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Experimental investigation of the heat exchange to a compartmentalized plug flow

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Abstract — A two-phase (air–water), tubular counterflow heat exchanger was developed for industrial unitary process applications to realize an active heat transfer control. A controlled train of air plugs was injected in a stratified smooth water flow to yield a compartmentalized, continuous two-phase flow whose stability was carefully monitored. Related Nusselt numbers were evaluated for single- and two-phase flows, at the same flow regimes. This comparison shows how the traveling, controlled air plug "turbulators" allowed for a large heat transfer rate increase, for all observed heating rates and phase velocity ratios. The analysis includes a comparison with the available literature correlations, and the efficiency of the adopted heat transfer control is finally inferred while a range of flow regimes is identified which optimizes the heat exchange. © 1999 Éditions scientifiques et médicales Elsevier SAS

active heat transfer control / heat transfer enhancement / two-phase heat exchanger / sectioned air–water flow / controlled air plugs

Nomenclature

c	specific heat	$J{\cdot}kg^{-1}{\cdot}K^{-1}$
G	mass flow rate	$kg{\cdot}s^{-1}$
h	convective heat transfer coefficient	$W{\cdot}m^{-2}{\cdot}K^{-1}$
k	conductivity	$W{\cdot}m^{-1}{\cdot}K^{-1}$
j	velocity	$m{\cdot}s^{-1}$
L	heat exchanger length	m
Nu	Nusselt number $= 2hr_{c1}/k_l$	(.)
r	radius	m
Re	Reynolds number $= 2G_c r_{c1}/\mu_l$	(.)
R_w	wall thermal resistance parameter defined in equation (1)	(.)
T	temperature	K

Greek symbols

μ	dynamic viscosity	$N{\cdot}s{\cdot}m^{-2}$
Ψ	enhancement parameter $= h_{tp}/h_l$	(.)

Subscripts

c	cold
g	gas
h	hot
i	inlet
l	single-phase (liquid)
o	outlet
tp	two-phase
w	wall
1	internal
2	external

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1. INTRODUCTION

The general nature of two-phase flows, with the aim of defining the various regimes of fluid dynamics and discussing the influence of these regimes on the realized heat transfer processes, has been widely investigated. The method of the regime map, as originally proposed by Baker [1] and Scott [2] and then extended by Mandhane et al. [3] and Taitler and Dukler [4], has been generally employed to characterize the sequence of the two-phase flow patterns for a variety of configurations, including the most common air–water flow in horizontal pipes. In *figure 1* the flow pattern terminology, as proposed by Barnea [5], is reproduced. Interest in such flows is

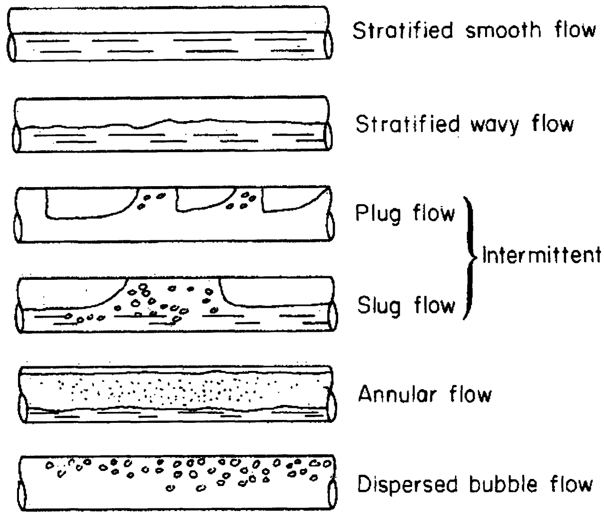

Figure 1. Flow patterns in two-phase horizontal flow.

still sustained in recent times, as proved by the extended pattern nomenclature proposed by Wong and Yau [6].

While the important influence of heat transfer which leads to phase-change has been fully described, starting from the work by Hewitt [7], few contributions have been found in the published literature, including the closure work by Shah [8], with reference to very small flow rate and tube diameter, intermittent subcooled (non-boiling) liquid flow associated with the transport of large bubbles or plugs, and the relative influence of the heat transfer rate departure from pure liquid situations.

Such cases are relevant in a number of unitary process configurations in the chemical, pharmaceutical or food technology industry. The use of an adequate gas flow to compartmentalize the liquid in the tubing may be used to prevent the development of laminar flow, and enable precise identification of the residence time in the heating section of the heat exchanger and a remarkable increase of the local convective heat transfer coefficient. The present work addresses the need to characterize such a peculiar heat exchange configuration.

2. EXPERIMENTAL SET-UP

A diagram, which shows the principal sections of the experimental set-up, is presented in *figure 2*.

Water was withdrawn from a reservoir storage by a rotary pump and sent to a mixing tee where a stable two-phase flow was established following the injection of a regulated pressurized air flow. In order to ensure the compartmentalization, as discussed in the following section, the gas flow was injected from the lower side of the mixing tee. Both flows were controlled by means of regulation valves. The resulting flow then traveled by a nylon tubing to an insulated, glass double-tube heat exchanger (inner-tube i.d. and thickness are $4.3 \cdot 10^{-3}$ and $1.0 \cdot 10^{-3}$ m, respectively; the outer-tube i.d. is $1.7 \cdot 10^{-2}$ m; the overall length is $2.0 \cdot 10^{-1}$ m), where it was heated by a counterflow of hot water, whose temperature and flow rate were controlled by means of a constant temperature circulator.

A constant-temperature boundary condition was ensured to the subject flow, as the outer mass flow rate was one order of magnitude larger than the inner one. Indeed, the temperature drop relative to the external tube was verified to be negligible for all explored conditions.

In order to monitor the flow pattern and the plug fraction (i.e. the ratio of volume flow rate of gas over that of liquid), a transparent tube was provided which was exposed to an aligned laser source (5 mW He : Ne) whose light could be intercepted by a photodiode. The recorded signal fluctuations relative to plug passages, due to the refractive index change, were then visualized by a double-trace oscilloscope and measured.

The volume flow rate of the liquid fraction was measured by intercepting the flow, at the heat exchanger outlet, by means of a graduated beaker and measuring the filling time, while that of the gas fraction was measured prior to the injection by means of a bubble soap flow meter, the related uncertainties being estimated equal to $\pm 1\,\%$ of each flow rate reading. Transferred thermal power was then determined by measuring the flow temperatures at the inner flow inlet and outlet. K-type ($\varnothing = 0.5$ mm) calibrated thermocouples were employed at both flow sides, and related signals were acquired by a PC-based PCL 818 HG Advantech card, with a sampling rate of 0.1 kHz. Hence, at least 1000 temperature data were acquired to yield for each value a mean which was affected by a standard deviation equal to $\pm 0.1\,\%$. Finally, the errors related to linear dimensioning and fluid and glass thermal properties were neglected.

3. RESULTS AND DISCUSSION

The experiments were performed with a liquid volume flow rate ranging from $0.6 \cdot 10^{-6}$ to $6.0 \cdot 10^{-6}$ $m^3 \cdot s^{-1}$, first by letting only liquid, then allowing for an additional constant gas volume flow of $1 \cdot 10^{-6}$ $m^3 \cdot s^{-1}$. As volume flow rates were sampled randomly, in case of two-phase flow the plug fraction had to be constantly monitored by

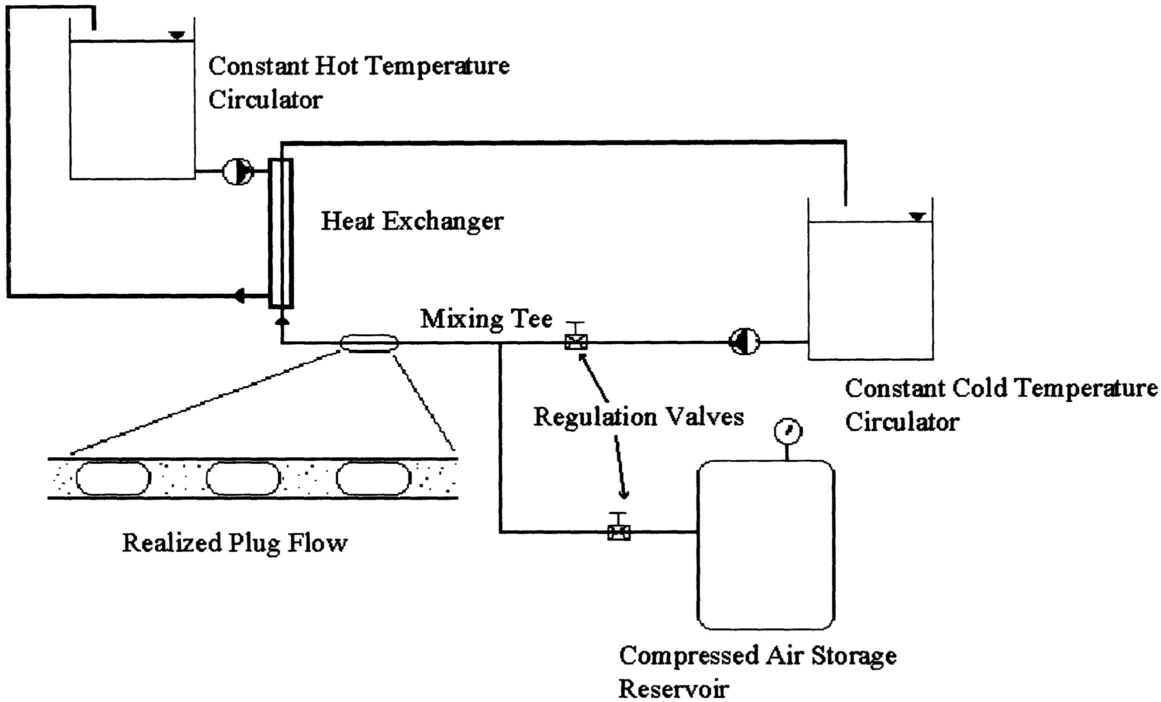

Figure 2. Experimental set-up.

means of the optical system to ensure the steady-state experimental conditions. It has to be considered that the air flow rate, due to its negligible contribution, was not taken into account in the determination of the overall energy balance which follows.

With the adopted flow rates, the flow conditions could be identified in the stratified smooth region of the Taitel and Dukler [4] flow condition map, as reported in *figure 3*. Nevertheless, the plug flow was realized by supplying air to the mixing tee, as mentioned earlier, providing that the gas upflow be carefully broken by the horizontal liquid current. The effect of the inherent surface tension at the wall of the tube, due to its relatively small cross-section and the adopted low flow rates, was also evident in favoring the partitioning. The air plugs had nearly the same cross-section as that of the tube and moved along with a nearly-symmetrical shape, as determined by the low gas flow rates employed. The absence of significant liquid transfer was verified by additional tests, in which a small quantity of dye was injected in one liquid plug and it was observed that no appreciable diffusion was produced in the neighboring plugs.

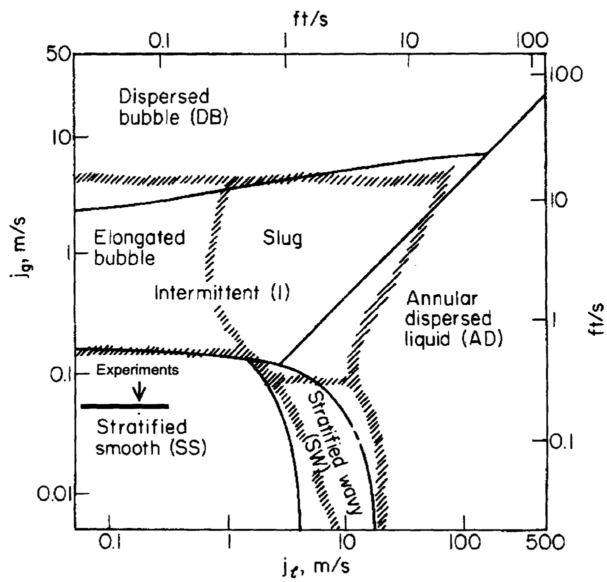

Figure 3. Flow regime transitions for air and water in a horizontal tube [4], with comparison with data by Mandhane [3], and location of the present experimental conditions.

With the adopted heat exchange configuration, it is necessary to evaluate firstly whether the problem may be treated as a conventional forced convection one, or whether adjustment should be made to account for the

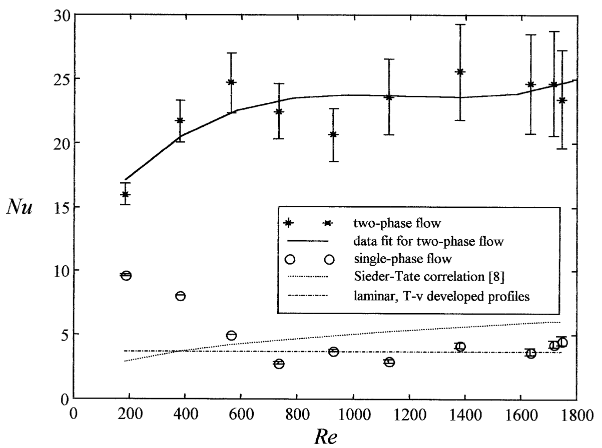

Figure 4. Measured *Nu* versus *Re* for single-phase and compartmentalized flows, with comparison with available correlations and related error bars.

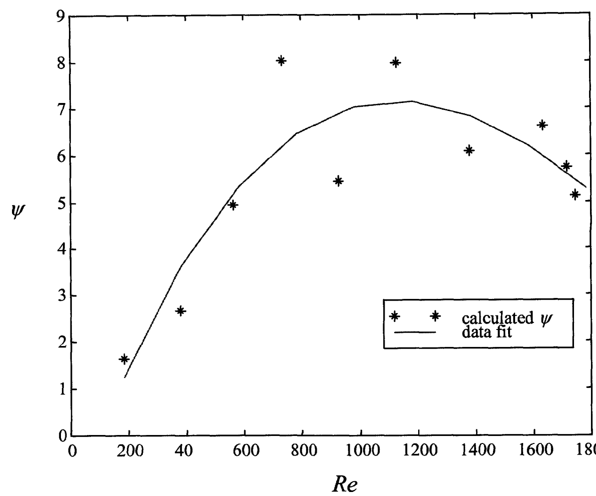

Figure 5. Calculated enhancement parameter Ψ with respect to flow regime.

conjugate effect due to temperature drop across the inner tube wall. To this end, the thermal resistance parameter R_w was calculated, relative to single-phase flow, as follows [9]:

$$R_\mathrm{w} = \frac{k_\mathrm{l}}{2\,k_\mathrm{w}}\left[1 + \left(\frac{r_2 - r_1}{r_1}\right)\right] \tag{1}$$

A three-fold higher value than the lower limit of 0.05 was found. Thus the average Nusselt number was evaluated with the following formulation:

$$Nu = \frac{\pi L k_\mathrm{l}(T_\mathrm{hi} - T_\mathrm{co}) - (T_\mathrm{ho} - T_\mathrm{ci})}{G_\mathrm{c} c(T_\mathrm{co} - T_\mathrm{ci}) \ln \frac{(T_\mathrm{hi} - T_\mathrm{co})}{(T_\mathrm{ho} - T_\mathrm{ci})}} - \frac{k_\mathrm{l}}{2\,k_\mathrm{w}} \ln\left(\frac{r_{\mathrm{c}2}}{r_{\mathrm{c}1}}\right) \tag{2}$$

The experimental results were collected for both single- and two-phase flows, for the same value of the driving parameter *Re* ranging from approximately 200 to 1 800, and grouped in *figure 4*. For the sake of comparison with customary adopted correlations, in the same graph the Sieder–Tate correlation, according to Shah [8], and the Nusselt number value for fully developed, constant surface temperature laminar flow are reported.

The experimental uncertainties which are pertinent to each measurement, for single- and two-phase flows, are also included in *figure 4*. The compounded uncertainties were obtained by summing each single uncertainty.

First, the expected departure of measured transfer rates from the reported correlations, in the lower range of adopted *Re*, due to the effect of the thermal entry length is clearly seen. A three-order best-fit polynomial is then employed to help visualize the progress of transfer rate to two-phase flow with *Re*. The remarkable enhancement in heat transfer recorded here is justified by the sequence of intermittent wetting of the whole of the tube, and consequent film drains in the interval between liquid plugs. While the single-phase flow is clearly laminar in the entire flow rate range, in the realized two-phase flow the traveling plugs continuously destroy the laminar sublayer, and consistently create turbulent wakes behind each one. Indeed, according to Shah [8], a turbulent regime is already attained with an *Re* as low as 170.

Results on the enhancement in heat transfer Ψ were also plotted with respect to *Re*, and are presented in *figure 5*, where a three-order best-fit polynomial is provided as well. The enhancement in heat transfer is evident, while an interesting range of flow regimes could be identified in order to optimize the configuration.

4. CONCLUSIONS

A simple experimental facility was set up to investigate the enhancement of heat transfer due to a plug flow. Measurements were carried out accurately to avoid flow instabilities. The presented results show that in unitary industrial heat exchange processes a remarkable increase of heat transfer may be attained by controlling the flow compartmentalization. The presented enhancement parameter Ψ also shows that an interesting potential exists to further investigate the adopted configuration in order to identify an optimal flow regime. Future work will also be performed to include the effect of inclined arrangements and relative Froude numbers, as well as the properties variation with temperature, in order to extend the general correlations provided by Shah [8].

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REFERENCES

[1] Baker O., Simultaneous flow of oil and gas, Oil Gas J. 53 (1954) 185.

[2] Scott D.S., Properties of Cocurrent Gas–Liquid Flow, Advances in Chemical Engineering 4, Academic Press, New York, 1963.

[3] Mandhane J.M., Gregory G.A., Aziz K.A., A flow pattern map for gas–liquid flow in horizontal pipes, Int. J. Heat Mass Tran. 1 (1974) 537.

[4] Taitel J., Dukler A.E., A model for predicting flow regime transitions in horizontal and near horizontal gas–liquid flow, AIChE J. 22 (1976).

[5] Barnea D., A unified model for predicting flow pattern transitions for the whole range of pipe inclination, Int. J. Multiphase Flow 13 (1987) 1.

[6] Wong T.N., Yau Y.K., Flow patterns in two-phase air–water flow, Int. Comm. Heat Mass Tran. 24 (1) (1974) 111.

[7] Hewitt G.F., Two-phase flow patterns and their relationship to two-phase heat transfer, in: Kakaç S., Veziroglu T.N. (Eds.), Two-phase flows and heat transfer, Hemispere Publ. Corp., New York, 1977.

[8] Shah M.M., Generalized prediction of heat transfer during two component gas–liquid flow in tubes and other channels, AIChE Symp. Series 77 (1981) 140.

[9] Shah R.K., London A.L., Laminar flow forced convection in ducts, in: Irvine T.F. Jr., Hartnett J.P. (Eds.), Advances in Heat Transfer – Supplement 1, Academic Press, New York, 1978.