HEAT TRANSFER AND FLOW FRICTION IN A RECTANGULAR DUCT WITH REPEATED MULTIPLE V-RIBS MOUNTED ON THE BOTTOM WALL

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Abstract

Measurements were carried out to investigate the enhancement in the heat transfer and the influence of the flow friction in a rectangular channel using periodic zigzag (multiple V-shaped) ribs affixed, in tandem, on the bottom surface of the channel, which was heated with a uniform heat flux. A horizontal rectangular channel with an aspect ratio, (height/width), of 1:4 was examined. The ribs were arranged in an in-line arrangement such that repeated V-shaped secondary flow cells were established. The present experiments were conducted for a range of Reynolds number from 15,000 to 75,000, rib height-to-pitch (e/p) ratio between 0.0375 and 0.15, and for three rib angles (α) of 30, 45 and 60 deg, respectively. An efficiency index of about 0.6 was observed for rib angles of 45 and 60 deg. at the lowest rib height to pitch ratio. New correlations were obtained for the influence of rib height-to-pitch ratio, rib angle, and Reynolds number on both Nusselt number enhancement ratio and flow friction factor ratio.

Keywords: Heat transfer enhancement, Multiple V-shaped ribs, Roughened surfaces, Rectangular duct flow

Nomenclature

SI units were used for the whole parameters within this paper.

c_p	air specific heat at constant pressure	V	axial flow mean velocity
D_H	hydraulic diameter of the duct $\{D_H=4(W{\times}H)/(2(W+H))\}$	W	test-duct width
		x	axial distance measured from inlet
e	rib height		
H	test duct height	**Subscripts:**	
h	heat transfer coefficient	f	for fluid
k	air thermal conductivity	m	mean value
L	test-section length	s	case of smooth duct flow
m	air mass flow rate	R	ribbed region
P	pressure	w	wall, surface
p	rib pitch	x	local value
q"	heat flux	x-Δx	test segment inlet
T	temperature	∞	fully developed conditions

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Greek letters:

α rib angle

Δ difference

ν air kinematic viscosity

ρ air density

Dimensionless Terms:

e/p   rib height-to-pitch ratio

F   Fanning friction factor, Eq.(4)

j   Colburn j-factor, (j=St Pr$^{2/3}$)

Nu   Nusselt number, Eq.(3)

Pr   Prandtl number

Re   Reynolds number (Re=VD_H/ν)

η efficiency index, $\eta=(Nu/Nu_s)/(F/F_s)$

1. Introduction

Enhancing the heat transfer rates in the internal flow passages of several thermal systems is an importunate request for saving power consumption and maintaining their structural integrity. This is particularly paramount in the engineering applications of elevated temperatures. Surface modifications like rib-roughening are commonly used in many applications such as compact heat exchangers and cooling systems in gas-turbine systems. In fact, numerous experimental and numerical studies have been reported on the commonly used rib configurations like solid square, rectangular, trapezoidal and perforated ribs, As examples, Sara et al. (2000), Demire et al.(1999), Sunden (1999), Abdel-Moneim et al. (2000a,b), Tsai et al.(1999), Hanafi et al.(2000). In general, the roughness with the whole known configurations, increases the heat transfer coefficient, but it also increases the pressure drop of the fluid flow. Therefore, it is a real challenge for investigators to find a new roughness geometry that will enhance heat transfer with a minimum increase in the flow friction.

In fact, the basic idea of the heat transfer enhancement is not only to disturb the velocity and temperature profiles close to the walls, but also to create a secondary flow that will exchange heat and momentum between the wall region and the core region. Nowadays, V-shaped ribbed duct is considered a new swirl flow geometry that creates a secondary flow and it enhances the heat transfer to a great extent. The previous investigations have shown that multiple V-shaped ribs may provide greater heat transfer enhancement than angled straight ribs and V-shaped ribs at Reynolds numbers below 2000. Olsson and Sunden (1997a) carried out experiments to study the flow field in ducts with different rib configurations. Parallel straight ribs, cross-straight ribs, parallel V-ribs, cross V-ribs and multiple V-ribs were investigated. It was found that, at Reynolds number below 2000 the secondary flow patterns are more stable when confined in square cells than confined in rectangular cells. The heat transfer data for these flow configurations were reported in Olsson and Sunden (1997b) and it was found that the swirl flow duct provides a significant increase in the j/f ratio at Re<2000. Olsson and Sunden (1998) carried out further experiments to investigate the influence of multiple V-shaped ribs in ducts. The tested ducts have rectangular cross sections with an aspect ratio of 1/8, and multiple V-ribs were attached to the wide walls in a staggered arrangement such that square secondary flow cells were established. Measurements were performed at Reynolds numbers ranging from 500 to 15,000, rib-to-duct height ratio between 0.1 and 0.2, rib pitch-to- duct height ratio from 3 up to 7 , and for a range of rib angles from 15 to 45 deg. It was found that a rib angle of 45 deg. provided the highest j/f ratio, while the increase of rib height or the decrease of the rib pitch decreases the j/f ratio. Han et al. (1991a,b) studied the effect of the rib angle orientation on the local heat transfer in a square channel with two opposite in-line ribbed walls for Reynolds number ranging from 15,000 to 90,000. The results showed that the V-shaped ribs perform better than the parallel and crossed ribs. Also, it was found that the crossed-ribs have the lowest heat transfer

enhancement and smallest pressure drop penalty, while the inverted V-shaped ribs generate the greatest friction.

The effect of V-shaped rib arrays on turbulent heat transfer and friction of fully developed flow in square channel was reported in Lau et al. (1991). This study was followed by two related studies Kukreja (1992) and Kukreja et al.(1993) including heat and mass transfer as well. The local heat/mass transfer in a square channel with full and V-shaped ribs were reported in Kukreja (1992) and Kukreja et al. (1993). Experiments have been carried out using naphthalene sublimation technique. Significant spanwise as well as streamwise variations of the local heat/mass transfer coefficient were found on the exposed surfaces of the ribbed walls in the oblique full rib and V-shaped rib cases.

To the best of the authors' knowledge, there have not been any investigations on enhancing the heat transfer in a rectangular duct using repeated zigzag ribs affixed on the bottom base in the open literature. Therefore, the objectives of the present study are:

i) To study, experimentally, the heat transfer and flow characteristics in fully developed turbulent flow in a rectangular channel ribbed with repeated multiple-V ribs on one its bottom wall.

ii) To investigate the effects of Reynolds number, rib angle, and rib intensity (rib pitch) on the flow and heat transfer.

iii) To obtain correlations for both the Nusselt number enhancement ratio and the Fanning friction factor ratio as functions of the investigated parameters.

2. Experimental Apparatus and Procedure

The experimental facility employed in the present investigation is an open-loop air flow circuit operated in the suction mode and oriented horizontally as shown schematically in Fig.(1-a). It consists of a blower assembly, test-section, and instrumentations to measure temperatures, pressure drop, airflow rate, and electrical power input. The details of the apparatus are depicted as follows:

(a) Air Blower: A centrifugal type air blower of 160 mm inlet section diameter driven by an AC motor of 3 hp capacity, is used to supply the system with air at the required flow rate. The airflow rate is controlled at the outlet of the air blower via a variable area outlet gate. A transition piece from a circular section of diameter 160 mm to a rectangular one with 160×40 mm is used to connect the air blower to the unheated exit duct of the test section.

(b) Test Section: The test duct, as shown in Figs.(1-b) and (1-c), is 1200 mm long and has a rectangular cross section of 40 mm by 160 mm which yields an aspect ratio of 1:4. Heat is supplied to the test section through only the principal bottom wall while the other three walls are well insulated. The heat-transferring wall is made of an aluminum plate of 6 mm thickness. The multiple V-ribs are made of aluminum wires and attached to the principal bottom wall of the test duct by using a thin layer of thermal glue. The V-ribs are fixed at downstream distance of 10 times the hydraulic diameter (≅600 mm) from the start of heating, as shown in Fig.(1-c), to assure that thermally fully developed flow condition is established. The rib pitch was changed from 20 to 40 to 80 mm by removing the intermediate ribs from the surface before the test. Aluminum plate and ribs are adopted in this work for their high thermal conductivity and machinability. Two main heater segments are used to heat the test section, one for the smooth region and the other for the ribbed one. The main heater segments are formed by winding Nickel-chromium wires uniformly around a thin mica sheet and sandwiched between another two mica sheets and then inserted between the aluminum plate and a 6-mm thick asbestos board. Another similar heater, which worked as a guard heater, is sandwiched between the

asbestos plate and the wooden base (20-mm thickness) to minimize the heat loss from the backside of the heated plate. The whole system is tightly fastened with through bolts to insure good contact. Details of the test duct cross section are shown in Fig.(1-b). Three pressure taps along the test duct are used for the static pressure drop measurements. The electric power input to the main and guard heaters can be controlled individually by a voltage regulator to provide a controllable constant heat flux. The test section is connected to the Bell-mouth intake through a gradual contraction, which is connected to the unheated entrance duct, as shown in Fig.(1-a). The unheated entrance duct is made of plexiglass plates of 5 mm thickness. This entrance duct served to establish hydrodynamically fully developed flow at the entrance of the heated duct. A calibrated orifice meter is used to measure the air flow rate through the test duct.

(c) Instrumentation: The test section was instrumented with calibrated copper-constantan thermocouples (type T) distributed along the length and across the span of the aluminum plate, highly concentrated at the ribbed region (the downstream half of the test duct) as shown in Fig.(1-d). These thermocouples are connected to a multi channel digital thermometer with scale division of 0.1 ^{o}C. Thermocouples, which could be traversed vertically, are used to measure the bulk mean air temperature entering and leaving the test section. Also, the airflow rate was measured by a calibrated orifice-meter connected to a digital micromanometer with scale division of 1.0 Pa. This digital micromanometer is also used to measure the pressure distribution along the test section. Electric power is supplied via voltage regulators to provide a controlled uniform heat fluxes for the smooth and ribbed regions. Also, a digital multimeter is used to measure the electric resistances, currents and voltages of the main and guard heaters, respectively.

Once the desired rib configuration was installed in the test section, the blower was switched on with the right amount of air flow rate. The heaters were also switched on and the experimental setup was allowed to reach steady state, which was obtained after about two hours. At this instant of time, readings of temperature, input powers, and pressure drop were recorded.

3. Data Analysis

The test duct is divided into 26 axial segments with lengths concentrated on the ribbed region (the downstream half of the test duct). The power input to the heaters was computed from the measurements of the voltage and resistance of the main heater segments and then the heat flux applied to each segment was obtained individually. During the tests, it was found that the ribbed region heat transfer capability was higher than that for the smooth one. In order to reduce the axial conduction between the smooth and rough walls, the heat input to the rough wall was controlled at about 1.5 to 2.0 of that for the smooth wall. This was accomplished by maintaining the temperature difference around the interface of the two regions not exceeds 0.5 ^{o}C. The heat lost by conduction through the insulation layers was neglected within the tested range of parameters. Also, the average heat flux to the test section was obtained by measurement of the average increase in the bulk temperature of flowing air across the test section. The average increase in the bulk temperature may be found by measuring the bulk temperatures at the inlet and exit of the test section. Since the values of the mass flow rate and the specific heat of air are known quantities, the heat flux may be calculated. This measurement was used just as a check of the heat flux measurement by the power input to the electrical heater. The heat balance was found to be valid within $\pm$ 4%.

Knowledge of the heat flux q" and the local surface temperatures of the test duct segments allowed for the computation of the local heat transfer coefficient h_x from the defining equation:

$$h_x = \frac{q''}{\overline{T}_{w,x} - \overline{T}_{f,x}}$$ (1)

where, q'' is the heat flux

$\overline{T}_{w,x}$ is the mean surface temperature of the test segment

$\overline{T}_{f,x}$ is the flow mean temperature at the duct segment and it is calculated by

averaging the segment inlet and exit temperatures: $\overline{T}_{f,x} = (T_{f,x} + T_{f,x-\Delta x})/2$

The flow inlet and exit temperatures, $T_{f,x-\Delta x}$ and $T_{f,x}$, are functionally related by successive heat balance for the duct segments as:

$$T_{f,x} = T_{f,x-\Delta x} + \frac{q''(W.\Delta x)}{m.c_P}$$ (2)

where, Δx is the test duct segment length. Also, the value of the measured inlet flow temperature is assigned to $T_{f,x-\Delta x}$ for the first test segment.

The local Nusselt number based on the equivalent (hydraulic) diameter (D_H) was evaluated as:

$$Nu_x = \frac{h_x D_H}{k_f} = \frac{q'' D_H}{k_f \left(\overline{T}_{w,x} - \overline{T}_{f,x}\right)}$$ (3)

The fully developed Nusselt number (Nu_∞) was evaluated by averaging the last five local values. Also, the flow Reynolds number based on the equivalent hydraulic diameter (D_H) was calculated as: $Re = VD_H / \nu$, where, V is the airflow mean velocity calculated from the mass flow rate divided by the density and cross-sectional area of the duct. The cross-sectional area is measured at the base of the roughness elements.

The Fanning friction factor (F) was calculated in terms of the flow frictional pressure drop along the smooth and ribbed regions as:

$$F_s = \left(\frac{\Delta P_s}{2\rho V^2}\right)\frac{D_H}{L_s}$$ (4-a)

$$F = \left(\frac{\Delta P_R}{2\rho V^2}\right)\frac{D_H}{L_R}$$ (4-b)

The thermophysical air properties (e.g. density, specific heat, viscosity, and thermal conductivity) are based on the mean film temperature.

An uncertainty analysis was performed to evaluate the accuracy of the present measurements. Percentages of uncertainties in the measurements of electric voltage, resistance, heating surface area, surface temperature, and flow mean temperature were 1.0, 0.6, 0.5, 1.0, and 1.0, respectively. Therefore, a total uncertainty of 4.1% was found for the heat transfer coefficient.

4. Results and Discussion

Prior to the ribbed duct experiments, fully developed heat transfer coefficient and flow friction were measured for the case of smooth duct. The results of these preliminary experiments were compared with the results given in the literature as shown in Fig.(2-a). The correlations selected for comparison include modification for gas flow in a rectangular duct.

A modified Dittus-Boelter correlation, ($Nu_s = 0.021 Re^{0.8} Pr^{0.4}$, MCAdames' correlation, as reference cited in Hwang (1998) was used to compare the heat transfer results while the correlation of Rohsenow and Choi (1969), ($F_s = 0.127 Re^{-0.3}$) was used to evaluate the flow friction data. It can be seen from Fig.(2) that the present measurements compare well with the previous correlations. In view of the satisfactory agreement the comparison has confirmed that the present experimental procedure is adequate and the obtained results are reliable. The present results for heat transfer and friction factor for smooth duct flow were correlated and the following correlations were obtained:

$$Nu_s = 0.02177 \ Re^{0.8} \tag{5}$$

and

$$F_s = 0.0567 \ Re^{-0.2} \tag{6}$$

These correlations are valid within $\pm6\%$ and ±13 % maximum deviations (for Nu_s and F_s, respectively) with the present experimental data within a range of Reynolds number from 15×10^3 to 75×10^3 for air.

The present measurements, for the case of ribbed duct, cover the cases of three different rib angles (α=30, 45 and 60 deg.), three different rib height-to-pitch ratios, (e/p=0.15, .075 and 0.0375) for a range of Reynolds number from 15000 up to 75000. The data corresponding to the thermally fully developed region were employed for the computation of the heat transfer coefficients. Also, the results for two rib angles and different pitches were compared with the extrapolation of the correlations of Olsson and Sunden (1998) for the Colburn j factor and Fanning friction factor and satisfactory agreements were found as shown in Fig.(3 & 4). Generally, higher values for the j factors were found for the ribbed duct compared with that for smooth one, this in fact confirms the establishment of heat transfer enhancement.

Figure (5) shows the fully-developed Nusselt number enhancement-ratio against Reynolds number for different rib configurations. In fact repeated strong shear layers may exist downstream of the V-ribs thus yields separation-reattachment flow patterns in periodic manners. As a result of the flow reattachment and impingement on the wall of the ribbed region, expected higher values for Nusselt number were found. Also, for the smallest rib angle, (α=30 deg.), i.e. at the higher angle of attack ($90 - \alpha$), small recirculating flow zones may take place upstream of the downstream V-ribs and this leads to further enhancement in the heat transfer as shown in Fig.(5).

The flow friction represented by the Fanning friction factor ratio is shown in Fig.(6) for the different rib configurations. Higher values for the friction factors were found, for the roughened duct in general, compared with that for smooth duct. Also, the flow friction was strongly influenced by the rib pitch where as no significant effect was found for the rib angle.

Higher heat transfer enhancement ratios and greater friction factors were found for the dense spaced ribs, (e/p ratio of 0.15) with rib angle of 30°. This may be attributed to the interaction and overlapping of the effect of both the downstream secondary flow and the upstream recirculating flow zones.

To better illuminate the flow and heat transfer characteristics, the efficiency index, which is defined by the ratio between the Nusselt number enhancement ratio to the friction factor ratio, was calculated as; $\eta = \dfrac{Nu / Nu_s}{F / F_s}$ and plotted against the Reynolds number at different rib geometries as shown in Fig.(7). The shape of the efficiency index curves reflects

H16

the different heat transfer and pressure drop mechanisms for different rib configurations. It was found that the efficiency index for this multiple V-rib configuration is, in general, greater than that for the helically roughened tubes and annulus, Abdel-Moneim et al. (2000 a, b).

Also, to evaluate the performance benefits of using the V-shaped ribs, the present results for the Nusselt number enhancement ratio (Nu_∞/Nu_s) as well as the Fanning friction factor ratio (F/F_s) were normalized as shown in Fig.(8). This was accomplished by dividing on the factor $\{(1+e/p)^a (1+\alpha/90)^b\}$ where the most appropriate values of the indices a,b were found by correlating the present results using the least squares method and the following correlations were obtained:

$$\frac{Nu_\infty}{Nu_s} = 0.881 \ Re^{0.057}\left(1+\tfrac{e}{p}\right)^{2.480}\left(1+\tfrac{\alpha}{90}\right)^{-0.257} \tag{7}$$

$$\frac{F}{F_s} = 0.771 \ Re^{0.136}\left(1+\tfrac{e}{p}\right)^{7.380}\left(1+\tfrac{\alpha}{90}\right)^{-0.830} \tag{8}$$

These correlations are valid, respectively, with $\pm$ 10% and $\pm$ 15% maximum deviations within the investigated ranges of the studied parameters, ($15000 \leq Re \leq 75000$, $0.0375 \leq e/p \leq 0.15$, and $30 \leq \alpha \leq 60^o$).

From the obtained correlations, Eqs.(7 & 8), one can easily find a roughness configuration that will yield a certain heat transfer enhancement and the corresponding increase in the pressure drop. Also, these correlations, were evaluated by simply dividing Eq.(7) by Eq.(8) and then checked with the corresponding experimental results as shown in Fig.(9). An envelope with $\pm$10% wide was found to cover most of the present experimental data.

5. Concluding Remarks

Turbulent heat transfer and flow friction characteristics in a rectangular duct with multiple zigzag (V-shaped) ribs arranged on one of its principal walls (bottom base) have been investigated experimentally. The effects of variation in the flow Reynolds number, rib height-to-pitch ratio (e/p) and rib angle (α) have been examined. The experimental results suggest the following conclusions:

1- Multiple V-shaped ribs cause secondary flow that interacts with the main flow over the ribbed wall of the duct. This promotes turbulence and yields a significant enhancement in the heat transfer and at the same time increases the flow friction.

2- Multiple V-shaped ribs provide higher values for the efficiency indices compared with those of the helical ribs.

3- The enhancement in the heat transfer is critically influenced by the rib angle depending on the creation origin and the relative movement of the secondary flow.

4- Higher values for both Nusselt number enhancement ratios and flow friction ratios were found for the dense spaced V-shaped ribs, (at e/p=0.15) with rib angle of 30°, where, a maximum enhancement in Nusselt number of about 1.8 was found corresponding to about 4 fold increase in the friction factor.

5- The greatest efficiency index was found for the wide spaced ribs, (at e/p=0.0375) with rib angle of 45 deg.

6- Empirical correlations were obtained for the influence of Reynolds number, rib angle (α), and rib height-to-pitch ratio (e/p) on both Nusselt number enhancement ratio and Fanning friction factor ratio.

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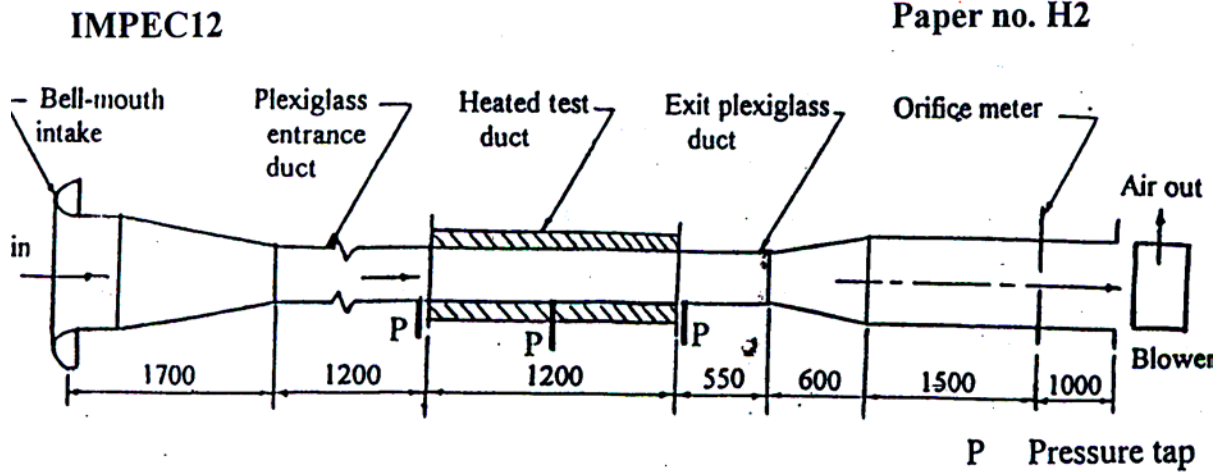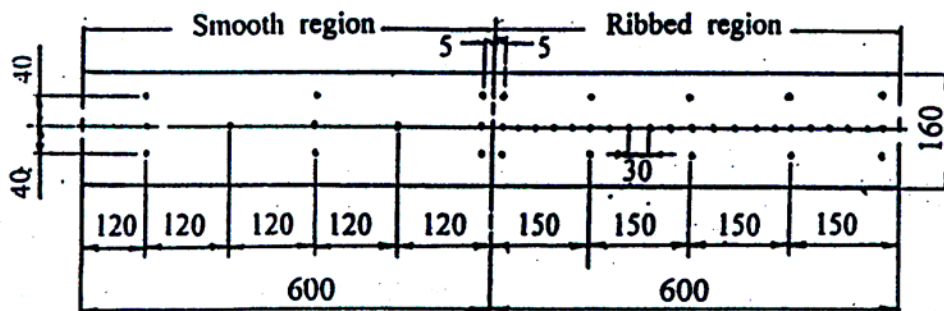
(a): Schematic diagram of the experimental apparatus
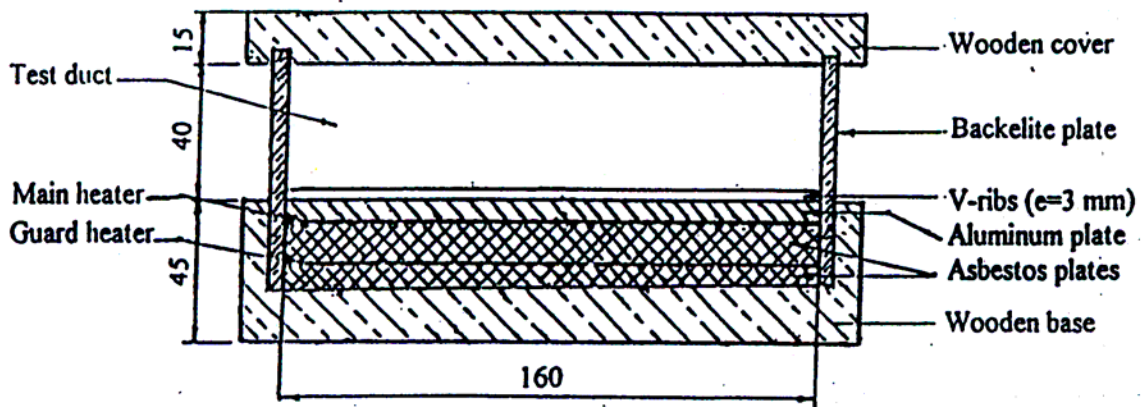
(b): Cross-sectional side view of the test duct
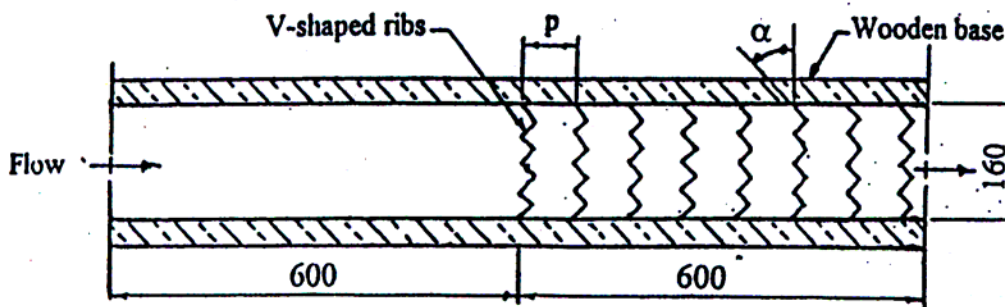
(c): Cross-sectional plan of the test duct

(d): Thermocouples distribution

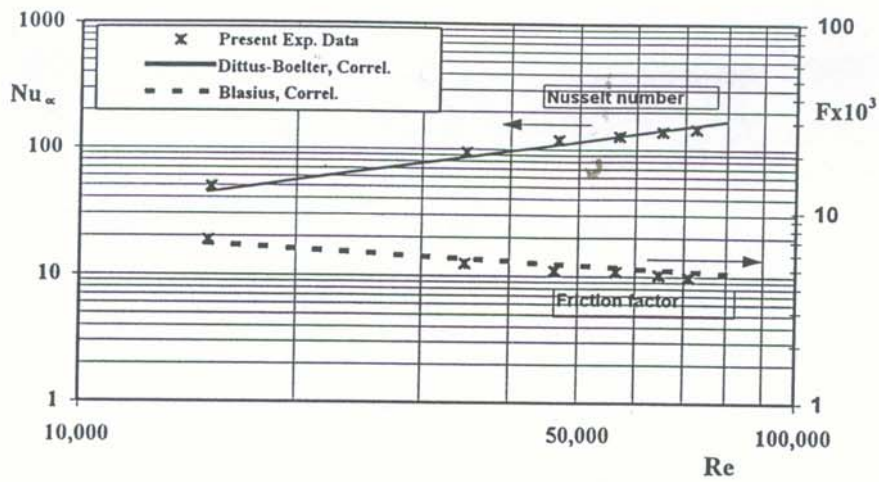Fig.(1): The Experimental apparatus

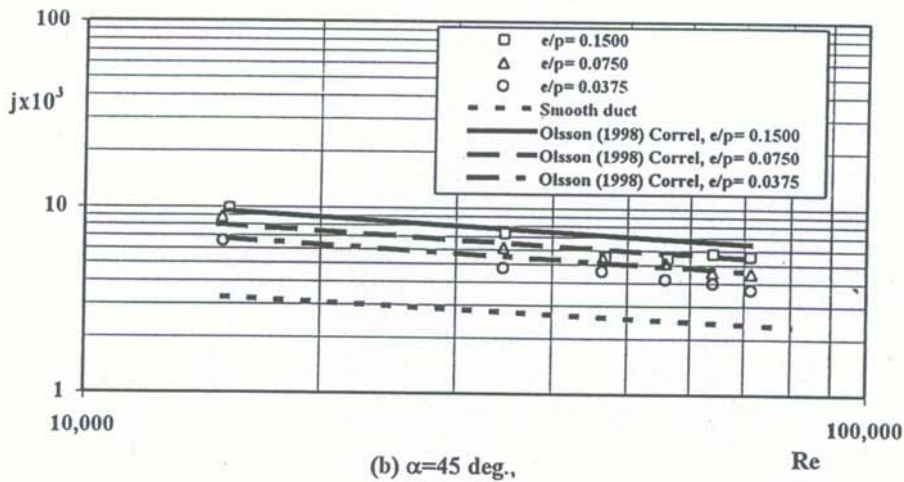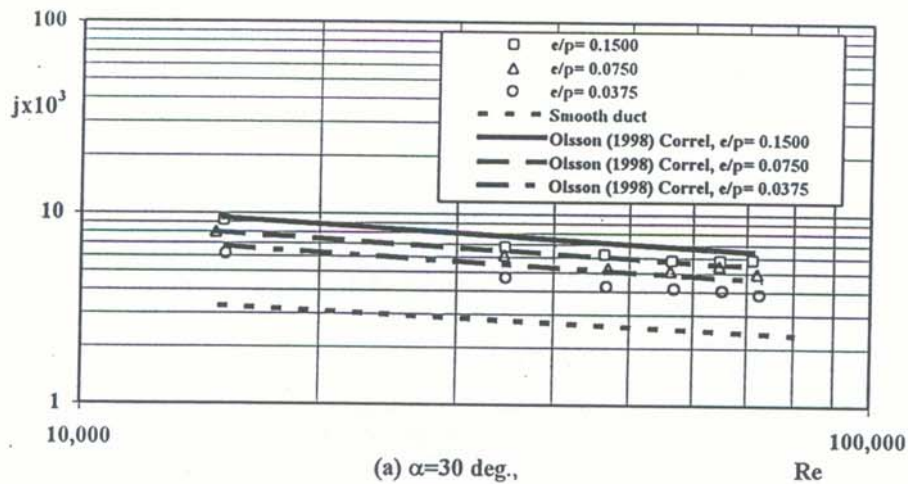Fig.(2): Comparison of the smooth duct results with previous correlations

(a) α=30 deg., Re

(b) α=45 deg., Re

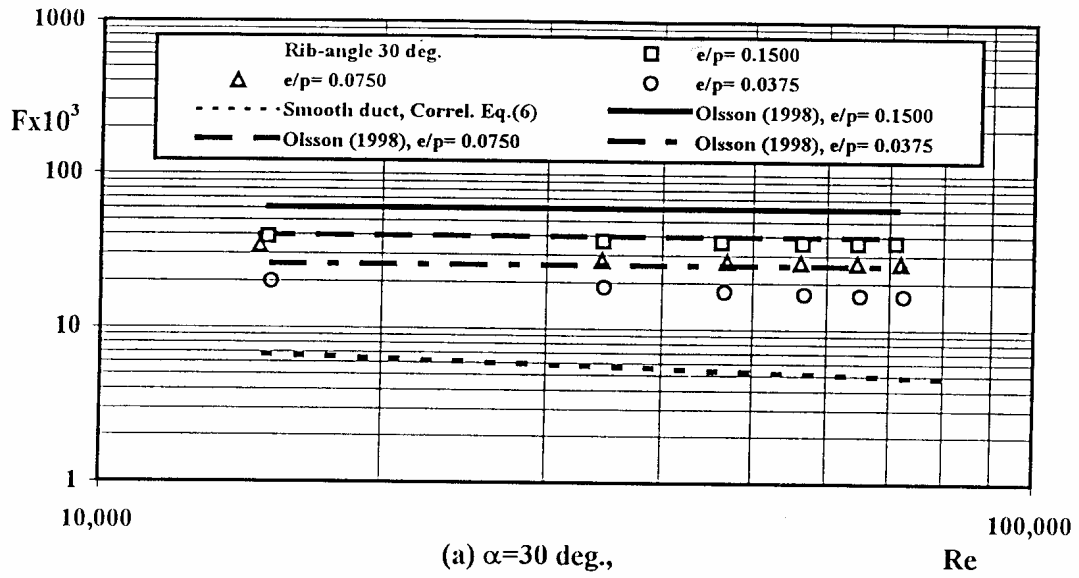Fig.(3): Colburn j-factor at different rib angles and different e/p ratios

(a) α=30 deg.,

(b) α=45 deg.,
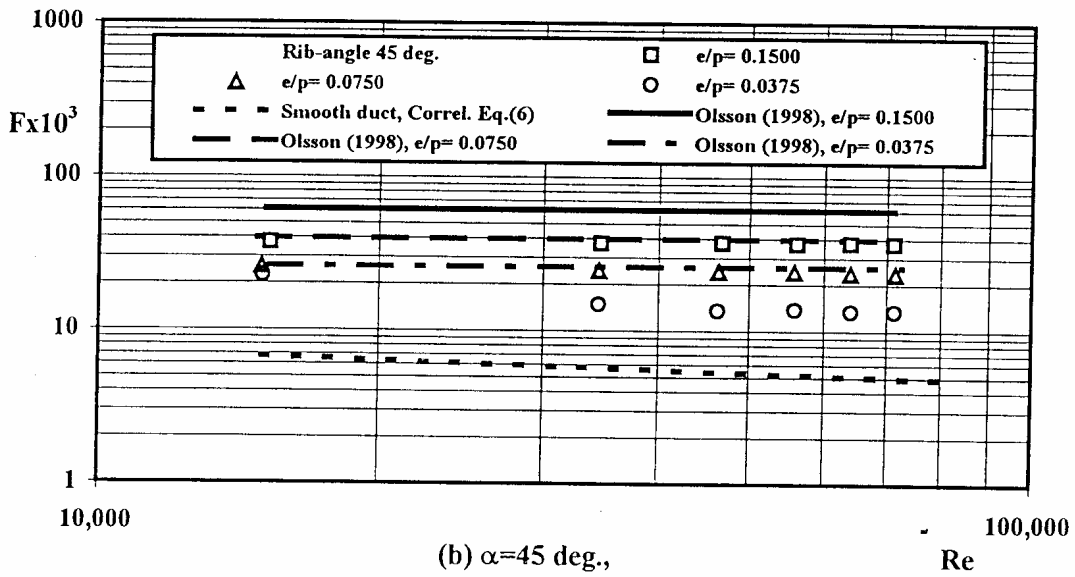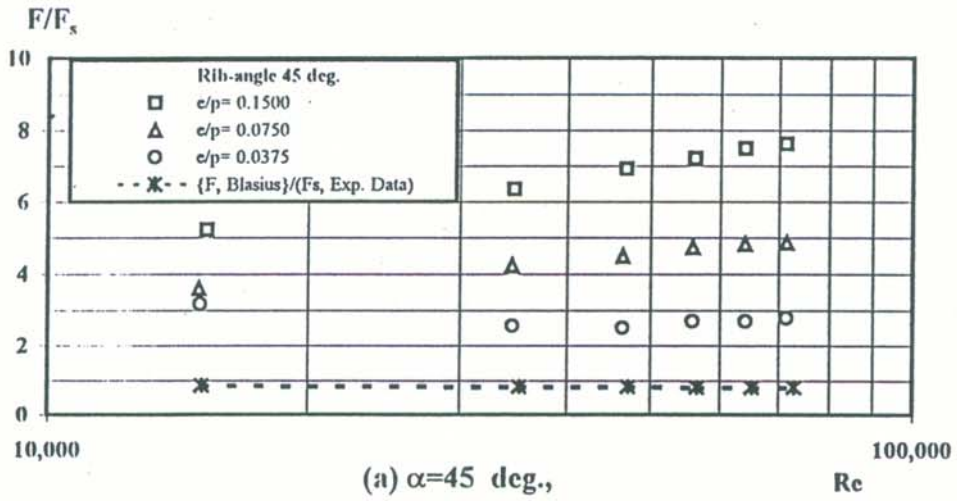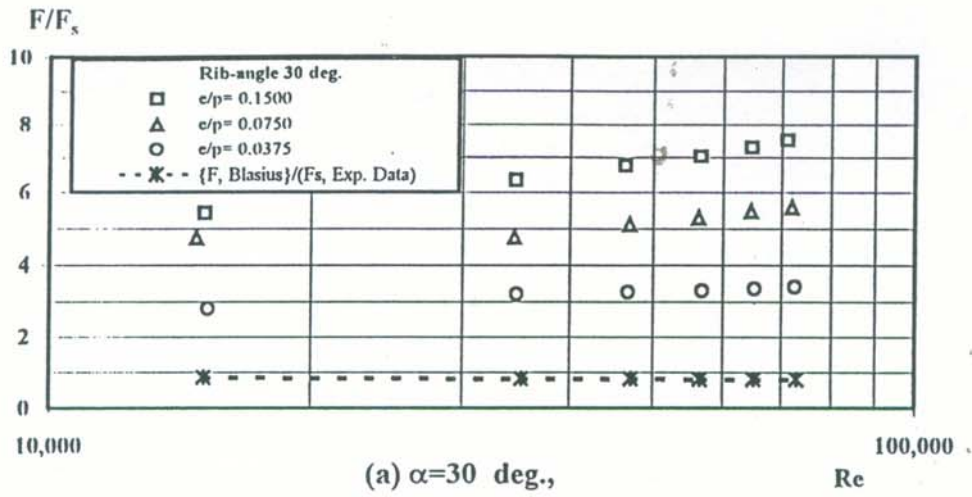
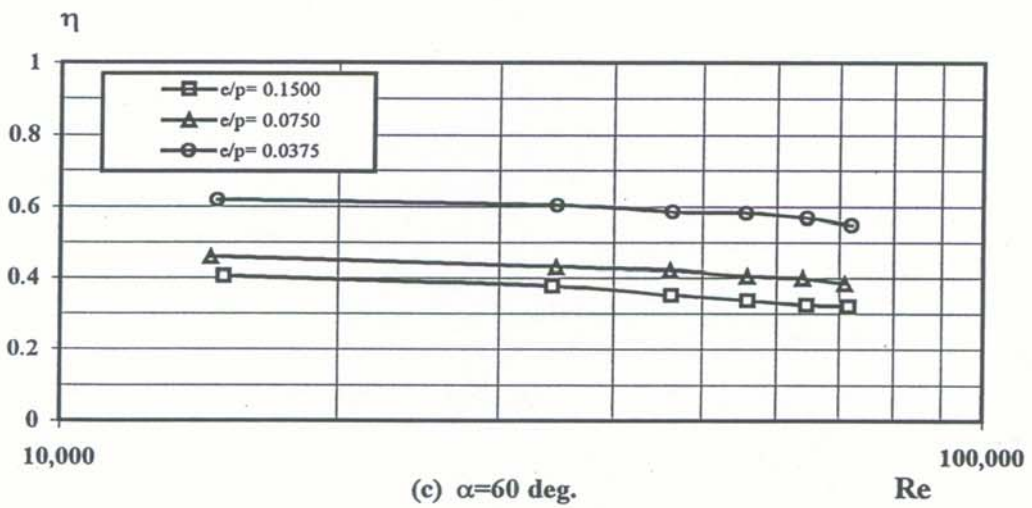Fig.(4): Fanning friction factor at different rib angels and e/p ratios

Nu_α/Nu_s

Rib-angle 30 deg.
- □ $e/p= 0.1500$
- △ $e/p= 0.0750$
- ○ $e/p= 0.0375$
- - - ✖ - - {Nu, Dittus-Boelter}/(Nus, Exp.Data)

(a) $\alpha=30$ deg.,   Re

Nu_α/Nu_s

Rib-angle 45 deg.
- □ $e/p= 0.1500$
- △ $e/p= 0.0750$
- ○ $e/p= 0.0375$
- - - ✖ - - {Nu, Dittus-Boelter}/(Nus, Exp.Data)

(b) $\alpha=45$ deg.,   Re

Nu_α/Nu_s

Rib-angle 60 deg.
- □ $e/p= 0.1500$
- △ $e/p= 0.0750$
- ○ $e/p= 0.0375$
- - - ✖ - - {Nu, Dittus-Boelter}/(Nus, Exp.Data)

(c) $\alpha=60$ deg.,   Re
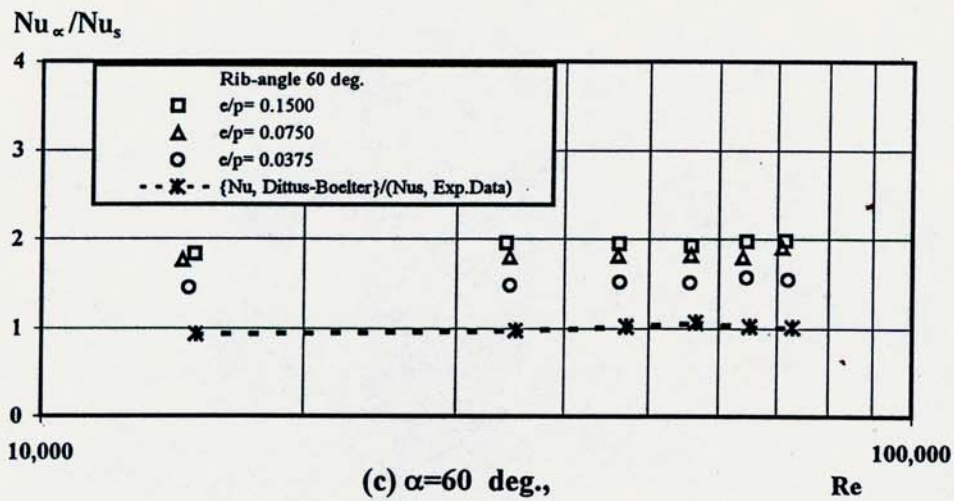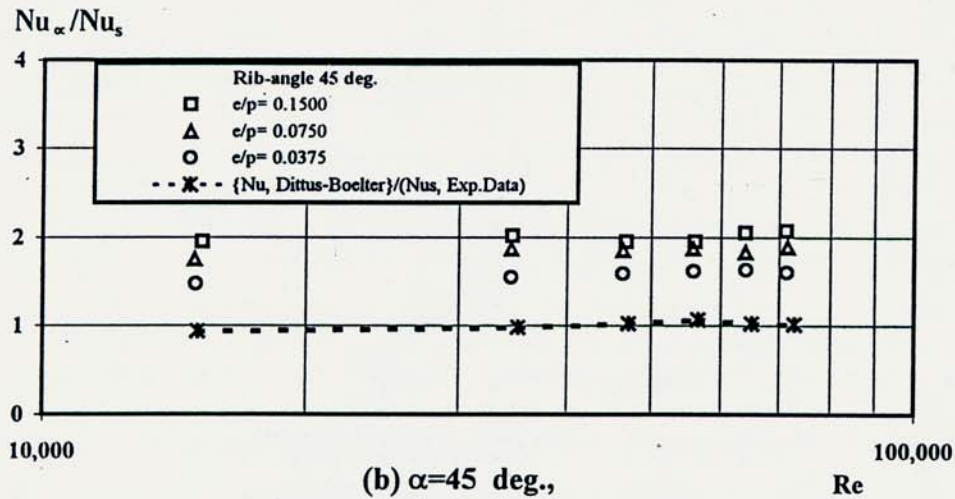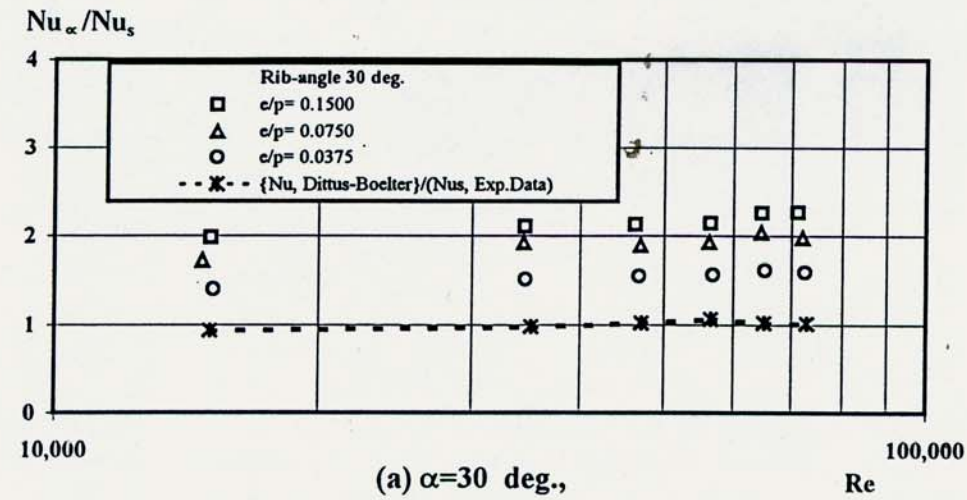
Fig.(5): Fully developed Nusselt number enhancement ratio
at different rib angles and e/p ratios

F/F$_s$

(a) α=30 deg.,   Re

F/F$_s$

(a) α=45 deg.,   Re

F/F$_s$

(a) α=60 deg.,   Re
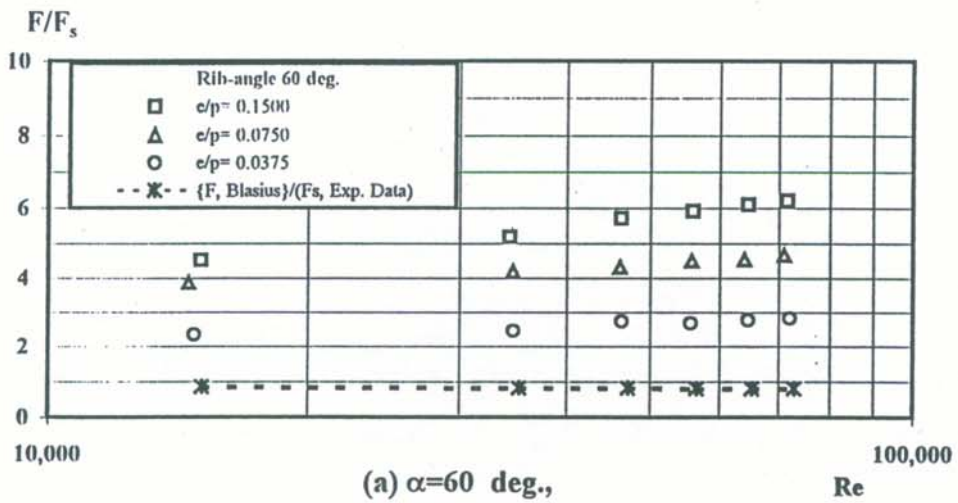
Fig.(6): Fanning friction factor ratio at different rib angels and e/p ratios
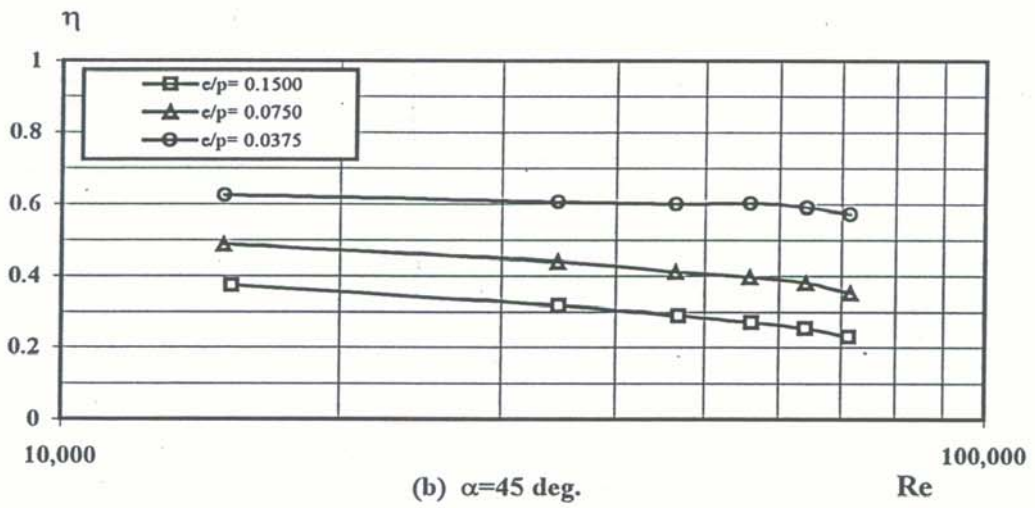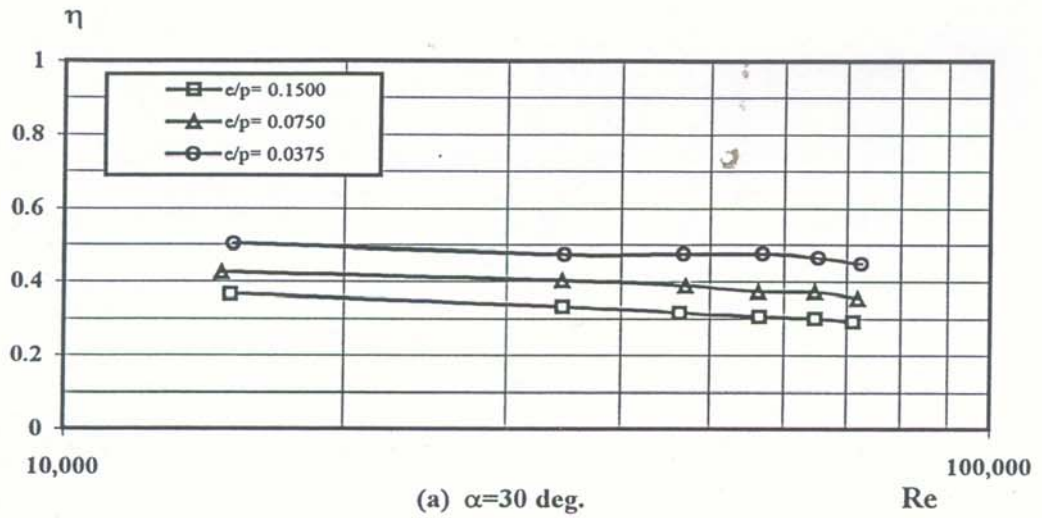
(a) α=30 deg.

(b) α=45 deg.

(c) α=60 deg.

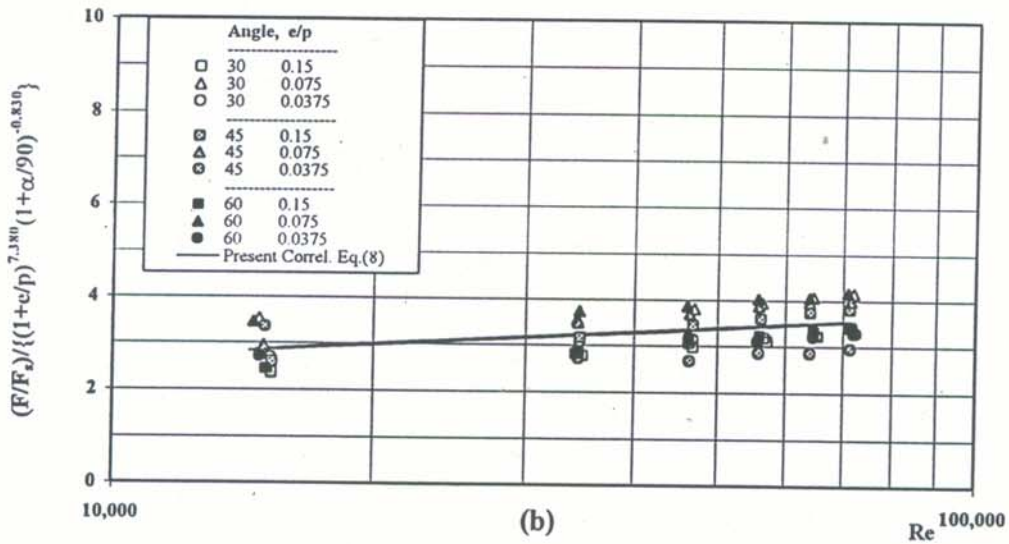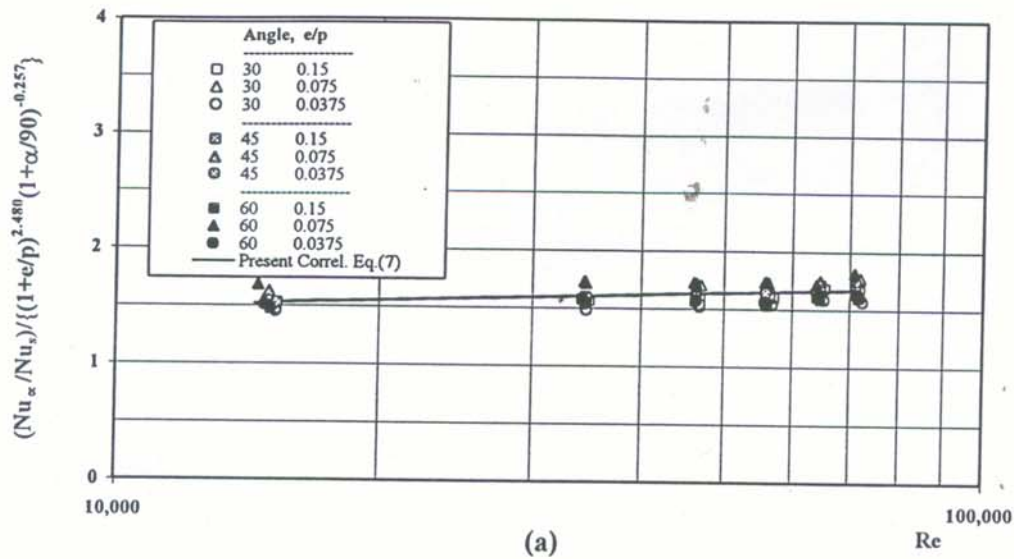Fig.(7): Roughness efficincy index for different rib-configurations

Fig.(8): Generalized: (a) Nusselt number enhancement ratio
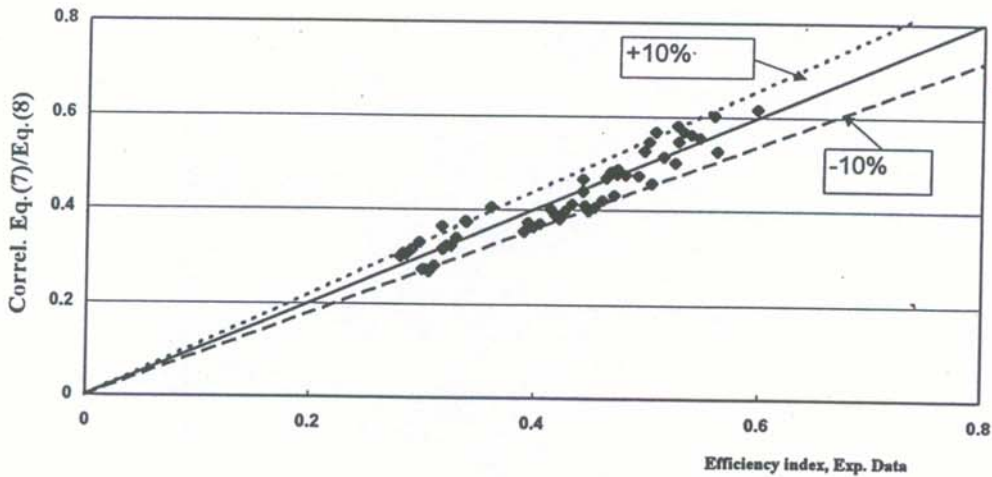(b) Fanning friction factor ratio, versus Reynolds number

Fig.(9): Efficincy index evaluation