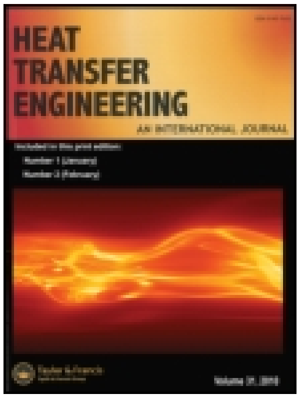

Heat Transfer Engineering
Publication details, including instructions for authors and subscription information:
http://www.tandfonline.com/loi/uhte20

Heat Transfer Coefficients for Turbulent Flow in Concentric Annular Ducts

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Published online: 10 Oct 2011.

Heat Transfer Engineering, 30(6):431–436, 2009

ISSN: 0145-7632 print / 1521-0537 online
DOI: 10.1080/01457630802528661

Heat Transfer Coefficients for Turbulent Flow in Concentric Annular Ducts

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On the basis of a large number of experimental data from the literature, correlations were developed for the heat transfer coefficient for turbulent flow in concentric annular ducts. A proven correlation for heat transfer in circular tubes was extended by factors that take into consideration the effect of the diameter ratio of the annulus and the different boundary conditions for heating or cooling.

INTRODUCTION

The simplest kind of a heat exchanger for transferring heat from one fluid to another is a double-pipe made up of two concentric circular tubes. One fluid flows through the inside tube and the other in concurrent or countercurrent flow through the annular passage.

Besides separate double-pipe heat exchangers, heating or cooling jackets for tubes make up this design. However, annular passages may occur, as in heat exchangers in other setups: to heat a fluid by electricity, a heating-rod may be put into the center of a circular tube, or the fluid in the annulus is heated or cooled from the outer tube and a dummy rod is placed in the center of the tube to increase the velocity of the fluid in the gap. Finally, the fluid flowing in the annulus may be heated or cooled from both sides, from the inner tube and from the outer tube.

To find the dimensions of the heat exchanger to solve a given problem, equations are needed for the heat transfer coefficient and the friction factor for the annulus.

The practicing engineer cannot follow the development of knowledge as published in the many journals and conference papers; rather, s/he has to rely on handbooks representing the state of the art. These are the basis for programs to simulate flow and heat transfer as well.

The *Heat Exchanger Design Handbook* (HEDH) [1], the *VDI-Heat Atlas* [2], and the *VDI-Waermeatlas* [3], inter alia, contain correlations on heat transfer and pressure drop in concentric annular ducts. In [1–3], the present author has published correlations on heat transfer for turbulent flow in concentric annuli. It is the matter of concern of this paper to show how these correlations have to be improved to represent the state of the art.

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DEFINITIONS

Figure 1 demonstrates the three different boundary conditions of heat transfer described above. Heat may be transferred at the inner wall of the annular duct while the outer wall is insulated (see Figure 1a, subscript i), at the outer wall while the inner wall is insulated (see Figure 1b, subscript o), or at both walls of the passage (see Figure 1c, subscript b).

The mean heat transfer coefficient over the whole length L of an annular duct is defined as

$$\alpha = \frac{\dot{q}}{\Delta T_{LM}} \tag{1}$$

in which the logarithmic mean temperature difference ΔT_{LM} is given by

$$\Delta T_{LM} = \frac{(T_w - T_{b,in}) - (T_w - T_{b,out})}{\ln[(T_w - T_{b,in})/(T_w - T_{b,out})]} \tag{2}$$

$T_{b,in}$ is the inlet bulk temperature, $T_{b,out}$ is the outlet bulk temperature of the fluid, and T_w is the temperature of the wall of the duct. The heat flux into the fluid is $\dot{q}$.

The friction factor of the duct is defined by the following equation:

$$\Delta p = f \frac{\rho\, u^2}{2} \frac{L}{d_h} \tag{3}$$

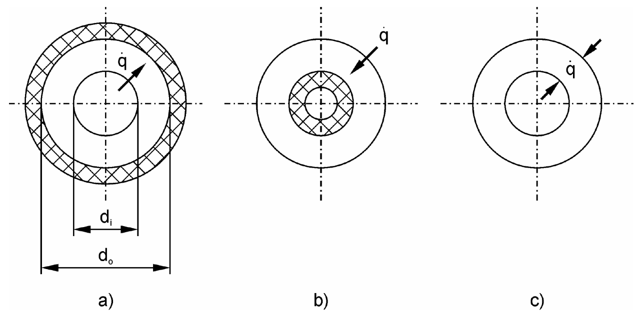

Figure 1 Boundary conditions for concentric annular duct flow: (a) heat transfer from the inner tube (outer tube insulated), (b) heat transfer from the outer tube (inner tube insulated), and (c) heat transfer from both tubes to the annular flow.

where Δp is the pressure loss of the fluid over the length L of the duct, ρ is the density of the fluid, u is the superficial velocity, and the hydraulic diameter of the annular duct is

$$d_h = d_o - d_i \tag{4}$$

where d_o is the inner diameter of the outer tube and d_i is the outer diameter of the inner tube (see Figure 1).

Heat transfer coefficients and friction factors in annular ducts depend on the ratio of the diameters d_i and d_o:

$$a = \frac{d_i}{d_o} \quad (0 \leq a \leq 1) \tag{5}$$

With annular flow, the maximum velocity is not in the middle of the gap but is shifted toward the inner wall of the duct as $a = d_i/d_o$ decreases.

The limiting case, $a = 0$, is a circular tube with an "infinitesimal thin wire" at the center [4]. The other limiting case of annular ducts, $a = 1$, is the parallel plate geometry.

Other dimensionless numbers are the Nusselt number:

$$Nu = \frac{\alpha\, d_h}{\lambda}, \tag{6}$$

where λ is the thermal conductivity of the fluid; the Reynolds number:

$$\mathrm{Re} = \frac{u\, d_h}{\nu}, \tag{7}$$

where ν is the kinematic viscosity of the fluid; and the Prandtl number:

$$\mathrm{Pr} = \frac{\nu}{\kappa}, \tag{8}$$

with the thermal diffusivity κ of the fluid.

The physical properties are to be calculated at the mean bulk temperature T_b:

$$T_b = \frac{T_{b,in} - T_{b,out}}{2} \tag{9}$$

CORRELATIONS IN THE LITERATURE

While equations on heat transfer for laminar flow in annular ducts could be derived theoretically [4], up to now, correlations for turbulent flow could only be developed by correlating experimental or numerical data. According to the different experimental facilities and measuring procedures, the experimental data are scattered and may show different slopes. The result is that one can find about as many correlations as experimental works. Dirker and Meyer [5] recently gave a compilation of eleven correlations and presented for their data a twelfth. (Note: against the claim of Dirker and Meyer [5], the correlation of Petukhov and Roizen [6] is based on the hydraulic diameter given here in Eq. (4). This is clearly set down [7] by Petukhov and Roizen. The rewriting of their correlation in [5] is not correct). Furthermore Dirker and Meyer [5] did not mention the correlations given in [1–3].

The idea followed up in [1–3] was to base a correlation for heat transfer in annular ducts on the correlation for heat transfer in turbulent pipe flow developed in [8]. This equation was extended by a factor F taking into account the dependence of the heat transfer coefficient on the ratio $a = d_i/d_o$ for the different boundary conditions.

Comparing a great number of experimental data on heat transfer in tubes with the correlations given in the literature, Gnielinski [8] found that a semi-empirical type of equation similar to that proposed by Prandtl [9] correlated the data best. The equation of Prandtl for fully developed turbulent flow is of the form

$$Nu = \frac{(f/8)\mathrm{Re}\,\mathrm{Pr}}{1 + 8.7\sqrt{f/8}\,(\mathrm{Pr}^n - 1)} \tag{10}$$

where f is the friction factor for turbulent flow. For smooth tubes, f is a function of the Reynolds number only, as in Eq. (14).

A number of modifications of Eq. (10) are to be found in the literature and may be summarized by the equation

$$Nu = \frac{(f/8)\mathrm{Re}\,\mathrm{Pr}}{k_1 + k_2\sqrt{f/8}\,(\mathrm{Pr}^n - 1)} \tag{11}$$

Petukhov and Kirillov [10] suggested that

$$k_1 = 1.07 + \frac{900}{\mathrm{Re}} - \frac{0.63}{(1 + 10\,\mathrm{Pr})}, \tag{12}$$

$k_2 = 12.7$, and

$n = 2/3$.

The available data were correlated best in the region of fully developed turbulent flow by Eq. (11) and k_1 reduced to $k_1 = 1$ [8]. Because Eq. (11) is based on a model for fully developed turbulent flow, it does not account for entrance effects. To overcome this disadvantage, Gnielinski [8] modified Eq. (11) by multiplying it with the entrance correction factor derived by Hausen [11].

The resulting equation reads

$$Nu = \frac{(f/8)\mathrm{Re\,Pr}}{1 + 12.7\sqrt{f/8}\,(\mathrm{Pr}^{2/3} - 1)}\left[1 + (\frac{d}{L})^{2/3}\right] \quad (13)$$

where $d = d_h$ is the diameter of the tube and L its length.

According to Konakov [12], the friction factor for turbulent flow in smooth tubes may be calculated from

$$f = (1.8 \log_{10} \mathrm{Re} - 1.5)^{-2} \quad (14)$$

The hydraulic diameter d_h for the annulus from Eq. (4) has to be used instead of the tube diameter d.

A comparison with a large number of experimental data heat transfer in annular ducts presented in [1] demonstrated that most of the data could be represented within a small error margin:

$$Nu = \frac{(f/8)\mathrm{Re\,Pr}}{1 + 12.7\sqrt{f/8}\,(\mathrm{Pr}^{2/3} - 1)}\left[1 + (\frac{d_h}{L})^{2/3}\right] F \quad (15)$$

with the friction factor f from Eq. (14),

$$F = 0.86\, a^{-0.16} \quad (16)$$

for the boundary condition, "heat transfer at the inner wall and the outer wall insulated (Figure 1a)," and

$$F = 1 - 0.14\, a^{0.6} \quad (17)$$

for the boundary condition "heat transfer at the outer wall and the inner wall insulated (Figure 1b)."

Petukhov and Roizen [6] found the factors F according to Eqs. (16) and (17) by correlating their data from [7].

IMPROVEMENT OF THE CORRELATION

The following two facts convinced the present author to re-evaluate the calculation procedure for the heat transfer coefficient during turbulent flow in annular ducts according to Eqs. (14) and (15):

1. The friction factor f of an annular duct differs from the friction factor of a circular tube according to Eq. (14). Because of the different velocity profile in annular flow, f itself depends on the ratio $a = (d_i/d_o)$.
2. Additional experimental data, especially the recently published data of Dirker and Meyer [5], should be taken into consideration.

Comparing a large number of experimental data on the friction factor of annular duct flows, Gnielinski [12] found

$$f_{ann} = (1.8 \log_{10} \mathrm{Re}^* - 1.5)^{-2} \quad (18)$$

where

$$\mathrm{Re}^* = \mathrm{Re}\frac{(1 + a^2)\ln a + (1 - a^2)}{(1 - a)^2 \ln a} \quad (19)$$

The direction of heat flux (heating or cooling the fluid) influences the heat transfer when the physical properties are temperature-dependent. No special investigations for annular ducts about this influence are known. Therefore, the correction factors for circular tubes are adopted.

For gases, the variation of fluid properties with temperature can be taken into account by simple power laws. The correction factor as multiplier to Eq. (15) can be written in the principal form

$$K = \left(\frac{T_b}{T_w}\right)^n \quad (20)$$

where T_b is the absolute bulk temperature of the gas and T_w is the absolute wall temperature.

The exponent n is $n = 0$ in case of cooling a gas $[(T_b/T_w) > 1]$. Heating the gas $[(T_b/T_w) < 1]$ leads to different values for each gas. Gnielinski [8] correlated the data collected for gases by taking $n = 0.45$ in the range $0.5 < (T_b/T_w) < 1.0$. According to the literature, n should be approximately equal to 0.15 for carbon dioxide and steam in the same range of temperature ratio.

For liquids, the variation of fluid properties with the temperature can be taken into account using

$$K = \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.11} \quad (21)$$

where Pr is the Prandtl number of the liquid at bulk temperature and Pr_w at wall temperature, respectively.

The experimental data collected from the literature shown in the following figures are correlated best if the original factor k_1 given by Petukhov and Kirillov [10] is used in the correlation for the Nusselt number.

Taking into consideration all the factors mentioned above, the correlation for the Nusselt number in annular ducts for turbulent flow ($Re > 10^4$) now reads

$$Nu = \frac{(f_{ann}/8)\mathrm{Re\,Pr}}{k_1 + 12.7\sqrt{f_{ann}/8}\,(\mathrm{Pr}^{2/3} - 1)}\left[1 + \left(\frac{d_h}{L}\right)^{2/3}\right] F_{ann}\, K \quad (22)$$

where, from Eqs. (12, 18), and (19), respectively,

$$k_1 = 1.07 + \frac{900}{\mathrm{Re}} - \frac{0.63}{(1 + 10\,\mathrm{Pr})}$$

$$f_{ann} = (1.8\ \log_{10} Re^* - 1.5)^{-2}$$

$$\mathrm{Re}^* = \mathrm{Re}\frac{(1 + a^2)\ln a + (1 - a^2)}{(1 - a)^2 \ln a}$$

The best correlation of the experimental data was achieved with

$$F_{ann} = 0.75\, a^{-0.17} \quad (23)$$

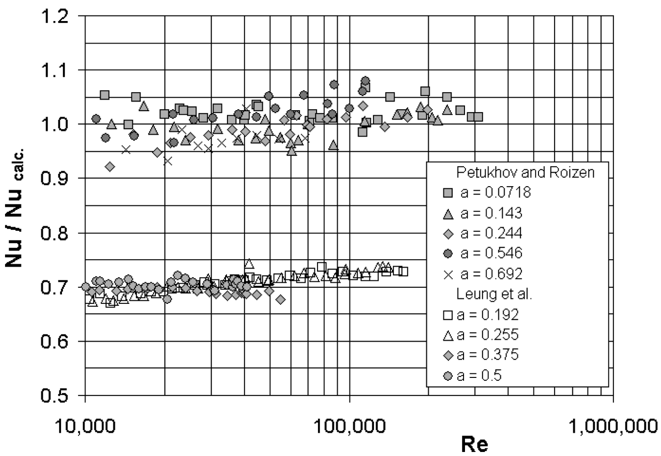

Figure 2 Ratio of the experimental Nusselt numbers to the Nusselt numbers from Eqs. (22, 23) as a function of the Reynolds number. The data of Petukhov and Roizen [7] and of Leung et al. [13] are for heat transfer to air heating from the inner tube (boundary condition (a) of Figure 1). The data of Leung et al. [13] are shown multiplied by the factor 0.7.

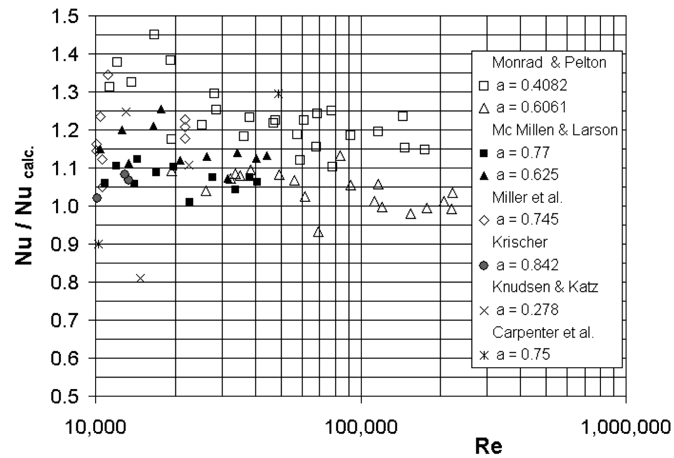

Figure 4 Ratio of the experimental Nusselt numbers to the Nusselt numbers from Eqs. (22, 23) as a function of Reynolds number. The data of Monrad and Pelton [17], McMillen and Larson [18], Miller et al. [19], Krischer [20], Knudsen and Katz [21], and Carpenter et al. [22] are for heat transfer to water heated from the inner tube (boundary condition (a) of Figure 1).

for the boundary condition "heat transfer at the inner wall and the outer wall insulated (Figure 1a)," and

$$F_{ann} = (0.9 - 0.15\ a^{0.6}) \tag{24}$$

for the boundary condition "heat transfer at the outer wall and the inner wall insulated (Figure 1b)."

No experimental data could be found for heat transfer from both walls to the annular flow (Figure 1c).

Figures 2–5 show the experimental data Nu in relation to the calculated values Nu_{calc} according to Eq. (22) for the boundary condition heat transfer at the inner wall, outer wall insulated. The data from Petukhov and Roizen [7] and of Leung et al. [13] for air in Figure 2 served as the main data to evaluate Eq. (23) with Eq. (22). The data covers a wide range of Reynolds number and annular diameter ratio (a). Equation (23) represents the data with only few exceptions within ±5%. The data of Leung et al. [13] shown in Figure 2 are multiplied by the factor 0.7 in order to be able to show them separately but on the same figure.

The experimental data of Lee and Barrow [14], Quarmby [15], and Zerban [16] for air on Figure 3 are more scattered than the rest of the data and do not give a uniform picture.

Most of the experimental data from the papers mentioned in Figure 4 are measured within the laminar region or the transition region between laminar and turbulent flow ($Re < 10^4$) and could not be used here. The data for water were measured by Monrad and Pelton [17], McMillen and Larson [18], Miller et al. [19], Krischer [20], Knudsen and Katz [21], and Carpenter et al. [22]. The recent experimental data of Dirker and Meyer [5] are shown in Figure 5. Depending on the value of the annular diameter ratio (a), the measured values are up to 15% lower than calculated from Eqs. (22, 23), while the dependency on the Reynolds number is correctly reproduced by Eq. (22). The authors themselves

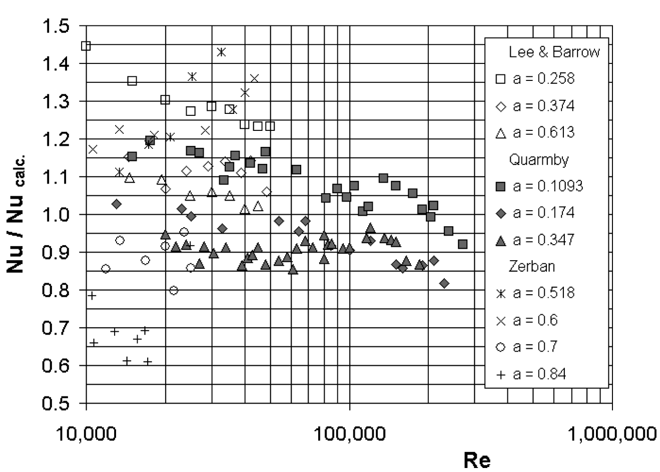

Figure 3 Ratio of the experimental Nusselt numbers to the Nusselt numbers from Eqs. (22, 23) as a function of Reynolds number. The data of Lee and Barrow [14], Quarmby [15], and Zerban [16] are for heat transfer to air heated from the inner tube (boundary condition "a" of Figure 1).

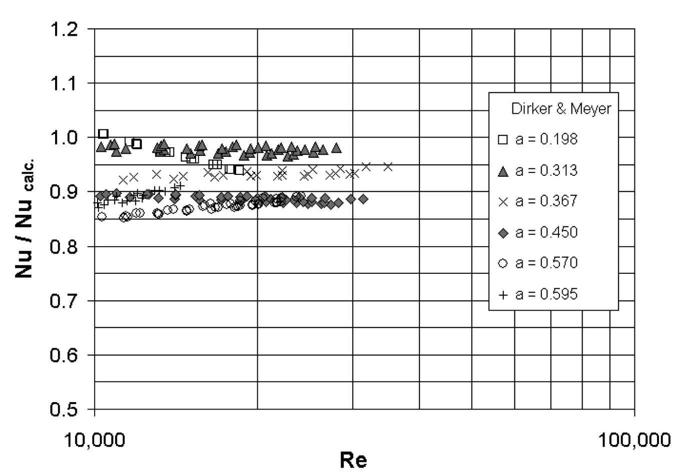

Figure 5 Ratio of the experimental Nusselt numbers to the Nusselt numbers from Eqs. (22, 23) as a function of Reynolds number. The data of Dirker and Meyer [5] are for heat transfer to water heated from the inner tube (boundary condition (a) of Figure 1).

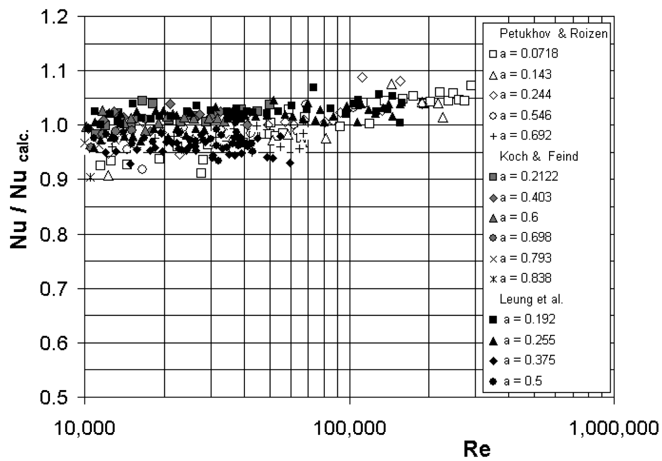

Figure 6 Ratio of the experimental Nusselt numbers to the Nusselt numbers from Eqs. (22, 24) as a function of Reynolds number. The data of Petukhov and Roizen [7], Leunget al. [13], and Koch and Feind [23] are for heat transfer to air heated from the outer tube (boundary condition (b) of Figure 1).

eliminated or ignored their experimental data measured in annular ducts with additional ratios a because they "did not agree with the general trend of the rest of the heat exchangers" (p. 42) [5]. Therefore, doubts may be registered on the precision of the experimental facility and the processing of the data. The difference between the experimental and the calculated data shall not be overrated.

Heating air from the outer wall, inner wall insulated (boundary condition Figure 1b), resulted in the experimental data of Petukhov and Roizen [7], Leung et al. [13], and Koch and Feind [23], which are shown in Figure 6 in relation to the data calculated from Eq. (22) with Eq. (24). There is a good correspondence between the data and the correlation. Few data have been published for heating water from the outer wall. The experimental values for this boundary condition calculated from Eqs. (22, 24) from Krischer [20] are shown in Figure 7. The correspondence is good for this case.

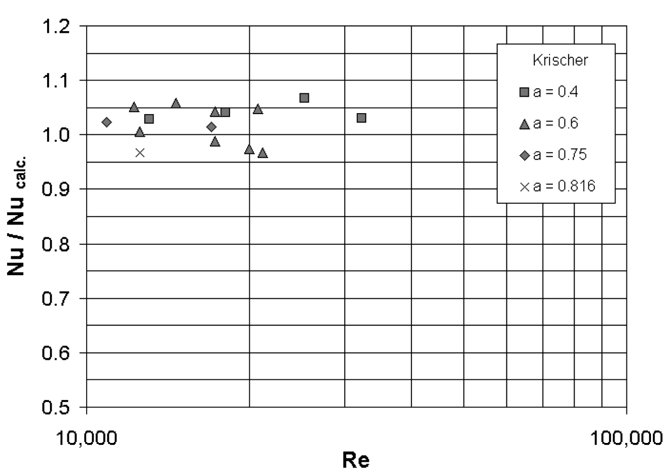

Figure 7 Ratio of the experimental Nusselt numbers to the Nusselt numbers from Eqs. (22, 24) as a function of Reynolds number. The data of Krischer [20] are for heat transfer to water heated from the outer tube (boundary condition (b) of Figure 1).

CONCLUSIONS

Heat transfer coefficients and friction factors for turbulent flow in annuli are different from those for tubes with circular cross-section because of the effect of the inner tube on the velocity profile. This is taken into account by the ratio $a = d_i/d_o$.

By evaluating a large number of experimental data from the literature, correlations were found to describe the effect of $a = d_i/d_o$. A correlation for heat transfer in circular tubes was extended by functions for the two boundary conditions (i.e., heating the annular flow from the inner tube or from the outer tube). In addition a correlation for the friction factor in annuli for turbulent flow was presented.

ACKNOWLEDGMENTS

The author would like to thank Prof. Matthias Kind, the director of the Institute for Thermal Process Engineering of the Universitaet Karlsruhe (TH), for giving him the opportunity to continue his scientific work. The author is also very grateful to Mr. Dirker for providing the complete set of data from his master's thesis.

NOMENCLATURE

a	annular diameter ratio, d_i/d_o
d	diameter of a tube, m
d_h	hydraulic diameter of annulus, $d_h = d_o - d_i$, m
d_i	outer diameter of the inner tube, m
d_o	inner diameter of the outer tube, m
f	friction factor, defined by Eq. (3); friction factor of a tube, Eq. (14)
F	factor to take into account the dependence on d_i/d_o, Eqs. (16, 17)
f_{ann}	friction factor of an annulus
F_{ann}	factor to take into account the dependence on d_i/d_o, Eqs. (23, 24)
K	factor to take into account the temperature dependence of fluid properties
k_1	factor in Eqs. (11, 12)
k_2	factor in Eq. (11)
L	length of a tube or annulus, m
n	exponent
Nu	Nusselt number
Δp	pressure loss, Pa
Pr	Prandtl number
$\dot{q}$	heat flux density, W m^{-2}
Re	Reynolds number
Re*	modified Reynolds number, Eq. (19)
T	temperature, K
ΔT_{LM}	logarithmic temperature difference, K
u	superficial velocity, m/s

Greek Symbols

α	heat transfer coefficient, W m^{-2} K^{-1}
κ	thermal diffusivity, m^2 s^{-1}
λ	thermal conductivity, W m^{-1} K^{-1}
ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}

Subscripts

b	bulk
in	inlet
out	outlet
w	wall

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