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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN SHELL-AND-TUBE HEAT EXCHANGERS WITHOUT BAFFLES

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Abstract – The influences of geometrical parameters on the shell side heat transfer in shell-and-tube heat exchangers are investigated by experiments using 32 different test heat exchangers. The test heat exchangers differ by number of tubes, length, shell and tube diameter, nozzle diameter and tube pitch. From the experimental results it can be confirmed that the influence of the tube pitch is small enough to be neglected in shell-and-tube heat exchangers used in real processes. The heat transfer rate of the longitudinal flow can be calculated from the correlation for turbulent flow in concentric annular ducts by inserting the porosity instead of the ratio of tube to shell diameter. The influence of the cross flow in the nozzle region increases with decreasing length of the heat exchangers. The heat transfer coefficients in the nozzle region are determined by comparing the overall heat transfer coefficients of the heat exchangers with that calculated from the correlation for the longitudinal flow. The results show that the heat transfer coefficient in the nozzle region is 40 % greater than that in the parallel region, if the length of the apparatuses is about 30 times the hydraulic diameter. A new correlation suitable for predicting the heat transfer coefficient is presented, which consists of a superposition of the Nusselt number for the flow in the nozzle region and that for the longitudinal flow.

Key words: Turbulence Flow, Shell-side, Heat Transfer Coefficients, Empirical Correlation, Shell-and-tube Heat Exchangers

INTRODUCTION

Shell-and-tube heat exchangers are found in many different applications in chemical industry, because they are easy to design and applicable for a wide range of temperature and pressure differences. Shell-and-tube heat exchangers without baffles are commonly used in the case of heating or cooling fluids that tend to polymerize or build deposits on any kind of wall, e.g. plate baffles.

For construction of those apparatuses it is necessary to be able to predict the pressure drop and shell-side heat transfer coefficients. Short [1943] investigated pressure drop and heat transfer in shell-and-tube heat exchangers with and without baffles. Donohue [1949] presented the following empirical correlation for predicting heat transfer coefficients in shell-and-tube heat exchangers without baffles using the experimental data obtained by Short [1943].

$$\left(\frac{\alpha \cdot d_T}{\lambda}\right) = 1.16 \cdot \left(\frac{d_h}{[\mathrm{m}]}\right)^{0.6} \cdot \mathrm{Pr}^{1/3} \cdot \left(\frac{\bar{u} \cdot d_T}{\nu}\right)^{0.6} \tag{1}$$

where d_h is the hydraulic diameter. The tube diameter and the hydraulic diameter as the geometric parameters are not sufficient to describe shell-and-tube heat exchangers with different number of tubes in divers arrangements. Moreover, the test heat exchangers of Short [1943] are relatively short (l/d_h=30.3 – 111.6) and the experiments were carried out in the range of the Reynolds number from 1000 to 30000. Therefore, the Eq. (1) is not appropriate for predicting heat transfer coefficients for fully developed flow in shell-and-tube heat exchangers.

Other investigators [Dingee et al., 1955; Miller et al., 1956; Wantland, 1956; Weisman, 1958; Presser, 1967; Rieger, 1969; Markocy, 1971; El-Genk et al., 1990] had attempted to introduce geometrical parameters in the following dimensionless relation which predicts heat transfer coefficients for fully developed turbulent flow in circular tubes.

$$\mathrm{Nu} = C \cdot \mathrm{Re}^m \cdot \mathrm{Pr}^n, \quad m \approx 0.8 - 0.91, \quad n \approx 0.33 - 0.44, \tag{2}$$

where C is a function of the ratio of pitch-to-tube diameter or the hydraulic diameter. The thermal or hydraulic equivalent diameter is used as the characteristic length for the Nusselt and Reynolds numbers. Eq. (2), however, is appropriate to calculate heat transfer coefficients for the flow in bundles with a great number of tubes like in nuclear reactor.

The elementary form of shell-and-tube heat exchangers is the double pipe made up of two concentric circular tubes. Three geometrical variables, i.e. length of the tubes (l), tube and shell diameter (d_T, d_S) are required to characterize the geometry. Commonly, the ratio of tube to shell diameter (d_T, d_S) and the ratio of length to hydraulic diameter (l/d_h) are used as non-dimensional geometric parameters in many heat transfer correlations for turbulent flow in concentric annular ducts [Petukhov et al., 1974; Gnielinski, 1991].

$$\mathrm{Nu} = \mathrm{Nu}\left(N_T, \frac{d_T}{d_S}, \frac{l}{d_d}\right) \qquad \text{where } N_T = l \tag{3}$$

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To characterize the geometry of shell-and-tube heat exchangers requires tube pitch (P), tube and shell diameter (d_T, d_S), length of the tubes (l) and tube arrangement. The peripheral diameter (d_p) can serve as the tube pitch (see Fig. 1). Therefore, the relationship between the Nusselt number and the non-dimensional geometric parameters can be given by the following form:

$$\mathrm{Nu} = \mathrm{Nu}\left(N_T, \frac{d_T}{d_S}, \frac{d_P}{d_S}, \frac{l}{d_h}\right) \qquad \text{where } N_T \geq 2 \tag{4}$$

In addition, the influence of the cross flow in the nozzle region has a great impact on the shell side heat transfer [Aicher and Kim, 1996], when the heat exchangers are short. In this work shell side heat transfer is investigated by means of experiments with 32 different test heat exchangers of symmetric and regular tube arrangements. The influence of the number of tubes, length of tubes, tube pitch, hydraulic diameter and the nozzles on the heat transfer is analysed to determine the geometric parameters dominating heat transfer in shell-and-tube heat exchangers.

APPARATUS AND EXPERIMENTAL SET UP

Experimental set up and procedure are explained in [Aicher and Kim, 1996]. The experiments were carried out with 32 different test heat exchangers which differ by

- number of tubes (N_T=1~91)
- shell diameter (d_S=34~82.5 mm)
- tube pitch (P=14~16 mm)
- tube diameter (d_T=8~40 mm)
- length of the heat exchangers (l=0.92~2.0 m; l/d_h=28.5~267.7)

Fig. 1 shows the cross section of the test heat exchangers with a small number of tubes. The geometric data of the all test heat exchangers are described in Table A in the Appendix. For easier notation they are characterized by a nine digit code

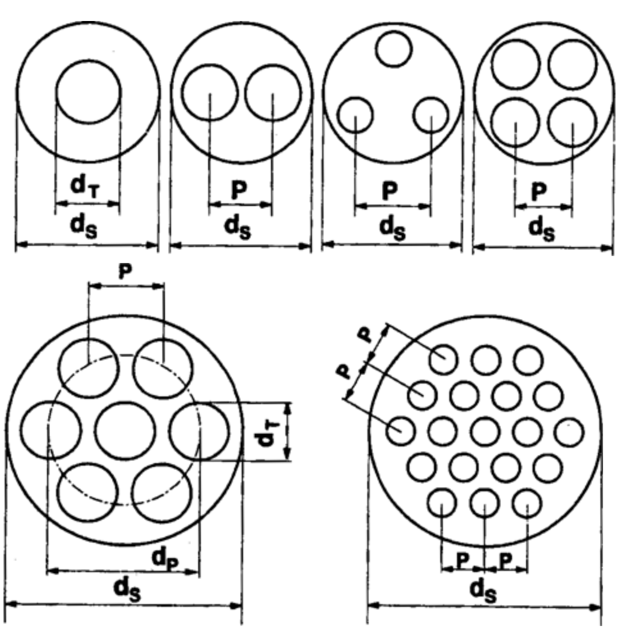

Fig. 1. Cross section of test heat exchangers with a small number of tubes.

(see first column in Table A). The first code indicates the length of the heat exchangers analysed. The next two digits give the number of tubes, the next three denote the shell diameter in mm, the following two digits denote the tube pitch, and the last two stand for the tube diameter. The cross section of the shell-and-tube heat exchangers D911501310 and D201502612 respectively is shown in Fig. 2. The geometry of these two test heat exchangers is similar to the geometry of the heat exchangers investigated by Short [1943] (see Table 1).

Inlet and outlet nozzles are mounted on the shell wall. The nozzle diameter is d_N=16 mm for the heat exchanger with a small shell ($d_S \leq 40$ mm) and d_N=35.9 mm for the other heat exchangers (d_S>40 mm). The nozzle diameter of the heat exchangers D201502612 and D911501310 is d_N=52 mm.

While the tube-side Reynolds number and fluid temperature are maintained at constant values (circa Re_T=25,000~80,000, θ_T~25 °C), the shell-side conditions are selected so that the measurements can be performed in a wide range of Reynolds numbers (500<Re<50,000) and Prandtl numbers (2<Pr<6). The nozzle effect can be determined by adjusting the same conditions during the measurements with two heat exchangers of same cross section but of different length.

RESULTS

1. Influence of the Cross-flow in the Nozzle Region

Aicher and Kim [1996] have investigated the influence of cross-flow on heat transfer in the nozzle region of double pipe heat exchangers. In this case, the important geometric parame-

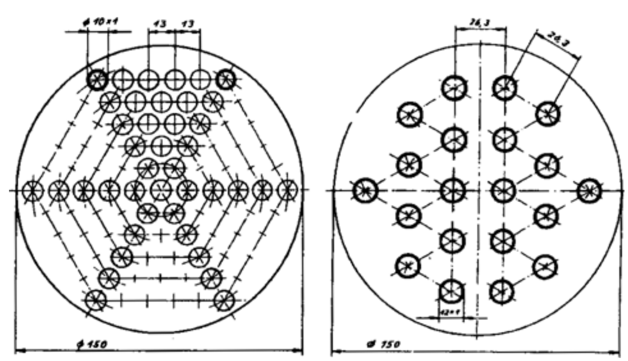

Fig. 2. Cross section of shell-and-tube heat exchangers D911501310 and D201502612.

Table 1. Geometric data of the heat exchangers investigated by Short [1943], Gentry et al. [1982] and Wantland [1956] (* The compact heat exchanger W100 has 100 tubes which are contained in square channel)

App. no.	l [mm]	N_T	d_S [mm]	P [mm]	d_T [mm]	d_h [mm]	P/d_T	l/d_h
S020	1524	20	153.9	27.78	12.70	50.25	2.19	30.3
S030	1524	30	153.9	22.23	15.88	25.66	1.40	59.4
S040	1524	40	153.9	19.84	12.70	26.10	1.56	58.4
S052	1524	52	153.9	17.46	12.70	29.28	1.83	52.0
S066	1524	66	153.9	15.08	12.70	13.19	1.19	115.5
S098	1524	98	153.9	12.70	9.53	13.65	1.33	111.6
G134	3023	134	260.0	17.46	12.70	23.53	1.38	128.5
W100*	1840	100	55.3	5.325	4.82	2.88	1.11	638.9

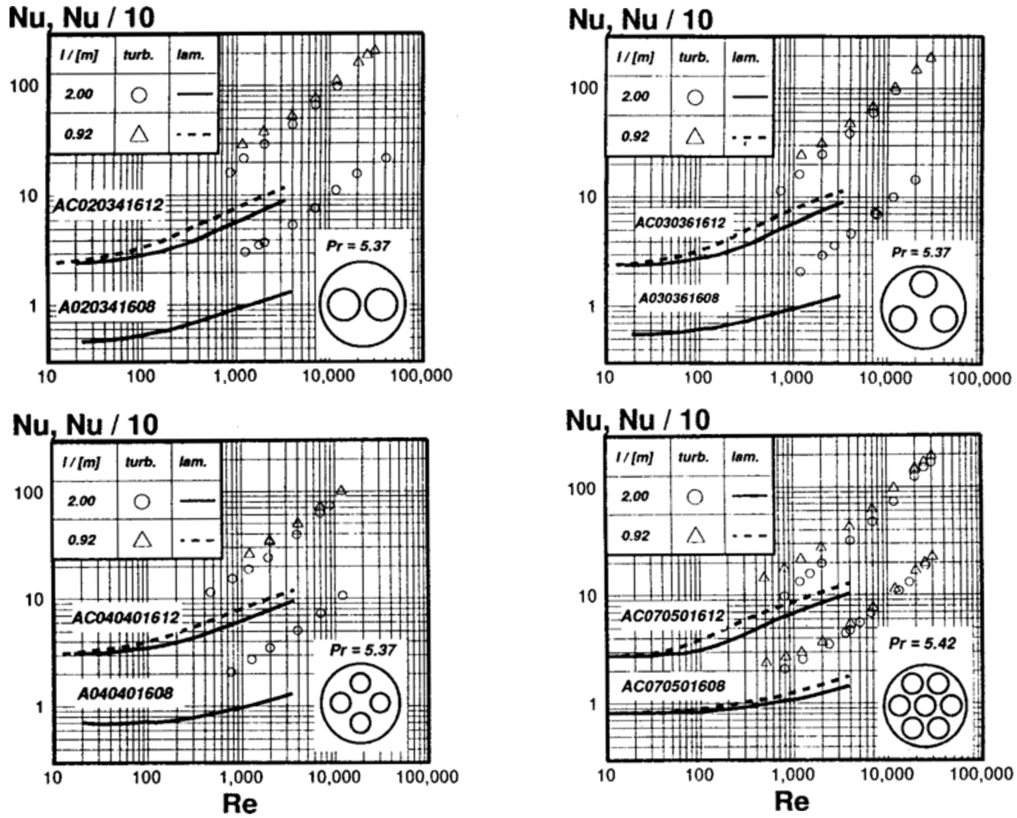

Fig. 3. Comparison of experimental results (symbols) with calculated Nusselt number for hydrodynamically developed laminar flow (lines) in a cylindrical shell containing a few of tubes (N_T=2, 3, 4 and 7).

ters are the non-dimensional length (l/d_h) and the ratio of the free cross section area of the nozzle to that of the shell. However, the heat transfer in the nozzle region in shell-and-tube heat exchangers with more than one inside tube depends on various geometric parameters, for example, tube pitch and arrangement.

Fig. 3 shows the experimental results (symbols) obtained using the different test heat exchangers with a small number of tubes (N_T=2, 3, 4 and 7). The tube pitch of the heat exchangers is at a constant value (16 mm), while the tube diameter d_T ranges from 8 to 12 mm, the length of the apparatuses is l=0.92 and 2 m. The Nusselt numbers for the heat exchangers with the small tube diameter (d_T=8 mm) are plotted as Nu/10 versus Re, and for the heat exchangers with a tube diameter of d_T=12 mm as Nu versus Re. The characteristic length in the Nusselt and Reynolds number is the hydraulic diameter. The shell side fluid temperature for these measurements was maintained at about 30 °C.

In Fig. 3, experimental results are compared with calculated Nusselt number (lines) for hydrodynamically developed laminar flow in a cylindrical shell containing a bundle of tubes presented in [Kim et al., 1993; Kim, 1994]. As can be seen in Fig. 3, the measured Nusselt numbers in the range of small Reynolds numbers ($500 \leq Re \leq 2300$) are higher than those calculated for hydrodynamically developed laminar flow. This effect is due to the influence of the cross-flow in the nozzle region. The flow in the nozzle region is not developed in axial direction, and the strong intensity of the turbulence promotes heat transfer.

It also can be recognized from Fig. 3 that the influence of the cross-flow in the heat exchangers with a small tube diameter (d_T=8 mm) is greater than in those with tube diameter d_T =12 mm. This is caused by the fact that the hydraulic diameter of the heat exchangers with small tubes is greater than that with larger tubes. Therefore, the non-dimensional length (l/d_h) of the heat exchangers with small tubes is smaller than that with larger tubes. In other words, under the same condition the influence of the cross-flow increases with decreasing non-dimensional length (l/d_h) of the heat exchangers.

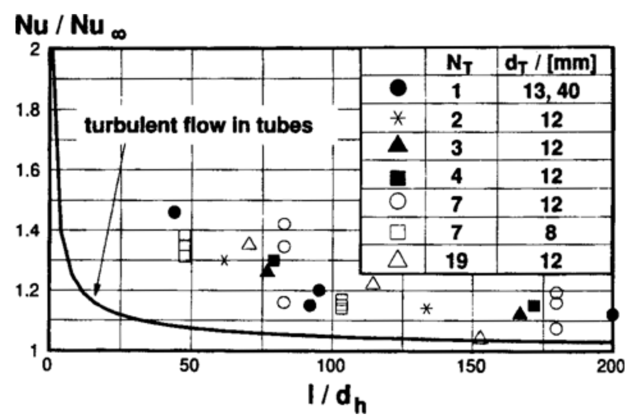

Fig. 4. Dependence of heat transfer in shell-and-tube heat exchangers on the length.

The difference of measured heat transfer coefficients between short and long heat exchangers increases with decreasing Reynolds number. It can be deduced from this fact that the influence of the turbulence generated by the cross flow on heat transfer in the nozzle region increases with decreasing Reynolds number. The heat transfer in the nozzle region depends on various geometric parameters, i.e. nozzle diameter, ratio of pitch to tube diameter, and tube arrangements.

Aicher and Kim [1996] presented the following equation for calculating the Nusselt number for fully developed flow Nu_∞ from their experimental data obtained using two heat exchangers with the same geometrical size except the length. The Nusselt number Nu_∞ depends only on the geometries of the cross section and the length of apparatuses.

$$Nu_\infty = \frac{(Nu \cdot l)_1 - (Nu \cdot l)_2}{(l)_1 - (l)_2} \tag{5}$$

where the indices 1 and 2 denote the long and short heat exchangers, respectively. The Nusselt number Nu_∞ was determined by experimental results obtained in their work. In Fig. 4 the ratios of the Nusselt numbers Nu/Nu_∞ are plotted as symbols versus the non-dimensional length l/d_h. The dependence of the heat transfer in the thermal entrance region on the length, for tubes or concentric annular ducts can be written as

$$\frac{Nu}{Nu_\infty} = 1 + \left(\frac{d_h}{l}\right)^{2/3} \tag{6}$$

In Fig. 4, the ratio of Nusselt numbers calculated from Eq. (6) is plotted as a solid line. Fig. 4 shows that the ratios of the Nusselt numbers measured in this work are higher than that calculated from Eq. (6). The deviation augments with decreasing non-dimensional length. This means that the shorter the heat exchangers are, the greater is the influence of the cross flow in the nozzle region.

2. Influence of the Tube Pitch and Tube Diameter

For the laminar flow in the parallel region the tube pitch (P) or the peripheral diameter (d_P) have a dominant impact on heat transfer [Kim et al., 1993; Kim, 1994]. The left part of Fig. 5 shows the experimental results obtained using six test heat exchangers with various tube pitches of P=14, 16 and 18 mm. In this diagram, the experimental results are compared to the values calculated from the equation of Dittus and Boelter [1956].

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \tag{7}$$

The left part of Fig. 5 also shows that tube pitch has a great impact on the Nusselt number for small Reynolds numbers. In this case the cross-flow in the nozzle region has a dominant impact on heat transfer. However, for large Reynolds numbers the Nusselt numbers depend only on the tube diameter. The Nusselt numbers of the heat exchangers with a tube diameter of 12 mm agree well with the Nusselt numbers calculated from Eq. (7), while those for the heat exchangers with the small tube diameter (d_T=8 mm) exceed the calculated values. In this work, the influence of the geometric parameters of the cross section on the heat transfer is investigated only in the range of large Reynolds numbers. In this case ($Re \geq 10{,}000$), the slope of the measured Nusselt number of all heat exchangers is 0.8 in double logarithmic diagram.

In the right part of Fig. 5, the experimental results for the long heat exchangers with tube diameters of d_T=8 and 12 mm are plotted in the form of Nu/Nu_{Ditt} versus the non-dimensional peripheral diameter ($D_P=d_P/d_S$). Additionally, the experimental results of Presser [1967] are shown. The ratio of tube-to-shell diameter of the heat exchangers investigated by Presser [1967] is $D_T=d_T/d_S=0.24$ and the non-dimensional peripheral diameter is D_P=0.504 and 0.620. For the case D_T=0.24 and D_P=0.48 the tubes touch each other. The right part of Fig. 5 also elucidates that the tube pitch only has impact on the heat transfer if the non-dimensional peripheral diameter is very small.

However, heat transfer rates in shell-and-tube heat exchangers are almost constant in a wide range of non-dimensional peripheral diameters. Therefore, its influence can be neglected for predicting heat transfer in practical shell-and-tube heat exchangers. Therefore, the relationship between the Nusselt number and the non-dimensional geometric parameters of the shell-and-tube heat exchangers, the Eq. (4), can be replaced by that for concentric annular ducts which is described by Eq. (3).

$$Nu = Nu\left(N_T, \frac{d_T}{d_S}, \frac{l}{d_h}\right) \quad \text{where } N_T \geq 2 \tag{8}$$

3. Comparison of the Experimental Results with the Correlations of Gnielinski

The left side in Fig. 5 shows that the measured Nusselt num-

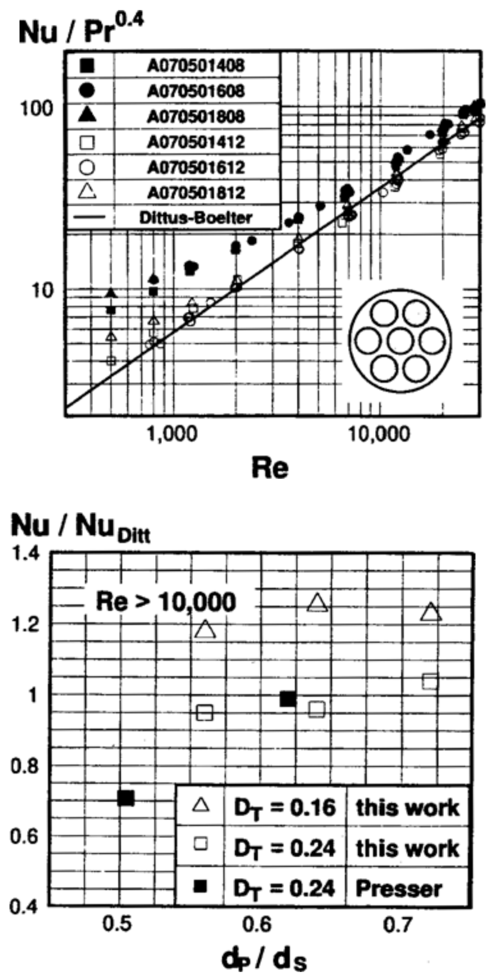

Fig. 5. Influence of tube pitch and tube diameter on the heat transfer in the shell-and-tube heat exchangers with seven tubes.

bers are parallel to the line calculated from Eq. (7) which describes the heat transfer of the fully developed turbulent flow in tubes. An equation that is valid for the thermal entrance region as well as for the fully developed turbulent flow has been presented by Gnielinski [1991]. This equation includes the term for the dependence on the length which has been noted in Eq. (6).

$$Nu_T = \frac{\frac{\xi}{8} \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot \sqrt{\frac{\xi}{8}} \cdot (Pr^{2/3} - 1)} \cdot \left[1 + \left(\frac{d_h}{l} \right)^{2/3} \right] \tag{9}$$

where ξ is the friction factor given by

$$\xi = [1.82 \cdot \log(Re) - 1.64]^{2} \tag{10}$$

In Fig. 6, the experimental results obtained from the heat exchangers with l=2 and 0.92 m respectively are plotted in the form of Nu/Nu_T versus the porosity as symbols. The Nusselt number Nu_T was calculated by using the hydraulic diameter instead of the tube diameter in Eq. (9) and (10). The porosity of shell-and-tube heat exchangers depends upon the number of tubes and the ratio of tube-to-shell diameter and can be calculated from the following equation.

$$\Phi = 1 - N_T \cdot \left(\frac{d_T}{d_S} \right)^2 \tag{11}$$

Gnielinski [1991] also presented a correlation to predict Nusselt numbers for turbulent flow in concentric annular ducts which depend on the ratio of tube-to-shell diameter. The correlation can be rewritten by using the porosity.

$$\frac{Nu_{ann}}{Nu_T} = 0.86 \cdot \left(\sqrt{N_T} \cdot \frac{d_T}{d_S} \right)^{-0.16} = 0.86 \cdot (\sqrt{1-\Phi})^{-0.16}$$

$$\text{where } N_T = 1 \tag{12}$$

In Fig. 6, the values calculated from Eq. (12) are plotted as a solid line. The Nusselt numbers of the test heat exchangers and the double pipe heat exchangers are greater than the values calculated from Eq. (12). The deviation for the short heat exchangers is greater than that for the long heat exchangers. The deviation becomes greater with increasing porosity. This is due to the fact that the ratio of length to hydraulic diameter augments with decreasing porosity, because the length is constant. Therefore, heat transfer in shell-and-tube heat exchangers can be described by Eq. (12), if the deviations are caused by the ef-

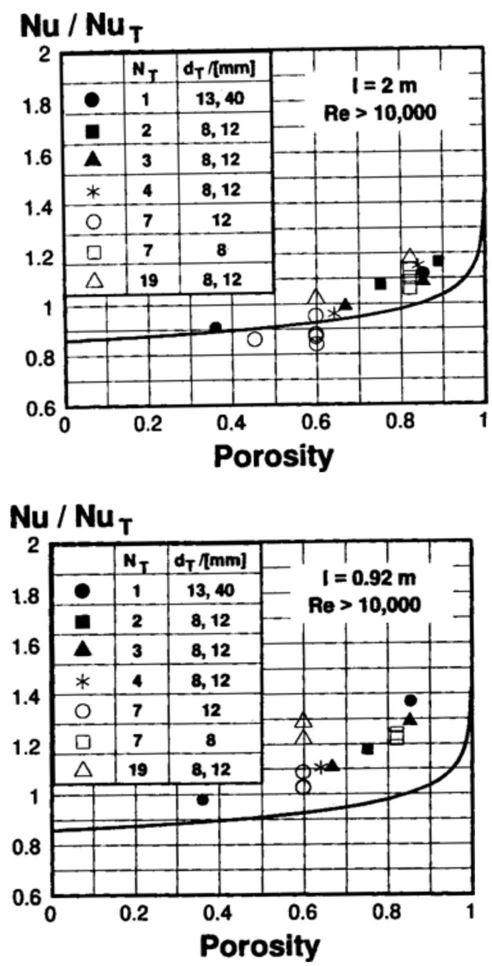

Fig. 6. Experimental results of heat exchangers with l=2 and 0.92 m, plotted in the form of Nu/Nu_T versus the porosity.

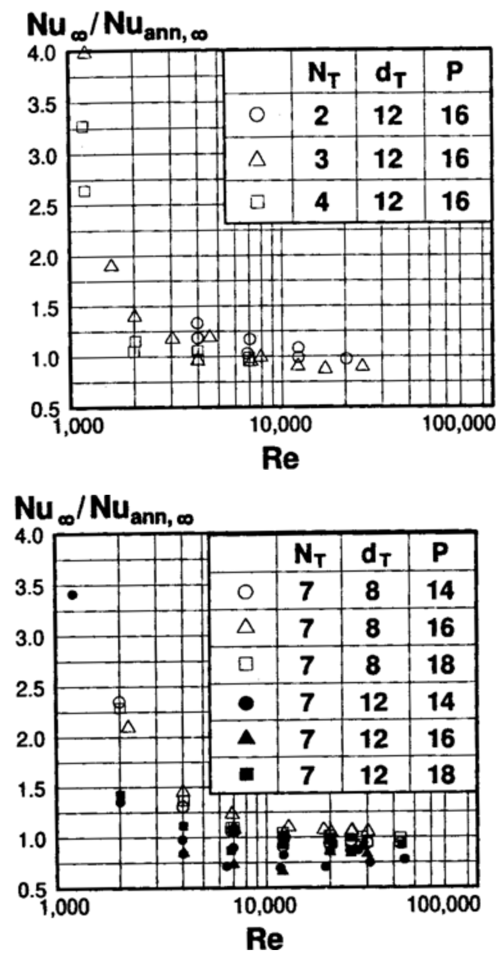

Fig. 7. Comparison of Nusselt numbers for fully developed turbulent flow in test heat exchangers Nu_∞ with those in concentric annular ducts $Nu_{ann,\infty}$.

fect of the cross flow in the nozzle region.

In Fig. 7, measured Nusselt numbers for fully developed turbulent flow in the test heat exchangers with a small number of tubes (N_T=2, 3, 4 and 7) Nu_∞ which can be calculated from the Eq. (5) are compared with those in concentric annular ducts $Nu_{ann,\infty}$. The Nusselt numbers Nuann, ∞ can be calculated from the following equation.

$$Nu_{ann,\infty}=0.86\cdot\left(\frac{d_T}{d_S}\right)^{-0.16}\cdot\frac{\frac{\xi}{8}\cdot(Re-1000)\cdot Pr}{1+12.7\cdot\sqrt{\frac{\xi}{8}}\cdot(Pr^{2/3}-1)} \quad (13)$$

As shown in Fig. 7, the values of the ratios of the Nusselt numbers $Nu_\infty/Nu_{ann,\infty}$ are about 1 for large Reynolds numbers. This means that the deviations in Fig. 6 are due to the effect of the cross flow in the nozzle region, and therefore, the heat transfer for longitudinal flow in shell-and-tube heat exchangers can be calculated from the Eq. (12). The heat transfer in the range of small Reynolds numbers can not be analysed because of the lack of the experimental data. In this region various geometrical and hydrodynamical parameters have an impact on the heat transfer.

4. Correlation

Aicher and Kim [1996] have presented the following correlation for predicting the heat transfer in double pipe heat exchangers, which includes the effect of cross flow in the nozzle region.

$$Nu=12\cdot Nu_T\cdot\frac{d_h}{l}+Nu_p \quad (14)$$

where Nu_p is the Nusselt number for longitudinal flow in double pipe heat exchangers. For shell-and-tube heat exchangers the Nusselt number Nu_p can be calculated from the Eq. (12) and can be rewritten by

$$Nu_p=Nu_T\cdot 0.86\cdot\left(\sqrt{N_T}\cdot\frac{d_T}{d_S}\right)^{-0.16}=Nu_T\cdot 0.86\cdot(\sqrt{1-\Phi})^{-0.16} \quad (15)$$

In this work, the experimental results are compared with the values calculated from Eq. (14) and (15). In Fig. 8, the ratios of the measured Nusselt numbers to those calculated from the Eq. (14) Nu_{mes}/Nu_{cal} are plotted versus the non-dimensional length (l/d_h).

Fig 8. shows that the measured Nusselt numbers for all test heat exchangers, except the heat exchangers B and C190831612, agree very well with those calculated from the correlation. For the heat exchangers B and C190831612 the ratio of the Nusselt numbers is only about 1.18. Therefore, the correlation (14) can be also used for predicting shell side heat transfer for turbulent flow (Re>10,000) in shell-and-tube heat exchangers.

Fig. 9 shows that the experimental results of other investigators [Short, 1943; Gentry et al., 1982; Wantland, 1956] are compared with the values calculated from Eq. (14). The geometric data of these heat exchangers are listed in Table 1.

In the left diagram in Fig. 9, the Nusselt numbers obtained by Short [1943] are plotted versus Reynolds number as symbols. Additionally, the Nusselt numbers of the heat exchangers D201502612 and D911501310 are added in the left diagram of Fig. 9. The Nusselt numbers calculated from the correlation (14) are added as solid lines. It can be seen that the experimental results obtained by Short [1943] do not agree well with the proposed correlation. This can be explained by the fact that the ratio of the pitch-to-diameter of heat exchanger S066 is very small (P/d_T=1.19), while the distance between the tube bundle and the shell is very large. Therefore, the heat transfer rate in this heat exchanger becomes smaller than that predicted by cor-

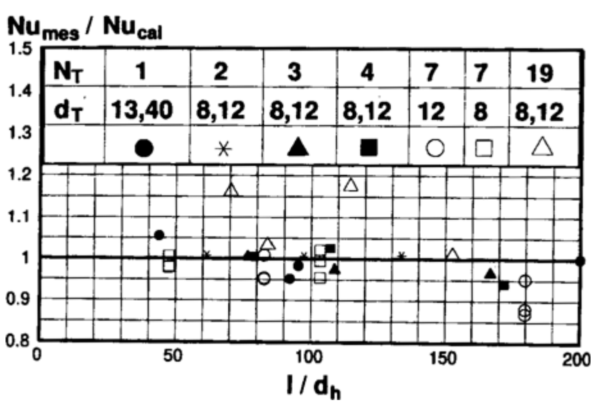

Fig. 8. Comparison of measured Nusselt numbers with those calculated from the Eq. (14) and (15) (Nu_{mes}/Nu_{cal} are plotted versus l/d_h).

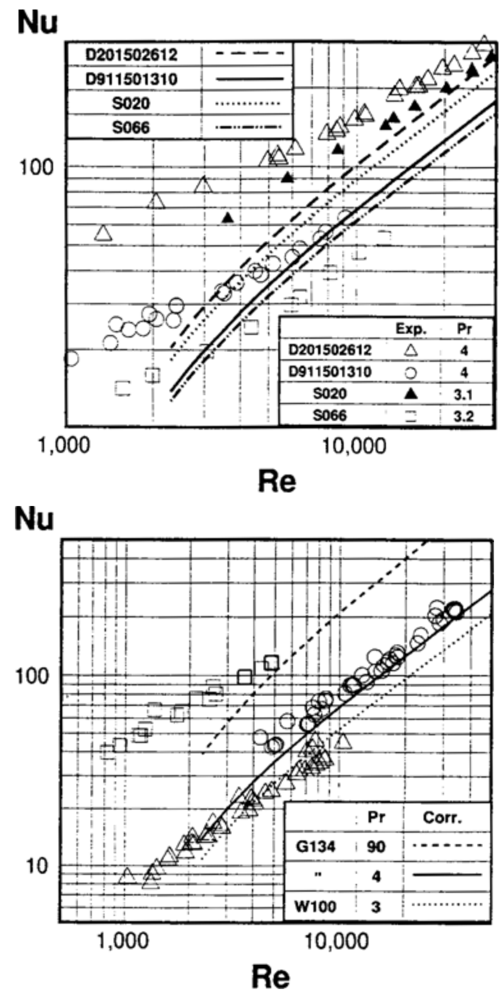

Fig. 9. Comparison of the experimental results (symbols) obtained by Short, Gentry et al. and Wantland with the Nusselt numbers calculated from the correlation (lines).

relation (14).

However, heat transfer rates in heat exchanger S020 and D 201502612 are greater than that expected from the correlation (14). The non-dimensional length of these heat exchangers is very small (l/d_h=30.3 and 28.5). In addition, the ratio of the cross section area of the nozzle to that of the shell side is 0.138. This means that the fluid velocity in the nozzle is about 7 times greater than that in shell side. Because of this, the effect of the cross flow in the nozzle region is greater than in other heat exchangers. This work does not include the experimental results of the other test heat exchangers investigated by Short [1943], because calculated and measured values agree quite well.

Gentry et al. [1982] have investigated the heat transfer in shell-and-tube heat exchangers with rod baffles. The compact heat exchanger investigated by Wantland [1956] has 100 tubes in square array which are contained in square channel. The right diagram in Fig. 9 shows that calculated and measured values agree quite well. However, the baffles and the tube arrangement have an impact on the heat transfer.

CONCLUSIONS

The influences of geometrical parameters on the shell side heat transfer in shell-and-tube heat exchangers are investigated by experiments using 32 different test heat exchangers which differ by the number of tubes, length, shell-and-tube diameter, nozzle diameter and tube pitch. The experimental results reveal that:

(1) The influence of the cross-flow increases with decreasing non-dimensional length (l/d_h) of the heat exchangers and with decreasing Reynolds number.

(2) For small Reynolds number the heat transfer of the heat exchangers depends on various geometric parameters, i.e. nozzle diameter, tube diameter, pitch, length of the heat exchangers, and tube arrangements. The influence of the geometric parameters can not be analysed because of the lack of the experimental data.

(3) The influence of the tube pitch can be neglected for predicting the heat transfer for turbulent flow ($Re \geq 10^4$) in practical shell-and-tube heat exchangers.

(4) The heat transfer rate of the longitudinal flow of the practical heat exchangers can be calculated from the correlation for turbulent flow in concentric annular ducts by inserting the porosity instead of the ratio of tube to shell diameter.

(5) The experimental results are compared with the values calculated from the correlation for predicting the heat transfer in double pipe heat exchangers, which includes the effect of cross flow in the nozzle region. The comparison shows that the correlation can be also used for predicting the shell side heat transfer for turbulent flow.

(6) The cross flow in the nozzle region has a great impact on the shell side heat transfer if heat exchangers are short. The results show that heat transfer coefficient in the nozzle region is about 40 % higher than that in parallel region, if the length of the apparatus is about l=30 d_h.

NOMENCLATURE

C : coefficient in Dittus and Boelter equation [-]
d_h : hydraulic equivalent diameter [m]
d_P,D_P : peripheral diameter [m], [-]
d_S : shell diameter [m]
d_T, D_T : tube diameter [m], [-]
l : length [m]
N_T : number of tubes [-]
P : tube pitch [m]
Nu : Nusselt number [-]
Pr : Prandtl number [-]
Re : Reynolds number [-]

Greek Letters

α : heat transfer coefficient [W/m^2 K]
Φ : porosity [-]
λ : heat conductivity [W/m K]
ν : kinematic viscosity [m^2/s]
ξ : friction factor [-]

Subscripts

∞ : fully developed turbulent flow
1 : long heat exchanger
2 : short heat exchanger
ann : concentric annular duct
cal : calculated
log : logarithmic mean value
N : nozzle
mes : measured
S : shell side
t : turbulent
T : tube side

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Appendix

Table A. Geometrical data of the test heat exchangers*

App. No.	l [m]	N_T	d_S [mm]	P [mm]	d_T [mm]	d_{TI} [mm]	d_h [mm]	d_N [mm]	P/d_T	l/d_h
A010340013	2.00	1	34.0	-	13.0	11.0	21.00	16.0	-	95.2
C010340013	0.92	1	34.0	-	13.0	11.0	21.00	16.0	-	43.8
A010500040	2.00	1	50.0	-	40.0	37.0	10.00	35.9	-	200.0
C010500040	0.92	1	50.0	-	40.0	37.0	10.00	35.9	-	92.0
A020341608	2.00	2	34.0	16.0	8.0	6.0	20.56	16.0	2.00	97.3
A020341612	2.00	2	34.0	16.0	12.0	10.0	14.97	16.0	1.33	133.6
C020341612	0.92	2	34.0	16.0	12.0	10.0	14.97	16.0	1.33	61.5
A030361608	2.00	3	36.0	16.0	8.0	6.0	18.40	16.0	2.00	108.7
A030361612	2.00	3	36.0	16.0	12.0	10.0	12.00	16.0	1.33	166.7
C030361612	0.92	3	36.0	16.0	12.0	10.0	12.00	16.0	1.33	76.7
A040401608	2.00	4	40.0	16.0	8.0	6.0	18.67	16.0	2.00	107.1
A040401612	2.00	4	40.0	16.0	12.0	10.0	11.64	16.0	1.33	171.8
C040401612	0.92	4	40.0	16.0	12.0	10.0	11.64	16.0	1.33	79.0
A070501408	2.00	7	50.0	14.0	8.0	6.0	19.36	35.9	1.75	103.3
A070501608	2.00	7	50.0	16.0	8.0	6.0	19.36	35.9	2.00	103.3
A070501808	0.92	7	50.0	18.0	8.0	6.0	19.36	35.9	2.25	103.3
C070501408	0.92	7	50.0	14.0	8.0	6.0	19.36	35.9	1.75	47.5
C070501608	0.92	7	50.0	16.0	8.0	6.0	19.36	35.9	2.00	47.5
C070501808	0.92	7	50.0	18.0	8.0	6.0	19.36	35.9	2.25	47.5
A070501412	2.00	7	50.0	14.0	12.0	10.0	11.13	35.9	1.17	179.7
A070501612	2.00	7	50.0	16.0	12.0	10.0	11.13	35.9	1.33	179.7
A070501812	2.00	7	50.0	18.0	12.0	10.0	11.13	35.9	1.50	179.7
C070501412	0.92	7	50.0	14.0	12.0	10.0	11.13	35.9	1.17	82.7
C070501612	0.92	7	50.0	16.0	12.0	10.0	11.13	35.9	1.33	82.7
C070501812	0.92	7	50.0	18.0	12.0	10.0	11.13	35.9	1.50	82.7
A070501714	2.00	7	50.0	17.0	14.0	10.0	7.62	35.9	1.21	262.4
A190831608	2.00	19	82.5	16.0	8.0	6.0	23.84	35.9	2.00	83.9
A190831612	2.00	19	82.5	16.0	12.0	10.0	13.11	35.9	1.33	152.6
B190831612	1.50	19	82.5	16.0	12.0	10.0	13.11	35.9	1.33	114.4
C190831612	0.92	19	82.5	16.0	12.0	10.0	13.11	35.9	1.33	70.2
D201502612	1.435	20	150.0	26.3	12.0	10.0	50.31	52.0	2.19	28.5
D911501310	1.435	91	150.0	13.3	10.0	8.0	12.64	52.0	1.33	113.5