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Experimental Investigation Of Heat Transfer Enhancement From Dimpled Pin Fin

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Abstract - Heat transfer enhancement over surface results from the depression forming recesses rather than projections. Generically, such features are known as dimples, and may be formed in an infinite variation of geometries which results in various heats transfer and friction characteristics. Heat Transfer enhancement using dimples based on the principle of scrubbing action of cooling fluid taking place inside the dimple and phenomenon of intensifying the delay of flow separation over the surface. Spherical indentations or dimples have shown good heat transfer characteristics when used as surface roughness. The technology using dimples recently attracted interest due to the substantial heat transfer augmentations it induces, with pressure drop penalties smaller than with other types of heat augmentation. The proposed work is concerned with experimental set up for enhancement of the forced convection heat transfer over the dimpled surface and without dimple surface. The objective of the present work is to find out the heat transfer coefficient, Reynolds number, nusselt number, efficiency of fin on dimpled surfaces and all the results obtained will be compared with those from a without dimple surface of same material surface.

Keywords: Dimpled pin fin, Experimental investigation, Heat transfer, etc.

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1. INTRODUCTION

1.1 INTRODUCTION TO HEAT TRANSFER

For well over a century, efforts have been made to produce more efficient heat exchangers by employing various methods of heat transfer enhancement. The study of enhanced heat transfer has gained serious momentum during recent years, however, due to increased demands by industry for heat exchange equipment that is less expensive to build and operate than standard heat exchange devices. Savings in materials and energy use also provide strong motivation for the development of improved methods of enhancement. When designing cooling systems for automobiles and spacecraft, it is imperative that the heat exchangers are especially compact and lightweight. Also, enhancement devices are necessary for the high heat duty exchangers found in power plants (i.e. air-cooled condensers, nuclear fuel rods).These applications, as well as numerous others, have led to the development of various enhanced heat transfer surfaces. The latter is particularly useful in thermal processing of biochemical, food, plastic, and pharmaceutical media, to avoid thermal degradation of the end product. On the other hand, heat exchange systems in spacecraft, electronic devices, and medical applications, for example, may rely primarily on enhanced thermal performance for their successful operation. The commercialization of enhancement techniques, where the technology has been transferred from the research laboratory to full-scale industrial use of those that are more effective and, has also led to a larger number of patents. The conversion, utilization, and recovery of energy in every industrial, commercial, and domestic application involve a heat transfer process. Some common examples are coming from domestic application to industrial ones. Improved heat exchange, over and above that in the usual or standard practice, can significantly improve the thermal efficiency in such applications as well as the economics of their design and operation. The engineering cognizance of the need to increase the thermal performance of heat based equipment's, thereby effecting energy, material, and cost savings as well as a consequential mitigation of environmental degradation had led to the development and use of many heat transfer enhancement techniques. These methods have in the past been referred to variously as augmentation and intensification, among other terms. In general, enhanced heat transfer surfaces can be used for three purposes: to make heat exchangers more compact in order to reduce their overall volume, and possibly their cost, to reduce the pumping power required for a given heat transfer process, or to increase the overall UA value of the heat ex changer. A higher UA value can be exploited in either of two ways:
(1) To obtain an increased heat exchange rate for fixed fluid inlet temperatures,
(2) To reduce the mean temperature difference for the heat exchange; this increases the thermodynamic process efficiency, which can result in a saving of operating costs. Enhancement techniques can be separated into two categories: passive and active. Passive Methods require no direct application of external power. Instead, passive techniques employ Special surface geometries or fluid additives which cause heat transfer enhancement. On the other hand, active schemes such as electromagnetic fields and surface vibration do require external power for operation. The majority of commercially interesting enhancement techniques are passive ones. Active techniques have attracted little commercial interest because of the costs involved, and the problems that are associated with vibration or acoustic noise. This paper deals only with gas-side heat transfer enhancement using special surface geometries. Special surface geometries provide enhancement by establishing a higher hA per unit base surface area.One method to increase the convective heat transfer is to manage the growth of the thermal boundary layer. The thermal boundary layer can be made thinner or partially broken by flow disturbance. Disruption of the laminar sub layer in the turbulent boundary layer is one of a particularly important heat transfer mechanism for augmenting heat transfer. The disruption can be obtained by using rough wall surfaces. In recent years, the concept of using an indented (dimpled) surface instead of protruding devices has gained attention

because of the combination of high heat transfer enhancement and a lower pressure loss penalty.The science and engineering of air-side heat transfer plays a critical role in the
design of compact heat exchangers. Typically, air-side thermal resistance constitutes about 80% of the total thermal resistance to heat flow. Commonly, densely packed fins
are used to increase the air-side surface area and also play the dual role of increasing the
heat transfer coefficient. This is accomplished by using various topologies such that the
thermal boundary layer is constantly regenerated either by interrupted surfaces and/or inducing self-sustained flow oscillations. Fins can be broadly categorized into continuous
surfaces, e.g. wavy fins and ribbed channels, and interrupted fins, e.g. strip fins and louvered fins. An additional aspect which any design has to be sensitive to is the friction penalty of achieving enhanced heat transfer. Hence, surface topologies which maximize heat transfer augmentation with minimal friction penalty are sought.The importance of heat transfer enhancement has gained greater significance in such areas as microelectronic cooling, especially in central processing units, macro and micro scale heat exchangers, gas turbine internal airfoil cooling, fuel elements of nuclear power plants, and bio medical devices. A tremendous amount of effort has been devoted to developing new methods to increase heat transfer from fined surface to the surrounding flowing fluid. Rib tabulator, an array of pin fins, and dimples have been employed for this purpose. In case of the electronics industry, due to the demand for smaller and more powerful products, power densities of electronic components have increased. The maximum temperature of the component is one of the main factors that control the reliability of electronic products. Thermal management has always been one of the main issues in the electronics industry, and its importance will grow in coming decades.

1.2 CLASIFICATION OF ENHANCEMENT TECHNIQUES

Heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib tabulators, protrusions, pin fins, and dimples. These heat transfer enhancement techniques have practical. Application for internal cooling of turbine air foils, combustion chamber liners and electronics cooling devices, biomedical devices and heat exchangers. The heat transfer can be increased by the following different Augmentation Techniques. They are broadly classified into three different categories:
(i) Passive Techniques
(ii) Active Techniques
(iii) Compound Techniques.

A. Passive Techniques These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power. These techniques do not require any direct input of external power; rather they use it from the system itself which ultimately leads to an increase in fluid pressure drop. They generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour except for extended surfaces.

B. Active Techniques These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases. In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer..

C. Compound Techniques A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger. When any two or more of these techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement. This technique involves complex design and hence has limited applications.

Rapid heat removal from heated surfaces and reducing material weight and cost become a major task for design of heat exchanger equipment’s like Cooling of I C engines. Development of super heat exchangers requires fabrication of efficient techniques to exchange great amount of heat between surface such as extended surface and ambient fluid. The present paper reports, an experimental study to investigate the heat transfer enhancement in rectangular fin arrays defined porosity equipped on horizontal flat surface in horizontal rectangular duct. The data used in performance analyses were obtained experimentally by varying PPI, different material inputs and fluids. The present work relates with the investigation of parameter of the fin to be optimized also the work elaborated with the help of CFD analysis for better results in terms of various charts those reflect the work necessity. Pin fins have a variety of applications in industry due to their excellent heat transfer performance, e.g., in cooling of electronic components, in cooling of gas turbine blades, and recently, in hot water boilers of central heating systems, etc. . In the two early studies by Sahiti et al., it was demonstrated that pin fin arrays offer the most effective way of enhancing the heat transfer rate within a particular heat exchanger volume. However, the pressure drops in such heat exchangers are usually much higher than those in others ; this defect greatly lowers the overall heat transfer performances of pin fin heat exchangers and as a result, their applications are restricted. In order to reduce the pressure drops and improve the overall heat transfer performances for pin fin heat exchangers, porous metal pin fin arrays may be used instead of traditional solid metal pin fin arrays. As porous media can significantly intensify the mixing of fluid flow and increase the contact surface area with fluid inside, it has been regarded as an effective way to enhance heat transfer by using porous media. The flow and heat transfer in porous pin fin heat exchangers for present study can be modelled as forced convective heat transfer in partially filled porous channels. The researches on forced convection with partially filled porous configurations have been investigated extensively in the last years. Hamid studied the laminar forced convection in a fully or partially filled porous channel containing discrete heat sources on the bottom wall. The Brinkman– Forchheimer extended Darcy model were used for the computations. He found that when the width of the heat source and the space between the porous layers were of same magnitudes as the channel height, the heat transfer enhancement in the partially filled channel was almost the same as that in the fully filled porous channel while the pressure drop was much lower. According to the authors' knowledge, almost no such attentions have been paid on this subject before.

2. LITERATURE SURVEY

Generally, heat transfer augmentation techniques are classified in three broad categories: active methods, passive method and compound method. Following are the some of experimental studies.

Chyu et al. studied the enhancement of surface heat transfer in a channel using two different concavities- hemispheric and tear drop. Concavities serve as vortex generators to promote turbulent mixing in the bulk flow to enhance the heat transfer at Re H = 10,000 to 50,000, H/d of 0.5, 1.5, 3.0 and δ/d =0.575. Heat transfer enhancement was 2.5 times higher than smooth channel values and with very low pressure losses that were almost half that caused by conventional ribs turbulators.

Moon et al. experimentally studied the effect of channel height on heat transfer performance and friction losses in a rectangular dimpled passage with staggered dimples on one wall. The geometry used was H/D = 0.37, 0.74, 1.11, 1.49 and ReH = 12,000 to 60,000. Heat transfer enhancement was roughly 2.1 times greater than the smooth channel configuration with H/D values from 0.37 to 1.49. The heat transfer augmentation was invariant with the Reynolds number and channel height. The increase in friction factor was 1.6 to 2.0 times less than the smooth channel. The pressure losses also remained approximately constant for the channel height.

Mahmood et al studied the flow and heat transfer characteristics over staggered arrays of dimples with δ/D=0.2. For the globally average Nusselt number, there were small changes with Reynolds number. Ligrani et al studied the effect of dimpled protrusions (bumps) on the opposite wall of the dimpled surface.

Mahmood et al experimentally showed the influence of dimple aspect ratio, temperature ratio, Reynolds number and flow structures in a dimpled channel at Re H = 600 to 11,000 and air inlet stagnation temperature ratio of 0.78 to 0.94 with H/D =0.20, 0.25, 0.5, 1.00. The results indicated that the vortex pairs which are periodically shed from the dimples become stronger and local Nusselt number increase as channel height decreases. As the temperature ratio Toi/Tw decreases, the local Nusselt number also increased.

Burgess et al experimentally analyzed the effect of dimple depth on the surface within a channel with the ratio of dimple depth to dimple printed diameter, equal to δ/D, 0.1, 0.2, and 0.3. The data showed that the local Nusselt number increased as the dimple depth increased due to an increased strength and intensity of vortices and three dimensional (3D) turbulent productions. Ligrani et al studied the effect of inlet turbulence level in the heat transfer improvement in walls with dimple ratio δ/D=0.1, showing that as the turbulence level was increased, the relative Nusselt number reduced due to the increased turbulent diffusion of vorticity.

Bunker et al use circular pipes with in-line arrays of dimples with δ/D=0.2 and 0.4 and surface area densities ranging from 0.3 to 0.7 to provide heat transfer enhancement and friction effects of dimples in a circular tube. Reynolds numbers for bulk flow were from 20,000 to 90,000. Heat transfer enhancement was 2times greater than the smooth circular tube in the case where the relative dimple depth was greater than 0.3 and array density was greater than 0.5. An increase of friction factor was approximately 4 to 6 times greater than the smooth circular tube, which was better than the rib turbulartors. Han studied the rotational effect of dimples in turbine blade cooling. In the case where a pressure drop is the main design concern, dimple cooling can be a good choice. Jet impingement over a convex dimpled surface was studied by Chang et al. showing an incremental increase in the relative Nusselt number (Nu/Nu0) up to

Pin- Fins are used to increase the heat transfer rate from surface to the surrounding fluid when 'h' value is generally smaller on the surface. Familiar examples are the circumferential fins around the cylinder of motor cycle engine & pin fins attached to the condenser tubes at the back of domestic refrigerator. In present pin-fins are normally used in different shapes & sizes depending upon its applications as shown in following figures. It is obvious that a fin surface sticks out from the primary heat transfer surface. The temperature difference with surrounding fluid will steadily diminish as one moves out along the fin. The design of the fins therefore required knowledge of the temperature distribution in the fin. The main objective of this experimental set up is to study temperature distribution in a simple pin fin.

Widespread application of compact heat exchangers in automotive, HVAC and refrigeration industries has necessitated the use of various ways to augment the air side heat transfer capacity. Interrupted surface is a way to enhance the heat transfer by reforming the thermal boundary layer. One of the most widely used designs among interrupted surfaces is the multilouvered geometry. During past decades many experimental and numerical studies have been conducted to demonstrate the flow characteristics . Beauvias was the first to perform flow visualization experiments on the louvered fin array. He showed that rather louvers actually acted as guides to redirect the flow in between them. Flow visualization performed by Davenport demonstrated two asymptotic flow regimes; duct directed flow, and louver directed flow. In the former the predominant flow is stream wise while in the latter the predominant flow is aligned with the louvers. Zhang and Tafti showed that the flow direction has significant implications on the overall heat capacity of the fin because of its strong effect on the heat transfer coefficient. Therefore it is important to be able to quantify the flow regime. In order to do so a parameter called "flow efficiency" is defined as the degree to which the flow is aligned to the louver direction . A 100% efficiency represents complete louver directed flow while 0% represents complete duct flow.

The importance of predominant flow directions was soon discovered in early multilouver studies . Zhang and Tafti showed that high flow efficiency enhances the heat transfer coefficient. Flow efficiency in general, is a function of Reynolds number and geometrical parameters like fin pitch, louver angle and louver thickness ratio [4,6]. Consequently a critical Reynolds number, after which the flow efficiency is independent of the louver angle, was identified. Achaichia et al studied the flow pattern along with k-epsilon turbulent model for high Reynolds numbers . However, high Reynolds turbulent models are not very accurate in unsteady laminar and low Reynolds number turbulence regimes encountered in compact heat exchangers. Zhang et. al and Tafti et al have studied the effects of flow oscillations in the form of large-scale vorticity on the heat transfer coefficient. Tafti investigated the transition from steady laminar to unsteady flow in a multilouvered fin array. The local heat transfer from the fin surface was found to be strongly affected by the large-scale vortices shed from the louver. Zhang and Tafti studied the effects of Intrafin and Inter-fin thermal wakes on the heat transfer capacity of the fin array. They suggested that thermal wake effects can be expressed as functions of the flow efficiency and the fin pitch-to-louver pitch ratio . In addition to interrupted surface concept of louvered fins, continuous fins like dimpled surfaces have proven to effectively enhance the heat transfer capacity. The usage of dimpled surfaces has received attention due to their good heat transfer characteristics and low pressure drop penalties compared to interrupted surfaces . One of the early studies was conducted by Afansayev et al. , where the effects of shallow dimples on flat plates on the overall heat transfer capacity and pressure drop were investigated. They reported a significant heat transfer enhancement (3040 %) at a low pressure drop cost. Ligrani et al. experimentally investigated the flow structure in dimpled surfaces and showed the existence of flow recirculation zone in the upstream half of the dimple. A region of low heat transfer was observed in the upstream half of the dimple cavity followed by a high heat transfer region in the flow reattachment region in the downstream half of the dimple and the flat landing downstream of dimple . Wang et al., identified a symmetric 3D horseshow vortex inside a single dimple using laminar flow simulations. Line et al., Isaev and Leonte`ev, Park er al., Won and Ligrani and Park and Ligrani investigated flow structure and heat transfer in dimpled channel in fully turbulent regimes using steady state RANS models.

In a study for a given base temperature, the effect of vacuum, angle of groove and number of thread per inch shows that there exist optimum angle of groove and number of thread per inch for which the heat loss per unit mass is maximum.i.e; heat loss is 1.2 to 1.4 times greater than radiating pin fin. The performance of heat exchanger can be improved by mounting protrusion on the surface. The rectangular winglet mounted on the triangular fins of the plate fin heat exchanger disturbs the flow structure and creates longitudinal vortices. Due to the existence of complex stream wise vortices system in the flow passage, the heat transfer between the fluid and its neighboring surface is significantly enhanced with a moderate pressure drop . It has been found that the grooved radiating fin loses approximately 1.23 times greater heat per unit area compared to the threaded fin. As pressure decreases heat loss reduces and contribution of radiation heat transfer on total heat loss increases . Study in different vortex generator shows that more complicated fin structure will lead to higher pressure drop and the pressure drop for dimple fin geometry is significantly higher than other fin types. The test result of heat transfer coefficient suggested that fin with dimple vortex generator is more beneficial than that of plain fin . Study of dimpled surface with uniform heat flux shows that heat transfer coefficient are relatively low on the leading edge of the dimple and high on the trailing edge and the flat area immediately downstream of the dimple . In numerical optimization of heat exchanger with brazed dimpled surface investigate friction factor and colburn factor according to parameter. The heat transfer and pressure drop increases as the dimpled diameter and channel height increases while they decreases as the dimple spacing and dimple pitch increased. The transverse distance had a greater effect on the enhanced heat transfer than the stream wise distance is shown by study on pin fin array with turbulent air flow . Natural convection heat transfer enhancement from a horizontal rectangular fin embedded with equilateral triangular perforation is investigated and compared with equivalent solid one. The gain in heat dissipation rate for perforated fin has strong relation with perforation dimension and lateral spacing . Mills and Ganesan also gave relation for heat transfer from plate with groove and protrusion using equivalent sand grain roughness and Stanton number . The experimental work was done by J.Nikuradse who used a surface coated with closely packed sand grains of mean diamet. As a consequence, other roughness patterns are often characterized in terms of equivalent sand grain roughness which gives the same friction factor in the fully rough regime. Whereas use of the equivalent sand grain roughness ensures that the correct friction factor is obtained in the fully rough regime.

The forced convective heat transfer in three-dimensional porous pin fin channels is numerically studied in this paper. The Forchheimer–Brinkman extended Darcy model and two-equation energy model are adopted to describe the flow and heat transfer in porous media. Air and water are employed as the cold fluids and the effects of Reynolds number (Re), pore density (PPI) and pin fin form are studied in detail. The results show that, with proper selection of physical parameters, significant heat transfer enhancements and pressure drop reductions can be achieved simultaneously with porous pin fins and the overall heat transfer performances in porous pin fin channels are much better than those in traditional solid pin fin channels. The effects of pore density are significant. As PPI increases, the pressure drops and heat fluxes in porous pin fin channels increase while the overall heat transfer efficiencies decrease and the maximal overall heat transfer efficiencies are obtained at PPI_20 for both air and water cases.Furthermore, the effects of pin fin form are also remarkable. With the same physical parameters, the overall heat transfer efficiencies in the long elliptic porous pin fin channels are the highest while they are the lowest in the short elliptic porous pin fin channels.

Markus R¨utten and Lars Krenkel -- The main physical mechanisms causing the enhancement of heat transfer is the generation and amplification of sufficiently strong longitudinal vortices which are interacting with the thermal boundary layer. The stratification of the thermal boundary layer near the heated walls is disturbed by these vortices. The convection of warmer fluid perpendicular to the heated wall and the mixing with colder fluid is intensified, and, additionally, further external momentum is transported into the inner boundary layer region.

K.S Dhanawade, H. S. Dhanawade -- The present paper reports, an experimental study to investigate the heat transfer enhancement in rectangular fin arrays with circular perforation equipped on horizontal flat surface in horizontal rectangular duct. The data used in performance analyses were obtained experimentally by varying flow, different heat inputs and geometrical conditions. The experiment covered Reynolds number range from 30006000, based on the flow average inlet velocity and hydraulic diameter. Clearance ratio (C/H) 0.45, inter-fin spacing ratio (S/H) 0.22, duct width 150mm, height100mm and fin size of both solid and perforated (weight reduction) were 100mm x 55mm x 3mm. For various heat inputs and flow rates values of

Reynolds and Nusselt number were obtained. The results of perforated fin arrays have been compared with its external dimensionally equivalent solid fin arrays. It shows that enhancement in heat transfer of perforated fin arrays than solid fin arrays.
Yang ,Wun ,Youn Chen,Wang -- Pressure drop and heat transfer characteristics of different dimple vortex generators arrangement are examined in this study. A total of five heat sinks were made and tested, including plain fin, dimple, two-group dimple, oblique dimple gap 4-12 fin, oblique dimple gap6-12 fin.
The tested results indicate that the pressure drop for dimple fin geometry is significantly higher than other fin types, followed by the two group dimple fin. The fin structure with oblique dimple shows slightly increase of pressure drop as compared to the plain fin surface. The results show that more complicated fin structure will lead to higher pressure drop. The observed IR image of the temperature measurement of the test fin configurations also proves the measurements. The results suggest that the fin with dimple vortex generators is more beneficial than that of plain fin geometry. The oblique dimple fin is especially useful for air-cooling applicable for electronic devices to achieve effective augmentations without suffering from significant pressure penalty.
Kore, V. Joshi, K.Sane -- An experimental investigation has been carried out to study heat transfer and friction coefficient by dimpled surface. The aspect ratio of rectangular channel is kept 4:1 and Reynolds number based on hydraulic diameter is varied from 10000 to 40000. The ratios of dimple depth to dimple print diameter is varied from 0.02 to 0.04 to provide information on the influences of dimple depth. The ratio of channel height to print diameter is 0.5. The heat transfer and friction factor data obtained is compared with the data obtained from smooth plate under similar geometric and flow conditions. It is observed that at all Reynolds number as depth increases from 0.2 to0.3, the normalized Nusselt number and thermal performance increases and then after when depth increase from0.3 to 0.4 normalized Nusselt number and thermal performance decreases. These are because of increase in strength and intensity of vortices and associated secondary flows ejected from the dimples.
Johann Turnow et al. Studied Vortex structures and heat transfer enhancement mechanism of turbulent flow over a staggered array of dimples in a narrow channel. It was found that the dimple package with a depth h to diameter D ratio of h/D = 0.26 provides the maximum thermo-hydraulic performance. The heat transfer rate could be enhanced up to 201% compared to a smooth channel . Yu Rao et al. investigate the effects of dimple depth on the pressure loss and heat transfer characteristics in a pin fin-dimple channel, where dimples are located on the end wall transversely between the pin fins. The study showed that, compared to the baseline pin fin channel, the pin fin-dimple channels have further improved convective heat transfer performance by up to 19.0%, and the pin fin-dimple channel with shallower dimples shows relatively lower friction factors by up to 17.6% over the Reynolds number range 8200 to 50,500. C. Bi et al. studied convective cooling heat transfer in mini-channels with dimples, cylindrical grooves and low fins. The results show that the dimple surface presents the highest performance of heat transfer enhancement . Chyu et al. studied the enhancement of surface heat transfer in a channel using two different concavities-hemispheric and tear drop. Concavities serve as vortex generators to promote turbulent mixing in the bulk flow to enhance the heat transfer at Re H = 10,000 to 50,000, H/d of 0.5, 1.5, 3.0 and δ /d =0.575. Heat transfer enhancement was 2.5 times higher than smooth channel values and with very low pressure losses that were almost half that caused by conventional ribs turbulators [13].S.A. Isaev studied Influence of the Reynolds number and the spherical dimple depth on turbulent heat transfer and hydraulic loss in a narrow channel. Detailed information gained from the presented computations can be used to get a deep insight into flow physics over dimpled surfaces and as a benchmark for validation of numerical and experimental methods [4].
Experimental Investigation of Heat Transfer Enhancement over the Dimple Surface under Forced Convection Sagar R Kulkarni1, Dr. R.G.Tikotka -- An experimental study has been conducted to study heat transfer over the dimpled plates under forced convection. The Reynolds number is varied in the range of 10000 to 30000 based on hydraulic diameter. The dimpled depth is varied from 0.2 to 0.4 keeping 0.1 frequency and keeping the dimple density constant in both the arrangements. The dimple print diameter is kept as 10mm where dimple depths are 2mm,3mm,4mm.A constant heat of 0.5 amperes was given. It is noticed that, heat transfer is augmented for a depth of 0..3 in both the arrangements.0.4 depth plate shown very least augmentation. The maximum thermal performance is for staggered arrangement and for 0.3 depth. The maximum thermal performance for inline arrangement is lowest value of thermal performance of staggered plate.
Experimental Investigation of Heat transfer by PIN FIN U S Gawai, Mathew V K, Murtuza S D -- Heat transfer enhancement over surface results from the depression forming recesses rather than projections. Generically, such features are known as dimples, and may be formed in an infinite variation of geometries which results in various heats transfer and friction characteristics. Heat Transfer enhancement using dimples based on the principle of scrubbing action of cooling fluid taking place inside the dimple and phenomenon of intensifying the delay of flow separation over the surface. Spherical indentations or dimples have shown good heat transfer characteristics when used as surface roughness. The technology using dimples recently attracted interest due to the substantial heat transfer augmentations it induces, with pressure drop penalties smaller than with other types of heat augmentation. The proposed work is concerned with experimental set up for enhancement of the forced convection heat transfer over the dimpled surface and flow structure analysis within a dimple. The objective of the present work is to find out the heat transfer rate and air flow distribution on dimpled surfaces and all the results obtained will be compared with those from a flat surface.
Experimental Analysis of Heat Transfer Enhancement Using Fins in Pin Fin Apparatus Allan Harry Richard.T.L --The aim of the present study is to improve the heat transfer characteristics and to investigate the performance of fin efficiency by using fins of different materials in pin fin apparatus. Here the system follows forced convection as the mode of heat transfer and it is the principle used in it. This experiment accomplished by using blower in the riser tube, which connects to the thermocouple which flows the air to the heater. From the heater the air gets heated and the air transfer to the the pin fin in it. This procedure followed for the fin of different materials, Reynolds number, Nusselts numbers are calculated and heat transfer coefficient and fin efficiencies are analyzed.
Investigation of Optimum Porous Pin Fin Parameter for Forced Convective Heat Transfer through Rectangular Channel Part-I Shrikant Vasantrao Bhunte1, Sanjay

3. BASICS OF HEAT TRANSFER

Heat transfer is the exchange of thermal energy between physical systems. The rate of heat transfer is dependent on the temperatures of the systems and the properties of the intervening medium through which the heat is transferred. The three fundamental modes of heat transfer are *conduction*, *convection* and *radiation*. Heat transfer, the flow of energy in the form of heat, is a process by which a system changes its internal energy, hence is of vital use in applications of the First Law of Thermodynamics. Conduction is also known as diffusion, not to be confused with diffusion related to the mixing of constituents of a fluid.

The direction of heat transfer is from a region of high temperature to another region of lower temperature, and is governed by the Second Law of Thermodynamics. Heat transfer changes the internal of the systems from which and to which the energy is transferred. Heat transfer will occur in a direction that increases the entropy of the collection of systems.

Thermal equilibrium is reached when all involved bodies and the surroundings reach the same temperature. Thermal expansion is the tendency of matter to change in volume in response to a change in temperature

3.1 Convection is the transfer of heat by the actual movement of the warmed matter. Heat leaves the coffee cup as the currents of steam and air rise. Convection is the transfer of heat energy in a gas or liquid by movement of currents. (It can also happen is some solids, like sand.) The heat moves with the fluid. Consider this: convection is responsible for making macaroni rise and fall in a pot of heated water. The warmer portions of the water are less dense and therefore, they rise. Meanwhile, the cooler portions of the water fall because they are denser.

3.2 Conduction is the transfer of energy through matter from particle to particle. It is the transfer and distribution of heat energy from atom to atom within a substance. For example, a spoon in a cup of hot soup becomes warmer because the heat from the soup is conducted along the spoon. Conduction is most effective in solids-but it can happen in fluids. Fun fact: Have you ever noticed that metals tend to feel cold? Believe it or not, they are not colder! They only feel colder because they conduct heat away from your hand. You perceive the heat that is leaving your hand as cold.

3.3 Radiation: Electromagnetic waves that directly transport ENERGY through space. Sunlight is a form of radiation that is radiated through space to our planet without the aid of fluids or solids. The energy travels through nothingness! Just think of it! The sun transfers heat through 93 million miles of space. Because there are no solids (like a huge spoon) touching the sun and our planet, conduction is not responsible for bringing heat to Earth. Since there are no fluids (like air and water) in space, convection is not responsible for transferring the heat. Thus, radiation brings heat to our planet.

3.4 Natural convection is a mechanism, or type of heat transport, in which the fluid motion is not generated by any external source (like a pump, fan, suction device, etc.) but only by density differences in the fluid occurring due to temperature gradients. In natural convection, fluid surrounding a heat source receives heat, becomes less dense and rises. The surrounding, cooler fluid then moves to replace it. This cooler fluid is then heated and the process continues, forming a convection current; this process transfers heat energy from the bottom of the convection cell to top. The driving force for natural convection is buoyancy, a result of differences in fluid density. Because of this, the presence of a proper acceleration such as arises from resistance to gravity, or an equivalent force (arising from acceleration, centrifugal force or Coriolis effect), is essential for natural convection. For example, natural convection essentially does not operate in free-fall (inertial) environments, such as that of the orbiting International Space Station, where other heat transfer mechanisms are required to prevent electronic components from overheating.

Natural convection has attracted a great deal of attention from researchers because of its presence both in nature and engineering applications. In nature, convection cells formed from air raising above sunlight-warmed land or water are a major feature of all weather systems. Convection is also seen in the rising plume of hot air from fire, oceanic currents, and sea-wind formation (where upward convection is also modified by Coriolis forces). In engineering applications, convection is commonly visualized in the formation of microstructures during the cooling of molten metals, and fluid flows around shrouded heat-dissipation fins, and solar ponds. A very common industrial application of natural convection is free air cooling without the aid of fans: this can happen on small scales (computer chips) to large scale process equipment.

3.5 Forced convection is a mechanism, or type of transport in which fluid motion is generated by an external source (like a pump, fan, suction device, etc.). It should be considered as one of the main methods of useful heat transfer as significant amounts of heat energy can be transported very efficiently.

3.6 FIN

A **fin** is a thin component or appendage attached to a larger body or structure. Fins typically function as foils that produce lift or thrust, or provide the ability to steer or stabilize motion while traveling in water, air, or other fluid media. Fins are also used to increase surface areas for heat transfer purposes, or simply as ornamentation.

Fins first evolved on fish as a means of locomotion. Fish fins are used to generate thrust and control the subsequent motion. Fish, and other aquatic animals such as cetaceans, actively propel and steer themselves with pectoral and tail fins. As they swim, they use other fins, such as dorsal and anal fins to achieve stability and refine their maneuvering.

3.7 TYPES OF FIN

1. Rectangular fin
2. Circular fin
3. Oval shape fin
4. Dorsal fin
5. Anal fin
6. Clocal fin
7. Adipose fin
8. Tail fin
9. Annular fin

10. Straight fin
11. Tapered fin

3.8 Fin Efficiency / Effectiveness

If the whole fin were at the wall (or root) temperature, the increase in heat transfer rate would be in direct proportion to the increase in surface area. Owing to the temperature gradient along a fin this is not acheivable.

3.9 Varying cross-section fins

Although analysis of these fins is more complex than for constant cross-section fins, it can be done, and graphs are available which give fin efficiency as a function of their geometry and dimensions. These can be used to estimate the heat transfer enhancement achieved by using these types of fins.

3.10 Convective Heat Transfer Coefficient (h)

The **heat transfer coefficient** or **film coefficient**, in thermodynamics and in mechanics is the proportionality constant between the heat flux and the thermodynamic driving force for the flow of heat (i.e., the temperature difference, d*T*):

$$h=(q/dt)$$

where:

q: amount of heat transferred (heat flux), W/m^2 i.e., thermal power per unit area, $q = d/dA$

h: heat transfer coefficient, W/(m^2•K)

dT: difference in temperature between the solid surface and surrounding fluid area, K.

It is used in calculating the heat transfer, typically by convection or phase transition between a fluid and a solid. The heat transfer coefficient has SI units in watts per squared meter kelvin: W/(m^{2}K).

The heat transfer coefficient is the reciprocal of thermal insulance. This is used for building materials (R-value) and for clothing insulation.

There are numerous methods for calculating the heat transfer coefficient in different heat transfer modes, different fluids, flow regimes, and under different thermohydraulic conditions. Often it can be estimated by dividing the thermal conductivity of the convectionfluid by a length scale. The heat transfer coefficient is often calculated from the Nusselt number (a dimensionless number). There are also online calculators available specifically for heat transfer fluid applications. Experimental assessment of the heat transfer coefficient poses some challenges especially when small fluxes are to be measured

3.11 Dimensionless Numbers

In dimensional analysis, a **dimensionless number** (or more precisely, a **number with the dimensions of 1**) is a pure number without any physical units; it does not change if one alters one's system of units of measurement, for example from English units to metric units. Such a number is typically defined as a product or ratio of quantities which do have units, in such a way that all units cancel.

The non dimensionalization of the governing equations of fluid flow is important for both theoretical and computational reasons. No dimensional scaling provides a method for developing dimensionless groups that can provide physical insight into the importance of various terms in the system of governing equations. Computationally, dimensionless forms have the added benefit of providing numerical scaling of the system discrete equations, thus providing a physically linked technique for improving the ill-conditioning of the system of equations. Moreover, dimensionless forms also allow us to present the solution in a compact way. Some of the important dimensionless numbers used in fluid mechanics and heat transfer are given below.

3.11.1 Reynolds Number (Re)

In fluid mechanics, the **Reynolds number** (**Re**) is a dimensionless quantity that is used to help predict similar flow patterns in different fluid flow situations. The concept was introduced by George Gabriel Stokes in 1851, but the Reynolds number is named after Osborne Reynolds (1842–1912), who popularized its use in 1883.

The Reynolds number is defined as the ratio of inertial forces to viscous forces and consequently quantifies the relative importance of these two types of forces for given flow conditions. Reynolds numbers frequently arise when performing scaling of fluid dynamics problems, and as such can be used to determine dynamic similitude between two different cases of fluid flow. They are also used to characterize different flow regimes within a similar fluid, such as laminar or turbulent flow:

- laminar flow occurs at low Reynolds numbers, where viscous forces are dominant, and is characterized by smooth, constant fluid motion;
- turbulent flow occurs at high Reynolds numbers and is dominated by inertial forces, which tend to produce chaotic eddies, vortices and other flow instabilities.

In practice, matching the Reynolds number is not on its own sufficient to guarantee similitude. Fluid flow is generally chaotic, and very small changes to shape and surface roughness can result in very different flows. Nevertheless, Reynolds numbers are a very important guide and are widely used.

Reynolds number interpretation has been extended into the area of arbitrary complex systems as well: financial flows, nonlinear networks, etc. In the latter case an artificial viscosity is reduced to nonlinear mechanism of energy distribution in complex network media. Reynolds number then represents a basic control parameter which expresses a balance between injected and dissipated energy flows for open boundary system. It has been shown that Reynolds critical regime separates two types of phase space motion: accelerator (attractor) and decelerator. High Reynolds number leads to a chaotic regime transition only in frame of strange attractor model.

The Reynolds number can be defined for several different situations where a fluid is in relative motion to a surface. These definitions generally include the fluid properties of density and viscosity, plus a velocity and acharacteristic length or characteristic dimension. This dimension is a matter of convention – for example radius and diameter are equally valid to describe spheres or circles, but one is chosen by convention. For aircraft or ships, the length or width can be used. For flow in a pipe or a sphere moving in a fluid the internal diameter is generally used today. Other shapes such as rectangular pipes or non-spherical objects have an *equivalent diameter* defined. For fluids of variable density such as compressible gases or fluids of variable viscosity such as non-Newtonian fluids, special rules apply. The velocity may also be a matter of convention in some circumstances, notably stirred vessels. The Reynolds number is defined below for each case.

Re=(pVD/u)=(VD/v)

P=density(Kg/m3)

V= velocity(m/s)

D=diameter(m)

U=dynamic viscosity

3.11.2 Nusselt Number (Nu):

In heat transfer at a boundary (surface) within a fluid, the Nusselt number (Nu) is the ratio of convective to conductive heat transfer across (normal to) the boundary. In this context, convection includes both advection and diffusion. Named after Wilhelm Nusselt, it is a dimensionless number. The conductive component is measured under the same conditions as the heat convection but with a (hypothetically) stagnant (or motionless) fluid. A similar non-dimensional parameter is Biot Number, with the difference that the thermal conductivity is of the solid body and not the fluid.

A Nusselt number close to one, namely convection and conduction of similar magnitude, is characteristic of "slug flow" or laminar flow. A larger Nusselt number corresponds to more active convection, with turbulent flow typically in the 100–1000 range.

The convection and conduction heat flows are parallel to each other and to the surface normal of the boundary surface, and are all perpendicular to the mean fluid flow in the simple case

Nu=(hL/K)

where *h* is the convective heat transfer coefficient of the flow, *L* is the characteristic length, *k* is the thermal conductivity of the fluid.

- Selection of the characteristic length should be in the direction of growth (or thickness) of the boundary layer; some examples of characteristic length are: the outer diameter of a cylinder in (external) cross flow (perpendicular to the cylinder axis), the length of a vertical plate undergoing natural convection, or the diameter of a sphere. For complex shapes, the length may be defined as the volume of the fluid body divided by the surface area.
- The thermal conductivity of the fluid is typically (but not always) evaluated at the film temperature, which for engineering purposes may be calculated as the mean-average of the bulk fluid temperature and wall surface temperature.

In contrast to the definition given above, known as *average Nusselt number*, local Nusselt number is defined by taking the length to be the distance from the surface boundary to the local point of interest.

Nu=(hx/K)

3.11.3 Prandntle Number(Pr):

The **Prandtl number** (**Pr**) or **Prandtl group** is a dimensionless number, named after the German physicist Ludwig Prandtl, defined as the ratio of momentum diffusivity to thermal diffusivity.That is, the Prandtl number is given as:

Pr=(uCp/K)

u=dynamic viscosity

Cp= Specific heat.

K= Thermal Conductivity.

4. EXPERIMENTAL SET-UP

The experimental setup of pin fin apparatus consists of following components:

1. Blower
2. U tube manometer
3. Voltmeter
4. Ammeter
5. Digital temperature indicator
6. Circular fin with dimple and without dimple
7. Heater
8. Duct
9. Orificemeter
10. Pipe fittings

Fig 4.1 Experimental Set up

4.1 Blower

Fig 4.2 Blower

A **centrifugal fan** is a mechanical device for moving air or other gases. The terms "blower" and "squirrel cage fan" (because it looks like a hamster wheel) are frequently used as synonyms. These fans increase the speed of air stream with the rotating impellers.

They use the kinetic energy of the impellers or the rotating blade to increase the pressure of the air/gas stream which in turn moves them against the resistance caused by ducts, dampers and other components. Centrifugal fans accelerate air radially, changing the direction (typically by 90°) of the airflow. They are sturdy, quiet, reliable, and capable of operating over a wide range of conditions.

Specifications of blower:

Amp: 0.5, HP: 0.14, Watt: 110, Voltage: 240 volt

Blower used for the production of pressurised air for the experimental purpose..in this experiment centrifugal type blower is used .

Outlet of the blower is connected to the main piping system with help of pipe fittings like valves and sockets

4.2 U Tube Manometer

Fig 4.3 Simple U tube Manometer

The fig shows the simple u tube manometer is used in the experimental for the purpose of measurement of air head. While experimentation u tube manometer gives the air head in terms of meter of water ,so air head is again finding with help of u tube manometer by considering the density of water and air. The fluid used in the manometer is water. The height of the column in terms of m of water is measured while experimentation displacement of fluid in left limb and right limb,finally taking that value as hw for the further calculation of pressure head in terms of m of air as follows:

Pa*ha=pw *hw

Pa = density of air (˘Kg/m3)
ha = pressure head in m of air
Pw= density of water (kg/m3)
hw= pressure head in m of water

4.3 Voltmeter

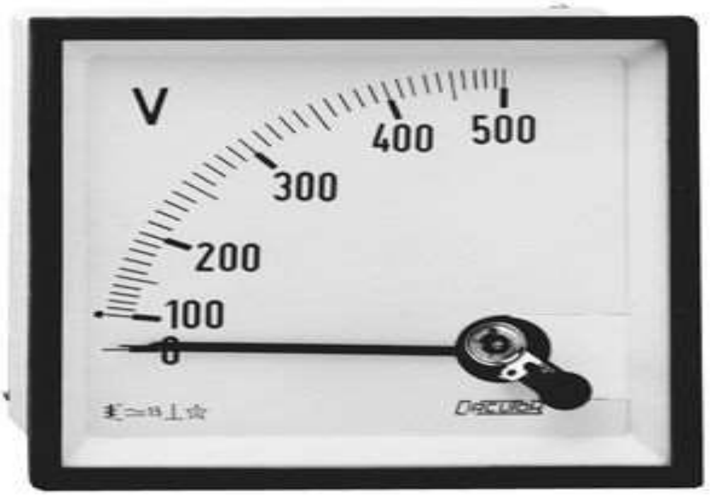


Fig 4.4 Voltmeter

A voltmeter, also known as a voltage meter, is an instrument used for measuring the potential difference, or voltage, between two points in an electrical or electronic circuit. Some voltmeters are intended for use in direct current (DC) circuits; others are designed for alternating current (AC) circuits. Specialized voltmeters can measure radio frequency (RF) voltage.

4.4 Ammeter

Fig 4.5 Ammeter

Among its many uses, electricity heats and lights our homes, makes our cars start up when we turn the key, and powers all our electronic devices. Sometimes we need to measure the electricity flowing through these devices. One of the instruments that can do this is the ammeter, which measures electric current. It gets its name from the standard unit of measurement for electric current, the ampere. Often you will see the word ampere shortened to amp. Nowadays, the job of an ammeter is often done with another, more versatile instrument called a multimeter, which can measure more than just current.When using an ammeter, it is very important that the instrument is correctly connected to the circuit.

4.5 Digital Temperature Indicator

Fig 4.6 Digital temperature Indicator

Microcontroller Unit (MCU) Devices are equipped with a temperature circuit designed to measure the operating temperature of the silicon die. The circuit's range of operating temperature falls between of -40°C and +85°C. The output of the circuitry is a voltage that is proportional to the device temperature and is connected internally to the device's Analog to Digital Converter (ADC). A channel is reserved for the temperature circuit output. Depending on the application, the Analog-to-Digital Converter result can be either compared directly against specific trip points, or used to determine the actual temperature by calculation, a look-up table or a combination of both.

4.6 Heater

It is an electrical element which is used for the heating the cylindrical fins. Their are two types of cylindrical fins are used i.e. With dimple and without dimple. When the electric supply is to be on at that time heater dissipates the heat on the fin material, and the distribution of heat is measured with the help of thermocouple wires which are connected to digital temperature indicator. The heater used in experimental setup is rated upto 230 voltage but dimmer is used for the varying the voltage condition and various heating positions.

Cylindrical Fin

Fig 4.6 (a) Positioning of cylindrical fin

Fig 4.6 (b) Positioning of cylindrical fin

Fig 4.6 (a) and (b) shows the positioning of the cylindrical fin which consist of fin with dimple and without dimple. The material used for the fin is an Aluminium (Al). The thermal conductivity of Al is 205 w/mk . The specific dimensions of cylindrical fin as follows:

Diameter =25mm

Length =160mm

For the effective heat transfer through forced convection while using cylindrical fin the dimple of oval shape are created on the surface of cylindrical fin. The dimensions of dimple as follows:

Diameter of dimple =20mm

Depth of dimple= 10mm

The fin is an extended surface which is cylindrical in shape. While working on experiment the air is forced on the surface of cylindrical fin . If dimples are not present the air directly travles around the cylindrical surface.

A Aluminium fin of circular cross section in fitted across a long rectangular duct. The other end of the duct is connected to the suction side of a blower and the air flows past the fin perpendicular to the axis. One end of the fin projects outside the duct and is heated by a heater. Extended surfaces of fins are used to increase the heat transfer rate from a surface to a fluid wherever it is not possible to increase the value of the surface heat transfer coefficient or the temperature difference between the surface and the fluid. Circumferential fins around the cylinder of a motor cycle engine and fins attached to condenser tubes of a refrigerator are a few familiar examples.

4.8 Duct System:

The rectangular duct is used in the experimentation model. Simply the material is used for the duct is acrylic. Because the acrylic is one of the best insulator and it must creates the barrier for the addition of heat. It means that it prevent s the addition of heat from atmosphere in the duct system. The dimensions of duct system are as follows:

Length= 300mm

Cross section= 160 mm*160 mm

The rectangular duct section cover with acrylic material and inside it the cylindrical fins are provided with the heater electric connections.

Fig 4.7 Duct system with acrylic

The outlet of the centrifugal blower is connected to the rectangular duct section with the help of pipe fittings as shown in fig 4.8

Fig 4.8 connection between duct and blower

The fig 4.8 consist of the pipe fittings such as socket, valve. The valve is used for the controlling the flow of air from blower to test section. The dimensions of pipe fitting are as follows:
Valve : 50mm
Pipe: 50 mm diameter
Length of pipe fittings with valve =185mm

4.9 Orifice meter

An **orifice plate** is a device used for measuring flow rate, for reducing pressure or for restricting flow (in the latter two cases it is often called a *restriction plate*). Either a volumetric or mass flow rate may be determined, depending on the calculation associated with the orifice plate. It uses the same principle as a Venturi nozzle, namely Bernoulli's principle which states that there is a relationship between the pressure of the fluid and the velocity of the fluid. When the velocity increases, the pressure decreases and vice versa

An orifice plate is a thin plate with a hole in it, which is usually placed in a pipe. When a fluid (whether liquid or gaseous) passes through the orifice, its pressure builds up slightly upstream of the orifice[1]:85–86 but as the fluid is forced to converge to pass through the hole, the velocity increases and the fluid pressure decreases. A little downstream of the orifice the flow reaches its point of maximum convergence, the *vena contracta* (see drawing to the right) where the velocity reaches its maximum and the pressure reaches its minimum. Beyond that, the flow expands, the velocity falls and the pressure increases. By measuring the difference in fluid pressure across tappings upstream and downstream of the plate, the flow rate can be obtained from Bernoulli's equation using coefficients established from extensive research

The average velocity V is defined so that it gives the correct rate of discharge when it is assumed constant over the vena contracta, or Q = VA. Then, we can write $V = C_vV_i$, where C_v is the coefficient of velocity. The coefficient of velocity is usually quite high, between 0.95 and 0.99. Combining the results of this paragraph and the preceding one, the discharge $Q = VA = C_vV_iC_cA_o = C_dA_oV_i$. C_d, the coefficient of discharge, allows us to use the ideal velocity and the orifice area in calculating the discharge.

Q= Cd a1 a0(2*g*ha)^1/2/(a1^2-a0^2)^1/2
Cd = Coefficient of discharge=0.64
a1= 1.017*10^-4 mm2
A0=1.530*10^-4 mm2
ha= pressure head in m of airg
g= 9.81 m/s2

5.1 Experimental Procedure

To study the temperature distribution along the length of a pin fin forced convection with dimple and without dimple, the procedure is as under

1. Start heating the fin by switching ON the heater and adjust dimmerstat voltage equal to 100 volts.
2. Start the blower and adjust the difference of level in the manometer with the help of gate valve.
3. Note down the thermocouple readings (1) to (4) at a time interval of 10 minutes.
4. When the steady state is reached, record the final reading (1) to (4) and also record the ambient temperature reading (5).
5. Repeat the same experiment with different manometer readings.

5.2 Precautions:

1. See that the dimmerstat is at zero position before switching ON the heater.
2. Operate the changeover switch of temperature indicator, gently.
3. Be sure that the steady state is reached before taking the final reading.
4. See that throughout the experiment, the blower is OFF.

5.3 OBSERVATIONS & CALCULATIONS

Specifications:

1. Duct size = 160mm x 160mm.
2. Diameter of the fin = 25mm.
3. Area of the orifice = a1=1.017*10^-4mm2; a0=1.530*10^-4mm2
4. Diameter of the delivery pipe = 50mm.
5. Coefficient of discharge (or orifice meter) Cd = 0.64.
6. Centrifugal Blower .14 HP single-phase motor.
7. No. of thermocouples on fin = 4.
   (1) to (4) indicated on temperature indicator.
8. Thermocouple (5) reads ambient temperature inside of the duct.
9. Thermal conductivity of fin material (Al) = 205w/m ^{0}C.
10. Temperature indicator = 0 – 300 ^{0}C with compensation of ambient temperature up to 50^{0}C.
11. Dimmerstat for heat input control 230V, 2 Amps.
12. Heater suitable for mounting at the fin end outside the duct = 400 watts (Band type).
13. Voltmeter = 0 – 100/200 V.
14. Ammeter = 0 – 2 Amps.

➢ **Observation Table:**

I) Without Dimple

hw=0.045m of water

SR.NO.	Volt	Current	T1	T2	T3	T4	T5
1	100	1	49	48	47	46	32.9
2	180	1.08	80	77	76	72	34.4
3	200	1.18	124	119	118	112	36.3
4	230	2	185	176	174	163	39.3

II) With Dimple

hw=0.045m of water

SR.NO.	Volt	Current	T1	T2	T3	T4	T5
1	100	0.80	57	48	45	44	36.2
2	150	1	76	67	61	59	38.2
3	200	1.08	121	101	88	85	42.2
4	230	1.2	174	139	118	112	46.3

6. RESULT AND DISCUSSION

Result Table - Without Dimple

SR NO	**Re**	**Nu**	**h**	**Efficiency%**
1	**167.94**	**6.69**	**7.25**	**95.46**
2	**204.71**	**7.342**	**8.25**	**94.87**

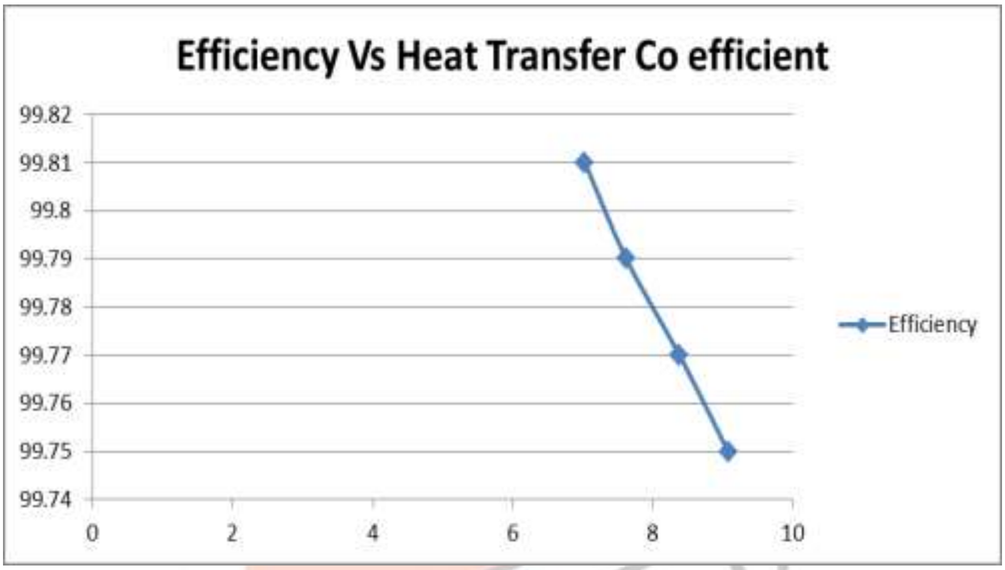
3	240.80	7.92	9.49	94.13
4	267.85	8.32	10.58	93.51

Result Table - With Dimple

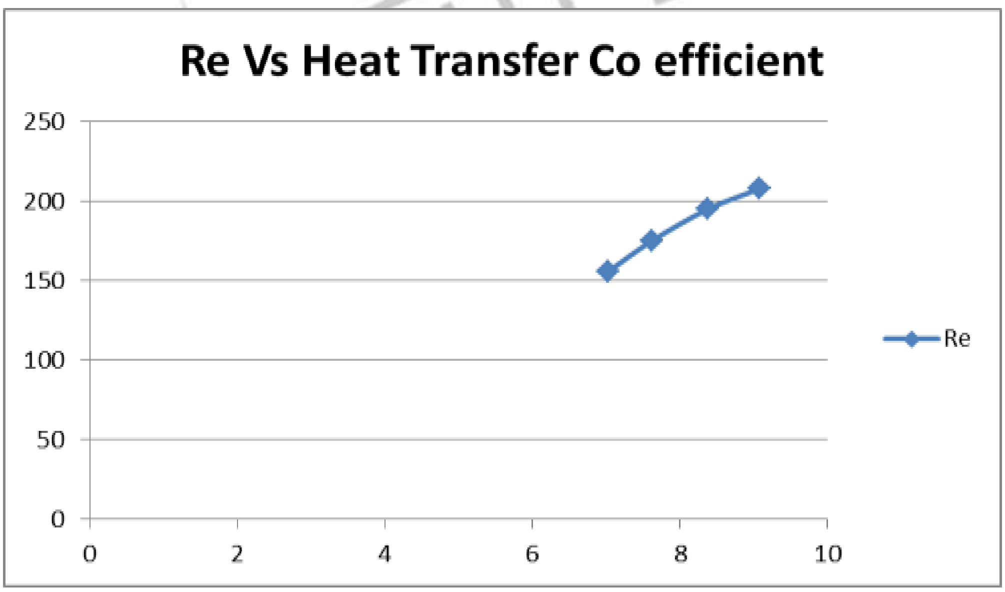
SR NO	Re	Nu	h	Efficiency%
1	156.06	6.47	7.03	99.81
2	175.31	6.83	7.62	99.79
3	195.29	7.18	8.38	99.77
4	208.05	7.39	9.074	99.75

Results graphs Without Dimples for various parameters

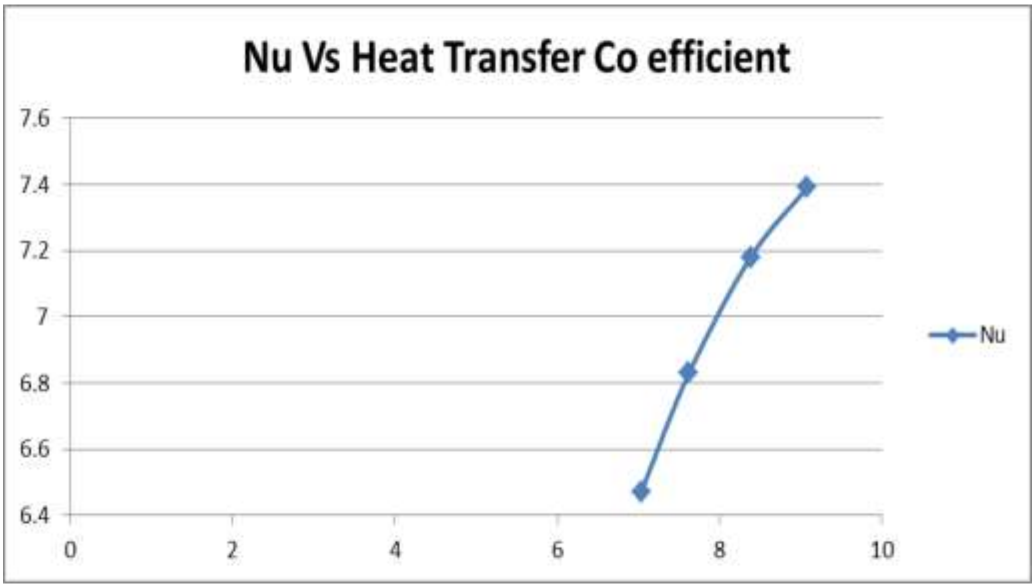
Graph 1



Graph 2

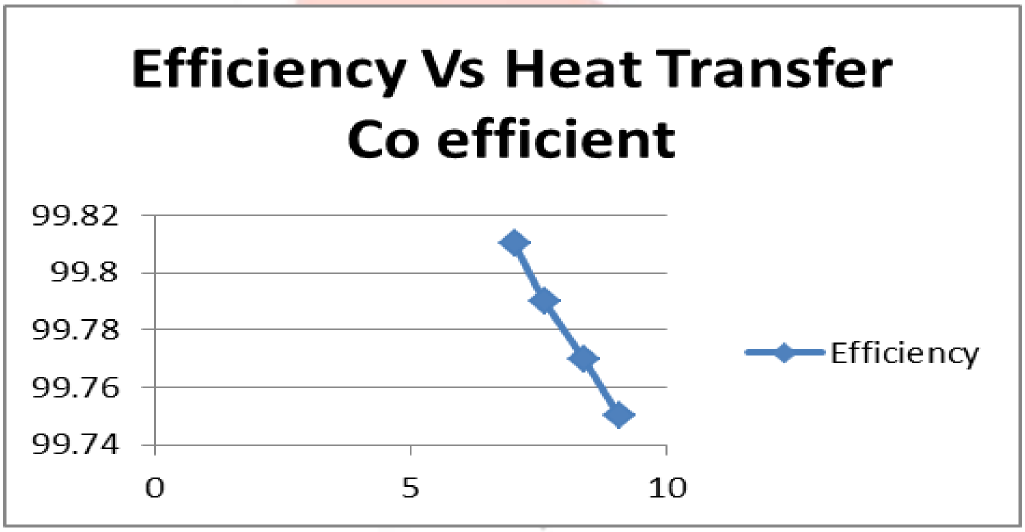


Graph 3

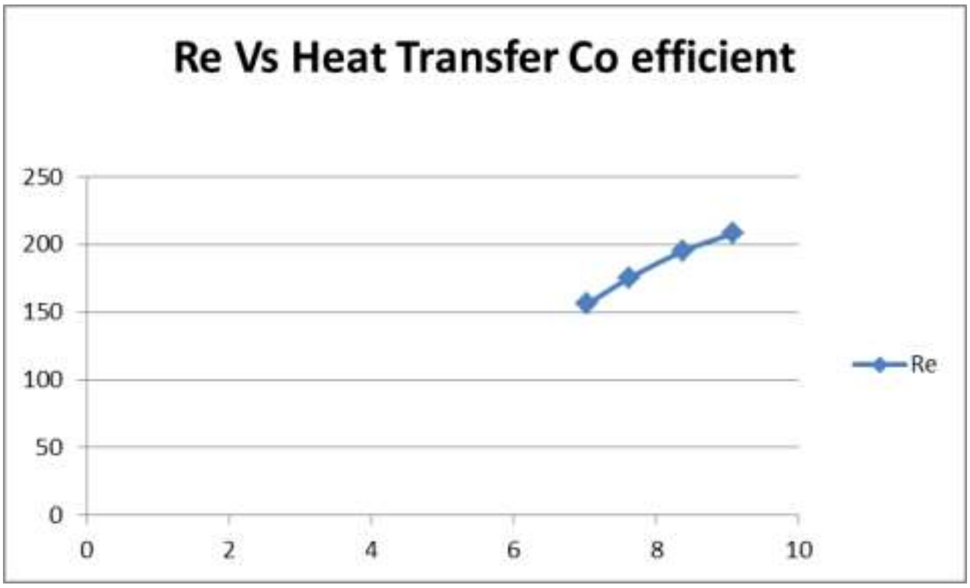


Results graphs With Dimples for various parameters

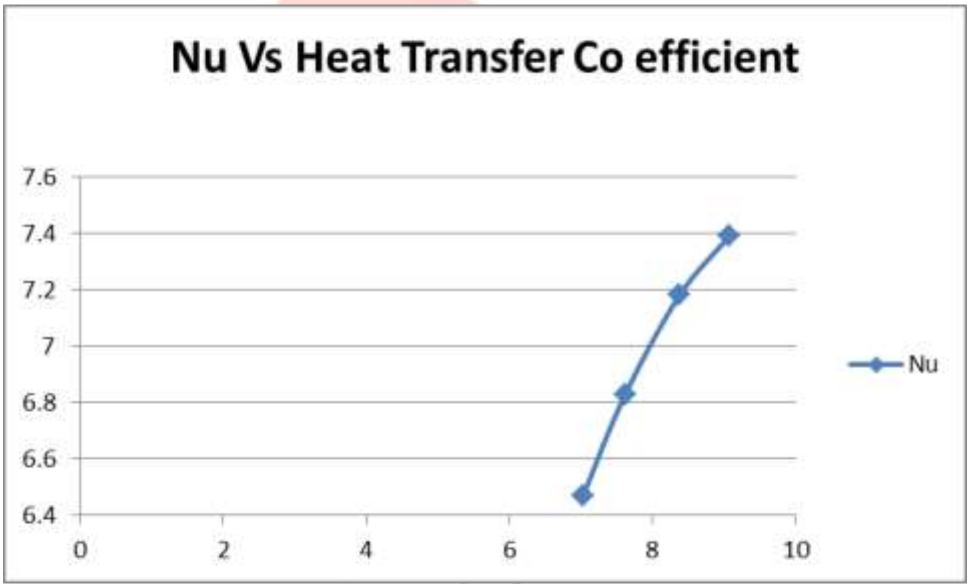
Graph 4



Graph 5



Graph 6



7. CONCLUSION

This study of heat transfer from Dimple pin fin of circular cross section has been done in this work. In this dissertation work the heat transfer enhancement with circular cross section shaped dimples has been experimentally investigated and results of these investigations are compared with each other. The conclusions drawn from this experimental investigation are listed below;

- From this study it can be concluded that with increase in Reynolds number the heat transfer coefficient of fins also increases .The maximum Reynolds number in circular fins is 267.85 without dimples and for dimples fin it is 208.05.
- It is also observed that with increases in Reynolds number the Nusselt number of the process also gets increased.
- As the objective of this study is to analyze effect of geometry (dimples) of the circular fins on the heat transfer. The experimental investigations conducted on annular fins of circular and elliptical cross section and results are compared to each other. It shows that Dimpled cross sectional circular fins transfer more heat and has high heat transfer coefficient in each case. It is observed that the maximum increase in heat transfer coefficient in circular fins with dimples is is 99.75% against the heat transfer coefficient without dimpled fins of 93.51.So, it can be concluded that the shape of geometry of the circular fin does affect the heat transfer characteristics of fins. And it can be also concluded that the Dimpled shape of the fin is more effective to transfer heat than circular shape fins.

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