Experimental Studies on Heat Transfer in the Annuli with Corrugated Inner Tubes

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Experimental heat transfer data for single-phase water flow in the annuli with corrugated inner tubes are presented. In the annuli with parallel flow, ten different annular arrangements are considered. For water flow rate in $1{,}700 < Re < 13{,}000$ regime, data for Nusselt numbers are presented. The results show significant effects of both the pitch to trough height ratio (P/e) and the radius ratio (r^*). As P/e becomes closer to 8 in the range below the radius ratio (r^*) of 0.5, Nusselt numbers increase. However, Nusselt numbers decrease in the range above the radius ratio (r^*) of 0.5 because flow reattachment position becomes farther in the narrower clearance.

Key Words : Heat Transfer Coefficient, Corrugated Annular Tube, Parallel Flow, Turbulent Flow, LMTD (Log-Mean Temperature Difference)

Nomenclature

A : Heat transfer area [m^2]
C_p : Specific heat [kJ/kg K]
D_b : Bore diameter [m]
D_e : Envelope diameter [m]
D_h : Annulus hydraulic diameter, $(D_{oi}-D_{vo})$ [m]
D_{oi} : Inner diameter of the outer smooth tube [m]
D_{vi} : Volume-based corrugated tube inner diameter [m]
D_{vo} : Volume-based corrugated tube outer diameter [m]
e : Corrugation depth, $(D_e-D_b)/2$ [m]
e^* : Nondimensional corrugation depth, e/D_{vo} [m]
f : Friction factor, $2\Delta PD_h/\rho V^2 L$
h : Heat transfer coefficient [W/m^2 K]
k : Thermal conductivity [W/m K]
L : Tube length [m]
N : Number of corrugation starts
Nu : Nusselt number, $h D_h/k$
P : Corrugation pitch [m], pressure [Pa]
P^* : Nondimensional corrugation pitch P/D_{vo} [m]
Pr : Prandtl number, $\mu C_p/k$
Q : Heat duty [W]
R : Heat transfer resistance [K/W]
Re : Reynolds number, $D_h V/\nu$
r^* : Annulus radius ratio, D_{vo}/D_{oi}
T : Temperature [K]
U : Overall heat transfer coefficient [W/m^2 K]
V : Flow velocity [m/s]
Vol : Volume [m^3]
θ : Corrugation helix angle, $\tan^{-1}(\pi D_{vo}/NP)$ [°]
θ^* : Nondimensional corrugation helix angle, $\theta/90$

Subscripts

a : Annulus
as : Smooth annulus
i : Inner tube, inlet
ci : Inlet at cold side
co : Outlet at cold side
hi : Inlet at the hot side

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ho : Outlet at the hot side
s : Smooth
t : Tube
tot : Total
w : Wall
r : Corrugated

1. Introduction

The convective heat transfer coefficients may be increased by artificially roughened surfaces, inlet vortex generators, vibration of the surface, application of electrostatic fields, and modification of the duct cross section and surface. Many of these techniques increase the heat transfer coefficient through a change in flow patterns. In the recent past, some attention has been given to heat transfer augumentation by means of spiral flutes, grooves, and ridges on heat exchanger tubes (Marto et al., 1979). The spirally fluted or corrugated tube is believed to enhance the convective heat transfer by introducing swirl into the bulk flow and/or periodic disruption of the boundary layer at the tube surface due to repeated changes in the geometry. Several geometric parameters such as the inner diameter, envelope diameter, ridge, pitch, shape, and the number of starts, must be specified to define a corrugated or fluted enhanced geometry completely. These dimensions are shown in Fig. 1. A change in any of these dimensions affects the flow and heat transfer characteristics of the tube. A survey of heat transfer and friction factor studies for spirally fluted tubing was also conducted by Bergles (1980). For laminar internal heat transfer in heating, 200 percent increases in the heat transfer coefficient and friction factor were reported. For turbulent internal flow, several studies were quoted with improvements of up to 400 percent above the plain tube heat transfer coefficients, but the pressure drop was as much as 10 times higher than for plain tubes. Heat transfer and pressure drop for tubes with single- and multiple- helix internal ridging were investigated by Withers (1980a, b). An empirical correlation for friction factor in terms of the Reynolds number, Re and a set of adjustable constants was proposed. Nakayama et al. (1983) performed an experimental investigation of heat transfer enhancement for water flowing through spirally ribbed tubes in a turbulent regime. They postulated that at low helix angles, the flow near the wall follows the rib profiles, while at high helix angles, it crosses the ribs. At intermediate angles, the flow changes from swirl-dominate flow to cross-over flow. Garimella and Christensen (1995a, b) and Garimella (1990) also addressed the hydrodynamic and heat transfer aspects of the annuli with spirally fluted inner tubes in counterflow. Flow mechanisms and pressure drop measurements were used to propose the friction factor correlations. These fluted-annulus friction factor correlations can be used to develop the Nusselt number correlations in conjunction with heat transfer data. While some research has been done by previous investigators on the spirally fluted enhanced geometries, there are deficiencies in understanding of heat transfer at spirally corrugated geometries. The objective of the present study is to experimentally investigate the heat transfer characteristics of annuli formed by placing a spirally corrugated tube inside a smooth outer tube in parallel flow.

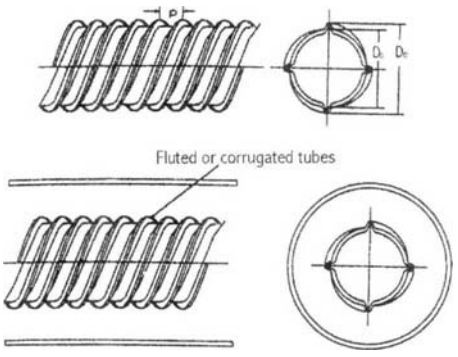

Fig. 1 Details of test section

2. Experimental Apparatus and Data Reduction

Six spirally corrugated tubes, one fluted tube, and three smooth tubes for benchmarking purpose, selected to achieve an adequate variation in all the relevant geometric variables, were used for

Table 1 Test matrix for temperature measurements

Annulus	Inner tube											Outer tube
	D_b [mm]	D_e [mm]	D_s [mm]	D_{vi} [mm]	D_{vo} [mm]	P [mm]	N	r^*	e^*	P^*	θ^*	D_o [mm]
T_{11}	7.9	9.5	9.5	7.68	9.28	2	1	0.368	0.022	0.22	0.96	25.2
T_{12}	7.9	9.5	9.5	7.52	9.12	5	1	0.362	0.044	0.55	0.89	25.2
T_{21}	11.0	12.5	12.5	10.79	12.39	6	1	0.492	0.048	0.48	0.92	25.2
T_{22}	10.0	12.5	12.5	10.93	12.53	10	1	0.497	0.056	0.80	0.84	25.2
ST_{21}	11.6	12.8	12.5	10.71	12.31	10	3	0.488	0.13	0.81	0.68	25.2
T_{31}	14.2	15.8	15.8	13.88	15.48	7	1	0.614	0.045	0.45	0.91	25.2
T_{32}	14.2	15.8	15.8	13.97	15.57	10	1	0.618	0.045	0.64	0.89	25.2
S_{01}	7.9	9.5	9.5	7.9	9.5			0.377				25.2
S_{02}	11.0	12.5	12.5	10.9	12.5			0.496				25.2
S_{03}	14.2	15.8	15.8	14.2	15.8			0.627				25.2

D_b : Bore diameter
D_e : Envelope daimeter
D_s : Original smooth tube diameter
D_{vi} : Volume-based grooved tube inner diameter
D_{vo} : Volume-based grooved tube outer diameter
P : Flute or corrugation pitch
N : Number of flute or corrugation starts
D_{oi} : Inner diameter of outer smooth tube
S : Smooth tube
ST_{21} : Spirally fluted tube
T : Spirally corrugated tube

the heat transfer test. Each tube was placed in a smooth outer tube, which enabled the testing of annuli with different radius ratio shown in Fig. 1. A matrix of all the annuli for which heat transfer tests were conducted is presented in Table 1. A schematic of the flow loop and test sections is presented in Fig. 2. City water was supplied to a settling tank equipped with an overflow line. In this tank, any dissolved air escaped to the atmosphere. The overflow line ensured a constant pressure head from the inlet to the test section. To achieve the objective of this study, a test rig that allowed testing over a wide range of flow rates was required. It is essential that the temperature changes in the individual fluid streams, and the approach temperature differences used for calculation of $LMTDs$, are large enough to minimize the errors due to measurement inaccuracies. Electric heating from the tube side or the tube wall itself was deemed impractical because of the convoluted corrugated tube geometry. Steam condensation on the tube side could not be used because for laminar flow at the annuli side, the low flow rate caused the cold fluid to approach the hot side temperature within a very short distance. Therefore, a substantial part of the heat exchanger does not contribute to heat transfer but is erroneously

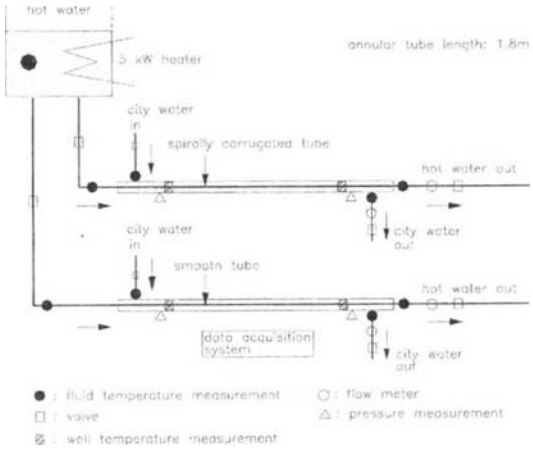

Fig. 2 Schematic diagram of experimental setup

included in the heat transfer area calculations. Near the outlet of annulus side, the water could start boiling. Thus the present method using the single phase fluid streams in both tubes was suitable. Laminar and turbulent heat transfer tests with reasonable temperature changes in the respective streams, as well as between the hot and cold side, can be performed by using single-phase water on both sides of the heat exchanger. An appropriately controlled tube-side inlet temperature and mass flow rate would provide a high

enough ratio of h_t/h_a without causing an excessive temperature rise in the annuli fluid. As the annulus-side flow rate is increased to achieve the turbulent Re values, the tube-side flow rate can also be increased to maintain high values of tube-side heat transfer coefficient (h_t)/ annulus-side heat transfer coefficient (h_a). Considering these factors, this method is chosen for conducting the heat transfer tests. This method does, however, require a means of calculating h_t with an accuracy that depends on the specific value of h_t/h_a in question. A cold water supply line and a supply of hot water were used for the tests. The city water line was split into two different streams. One stream was used as the cold water inlet to the annulus side of the test section. The other stream was heated by an electrical heater (Cartridge type, 5 kW) to provide the desired flow rate of hot water at the required inlet temperature. The hot water passed through the heat exchanger and exited to the drain. Cold water from the outlet of test section was supplied to the combination of control valves, from which it flowed through a flow meter of cumulative type.

The measured flow rate was used to calculate the heat duty in the heat exchanger and the annulus Re. The cold water flowed through the parallel flow heat-exchanger test section and exited to the drain. The test section consisted of a 1.8-m-long heat exchanger, insulated on the outside, formed by placing the various tubes inside a smooth outer tube. We measured the inlet and outlet temperatures of the tube side and shell side for overall energy balance by a data acquisition system. Thermocouple ports (for K-type thermocouples) were also provided at 30 cm and 150 cm from the ends of the inner tube. Numerical values and graphs of temperatures as a function of time were displayed and recorded on a computer to allow confirmation of the steady state. During the tests, the ratio of tube-side to annulus-side flow was kept at about five whenever feasible. Because the annulus-side flow area was typically larger than that of the tube side, this ratio of mass-flow rates resulted in higher ratio of heat transfer resistances. This control strategy helped in minimizing the sensitivity of the deduced annulus-side heat transfer coefficient (h_a) values to errors in the tube-side correlations. The Reynolds number was calculated from the measured flow rate based on the hydraulic diameter, D_h. The flow velocity was calculated by using the cross-sectional flow area. It should be noted that the corrugated tube did not have a circular cross section ; therefore, a diameter that represented the average cross-sectional area of the corrugated tubes was calculated from the volume of water required to fill a given length of tubing as follows :

$$D_{vi}=\sqrt{\frac{4\overline{Vol}}{\pi L}} \tag{1}$$

The volumetric outside diameter, D_{vo}, which is the quantity of interest for the annulus side, is calculated by adding twice the tube thickness to D_{vi}. The heat duties of the two fluid streams were calculated from the flow rates and the temperature changes in the respective streams from the inlet to the outlet. The discrepancy between Q_a and Q_t was less than 6% even at the worst situation. The overall heat transfer coefficient UA_{tot} was calculated as follows :

$$UA_{tot}=\frac{Q_{tot}}{LMTD_{tot}} \tag{2}$$

$$LMTD=\frac{(T_{ho}-T_{co})-(T_{hi}-T_{ci})}{\ln[(T_{ho}-T_{co})/(T_{hi}-T_{ci})]} \tag{3}$$

where $LMTD$ means the log-mean temperature difference in parallel flow. The subscripts tot represents the total heat exchanger, in addition, h_i and h_o refer to the inlet and outlet at the hot fluids and c_i and co refer to the inlet and outlet at the cold fluids, respectively. Heat duties, $LMTDs$, and UAs are also calculated by using the equations shown above for this fully developed heat exchangers for each data point for all the annuli. The overall U value ($UA/\pi D_{vo}L$) is comprised of the conductance of the tube side, the tube wall, and the annulus side. Tube side flow is always maintained in the turbulent regime, and the corresponding tube-side Nu for $Re>700$ is given by Ravigururajan and Bergles (1985) as follows :

$$\frac{Nu_r}{Nu_s}=(1+[2.64Re^{0.036}e^{*0.212}P^{*-0.21}\times(\theta^*)^{0.29}\mathrm{Pr}^{-0.024}]^7)^{1/7} \quad (4)$$

In the correlation heat transfer data for smooth tubes, several investigators have used an equation that incorporated the corresponding friction factor. One such example is the Petukhov and Popov correlation (1963). Because the Nusselt numbers for fluted annuli showed a departure from laminar behaviour at Reynolds numbers as low as 700, it was assumed that such an approach would be applicable for $700<\mathrm{Re}<40{,}000$ with a modification as follows (Garimella and Christensen, 1995);

$$Nu_s=\frac{(f_s/8)RePr}{1+12.7\sqrt{f_s/8}\,(Pr^{2/3}-1)} \quad (5)$$

where f_s is the smooth-tube friction factor (Petukhov, 1970):

$$f_s=4[1.58\ln(Re)-3.28]^{-2} \quad (6)$$

These equations were applied to calculate the tube-side heat transfer coefficient based on the corrugated-tube outer volumetric diameter as follows:

$$h_t=\frac{Nu_r\cdot k}{D_{vi}}(D_{vi}/D_{vo}) \quad (7)$$

The tube wall presents the following resistance to heat transfer:

$$R_w=\frac{D_{vo}}{2k_w}\ln(D_{vo}/D_{vi}) \quad (8)$$

From the above equations, the heat transfer coefficient of the annulus (h_a) was calculated from the following equation;

$$U=\frac{1}{1/h_t+R_w+1/h_a} \quad (9)$$

and Nusselt number in the annulus is given by

$$Nu_a=h_aD_h/k \quad (10)$$

Uncertainties in the reported values of Nu were evaluated by using the compounding-of-errors technique (Kline and McClintock, 1953). The uncertainty in the calculation of Nu consists of three major components, uncertainty in U, and uncertainty in deriving Nu from U, and the h_t prediction. From the correlation by Ravigururajan and Bergles (1985), the uncertainty in the tube-side resistance may be conservatively estimated to be $\pm25\%$. By combining this with the uncertainty in U calculated above, the uncertainty of Nu is $\pm16\%$.

3. Results and Discussion

Three smooth-tube annuli were tested for benchmarking purpose as shown in Fig. 3. The Nusselt numbers were higher with smaller radius ratio (r^*). This phenomena might be attributed to the definition of Reynolds number. The data correlated by Eq. (11) and Garimela and Christensen's experimental results (1995) for smooth-tube annuli were included for a comparison. The graph shows good correspondence between the present results and the literature values.

These researchers presented turbulent heat transfer data for $Re<5{,}000$. The data from Kays and Leung (1963) were correlated by Garimela and Christensen (1995) and the resulting equation was assumed also to apply for Re values between 3,000 and 5,000, with recognition that the potential for error for $Re<5{,}000$ would be high. The correlation was as follows;

$$Nu_{as}=0.22Pr^{0.5}Re_{as}^{0.8} \quad (11)$$

where the subscript as refers to a smooth annulus. Kays and Crawford (1993) reported that Nusselt number was 6.18 in the fully developed laminar annular tube with radius ratio (r^*) of 0.5. However, Nusselt number in the present work was much higher than 6.18 for $Re<2{,}300$. Thus, it is needed to investigate the transition region between laminar and turbulent flow. Garimella (1990) and Garimella and Christensen (1993) showed from the flow visualization tests of flute-tube annuli that the fluid did not flow exclusively in trough and crest zones near an enhanced-tube wall and in a purely axial flow away from the wall, but in a spiral pattern, alternatively crossing over the trough and crest. Their tests also confirmed that a Reynolds number of 800 adequately delineated laminar and turbulent flow. Therefore, it seemed that the present work covering annuli with corrugated core tubes would have a similar flow pattern.

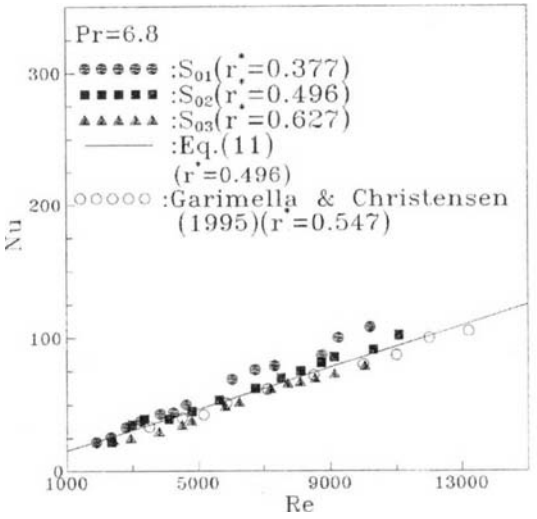

Fig. 3 Nusselt numbers for smooth annuli

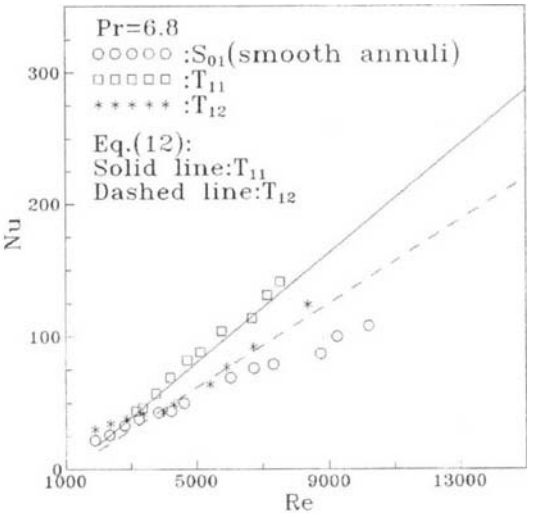

Fig. 4 Nusselt numbers for corrugated annuli, T_{11} and T_{12}

Figure 4 shows the variations of Nusselt numbers in the annuli having the corrugated core tubes, T_{11} and T_{12}. The Nusselt numbers in T_{11} of pitch to trough height ratio $(P/e)=10$ are somewhat higher than in T_{12} of 12.5. This phenomena have similar tendencies to Lawn's results (1974) that Nusselt numbers become highest around $P/e=8$ and lower at the situation farther away from $P/e=8$. This feature is in a line with Wilkie (1966) and Ahn et al. (1994) showing that the highest heat transfer occurs when the roughness pitch and flow reattachment are equal. We derived the empirical correlations for the annuli having the corrugated core tubes as follows ;

$r^*<0.5$:

$$Nu_a=(0.115Pr+0.1987)(-4.94652\times10^{-7}Re^2+0.026896Re-36.25)\times[0.0211(P/e)^2-0.528(P/e)+3.742]r^{*-0.55} \quad (12)$$

$r^*>0.5$:

$$Nu_a=(0.115Pr+0.1987)(7.268\times10^{-8}Re^2+0.01054Re+15.3)\times[0.0281(P/e)+0.499]r^{*-0.55} \quad (13)$$

The solid and dashed lines indicate the data obtained from the empirical correlations for T_{11} and T_{12}. The empirical correlations provide a fairly good representation of variations of experimental data. The Nusselt numbers in the annuli with corrugated core tubes of T_{21} and T_{22}, and with spirally fluted core tube of ST_{21} are shown

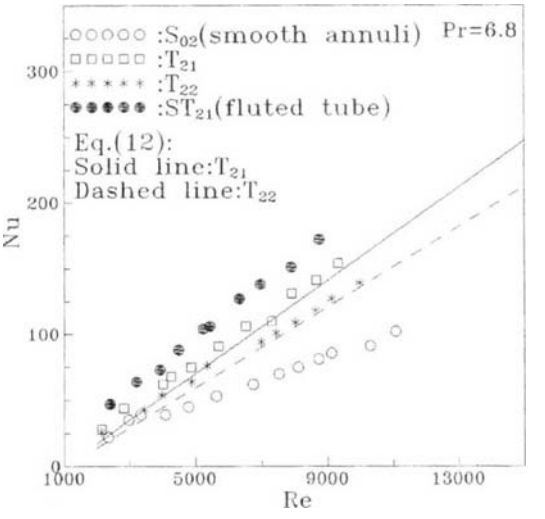

Fig. 5 Nusselt numbers for corrugated and fluted annuli, T_{21}, T_{22} and ST_{21}

in Fig. 5. The values in ST_{21} are much higher than in T_{21} and T_{22}. This effect might be caused by the fact that the trough, corresponding to the roughness height, in the spirally fluted tube has higher. Nusselt numbers for T_{31} and T_{32} are indicated in Fig. 6. Careful inspection of this figure further shows that, being different from Figs. 4 and 5, the heat transfer in T_{32} having pitch to trough height ratio (P/e) of 14.2 is higher than in T_{31} having P/e of 10. This feature is attributed

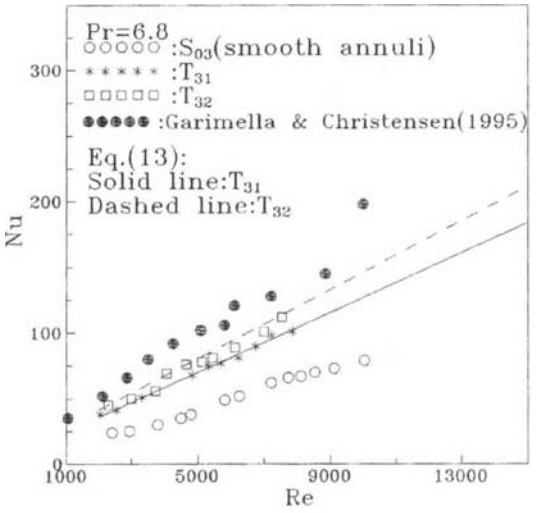

Fig. 6 Nusselt numbers for corrugated annuli, T_{31} and T_{32}

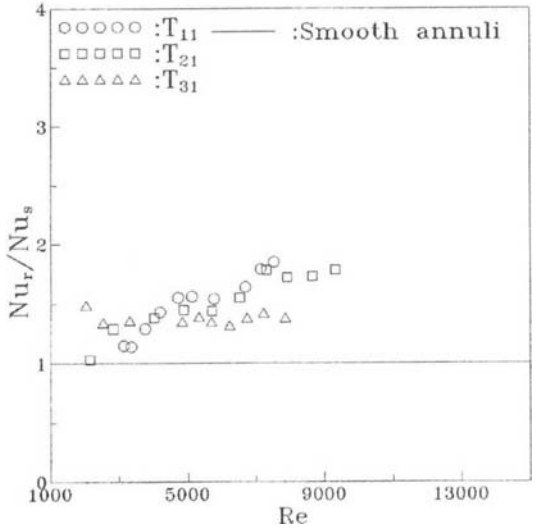

Fig. 7 Nusselt numbers normalized by smooth annuli

to the fact that the flow reattachment occurs at the position farther away from the vicinity of the ridge in the annuli with higher radius ratio. For a comparison, Garimella and Christensen's data (1995), covering the annuli with spirally fluted core tube, are involved. It is shown that the same features as in Fig. 5 are retained. Nusselt numbers non-dimensionalized by a smooth channel are represented in Fig. 7. The data for the smooth annuli using Eq. (11) are indicated as a reference. The effects of Nusselt numbers on the radius ratio were not simple for the flow in the annuli with square-ribbed surface roughness elements on the outer tubes only (Ahn et al., 1994). The normalized Nusselt numbers increased with the radius ratio. However, in the present study, normalized Nusselt numbers usually increased with decreasing the radius ratio. The reason for discrepancy between the two cases is not known for the moment because lack of flow data over the annuli with rough surfaces makes it difficult to speculate on probable cause.

4. Conclusion

The present work shows the experimental heat transfer data for single-phase water flow in the annuli with corrugated inner tubes. In the annuli with parallel flow, ten different annular arrangements for water flow rate in the $1{,}700 < Re < 13{,}000$ regime are considered. The following conclusions can be drawn:

(1) In the range below the radius ratio (r^*) of 0.5, the Nusselt numbers in T_{11} of pitch to trough height ratio $(P/e) = 10$ are somewhat higher than in T_{12} of 12.5. It is because the radial directional fluctuating components are strongly produced in the equality between the roughness pitch and flow reattachment distance.

(2) However, the heat transfer in T_{32} having pitch to trough height ratio (P/e) of 14.2 is higher than in T_{31} having P/e of 10 in the range above the radius ratio (r^*) of 0.5 because the flow reattachment occurs at the position farther away the vicinity of the ridge in the annuli with higher radius ratio.

(3) The Nusselt numbers in the annuli with spirally fluted core tube of ST_{21} are much higher than in the annuli with corrugated core tubes of T_{21} and T_{22} because the trough, corresponding to the roughness height, is higher in the spirally fluted tube.

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