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

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Thesis submitted in fulfilment of the requirements for the award of the degree of Bachelor of Mechanical Engineering with Automotive Engineering

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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”
ACKNOWLEDGEMENT

بسم الله الرحمن الرحيم

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ABSTRACT

This paper presents 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. Using material called mild steel and its fins are assembled 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 used 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, further experimentations are needed by using aluminum for the cylinder tube material and different length of cylinder tube, 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 sirip 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 aloi 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.

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 use 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 given size heat exchanger. This goal can be achieved 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).

![The dimension of internally finned tube](image)

**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:

\[ N u = C R_a c_2 a c_3 \]
Where, I need to find ad calculate the coefficient of $C_1, C_2$ and $d_3$.

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.

\[
\frac{\partial}{\partial r}(r u) + \frac{\partial}{\partial x}(r v) = 0 \tag{1}
\]

\[
u = \frac{K}{\mu} \left( \frac{\partial p}{\partial x} + g \right) \tag{2}
\]

\[
v = \frac{K}{\mu} \frac{\partial p}{\partial r} \tag{3}
\]
\[ u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial r} = \alpha \left[ \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial r^2} \right] \] \hspace{1cm} (4)

The boundary condition for this problem is:

\[ \begin{align*}
v &= 0, T &= T_a \quad r = R \\
u &\to 0, T &\to T_a \quad r \to \infty
\end{align*} \hspace{1cm} (5) \]

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:

\[ \frac{q_c}{q} = \frac{\left[ -\frac{\partial (\xi \cdot \lambda)}{\partial \lambda} \right]}{\left[ -\frac{\partial (0 \cdot \lambda)}{\partial \lambda} \right]} \] \hspace{1cm} (7)

Where the curvature parameter \( q_c \) and \( q \) are defined as:

\[ \xi = \frac{2x}{R \left( R_{\alpha x} \right)^{1/2}} \] \hspace{1cm} (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 be calculated using:

\[ Nu = 0.565 R_{Da}^{1/2} \]  

(9)

Where:

\[ Nu = \frac{hD}{k} \]

\[ R_{Da} = \frac{g d K (T - T_e)}{v \alpha_m} \]

\[ \alpha_m = \text{thermal diffusivity for porous media} \]

\[ h = \text{average heat transfer coefficient} \]

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 \( 0.001 < R e_m a x < a \)

\[ Nu = \mu^0.87 \pm 0.618 R_{Da}^{0.69} + 8.54 \times 10^6 G e G e c \left( \frac{D}{D_{Da}} \right)^{0.17} \] for \( 0.001 < R e_m a x \) \( \leq 3 \) (10)

\[ Nu = \mu^0.87 \pm 0.766 R_{Da}^{0.3} \left( \frac{C_1}{C_2} \right)^{0.17} \] for \( 3 < R e_m a x \leq 100 \) (11)

Where:

\[ Nu = \frac{hD}{k} \]

\[ G e = \frac{g D}{C_p} \]

\( C_1 \) 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 $0^\circ \leq \gamma \leq 15^\circ$ 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 $Pr = 0.7$ 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 $8 \leq N \leq 54$, $0.015 \leq P_r \leq 0.1$ and $0^\circ \leq \gamma \leq 15^\circ$. Friction factors and Nusselt numbers were measured in the range of $7,000 \leq Re \leq 70,000$, $0.62 \leq \mu_w \leq 1.35$ and $3.5 \leq P_r \leq 5.0$ 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, Vlankancic’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”.