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EXPERIMENTAL STUDY OF HEAT TRANSFER AUGMENTATION IN PIPES WITH HELICAL TURBULENCE PROMOTERS USING THE NAPHTHALENE SUBLIMATION TECHNIQUE

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Abstract. *This work presents an experimental study of the influence of internal helical turbulence promoters on the heat transfer enhancement in a circular pipe submitted to a forced convection turbulent flow of air. The naphthalene sublimation technique was used to determine the heat transfer coefficient based on the heat-mass transfer analogy. The performance of the turbulence promoters was investigated for two different aspect ratios and the Reynolds number was varied between 10000 and 60000. In addition, pressure drop measurements were made for each run.*

Keywords: *Turbulence promoters, Naphthalene sublimation technique, Heat-mass transfer analogy*

1. INTRODUCTION

Due to the large number of industrial applications of heat and mass transfer phenomena, as in nuclear and processing reactors, furnaces, boilers and heat exchangers, and the availability and cost of raw materials and energy resources, it is of great importance the development of techniques that lead to more compact and efficient heat transfer equipments.

In such case, many techniques have been developed and widely disseminated in the last years. Turbulence promoters such as twisted tapes, helical rib roughness or helically coiled wires inserted inside ducts can be employed either to reduce the size of an equipment or to increase its heat transfer rate. Theoretical and experimental works have been done to study the effects of twisted tapes and rib roughness, but the number of works about helical wires is still scant.

Kumar and Judd (1970) performed a review about heat transfer with coiled wire turbulence promoters and presented an experimental investigation of the effect of coiled wires of different aspect ratios. Analysis of the computed results indicated that heat transfer rate was increased by as much as 280%, though at the cost of much larger increase in frictional power loss.

Gee and Webb (1980) presented the experimental study of the single-phase forced convection in a circular tube containing a two-dimensional helical rib roughness. Sethumadhavan and Raja Rao (1983) presented results from experimental investigations of heat transfer and fluid friction in a duct tightly fitted with helical wire coil inserts of several aspect ratios. Chiou (1987) performed an experimental investigation of the augmentation of forced convection heat transfer in a circular tube using spiral spring inserts. By means of analysis of temperature fluctuations in a heated circular pipe with helical turbulence promoters, Blanco (1996) and Blanco and Möller (1996) show that the structure of the turbulent flow an the heat transfer near the wall are strongly affected by this type of device. More recently, the experimental results of Vicari (1996) and Vicari and Möller (1997 - a and b) presented the characteristics of the turbulent flow inside pipes with helical turbulence promoters.

In general, these works show the characteristics of the turbulent fluid flow and heat transfer, i.e., pressure drop, velocity and turbulence characteristics distribution, and confirm the fact that the introduction of this kind of devices increases the momentum, heat and mass transfer. They also show that helical turbulence promoters increase the main flow velocity and produce helical flow near the walls superimposed to an axially directed central core flow. Friction velocity increases more rapidly than turbulence intensities and Reynolds stresses, leading to lower dimensionless values of these quantities in smooth pipes.

Moreover, the helical wire turbulence promoter behaves like roughness elements which affect the velocity distribution, the turbulence level and the turbulent wall shear. These effects result in increased eddy diffusive heat transfer, increased frictional power loss and provide a more homogeneous flow.

The naphthalene sublimation method can be used to study, with good confidence, mass and heat transfer in many applications, but with certain restrictions. This method is particularly useful in complex flows and geometries and for flows with large gradients in wall transport rate. Mass transfer boundary conditions, analogous to isothermal and adiabatic walls in convective heat transfer, can be easily imposed. Furthermore, by imposing these boundary conditions, the nature of mass transfer avoid the presence of errors analogous to conductive losses in a wall. For the measurements, the test specimen can be easily prepared by several methods, including dipping, machining, spraying and casting. The heat transfer coefficient can be determined from the measured mass transfer results with good confidence via the heat-mass transfer analogy.

Souza Mendes (1991) reviewed the naphthalene sublimation technique in detail. In his paper, the fundamentals of the method are discussed and its basic characteristics and typical procedures are presented. Goldstein and Cho (1995) also presented a review of mass transfer measurements using naphthalene sublimation technique.

The purpose of the present work is to present the results of an experimental investigation, using the naphthalene sublimation technique, about the influence of internal helical turbulence promoters, when inserted in circular pipes, in the heat transfer and frictional power loss enhancement.

2. EXPERIMENTAL APPARATUS

A partial view of test section is shown in Figs. 1 and 2. A typical arrangement for forced convection experiments consists of an open-loop flow circuit. The circuit is constituted by a 5930 mm long horizontal PVC pipe, with 65 mm internal diameter, and by a centrifugal fan.

Turbulence promoters were obtained by the insertion of helical springs made of copper wires of two different diameters and constant pitch.

Pressure taps were installed along the pipe to measure the pressure drop, both in the smooth section and in the section containing the turbulence promoter. Two electronic pressure transducers ARA 200 (Hartmann & Braun), with operation range from about 0,5 to 2,5 mbar and 0,8 to 4,0 mbar, were applyed. The pressure drop results were obtained as current values from digital multimeters (Metex, model M-4650B).

To measure the air temperature inside the pipe, a mercury thermometer (Incoterm), with operation range between –10 and 50^0C and 0.2^0 C of accuracy, was used. The measurement of the temperature in the surface of the test specimen was done with the aid of NTC nickel sensors, with 10 kΩ resistance, connected to a constant 5V energy source and to a digital multimeter (Metex, model M-4650B).

To weigh the specimens, a precision balance (Bosch, model SAE 200), with 210 g of capacity and 0.1 mg of accuracy, was used. The duration of each run was measured with a digital chronometer (Superatic).

The calibration of the temperature sensors were carried out inside an adiabatic chamber made with expanded polyurethane, where the thermometer and the temperature sensor (connected to the energy source) were inserted. Whereas water was used as the working fluid and with the objective to protect the sensor, both were involved by a rubber balloon.

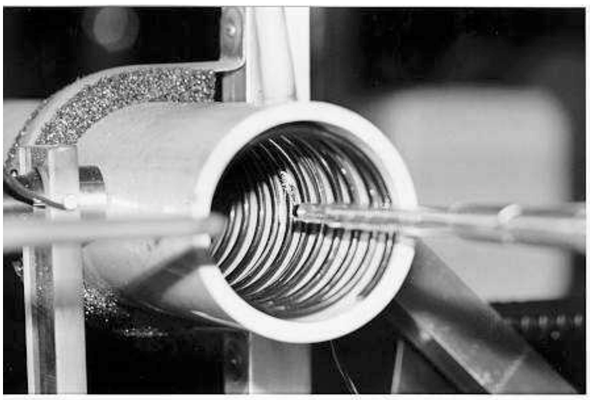

Figure 1 – Detail of the tube with helical turbulence promoter and thermometer.

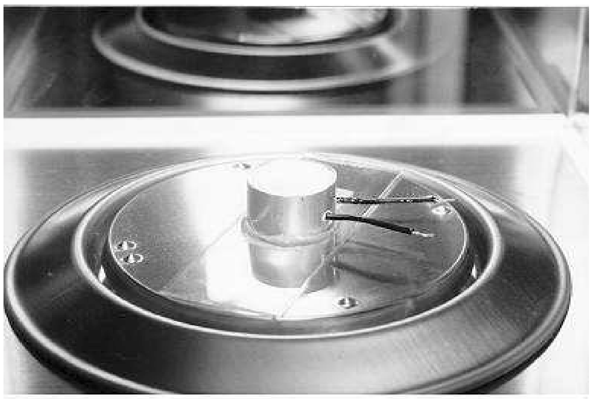

Figure 2 – Weighing of the test specimen.

3. EXPERIMENTAL PROCEDURE

The experiment consisted in submitting the test specimen, previously weighed in a precision balance, to the turbulent flow, at the same time that a second test specimen (called henceforth as reference specimen) is exposed to the ambient conditions (natural convection).

First, molds of glass and aluminum were fabricated and cleaned with a cotton drenched with isopropyl alcohol. While the naphthalene powder was melted, a temperature sensor was inserted in the correspondent mold. The naphthalene was then poured into the mold. After solidification, the surface of the test specimen was prepared with the help of a spatula and a sandpaper. The test specimen consisted of the solid naphthalene and its own mold which was used as a support.

Before the beginning of each run, the test specimen and the reference specimen were weighed and the temperature sensor was connected to the energy source and to the current reading system. The mold was inserted in a hole drilled in the wall of the pipe and the reference specimen was placed near the test section. The care of positioning the naphthalene surface flush to the wall of the duct was taken. After the end of the run, whose duration was approximately 90 minutes, the test specimen was removed of the test section and, as well as the reference specimen, it was weighed again.

4. DATA REDUCTION

The averaged mass transfer coefficient k_C was determined by:

$$k_C = \frac{m}{\left(C_{Aw} - C_{Af}\right)A} \tag{1}$$

where k_C is the mass transfer coefficient (m/s), C_{Aw} e C_{Af} are the naphthalene concentration at the wall and the naphthalene concentration in the turbulent core (kg/m^3), A is the mass transfer area (m^2) and m is the mass transfer rate (kg/s), defined by

$$m = \frac{\Delta m}{\Delta t} \tag{2}$$

in which Δm is the sublimated mass (kg) and Δt is the duration of each run (s). $C_{Aw} = 0$ for a naphthalene vapor free mainstream. Since the vapor concentration (and saturated vapor pressure) of naphthalene at the surface is constant, the wall boundary condition is equivalent to an isothermal boundary condition in the heat transfer process.

The naphthalene vapor concentration at the wall is obtained through the perfect gas law, given by:

$$C_{Aw} = \frac{\overline{M} P_v}{R T_w} \tag{3}$$

where $\overline{M}$ is the naphthalene molecular weight (kg/kgmol), P_v is the naphthalene vapor pressure at he wall temperature (Pa), T_w is the wall temperature (K) and R is the ideal gas constant for naphthalene (N.m/kgmol.K).

The saturated vapor pressure required to calculate the naphthalene wall concentration, i.e., the concentration in the surface test specimen, is given by the Ambrose equation (1975) due to the fact that it furnishes intermediate results among those given by the others authors. The Ambrose equation is defined by

$$T_w \log P_v = \left(\frac{1}{2}\right) A_0 + \Sigma_1^3 \left[A_s E_s(x)\right] \tag{4}$$

where

$$x = \frac{\left(2T_w - 574{,}3\right)}{114} \tag{5}$$

and

$$E_1(x) = x \tag{6}$$

$$E_2(x) = 2x^2 - 1 \tag{7}$$

$$E_3(x) = 4x^3 - 3x \tag{8}$$

being $A_0 = 301{,}6247$, $A_1 = 791{,}4937$, $A_2 = -8{,}2536$ and $A_3 = 0{,}4043$.

The Sherwood number was used to avoid the uncertainty of the diffusion coefficient. It was determined as:

$$Sh = \frac{k_C d}{D_{AB}} \tag{9}$$

where d is the diameter of the test specimen (m) and D_{AB} is the mass diffusivity (m^2/s).

When the molecular diffusion coefficient and Schmidt number given by different authors are compared, including theoretical and empirical correlations, the discrepancy between them is greater than 26%; recent results show a discrepancy of approximately 6%. The resulting correlations are given by:

$$D_{AB} = 0{,}0681\left(\frac{T_f}{298{,}16}\right)^{1{,}936}\left(\frac{101325}{P}\right) \tag{10}$$

and

$$Sc = 2{,}28\left(\frac{T_f}{298{,}16}\right)^{-0{,}1526} \tag{11}$$

where T_f is the turbulent core temperature, P is the ambient pressure and Sc the Schmidt number, being T_f in K and P in Pa.

The heat-mass transfer analogy yields:

$$Nu = Sh\left(\frac{Pr}{Sc}\right)^{1/3} \tag{12}$$

being Nu the Nusselt number and Pr the Prandtl number.

The Reynolds number for flow through the pipe follows the conventional definition:

$$Re = \frac{\bar{u}_m D}{\nu} \tag{13}$$

where $\bar{u}_m$ is the average spatial velocity (m/s), ν is the cinematic viscosity (m^2/s) and D is the diameter of the tube (m).

Whereas the fully developed flow in a smooth pipe, we can analyze the stresses distribution through a control volume and obtain the friction factor defined as:

$$\lambda = \frac{D}{\frac{\rho \bar{u}_m^2}{2}} \left(\frac{\Delta P}{L} \right) \tag{14}$$

where ρ is the naphthalene vapor density (kg/m^3), $\Delta P/L$ is the pressure drop per unit of length (Pa/m).

Maubach (1970) proposed the following equation for the friction factor calculation in smooth pipes:

$$\frac{1}{\sqrt{\lambda}} = 2{,}035 \log\left(Re\sqrt{\lambda}\right) - 0{,}989 \tag{15}$$

Equations (14) and (15) were used to calculate the Reynolds number and the average velocity from pressure drop data.

To verify the adequacy of measurement technique, a heat-momentum analogy was used and compared with the heat-mass analogy results. The equation used in the heat-momentum analogy is defined by Gnielinski (1976) as:

$$Nu = \frac{(\lambda/8)(Re - 1000)Pr}{1 + 12{,}7\sqrt{(\lambda/8)}\left(Pr^{2/3} - 1\right)}\left[1 + \left(\frac{D}{L}\right)^{2/3}\right] \tag{16}$$

where

$$\lambda = \left(1{,}82 \log Re - 1{,}64\right)^{-2} \tag{17}$$

The theoretical Nusselt number, which classic deduction is done by taking into consideration a fully developed flow, must be corrected by a factor that includes the effect of the thermally developing flow and the ratio of the mass-momentum transfer phenomena characteristic lengths (d for mass transfer and D for momentum transfer).

5. RESULTS AND DISCUSSION

Before inserting the turbulence promoters, measurements were performed in the smooth pipe with the purpose of verifying the adequacy of measurement technique to the specific problem.

Figure 3 presents a comparison between Nusselt number obtained via mass and momentum transfer measurements in the smooth pipe, i.e., the pipe without turbulence promoters. The agreement of the results is very good, showing that the experimental technique can be used with a good accuracy in this study, being suitable for the proposed experimental study with turbulence promoters inserted inside a duct. The data were obtained via Chilton-Colburn analogy (through the naphthalene sublimation technique) and via momentum-heat transfer analogy, using the Gnielinski´s equation.

Moreover, the friction factors determined via mass-momentum transfer analogy are 20% greater than those obtained from pressure drop measurements, as shown in Fig. 4. Thus, this is the uncertainty of the friction factor measurements.

A comparison between Nusselt number for smooth tube and for tube containing turbulence promoters with two different aspect ratios is shown in Fig. 5. The presence of the turbulence promoters increase the Nusselt number values from 450 to 650 %, showing the efficiency of the helical turbulence promoters for the heat transfer augmentation. This is reflected also in the results of the correspondent friction factor augmentation, Fig. 6, which was of the order of 250 to 300% when compared to the smooth pipe.

Since Nusselt number is proportional to Reynolds number, Eq. (16), heat transfer coefficients are proportional to the flow velocity inside the limits of validity of this equation, but the pressure drop is proportional to the square of the flow velocity, so that the increase on the heat transfer rates using helical turbulence promoters can occur with very high pumping costs.

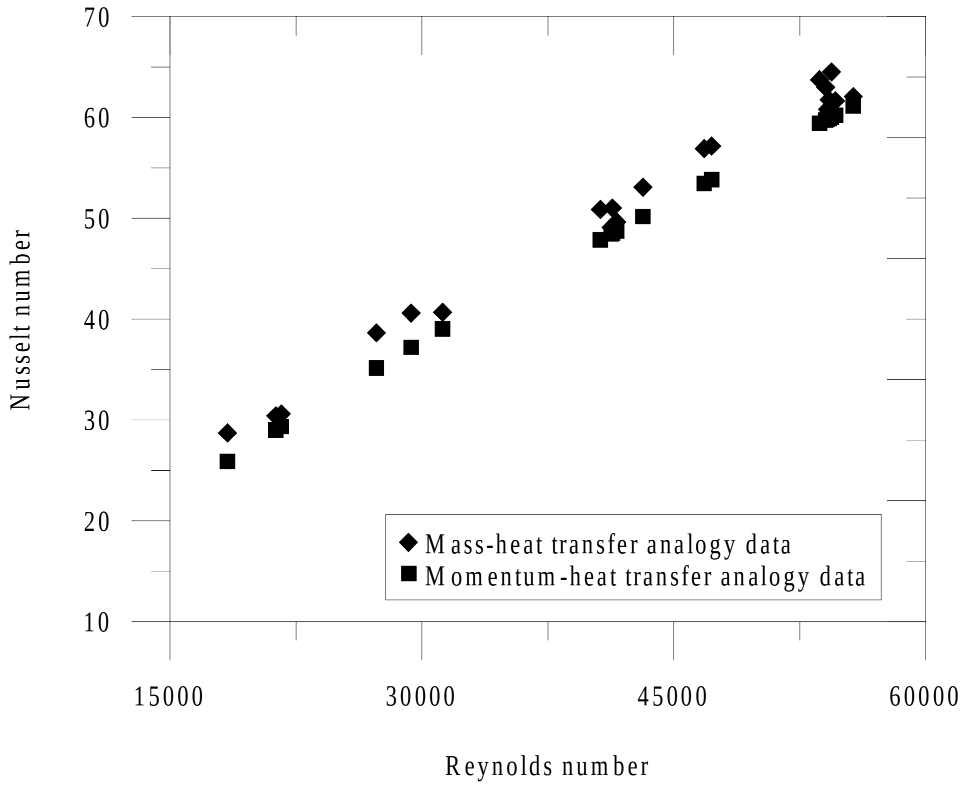

Figure 3 – Comparison of Nusselt number obtained via mass and momentum transfer measurements.

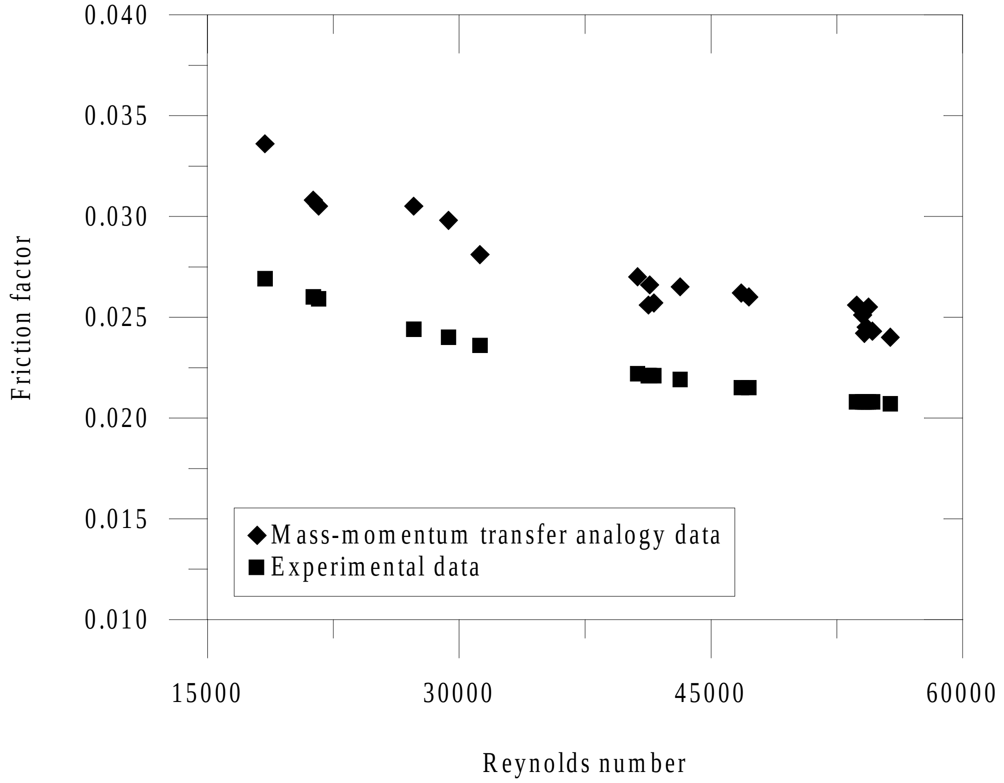

Figure 4 – Friction factor for smooth tube. Comparison between friction factor obtained via pressure drop and via mass-momentum analogy.

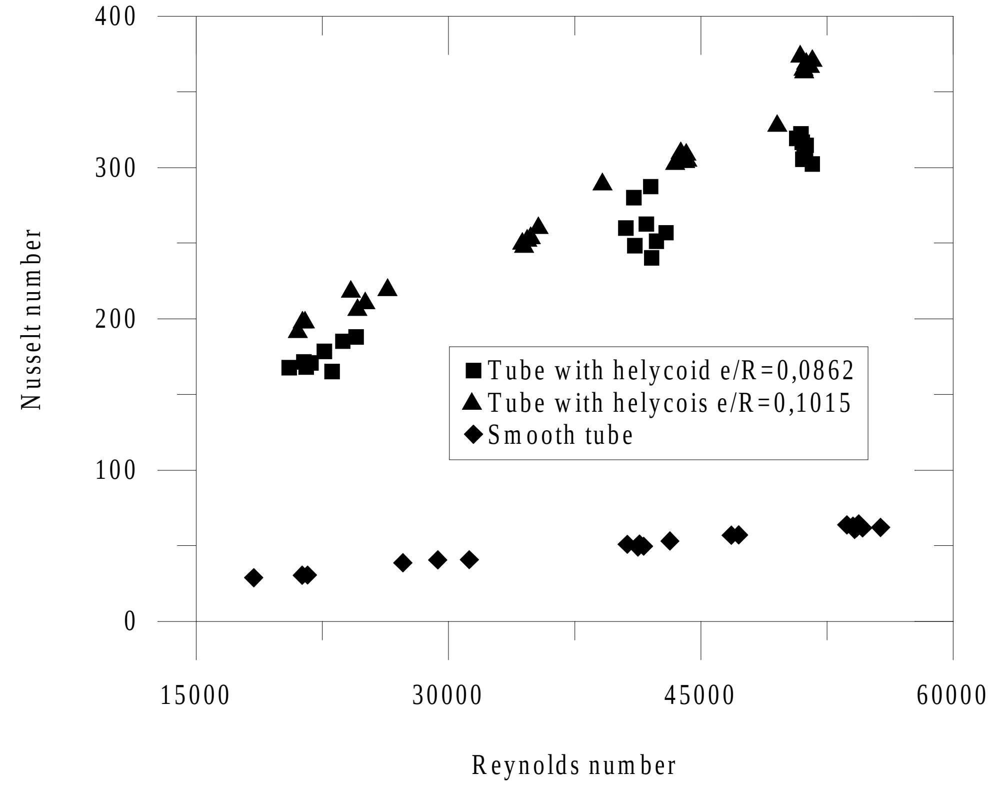

Figure 5 – Comparison between Nusselt number for smooth tube and for tube containing turbulence promoters with two different aspect ratios.

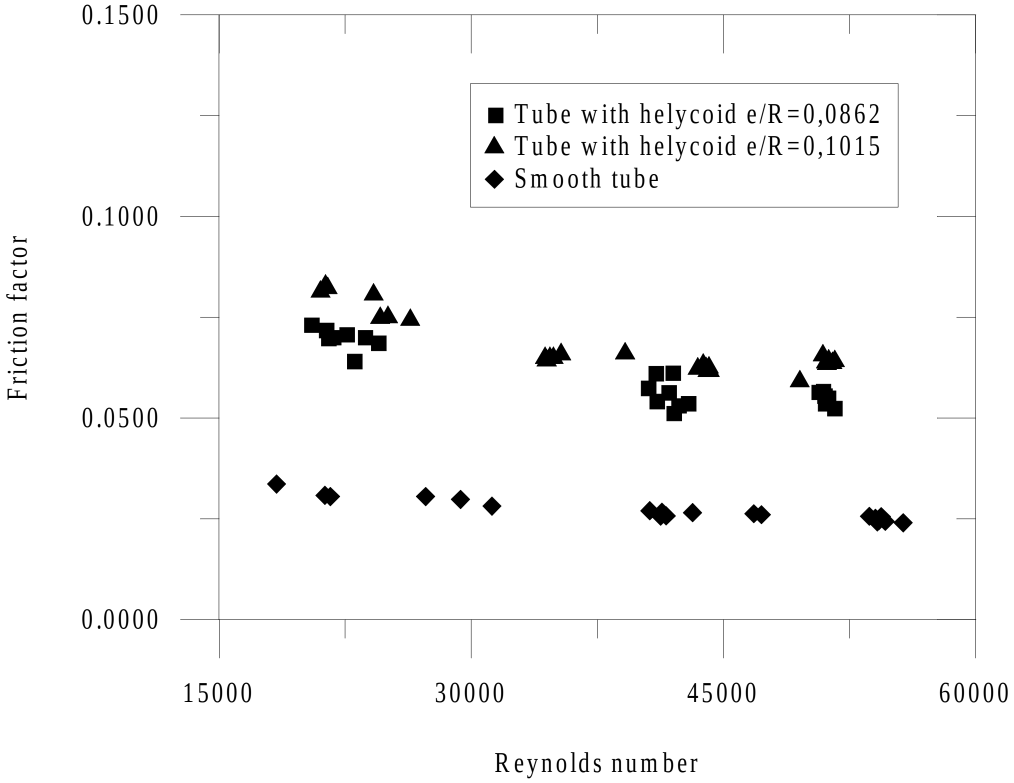

Figure 6 – Comparison between friction factor for smooth tube and for tube containing turbulence promoters with two different aspect ratios.

6. CONCLUSIONS

This paper presents the experimental study of the heat transfer augmentation in a pipe with internal helical turbulence promoters using the naphthalene sublimation technique.

The results of measurements in a smooth pipe show that the experimental technique was adequate to the proposed study with very reliable results of Nusselt numbers and friction factors.

The presence of the helical turbulence promoters increase the Nusselt number, of about 450 to 650 %, compared to the smooth pipe. This is accompanied by a growth of the friction factor of about 250 to 300%, which can lead to very high pumping costs.

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