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Evaluation of heat transfer parameters in packed beds with cocurrent downflow of liquid and gas

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Abstract

The results of an experimental investigation on heat transfer from a packed bed with cocurrent gas–liquid downflow to the wall are presented and analyzed in this contribution. The measurements cover the range of operating variables corresponding to the so-called trickle regime in beds presenting aspect ratios (tube to particle diameter ratio) from 4.67 to 34.26. Water and air were employed as model fluids. The heat transfer process was first analyzed by means of a two-dimensional pseudohomogeneous plug-flow model with two parameters, the effective radial thermal conductivity (k_{er}) and the wall heat transfer coefficient (h_w). k_{er} is well correlated with liquid and gas Reynolds numbers and particle diameter, except for the lowest experimental aspect ratio (4.67). Instead, a meaningful correlation of h_w stands only for aspect ratios larger than 15. These results are analyzed and the evidence points out to sustain the hypothesis that the model fails at low aspect ratios because an apparent contact resistance ($1/h_w$) can no longer accommodate the effects of significant fluid bypassing and finite size of the near-wall region. The experimental set of data were also used to develop a correlation for the overall heat transfer coefficient (h_T), which can be employed satisfactorily to predict heat transfer rates in the whole range of variables here investigated.

Keywords: Heat transfer; Trickle-bed reactors; Thermal parameters; Radial temperature profile; Effective thermal conductivity; Wall heat transfer coefficient

1. Introduction

In some industrial processes such as the synthesis of methyl-isobutyl ketone (MIBK) or the Shell Middle Distillates Synthesis process, aimed to convert natural gas into synthetic hydrocarbon employing an advance Fischer–Tropsch technology (Krishna & Sie, 1994), multitubular trickle-bed catalytic reactors are employed. The geometrical configuration of these reactors allows part of the reaction heat to be transferred to an external coolant.

In spite of the importance of assessing correctly the heat transfer rates and thermal behaviour in these and other industrial processes, the amount of investigations reported in the open literature about heat transfer within packed beds with two-phase flow is scarce. Considering specifically the works devoted to studies with cocurrent downflow (Matsuura, Hitaka, Akehata, & Shirai, 1979a, b; Hashimoto, Muroyama, Nagata, & Fujiyoshi, 1976; Muroyama, Hashimoto, & Tomita, 1977; Specchia & Baldi, 1979; Lamine, Gerth, Le Gall, & Wild, 1996), the results show noticeable disagreement or only partial information is provided.

The goal of this contribution is to present own experimental results on heat transfer from packed beds with cocurrent two-phase downflow to the tube-wall and to analyse them under two perspectives: the values of effective radial thermal conductivity (k_{er}) and wall heat transfer coefficient (h_w) for the two-dimensional pseudohomogeneous plug-flow model and the values of the overall heat transfer coefficient (h_T) accounting just for the observed heat transfer rates.

In both cases, the effect of gas and liquid flow rates within the trickle (low interaction) regime and the aspect

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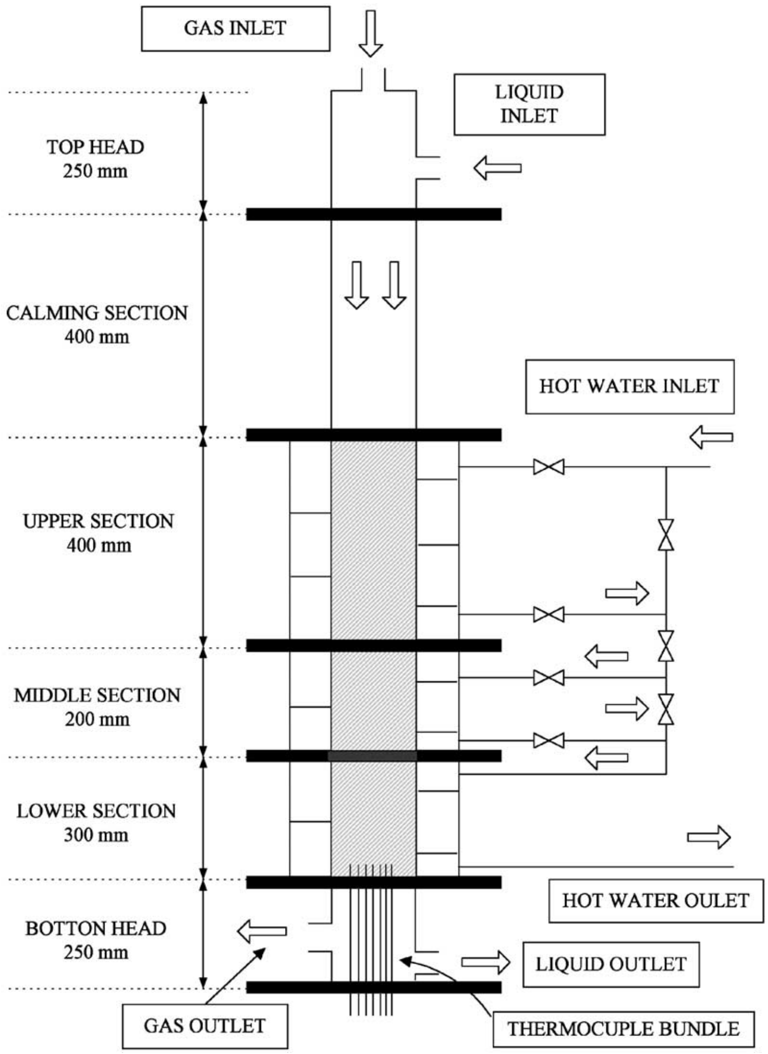

Fig. 1. Sketch of the experimental apparatus.

ratio (tube to particle diameter ratio) from 4.67 to 34.26 is analysed in relation to previous works with the final purpose of developing suitable predictive correlations. The objective is fulfilled as regards to h_T, but the meaning of the two-dimensional pseudohomogeneous plug-flow model seems to be restricted to the higher range of aspect ratios. Possible causes for this behaviour are discussed.

2. Experimental set-up and procedure

The experimental study was carried out on a 51.4 mm internal diameter tube shown in Fig. 1. Air and water flowed downwardly through beds of ballotini tightly sized in four diameters, 1.5, 3.0, 6.3 and 11 mm. Four aspect ratios thus arose, $a = 4.76, 8.16, 17.13, 34.26$. The liquid flow rate was varied between 0.3 and 1.0 l min^{-1} and that of air between 3.0 and 30.0 l min^{-1} at nearly atmospheric pressure. These ranges were restricted to operating conditions within the trickle regime, whose confirmation was obtained by flow maps reported in the literature and in many instances by direct visual observation (Mariani, 2000).

The heating fluid was water fed at about 80°C into a three-section jacket. These sections are identified as upper, middle and lower sections (Fig. 1). The hot water could pass through one, two or all of the three sections, thus allowing three lengths of active heat transfer surface, 270, 470 and 870 mm. The hot water circulates in a close loop including an electric heater. Water and air were fed in the bed without recirculation.

The top of the tube (calming section in Fig. 1) was also packed with spheres to provide uniform temperature and flow distributions at the inlet of the jacketed zone.

The following temperature measurements were carried out: at the inlet and outlet of each section of the jacket (although the water flow was high enough to maintain nearly isothermal conditions within the jacket), at the bed inlet and outlet in the liquid phase, at nine points distributed radially and angularly on the bed cross section at about 30 mm above the plate sustaining the bed, and at three axial positions within the tube-wall. All temperature readings were recorded by a data acquisition system.

For each experimental condition, defined by given packing size, liquid and air flow rates and heat transfer length, between 4 and 8 replicates were performed. For each replicate, the bed was first fluidized by water. Each run demanded about 3–4 h. After reaching steady state conditions, the recorded sets of temperatures (about 2000 in each run) were averaged to obtain the values to be used for analytical purposes.

Some other experimental details concerning the experimental set-up and operating procedure can be found in Mariani (2000).

3. Results from applying the two dimensional pseudohomogeneous model

The two-dimensional pseudohomogeneous plug-flow (2DPPF) model introduces two effective heat transfer parameters intended to characterize heat transport within fixed beds: the effective radial thermal conductivity (k_{er}) and the wall heat transfer coefficient (h_w). The 2DPPF model has been extensively used for packed beds with one-phase flow and it was always employed in those works related to packed beds with cocurrent liquid-gas downflow.

According to the 2DPPF model, the temperature distribution within the bed can be obtained as a series solution (Wakao & Kaguei, 1982). A constant temperature in the heating fluid was assumed. The inlet bed temperature profile was established from the known value at the bed axis, an unknown coefficient C_1 for the first term of the series and ratios (C_j/C_1) fixed according to those arisen for an uniform inlet profile for the remaining coefficients C_j ($j>1$). Thus, the unknown parameters were k_{er}, h_{wc} and C_1. The coefficient h_{wc} arises from adding the thermal resistance at the wall and that on the jacket side, $1/h_{wc}=1/h_w+1/h_c$. The coefficient h_c was evaluated from measured temperatures in the jacket fluid and at the wall. Values of h_w were typically ten times smaller than h_c.

The values of k_{er}, h_{wc} and C_1 were estimated by fitting the experimental sets of temperatures measured on the cross section near the bed exit (see Section 2). Values of k_{er}, h_{wc} were considered independent of the heating length, and values of C_1 were constrained to a linear function of the liquid flow rate for each particle diameter. The Greg Software Package (Stewart, Caracotsios, & Sørensen, 1992) was employed for the regression analysis.

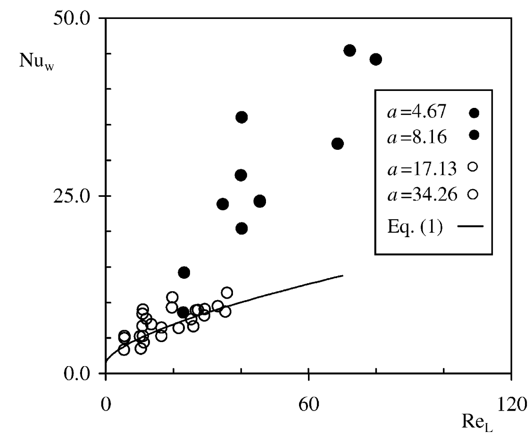

Fig. 2. Experimental and predicted values of Nu_w vs liquid Reynolds number, Re_L.

3.1. Wall heat transfer coefficient

The results for h_w were not significantly sensitive to the gas flow rate, indicating that the mechanisms responsible for the behaviour of h_w were dominated by the liquid phase. Assuming that h_w reflects the behaviour of a liquid boundary layer on the wall subjected by the presence of particles, $Nu_w=h_w d_p/k_L$ should be a function of the liquid Reynolds number, Re_L. The fitted values of Nu_w, distinguishing those for the smaller particles, $d_p=1.5, 3.0$ mm ($a=34.26, 17.13$), and for the larger particles, $d_p=6.3, 11$ mm ($a=8.16, 4.76$), are plotted in Fig. 2.

The effect of Re_L is clearly different for both data sets and the expected single functionality with Re_L can apparently be ruled out. While a classical dependency $Nu_w \propto Re_L^{\alpha}$ with $\alpha<1$ (see Eq. (1) described below) is suitable for the smaller particles, Nu_w increases with Re_L in a nearly proportional way for the larger particles.

The reason for this behaviour should be found in the plug flow assumption of the model. It is well known that the zone from the wall to around half particle diameter, hereafter named wall-zone, shows a larger void fraction than the rest of the bed (Mariani, Mazza, Martínez, & Barreto, 1998, recently reviewed the subject). A higher permeability is then expected in that zone, a feature that at least for one-phase flow has been clearly demonstrated (see e.g. Giese, Rottschäfer, & Vortmeyer, 1998). While the amount of fluid flow in the wall-zone can be justifiably ignored in the frame of the 2DPPF model at large values of a, the effect can be hardly neglected at low values: the wall-zone represents 5.7% of the total cross section area for $a=34.26$ and 38.2% for $a=4.76$. It is then reasonable to believe that the high values of Nu_w (Fig. 2) resulting from applying the 2DPPF model arise mainly as a consequence of ignoring the extra amount of fluid flowing in the wall-region, which can be actually heated more easily than the assumption of uniform flow distribution allows.

An indirect evidence of the increase flow in the wall-zone was observed in the course of the experiments. Temperatures measured at the bed exit correspond to a mix cup temperature (as the corresponding thermocouple was place in the liquid collector at the bottom of the bed). In most of the experiments with the two larger particles these values were higher than those obtained by averaging on the cross section the readings of the nine thermocouples placed just before the bed exit. The latter estimation ignores the hotter extra flow in the wall-zone. Instead, for the two smaller particles, either both measurements matched or the mixed cup temperature was in some instances even slightly lower (probably because of some heat losses from the bottom head of the tube).

The effect of the wall-zone on heat transfer features has been frequently reported for one-phase flow (e.g. Dixon, Diconstanzo, & Soucy, 1984), but to our knowledge it has not been previously recognized for two-phase flow. Considering this apparent limitation of the 2DPPF model, we have only considered the smaller particles, $d_p = 1.5$ and 3 mm, to correlate Nu_w values with the experimental variables. Employing the type of dependency of Nu_w used frequently for one-phase flow (Lemcoff, Pereira Duarte, & Martínez, 1990) and in some studies for two-phase flow (Specchia & Baldi, 1979; Muroyama et al., 1977), we obtained

$$Nu_w = Nu_{w0} + 0.471 Pr_L^{1/3} Re_L^{0.65}, \tag{1}$$

for $a > 15$, $Re_L < 40$.

The usual Chilton–Colburn dependency of Nusselt number on Prandtl number was assumed in Eq. (1). Hence, the appearance of the factor $Pr_L^{1/3}$. The value of the Nusselt number without fluid flow, Nu_{w0}, was a priori estimated, as reported in Mariani (2000). Therefore, only the Re_L exponent and the factor 0.471 were adjusted.

The deviations between experimental values (Fig. 2) and those from Eq. (1) and from other correlations presented in bibliography are given in Table 1, in terms of the average error

$$\sum_{i=1}^{N} |(Nu_{w,i} - Nu_{w,i}^{\text{pred}})/Nu_{w,i}|/N$$

(N is the number of experimental observations).

As expected, Eq. (1) provides the best fitting of our Nu_w data set. The other correlations showed large deviations with respect to the present data and also among themselves. On one hand, this can be a consequence of the inherent difficulties in estimating h_w from experimental temperature profiles, a fact that has been frequently observed for one-phase flow (e.g. Lemcoff et al., 1990). Besides, some of the previous correlations are based on data showing some deficiencies, because of the experimental procedure, the process to reduce the experimental data or the range of operating conditions, which sometimes were outside the trickle regime. These aspects are treated in detail by Mariani (2000).

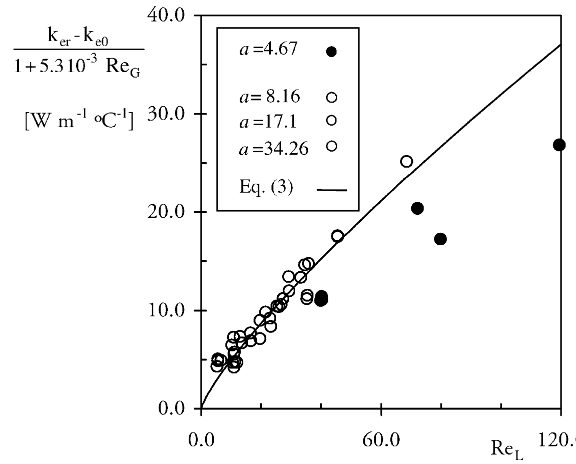

Fig. 3. Experimental and predicted values of k_{er} vs liquid Reynolds number, Re_L.

3.2. Effective radial thermal conductivity

The fitted values of k_{er} showed a weak, but definite, dependency on the gas Reynolds number Re_G. An increase of about 30% was observed from the lowest to the highest values of Re_G. Nonetheless, the strongest effect was that of Re_L. Assuming that k_{er} is the result of adding contributions from the bed without fluid-flow and, mainly, from the liquid-flow lateralization, the following expression was adopted:

$$k_{er} = k_{e0} + b k_L (1 + c Re_G) Re_L^d Pr_L, \tag{2}$$

where b, c and d are fitting parameters and k_{e0} is the contribution without fluid-flow, estimated as detailed by Mariani (2000). This contribution was at least one order of magnitude lower than the flow contribution. The term $(1 + c Re_G)$, with $c > 0$, represents an enhancement effect of the gas-flow to liquid-flow lateralization.

The experimental data and the results from Eq. (2) after fitting all data except those for $d_p = 11$ mm ($a = 4.76$) are given in Fig. 3. Eq. (2) thus becomes

$$k_{er} = k_{e0} + 0.281 k_L (1 + 5.3\,10^{-3} Re_G) Re_L^{0.81} Pr_L, \quad a > 8. \tag{3}$$

The values of k_{er} for $d_p = 11$ mm, also plotted in Fig. 3, are evidently lower than the trend (quantified by Eq. (3)) followed by the remaining packings. This can be most probably due to the suspected inappropriateness of the 2DPPF model at low aspect ratios. In fact, at the

Table 1
Comparison between values of k_{er} and Nu_w from the present experiments with the values from different correlations

Correlation	Average error (%)	Points with positive deviation	Points with negative deviation
Nu_w			
Matsuura et al. (1979b)	85.0	26	0
Specchia & Baldi (1979)	29.0	19	7
Muroyama et al. (1977)	58.7	26	0
Equation (1)	17.4	17	9
k_{er}			
Matsuura et al. (1979a)	26.7	4	31
Specchia & Baldi (1979)	143.2	2	33
Hashimoto et al. (1976)	42.6	4	31
Chu & Ng (1985)	16.5	20	15
Lamine et al. (1996)	31.1	7	28
Equation (3)	11.4	21	14

lowest aspect ratio (e.g. $a = 4.76$) the increased flow in the wall-zone would make the actual liquid superficial velocity in the bulk of the bed significantly decrease below the uniform value on which Re_L is based. Therefore, Eq. (3) will predict values higher than the actual ones at low aspect ratios, $a < 8$ according to the tested set.

A comparison between Eq. (3) and former correlations for k_{er} with the present data (excluding those for $d_p = 11$ mm) is given in Table 1, based on the average error $\sum_{i=1}^{N} |(k_{er,i} - k_{\mathrm{er},i}^{\mathrm{pred}})/k_{er,i}|/N$. Comments similar to those made for the values of Nu_w apply. However, Eq. (3) for k_{er} is more precise than Eq. (1) for Nu_w. This fact arises because the spread of h_w values is larger than for k_{er}. In turn, this may be a consequence that k_{er} responds physically better to actual heat transfer mechanisms than h_w. It should also be noted that a fair agreement is found between the present data and values predicted by the theoretically developed expression of Chu and Ng (1985) for k_{er} (Table 1).

4. Overall heat transfer coefficient

The observed heat transfer rates can be expressed in terms of an overall heat transfer coefficient, h_T, from the cocurrent two-phase flow to the wall. Following the usual definition, when the coolant temperature T_c may be regarded as being uniform, h_T is expressed as

$$h_T = \left\{ \frac{2\pi RL}{W} \left[\ln \left(\frac{T_c - \bar{T}_0}{T_c - \bar{T}} \right) \right]^{-1} - \frac{1}{h_c} \right\}^{-1}, \tag{4}$$

where $\bar{T}_0$ and $\bar{T}$ are the average temperatures at the bed inlet and outlet, and W (J s^{-1} °C^{-1}) is the average (between inlet and outlet conditions) increase of the enthalpy of the gas–liquid stream, assumed at equilibrium,

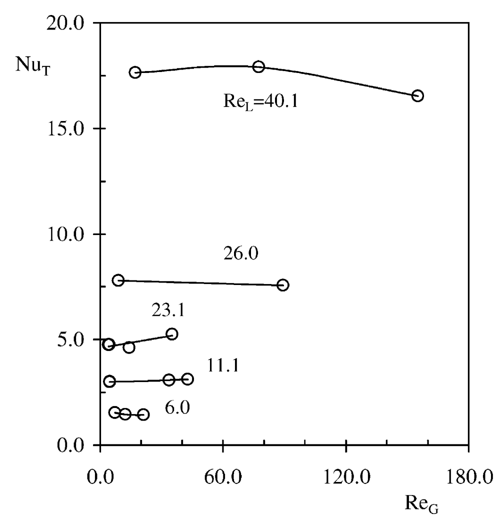

Fig. 4. Experimental values of overall Nusselt number, Nu_T vs gas Reynolds number, Re_G

per unit degree centigrade. Therefore, W takes into account the rate of liquid vaporization while the mixture is heated up.

The values of h_T calculated from Eq. (4) reveal that the effect of the gas flow rate was almost negligible and it does not follow a definite trend. This can be appreciated in Fig. 4, where $Nu_T = h_T d_p / k_L$ is plotted against Re_G at different levels of Re_L.

The values of Nu_T are plotted in Figs. 5A and B against Re_L. Some conditions involving the shortest or the medium heating lengths (270 and 470 mm) were not included, as they evidenced thermal entry effects (temperature profiles were not completely developed). Therefore, the data in Figs. 5A and B correspond to asymptotic h_T values.

The evident effect of the aspect ratio on Nu_T can be interpreted as due to the increase of k_{er} with d_p (for a

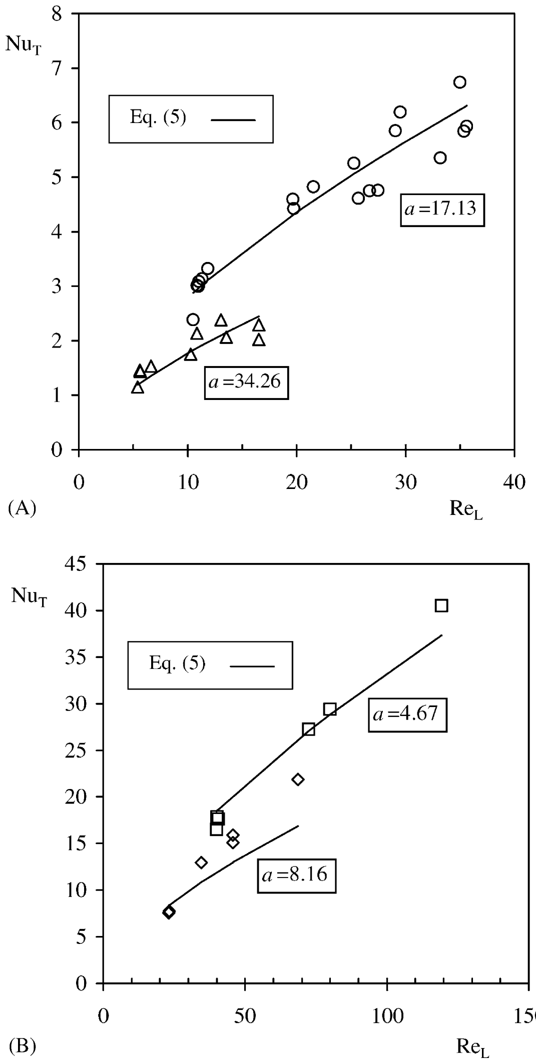

Fig. 5. Experimental and predicted overall Nusselt number, Nu_T vs liquid Reynolds number, Re_L. (A) Higher aspect ratios. (B) Lower aspect ratios.

given tube radius, R) or to the increase of the thermal path with R (for a given d_p). In addition, the apparent effect of the wall-zone should stress this trend at low aspect ratios.

The values of Nu_T were correlated with operating conditions by means of the following expression

$$Nu_T = \frac{h_T d_p}{k_L} = \left[3.87 - 3.77\exp\left(\frac{-1.37}{a}\right)\right] Re_L^{0.643} Pr_L^{1/3}, \tag{5}$$

where the effect of Pr_L was assumed in a similar way as made in Eq. (1).

The use of Eq. (5) is restricted to operations in the trickle regime and, according to the ranges covered in this investigation, to $a > 4.7$ and $5.4 < Re_L < 119.6$. Its average deviation, expressed as $\sum_{i=1}^{N} |(h_{T,i} - h_{T,i}^{\text{pred}})/h_{T,i}|/N$, is less than 9%. Therefore, it is concluded that Eq. (5) can suitably represent the experimental data.

To our knowledge, there is no other attempt in bibliography to correlate values of h_T. The contributions on the subject focused the analytical efforts on the parameters of the 2DPPF model, as described in Section 3. We believe that the importance of knowing h_T should not be underestimated. Actually, for a heat exchange process without chemical reactions, radial temperature profiles are not usually relevant and the knowledge of h_T suffices to estimate the heat transfer rate. For multitubular bed reactors h_T is the only thermal parameter needed by a one-dimensional pseudohomogeneous plug-flow model, which in spite on being a first approximation approach, its usefulness cannot be ruled out for some applications and systems with moderate reaction heat effects.

5. Conclusions

Results from an experimental investigation on heat transfer from a packed bed with cocurrent gas-liquid downflow to the wall have been presented. The measurements cover the range of operating liquid and gas flow rates of the trickle (low interaction) regime in beds presenting aspect ratios (tube to particle diameter ratio) from 4.67 to 34.26.

Values of radial effective thermal conductivity and wall heat transfer coefficient, according to a two-dimensional pseudohomogeneous plug-flow model, have been inferred by regression of the temperature distribution on the bed cross section. The analysis of the results showed evidences that the 2DPPF model is not appropriate for low aspect ratios, a fact which has already been pointed out for one-phase flow. The increased flow in the region nearby the bed wall is strongly suspected to be the key factor for the limitations of 2DPPF at low aspect ratios.

Consequently, correlations based on the present experimental results for h_w and k_{er} (Eqs. (1) and (3)) were proposed for aspect ratios above 15 for h_w and above 8 for k_{er}. Although it would be possible to predict h_w and k_{er} at lower aspect ratios, it is believed that the use of the 2DPPF model cannot be recommended in reactor modelling, as it will fail in predicting actual temperature profiles resulting by the coupling of reaction and heat transport effects. The correlation obtained for k_{er} is more precise than that for h_w, a result suggesting that k_{er} responds physically better than h_w to actual heat transfer mechanisms.

The observed heat transfer rates were expressed in terms of an overall heat transfer coefficient. An expression proposed to estimate h_T (Eq. 5) was able to fit with good precision the whole set of experimental data. To our knowledge, no expression for h_T was previously presented.

As a general observation, the flow rate of the gas phase, within the range of values allowed by the low interaction regime, shows little effect on the heat transfer rate or on the thermal parameters. This finding can be ascribed to the very low mass-flow contribution of the gas phase and to the low interaction feature of the trickle regime.

We believe that the present set of data for h_w, k_{er} and h_T can be regarded as being reliable, as they thoroughly cover the range of gas and liquid flow rates in the trickle regime and much care was taken in the experimental and regression procedures. Nonetheless, limitations are worth remarking. The questionable applicability of 2DPPF model at low aspect ratios prompts for flow distribution measurements and for the development of alternative models describing temperature and flow distributions. Experiments with particle shapes other than spherical and with different liquid properties are also needed to complete a body of basic experimental data.

Notation

a	bed to particle diameter ratio, dimensionless
C_P	specific heat, J kg^{-1} °C^{-1}
d_p	particle diameter, m
G	superficial mass velocity, kg m^{-2} s^{-1}
h_w	wall heat transfer coefficient, W m^{-2} °C^{-1}
h_c	jacket heat transfer coefficient, W m^{-2} °C^{-1}
h_T	overall heat transfer coefficient in the bed, W m^{-2} °C^{-1}
k	fluid thermal conductivity, W m^{-1} °C^{-1}
k_{er}	effective radial thermal conductivity, W m^{-1} °C^{-1}
Nu_w	Nusselt number at the wall, $h_w d_p/k_L$, dimensionless
Nu_T	overall Nusselt number, $h_T d_p/k_L$, dimensionless
Pr	Prandtl number, $C_P \mu/k$, dimensionless
Re	Reynolds number, $G d_p/\mu$, dimensionless
R	tube radius, m
μ	dynamic viscosity, Pa s

Subscripts

c	coolant
G	gas
L	liquid
w	wall

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