THE OVERALL HEAT TRANSFER CHARACTERISTICS OF A DOUBLE PIPE HEAT EXCHANGER: COMPARISON OF EXPERIMENTAL DATA WITH PREDICTIONS OF STANDARD CORRELATIONS

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(Received: September 24, 2001 – Accepted in Revised Form: October 7, 2002)

Abstract The single-phase flow and thermal performance of a double pipe heat exchanger are examined by experimental methods. The working fluid is water at atmospheric pressure. Temperature measurements at the inlet and outlet of the two streams and also at an intermediate point half way between the inlet and outlet is made, using copper-constantan thermocouple wires. Mass flow rates for each stream are also measured using calibrated rotameters. Heat is supplied to the inner tube stream by an immersion heater. The overall heat transfer coefficients are inferred from the measured data. The heat transfer coefficient of the inner tube flow (circular cross section) is calculated using the standard correlations. The heat transfer coefficient of the outer tube flow (annular cross section) is then deduced. Higher heat transfer coefficients are reported in the laminar flow regime in comparison to the predictions of standard correlations for straight and smooth tubes. The reasons for this discrepancy are identified and discussed. Experimental errors in measuring temperatures and mass flow rates are studied and their effects on the heat transfer coefficients are estimated. Experimental results for the range of operating conditions used in this work show that the outer tube side heat transfer coefficients are smaller than the inner side heat transfer coefficients by a factor of almost 1.5 and 3.4 in counter flow and parallel flow arrangements, respectively. The agreement with predictions is very good for the counter flow arrangement, but not very good for the parallel flow arrangement.

Key Words Double Pipe Heat Exchanger-Heat Transfer Coefficient-Inner Tube Flow-Outer Tube Flow

چکیده جریان یک فاز و عملکرد حرارتی یک مبدل حرارتی دو لوله ای با استفاده از روشهای تجربی مورد بررسی قرار می‌گیرد. سیال عامل آب در فشار اتمسفری است. اندازه گیری درجه حرارت در مقاطع ورودی و خروجی و یک نقطه میانی بین مقاطع ورودی و خروجی با استفاده از سیمهای ترمو کوپل مس-کنستانتان انجام می‌شود. شدت جریان جرمی هر یک از دو سیال نیز با استفاده از روتامترهای مدرج شده اندازه گیری می‌شود. ضریب انتقال حرارت کلی از روی داده های تجربی محاسبه می‌شود. ضریب انتقال حرارت جریان لوله داخلی (مقطع دایره) با استفاده از روابط استاندارد بدست می‌آید. ضریب انتقال حرارت جریان لوله خارجی (مقطع حلقوی) از رابطه بین مقاومتها در یک جدار مرکب استوانه ای حاصل می‌شود و با مقادیر حاصل از روابط استاندارد مقایسه می‌گردد. ملاحظه می‌شود که ضرایب انتقال حرارت تجربی به میزان قابل ملاحظه ای از آنچه که روابط استاندارد برای یک لوله مستقیم و نرم پیش بینی می‌کنند بیشتر است. علل این اختلاف شناسایی و مورد بحث قرار می‌گیرد. عوامل خطا در اندازه گیری درجه حرارتها و جریانهای جرمی دو سیال مطالعه و تاثیر آن بر ضرایب انتقال حرارت تخمین زده می‌شود. نتایج تجربی در محدوده درجه حرارتها و جریانهای جرمی مورد استفاده در این پژوهش نشان می‌دهند که ضرایب انتقال حرارت جریان لوله خارجی بطور متوسط به نسبت ۱/۵ برای مبدل با جریانهای غیر هم جهت و ۳/٤ برای مبدل با جریانهای هم جهت از ضرایب انتقال حرارت جریان لوله داخلی کوچکتر است. این نسبتها با نسبتهای قابل پیش بینی توسط روابط استاندارد برای مبدل با جریانهای غیر هم جهت مطابقت نزدیک و برای مبدل با جریانهای هم جهت انحراف شدید دارد.

1. INTRODUCTION

Double pipe heat exchangers are the simplest recuperators in which heat is transferred from the hot fluid to the cold fluid through a separating cylindrical wall. It consists of concentric pipes separated by mechanical closures. Inexpensive, rugged and easily maintained, they are primarily adapted to high-temperature, high-pressure applications due to their relatively small diameters.

Double pipe heat exchangers have a simple construction. They are fairly cheap, but the amount of space they occupy is generally high compared with the other types. The amount of heat transfer per section is small, that makes the double pipe heat exchangers a suitable heat transfer device in applications where a large heat transfer surface is not required.

Although the performance and analysis of double pipe heat exchangers have been established long time ago, Abdelmessih and Bell [1] have taken a closer look to these exchangers recently. They have summarized some of the existing laminar flow heat transfer correlations in circular, horizontal, straight tubes. They have studied the effects of natural convection upstream of the bend and also the effects of secondary flow downstream the bend.

For a fully (or almost fully) developed velocity profile in the straight tube (upstream of the bend), where the thermal profile is not fully developed under any conditions, Abdelmessih and Bell [1] found that both forced and natural convection contribute to the heat transfer process according to the following correlation:

$$Nu = [4.36 + 0.327(Gr\,\mathrm{Pr})^{1/4}](\frac{\mu_b}{\mu_w})^{0.14} \qquad (1)$$

where all physical properties (except μ_w) are evaluated at the local bulk temperature. Nu is the local peripheral average Nussult number. The term representing the forced convection effect (4.36) will be recognized as the analytical result for fully developed laminar flow with constant properties and constant wall heat flux. The data used to generate Equation.1 covered the following ranges:

$$\begin{aligned} &120 \le \mathrm{Re} \le 2500 \\ &3.9 \le \mathrm{Pr} \le 110 \\ &2500 \le Gr \le 1130000 \\ &27 \le \frac{X}{d_i} \le 171 \end{aligned} \qquad (2)$$

Downstream from the bend, in addition to the forced and natural convection contributions, there is a secondary flow contribution. Adding the term correlating this effect gives the final Equation [5]:

$$Nu = [4.36 + 0.327(Gr\,\mathrm{Pr})^{1/4} + 1.955\times 10^{-6} \times \mathrm{Re}^{1.6}\, De^{0.8} e^{-0.07259(X/d_i)}] \times (\frac{\mu_b}{\mu_w})^{0.14} \qquad (3)$$

The data covered the following ranges:

$$\begin{aligned} &120 \le \mathrm{Re} \le 2500 \\ &40 \le \mathrm{Pr} \le 110 \\ &2500 \le Gr \le 50000 \\ &1.95 \le \frac{X}{d_i} \le 145 \\ &4.8 \le \frac{R_c}{r_i} \le 25.4 \end{aligned} \qquad (4)$$

Equation 3 has three limiting cases, as the curvature tends to zero, the Dean number tends to zero and Equation 3 reduces to Equation 1. The second case is the absence of natural convection, i. e., the Grashof number reduces to zero. The third case is for the fully developed velocity and temperature profiles in a straight tube and the absence of natural convection, i. e., both the Grashof number and the Dean number tend to zero, then Equation 3 reduces to:

$$Nu = 4.36(\frac{\mu_b}{\mu_w})^{0.14} \qquad (5)$$

For a nearly uniform wall heat flux, there will always be a natural convection contribution; i. e. Grashof number will not approach zero, unless gravity approaches zero.

The natural convection upstream of the bend (second term in Equation 1), the natural convection downstream of the bend (second term in Equation 3) and the secondary flow downstream of the bend (third term in Equation 3) contribute to the heat transfer process [1], [2], [3] and [4]. The present study is aimed to investigate the effect of these phenomena quantitatively by measuring the heat transfer coefficients and estimating their deviation from the predictions of standard correlations for straight tubes. It should be mentioned that, some of this deviation could be because of surface roughness of the tubes, as the standard correlations are generally for the smooth tubes. The question that what fraction of this deviation is because of

TABLE 1. The Exchanger Geometrical Data.

The inner tube inner diameter	16.5 mm
The inner tube outer diameter	21.5 mm
The outer tube inner diameter	27.5 mm
The inner tube height	650 mm
The outer tube height	600 mm
Total exchanger length	1500 mm
External tube area	0.101316 m^2
The tube material	steel

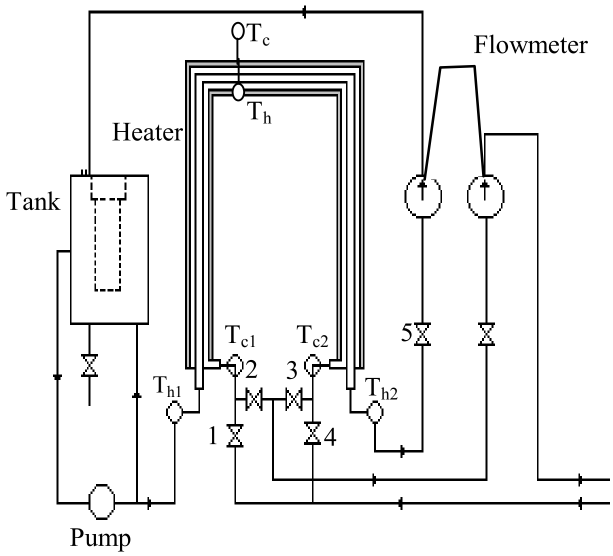

Figure1. Schematic diagram of the rig circuit.

natural convection, secondary flows, and surface roughness is still unanswered.

2. EXPERIMENTAL SETUP

The experiments, on which this work is based, were performed for the case of nearly uniform heat flux at the surface. This condition of uniform heat flux is probably closer to present the practical condition of operating double pipe heat exchangers in concurrent flow, where the outside fluid heats the fluid in the pipe, or vice versa. Constant heat flux should not be confused with constant surface temperature, where the latter is closely approximated when there is a phase change in one of the fluids, or when the fluids are in co-current flow.

The experimental rig was designed and constructed in the Heat Transfer Laboratory, Department of Mechanical Engineering, Kerman University. A schematic diagram of the rig circuit is shown in Figure. 1. The double pipe heat exchanger is in the vertical position; it is bent 90 degrees twice.

Some geometrical data about the exchanger are listed in Table 1.

An immersion heater heats water. A constant speed pump is used to pump the hot water from the tank into the inner tube. Water returns to the tank through valve 5. The cold water is supplied through the mains and drains through valves 2 and 3 in the counterflow and parallel flow conditions respectively. Shutting the valves 1 and 3 and opening the valves 2 and 4 achieve the counterflow conditions, while the parallel flow shutting the valves 2 and 4 and opening the valves 1 and 3 obtain these conditions.

Temperatures are measured at the inlet and outlet regions of the exchanger using copper-constantan thermocouple wires. The locations of thermocouples are so designed that the cold and hot stream temperatures at each terminal are measured at the same cross section. The insulating material covering the outer tube is 1.5 cm thick. It can be assumed that no heat from the hot stream dissipates into the atmosphere.

3. EXPERIMENTAL PROCEDURE, RESULTS AND ERRORS

Experimental Procedure The overall characteristics of the exchanger unit are investigated experimentally. The steps taken are as follows:

1. Measuring the temperatures of water at the inlet and outlet sections and also at an intermediate point half way between the inlet

TABLE 2. Temperature and Flow Rate Measurements.

	T_{h1}[°C]	T_{h2}[°C]	T_{hm}[°C]	W_h[kg/s]	T_{c1}[°C]	T_{c2}[°C]	T_{cm}[°C]	W_c[kg/s]
1	71.4	61.6	66.4	0.072	25.1	41.3	32.4	0.041
2	69.5	60.4	64.1	0.072	25.9	34.8	26.6	0.067
3	65.6	57.3	61.2	0.072	23.2	33.4	26.3	0.055
4	61.5	54.4	58.3	0.072	24.0	32.5	26.7	0.059
5	63.7	57.3	60.1	0.072	22.3	32.2	27.2	0.042
6	64.4	57.5	60.4	0.072	21.7	33.6	27.8	0.037
7	68.4	62.6	66.3	0.072	26.9	35.7	30.1	0.042
8	66.1	58.9	62.2	0.072	21.5	31.6	26.9	0.049

and outlet for each stream, using copper-constantan thermocouple wires.

2. Measuring the water flow rate for each stream using calibrated rotameters. Rotameters have been tested manually by measuring the amount of fluid collected in a vessel in a certain amount of time at room temperature. Rotameters have stainless steel floats.
3. Calculating the overall rate of heat transfer in the exchanger assuming heat losses from the outer tube stream to be negligible. Therefore the overall rate of heat transfer is equal to either the heat released from the hot stream or the heat absorbed by the cold stream, namely:

$$Q = (WC\Delta T)_c = (WC\Delta T)_h \tag{6}$$

4. Calculating the log-mean temperature difference between the two streams. The total heat transfer rate from the hot fluid to the cold fluid in the exchanger is expressed as:

$$Q = UA(LMTD) \tag{7}$$

5. Calculating the overall heat transfer coefficient at different operating conditions assuming to be constant throughout the exchanger, using Equation 7.
6. Calculating the film heat transfer coefficient for the inner tube side flow, using the Dittus-Boelter [6] correlation:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n} \tag{8}$$

The value of n is 0.3 if the inner tube side fluid is being cooled and 0.4 if the inner tube side fluid is being heated. The Dittus-Boelter correlation is valid for fully developed turbulent flow (Re>10000) in smooth tubes for fluids with Prandtl numbers ranging from about 0.6 to 100 and with moderate temperature differences between the wall and the fluid conditions. The other restriction for Equation 8 is that it is used when constant heat flux boundary condition is applied.

7. Calculating the film heat transfer coefficient for the outer tube side from Equation 17.

Experimental Results The experimental procedure for each run was to set a pre-defined temperature and flow rate for the hot water stream, set the cold water flow rate and then wait for the steady state conditions to be reached. Following steps 1 to 7 for each run provides a value for the heat transfer coefficient of the outer tube flow. Repeating the experiment for different operating conditions, results in a set of tabulated data. Table 2 contains a range of 8 operating conditions measured as described in steps 1 and 2. The first 4 rows correspond to counterflow conditions while the second 4 rows correspond to parallel flow conditions.

Table 3 contains the heat transfer characteristics calculated as described in steps 3 to 5. Table 3 corresponds to the data listed in Table 2. Calculating the film heat transfer coefficient for the tube side flows as mentioned in step 6, requires one to know viscosity, Reynolds number, Prandtl

TABLE 3. Heat Transfer Characteristics of the Exchanger.

	Q_h [kW]	LMTD[°C]	$U_\circ [W/m^2\,°C]$	Q_c [kW]	$1-Q_c/Q_h$
1	2.956	33.20	879	2.755	6.8%
2	2.745	34.60	783	2.512	8.5%
3	2.504	33.14	746	2.366	5.5%
4	2.142	29.69	712	2.093	2.3%
5	1.931	32.40	588	1.761	8.8%
6	2.082	32.40	634	1.851	11.1%
7	1.750	33.67	513	1.544	11.8%
8	2.172	35.25	608	2.094	3.6%

TABLE 4. Inner Tube Side Heat Transfer Coefficient Deductions.

	$\mu [kg/m.s]$	Re	Pr	Nu	k[W/m.°C]	$h_i [W/m^2\,°C]$
1	0.000439	12585	2.802	59.67	0.658	2380
2	0.000453	12196	2.896	58.75	0.657	2344
3	0.000473	11681	3.037	57.58	0.654	2283
4	0.000493	11207	3.178	56.48	0.651	2229
5	0.000480	11510	3.084	57.19	0.653	2263
6	0.000480	11510	3.084	57.19	0.653	2263
7	0.000439	12585	2.802	59.67	0.658	2380
8	0.000466	11856	2.990	58.02	0.655	2303

number, Nusselt number, and conductivity of water. These data for operating conditions corresponding to Table 2 are listed in Table 4. The inner tube side heat transfer coefficients based on Equation 8 and the outer tube side heat transfer coefficients based on Equation 17 are listed in Tables 4 and 6 respectively.

Experimental Errors The accuracy for measured heat transfer coefficients is affected by the effectiveness of thermal insulation, the amount of heat lost to the ambient, the accuracy of the thermocouple system, and the accuracy of rotameters. Simple one-dimensional calculations clearly indicate that 99% of the heat transferred from the inner tube flow goes to the outer tube flow and only 1% is lost into the insulation material. It is estimated that the thermocouple system including the thermocouple wire variations, digital voltmeter characteristics and all associated measurements, communications and transformation procedure is able to give readings of $\pm 0.1°$C about the true temperature. The rotameters used to measure the inner tube and outer tube flow rates were quoted by the manufacturer as being able to measure to $\pm 2\%$ of the readings. Thus for a typical run, where the stream temperature difference is about 18°C, an error of about:

$$\pm\sqrt{(0.01)^2 + (\frac{0.2}{18})^2 + (0.02)^2} = \pm 1.5\%$$

TABLE 5. Outer Tube Heat Transfer Properties.

	μ $[kg/m.s]$	Re	Pr	k [W/m°C]
1	0.000795	1339	5.414	0.6174
2	0.000863	2016	5.93	0.6101
3	0.000877	1629	6.034	0.6087
4	0.000863	1776	5.93	0.6101
5	0.000863	1265	5.93	0.6107
6	0.000850	1131	5.827	0.6116
7	0.000822	1326	5.621	0.6145
8	0.000863	1475	5.93	0.6107

is expected for the evaluated heat transfer coefficients. The errors are much greater when the stream temperature differences are lower. For example, for the stream temperature difference of 3°C, the estimated error in the evaluated heat transfer coefficients is:

$$\pm\sqrt{(0.01)^2 + (\frac{0.2}{3})^2 + (0.02)^2} = \pm 7\%$$

The mean heat transfer coefficient, the variance, the standard deviation and the coefficient of variation for both counter-flow and parallel-flow arrangements are listed in Table 7. It can be seen that for counter-flow arrangement, 75% of the data are within one standard deviation from the mean and 100% of the data are within two standard deviations from the mean. For parallel-flow arrangement, 50% of the data are within one standard deviation from the mean and 100% of the data are within two standard deviations from the mean.

4. COMPARISON OF DATA WITH STANDARD CORRELATIONS

The Hausen Correlation The Hausen correlation [7] may be used for the outer tube flow, in this case the hydraulic diameter is:

$$D_e = D_i - D_o \tag{9}$$

where D_i is the outer tube inner diameter and D_o is the inner tube outer diameter, for the case at hand D_e is 6 mm. The Hausen correlation reads as follows:

$$Nu = 3.66 + \frac{0.0668Gz}{1 + 0.04Gz^{\frac{2}{3}}} \tag{10}$$

where Gz is the Graetz number defined as follows:

$$Gz = \mathrm{Re}\,\mathrm{Pr}\,D_e / L \tag{11}$$

The results appeared in Table 6 have been multiplied by 1.2 to take care of the uniform heat

TABLE 6. Comparison between the Experimental Data and the Standard Correlations for Laminar Flow in the Outer Tube.

	$h_o[W/m^2.°C]$ experimental data, Equation $(A.5)$	Gz=RePrD$_e$/L Graetz number	$h_o[W/m^2.°C]$ Equation 10	$h_o[W/m^2.°C]$ Equation 12	Kays [14] and Sellars et al [12]
1	1861	29.00	625.2	705.6	473
2	1496	47.82	702	823.2	549
3	1395	39.33	664.8	770.4	507
4	1303	42.13	678	789.6	519
5	933	30.00	624	705.6	489
6	1054	26.36	607.2	676.8	459
7	741	29.81	626.4	709.2	481
8	974	35	646.8	742.8	499

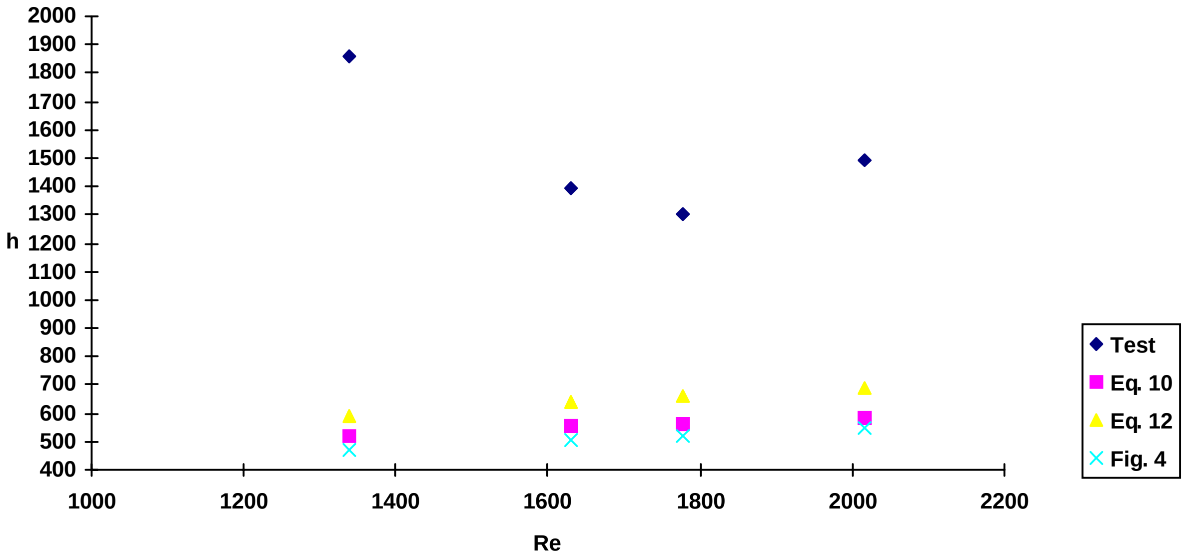

Figure2. The outer tube side heat transfer coefficients-Counterflow.

flux boundary condition at the surface of the existing heat exchanger. This is because the Hausen correlation is for isothermal wall.

Table 5 includes viscosity, Reynolds number, Prandtl number, Nusselt number and conductivity of the outer tube stream corresponding to the operating conditions listed in Table 2.

The experimental heat transfer coefficients are then calculated using Equation 17 and inserted in Table 6.

The outer tube heat transfer coefficients evaluated according to the Hausen correlation and using the data listed in Table 5 are included in Table 6. The outer tube heat transfer coefficients can now be compared with those evaluated experimentally. The comparison is illustrated in Figures 2 and 3 for the counterflow and parallel flow conditions respectively. The points indicated by symbol ■ are predictions of the Hausen correlation, while the points indicated by symbol ♦ are the experimental results.

It should be mentioned that some different combinations of standard correlations have been recommended to predict the film heat transfer coefficients of inner tube flow and outer tube flow by different workers. For example the ESDU [5] and the Kern [10] correlations are used to predict the inner tube side and the outer tube side heat transfer coefficients, respectively in the TASC [11] computer program.

It is evident that the results obtained by the independent workers are different from each other.

Table7. Statistical Analysis of Experimental Data for Heat Transfer Coefficient of Flow in the Outer Tube

	Mean Heat Transfer Coefficient	Variance	Standard Deviation	Coefficient of Variation
Counter-flow	1514	44854	212	0.14
Parallel-flow	926	13240	115	0.124

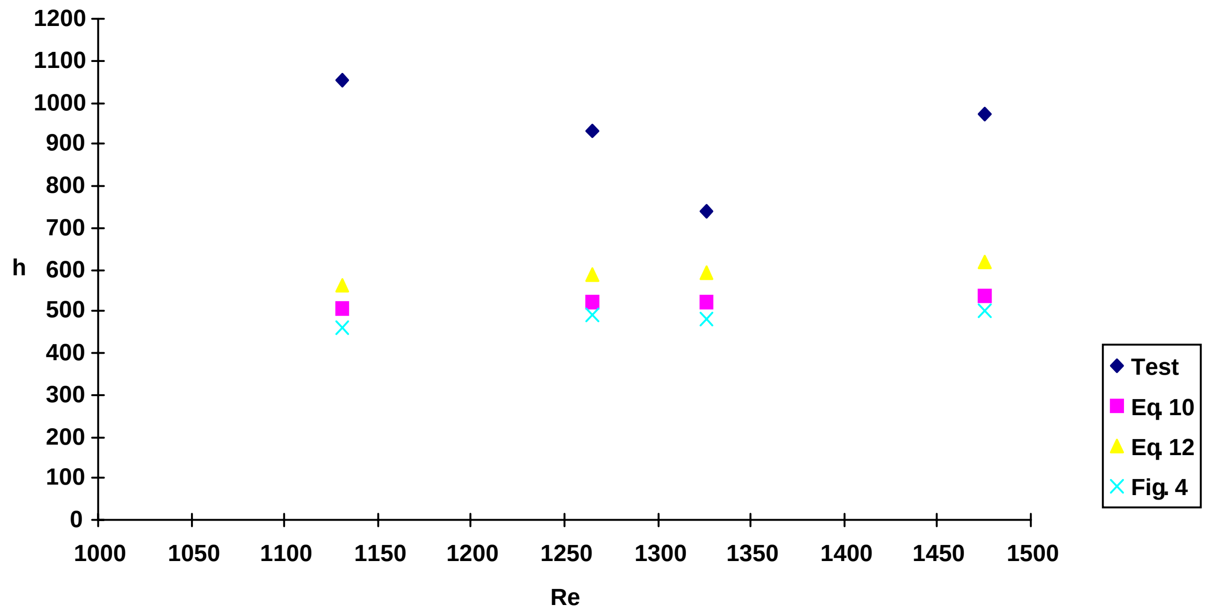

Figure 3. The outer tube side heat transfer coefficients-Parallel flow.

The Dittus-Boelter [6] and Hausen [7] correlations have been used to predict the film heat transfer coefficients of inner tube flow and outer tube flow, respectively, and the results have been compared with those deduced experimentally in Table 6. Two other correlations are used to predict the film heat transfer coefficients of the outer tube side in the same manner as the Hausen correlation was applied and the results will be compared with those deduced experimentally in Table 6. In these two cases the Dittus-Boelter [6] correlation is monotonically used to predict the film heat transfer coefficients of the inner tube side.

The Sieder-Tate correlation The Sieder-Tate [12] correlations have been used to design the double pipe heat exchangers since 1950 and they are strongly recommended by Kern [10] in his old but reliable text. The Sieder-Tate correlations can be used for predicting the film coefficients of flow in both the inner tube side and the outer tube side of a double pipe heat exchanger. They can be used for both heating and cooling of a number of fluids, principally petroleum fractions, in horizontal and vertical tubes:

$$Nu = 1.86 \mathrm{Re}^{\frac{1}{3}} \mathrm{Pr}^{\frac{1}{3}} \left(\frac{D}{L}\right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{12}$$

The results appeared in Table 6 have been multiplied by 1.2 to take care of the uniform heat flux boundary condition at the surface of the existing heat exchanger. This is because the Sieder-Tate correlation is for isothermal wall. Equation 12 applies for laminar flow (Re<2100). L is the total heat transfer length. The outer tube heat transfer coefficients are recalculated based on Equation 12. The results are included in Table 6 and the comparison is shown in Figures 2 and 3 for the counterflow and parallel flow conditions respectively. The points indicated by the symbol Δ are predictions of the Sieder-Tate correlation, while the points indicated by the symbol ♦ are the experimental results.

The Heat Exchanger Design Handbook, HEDH [13], has also recommended the Sieder-Tate correlations to be used for predicting the film coefficients of single-phase flow in both the inner tube and the outer tube of a double pipe heat exchanger.

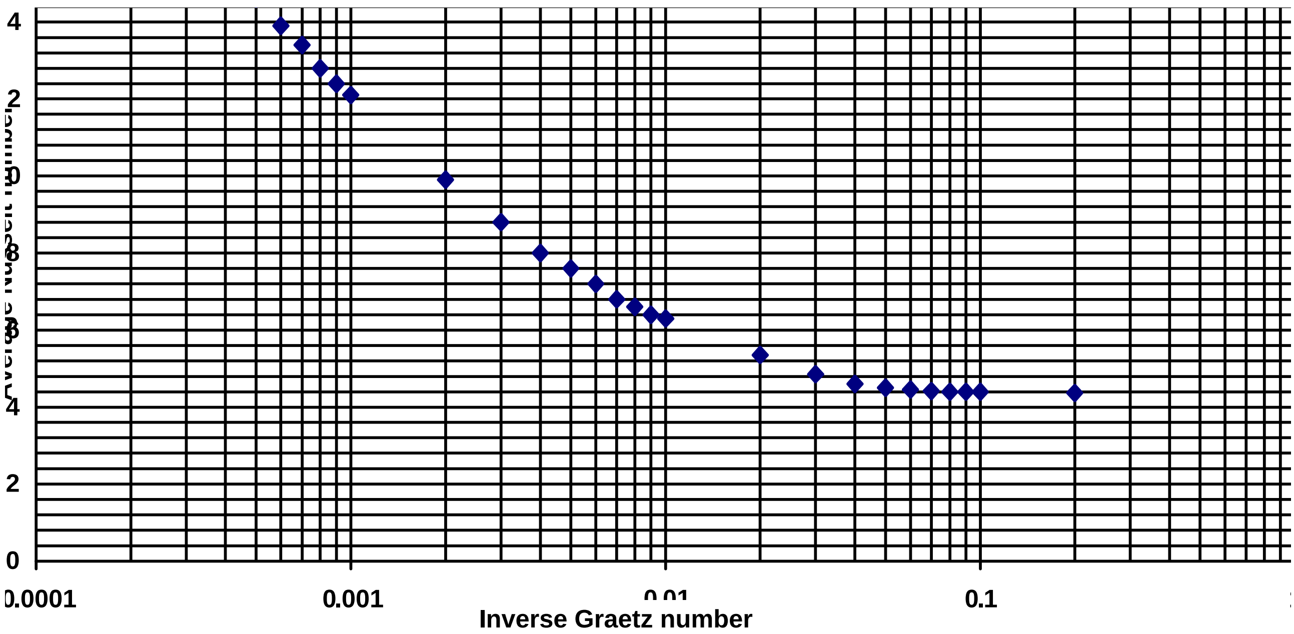

Figure 4. Average Nusselt numbers for circular tube thermal entrance regions in fully developed laminar flow [11].

The Kays and Sellars, Tribus and Klein Predictions Kays [8] and Sellars, Tribus, and Klein [9] calculated the total and average Nusselt numbers for laminar entrance regions of circular tubes for the case of a fully developed velocity profile. The results of their analyses are shown as the variation of average Nusselt number in terms of the inverse of Graetz number [14]. Figure 4 is a graphical presentation of the average Nusselt numbers for circular tube thermal entrance region in fully developed laminar flow, deduced by Kays, and Sallars [14], introduced here as a source of comparison. The outer tube heat transfer coefficients are recalculated based on the above predictions. The results are included in Table 6 and the comparison is shown in Figures 2 and 3 for the counterflow and parallel flow conditions respectively. The points indicated by the symbol × in Figures 2 and 3 are predictions of the Kays and Sellars et al (extracted from Figure 4), while the points indicated by the symbol ♦ are the experimental results.

5. CONCLUSIONS

The outer tube side heat transfer coefficients deduced from experimental data are compared with those evaluated based on standard correlations. The comparison is illustrated schematically in Figures 2 and 3 for counterflow and parallel flow conditions, respectively.

In both the counterflow and parallel flow conditions, all three standard correlations predict lower heat transfer coefficients compared with the experimental results. The Sieder-Tate [12] correlation predicts the highest values among the three standard correlations; they are still lower by a factor of 1.04 to 2.64. That means, the standard correlations for laminar flow in the outer tube side in which the Nusselt numbers are proportional to $Re^{0.33}$ underestimate the heat transfer coefficients. The differences between the predictions of standard correlations (Equation 10, Equation 12 and Figure 4) as shown in Figures 2 and 3 are not considerable. This close agreement for empirical correlations is rather strange. In fact a simple or minor error could have brought the results together.

The outer tube side Reynolds numbers are lower by a factor of 6 to 10 (third column in Tables 5 and 6), but the hydraulic diameter is lower by a factor of nearly 3 (Table 1). Therefore, if a Dittus-Boelter type of correlation (Equation 8) is valid for both the outer and inner tube side streams, the outer tube side heat transfer coefficients would be expected to be smaller than the inner tube side heat transfer coefficients by a factor of:

$$\frac{h_i}{h_o} = \left(\frac{\mathrm{Re}_i}{\mathrm{Re}_o}\right)^{0.8} \left(\frac{D_o}{D_i}\right) \cong (8)^{0.8} \left(\frac{1}{3}\right) \cong 1.7$$

The above ratio for the experimental results ranges between 1.28 and 1.71 for the counterflow arrangement, and between 2.15 and 4.62 for the parallel flow arrangement. The agreement with predictions is very good for the counterflow arrangement, but not very good for the parallel flow arrangement. The discrepancy may be because of four reasons. *Firstly*, there was probably heat transfer in regions between the thermocouple wires and the exchanger terminals, so that the actual heat transfer area was larger than calculated. *Secondly*, there was probably a higher coefficient in certain regions, such as in turnaround region, than predicted by straight pipe equation. *Thirdly*, the effect of natural convection in internal flows, especially when the forced and free convection currents are in the same direction (aiding flow), can enhance the heat transfer coefficients by a factor of 1.41compared with the case when the heat transfer mechanism is assumed strictly on the basis of laminar forced convection [solved example in 14]. *Fourthly*, the standard correlations are generally presented for the smooth heat transfer surfaces, while in a real exchanger the heat transfer surfaces are not smooth and those results in higher heat transfer coeffficiets.

Temperatures have been rounded to the nearest decimal. Stream temperature differences ranged from 3 to 18 degrees. Stream flow rates are likely to be in error by $\pm$ 2 percent. The heat loss through the thermal insulation is about 1%. With this error analysis, the error in the measured heat transfer coefficients is 1.5-7 percent.

Hewitt et al [15] and also HEDH [13] have recommended that the Dittus-Boelter correlation (Equation 8) with n=0.4 to be used for predicting the film coefficients of both the inner tube side and the outer tube side flows for both laminar and turbulent flows. Using the Dittus-Boelter correlation with $\mathrm{Re}^{0.8}$ for the laminar flow in the outer tube side increases the heat transfer coefficients by a factor 2 to 3. That makes us believe that Hewitt et al [15] and also HEDH [13] are correct.

The experimental heat transfer coefficients do not have a direct relationship with the outer tube side Reynolds numbers. This behaviour is not repeated by any of the standard correlations. This is because the experimental heat transfer coefficients are governed by the overall heat transfer coefficients (Equation. 17), rather than by the Reynolds number.

The purpose of this article is to recognise the mechanisms of heat transfer that occur in double pipe heat exchangers and to report higher heat transfer coefficients in the laminar flow regime. The phenomena that contribute to the heat transfer enhancement are mainly natural convection, secondary flows, and surface roughness. There are of course other reasons for this enhancement that need to be studied experimentally. The question that what fraction of this enhancement is because of natural convection, secondary flows, and surface roughness is still unanswered.

6. NOMENCLATURE

A	Tube area exposed to heat transfer, m^2
C	Stream specific heat, $J / kg.°C$
d_i	Inner tube inner diameter, m
D_i	Outer tube inner diameter, m
D_o	Inner tube outer diameter, m
De	Dean number $= \mathrm{Re}\sqrt{r_i / R_c}$
Gr	Grashof number
Gz	Graetz number
h	Film heat transfer coefficient, $W / m^2.°C$
k	Thermal conductivity, $W / m.°C$
L	Total length exposed to heat transfer, m
LMTD	Logarithmic Mean Temperature

	Difference, $^\circ C$
n	Exponent in Equation. 8
Nu	Nusselt number
Pr	Prandtl number
Q	Heat transfer rate, W
r_i	Inner tube inner radius, m
r_o	Inner tube outer radius, m
R_c	Bend radius measured to centerline of U-tube, m
Re	Reynolds number
T	Stream temperature, $^\circ C$
U	Overall heat transfer coefficient, $W/m^2.^\circ C$
W	Stream mass flow rate, kg/s
μ	Dynamic viscosity, kg/m.s

7. SUBSCRIPTS

c	Cold
e	Equivalent
i	Inner
h	Hot
o	Outer
w	Wall

8. APPENDIX

The heat transfer rate in a composite cylindrical wall is expressed in terms of the total temperature difference and the resistance of different layers:

$$Q = (T_h - T_c) / \left(1/h_i A_i + (\ln r_o / r_i)/(2\pi kL) + 1/h_o A_o\right) \tag{13}$$

The overall heat transfer coefficient is defined as the inverse of the sum of resistances to heat flow, so that Equation 13 reduces to:

$$Q = UA(T_h - T_c) \tag{14}$$

where U can be defined in terms of either the internal or external tube areas. In each case Equation 14 applies:

$$Q = U_i A_i (T_h - T_c) = U_o A_o (T_h - T_c) \tag{15}$$

The values of U_i and U_o are as follows:

$$U_i = 1/ \left(1/h_i + (A_i \ln r_o / r_i)/(2\pi kL) + A_i / h_o A_o\right) \tag{16}$$

and

$$U_o = 1/ \left(A_o / A_i h_i + (A_o \ln r_o / r_i)/(2\pi kL) + 1/h_o\right) \tag{17}$$

$T_h - T_c$ is the temperature difference between the hot and cold fluids at a local point. This temperature difference varies with position along the path of flow. To determine the rate of heat transfer between the hot and cold fluids is therefore a complicated matter. In practice it is convenient to use an average effective temperature difference for the entire heat exchanger. Only if the overall heat transfer coefficient, U, is constant, this average effective temperature difference turns out to be the logarithmic mean temperature difference, LMTD, defined as:

$$LMTD = \left((T_{h2} - T_{c2}) - (T_{h1} - T_{c1})\right) / \left(\ln (T_{h2} - T_{c2})/(T_{h1} - T_{c1}) \right) \tag{18}$$

The subscripts used in Equation 18 are consistent with those used in Table 2 and therefore apply for parallel flow conditions only. In counterflow conditions, however, the subscripts 1 and 2 for the cold fluid temperatures must be replaced.

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