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HEAT TRANSFER IN FORCED CONVECTION THROUGH SQUARE DUCTS

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ABSTRACT

This paper presents an experimental study to investigate the heat transfer characteristics and friction factor in developing turbulent flow through a ribbed square duct and a non-ribbed square duct. The ducts are made of 16mm thick bakelite sheet. The bottom surface of the ribbed wall has rib pitch to height ratio of 6. Both of the non-ribbed and ribbed ducts are heated by passing alternating current to the heater placed under them. The constant heating is controlled using a digital temperature controller and a variac. The results of ribbed duct are compared to the results of the non-ribbed duct under the similar experimental conditions. It is found that the heat transfer in ribbed duct is better than the non-ribbed duct. It is also evident that in the ribbed duct Nusselt number increases by 19.68 percent and mean friction factor increases by 7.68 percent over that of the non-ribbed duct.

INTRODUCTION

In recent years, energy and material saving considerations have promoted an expansion of the efforts aimed at producing more efficient heat exchanger equipment through the augmentation of heat transfer. The potentials of heat transfer augmentation in engineering applications are numerous. An increase in the efficiency of the heat exchanger through augmentation may result in considerable saving in the material. In various enhancement techniques like; roughened surfaces, extended surfaces, enhancement devices as twisted taps and coiled tubes have been technically identified for possible use in heat transfer augmentation in different important engineering applications. Among those augmentation techniques, people are now showing more interest with large-scale surface roughness in the form of ribs of different geometry and orientations called turbulent promoters. These promoters at the wall surface help to destroy the viscous sub-layer and buffer layer, as a result heat transfer is enhanced.

In many engineering applications, uses of non-circular ducts have greater advantages over circular ducts. To have a clear understanding of heat transfer augmentation and flow characteristics in non-circular ducts a number of experimental studies have been carried out by different researchers. Akhanda[1] has carried out an experimental study on ribbed surfaces with eight different geometry of the test specimen. James,[2] Melling[3] and Keys[4] have conducted experimental studies with fully developed turbulent flow through non-circular smooth ducts. Ali,[5] Hirota,[6] Han,[7] Fijita[8] and Zhang,[9] have investigated heat transfer augmentation and flow characteristics with artificially prepared rib-roughened non-circular ducts for fully developed turbulent flow and recommended suitable mechanisation of heat transfer augmentation techniques.

Hong[10] and Monohar[11] have carried out experimental study on convection heat transfer for laminar flow in the entrance region on circular tubes. Sparrow[12] has conducted an investigation with laminar forced convection heat transfer in the entrance region of non-circular ducts. Khan[13-17] has carried out experimental investigations to have an understanding on heat transfer augmentation technique and friction characteristics in turbulent flow in the developing region of a rib-roughened square duct. The present investigations have been conducted to compare the heat transfer characteristics in square ducts of ribbed wall with non-ribbed wall.

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EXPERIMENTAL SETUP AND TEST PROCEDURES

An experimental apparatus has been designed and constructed to investigate the heat transfer augmentation in developing flow through a square duct as shown in Figure 1. Air at room temperature and at atmospheric pressure is introduced into the square duct through a contraction. The equivalent diameter of the duct is 50mm and its overall length is 5755mm, which are divided into three sections. They are the hydrodynamic developing duct of length 3000mm, the developed duct sections of length 1830mm and the rear duct sections of length 925mm. It was ascertained that the fully developed velocity profiles were attained about 50D (2500 mm) downstream from the entrance of the developing duct Bakelite sheet of 16mm thick is used to make its side walls and supporting wall. The top wall is made of 12.5mm thick clear perspex sheet and is fastened with the side walls by means of Allen bolts. A flexible duct (damper) is placed at the outlet of the rear duct. A diffuser of length 920mm is introduced between the outlet of the flexible duct and the inlet of the fan motor having 2.75 hp and 2900 rpm. A silencer is mounted at the outlet of the fan motor to minimize creating sound and vibration. A butter-fly valve is set-up at the other end of the silencer to control the flow rate of air for measuring the required Reynolds number.

At the beginning of every set of experiment, a specified target temperature is fixed at the temperature controller. The voltage and current supplied to the heater are recorded using a digital voltmeter and a digital ammeter and thus heat generated in the heater is calculated. Eleven thermocouples are fitted at the bottom surface of the aluminium wall to

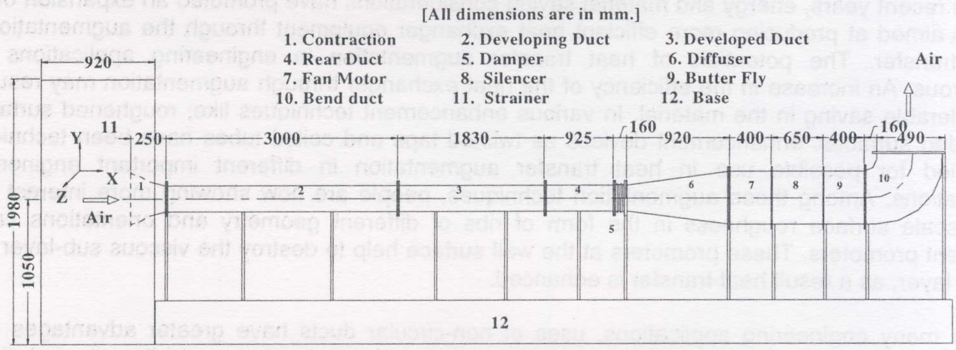

Figure.1 The Schematic Diagram of the Experimental Apparatus

measure its outer surface temperatures. The thermocouples are distributed at a pitch of 282mm, starting at 90mm from the leading end and along the bisector of the bottom surface. A digital thermocouple thermometer equipped with a selector switch is used to record the temperatures at their respective positions. The mean temperature of the bottom surface is calculated by averaging the recorded temperatures and the mean temperature of its inner surface is corrected by using one dimensional conduction equation.

The temperatures and velocities of the air flowing in the test duct are measured for 121 specified points across its inlet and outlet. From the recorded data the average temperature and average velocity at inlet and outlet are calculated for every set of experiment. Thus, the mean temperature and mean velocity of flow are evaluated by averaging these average temperatures and average velocities. For each set of experiments, static pressures at wall are measured from eleven pressure-taps. The taps are mounted at a pitch of 288mm in the range between X/D=3 to X/D=57 and at the same level of 25mm below its top-side. From the pressure difference along the length of the test duct the corresponding friction factors are evaluated.

After completing the experiments on the non-ribbed wall as the bottom surface of the duct as shown in Figure 2. It is then replaced with the ribbed aluminium wall for the next sets of experiments. The square sized rib having the dimensions of 1mm×1mm×50mm are prepared on aluminium plate by metal removing process as shown in Figure 3. The ribs are oriented at 90° with the flow direction. The maximum uncertainties involve in measured values are estimated as 11.5 percent.

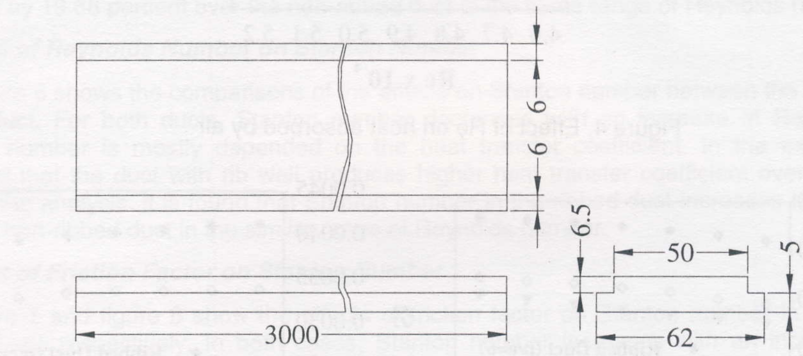

Figure 3. Orthographic views of the smooth wall

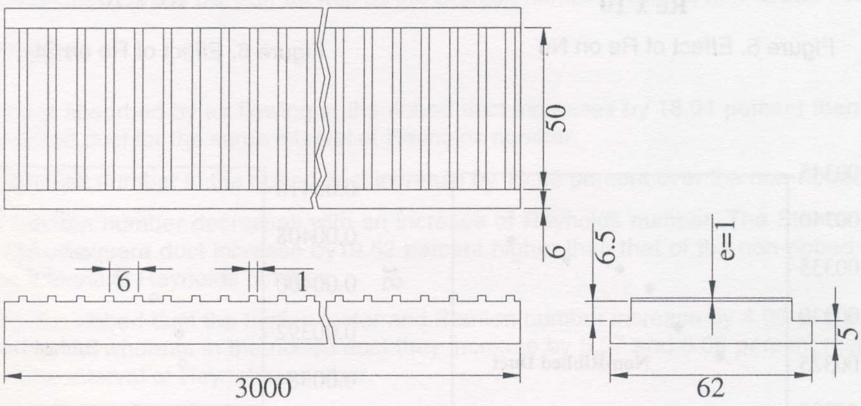

Figure 3. Orthographic views of the ribbed wall

RESULTS AND DISCUSSIONS

A. *Effect of Reynolds Number on Heat Absorbed by Air*

Figure 4 exhibits comparisons on the effect of Reynolds number between heat absorbed by air flowing through the ribbed and non-ribbed ducts. It is seen that the heat absorbed by air flowing in

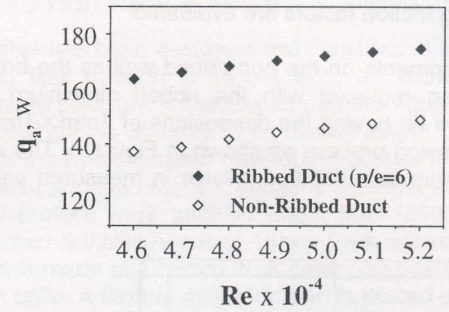

Figure 4. Effect of Re on heat absorbed by air

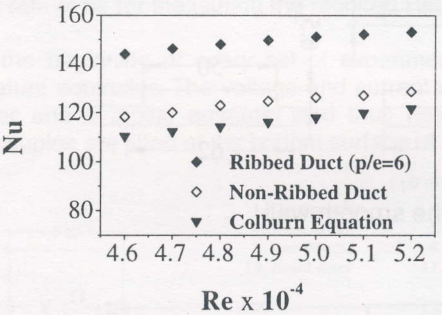

Figure 5. Effect of Re on Nu

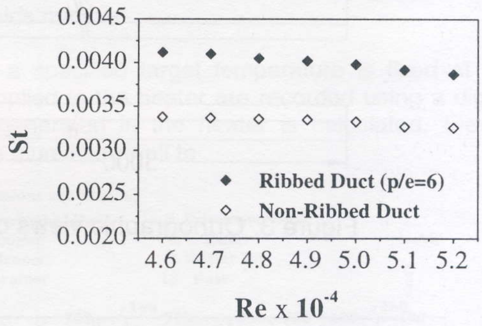

Figure 6. Effect of Re on St

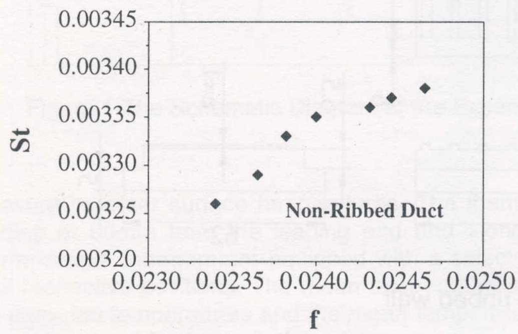

Figure 7. Effect of f on St Non-Ribbed

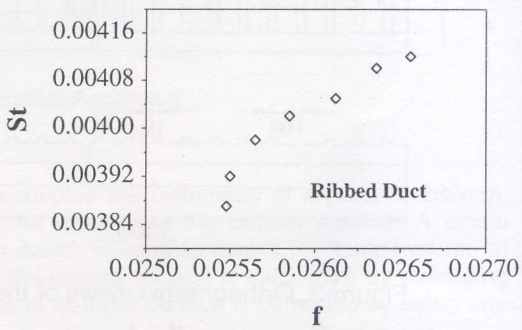

Figure 8. Effect of f on St on-ribbed Duct

the ribbed duct is more than that of the smooth duct and it is found that heat absorbed by air flowing through the ribbed duct increases by 18.01 percent over the air flowing through the non-ribbed duct in the same range of Reynolds number.

B. *Effect of Reynolds Number on Nusselt Numbers*

Figure 5 shows the comparisons of the effect of Reynolds number on Nusselt numbers between the ribbed and non-ribbed ducts and the correlations available in the literature. Nusselt number increases for both ducts with an increase of Reynolds number. Ribs break both the laminar and viscous sub layers and created local wall turbulence due to flow separation and reattachment between the ribs, which greatly enhances the heat transfer. Thus, ribbed duct yields higher amount of heat to the air flowing through the duct, which yields the higher heat transfer coefficient as well as higher Nusselt number. In the analysis it is observed that the Nusselt number in the ribbed duct increase by 19.68 percent over the non-ribbed duct in the same range of Reynolds number.

C. *Effect of Reynolds Number on Stanton Number*

Figure 6 shows the comparisons of the effects on Stanton number between the ribbed and non-ribbed duct. For both ducts, Stanton number decreases with an increase of Reynolds number. Stanton number is mostly depended on the heat transfer coefficient. In the experiments, it is observed that the duct with rib wall produces higher heat transfer coefficient over the non-ribbed duct. In the analysis, it is found that Stanton number in the ribbed duct increases by 19.52 percent over the non-ribbed duct in the similar range of Reynolds number.

D. *Effect of Friction Factor on Stanton Number*

Figure 7 and figure 8 show the effects of friction factor on Stanton number in non-ribbed and ribbed ducts respectively. In both cases, Stanton number increases with an increase of friction factor. It is found in the non-ribbed duct that friction factor and Stanton number increase by 4.06 and 3.55 percent respectively, whereas in the ribbed duct they increase by 5.07 and 6.06 percent respectively in the same range of Reynolds number. This is because, ribbed duct breaks both the laminar and viscous sub layers and created local wall turbulence due to flow separation which greatly enhances the heat transfer as well as the Stanton number over the non-ribbed duct.

CONCLUSIONS

- The heat absorbed by air flowing in the ribbed duct increases by 18.01 percent than that of the non-ribbed duct for the same interval of Reynolds number.
- The Nusselt number in the ribbed duct increase by 19.68 percent over the non-ribbed duct.
- The Stanton number decreases with an increase of Reynolds number. The Stanton number in the ribbed square duct increase by19.52 percent higher than that of the non-ribbed duct in the same interval of Reynolds number.
- In the non-ribbed duct the friction factor and Stanton number increase by 4.06 and 3.55 percent respectively, whereas in the ribbed duct they increase by 5.07 and 6.06 percent respectively in the same interval of Reynolds number.

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NOMENCLATURE

D	Hydraulic diameter, mm
e	Height/Width of ribs, mm
f	Friction factor
h	Heat transfer coefficient, W/m^2 ^{o}C.
Nu	Nusselt number
Pr	Prandtl number
p	Pitch between two adjacent ribs, mm
p/e	Rib pitch to height ratio
q_a	Heat absorbed to the air, watt
Re	Reynolds number
St	Stanton number
T_m	Mean temperature of air ^{o}C

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