HEAT TRANSFER ENHANCEMENT IN A TANGENTIALLY INJECTED SWIRL FLOW THROUGH A ROUGHENED TUBE

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ABSTRACT

Heat transfer enhancement and flow friction were experimentally investigated for a swirl flow that enters tangentially in a tube roughened by a helical wire. The effects of tube roughness intensity, tangential to total momentum flux ratio and Reynolds number were studied. Measurements were conducted in a horizontal tube roughened with helical-wires at two different diameter to pitch ratios of 0.025 and 0.050, respectively, within a range of Reynolds number from 10×10^3 to 60×10^3, and at different values of the tangential to total momentum flux ratios of 1.73, 2.94, 3.46 and 5.88, respectively. An enhancement in the heat transfer coefficient accompanied with excess friction was observed within the investigated ranges of the different parameters. Also, it was found that the efficiency index is critically dependent on the roughness Reynolds number, momentum flux ratio and tube roughness intensity and it is almost less than that for the rough plain tubes. New correlations for Nusselt number enhancement ratio and the friction factor ratio were obtained as functions of momentum flux ratio, d/p ratio and Reynolds number .

KEYWORDS:
Tangential injected swirl flow-Roughened tubes-Flow friction- Heat transfer enhancement.

1. INTRODUCTION

Swirl flow is considered one of the techniques that has been used successfully to enhance heat transfer in single-phase flow forced convection at a moderate range of Reynolds number up to 50×10^3. Also, rough surfaces are commonly used to augment heat transfer and it has been found that roughness provides a substantial augmentation in the heat-transfer. In fact, a lot of published researches have been carried out to evaluate the enhancement in heat transfer either by swirl or roughness individually and few of them take both the two effects (swirl and roughness) in considerations. Therefore, the combined effect of both swirl and roughness was proposed to be investigated within the present experimental study. In accordance, the main objective of the present work is to explore the effect of the tangentially-injected swirl flow in roughened tubes on heat transfer and pressure drop characteristics.

Heat transfer and flow characteristics in a tangential injected swirl flow in smooth tubes at which a part of the fluid was entered axially and the remainder was injected tangentially at different momentum flux ratios have been investigated by Dhir and Co-Worker[1-4]. Generally, it was found that the tangential to total momentum flux ratio (the ratio between the tangential momentum flux associated with the injected fluid and the total axial momentum flux) is the dominant parameter that affect the enhancement in heat transfer.

Experiments were performed by Dhir et al.[2] to study the enhancement in the heat transfer for air flow forced convection using single and multi-stage tangential injection. The results showed that the enhancement in heat transfer is strongly dependent on the momentum flux ratio but is weakly dependent upon Reynolds number. Also, for multi-stage tangential injection, made by introducing swirl flow at different axial locations, further enhancement in heat transfer was found.

Change and Dhir [3] carried out an experimental study for the heat transfer enhancement for a turbulent flow with tangentially injected swirl flows in vertical tubes. It was found that the momentum flux ratio is the most effective parameter that enhances heat transfer while Reynolds number affects it weakly and the effect of the fluid Prandtl number is very weak. Also, the injector tube diameter and number of injectors do not affect the heat transfer enhancement. The effect of injection include swirl flow on single and two phase heat transfer using water as a working fluid was investigated by Guo and Dhir [4]. It was found that, on a constant pumping power basis, a net enhancement of about 20% could be achieved experimentally at a momentum flux ratio of about 9.6.

Abdel-Moneim et al. [5] carried out an experimental study to investigate the enhancement in heat transfer by using tangential injection swirl in a two-phase gas-solids suspensions flow. The effects of solids-loading-ratio, tangential to total momentum flux ratio and Reynolds number were investigated. The tangential to total momentum flux ratio and Reynolds number were found to be the most dominant parameters that affect the heat transfer coefficient for both single and two-phase flows. Also, for single-phase air flow, Nusselt number enhancement ratio up to 1.6 was obtained corresponding to five fold increase in the pressure drop.

Aoyama et al. [6] devised a turbulence promoter composed of a row of thin plates twisted by 90 degree alternately in different directions. The heat transfer coefficient and pressure loss were measured in an air flow through a circular smooth tube around which the promoter was set. Six types of promoters were examined and the heat transfer coefficients were enhanced. The result was explained by two enhancement mechanisms, namely some swirl motions and raised turbulence intensity, as the flow proceeds along the twisted plates, many axial vortices are produced and a flow-separation occurs at the lateral edge of the plates. The heat transfer performance was evaluated in terms of the area goodness factor and the increase in Nusselt number under the same pumping power.

The effect of curvature-induced secondary motion on turbulent heat-mass transfer and pressure drop characteristics were investigated by Ohadi [7] for a straight tube situated downstream of bends with circular cross section. Bends with turning angles of 0, 30, 60, and 90 deg were employed. It was found that with the bend in place, the effect of curvature-induced secondary motion resulted in enhancement or reduction of heat-mass transfer coefficients accompanied with an increase in the friction factor for all cases studied.

Durmus and Ayhan [8] investigated the heat transfer characteristics for water flowing in axially rotating pipe with a swirl motion at different rotating speeds. The experiments were carried out for 0, 250, 520 and 850 rpm. The swirl motion was produced by four opposite positioned slots located at the pipe entrance length. The augmentation of heat transfer was found to be a function of the slots number, rotating speed and Reynolds number and it can be as much as three times the value for non-swirl flow. Experiments were also, conducted by Durmus and Ayhan [9] to evaluate the usage of rotating pipes with helical deformed finning on the outer surface of a hot-cold water heat exchanger. To create swirl flow, four slots were made at inlet. It was found that the effectiveness of the heat exchanger is improved and the intensity of swirl is increased by increasing the rotating speed for fixed dimensions of slots within a range of Reynolds number from 3000 to 20000.

Experiments were carried out by Abdel-Moneim and El-Shamy [10] to investigate the heat transfer enhancement and the frictional pressure drop for a single-phase flow of air in a tube roughened with a helical-wire with different pitches within a range of Reynolds number from 20×10^3 to 70×10^3. The results show that a maximum increase in the average Nusselt number of about 70% was observed corresponding to six fold increase in the friction factor for tube with concentrated roughness (d/p=0.05). Also it was found that the Reynolds number and the pitch of the helical rib are the most dominant parameters that affect heat transfer and flow friction coefficients.

2. EXPERIMENTAL APPARATUS

An experimental set up was designed and constructed to make it available to carry out the present heat transfer and flow measurements. It consists mainly of: air passage, test-section including the roughness coil, tubular injectors and measuring instruments as shown in Fig.(1-a).

The test-section was made of a horizontal circular brass tube of 40 mm inner diameter, 3.5 mm thickness and 3 m long, representing a length-to-diameter ratio of about 75.0. Also, a hydrodynamic entry length of 2.5 m was allowed upstream of the tubular injectors which were located individually at the test-section inlet. The test section was heated by means of an electric heater and also a guard heater was used to prevent heat loss. The roughness was formed by winding a long wire of stainless steel of 1 mm diameter with a predetermined pitch on a removable wetted-wooden rod of 37.5 mm diameter using a precision lathe. The roughness coil was fitted tightly on the tube inner surface by its expansion due to the winding stiffness as shown in Fig.(1-b). The helical pitch was unaffected by coil expansion due to the small clearance between the removable rod and the tube inner diameter. Also, the wooden rod was preheated and dried before removing process to increase clearance and make it easy that avoids disturbance of the helical pitch due to rod end sliding.

Air as a working fluid was injected tangentially through a set of tubular injectors with two different injection-tube inner diameters of 16.5 and 21.5 mm, respectively. The injectors were fabricated and assembled such that two injection-tubes with 150 mm length for each were fixed tangentially and opposed diametrically on a base tube of 300 mm length and 40 mm inner diameter (the same inner diameter of the test-section). Also, the injection-tubes were fixed perpendicular to the base tube axis. The injector's base tubes were connected individually to the inlet end of the test tube via a short flexible connection. Details for one of the used injectors are shown in Fig.(1-c).

Thermocouples were used to measure the temperature of the inner surface of the test-section at 40 different test locations distributed axially along the test-section wall with relatively high concentration at entrance. At each test location, at least two diametrically opposite top-bottom, thermocouples were fixed on the tube wall to check the axisymetry of the flow. The used thermocouples were made of copper-constantan wires of 0.5 mm diameter and connected to a digital thermometer having one decimal point (sensitive to $0.1^{o}C$). The air flow radial temperature profile was measured via radiation shielded thermocouples mounted on vertical traverse mechanisms at tube inlet and exit, respectively. To avoid flow disturbance, each of the thermocouples probe with its shield was resided in a tubular cavity that was assembled normal to the test tube such that the probe was presented only in the flow field at the instant of reading.

The air flow rate was measured by a calibrated orifice and the readings of its head difference was indicated by an inclined differential manometer. Also, a system of U-tube manometers was used for measuring the axial pressure distribution along the test section.

3. EXPERIMENTAL PROCEDURE AND METHOD OF CALCULATIONS

Experiments were conducted using injectors with either double or single entry condition to perform different momentum flux ratios at different roughness intensities.

The momentum flux ratio, which is defined as the ratio between the momentum flux of the injected flow through the injectors and the total momentum flux through the test-tube, can be expressed as:

$$\frac{M_t}{M_T} = \frac{m_t^2 A}{m_T^2 A_j} \tag{1}$$

In fact, the present experiments were carried out while the total mass flow rate of working fluid was injected tangentially to the test section. Hence the ratio m_t/m_T is equal to unity and consequently the momentum flux ratio can be simply expressed by the area ratio A/A_j. Therefore, values for the momentum flux ratios of 1.73 and 3.46 were assigned corresponding to double and single entry injectors of 21.5 mm diameter while values of 2.94 and 5.88 were assigned for double and single entry injectors of 16.5 mm diameter, respectively.

Moreover, the test tube was divided into 16 segments with different axial lengths with high concentration at the entrance region. The local heat transfer coefficient for each tube segment was calculated simply by,

$$h_x = \frac{q''}{T_{w,x} - T_{f,x}} \tag{2}$$

where,

- q'' is the heat flux, calculated from the readings of the main heater circuit (voltage, current and power factor) and the tube surface area.
- $T_{w,x}$ is the local inner surface mean temperature of the tube segment (the mean value between the top and bottom thermocouples readings)
- $T_{f,x}$ is the local flow mean temperature at the tube segment, calculated from the heat balance for each segment and checked with the mean value calculated by integrating the measured radial temperature profiles at tube inlet and exit, respectively.

The local Nusselt number based on the local heat transfer coefficient was calculated by,

$$Nu_x = \frac{h_x D}{k} \tag{3}$$

and the average Nusselt and Stanton numbers were calculated by averaging the local values as follows:

$$\overline{Nu} = \frac{1}{L}\int_{x=0}^{L} Nu_x dx \tag{4}$$

$$St = \frac{\overline{Nu}}{Re\,Pr} \tag{5}$$

where, Re is the flow Reynolds number which was calculated by,

$$Re = \frac{4m}{\pi D \mu} \tag{6}$$

and Pr is the Prandtl number.

Also, the air properties that included within the calculated parameters were taken at an average temperature of $(T_{in}+T_{out})/2$.

The Fanning friction factor in terms of the flow frictional pressure drop along the test section was calculated by,

$$F = \frac{\Delta P}{2\rho V^2} \frac{D}{L} \tag{7}$$

4. RESULTS AND DISCUSSION

To check the validity and establishment of the present experimental technique, preliminary experiments were carried out for the following three cases:

a) smooth plain-tube flow,

b) rough plain-tube flow,

and c) tangentially-injected swirl flow in a smooth tube.

These preliminary experiments were conducted within a range of Reynolds number from 10×10^3 to 60×10^3 and in the case of rough plain-tube flow, two different d/p ratios of 0.025 and 0.050 were tested. The heat transfer results of the preliminary experiments for the cases of smooth-tube flow were compared with pervious correlations as shown in Fig.(2-a). The comparison shows an excellent agreement with that of Sparrow [13] for plain-tube flow and that of Abdel-Moneim et al.[5] for tangentially injected swirl flow with M_t/M_T ratio of 5.88, respectively, thereby establishing the used experimental technique. Beside establishment of the present experimental technique the preliminary experiments were carried out to perform references for comparative evaluation of the present results of the tangentially injected swirl flow in the roughened tube.

The main groups of experiments were carried out for the tangentially injected swirl flow in the roughened tube for two different d/p ratios at different M_t/M_T ratios and at different Reynolds numbers. The variations of the average Nusselt number for the different M_t/M_T ratios at the two d/p ratios of 0.025 and 0.050 are shown in Fig.(2-b,c). In fact the inlet swirl due to tangential entry may be conserved for a long axial distance due to the presence of the secondary flow that created by the helical-effect of the roughness-coils. This may influences the velocity distribution, flow turbulence and increases the actual flow pass. This in accordance enhances the heat transfer coefficient. This effect was drawn from the axial distributions of the local Nusselt number which indicate longer thermal developing lengths compared with that for plain tube flow.

The influence of the tangential flow entry condition (single or double entry) represented by the ratio M_t/M_T on the average Nusselt number is also, depicted in Fig.(2). It was observed that, when using single entry injectors the enhancement in heat transfer coefficient is more significant compared with that when using double entry injectors at the same injected mass flow rate. This in fact due to, for the same injected mass flow rate, the usage of single entry injector makes the flow to enter with a velocity of 4 times that of a similar double entry injector (with the same diameter). This higher inlet flow velocity improves the inlet swirl and in accordance keeps the helical flow intensity for a longer axial distance that leads to an enhancement in the heat transfer. This effect becomes more significant at higher roughness intensity (for d/p=0.050) as shown in Fig.(2-c). To evaluate the heat transfer enhancing effect of both the condition of entry and the swirl intensity, Stanton number was calculated and plotted versus Reynolds number as shown in Fig.(3). Generally, the results indicate normal enhancement in the heat transfer over that of the plain tube.

On the flow friction side, higher values for the flow friction factors were observed, in general, compared with that for plain tube flow regardless the injector size and the condition of entry as shown in Fig.(4). Also, it was found that the increase in the tangential inlet swirl increases the flow friction with a higher extent compared with that results by increasing the tube roughness.

The flow friction and heat transfer data were formulated in terms of Nikuradse's friction similarity factor F^+, cited in Schlichting [11], and Dipprey and Sabersky's [12] heat transfer similarity factor H^+ which have been reported by Abdel-Moneim and El-Shamy [10] and expressed as follows:

$$F^+ = \sqrt{\frac{2}{F}} + 2.5\ \ell n\left(\frac{2d}{D}\right) + 3.75 \tag{8}$$

$$H^+ = \left(\frac{F}{2\ St} - 1\right) / \sqrt{\frac{F}{2}} + F^+ \tag{9}$$

These factors were then plotted against the roughness Reynolds number Re^+ in which the rib height was represented by the roughness wire diameter and it was calculated as:

$$Re^+ = Re\frac{d}{D}\sqrt{\frac{F}{2}} \tag{10}$$

Figure (5) shows the variation of friction similarity factor F^+ with the roughness Reynolds number Re^+ for the two different d/p ratios at different M_t/M_T ratios. The present results indicate that F^+ is critically dependent on Re^+ in general. It reaches a maximum value at a value of Re^+ depending on both M_t/M_T and d/p ratios. Also, the maximum value of F^+ decreases with the increase in either the ratio of d/p or M_t/M_T. Moreover, the value of Re^+ at which F^+ reaches its maximum decreases with increasing the ratio M_t/M_T. The present data of F^+ were introduced in calculating the heat transfer similarity factor H^+ and the effect of the momentum flux ratio for the two different d/p ratios on H^+ is shown in Fig.(6). It was found that H^+ is slightly affected by the M_t/M_T ratio and its envelope of variation was linearly correlated by,

$$\mathbf{H^+ = \alpha \pm \beta} \tag{11}$$

where,

α is the base value and it is a function of d/p ratio,

$\boldsymbol{\alpha = 30 - 200\ (d/p)}$

and β is the fluctuation and it is a function of both d/p ratio and Re^+,

$\boldsymbol{\beta = 7.5 - 0.0416\ Re^+}$ for d/p = 0.025

and $\boldsymbol{\beta = 12.5 - 0.0694\ Re^+}$ for d/p = 0.050

This correlation is valid within $\pm 5\%$ maximum deviation with the present experimental data within the investigated range of Reynolds number from 10×10^3 to 60×10^3 and for momentum flux ratio ranging from 1.73 to 5.88.

The efficiency index which is defined by the ratio between the Stanton number enhancement ratio ($St/St_{0,p.t.}$) to the friction factor ratio ($F/F_{0,p.t}$), $\eta = \dfrac{\mathbf{St/St_{0,p.t.}}}{\mathbf{F/F_{0,p.t.}}}$ was employed to evaluate the performance benefits of the rough tube and the results are shown in

Fig.(7). It was found that the efficiency index for the plain tube flow is almost greater than that for the tangentially injected flow regardless the values of Re^+, d/p and M_t/M_T ratios. Also, the efficiency index is critically dependent on Re^+ and it reaches its maximum at a value of Re^+ depending on both d/p and M_t/M_T ratios. However, the usage of single entry injectors enhances the heat transfer it also increases the flow friction with higher extent compared with that of double entry ones. Therefore, the efficiency index indicates the contrast. Figure (7) shows that the values of η for the double entry injectors which are in general grater than that of the single ones. This effect was observed for the whole investigated ranges of d/p and M_t/M_T ratios. This in fact may attributed to the higher rate of momentum change due to the higher incoming-flow velocity in case of using single entry injectors thus leads to further increase in the flow friction and pressure drop.

The Nusselt number enhancement ratio ($Nu/Nu_{0,\ p.t}$) and the friction factor ratio ($F/F_{0,p.t.}$) were correlated utilizing the present experimental data and the following correlations were obtained:

$$\frac{Nu}{Nu_{0,p.t.}} = Re^{0.1634}\left(\frac{d}{p}\right)^{0.4547}\left(\frac{M_t}{M_T}\right)^{0.405} \tag{12}$$

$$\frac{F}{F_{0,p.t.}} = 27.0\left(\frac{d}{p}\right)^{0.5}\left(\frac{M_t}{M_T}\right)^{0.5} \tag{13}$$

These correlations are valid within $\pm$8% maximum deviation with the present experimental data within a range of Reynolds number from 10×10^3 to 60×10^3, for two helix wire diameter to helical pitch ratios of 0.025 and 0.050, and for momentum flux ratio ranging between 1.73 and 5.88.

5. UNCERTAINTY ANALYSIS

Generally, the accuracy of the experimental results is governed by the accuracy of the individual measuring instruments. Also, the accuracy of an instrument is limited by its minimum division. The uncertainties existed in both the heat transfer coefficient (Nusselt number) and in the flow velocity (Reynolds number) were estimated following the differential approximation method. For a typical experiment the maximum uncertainty in measuring the main heater input voltage, the main heater electric resistance, the wall temperature, the flow mean temperature and the tube inner surface area were 0.5%, 0.14%, 0.25%, 0.25% and 0.25%, respectively. These were combined in a summation form as: {2 (0.5%)+ 0.14%+ 0.25%+0.25%+0.25%} to give a maximum overall error of 1.89% in the reported heat transfer coefficient or Nusselt number. Moreover, a maximum uncertainty of 3.0% was estimated in the reported Reynolds number due to the measurement of the air discharge using an orifice instrumented with an inclined differential manometer.

6. CONCLUSIONS

In view of the present experimental results the following conclusions were drawn:

1- The swirl associated with the tangential entry is conserved for a long distance through a tube roughened with a helical wire due to the creation of the secondary helical flow for the investigated range of Reynolds number from 10×10^3 to 60×10^3.

2- A maximum Nusselt number enhancement ratio of about 3.15 corresponding to a 14.6 fold increase in the friction factor was obtained for the tube with d/p=0.050.

3- The efficiency index is critically dependent on Re^+ and it is almost less than that for plain tube flow.

4- Generally, the efficiency index for double entry injectors is greater than that for single entry injectors.

5- The efficiency index reaches a maximum value of about 0.25 at a value of Re^+ of about 70 for the rough tube with d/p=0.025 and for the case of double entry injector with D_j/D=0.537 and M_t/M_T=1.73.

6- New correlations were obtained utilizing the present results for the Nusselt number enhancement ratio and the friction factor ratio for the tangentially injected swirl flow in a tube roughened with helically-coiled wires.

7. NOMENCLATURE

SI system of units was applied for the whole parameters used in this paper.

A	cross-sectional area of the test-section
A_j	total cross sectional area of the injectors
C_p	working fluid (air) specific heat
D	inner diameter of the test-section
D_j	inner diameter of the injector
d	diameter of the roughness wire
h	heat transfer coefficient
k	thermal conductivity of air
L	tube length
M_T	total momentum flux rate of the flow
M_t	tangential momentum flux rate of the injected flow.
m	mass flow rate of air
m_t	total mass flow rate through the tube
m_T	mass flow rate of the injected flow
P	pressure
p	helical pitch
q"	heat flux
T	temperature
V	axial flow velocity
x	axial distance from tube inlet

Greek letters:

α	function used in correlation, Eq.(11)
β	function used in correlation, Eq.(11)
Δ	difference
μ	dynamic molecular viscosity
ρ	density
η	rough-tube efficiency index

Subscripts:

d	double entry injector
in	at tube inlet
j	for injectors
m	mean value
o	case of smooth tube flow
out	at tube exit
p.t.	case of plain tube
s	single entry injector
T	total
t	tangential
w	for tube inner surface
x	local value

Superscripts:

—	average value.
+	similarity factor

Dimensionless Terms:

d/p	roughness wire-diameter to the helical-pitch ratio
F	friction factor, $\dfrac{\Delta P}{2\rho V^2}\dfrac{D}{L}$
F^+	Nikuradse's friction similarity factor
H^+	Sabersky's heat transfer similarity factor
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
Re^+	roughness Reynolds number
St	Stanton number

REFERENCES

1- Guo, Z and Dhir, V. K., "Single and Two-Phase Heat Transfer in Tangential Injection-Induced Swirl Flow", Int. J. Heat and Fluid Flow, Vol. 10, No.3, pp.203-210, September 1989.

2- Dhir, V. K., Tung V. X., Change, F. and Yu, J., "Enhancement of Forced Convection Heat Transfer Using Single and Multi-Stage Tangential Injection," ASME HTD, Vol. 119, pp.61-68, 1989.

3- Change, F. and Dhir, V.K., "Heat Transfer Enhancement Turbulent Flow Field in Tangentially Injected Swirl Flows in Tubes.", ASME HTD, Vol. 256, pp. 37-48,1993.

4- Guo, Z. and Dhir, V.K.", Effect of Injection Induced Swirl Flow on Single and Two-Phase Heat Transfer.", ASME HTD, Vol. 81, pp.77-84, 1987.

5- Abdel-Moneim, S.A., Zahran, M.S., Berbish, N.S. and Abd-Rabbo, M.F. "Heat Transfer and Pressure Drop in a Tangentially Injected Swirl Flow through a Horizontal Tube", Eng. Research Jour., Vol. 2, PP. 220-231, Feb. 1995.

6- Aoyama, Y. Hijikata, K., Futagami, K. and Nomoto, Y. ,“ Turbulent Heat Transfer Enhancement by A Row of Twisted Plates”, Trans. JSME, Vol. 53, No. 487, pp.991-996, 1987.

7- Ohadi, M..M., “Turbulent Swirl Affected Heat Transfer and Pressure Drop Characteristics in A Straight Tube Situated Downstream of A bend”, Diss. Abst. Int. Science and Eng., Vol. 47, No. 8, P.299, Univ. of Minnesota, MN, USA, 1987.

8- Durmus, A. and Ayhan, T., “Heat Transfer Characteristics in Rotating Tubes with Swirl Generator”, ASME Pet. Div. Publ., (PD), New York, ASME Vol. 64, No. 8-3, pp.703-709, 1994.

9- Durmus, A. and Ayhan, T., “Model of Heat Exchanger of which Rotating Inner Tube with Helical Deformed Finning and Swirl Generator at Inlet”, ASME Pet. Div. Publ., (PD), New York, ASME Vol. 64, No. 8-3, pp.695-701, 1994.

10-Abdel-Moneim, S. A. and El-Shamy, A. R. "Heat Transfer and Flow Characteristics in Helically Rib-Roughened Tubes", Proceedings of 11th International Mechanical Power Engineering Conference (IMPEC 11), Vol. 1, pp. H60-H74, Feb.5-7, 2000.

11- Schlichting, H., “Boundary Layer Theory”, 7th ed., McGraw-Hill, New York, 1979.

12- Dipprey, D.F. and Sabersky, R. H., “Heat and Momentum Transfer in Smooth and Rough Tubes”, Int. J. Heat Mass Transfer, Vol. 20, pp.329-353, 1963.

13- Sparrow, E. M., Holman, T. .M. and Siegel, R. S., “Turbulent Heat Transfer in Thermal Entrance Region of A Pipe with Uniform Heat Flux”, Applied Scientific Research, Section A, Vol. 7, P.37, 1957.