Experimental study of gas-liquid heat transfer in a 2-phase flow in a packed bed

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Abstract. The paper deals with an interfacial heat exchange between gas (air) and liquid (water) flowing countercurrently through a packed bed. The experimental investigation was performed with the use of column of 0.1 m inner diameter and filled in with glass Raschig rings. Loads of working media ranged between 0.018÷0.14 ×10⁻¹ m³·(m²·s)⁻¹ and 0.0007÷0.0053 m³·(m²·s)⁻¹ for gas and liquid phases, respectively. It was found that interfacial Nusselt number is mainly dependent on the gas load, noticeably influenced by the temperature difference between phases and practically independent of the liquid load.

1. Introduction

Countercurrent gas-liquid flow through the packed bed finds its application in chemical engineering, in particular in heat and mass transfer processes as rectification [1], absorption [2,3], stripping [4] and cooling [5]. In these processes, the packing material is required to provide a large contact area between flowing media as well as to ensure a sufficient liquid holdup required to reach the expected process efficiency [6].

The description of the countercurrent gas liquid flow inside the packed bed is a challenging task even if isothermal conditions are assumed. The complex geometry of the bed composed of random packing material, leads to unpredictable liquid behaviour. The flow pattern strongly influences the phase interactions and in turn decides about the pressure drop and liquid holdup responsible for efficiency of transfer process [7]. If the process is additionally nonisothermal the significant changes in material properties of both phases have to be taken into account.

Though, the flow hydrodynamics is frequently undertaken by many researchers [8], much less works is dedicated to the heat transfer phenomena occurring in packed bed. The reported papers concerned the fundamental study of the effective radial heat transfer coefficient [9], the bed to wall heat transfer coefficient [10] and the particle to fluid heat transfer coefficient [11]. According to the authors knowledge there is only one work dedicated to this subject of the interfacial gas-liquid heat transfer processes occurring inside the packed bed. Heidari et al. [12] analysed interfacial heat transfer in simplified model of trickle bed reactor (TBR) of 60 mm inner diameter with fourteen spheres of identical diameter equal to 40 mm onto the bed axis. The co-current gas-liquid flow was considered. The impact of selected group numbers on interfacial Nusselt number was examined showing heat transfer enhancement with increasing gas Reynolds and Prandtl numbers and its inhibition with liquid Reynolds, Prandtl and Eötvös.
The aim of the present research is the parametric study of the interfacial gas-liquid heat exchange under the varying countercurrent flow conditions including gas and liquid flow rates and interphase temperature difference.

2. Experimental procedure
The experiment was performed with the use of a small laboratory test rig schematically shown in Fig. 1. The distilled water and the ambient air were used as flowing media. The water was pumped from the main container (5) and flowed through the solid particles filter (8), the flow meter (17) and the distributor (21), ensuring the uniform wetting of the packing surface to the column (19). Afterwards, it flowed freely thanks to the gravity force through the packed bed (20) and it was collected in the tank (12) from which it flowed back to the main container (5). Thus the water circulated in a closed loop and its flow was enforced by the pump (9) and controlled by operating valves (7) and (11). The column was filled with glass Raschig rings (20). The characteristics of the packing element are collected in Tab. 1.

![Figure 1. Schematic diagram of experimental facility](image)

The air flow, indicated in Fig. 1 by grey lines, was enforced by a vacuum pump (10). The air was sucked to the column at its bottom (see grey arrows in Fig. 1) and flowed countercurrently to the liquid upward the packed bed. Then air leaved the column through the outlet (23) and reached the condenser (16) where the water vapour was condensed and collected in tank (14), whereas the air passed through the gas flow meter (15) and quit the test rig.
Temperature of the water in main container was kept constant thanks to the temperature controller (3) connected to a thermocouple (4) and a heater (6). The temperature of water feeding and leaving the column was measured with the use of sensors (24) and (31), respectively. The gas temperature entering and leaving the packed bed and at the outlet of gas flow meter (15) was measured with the use of sensors (30), (25) and (18). The signals from each sensor were delivered through the analogue to digital converter (2) to the personal computer (1) where were further post processed with the use of LabVIEW software. It should be remarked that air humidity was also analysed and measured with sensors (29) and (26).

The accurate measurements of liquid and gas temperatures were of the high importance, thus it had to be performed with a particular care. For that purpose steady state conditions had to be achieved in terms of media fluxes and their inlet and outlet temperatures. In order to assess whether the thermodynamic equilibrium was reached the temperatures from all sensor were monitored in time. The average time required to achieve steady state conditions for a single process configuration (fluxes and inlet temperatures of media) was equal to about one hour. The study was performed for a wide range of media fluxes, inlet water temperature and for an ambient inlet temperature of the air. The complete set of operating parameters is given in Tab. 2.

### Table 2. Operating parameters used in the experiment.

| parameter             | symbol | unit          | value          |
|-----------------------|--------|---------------|----------------|
| inlet liquid load     | \( u_L \) | \( \text{m}^3/\text{m}^2/\text{s} \) | \( 0.71 \div 5.31 \times 10^{-3} \) |
| inlet gas load        | \( u_G \) | \( \text{m}^3/\text{m}^2/\text{s} \) | \( 0.0177 \div 0.1415 \) |
| inlet liquid temperature | \( T_{L,\text{in}} \) | \( ^\circ\text{C} \) | \( 30 \div 70 \) |
| inlet gas temperature  | \( T_{G,\text{in}} \) | \( ^\circ\text{C} \) | \( 21 \pm 1 \) |

### 3. Interfacial heat transfer estimation

The objective of the present study was to determine the heat flux exchanged between phases and then to quantify the heat transfer process by the most common group numbers. It should be noted that, during the experiment, the water vapour mass fraction in air was at the level between 3÷5% (with some exceptions when it could reach 8%). Thus, it could be assumed that neither the specific heat \( c_G \) of air nor the heat transfer coefficient \( h \) depend on temperature \( T \) and air humidity \( X \) and consequently on vertical position \( x \):

\[
h \neq f(X, T_G, T_L) \Rightarrow h \neq f(x) \quad \text{and} \quad c_G \neq f(X, T_G, T_L) \Rightarrow c_G \neq f(x) \tag{1}
\]

where indexes \( G \) and \( L \) correspond to gas and liquid phase, respectively. The heat flux transferred from the liquid phase can be written as:

\[
\dot{Q}_L = \dot{Q}_{es} + \dot{Q}_{vap} + \dot{Q}_{loss} \tag{2}
\]

In the relation (2) \( \dot{Q}_{es} \) stands for the heat flux exchanged between phases, \( \dot{Q}_{vap} \) is the heat flux consumed by evaporating water, and \( \dot{Q}_{loss} \) represents heat loses through the column wall. It should
be remarked that the column wall was carefully isolated so the last term in relation (2) can be neglected. Assuming that the humidity of air $X$ changes linearly with coordinate $x$ the elementary portion of heat flux exchanged between phases can be expressed as:

$$\, d\dot{Q}_{ex} = h \cdot \Delta T \cdot ds = h \cdot \Delta T \cdot a \cdot S \cdot dx$$

(3)

where $ds$ is the elementary surface of packed bed, $a$ its specific surface area, $S$ represents column cross sectional area and $\Delta T$ stands for the temperature difference between phases (see Fig. 2). Integration along the packing section of height $L$ gives total heat flux $\dot{Q}_{ex}$:

$$\dot{Q}_{ex} = \int_0^L d\dot{Q}_{ex} = a \cdot S \cdot h \int_0^L (T_L - T_G) \, dx$$

(4)

In principle in a countercurrent heat exchange system temperatures vary in a logarithmic manner.

Figure 2. Assumed temperature distributions for both phases along the column

However, taking into account relatively low temperature ranges, their linear dependences (see Fig. 2) were assumed, i.e.:

$$T_L, T_G - x \Rightarrow \Delta T - x$$

(5)

allowing to integrate expression (4) yielding:

$$\dot{Q}_{ex} = a \cdot S \cdot h \cdot L \left[ \left( T_{L,\text{in}} + T_{L,\text{out}} \right) - \left( T_{G,\text{in}} + T_{G,\text{out}} \right) \right] = a \cdot V_{\text{col}} \cdot h \left( \overline{T_L} - \overline{T_G} \right) = a \cdot V_{\text{col}} \cdot h \cdot \overline{\Delta T}$$

(6)

where $V_{\text{col}}$ represents the packing section volume, and overbar $\overline{()}$ denotes value averaged along the column. The elementary heat flux absorbed by the gas phase equals to:

$$d\dot{Q}_G = \dot{m}_G \cdot c_G \cdot d\Delta T_G$$

(7)

where:

$$d\Delta T_G = \frac{dx}{L} \Delta T_G$$

(8)

Assuming linear growth in water vapour mass flux $\dot{m}_{\text{vap}}$ with position $x$, the gas mass flux $\dot{m}_G$ writes:

$$\dot{m}_G = \dot{m}_{\text{air}} + \frac{x}{L} \dot{m}_{\text{vap}}$$

(9)
where \( \dot{m}_{\text{air}} \) is the humid air mass flux at the inlet. Using (8) and (9) one may integrate expression (7) and arrive finally to heat flux \( \dot{Q}_G \):

\[
\dot{Q}_G = \int_0^L \int_0^L \left( x \cdot \dot{m}_{\text{vap}} + \dot{m}_{\text{air}} \right) c_G \frac{\Delta T_G}{L} dx = \left( \dot{m}_{\text{air}} + \frac{1}{2} \dot{m}_{\text{vap}} \right) c_G \cdot \Delta T_G
\]

Assuming equality of heat fluxes \( \dot{Q}_{\text{ex}} \) and \( \dot{Q}_G \) (expressions 6 and 10) the interfacial heat exchange coefficient can be determined as:

\[
h = \frac{c_G \cdot \Delta T_G}{a \cdot V_{\text{col}} \cdot \Delta T} \left( \dot{m}_{\text{air}} + \frac{1}{2} \dot{m}_{\text{vap}} \right)
\]

The mass fluxes in expression (11) can be calculated taking into account the gas flow meter reading and the measurements of gas temperature and humidity at the inlet/outlet to the packing section.

4. Results

The heat transfer measurements were conducted according to the conditions summarised in Tab. 2. The heat absorbed by the gas phase as a function of gas and liquid loads (superficial velocities) for 3 different inlet liquid temperatures (i.e. 30°C, 50°C, and 70°C) is presented in Fig. 3a. As can be seen the heat flux is practically independent of the liquid load while it grows with increasing gas load and temperature difference between phases. It is worth to note that the heat fluxes \( \dot{Q}_{\text{ex}} \) varies with gas load in linear manner whereas its dependence on inlet liquid temperature (with the fixed inlet gas temperature) is slightly nonlinear, i.e. exceeding its growth proportional to the \( \Delta T \) increase.

**Figure 3.** Heat exchanged between phases (a) and Nusselt number (b) versus superficial liquid and gas velocities for three different inlet liquid temperatures. Inlet gas temperature was fixed at 21°C.

The measurement results from were then recalculated to the form showing the intensity of heat exchange between phases. Instead of interfacial heat transfer coefficient \( h \) it was decided to apply use Nusselt number relating the total heat transfer to the conductive heat transfer according to:

\[
Nu = \frac{h \cdot d_e}{k_L}
\]

where \( k_L \) is liquid conductivity and \( d_e \) represents the characteristic lengthscale of the packed bed:

\[
d_e = a^{-1}
\]
The Nusselt number distribution in the same coordinating system is presented in Fig. 3b. Similarly to \( \dot{Q}_c \) Nusselt number increases with gas superficial velocity which is in agreement with observations of Heidari et al. [12] devoted to the trickle-bed reactor. In contrary to the heat flux \( Nu \) indicates slight dependence on liquid superficial velocity which becomes especially evident for the lowest loads. This could be attributed to either increased experimental uncertainty both in terms of heat flux and flow rate measurements or the qualitative change of the phenomenon. Extremely low liquid superficial velocities lead to extended liquid passage times through the column which may contribute to relatively increased heat loses to the environment and thus influence the resulting heat transfer coefficient. It is also worth noticing that Nusselt number shown in Fig. 3b depends on gas-liquid temperature difference. One could rather expect that the 3 surfaces corresponding to different inlet liquid temperatures collapse into one common distribution with insignificant scatter of data points. Instead the surfaces diverge with increasing gas load. For instance in the high liquid load range the rise in inlet liquid temperature from 30°C to 50°C results in \( Nu \) change from 0.45 to 0.52 (appr. 15% growth), while further rise in \( T_{L,in} \) to 70°C results in \( Nu = 0.62 \) (19% growth). The possible explanation of \( Nu \) variation with \( \Delta T \) may be the change of the contact area between phases due to varying liquid viscosity leading in turn to better surface wettability. In the calculation procedure (see Eq. 11) the contact area between phases was assumed to be equal to the packed bed specific surface area, i.e. perfect wettability. As discussed in [13] the wetting efficiency of the packed bed changes with varying liquid load as well as with liquid physical properties. However, it is still a challenge to quantify this parameter especially when different packing element types and sizes are considered.

In order to generalise the results presented in Fig. 3 the Nusselt number should be expressed as a function of the other group numbers. According to [14] the Reynolds, Galileo, Prandtl and Eötvös numbers are regarded as the most relevant group numbers to describe the heat transfer processes in a 2-phase flow system. According to [5] the liquid Reynolds number for packed beds takes the form:

\[
\text{Re}_L = \frac{u_L \cdot \rho_L}{\mu_L} \quad (14a)
\]

while for the gas phase it includes additionally packed bed porosity \( \varepsilon \):

\[
\text{Re}_G = \frac{u_G \cdot d_p \cdot \rho_G}{(1 - \varepsilon) \mu_G} \quad (14b)
\]

In definition (14b) \( d_p \) is the specific dimension defined as:

\[
d_p = 6 \frac{1 - \varepsilon}{a} \quad (15)
\]

The Galileo number expressing the relation between the gravity and viscous forces writes:

\[
Ga = \frac{\rho_L \cdot g \cdot d_l^2 \cdot \varepsilon^2}{\mu^2 (1 - \varepsilon)^3} \quad (16)
\]

The Prandtl number takes thermal and viscous diffusion rates into account:

\[
\text{Pr}_L = \frac{H_L \cdot \rho_L}{k_L} \quad (17)
\]

And finally the Eötvös number definition, balancing gravity and surface tension forces:

\[
Eö = \frac{\rho_L \cdot g \cdot d_l^2 \cdot \varepsilon^2}{\sigma (1 - \varepsilon)^2} \quad (18)
\]

where \( \sigma \) is the gas-liquid surface tension and \( d_l \) is a characteristic lengthscale of the packing material given as [15]:
\[ d_l = \frac{d_e \cdot e}{1 - e} \]  

(19)

In order to describe the interfacial gas-liquid heat transfer in trickle-bed reactor the following relationship was considered:

\[ Nu = Re_L^A \cdot Re_G^B \cdot Ga_L^C \cdot Ga_G^D \cdot Pr_L^E \cdot Eo^F \]  

(20)

where the group numbers exponents were unknown and they had to be found using regression analysis to fit the \( Nu \) distribution from Fig. 3b in the most accurate way. Detailed analysis of various group number combinations allowed to fit the measurement data with the following correlation:

\[ Nu = Re_L^{-0.03717} \cdot Re_G^{1.1045} \cdot Ga_L^{-1.3483} \cdot Pr_L^{1.2001} \cdot Eo^{2.0911} \]  

(21)

Figure 4 presents the Nusselt number calculated with the use of formula (21) against the experimental data. In order to make it easier to interpret the results the solid lines corresponding to \( \pm 10\% \) error are plotted in the graph. As can be seen the proposed correlation fits the measured data with very high accuracy characterised by the correlation coefficient equal to 0.998. Very few data points lie outside the \( \pm 10\% \) limits and they correspond to the lowest liquid load range where the increased measurement uncertainty may be observed.

![Figure 4. Comparison of the experimental and modelled Nusselt number values](image)

5. Summary

Interfacial heat flux in the countercurrent gas-liquid flow through the packed bed filled with Raschig rings was investigated. It was found that calculated Nusselt number is strongly dependent on the gas load, noticeably dependent on the temperature difference between phases and slightly dependent on the liquid load. The correlation for Nusselt number taking into account the effects of inertia, gravity, surface tension, viscous and thermal diffusion represented by Reynolds, Prandtl, Galileo and Eötvös numbers was proposed. It has been found that the proposed correlation for Nusselt number fit the experimental data with very good accuracy, i.e. adequately reflecting the physical processes taking place in a porous media tricked with liquid phase. Slight scatter of data points was observed for the lowest liquid loads most likely due to increased measuring uncertainty.
The papers show the need for further research work to provide better understanding of the heat transfer processes in complex geometrical constraints. The wetting efficiency of the packed bed seems one of the most important factors governing the interfacial heat transfer. It is planned to continue the work in a new test facility allowing to study much wider gas and liquid loads as well as different random packing element types and sizes.

6. References
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