/fin}}{[(C/4m)Ra_L{}^m]_{w/ofin}} \tag{30}$$

Second concept that has being using by Suhil is fin efficiency, η is defined as the actual rate of heat transfer from the fin to the maximum possible heat transfer rate from the fin:

$$\eta = \frac{q_{fin}}{q_{max}} = \frac{\bar{h}A(\bar{T}_w - T_\infty)}{\bar{h}_{max}A(\bar{T}_w - T_\infty)} \tag{31}$$

By deriving an expression for maximum average Nusselt number for a horizontal porous fin attached to a vertical wall under natural convection condition, he found that

$$Nu_L = \frac{hL}{k_{eff}} = 0.8165\sqrt{Ra_{2L}} \tag{32}$$

Where,$Ra_{2L} = \frac{g\beta(T_b - T_\infty)KL}{\alpha v k_r}$ (33)

The definition of Ra_{2L} given in equation (33) can be modified to constant heat flux conditions using

$$q" = k\frac{(T_b - T_\infty)}{L} \text{ and becomes } Ra_{2L} = \frac{g\beta q"KL^2}{\alpha\nu k_{eff}} \tag{34}$$

And the value of $\overline{h_{max}}$ can be found from equation (32) becomes:

$$h_{max} = \frac{k_{eff}}{L}(0.8165\sqrt{Ra_{2L}}) \tag{35}$$

4.3 DISCUSSIONS

4.3.1 Discussion for this Project

The result and discussion that can be found from experiment natural convection heat transfer in a vertical internally finned tube is mostly the surface and fin temperature for cylinder tube without fin is higher than surface and fin temperature for cylinder tube with fin.

Since the temperature for the cylinder tube without fin is higher than with fin, the heat flux and heat transfer is also higher than with fin. The Nusselt number is also high for cylinder tube without fin.

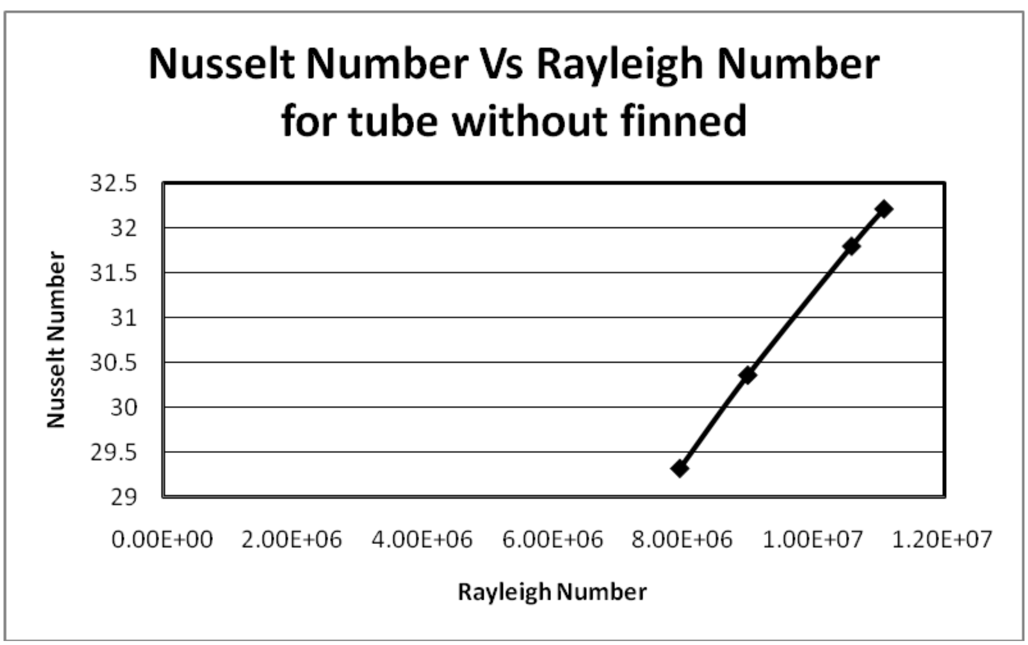


Figure 4.1: Graph Nusselt Number Vs Rayleigh Number for Cylinder Tube without fins

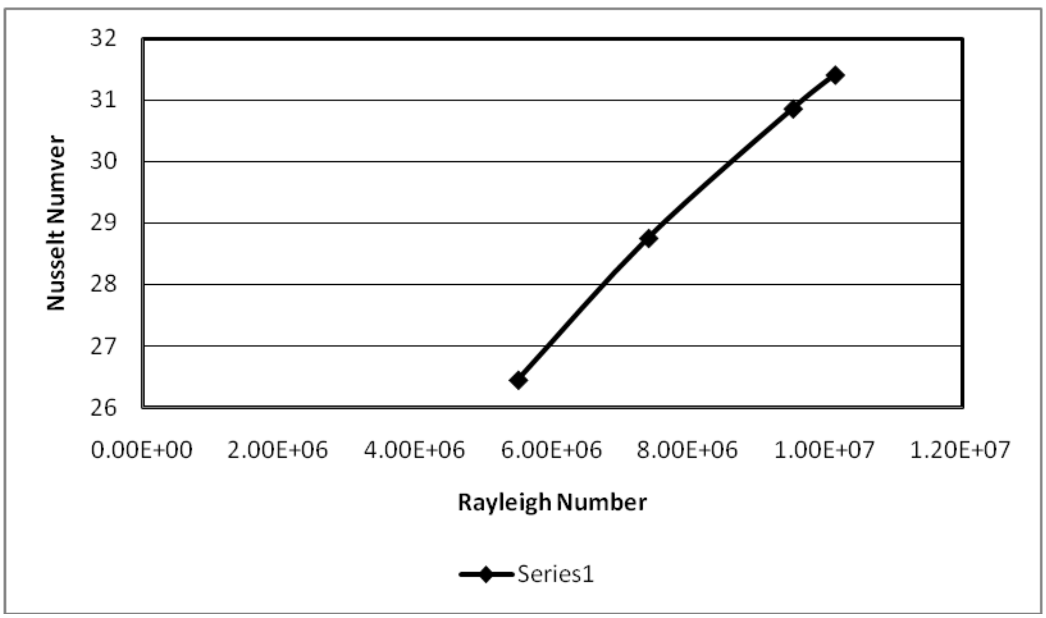


Figure 4.2: Graph Nusselt Number Vs Rayleigh Number for Cylinder Tube with fins

From Figure 4.1, shown that graphs for Nusselt number against Rayleigh number for cylinder tube without fins. From the graph can concluded that Nusselt number increase uniformly with Rayleigh number.

For graph Figure 4.2 has shown Nusselt number against Rayleigh number for cylinder tube with fins. The Nusselt number for this graph also increase uniformly with Rayleigh number but for the slope, Figure 11 shown sharply slope than graph figure 12.

4.3.2 Discussion from Experiment by Suhil

As we know in the early chapter, the experiment for this project is not really followed the previous experiment by Suhil so there is not too many have to discuss since the result also cannot being used. So, here is the discussion that has been made by Suhil for his experiment.

There is two folds has been presented by Suhil. Which are the data collected from vertical cylinders without using porous material (clear cylinder case) and the data collected when porous fins are attached to the surface of the cylinder

4.3.2.1 Heat Transfer from Vertical Cylinder without Porous Material

These cases are originally initiated in order to verify the experimental setup and to compare our results with the correlations obtained by other researches. Measured data were collected for different heat fluxes and two different cylinder diameters.

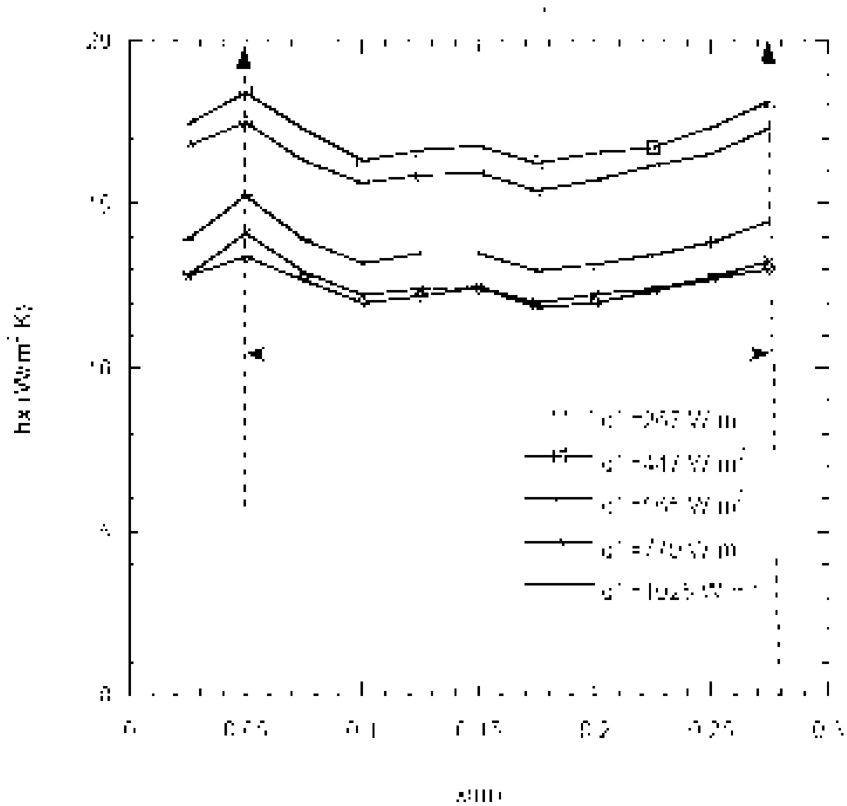


Figure 4.3: The variation of local heat transfer coefficient along the cylinder for selected heat fluxes, Suhil [50]

Figure 4.3 show that the value of the heat transfer coefficient depends on the heat flux. It should be the value of h must be independent on the heat flux. This figure also indicated that an x increases the value of h decreases until the middle of the cylinder where beyond that point as x increases h increases.

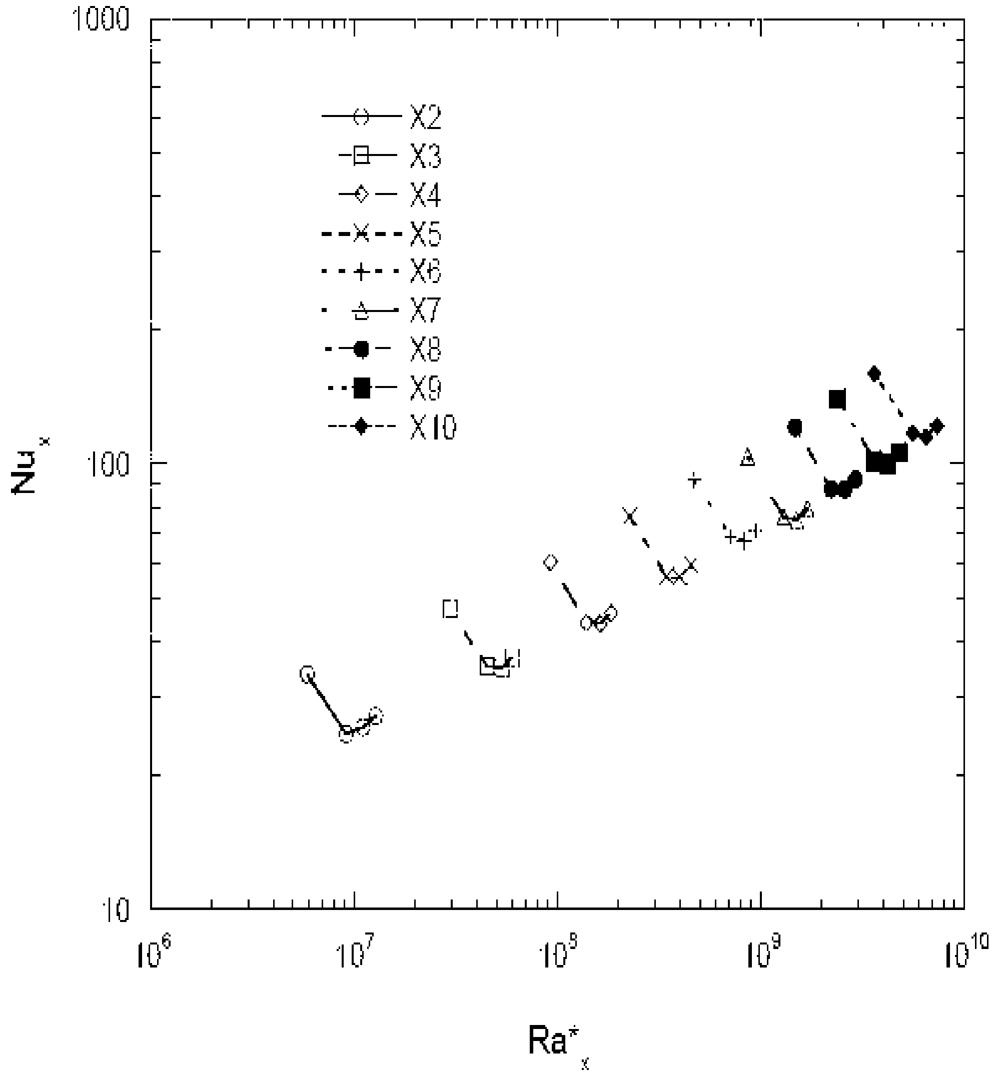


Figure 4.4: The variation of Nusselt number along the cylinder with the variation of the modified Ra at different locations, Suhil [50]

Figure 4.4 shows the distribution of the local Nusselt number with the variation of the modified Nusselt number at different x locations. The data at each x location are obtained by varying the heat flux. This figure shows that at each location when the heat flux is low the value of Nux is large and as the heat flux increases the value of Nux is almost constant. This means the heat transfer characteristics are changed from laminar (at low value of heat fluxes) to turbulent (at high values of heat fluxes).

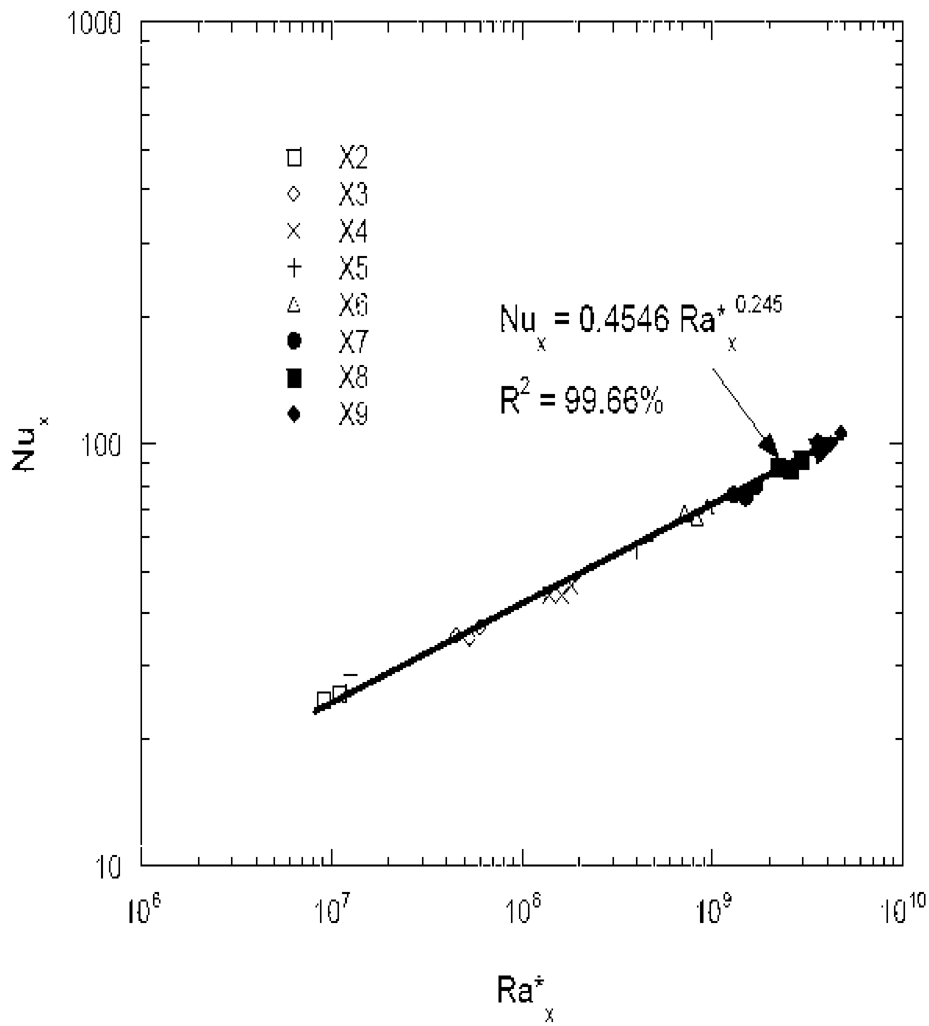


Figure 4.5: The variation of the local Nusselt number with the variation of the modified Ra in the turbulent region (D=50 mm), Suhil [50]

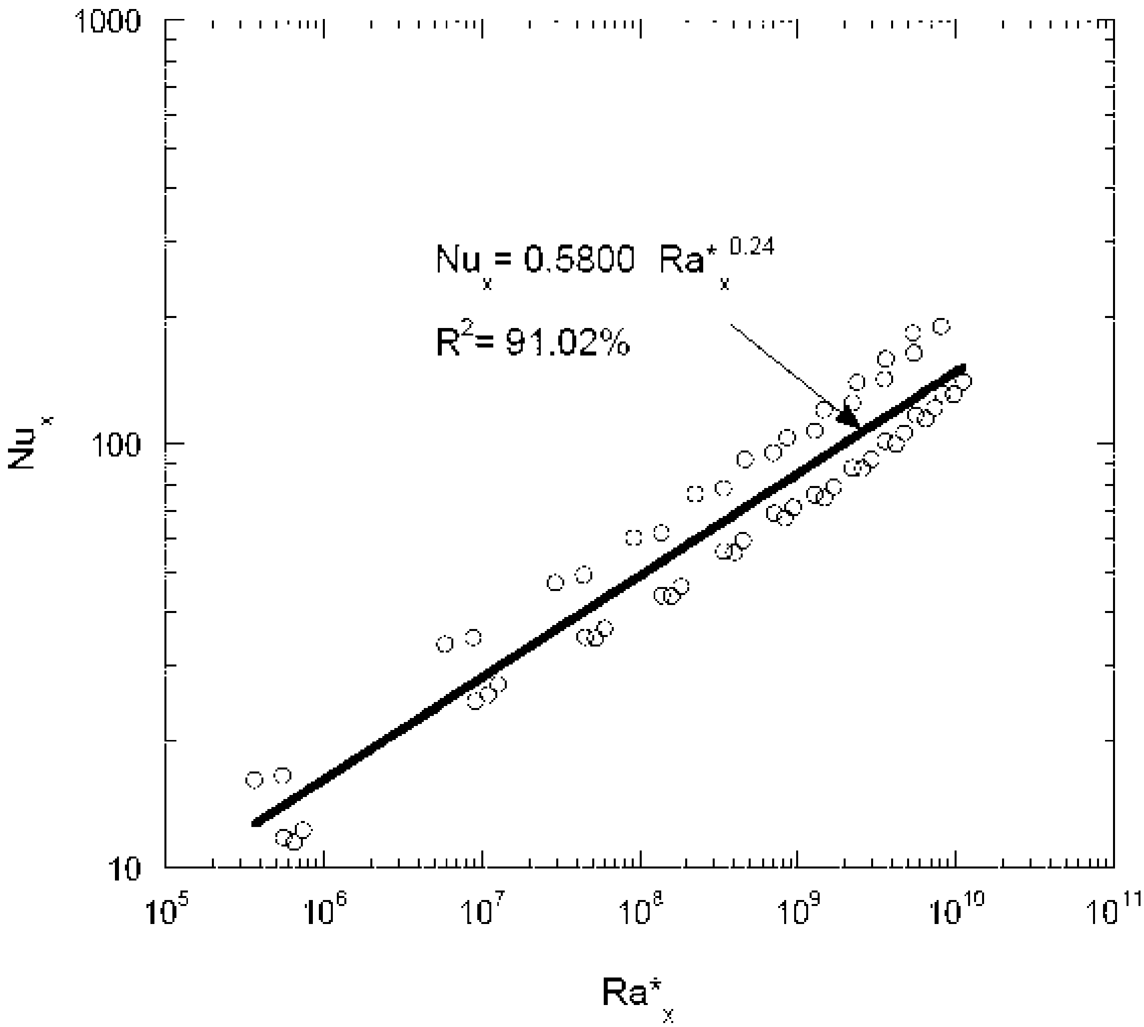


Figure 4.6: The variation of the local Nusselt number with the variation of the modified Ra for all heat fluxes (D=50 mm), Suhil [50]

To separate these regions the data in the laminar region is extracted and the data in the turbulent region is plotted in Figure 4.5, with a correlation coefficient (R-squared) of 96.7%. On the other hand if we correlate the complete data in the range of Ra shown in Figure 14, the following correlation is obtained (see Figure 4.6).

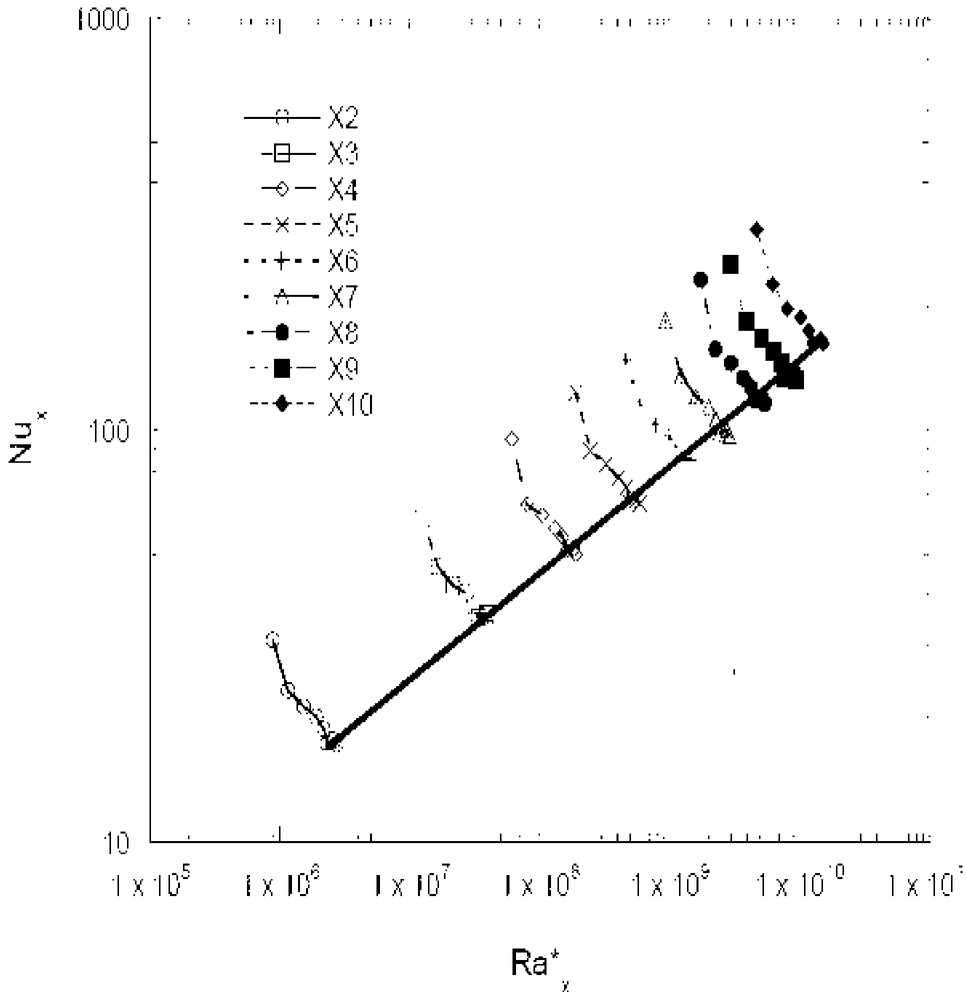


Figure 4.7: The variation of Nusselt number along the cylinder with the variation of the modified Ra at different locations (D=80mm)

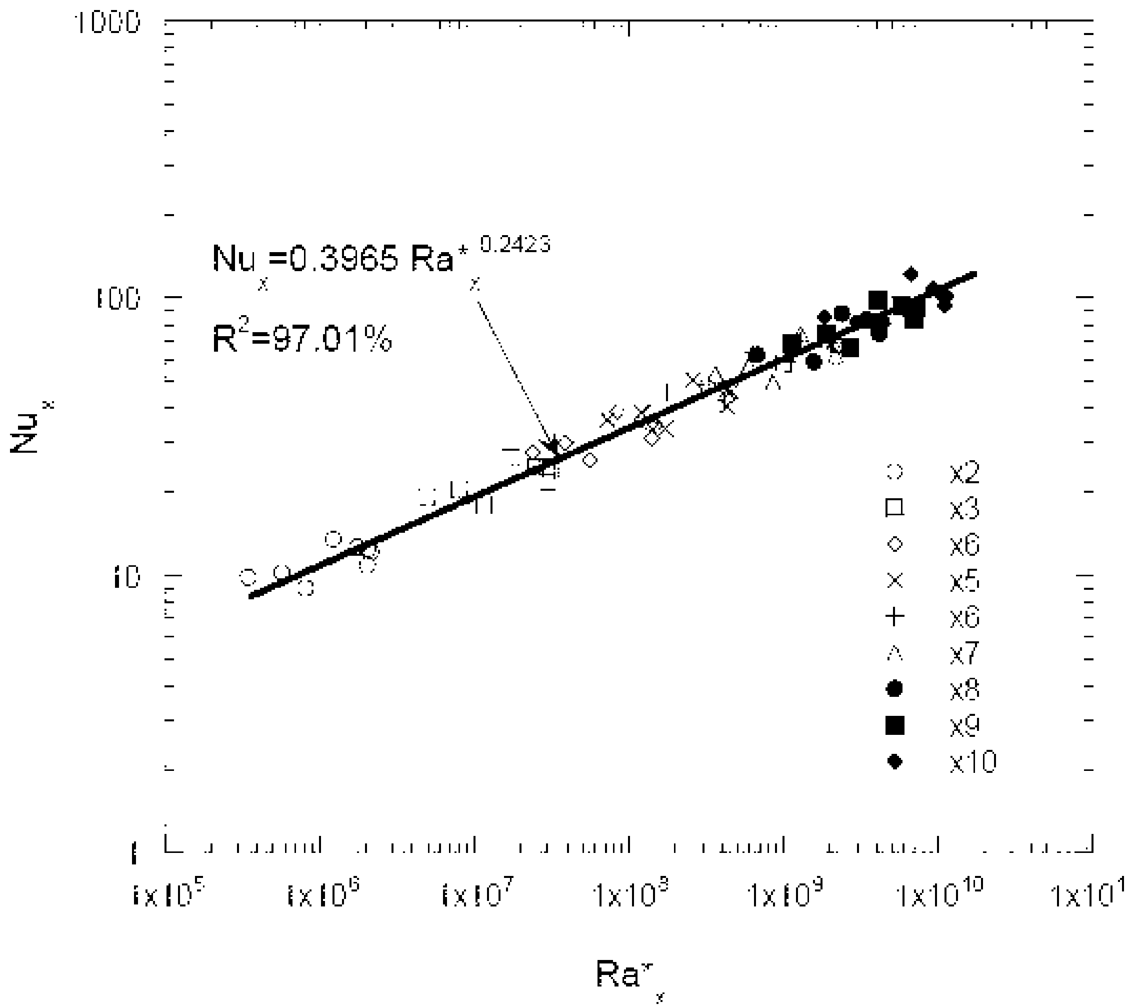


Figure 4.8: The variation of the local Nusselt number with the variation of the modified Ra in the turbulent region at different locations along the cylinder (D=80 mm), Suhil [50]

From Figure 4.7 and Figure 4.8 can see that the different flow which is Figure 4.8 using turbulent flow. There is a different of Nusselt number between that two figures which for Figure 4.7 Nusselt number much more higher than using turbulent flow.

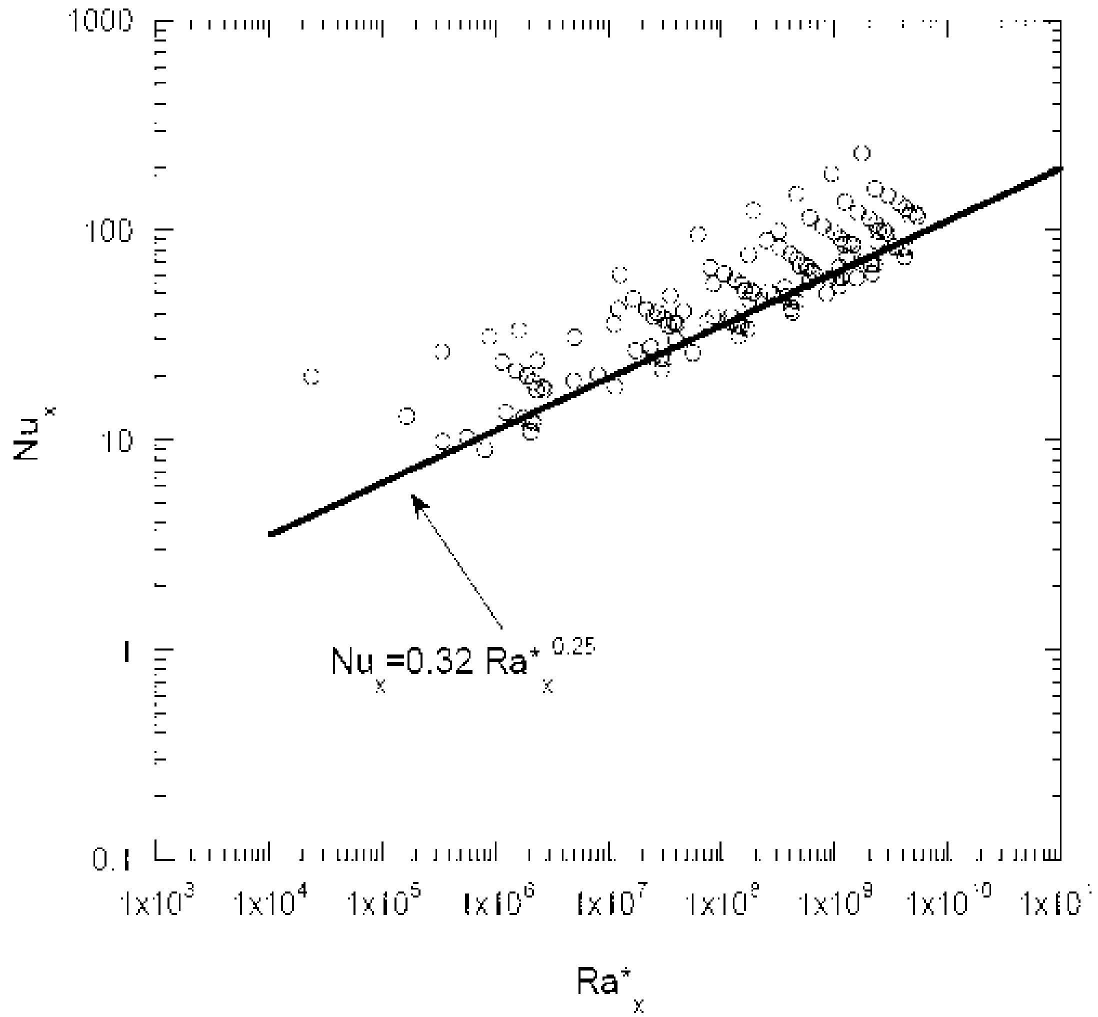


Figure 4.9: The variation of the local Nusselt number with the variation of the modified Ra at all tested locations along the cylinder and for all heat fluxes (D=80 mm),Suhil [50]

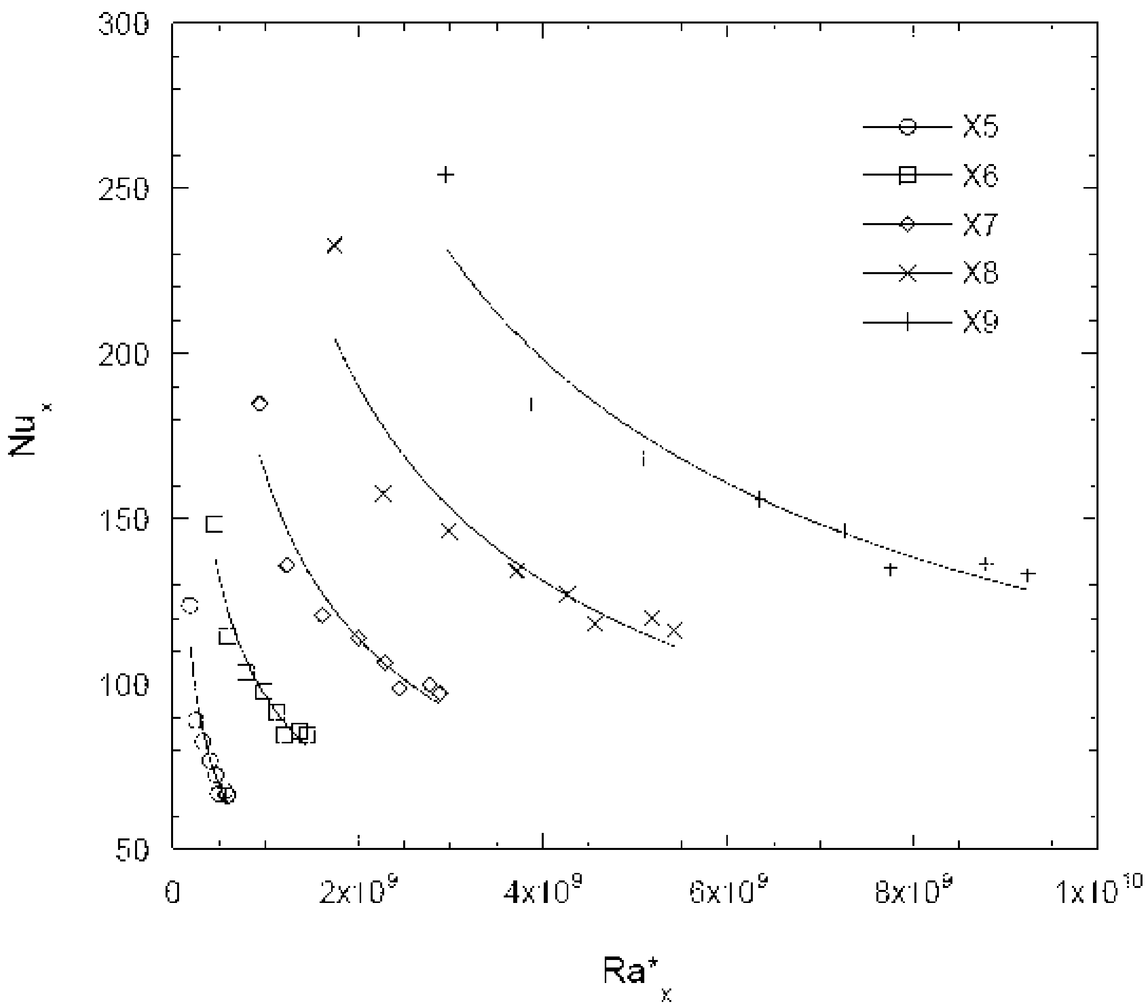


Figure 4.10: The variation of the local Nusselt number with the variation of the modified Ra in the laminar region at selected locations along the cylinder (D=80 mm), Suhil [50]

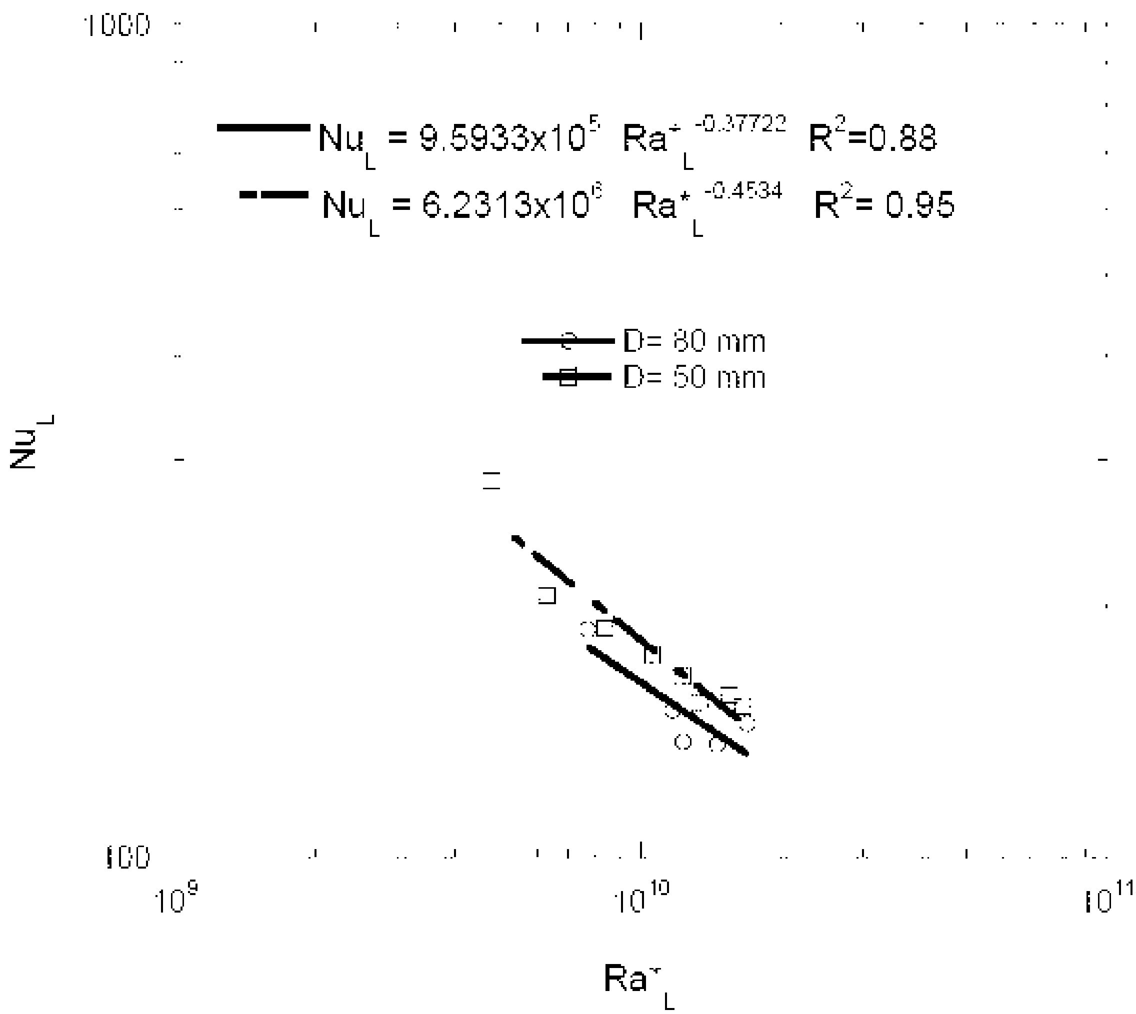


Figure 4.11: The variation of the averaged Nu with the variation of Ra*L for two cylinders, Suhil [50]

Figure 4.10 shows the variation of Nusselt number with the variation of the modified Ra in the laminar region of the 80 mm diameter cylinder at selected locations on the cylinder. It is clear that the value of Nux is a strong function of the heat flux and decreases as the heat flux increases. To compare the results obtained for both cylinders, the values of the averaged NuL calculated using equation (24) are obtained and plotted in Figure 4.11. It is clear from this figure that there is an effect to the cylinder radius on the average heat transfer coefficient and as the radius of the cylinder decreases the average heat transfer coefficient increases.

4.3.2.2 Heat Transfer for Vertical Cylinder with Porous Fins

This is the other one fold that has been studied from Suhil's experiment. The experiments involving porous materials took two tracks.

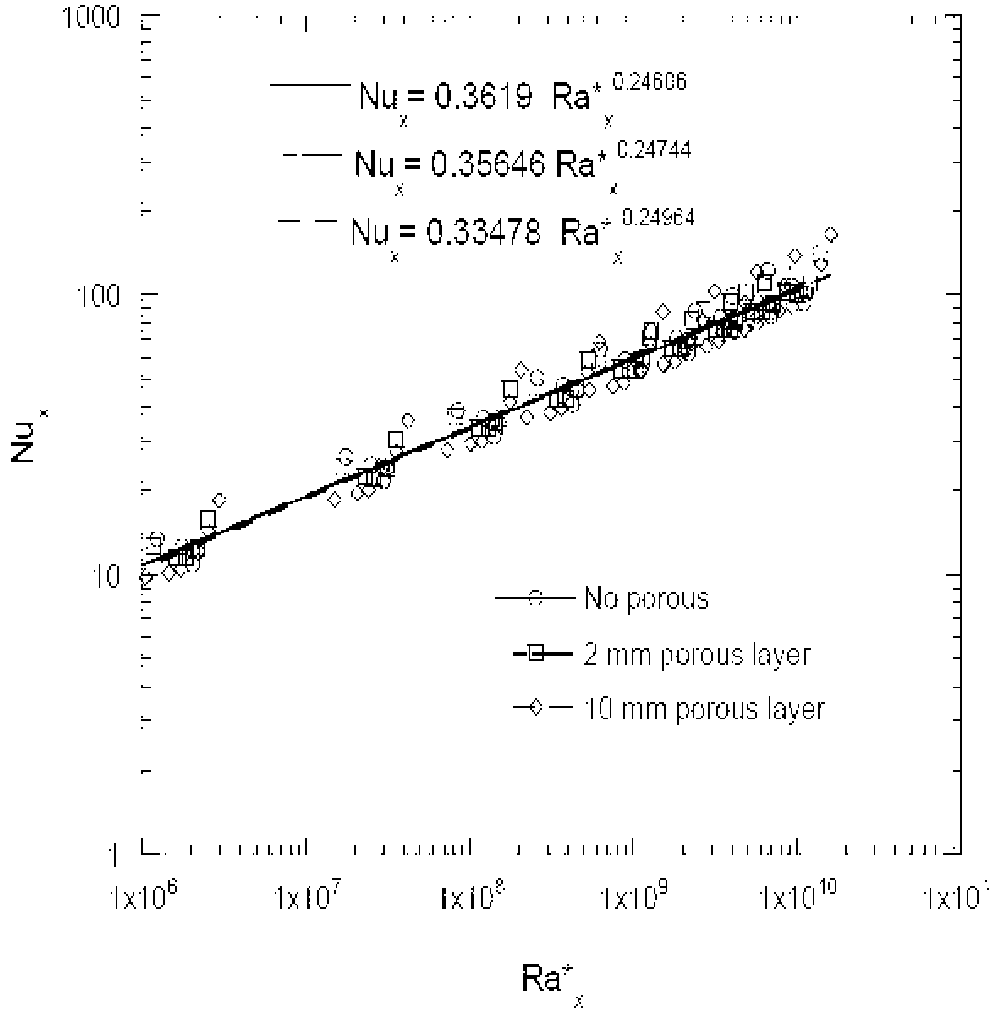


Figure 4.12: A comparison between the variation of the Nux with the variation of Ra*x for the 80 mm cylinder for different porous layers (Porous B), Suhil [50]

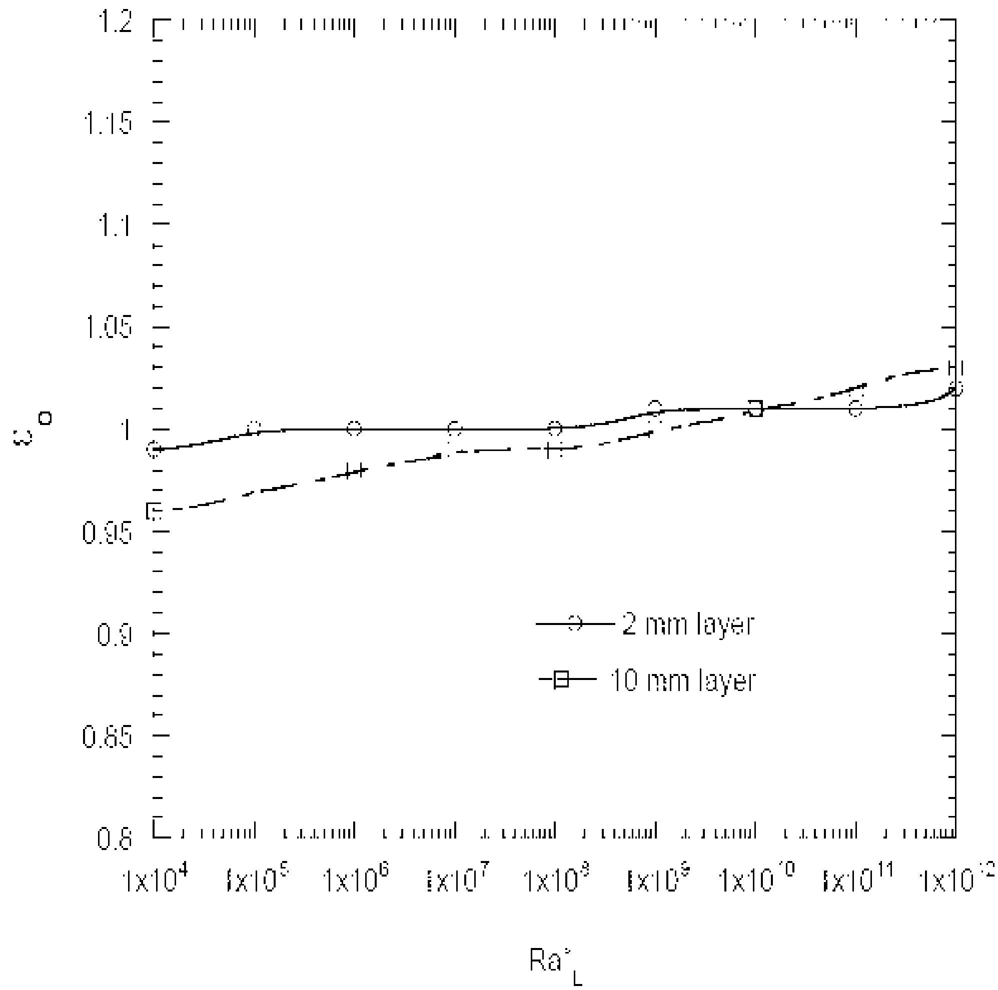


Figure 4.13: Variation of surface effectiveness with the variation of Ra*L for the 80 mm cylinder for different porous layers thicknesses (Porous B), Suhil [50]

For Figure 4.12 show that the different of using porous layer and not using porous in Nusselt number. From figure can see that using 2mm porous layer is much better for Nusselt number. For Figure 4.13, shown that the surface effectiveness by using difference porous layer.

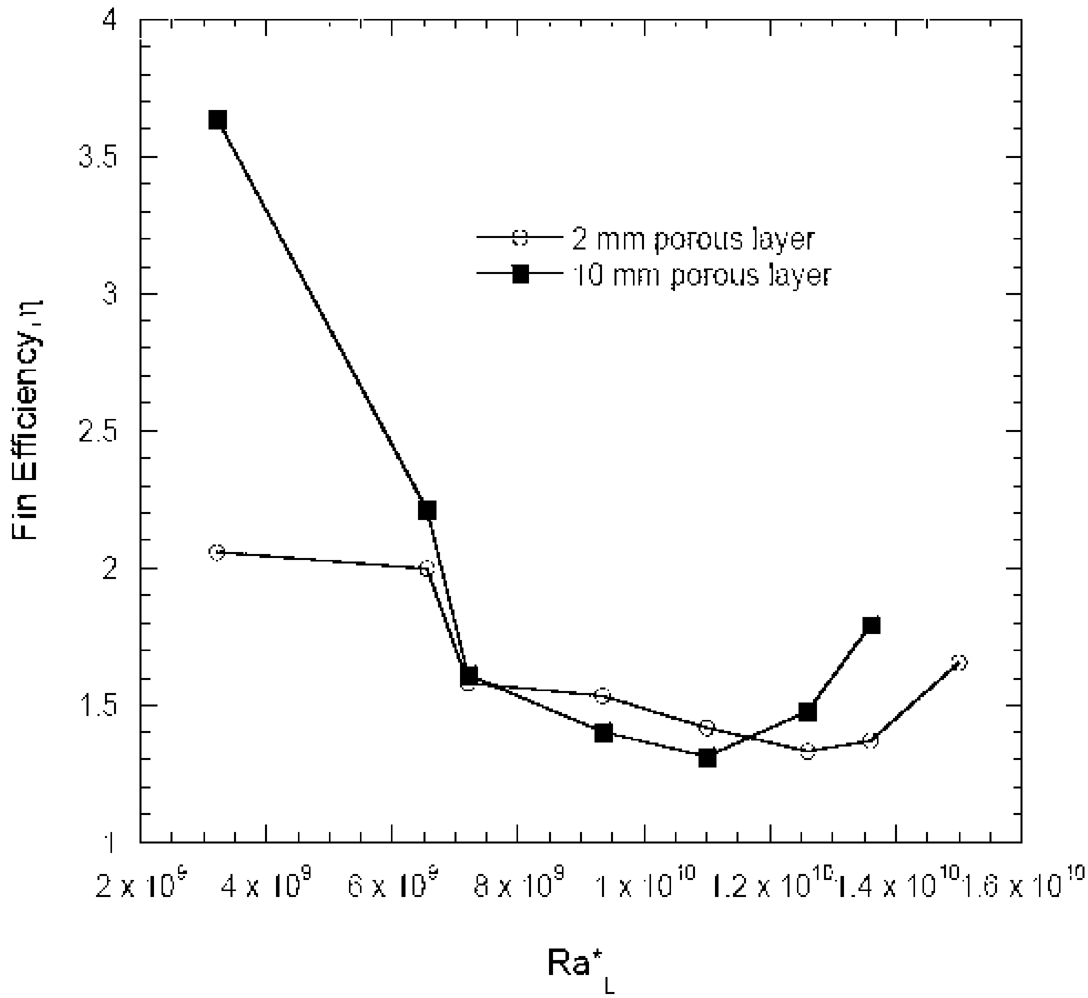


Figure 4.14: The fin efficiency for the porous layers covering the 80 mm vertical cylinder, Suhil [50]

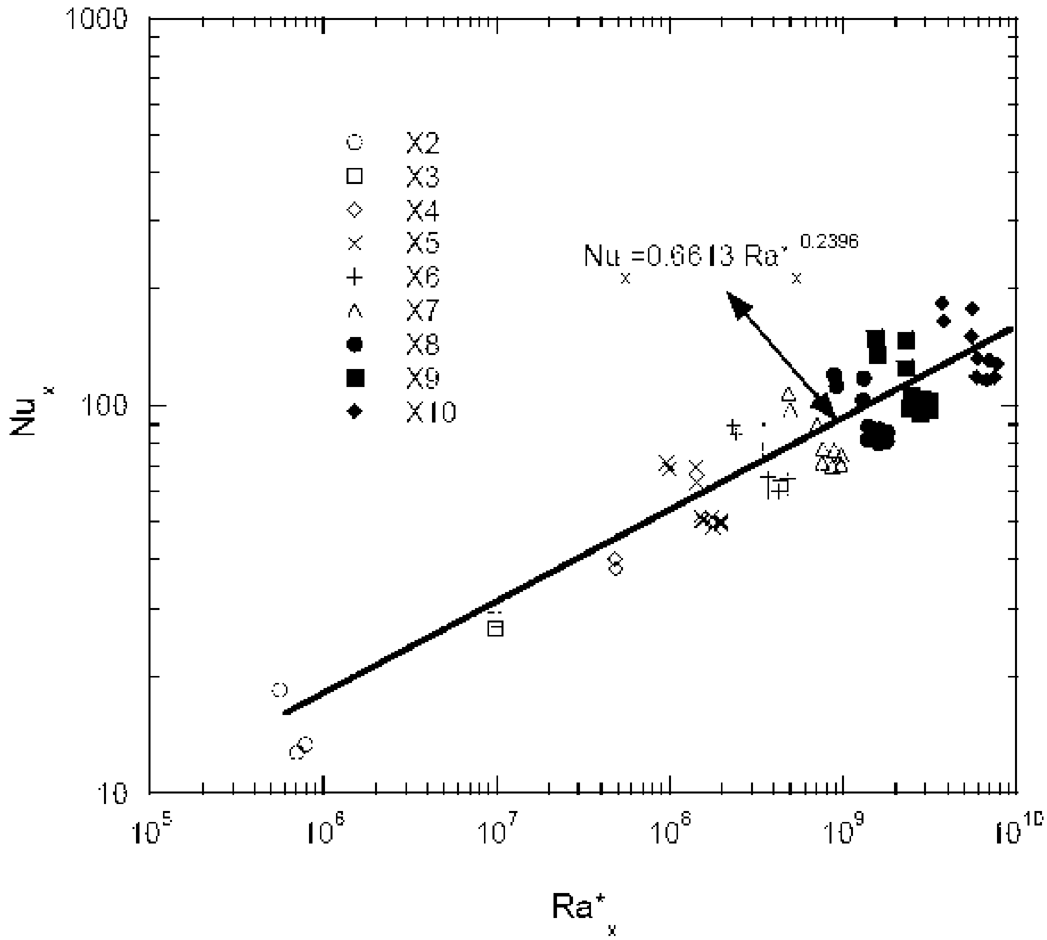


Figure 4.15: The variation of the local Nusselt number with the variation of the modified Ra at all locations for single short fin (D=50 mm, Porous A), Suhil [50]

By comparing the correlations given Figure 4.9 and Figure 4.15, it is clear that using one single fin gives better heat transfer than using a single porous layer covering the complete cylinder. However, this enhancement is still smaller than the anticipated one.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 INTRODUCTION

An experimental study of natural convection heat transfer in a vertical internally finned tube is carried out. There are two different condition has been investigate which is cylinder tube without fins and cylinder tube with fins.

Since, the experiment for this project is not really followed the previous experiment, it is concluded that in this experiment, cylinder tube with fins have much better heat transfer than without tube.

5.2 CONCLUSION

The conclusion for this experiment is heat transfer without fins should be lower than cylinder tube with fins. The heat flux and Nusselt number also should be lower than cylinder tube with fins.

The conclusion from the experiment by Suhil, he has investigated two different cylinder diameters. It is found that when no fins are used as the cylinder diameter decreases the heat transfer increases. Also, for the same case it is found that the value of the averaged Nu number is a strong function of q”. At any location along the cylinder the heat transfer regime changes from laminar to turbulent as *q*” increases regardless of the value of the modified Nu. On the other hand, it is found that when the porous fins are used no noticeable enhancement on the heat transfer is observed. The reason for that could be that the porous material used here has a low value of the permeability, K, and, therefore, increases the resistance to the flow which reduces the flow rate of the fluid passing near the cylinder which in turn reduces the heat transferred from the cylinder.

Finally, Suhil has conclude that, the role of attaching porous fins to a vertical cylinder, further experimentations are needed for porous materials having high value of the permeability, K.

5.3 RECOMMENDATION

5.3.1 Material

The material that has being using in this experiment is mild steel only. Next experimental can use other material such as aluminum to compare with mild steel material. Aluminum - in its metallic form does not exist naturally. It is found only in combination with other minerals in the form of silicate and oxide compounds which make up about 8 per cent of the earth's crust. Aluminum is the third most common crustal element and the most common crustal metal on earth. These mineral compounds are very stable and it took many years of research to find a way to remove the metal from the ore minerals in which it is found.

Aluminum can be very strong, light (less than one third the specific gravity of steel, copper or brass), ductile, and malleable. It is an excellent conductor of heat and electricity. Polished aluminum has the highest reflectivity of any material - even mirror glass. It can be cast, rolled or extruded into an infinite variety of shapes. It has unique barrier properties as a packaging material, it resists corrosion and it can be recycled Ð again and again and again, with no loss of quality or properties. Mixed with small, often minute, quantities of other materials such as iron, silicon, zinc, copper, magnesium, tin, titamium, lithium,chromium, tungsten, manganese, nickel, zirconium and boron, it is possible to produce an array of alloys with very different physical properties.

The main negative aspect to its use is the great amount of energy that is needed in order to refine it from its common ore, bauxite. The recycling of Aluminum scrap metal saves over ninety percent of the energy necessary to divide Aluminum from bauxite. Aluminum is only about one third as dense as iron, but some of its alloys, such as duraluminum are as strong as mild steel. Duraluminum is formed from 94.3 percent Aluminum, 4 percent copper, 0.5 percent manganese, 0.5 percent magnesium, and 0.7 percent silicon. Even though it is a great deal stronger than pure Aluminum, this alloy is not as resistant to corrosion and is frequently clad with pure Aluminum.

Due to its lightness and strength, Aluminum is used a lot in the construction of aircraft. It has high electrical conductivity, 80 percent of which consists of copper, and is used instead of copper in huge electrical conductors. Its disadvantage there is its propensity to become oxidized at contacts, and has need of the use of contacts coated with an antioxidant. Aluminum is a very active metal, and something that cannot frequently be seen is that it quickly creates a barrier oxide layer on uncovered surfaces, holding back its contact. When Aluminum is very strongly heated it burns very rapidly in air, and when it is in the form of a fine dust it can also be explosive.

5.3.2 Experiment Setup and Apparatus

The recommendation for next experiment is the experiment setup. Setup the experiment due to the previous experiment. It is using standard methods of experiment of natural convection heat transfer. An apparatus for the experiment also used the previous experiment such as using a electricity, portable thermocouple that can put inside the tube.

Besides that, the cylinder tube and fins have to connect to the tube surface during the fabrication. Such as using a wire cut to make fins that connected to the body of the cylinder. The next experiment, can make much longer of the length for cylinder tube and make fins more variable. Such using more difference of thickness of fins.

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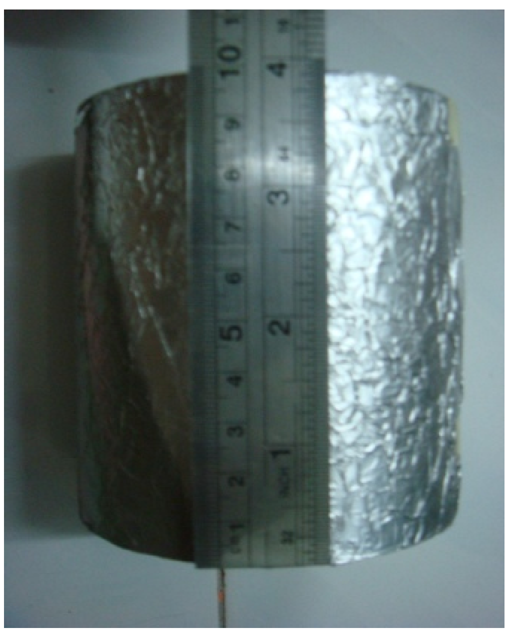
APPENDIXES

APPENDIX A

Measurement of Finned Tube

Fins Measurement



Tube Measurement

Diameter in and out measurement

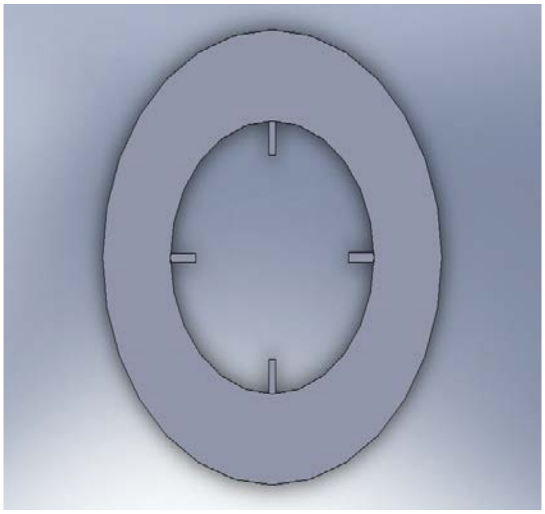
APPENDIX B

Finned Tube Cylinder

Cylinder Tube

Finned Tube

Cylinder Tube using Solid-Work



Finned Tube using Solid-Work

APPENDIX C

The Way to Measure Temperature

Measure fin using Laser Thermometer

Measure Tube Su
Thermo

Measure inside Tube using Digital Thermocouple

Laser Thermometer and Digital Thermocouple

APPENDIX D

GANTT CHART FOR UNDERGRADUATE RESEARCH PROJECT 1

Work progress	W 1	W 2	W 3	W 4	W 5	W 6	W 7	W 8	W 9	W 10	W 11	W 12	W 13	W 14	W 15	W 16
Discussion on assigned research title																
Briefing on undergraduate PSM 1																
Preparation if 1^{st} draft																
Submission of 1^{st} draft																
Submission proposal																
Correction and preparation final draft																
Submission of final draft, evaluate of final draft																
Return back final draft, correction and preparation for PSM 2																
Present to panel																

GANTT CHART FOR UNDERGRADUATE RESEARCH PROJECT 2

Work progress	W 1	W 2	W 3	W 4	W 5	W 6	W 7	W 8	W 9	W 10	W 11	W 12	W 13	W 14	W 15	W 16
Discussion about the method																
Discuss again about expected result																
Prepare the material and apparatus																
Fabricate the cylinder finned tube																
Run the experiment																
Discuss the result with supervisor																
Prepare the final draft report																
Submission of final draft, evaluate of final draft																
Correction and preparation final draft																
Presentation																
Submit the final report of thesis																