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Experimental &Theoretical Analysis Of Heat Transfer Augmentation From Dimpled Surface

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ABSTRACT
In the present work the heat transfer characteristics and the pressure drop of the forced convection apparatus of six dimpled plates is studied. Six test plates with varying dimple densities; by varying the input voltage Nusselt No. variation was recorded. It is found that Nusselt No. increases as the dimple density increases .Also it was found that percentage increase in Nusselt No. is greater for staggered dimple arrangement. The sample experimental results obtained are presented in graphical forms as shown in Figure shows the calculated results based on the observations to show the comparative Nusselt numbers enhancements with that obtained with different parameters combinations.

Dimpled typical technique that offers a higher heat transfer increase at the cost of mild pressure drop penalty. This study investigates the heat transfer characteristics of Plate with dimpled surface. Over the past couple of years the focus on using concavities or dimples provides enhanced heat transfer has been documented by a number of researchers.

I. NOMENCLATURE

- A Effective heat transfer surface area , m^2
- C_p Specific heat at constant pressure for the air, $Jkg^{-1}K^{-1}$
- C_d Coefficient of discharge for orifice
- D Pipe diameter ,m
- d Orifice Diameter, m
- g Acceleration of gravity, $msec^{-1}$
- H Height of test plate, m
- h Convection heat transfer coefficient, $Wm^{-2}K^{-1}$
- K Thermal conductivity of gas, $Wm^{-1}K^{-1}$
- L Characteristics length of plate, mm
- m Mass flow rate, kg/s
- Nu Nusselt number
- Nuo Nusselt number obtained for plain plate
- Nu18 Nusselt number obtained dimpled plate with 18 number of dimples
- Nu20 Nusselt number obtained dimpled plate with 20 number of dimples
- Nu30 Nusselt number obtained dimpled plate with 22 number of dimples
- Nu50 Nusselt number obtained dimpled plate with 24 number of dimples
- Nu56 Nusselt number obtained dimpled plate with 33 number of dimples
- Nu/Nuo Baseline Nusselt number ratio
- q Heat transfer rate, W/m^2
- Re Reynolds number
- T Temperature, ^oC
- T_b Mean bulk temperature, ^oC
- T_{in} Air inlet temperature, ^oC
- T_{out} Air outlet temperature, ^oC
- T_s Average surface temperature, ^oC
- w Width of test plate, mm

II. INTRODUCTION

Extensive research effort has been focused on reducing the consumption of nonrenewable energy. Improving the efficiency of the universal process of heat exchange is one such area which continues to attract a lot of attention. Enhancing the efficiency of heat transfer is useful in a variety of practical applications such as macro and micro scale heat exchangers, gas turbine internal airfoil cooling, fuel elements of nuclear power plants, powerful semiconductor devices, electronic cooling, combustion chamber liners, biomedical devices, etc. Compact heat exchangers and gas turbine internal airfoil cooling are two applications which have been the subject of study for a number of researchers over the recent years.

Compact heat exchangers are used extensively in the trucking industry as radiators to reduce the excess thermal energy. Improved efficiency of compact heat exchangers can permit smaller radiators leading to smaller frontal area and thus can lead to substantial fuel saving. In a compact heat exchanger there are three important aspects of heat transfer. The first aspect to consider is the convection of heat from the fluid to the tube wall of the heat exchanger. The heat is then conducted through the walls of the tube. Finally, the heat is removed from the tube surface by convection to the air flowing through it. Air-side resistance to heat transfer in compact heat exchangers comprises between 70-80 percent of the total resistance and hence any improvement in the efficiency of a compact heat exchangers is focused on augmenting the air side convective heat transfer.

Heat exchangers are widely used in various thermal power plants, means of transport, heating and air conditioning systems, electronic equipment's, and space vehicles. In all these applications, improvements in the efficiency of heat exchangers can lead to substantial cost, space and material savings. Therefore, considerable research work has been done in the past to seek effective ways to increase the efficiency of heat exchangers [10].

The study of improved heat transfer performance is referred to as heat transfer enhancement, augmentation, or intensification. In general, this means an increase in heat transfer coefficient. For both single-phase and two-phase heat transfer, effective heat transfer enhancement techniques have been reported. However, in the present work wire coils inserts are used in the inner tube to increase heat transfer rate.

Use of dimpled surface can significantly intensify the heat transfer enhancement. Some typical wire insert are water tube boilers, condensers, evaporators, radiators, heat exchangers in chemical and textile industries, compact heat exchangers etc. Using coil wire insert tubes not only increase the swirl flow but also reduce the hydrodynamic resistance for the fluid flow over the surface. The swirls formed inside the tube results in thinning and disturbing the thermal boundary layer formed over the surface during coolant flow and hence, increasing the turbulence. And serve ultimately to bring about enhancement of heat transfer between the fluid and its neighboring surface at the price of less increase in pressure penalty.

III. LITERATURE REVIEW

A variety of experimental and analytical works has been carried out on enhancement of heart transfer. Especially the heat transfer enhancement by forced convection have concerned by many researcher and practitioners. Relevant literature pertaining to this title reviewed from different points is as below:

R.K.Ali[1] investigated experimentally the heat transfer from a heat source simulating an electronic chip mounted on a printed circuit board placed downstream of a guide fence on the lower wall of the flow passage with two different aspect ratios (H/W = 0.3 and 1). The channel height to the heat source height ratios (H/B) are of 10 and 3. The effect of the guide fence height (b) and the spacing between the guide fence and the heat source (S) were investigated. The guide fence was orientated such that guide fence extension point was varied from the midpoint of the front face of the heat source to the endpoint of the side face at 5000 6 ReL 6 30,000. The results for the heat source without guide fence displayed noticeable difference when compared with the flow over smooth plate placed on the lower wall of the flow passage. An enhancement in the convective heat transfer coefficient up to 20% is obtained when decreasing the flow passage height to the heat source height ratio from 10 to 3. Also, higher Nusselt number is located at the front face and the vertical sides of the heat source compared with that of the top surface. Nusselt number increases with the increase in both Reynolds number and the guide fence height while the effect of spacing between the guide fence and the heat source depending on the guide fence height. Correlations for the average Nusselt number were obtained utilizing the present measurements within the investigated range of the different parameters.

Arthur Bergles[2] investigated Phase-change processes, such as pool and flow boiling, are generally very effective modes of heat transfer. However, the demands of modern thermal systems have required the development of methods to enhance boiling systems. While heat fluxes above 108W/m2 have been accommodated in carefully controlled situations, the required fluid and the convective conditions usually dictate maximum heat fluxes several orders of magnitude lower. Two major contemporary areas, enhanced surfaces for pool boiling and enhanced surfaces and inserts for forced convection boiling /vaporization, are discussed, as they facilitate the attainment of high heat fluxes. In addition to these passive techniques, active techniques and compound techniques are mentioned. The taxonomy of enhanced heat transfer is covered, and recommendations are given for future work.

Yaroslav Chudnovsky[3] investigated vortex heat transfer enhancement of heat transfer by a system of 3D surface cavities (dimples) having specific geometry, dimensions and mutual orientation Each dimple acts as a "vortex generator" which provides an intensive and stable heat and mass transfer between the dimpled surface and gaseous heating media. The obtained method includes mixing of the fluid by creating artificial surface such as dimples or other techniques.

Qiuwang Wang , Qiuyang Chen, Ling Wang[4] studied technique to enhance heat transfer in cooling channels of plate-type fuel elements in reactor cores, the experimental research is conducted on the heat transfer and pressure drop in horizontal narrow rectangular channels with mounted longitudinal vortex generators (LVGs) for water flow with Prandtl number Pr = 4–5. The parameters examined were: flow velocity from 0.5 to 3.4 m/s, Reynolds number from 3000 to 20,000, heat flux 43.6 kW/m2, maximum system pressure 1.3 atm, and viscosity ratio from 1.05 to 1.2. It is found that the LVGs could greatly improve the heat transfer rate by 10–45%. Thermal performance is compared under three constraints, i.e., identical mass flow rate (IMF), identical pressure drop(IPD) and identical pumping power (IPP). It is found that the heat transfer performance of channel with LVGs on two sides are better than those on one side. Application of LVGs to plate-type fuel element is a potential technique for next generation advanced nuclear reactors concepts.

Srinath V. Ekkad , David Kontrovitz [5]has presented detailed heat transfer distributions are presented over a jet impingement target surface with dimples. Jet impingement by itself is an extremely effective heat transfer enhancement technique. This study investigates the effect of jet impingement on a target surface with a dimple pattern. The effect of dimple location, underneath the jets or between the jets, is investigated. The effect of dimple depth is also investigated. The average jet Reynolds number is varied from 4800 to 14 800. The heat transfer measurements are obtained using the transient liquid crystal technique. Results for dimpled target surfaces are normalized with data for plane target surfaces to determine whether the presence of dimples enhances heat transfer. Results show that the presence of dimples on the target surface, in-line or staggered with respect to jet location, produce lower heat transfer coefficients than the non-dimpled target surface. The bursting phenomena associated with flow over dimples produces disturbances of the impingement jet structures resulting in lower levels of heat transfer coefficients on the target surface.

A.Slanciauskas [6]studied the effect of roughness elements on heat transfer in fluids is different from their effect in gases because of the different thermal resistance of the viscous sub layer. The distinctive feature of the process was examined in every detail and generalized by two rules deduced, one for fluids with Pr> 5 and another for gases. For fluids, small equally dispersed roughness elements can disturb the viscous sub layer sufficiently. For gases, the enhancement is reached by creation of many reattachment zones after obstacles. Two- to three-fold heat transfer enhancement within the limits of Reynolds analogy is attainable.

IV. EXPERIMENTAL SET UP

Fig: Experimental Setup

Blower is used to supply air. The heat exchange module developed, was connected to air flow bench to force the air parallel to the dimpled test surface. The strip plate heater fixed at the bottom of the test plate, was connected to power socket through dimmer stat. Dimmer stat readings were varied to give the required heat input to the test plate. Calibrated Copper-Constantan thermocouple wires were used to measure the temperatures. Provisions were made to fix the thermocouple junction on the test surface. Temperatures of air at inlet and outlet of the heat exchange module are also measured Digital temperature indicators were used to show the temperature readings (in ^{o}C) recorded by thermocouple wires. Photo shows the image of experimental set-up.

V. EXPERIMENTAL PROCEDURE

In this section the procedure for the experimentation is outlined thoroughly. As mentioned in the earlier sections seven mild steel plates of dimensions 150x100x12 mm^3 were fabricated. Six plates were dimpled with different dimple densities and schemes of dimple arrangements, using round nose tool. Tests for forced convection heat transfer measurement were decided to conduct on the plain and dimpled surfaces. Plate heater was fixed at the bottom of the test plate. The depth of the dimple was kept as 3 mm to maintain δ/D ratio as 0.33. Later on depth of dimple was increased to 4 mm to give δ/D ratio as 0.41. The procedure for experimentation used is explained stepwise as follows

1. Switch on both the temperature indicators to ensure that both of them indicate accurate temperature readings initially.
2. Start the power supply to heater at required dimmer stat reading.
3. After achieving a suitable temperature (say 60^{o}C) at the test surface, ensure that temperatures recorded at all the test surface points are same. Start the blower to force the air flow over the test surface.
4. Adjust the regulating valve manually to adjust the mass flow rate of air to indicate the desired pressure differential reading on water manometer.
5. After achieving the steady state condition note down temperatures at different points on test surface, inlet and outlet air temperatures and pressure drop readings across the test section.
6. The readings are checked for repeatability.
7. For the same heat input to the test surface, repeat the procedure for flow rates of air i.e. at 04 cm, 06 cm and 08 cm of water column difference.
8. Change the heat input by varying dimmer stat readings and repeat the whole procedure. The dimmer stat readings were varied as 60 Volts, 80 Volts and 100 Volts .
9. The same procedure is repeated for the all test plates.

VI. MATHEMATICAL MODELLING

T_s =((T1+T2+T3+T4+T5)/5) ,
T_a= ((Tai+Tao)/2)

$$Q = C_d \frac{a_1\, a_2 \sqrt{2gH1 \frac{\rho_w}{\rho}}}{\sqrt{a1^2 - a2^2}} ,$$

m = ρ Q

$$m = \rho\, a_1 V, \qquad \therefore V = \frac{m}{\rho a_1}, \qquad Re = \frac{\rho V d_1}{\mu}$$

$$q = h\, A_s\, (T_s - T_{bm}) = m\, C_p\, (T_6 - T_7) \qquad \therefore h = \frac{m\, C_p\, (T_6 - T_7)}{A_s\, (T_s - T_{bm})}$$

$$Nu = \frac{h\, d_1}{k}, \quad f = \frac{\Delta P}{\left(\frac{\rho V^2}{2}\right)\left(\frac{L}{d_1}\right)}$$

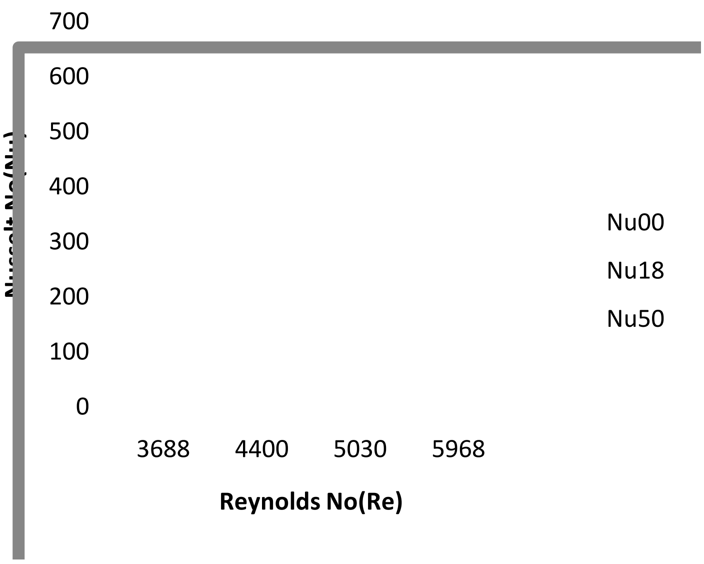

Figure 2: Reynold number v/s Nusselt number

VII. RESULTS AND DISCUSSION

Looking to the nature of the curves, it is found that plate with 50 dimples is having highest average heat transfer coefficient and hence highest average Nusselt number, in all the heat input conditions in the range of Reynolds number from 3688 to 5968.

VIII. CONCLUSION

The main conclusions are summarized as:

1. Heat transfer rate from the test surface increases with increase in mass flow rate of flowing fluid and heat input.
2. The use of dimples on the surface results in heat transfer augmentation in forced convection heat transfer with lesser pressure drop penalty.
3. The value of maximum Nusselt number obtained for staggered arrangement of dimples is greater than that for inline arrangement, keeping all other parameters constant. It shows that for heat transfer enhancement staggered arrangement is more effective than the inline arrangement.
4. At all Reynolds number considered Nusselt number augmentation increases as the dimple density of test plates increases (all other experimental and geometric parameters are kept constant).This is because the more number of dimples produce:(i) increase in the strength and intensity of vortices and associated secondary flows ejected from the dimples (ii) increases in the magnitudes of three-dimensional turbulence production and turbulence transport. But the percent increase in Nusselt number enhancement per unit percent increase in area decreases beyond a particular value of dimple density. More number of dimples beyond a particular value is believed to trap fluid which then acts as a partially insulating pocket to decrease the rate of Nusselt number enhancement with increase in further dimple density. It also results in decrease in rate of Nusselt number enhancement after a certain value of dimple density of plate (here 22 numbers of dimples for staggered arrangement and 24 numbers of dimples for inline arrangement). Thus it can be commented that the optimum value of dimple density lies in between 22 and 33 numbers of dimples for staggered arrangement and in between 24 and 35 for inline arrangement of dimples on the considered surface area.
5. At all Reynolds number considered Nusselt number augmentation increases with increase in dimple depth .But the rate of increase in Nusselt number per unit increase in surface area are low after increasing the dimple depth beyond a certain value. This is attributed to larger region of stronger re-circulating flow developed due to dipper dimple. The strong recirculating flows produced believe to trap the fluid which again acts as partially insulating zones results in lowering the rate of increase of Nusselt number enhancement. This concludes that there lies a optimum value of dimple depth and corresponding δ/D value where the rate of Nusselt number enhancement is maximum

IX. SUGGESTIONS FOR FUTURE WORK

1. In the present work experiments were carried for limited and low range of Reynolds number. This could be extended for the higher range of Reynolds number and using larger dimensions of the test surface.
2. Different shapes like rectangular or triangular shapes of dimples can be used instead of circular dimples on the test surface.
3. Performance of combination of above mentioned shaped dimples can be experimented and compared.
4. Test plate material can be changed such as copper (which is very good conductor of heat) and performance is compared with different material combinations

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