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THE EFFECT OF SURFACE ROUGHNESS ON THE HEAT TRANSFER THROUGH THE EVAPORATOR TUBES

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ABSTRACT:
The present work is an experimental investigation to study the effect of surface roughness in the evaporator tubes on the heat transfer coefficient. The surface roughness are achieved by using mechanical and chemical treatments. The experiments are performed by using R-11 inside copper tubes for seven grades of surface roughness ranging from $R_a = 0.05$ to 1.5 µm. It has become clear that the heat transfer coefficient remarkably increases with the increase of the surface roughness. A detailed comparison of the experimental results shows the influence of the different surface conditions achieved by mechanical and chemical treatment on the absolute value of average heat transfer coefficient. Correlations for the enhancement of average heat transfer coefficient and also the average temperature difference as function of arithmetic average height R_a are developed. The present experimental results show good agreement with other published experimental data.

KEY WORDS: Forced convection–roughned surfaces-refrigerant 11.

1. INTRODUCTION

In heat transfer and fluid (refrigerant) flow applications, the existence of highly efficient thermal systems presents an important aim of most designers and researchers. This will bring about some saving in volumes and reduces overall weights. In practice, several methods and techniques have been utilized to achieve this goal. Among these techniques is the inclusion of flow of refrigerant through roughened tubes.

Several experimental studies have investigated evaporation inside smooth tubes as (Kandlikar 1998) and (Kandlikar 1990). Kandlikar (1998) presented a theoretical analysis model to predict the heat transfer coefficients for different mixtures as (bezene/methanol, R-23/R-13, and R-22/R-12) and Kandlikar (1990) developed a simple correlation for predicting saturated flow boiling heat transfer coefficient inside horizontal and vertical smooth tubes.

Many researchers beginning with (Altman et al. 1960), (Schlager et al. 1990) and (Nae-Hyun Kim and Webb, 1993) have investigated the heat transfer augmentation and pressure drop in internally finned tubes.

Rough surfaces are commonly used to augment heat transfer especially for single-phase flow. Roughening the boiling surface is one obvious method of providing a potentially greater number of nucleation sites. Many methods of providing surface roughness have been used in boiling experiments. In some studies, the surface has been roughened by using various grades of emery clothes or by chemical treatments. Nucleation occurs at gas-filled cavities in the heating surface and the superheat, at which nucleation occurs, depends critically on the size and distribution of these cavities and on the temperature gradient away from the heating surface. For forced refrigerant flow through evaporator the methods for improve the performance lies in the bubbly flow region depend on achieving nucleation at a lower superheat than with a normal smooth surface. Zurcher et al. (1999) and Wambsganss et al. (1993) illustrated that the heat transfer coefficient on the wetted portion of the tube is comprised of the nucleate boiling and the convective boiling contributions.

An experimental study carried out by Bier et al. (1979) to investigate the heat transfer for five horizontal copper plates with different surface finish to boiling refrigerants R-11 and R-115. They showed that an increase in the heat transfer coefficient with the increase in the roughened surfaces parameters with both mechanical and chemical treatment.

Kaneyasu Nishikawa et al. (1982) studied the effect of pressure and the effect of surface roughness on boiling heat transfer by using freons R-21, R-113 and R-114. It has become clear that the heat transfer coefficient remarkably increases at high-roughened surfaces and refrigerant pressure.

Shreif and Osman (2000) carried out an experimental work to study the effect of swirl flow on average heat transfer coefficient along the copper tube with a constant heat flux. The test section is a horizontal electrically heated tube with a constant heat flux. They investigated a general correlation for smooth tube average heat transfer coefficient as a function of mass flux (G) and heat flux q as:

$$\overline{h}_{sm} = a\, q^{b} \qquad (1)$$

Where a and b are function of the mass velocity G, (kg/m^2.s),

$$a = 0.12355 + 0.0019\,(G)^{1.32}$$

$$b = 0.9073 - 0.01078\,(G)^{0.5}$$

The purpose of this paper is studding the effect of surface roughness by using mechanical and chemical methods on the heat transfer coefficient.

2. EXPERIMENTAL RIG AND INSTRUMENTATIONS

Figure (1) shows the experimental test rig that was used to study the effect of roughened tubes on the heat transfer coefficient. The test rig have the facility of changing the test section with seven tubes have different surface roughness one of them is the smooth tube. The experimental apparatus consisted of a horizontal copper test section with R-11 flowing in a closed loop as shown in Fig. (1). Smooth tube was used for calibrating the experimental setup and also to compare the enhancement obtained in heat transfer. All tubes were made of hard copper material. The refrigerant flow was measured by using a flow meter. To control the sub-cooling temperature at the exit of the water-cooled condenser by regulating the cooling water quantity before entering the refrigerant in the flow meter. The refrigerant flow was regulated in liquid phase by a manually operated throttle valve to a prescribed evaporating pressure (which equals approximately 1 bar pressure gauge). The ranges of the tested mass velocities are 17, 33, 47 and 60 kg/m^2.s. The test section is heated by means of hot water flow in the annulus between the test tube and the outer insulated steel tube for a length of 440 and inner diameter of 38.2 mm. The experiments generally proceeded by increasing heated capacity for the heated section by increasing the circulating water temperature and water rate. The hot water circuit, Fig. 1, comprises the components: electrically heated water tank with thermostatic heater (1.5 kW) has a capacity of approximately 30 liters (its base diameter 250 mm and height 600mm), water circulating pump, and two throttle valves. The rate of water flow was established by proper bypassing of the main circuit. A schematic drawing of the experimental evaporator is shown in Fig. 2. The test section was carefully centered inside the outer steel tube with sets of three screws at three intervals along the length. To minimize the radial heat loss, the outside steel tube covered by 5-mm asbestos, thickness, 25-mm glass wool thickness and aluminum foil. The outer and inner insulation surface temperatures are measured for different axial positions along the tube axis to calculate the radial heat loss. Heat transfer tests were run over a range of heat fluxes covering values from 1100 to 17950 W/m^2. Figure 2 shows a cross-section of the horizontal test-section, which was made from a copper tube of 10.2 mm-inner diameter, 1.25-mm thickness and 440 mm long. At each heated water mass flow rate setting (is measured by collecting the discharge water in a calibrated vessel in a certain time), data were recorded concerning the test-section refrigerant flow rate, outer copper tube wall temperatures, refrigerant bulk (saturation) temperature and the inlet and exit heating water temperatures.

Heat input was determined from the measured water heat lost as corrected for heat loss through outer wall, based on the average inner and outer insulation temperatures. Measurements are performed at five axial stations along the test-section of 110 mm apart. Calibrated thermocouples were fixed at each test position (5 different test positions) with the hot junction of each is soldered onto the outside copper tube surface. The calibrated thermocouples were made from copper-constantan wires of 0.3-mm diameter. The reading were taken by using multi-meter with ± 0.25 °C accuracy. The effect of axial heat conduction in the copper cylinder was confirmed to be much smaller than the radial heat conduction by the heat conduction calculation, Shinju Suzuki and Satoshi Kumagai (1996).

The test facilities and the methods of calculation of the tested parameters are described in detail in Sherif and Osman (2000).

The tested tubes have seven different surface roughness, one of them is the smooth one has $R_a = 0.05$ μm, (arithmetic average height). Two different types of surface roughness tubes with similar diameters were made. The first one is made by using an emery clothes of different roughness to obtain R_a values of 0.2, 0.6 and 1.5 μm. The second tubes surfaces is performed by using acid pitted for the values of $R_a = 0.1$, 0.25 and 0.7 by filling the tubes with a nitric acid for different times intervals. The chemical treatment is used instead of mechanical process for bends and long pipes. The tubes surface roughness are measured by the recorder traces of the profilometer. Samples of the measured surfaces are shown in Fig. 3. Also, photoghraphs for the tested surface roughness by optical microscope are shown in Fig. 4. The calculated above parameters for surface roughness are compared with that of smooth tube.

3. EXPERIMENTAL RESULTS AND DISCUSSION

The average heat transfer coefficient $\bar{h}$ is calculated by using the experimental data at different heat flux, q, and different mass velocity, G. It was plotted for smooth tube against the heat flux, q, for the four tested values of mass velocity G, Fig. 5. The heat transfer coefficient in the tested range increases with an increase of both heat flux and mass flows rate. To check the validity of the present results, it is necessary to compare the present results with those of the previous workers. This result agrees with the obtained results by Sherif and Osman (2000). A comparison of the predicted correlation by Sherif and Osman (2000) for smooth electrical heated tube with R-11 (correlation (1)) and the present experimental results is shown in Fig. 6. It can be seen that a good level of agreement is obtained for the experimental results with correlatiion (1) within 10% deviation.

Of special interest for studies of heat transfer augmentation is the question of how much the heat transfer coefficient is increased relative to an equivalent smooth tube at similar conditions. The effect of heat flux on average heat transfer coefficient for roughened tubes by emery clothes and by chemical treatment is shown in Figs. 7 and 8 respectively. There are significant increases in the roughened tubes average heat transfer coefficient compared to the smooth flow. The increase was much higher at higher values of roughness (R_a) as a result of increasing the nucleation sites.

Generally, the effect of surface roughness on average heat transfer coefficient can be combined in figure (Fig. 9) for the two methods of roughened tubes. It can be seen that the enhancement in heat transfer coefficient was as high as 107 percent for $R_a = 0.1$ μm and about 121 percent for $R_a = 1.5$ μm as compared with smooth tube flow, depending on the increase in the nucleation sites.

Furthermore, the present experimental data for roughened tubes combined with the smooth tubes data, Fig. 9, can be correlated to relate the enhancement in average heat transfer coefficient ($\overline{h}_r / \overline{h}_{sm}$) at the same heat flux with the arithmetic average height, R_a, as:

$$\overline{h}_r / \overline{h}_{sm} = 0.9559 + 0.314\,(R_a)^{0.6} \qquad (2)$$

where $\overline{h}_r$ is the roughened tubes average heat transfer coefficient.

$\overline{h}_{sm} = a\,q^b$ (correlation (1)) at the same heat and mass fluxes.

This correlation is valid in the tested ranges of R_a. Equation (2) gives good agrement with the present experimental results in Fig. 9 within ± 10 % deviation as shown in Fig. 10.

A limited number of literatures were found for mechanical and chemical roughened tubes. The comparison between the present and the previous experimental results are shown in Figs. 11 and 12. Figure 11 shows that the present result for emery clothes give good agreement with Bier et al. (1979) and Nishikawa et al. (1982) results. Also, similar trends for the present results and Bier et al. (1979) results are obtained Fig. 12. The differences in results in Figs. 11 and 12 are as a result of changing of the tested pressure and operating conditions.

The effect of heat flux q on the temperature difference between the tube average surface temperature and average fluid bulk temperature for smooth and roughened tubes are shown in Figs.13 and 14. There is a significant increase in the flow temperature difference with the increase in the heat flux. The average temperature difference decreases with the increase in the refrigerant mass velocity, Fig.13. Also, there is a decrease in the roughened tubes flow temperature difference with the increase in R_a, Fig. 14, as a result of increasing the convective heat transfer coefficient due to the increase in nucleation sits. Figure 15 shows that the average temperature difference increases with the increase of average heat transfer coefficient as a result of the increasing rate in heat flux is greater than the increase of each average temperature difference and average heat transfer coefficient. Data given in Fig. 15 can be correlated to relate the average roughened temperature difference to smooth value ratio ($\Delta T/\Delta T_{sm}$) as function of arithmetic average height, R_a, as:

$$\Delta T/\Delta T_{sm} = 1 - 0.974\,\ln(R_a/R_{asm}) \qquad (3)$$

Where the average temperature differences ratio ($\Delta T/\Delta T_{sm}$) is calculated at the same average heat transfer coefficient and mass flux.

ΔT is average roughened temperature difference.

ΔT_{sm} is the average temperature difference for smooth tube and can be calculated at any value of mass velocity G and average heat transfer coefficient $\overline{h}_{sm}$ from Eq. (1) by replacing q by ($\Delta T_{sm} \times \overline{h}_{sm}$).

Figure (16) shows a comparison between the present correlation (3) and the present experimental results. The experimental data were found to correlate to Eq. (3) within ±11%.

Also, it is clear from this figure that the average temperature difference ratio ($\Delta T/\Delta T_{sm}$) calculated at the same $\bar{h}$ is independent on the average heat transfer coefficient for the tested range of R_a.

The local heat transfer coefficient versus quality for the tested six roughed tubes and the reference smooth tube are compared in Fig. 17 at the same heat flux and mass flux. Heat transfer coefficient are more dependent on heat flux in regions of lower quality, which represents a bubbly flow region, as compared to higher quality, which represents a mist flow region. For each heat flux case, the heat transfer coefficient decreases at a certain quality and then increases approximately linearly with quality. The same behavior obtained by Reid et al. (1991). As shown in Fig. 17 there is an enhancement in the roughed tube heat transfer coefficient compared to the smooth flow. The increase in heat transfer coefficients was much higher at higher values of surface roughness, but the exact enhancement factor cannot be determined because of changes in the heat flux values.

4. CONCLUSIONS

1. There are significant increases in the roughed tubes average heat transfer coefficient compared to the smooth flow. The increase was much higher at higher values of roughness (R_a) as a result of increasing the nucleation sites. The enhancement in heat transfer coefficient was as high as 107 percent for $R_a = 0.1$ μm and about 121 percent for $R_a = 1.5$ μm as compared with smooth tube flow, depending on the increase in the nucleation sites.

2. The enhancement in average heat transfer coefficient as function of arithmetic average height, R_a, can be correlated as:

   $$\bar{h}_r / \bar{h}_{sm} = 0.9559 + 0.314\,(R_a)^{0.6}$$

   where $\bar{h}_r$ is the roughed tubes average heat transfer coefficient.
   $\bar{h}_{sm} = a\,q^b$ (correlation (1)) at the same heat and mass fluxes.

   The investigated correlation gives good agreement with the present results within ±10%.
3. There is a decrease in the roughened tubes flow temperature difference with the increase in R_a as a result of increasing the convective heat transfer coefficient due to the increase in nucleation sites.
4. The average temperature difference ratio ($\Delta T_r/\Delta T_{sm}$) calculated at the same $\bar{h}$ is independent on the average heat transfer coefficient for the tested range of R_a.
5. The average roughened temperature difference to smooth value ratio ($\Delta T_r/\Delta T_{sm}$) can be correlated form the present experimental results as function of the tested arithmetic average height, R_a, within ±11% deviation as:

   $$\Delta T/\Delta T_{sm} = 1 - 0.974\,\ln(R_a/R_{asm})$$

   Where the average temperature differences ratio ($\Delta T/\Delta T_{sm}$) is calculated at the same average heat transfer coefficient and mass flux.
6. Heat transfer coefficient are more dependent on heat flux in regions of lower quality, which represents a bubbly flow region, as compared to higher quality, which represents a mist flow region.

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NOMENCLATURE

D_i	The inner diameter of the test-section (mm).
D_o	The outer diameter of the test-section (mm).
G	Mass velocity (flux) of refrigerant (kg/m^{2}s).
h	Convective heat transfer coefficient (W/m^2. K).
L	Heated length of the test-section tube (m).
Q	Input power by the hot water, (W).
q	Heat flux (W/m^2).
R_a	Arithmetic average height, (μm).
R_p	Maximum height of the profile above the mean line, (μm).
T_{si}	Tube inner surface temperature (K).
T_{so}	Tube outer surface temperature (K).
T	Temperature (K).
X	Magnification ratio.

x Refrigerant quality.

Greek letters

Δ	Difference.
ρ	Density.

Subscripts

m	Bulk temperature.
r	Roughened tubes.
si	Inner tube surface
sm	Smooth tube
so	Outer tube surface

Superscripts

$\overline{}$	Arithmetic Average value along the tube length

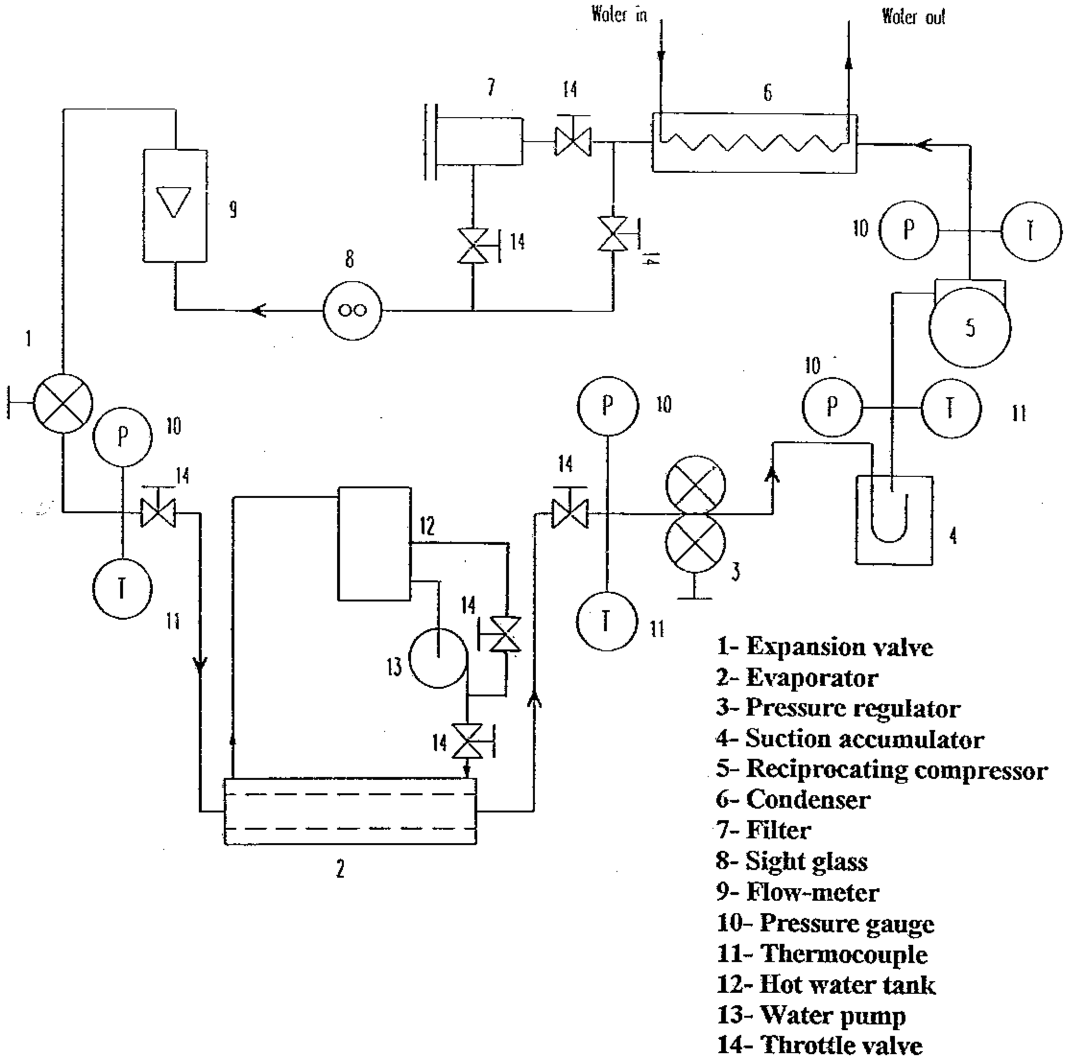

Fig. 1: The Experimental Test Rig.

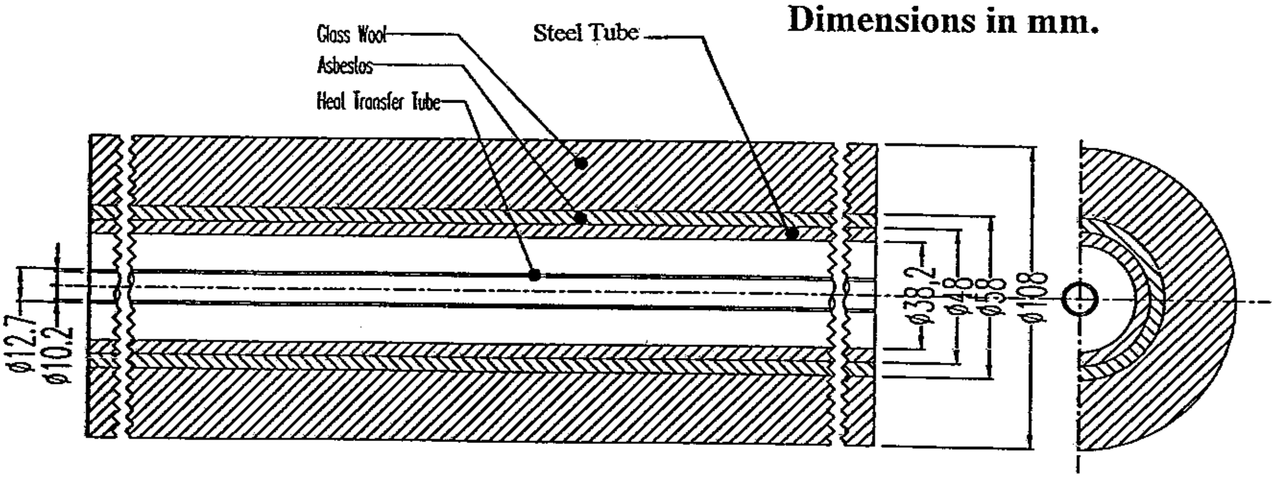

Fig. 2: Details of The Tested Tube.

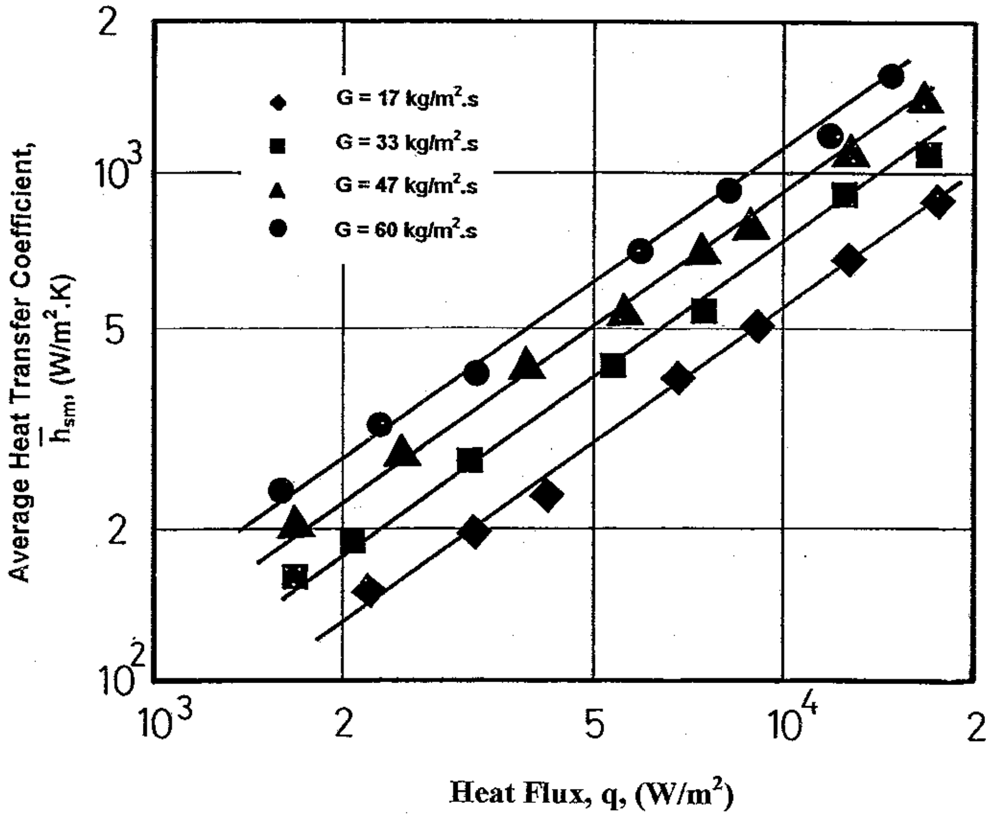

Fig. 5: Effect of Heat Flux on average Heat Transfer Coefficient For Smooth Tube ($R_a = 0.05\ \mu m$).

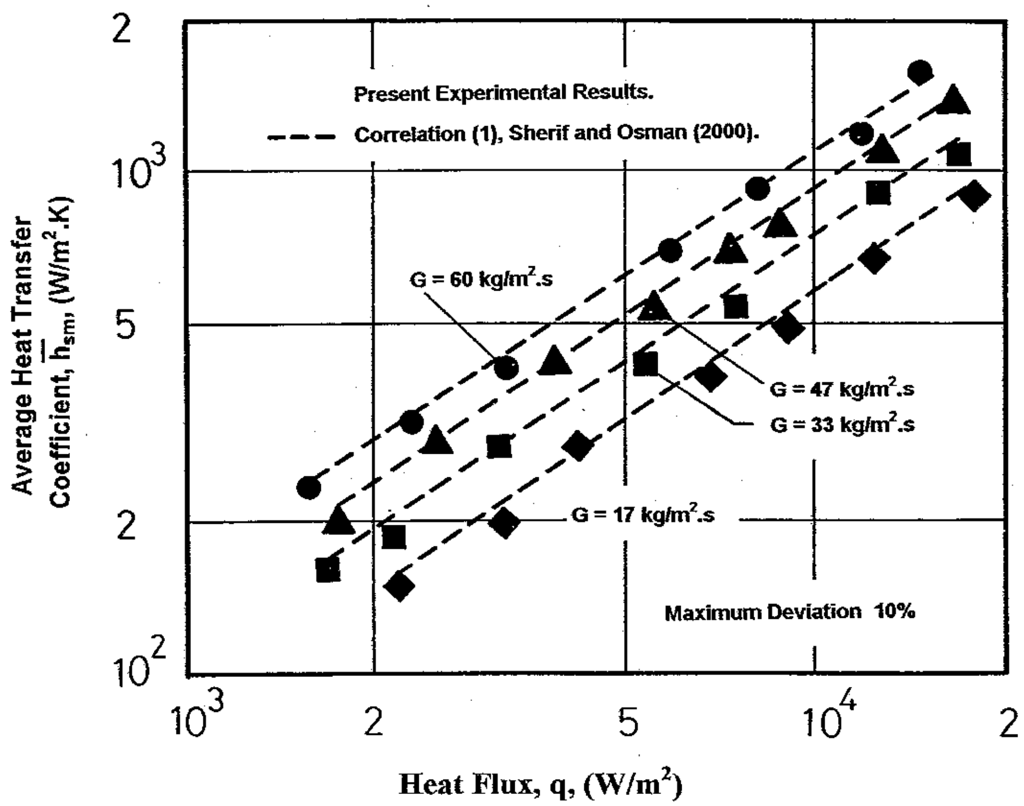

Fig. 6: Comparison Between The Present Experimental Results and Correlation (1) For Smooth Tube ($R_a = 0.05\ \mu m$).

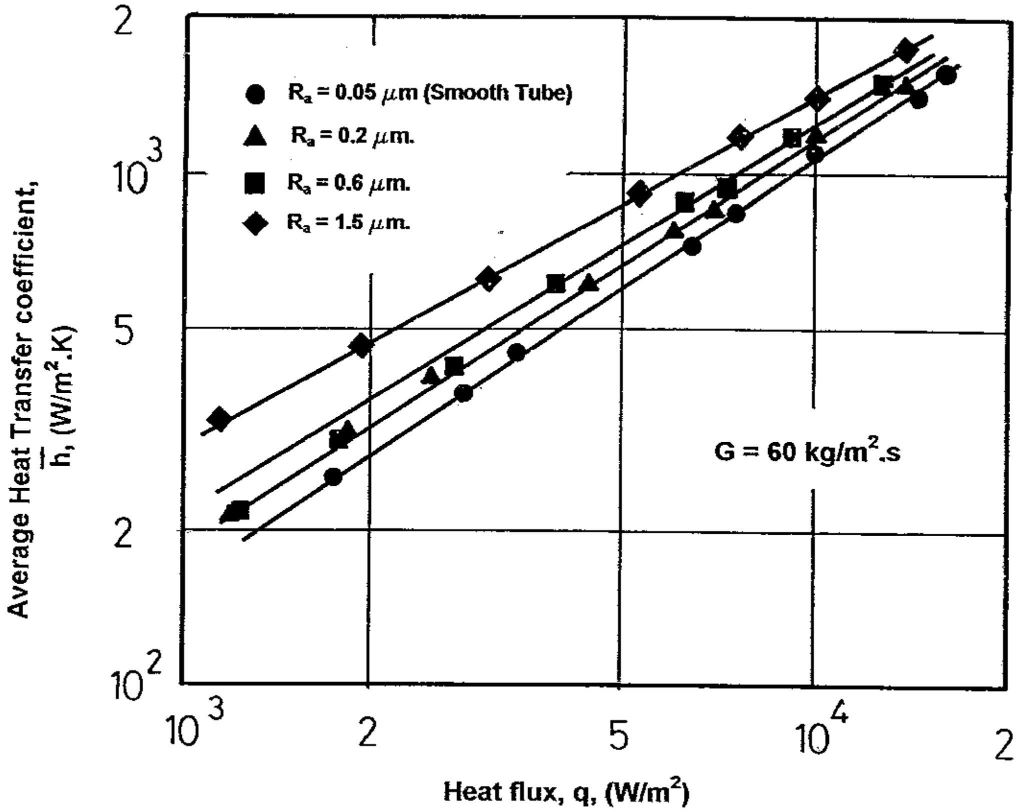

Fig. 7: Variation of Average Heat Transfer coefficient Versus Heat Flux.

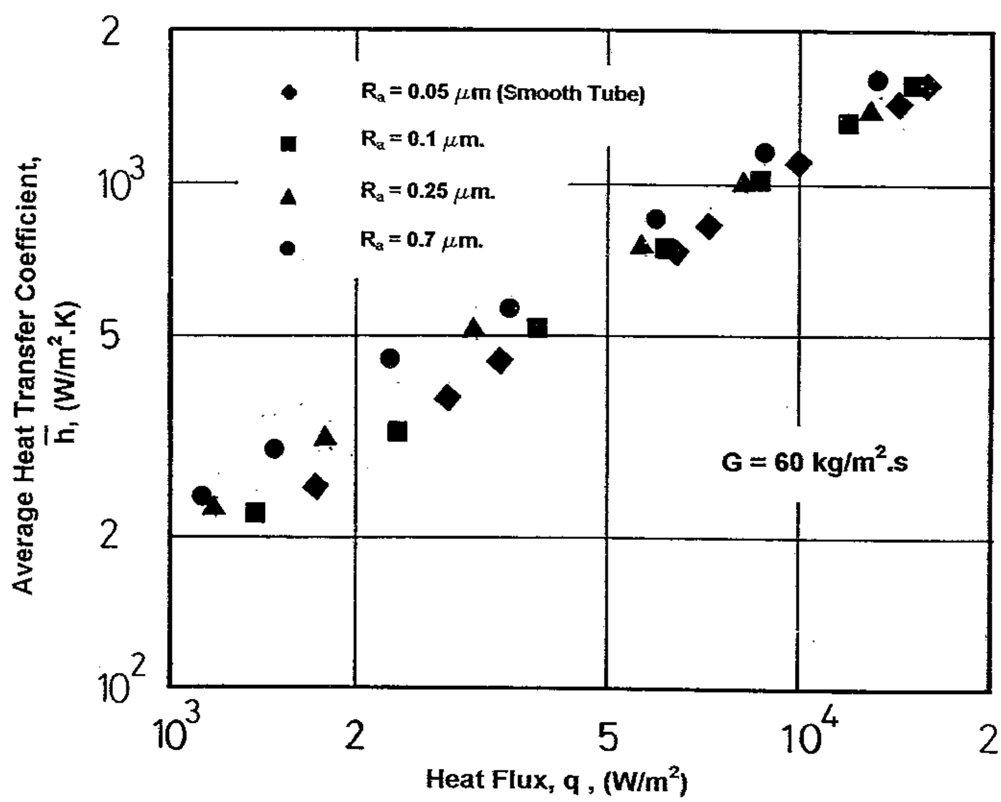

Fig. 8: Variation of Average Heat Transfer Coefficientt Versus Heat Flux For Chemical Treatment.