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Performance of Heat Transfer and Pressure Drop in a Spirally Indented Tube

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In an effort to develop a heat transfer enhancement technique for low temperature applications such as utilization of LNG cold energy, an experiment was carried out to evaluate the heat transfer and the pressure drop performance for a spirally indented tube using ethylene-glycol and water solutions and pure water under horizontal single-phase conditions. The test tube diameter was 14.86 mm and the tube length was 5.38 m. Heat transfer coefficients and friction factors for both inner and outer surfaces of the test tube were calculated from measurements of temperatures, flowrates and pressure drops. Correlations of heat transfer coefficients in the spirally indented tube, which were applicable for laminar and turbulent regimes were proposed for inner and outer surfaces. The correlations showed that heat transfer coefficients for the spirally indented tube were much higher than those for smooth tubes, increased by more than 8 times depending upon the Reynolds number. The correlations were compared with other correlations for various types of surface roughness. The effect of the Prandtl number on the heat transfer characteristics was discussed. The critical Reynolds number from the laminar flow to the turbulent flow inside the spirally indented tube was found to be around Re=1, 000.

Key Words : Spirally Indented Tube, Heat Transfer Coefficient, Double-Pipe Heat Exchanger, Friction Factor

Nomenclature

C_{ph} : Specific heat of hot fluid
D_i : Inside tube diameter
D_o : Outside tube diameter
f : Darcy friction factor
G_h : Mass flow rate of hot fluid
h : Heat transfer coefficient
k_w : Tube wall thermal conductivity
Pr : Prandtl number
Q : Heat transfer rate
Re : Reynolds number
T_{pf} : Mean cold fluid temperature
T_{hf} : Mean hot fluid temperature
$T_{hf,i}$: Mean inlet hot fluid temperature
$T_{hf,o}$: Mean outlet hot fluid temperature
T_w : Average wall temperature

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1. Introduction

As a clean energy, liquefied natural gas (LNG) has become a popular energy resource for residential heating and air-conditioning as well as for industrial processes. At the stage of vaporizing process, LNG cold energy of about 8,400 kJ per unit kilogram at a temperature ranging between −160℃ and −170℃ is disspated to the surrounding. Utilization of this cold energy has become a matter of concern in the LNG industry for energy saving as well as cost reduction to obtain extremely low temperatures. Examples of LNG cold energy utilization may be found in the fields of cryogenic storehouse, cold energy generation, air separation, and freezing treatment of wastes. Heat exchanger systems such as double-pipe types and shell-and-tube types are widely used to take advantage of LNG cold energy. To improve heat transfer performance of such heat exchangers, a suitable enhancement technique should be employed.

In recent years, significant concern has been focussed on techniques for augmenting heat transfer coefficients in the heat transfer equipments. Among these, heat transfer augmentation by means of integral roughness technique has been of commercial interest for increasing the efficiency of process heat exchangers. Typical examples of this technique include a corrugated tube, a spirally indented tube and a wire-coiled insert. This technique is to provide artificial roughness on the heat transfer surface to disturb the flow boundary layer, resulting in reduction of a convective thermal resistance through the viscous sublayer.

Numerous studies of heat transfer enhancement through surface roughness have been performed, over the years, producing a variety of correlations. For instance, Whither (1980) and Li et al. (1982) developed analogy-based correlations whereas Yorkshire (1982) and Ravigururajan and Bergles (1986) proposed a statistical type of correlations. Lee et al. (1993) suggested a Dittus-Boelter type correlations for a spirally twisted tube and an internally finned tube using R-113. These correlations aimed at predicting the heat transfer coefficients for various types of integral roughened tubes over a wide range of flow conditions.

The present study deals with an experimental work of heat transfer enhancement using a spirally indented tube with an ethylene-glycol and water (EG-water), which may be a potential surface roughness technique in a heat exchanger for LNG cold energy utilization. Correlations of heat transfer coefficients in the spirally indented tube are proposed for inner and outer surfaces. The correlations are compared with other correlations for various types of surface roughness. The effect of the Prandtl number on the heat transfer characteristics is discussed and the transition feature from the laminar flow to the turbulent flow

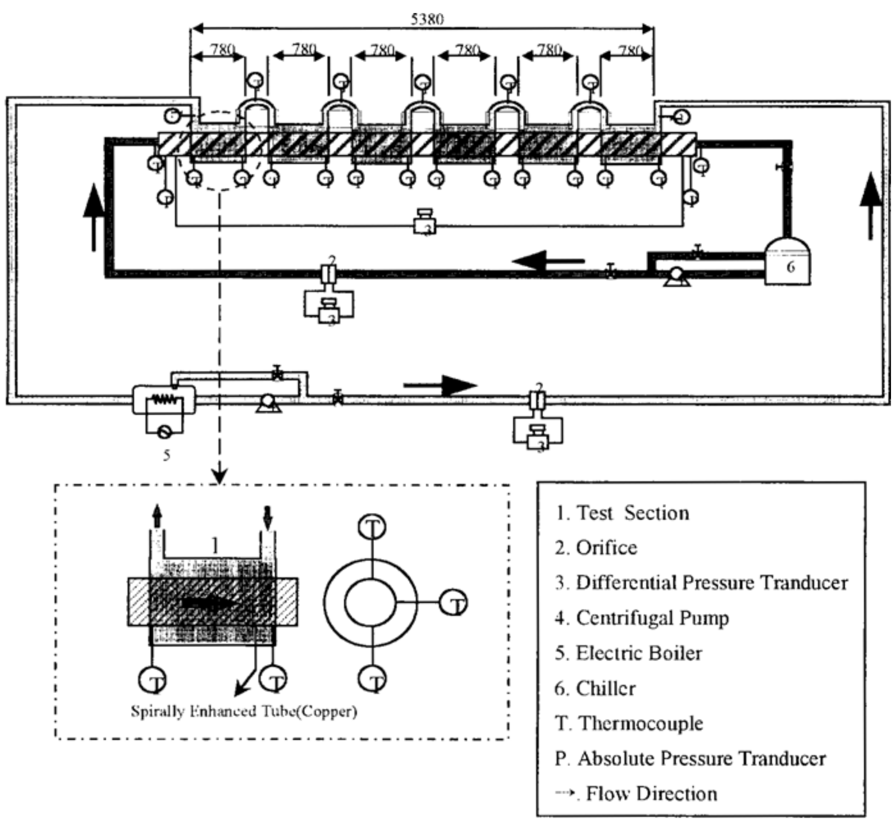
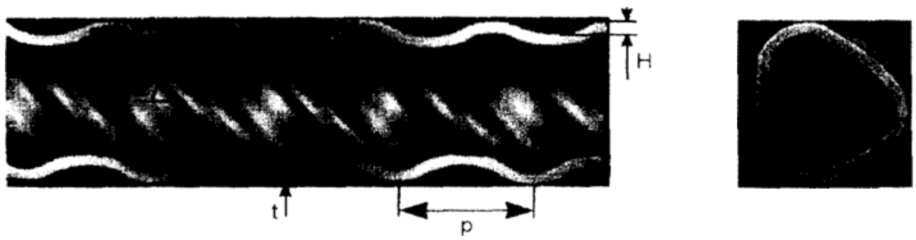


Fig. 1 Schematic diagram of experimental equipment.

Table 1 Experimental conditions in the present study.

Parameter	Inner side (EG Solution)	Outer side (EG Solution)	Inner side (Water)
Inlet pressure (MPa)	0.13 ~ 0.30	0.10 ~ 0.11	0.13 ~ 0.28
Mass flux (kg/m²s)	580 ~ 2,000	220 ~ 520	320 ~ 1771
Inlet temperature (℃)	-13.0 ~ -3.3	20.0 ~ 21.0	12.4 ~ 16.1
Pr	39 ~ 120	34 ~ 40	8 ~ 9
Re	500 ~ 5,000	800 ~ 2,200	4, 000 ~ 25,000



Unit : mm

Material	Outside diameter	Thickness (t)	Angle of inclination (α)	Pitch (p)	Depth of spiral (H)
Copper	15.88	1.02	50°	13	1.6

Fig. 2 Specifications of a spirally indented tube.

inside the spirally indented tube is also mentioned.

2. Experimental

2.1 Apparatus

An experimental facility was constructed to investigate the heat transfer and the pressure drop performance of a spirally indented tube. A schematic diagram of the experimental apparatus is shown in Fig. 1.

It includes a test tube, an electrical boiler, a chiller, and two circulating pumps. The test tube was made of two concentric tubes and the inner tube was a spirally indented tube. Ethylene-glycol and water (EG-water) solutions and pure water were used as working fluid which are frequently used in heat exchanger systems for utilization of LNG cold energy. The total length of the test tube is 5.38 m and the diameters of the inner and outer tubes are 14.86 mm and 32.10 mm, respectively. The specification of the spirally indented tube is indicated in Fig. 2.

The test tube was divided into six 0.78 m long subsections, as shown in Fig. 1. There subsections were arranged in series and mixing chambers for EG-water solution flowing outside the tube were placed between the subsections. Heat was added in the electrical boiler to the EG-water solution circulating outside the tube in a counter-flow mode. The EG-water solution heated in the test tube was cooled in a chiller to maintain steady-state operating conditions in the system. The operating conditions performed in the present study are summarized in Table 1.

The flow rates of both fluids were controlled by pressure regulation valves and were measured by two sets of an orifice and a differential pressure transducer. The uncertainty in the flow rate measurement was $\pm 0.5\%$. Temperatures of the inlet and the outlet of the test tube for both cooling and heating mediums were measured by means of copper-constantan thermocouples. The test tube wall temperature were also measured to calculate the heat transfer coefficient for both the inner side and the outer side of the tube. The uncertainty in the temperature measurement was within ±0.1℃. The pressures at the inlet and the outlet of the test tubes were measured with pressure taps and pressure transducers. The uncertainty in the pressure measurement was within ±0.3%. The signals from thermocouples and pressure transducers were logged by a data logger and then were processed by a computer-based data acquisition system.

2.2 Experimental procedures and data reduction

Performance test for the heat transfer and the pressure drop was carried out for various combinations of operating conditions, as summarized in Table 1. In each test run, data of temperatures, flow rates, and pressure drops on both hot and cold fluids were collected after steady-state conditions were established. In general, approximately 6~7 hours were required for reaching a steady state after the system started up. The heat transfer rates calculated by energy balance equations for both hot and cold fluids were found to agree within 5% error margin.

The data reduction was carried out based on a segmental energy balance. The section-average heat transfer coefficient for inner wall side of the spirally indented tube was calculated from the following equation.

$$h=\left[\frac{\pi D_i \Delta L(T_w-T_{cf})}{Q}-\frac{D_i}{2k_w}\ln\left(\frac{D_o}{D_i}\right)\right]^{-1} \quad (1)$$

where T_w is the average wall temperature, T_{cf} is the mean cold fluid temperature. The heat transfer rate, Q, in Eq. (1) can be calculated from the energy balance.

Table 2 A summary of conditions of correlations used for comparison.

	Author	Medium	Pr	Re	Tube types	Correlation eqs.
Inside	Present Study	Ethylene Glycol	39-120	500-5,000	Spirally Indented	$Nu=0.13Re^{0.67}Pr^{0.33}$
		Water	8-9	4,000-25,000		
	Kaushik and Azer (1986)	R-113 (Liquid)	10	3,000-8,000	Finned	$Nu/Pr^{0.4}=0.07Re^{0.68}F_1^{-1.50}F_2^{1.76}F_3^{16.55}$ $F_1=[1-(4nbt)/(\pi D_i^2\cos\alpha)]/[1-2b/D_i]^2$ $F_2=(\pi D_i)/[\pi D_i+2nb/\cos\alpha]$ $F_3=\sec\alpha$
	Hong and Bergles (1976)	Ethylene Glycol	84-192	13-390	Twisted Tape INserts	$Nu_s=5.172[1+5.481\times10^{-3}Pr^{0.7}\cdot(Re_s/y)^{1.25}]^{0.5}$
		Water	3-7	83-2,460		
	Trupp and Haine (1989)	Air	0.69	500-10,000	Finned	Not available (Graphic form)
	Obot et al. (1991)	Air	0.71	800-50,000	Spirally Fluted	Not available (Graphic form)
Outside	Present Study	Ethylene Glycol	34-40	800-2,200	Spirally Indented	$Nu=0.006Re^{1.2}Pr^{0.4}$
	Kaushik and Azer (1986)	Water	10	800-3,500	Knurled	Not available (Graphic form)

$$Q = G_h C_{ph} (T_{hf,i} - T_{hf,o}) \quad (2)$$

where T_{hf} is the mean hot fluid temperature, G_h is the mass flow rate of hot fluid. Meanwhile, the section-average heat transfer coefficient for outer wall side of the spirally indented tube was calculated as follows:

$$h = \frac{Q}{\pi D_o \Delta L (T_{hf} - T_w)} \quad (3)$$

Finally, the Darcy Friction factor was evaluated from the following equation.

$$f = \frac{\Delta P}{(L/D)(\rho V^2/2)} \quad (4)$$

3. Results and Discussion

Data of the heat transfer coefficients for both inner and outer surface of the spirally indented tube and the friction factor for the inner surface were obtained for various flow conditions. The range of Reynolds number in the present study is approximately 500 to 5,000 for ethylene-glycol and water solution and 4,000 to 25,000 for pure water. The transition from laminar flow to turbulent flow and the effect of the Prandtl number on heat transfer will be discussed. Correlations of heat transfer coefficient for the spirally enhanced surface will be proposed and will be compared to those for other modes of heat transfer enhancement. Table 2 shows a summary of conditions of other correlations considered in the present study for comparison.

3.1 Friction factor

Friction factors inside the spirally indented tube were calculated from the data of pressure drop measurement, using Eq. (4). Figure 3 shows a comparison of the present data with data of Hong and Bergles (1976) and Obot et al. (1991) as well as the laminar equation. As expected, friction factors in the present test tube are much higher than the values calculated by the laminar equation. It is shown that the friction factor for the Reynolds number less than approximately 1, 000 decreases in a linear mode and is inversely proportional to the Reynolds number. The present data are comparable to data of Hong and Bergles (1976) for twisted-tape inserts while they are higher than data of Obot et al. (1991) for a spirally fluted tube. However, both data set also have much higher values than those predicted by the laminar equation.

3.2 Heat transfer coefficient

Heat transfer coefficients for the inner-side and the outer-side of the spirally indented tube were calculated, using Eqs. (1), (2) and (3).

Figures 4 and 6 show the Nusselt number for both sides, respectively, along with the analytical results for a smooth surface. Three different Prandtl number ranges of fluid (Pr=8~9, 39~41, 110 ~120) were tested in the present study. It is noted in Fig. 4 that the Nusselt number for the spirally indented tube is much higher than that for a smooth tube, showing an increase of more than 8 times. This exceptional augmentation may be attributed in part to the flow regime transition from the laminar flow to the turbulent flow in the spirally enhanced pipe even at moderate Reynolds numbers. It is generally acknowledged that the transition takes place at $Re = 2{,}300$ in a smooth

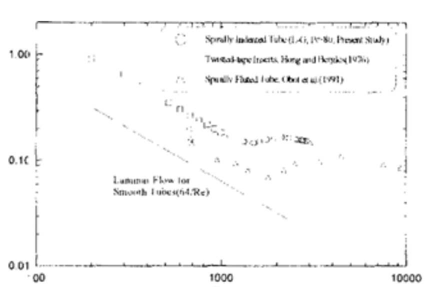


Fig. 3 Friction factors for a spirally indented tube.

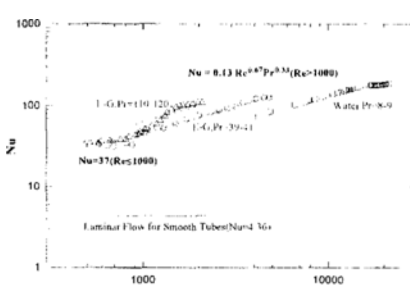


Fig. 4 Correlation of heat transfer coefficient for inner side surface.

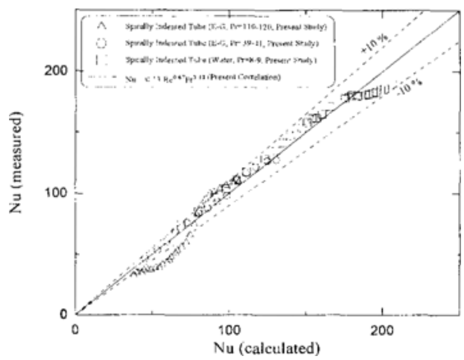


Fig. 5 Comparison of present correlation with experiment data.

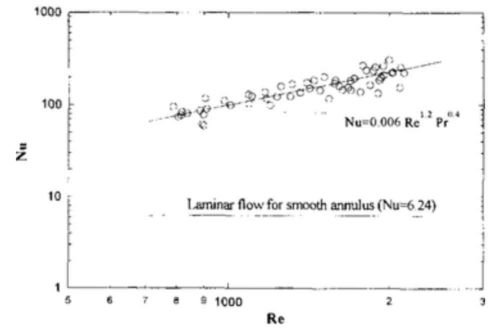


Fig. 6 Correlation of heat transfer coefficient for outer side surface.

pipe, while the transition appears to expedite in the spirally enhanced pipe, occurring around $Re=1{,}000$ as shown in Fig. 4. This may be also confirmed from the friction factor data, as indicated in Fig. 3. It is well known that the critical Reynolds number varies from 1,000 to 10,000 even for a smooth tube depending upon an initial disturbance preexisting when the flow is introduced at the entrance. (White, 1987) Sufficient data of the critical Reynolds number for the integral roughened surface were not found in the literature but Uttawar and Raja Rao (1985) reported a possibility of a smaller critical Reynolds number. They argued that the critical Reynolds number for wire-coiled insert geometries using an oil with very high Prandtl number could be 200. Blumenkrantz and Taborek (1974) observed the critical Reynold number of 1, 000 for Turbotech spirally grooved tube. This number seems to be very close to the result obtained in the present study.

Two different correlations are suggested depending upon the flow regime. For laminar region ($Re \leq 1{,}000$) the Nusselt number is independent of the Prandtl number and the Reynolds number and thus, a simple correlation of constant Nu-type are obtained. In turbulent flow region, the Nusselt number is a function of the Reynolds number and the Prandtl number. A conventional type correlation is proposed based upon a regression technique of the experimental data.

$$Nu=37.0 \text{ for laminar region } (Re\ 1{,}000) \quad (5)$$

$$Nu=0.13\ Re^{0.67}Pr^{0.33} \text{ for turbulent region } (Re>1{,}000) \quad (6)$$

A direct comparison of Eq. (6) with the present data is shown in Fig. 5. Most of the data are within $\pm 10\%$ error margin of the correlation. The exponent of the Prandtl number was found to be 0.33 using three different Prandtl number range data set as mentioned earlier.

Blumenkrantz and Taborek (1971) showed that a spirally enhanced tube provides up to 200% enhancement for heating a viscous fluid in laminar flow. Richards et al. (1987) tested 12 different spirally enhanced tubes in turbulent flow using steam on the outer surface. They suggested conventional type correlations with a fixed value of the exponent on the Reynolds number, that is, 0.8. However, the validity of the 0.8 exponent has not been confirmed. (Webb, 1994)

Figure 6 shows the Nusselt number for the outer side with the analytical expression for laminar flow in the smooth annulus. The Nusselt number for the spirally enhanced surface is much larger than that for a smooth surface of $Nu=6.24$ (Lundberg et al., 1963). This is probably due to the flow transition to turbulent region in the annulus, although the Reynolds number is below the generally known critical value, that is, 2, 300. The surface roughness of the spirally enhanced pipe disturbs the boundary layer, particularly very thin viscous sublayer causing a rapid the decrease of the thermal resistance in the vicinity of the wall. A single correlation for the outer surface can be given as the following.

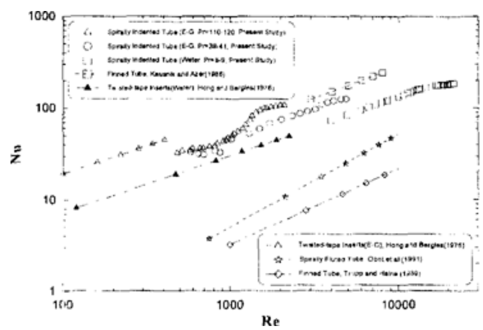


Fig. 7 Comparison with other correlations (Nu –Re) for inner surface data.

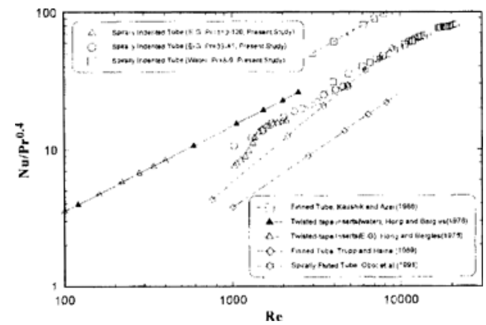


Fig. 8 Comparison with other correlations ($Nu/Pr^{0.4}$ – Re) for inner surface data.

$$Nu = 0.006\ Re^{1.2}\ Pr^{0.4} \tag{7}$$

Note that the entrance length effect exists in the annulus flow, but the length can be neglected compared to the total subsection length.

3.3 Comparison with other correlations

The conditions of present correlations are compared with other available correlations as summarized in Table 2. The objectives of this comparison are to reveal the heat transfer performance of various enhancement techniques on a common basis and to investigate the effect of the Prandtl number on the heat transfer characteristics.

Figure 7 shows a comparison of the present data on inner wall with various correlations for finned tubes, twisted-tape inserts and spirally fluted tube on the (Nu, Re) plane. The same data and correlations were redrawn to investigate the influence of the Prandtl number on the ($Nu/Pr^{0.4}$, Re) plane, as shown in Fig. 8. It is seen that the dependency of the Nusselt number on the Reynolds number is very similar, regardless of the enhancement mode summarized in Table 2. The enhancement technique of twisted-tape inserts exhibited, in general, a better heat transfer performance. It is considered that the twisted-tape inserts to disturb the boundary layer and promote a turbulent mixing of thermal energy in the flow core. Meanwhile, the spirally enhanced surface and the spirally fluted surface only have a function to disturb the boundary layer in the vicinity of the wall, reducing the convective thermal resistance through the viscous sublayer. The finned tube may have one function or both functions, depending upon the fin geometry and the relative height.

It is very interesting to investigate the influence of the Prandtl number on heat transfer on integral roughened surfaces due to a mechanism of boundary layer development on the surfaces which is different from that on the smooth surfaces. Several works (Ravigururajan and Bergles, 1995 ; Rabas and Arman, 1992) have been done on this subject but contradictory results were be found on a value of the exponent of the Prandtl number in a Dittus-Boelter type correlation. For example, Hong and Bergles (1976) suggested 0.4 as the Prandtl number exponent based on the experimental results of two different fluids in a circular pipe with twisted tape inserts. Using five different geometries with transverse, rectangular ribs, Webb, et al. (1971) showed that the exponent varied, between 0.4 and 0.6 depending upon the flow condition. On the other hand, Bergles and Hau (1979) indicated that the Prandtl number exponent did not vary from a value of about 0.33 based on the experimental results in a pipe with transverse, square disruptions. In the present study, a Prandtl number exponent of 0.33 seems to be reasonable, as shown in Fig. 5. based on experimental data of EG-water solutions with Pr $=39\sim41$ and $110\sim120$ and pure water with Pr$=8\sim9$.

Figures 9 and 10 show a comparison of the present data for the outer side with water data of Kaushik and Azer (1986) for knurled surface.

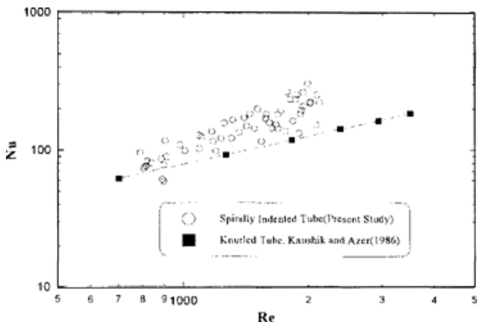


Fig. 9 Comparison with other correlations (Nu -Re) for outer surface data.

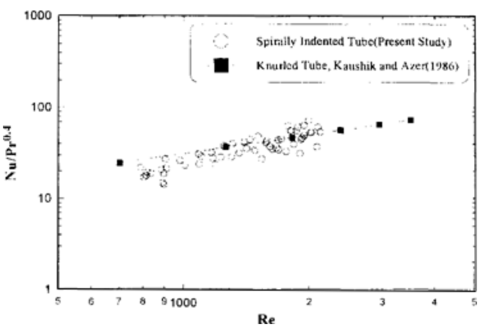


Fig. 10 Comparison with other correlations ($Nu/Pr^{0.4}-Re$) for outer surface data.

The Nusselt number for the spirally enhanced surface is higher than that for a knurled surface mainly due to higher Prandtl number, as in Fig. 9 but both enhancement techniques seem to be similar in heat transfer performance, as seen in Fig. 10.

4. Conclusions

An experiment was carried out to evaluate the heat transfer and the pressure drop performance for a spirally indented tube using ethylene-glycol and water solutions and pure water under horizontal single-phase conditions. The results showed that the heat transfer coefficients for the spirally indented tube was increased to more than 8 times of those for smooth tube. The friction factors were also increased considerably but the heat transfer augmentation effect was more dominant. The transition from the laminar flow to the turbulent flow inside the spirally indented tube was found to take place around $Re=1,000$. Correlations of heat transfer coefficients in a power-law relationship of the Nusselt number, the Reynolds number and the Prandtl number were proposed, which could be available for practical applications. The correlations were compared with those for various types of surface roughness.

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