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The overall heat transfer characteristics of a double pipe heat exchanger: comparison of experimental data with predictions of standard correlations

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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 are made. 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.

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, A. N., and Bell, K. J. [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:

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

where all physical properties (except μ_b) are evaluated at the local bulk temperature. Nu is the local peripheral average Nusselt 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 Eq.1 covered the following ranges:

$$120 \le Re \le 2500$$
$$3.9 \le Pr \le 110$$
$$2500 \le Gr \le 1130000 \quad (2)$$
$$27 \le \frac{X}{d_i} \le 171$$

Eq. 1 has an absolute average percent deviation from the data of 5.98%.

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 [2]:

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

The data covered the following ranges:

$$120 \le Re \le 2500$$
$$40 \le Pr \le 110$$
$$2500 \le Gr \le 50000 \quad (4)$$
$$1.95 \le \frac{X}{d_i} \le 145$$
$$4.8 \le \frac{R_c}{r_i} \le 25.4$$

Eq. 3 has an average absolute deviation of 9.9% in comparison to the data. Eq. 3 has three limiting cases, as the curvature tends to zero, the Dean number tends to zero and Eq. 3 reduces to Eq. 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 Eq. 3 reduces to:

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

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.

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 representing 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 Fig. 1. Some geometrical data about the exchanger are listed in Table 1. The double pipe heat exchanger is in the vertical position; it is bent 90 degrees twice.

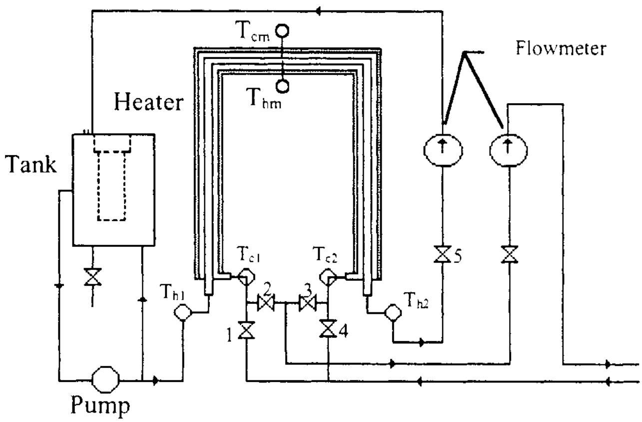

Fig1 Schematic diagram of the rig circuit

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

Water is heated by an immersion heater. 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. The counterflow conditions are achieved by shutting the valves 1 and 3 and opening the valves 2 and 4, while the parallel flow conditions are obtained by shutting the valves 2 and 4 and opening the valves 1 and 3.

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 and results

3.1 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 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:

$$Q = (WC\Delta T)_c = (WC\Delta T)_h \quad (6)$$

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 Eq. (*A*.5).
6. Calculating the film heat transfer coefficient for the inner tube side flow, using the Dittus-Boelter [3], correlation:

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

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 Eq. 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 Eq. (*A*.2).

3.2 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 2. Temperature and flow rate measurements

	Th1 [°C]	Th2 [°C]	Thm [°C]	Wh [kg/s]	Tc1 [°C]	Tc2 [°C]	Tcm [°C]	Wc [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

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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.

Table 3. Heat transfer characteristics of the exchanger

	Q [kW]	LMTD [°C]	$U_o [W/m^2 °C]$	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%

Calculating the film heat transfer coefficient for the tube side flow as mentioned in step 6, requires one to know viscosity, Reynolds number, Prandtl 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 Eq. 8 and the outer tube side heat transfer coefficients based on Eq (*A*.2) are listed in Tables 5 and 6 respectively.

Table 4. Inner tube side heat transfer coefficient deductions

	μ [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

4 Comparison of data with standard correlations

4.1 The Hausen correlation:

The Hausen correlation [4] 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:

$$Nu = 1.20(3.66 + \frac{0.0668Gz}{1 + 0.04Gz^{\frac{2}{3}}}) \tag{10}$$

where Gz is the Graetz number defined as follows:

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

The factor 1.20 appeared in Eq. 10 takes care of the uniform heat 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.

Table 5. Outer tube heat transfer properties

	μ[kg/m.s]	Re	Pr	Nu	k[W/m°C]
1	0.000795	1339	5.414	6.92	0.6174
2	0.000863	2016	5.93	32.34	0.6101
3	0.000877	1629	6.034	41.85	0.6087
4	0.000863	1776	5.93	44.20	0.6101
5	0.000863	1265	5.93	24.50	0.6107
6	0.000850	1131	5.827	4.82	0.6116
7	0.000822	1326	5.621	5.89	0.6145
8	0.000863	1475	5.93	24.50	0.6107

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 Figs. 2 and 3.

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, Eq. (*A*.5)	Gz= $RePrD_e/L$ Graetz Number	h_o $[W/m^2.°C]$ Eq. 10	h_o $[W/m^2.°C]$ Eq. 11	Kays[5] and Sellars, and Klein[6]
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

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 [2] and the Kern [7] correlations are used to predict the inner tube side and the outer tube side heat transfer coefficients, respectively in the TASC [8] computer program.

It is evident that the results obtained by the independent workers are different from each other. The Dittus-Boelter [3] and Hausen [4] 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 [3] correlation is monotonically used to predict the film heat transfer coefficients of the inner tube side.

4.2 The Sieder-Tate Correlation:

The Sieder-Tate [9] correlations have been used to design the double pipe heat exchangers since 1950 and they are strongly recommended by Kern [7] 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.

$$Nu = 1.20[1.86\, Re^{\frac{1}{3}}\, Pr^{\frac{1}{3}} (\frac{D}{L})^{\frac{1}{3}} (\frac{\mu}{\mu_w})^{0.14}] \tag{12}$$

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$$Nu = 1.20[0.027 Re^{0.8} Pr^{\frac{1}{3}} (\frac{\mu}{\mu_w})^{0.14}] \qquad (13)$$

The factor 1.20 appeared in Eqs. 12 and 13 takes care of the uniform heat flux boundary condition at the surface of the existing heat exchanger. This is because the Sieder-Tate correlations are for isothermal wall. Eq. 12 applies for laminar flow (Re 2100), while Eq. 13 takes care of turbulent flow. L is the total heat transfer length. The outer tube heat transfer coefficients are recalculated based on Eq. 12. The results are included in Table 6 and the comparison is shown in Figs. 2 and 3.

The Heat Exchanger Design Handbook, HEDH [10], 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.

4.3 The Kays and Sellars, Tribus and Klein predictions:

Kays [5] and Sellars, Tribus, and Klein [6] 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 these analyses are shown in Fig. 4 in terms of the inverse of Graetz number [11]. 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 Figs. 2 and 3.

Fig. 2. The shell side heat transfer coefficients-Counterflow

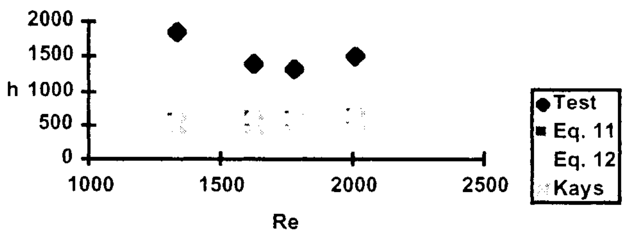


Fig. 3. The shell side heat transfer coefficients-Parallel flow

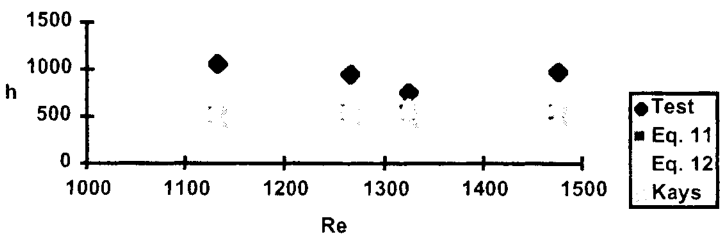
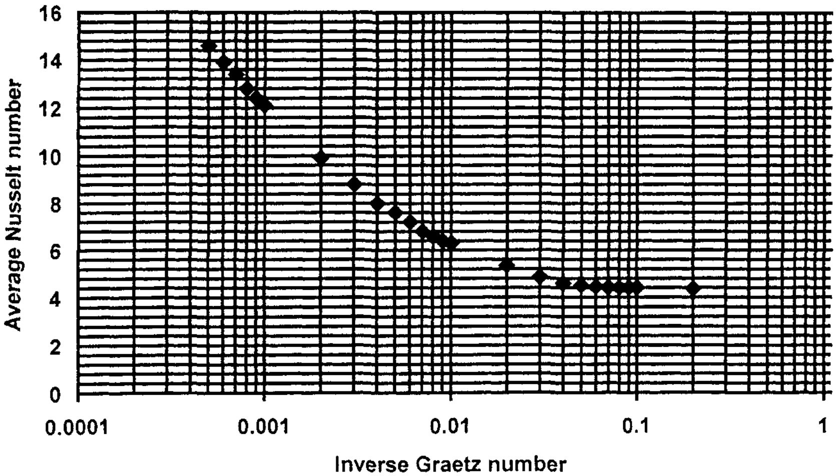


Fig. 4. Average Nusselt numbers for circular tube thermal entrance regions in fully developed laminar flow [12]



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 Figs. 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 [9] 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 outer tube side Reynolds numbers are lower by a factor of 6 to 10, but the hydraulic diameter is lower by a factor of nearly 3. Therefore the outer tube side heat transfer coefficients would be expected to be similar to the tube side heat transfer coefficients. The agreement with predictions is not bad. The discrepancy may be because of three 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 1.41 compared with the case when the heat transfer mechanism is assumed strictly on the basis of laminar forced convection [11].

Temperatures have been rounded to the nearest decimal. Stream temperature differences ranged from 3 to 18 degrees. Heat loads, deduced from these temperature differences are likely to be in error by 1 to 2 percent. With this error in heat load, the error in the measured heat transfer coefficients is 3-6 percent.

Hewitt et al [12] and also HEDH [4] have recommended that the Dittus-Boelter correlation 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 $Re^{0.8}$ 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 [6] and also HEDH [4] 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 (Eq. 5), 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.

References

[1] Abdelmessih, A. N., and Bell, K. J. :Effect of Mixed Convection and U-Bends on the Design of Double Pipe Heat Exchangers, Heat Transfer Engineering, vol. 20, no. 3, 1999.

[2] Engineering Science Data Unit, Paper 67016, 1967; Paper 68007, 1968; Paper 68006 with amendment A, 1970; 4, Hamilton Place.

[3] Dittus, F. W., and Boelter L. M. K. : University of California (Berkeley) Pub. Eng., vol. 2, p. 443, 1930.

[4] Hausen, H.: Darstellung des Warmeuberganges in Rohren durch verallgemeinerte potenzbeziehungen, VDI Z., no. 4, p.91, 1943.

[5] Kays, W.M.: Convective heat and Mass Transfer, pp. 187-190, McGraw-Hill, New York, 1966.

[6] Sellars, J.R., Tribus, M., and Klein J.S.: Heat transfer to laminar flows in a round tube or flat conduit: The Graetz problem extended, Trans. ASME, vol.78, p.441, 1956.

[7] Kern, D. Q. :Process Heat Transfer, MacGraw-Hill, New York, 1950.

[8] HTFS News Sheet, no.6, December 1985.

[9] Sieder, E. N., and Tate, F. E. : Heat Transfer and Pressure Drop in Tubes, Ind. Eng. Chem., vol. 28, pp. 1429-1436, 1936.

[10] Schlunder, E. U., Editor-in-Chief: Heat Exchanger Design Handbook,

Transactions on Modelling and Simulation vol 30, © 2001 WIT Press, www.witpress.com, ISSN 1743-355X

Section 3.2, Hemisphere Publishing Corporation, New York, 1989.
[11] Holman, J.P.: Heat transfer, eighth edition, McGraw-Hill, New York .
[12] Hewitt, G. F., Shires, G. L., and Bott, T.R. :Process Heat Transfer, CRC Press, London, ISBN 0-8493-9918-1,1994.
[13] Lord, R. C., Minton, P. E., and Slusser, R. P., Union Carbide Corp.: "Design of Heat Exchangers" in Process Heat Exchange, MacGraw-Hill, New York, 1970.

Appendix A

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

$$Q/A = (T_h - T_c)/\left(1/h_iA_i + (\ln r_o/r_i)/(2\pi kL) + 1/h_oA_o\right) \quad (A.1)$$

The overall heat transfer coefficient is defined as the inverse of the sum of resistances to heat flow, so that Eq.(*A*.1) reduces to:

$$Q = UA(T_h - T_c) \quad (A.2)$$

where U can be defined in terms of either the internal or external tube areas. In each case Eq. (*A*.2) applies:

$$Q = U_iA_i(T_h - T_c) = U_oA_o(T_h - T_c) \quad (A.3)$$

The values of U_i and U_o are as follows:

$$U_i = 1/\left(1/h_i + (A_i \ln r_o/r_i)/(2\pi kL) + A_i/h_oA_o\right) \quad (A.4)$$

And

$$U_o = 1/\left(A_o/A_ih_i + (A_o \ln r_o/r_i)/(2\pi kL) + 1/h_o\right) \quad (A.5)$$

$T_h - T_c$ 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:

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

$$(\ln[(T_{h2} - T_{c2})/(T_{h1} - T_{c1})]) \quad (A.6)$$

The subscripts used in Eq.(*A*.6) 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.