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Low Reynolds number heat and mass transfer measurements of an overall counterflow, baffled, finned-tube, condensing heat exchanger

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Abstract—The heat and mass transfer performance of a condensing heat exchanger is characterized. The study is conducted on an indirect contact, direct transfer, overall counterflow, baffled, copper, finned-tube heat exchanger. Heat and mass transfer data are correlated, and a comparison is made to other studies. Interesting behavior and contrasts of the low Reynolds sensible heat transfer in the condensing and non-condensing cases is observed, and a mechanism explaining the physics of these observations is hypothesized.

INTRODUCTION

HEAT EXCHANGERS are very commonly used by the heat transfer engineer, and comprehensive texts have been written regarding the design and analysis of heat exchangers (e.g. Kays and London [1]). Often, the application involves condensation. Such is the case in condensing boilers where combustion products are condensed to recover the latent energy. Unfortunately, there are few good correlations in the literature for such situations, with none addressing low Reynolds number heat and mass transfer from a condensing water and air mixture to a high integral fin-tube heat exchanger typical of the one in this study. Earlier reports of additional aspects of the work now addressed were given in Idem *et al.* [2, 3]. The first examines the effect of 'T-baffles' on the performance of a non-condensing heat exchanger geometrically similar to the one in this study. The second addresses condensing and non-condensing performance of an unbaffled configuration. The results now presented consider the case of a condensing, baffled heat exchanger, at low to moderate Reynolds numbers.§

Elmahdy and Mitalas [4] used a correlation of the form

$$J = C_1 \, Re^{C_2} \tag{1}$$

to characterize heat exchanger performance. Using fully dry and fully wet data, the authors determined the constants of equation (1) for a dry and wet J, respectively. The authors employed an area weighting scheme to predict the performance of partially wet heat exchangers. The results were in agreement with experimental data within about 4% of the overall heat transfer for 4 and 8 row plate-finned heat exchangers.

Results from non-condensing heat exchanger experiments were used to predict equivalent enthalpy transfer coefficients by Senshu *et al.* [6]. The authors reported agreement between predicted and measured enthalpy transfer coefficients of $\pm 5\%$.

Rudy and Webb [7] reported a technique of measuring the condensate retention during condensation on an integral low-finned tube from a quiescent medium. The results verified that condensate retention was intensified for closer fin spacing. The authors proposed that the gravity drained model of Beatty and Katz [8] was inadequate by virtue of its neglect of surface tension effects. In a later study Webb *et al.* [9] used the Beatty–Katz model to predict the performance of a spine finned tube and found the model to overpredict the performance by about 40%. The discrepancy was again attributed to the surface tension effects. A model to predict condensate retention on integral low-finned tubes was proposed by Rudy and Webb [10], and comparisons with experimental data showed agreement within 10% over most of the experimental range. Webb *et al.* [11] employed the results of these studies to develop a surface tension drained model of heat transfer on integral low-finned tubes. The model neglected vapor shear or condensate inundation effects, and predicted heat transfer coefficients within 20% of experiments.

The effects of condensate inundation have been explored by several investigators. Fujii [12] discussed two models of heat transfer with condensate inundation, but both require knowledge of a parameter describing how falling condensate spreads over the subsequent tube.

Yau *et al.* [13] undertook experimentation to study

§ A third report (Idem and Goldschmidt [5]) directly addressing the effect of 'gull-wing' baffles is currently under review for publication by the American Society of Heating, Refrigerating, and Air-conditioning Engineers.

NOMENCLATURE

- A area [m^2 (ft.^{2})]
- b slope of enthalpy curve [kJ kg^{-1} °C^{-1} (Btu lb$_m^{-1}$ °F^{-1})]
- c_p specific heat [kJ kg^{-1} °C^{-1} (Btu lb$_m^{-1}$ °F^{-1})]
- C_D drag coefficient
- d diameter [m (ft.)]
- f friction factor [dimensionless]
- F force on retained condensate [N (lb$_f$)]
- g acceleration due to gravity [m s^{-2} (ft. s^{-2})]
- G mass flux [kg s^{-1} m^{-2} (lb s^{-1} ft.$^{-2}$)]
- h heat transfer coefficient [W m^{-2} °C^{-1} (Btu h^{-1} ft.$^{-2}$ °F^{-1})]
- h_h enthalpy transfer coefficient [kW kg^{-1} m^{-2} °C^{-1} (Btu lb$_m^{-1}$ ft.$^{-2}$ °F^{-1})]
- i enthalpy [kJ kg^{-1} (Btu lb$_m^{-1}$)]
- J Colburn J-factor [dimensionless]
- J_h enthalpy transfer Colburn J-factor [dimensionless]
- L characteristic length [m (ft.)]
- m mass [kg (lb$_m$)]
- M fin efficiency parameter [dimensionless]
- Nu Nusselt number [dimensionless]
- p pressure [Pa (psi)]
- Pr Prandtl number [dimensionless]
- Q heat transfer rate [W (Btu h^{-1})]
- r radius [m (ft.)]
- Re Reynolds number [dimensionless]
- S_F fin density [fin m^{-1} (fin ft.$^{-1}$)]
- S_L longitudinal pitch [m (ft.)]
- S_T transverse pitch [m (ft.)]
- T temperature [°C (°F)]
- U overall heat transfer coefficient [W m^{-2} °C^{-1} (Btu h^{-1} ft.$^{-2}$ °F^{-1})]
- U_h overall enthalpy transfer coefficient [kW kg^{-1} m^{-2} (Btu lb$_m^{-1}$ ft.$^{-2}$)]
- V velocity [m s^{-1} (ft. s^{-1})].

Greek symbols

- α, β, γ dimensions of retained condensate (see Fig. 8) [m (ft.)]
- δ thickness [m (ft.)]
- η efficiency [dimensionless]
- ρ mass density [kg m^{-3} (lb$_m$ ft.$^{-3}$)]
- σ_w surface tension of water [N m^{-2} (lb$_f$ ft.$^{-2}$)]
- σ minimum/frontal area [dimensionless]
- ω humidity ratio.

Subscripts

- c cold fluid
- f fin
- fr frontal
- h hot fluid
- H hydraulic
- i inside or inlet
- l liquid (water)
- min minimum
- o outside or outlet
- st at saturation
- v vapor (air/water).

Superscript

- $\bar{}$ mean.

the impact of fin spacing on the performance of condensing tubes. It was found that heat transfer enhancement was dependent on fin spacing, and may be influenced by the Taylor instability wavelength. Furthermore, the impact of vapor velocity was determined to be insignificant in the range of the experimental work.

The effect of fin shape was investigated by Marto *et al.* [14]. The authors found that while a parabolic fin shape consistently had the best performance, none of the fin shapes tested exhibited differences greater than the experimental error.

Fujii *et al.* [15] suggested a method of predicting tube bank performance from the performance of a single tube, and the results were accurate to within 10%. The influence of oncoming vapor velocity on the condensation of steam was studied by Fujii *et al.* [16].

Although there has been a tremendous amount of work done in condensing heat exchanger research, as indicated by the partial survey above, research in this area remains ardent. This is particularly so in the area of condensation on integral high-finned tubes, which are being commonly used. The purpose of the present study is to quantify the heat and mass transfer performance of such tubes in a baffled heat exchanger, and to compare this performance to correlations available in the literature (the unbaffled configuration was reported in Idem *et al.* [3]).

EXPERIMENTAL SET-UP

The test set-up, detailed in Idem *et al.* [3], and Idem [17], consisted of a wind tunnel and dedicated instrumentation. Turning vanes, a louvered mixing device, and a fibrous screen upstream of the test heat exchanger were used to obtain uniform flow conditions. The temperature of the water flowing through the heat exchanger was set by mixing 'city' water with water stored in a tank which could be heated or cooled. The approach air temperature and humidity level were set by adjusting the output of four resistance heaters and injecting steam. All walls downstream of the heaters, and the heat exchanger, were well insulated.

Flow nozzles were used for measuring the flow rate of air. An externally mounted psychrometer was used to measure the moisture content upstream of the heat exchanger. A similar psychrometer was located downstream of the heat exchanger in the bottom elbow of the duct. By making a simple mass balance, the amount of condensed water vapor could be deduced. As an added precaution, a third psychrometer was placed in the inclined section of the duct. The effects of additional condensation occurring on the walls of the duct and re-evaporation of the condensate in the bottom elbow were determined to be negligible.

Temperature measurements of the air stream were obtained by means of three thermocouple grids within the wind tunnel. Measurements were made upstream and downstream of the heat exchanger, with the third grid providing quantification of the heat losses to the environment. These heat losses were found to be negligible.

With the experimental uncertainties given in Idem *et al.* [3], the method of Kline and McClintock [18] was employed to evaluate the uncertainties over the range of the experiments reported here. For a typical case, the Colburn J-factors were found to be within 7.9%, and the Reynolds numbers within 8.0% of the reported values. This represents a slight improvement over the uncertainties reported in the data of Idem *et al.* [3].

HEAT EXCHANGER DATA REDUCTION AND REPRESENTATION

All of the testing now reported is for a baffled heat exchanger (Fig. 1). (A plan or side view would show the baffles to extend the full length of the tubes.) The 'gull-wing' shaped baffles are slightly different from the 'T-baffle' configuration reported in Idem *et al.* [2], but are the same as described in ref. [5]. A four tube, two pass cross flow heat exchanger was tested in overall counter flow, with the water flow split evenly among each tube in a pass. The fin diameter, tube inside diameter, and outside diameter were 0.039 m (0.127 ft.), 0.016 m (0.053 ft.) and 0.019 m (0.063 ft.). The average fin density, tube transverse pitch, longitudinal pitch, and average fin thickness were 270.1 fins m^{-1} (82.3 fins $ft.^{-1}$), 0.040 m (0.131 ft.), 0.039 m (0.127ft.) and 0.0006 m (0.00196 ft.), respectively.

The Reynolds number used in correlating the data was based on the air velocity at the minimum flow area, and the hydraulic diameter. The ratio of the hydraulic diameter to the equivalent flow length, L (measured from the leading edge of the first tube row, to the leading edge of a hypothetical tube row that would follow the last tube row), is

$$\frac{d_H}{L} = 4\sigma\frac{A_{fr}}{A_o} \tag{2}$$

where

$$\sigma = \frac{A_{min}}{A_{fr}} = \frac{S_T - d_o - [(d_f - d_o)\delta S_F]}{S_T}$$

and

$$\frac{A_{fr}}{A_o} = \frac{S_T}{\pi\left[\frac{d_f^2 - d_o^2}{2} + d_f\delta\right]S_F + \pi(1-\delta S_F)d_o}.$$

Thus, correlations based on hydraulic diameter are independent of tube length and heat exchanger width.

It is customary to regard the total performance of condensing heat exchangers to be comprised of the sensible and latent energy portions. The approach here will be to regard the sensible heat transfer, and the combined simultaneous heat and mass transfer separately. The enthalpy difference will serve as the

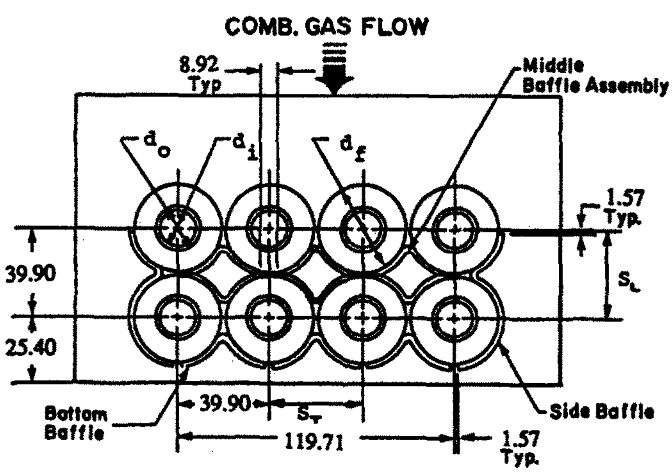

FIG. 1. Heat exchanger assembly.

driving potential for the combined heat and mass transfer, and the temperature difference will serve as the driving potential for the sensible heat transfer, in the usual sense.

Sensible heat transfer

The total sensible heat transfer of a multi-pass, finned-tube heat exchanger may be given by

$$Q = FU_o A_o \Delta T_{lm} \tag{3}$$

where U_o is the overall heat transfer coefficient, A_o the overall outside heat transfer area, and ΔT_{lm} the log-mean temperature difference for the flow geometry being studied. For this heat exchanger, under the conditions studied, the correction factor, F, was always unity. The log-mean temperature difference for a counterflow heat exchanger is given by

$$\Delta T_{lm} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\ln\left[\dfrac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}}\right]}. \tag{4}$$

In most experimental works the heat transfer and temperatures are measured, the area is known, and the overall heat transfer coefficient is determined from equations (3) and (4). In an attempt to generalize the results, so that they may be applied to other similar heat exchangers, the outside heat transfer coefficient is desired. The overall heat transfer coefficient is related to the outside (as well as inside) heat transfer coefficient (for steady state) by

$$U_0 = \frac{1}{\dfrac{A_o}{A_i}\left(\dfrac{1}{h_i} + F_i\right) + \dfrac{A_o}{A_i}\dfrac{r_i}{k_w}\ln\left(\dfrac{r_o}{r_i}\right) + \dfrac{1}{\eta_o}\left(\dfrac{1}{h_o} + F_o\right)} \tag{5}$$

where F_o and F_i are fouling factors, taken as zero for clean heat exchangers. The surface efficiency, η_o, is given by

$$\eta_o = 1 - \frac{A_f}{A_o}(1 - \eta_f) \tag{6}$$

η_f, is the fin efficiency, and represents the ratio of actual heat transferred to the maximum possible heat transfer.

For the straight fin of length l (Kays and London [1])

$$\eta_f = \frac{\tanh (Ml)}{Ml} \tag{7}$$

with

$$M^2 = \frac{h_o P}{kA}. \tag{8}$$

In this expression, h_0 is the heat transfer coefficient, P the perimeter, k the thermal conductivity, and A the cross-sectional area of the fin. Schmidt [19] demonstrated that a circular fin could be modeled as a straight fin, if the length, l, was replaced by

$$l_f = \tfrac{1}{2}(d_o - d_i)\left(1 + 0.35 \ln\left(\frac{d_o}{d_i}\right)\right). \tag{9}$$

Thus, from heat transfer measurements, U_o is determined using equation (3). A value of the outside heat transfer coefficient is guessed, and the fin efficiency is calculated using equations (7)–(9). With U_o and η_f, the outside heat transfer coefficient may be calculated using equations (5) and (6). To accomplish this, the inside heat transfer coefficient, h_i, is calculated using the correlation suggested by Nusselt [20] for developing turbulent flow

$$Nu = 0.036 Re^{0.8}\, Pr^{1/3}\left(\frac{d_i}{L}\right)^{0.055} = \frac{h_i d_i}{k} \tag{10}$$

where d_i is the tube diameter, and L the length. The water flow rate through the heat exchanger was always high enough to ensure turbulent flow, thus making the inside heat transfer coefficient as high as possible.

The 'guess' of the outside heat transfer coefficient is updated, and this process is iteratively employed until the error between successive guesses is sufficiently small. A value for the Colburn J-factor is determined from

$$J = \left(\frac{h_o}{Gc_p}\right) Pr^{2/3}. \tag{11}$$

Simultaneous heat and mass transfer

In a fashion analogous to the sensible heat transfer, the combined heat and mass transfer can be shown to be represented by [17, 21, 22]

$$Q = U_h A_o \Delta i_{lm} \tag{12}$$

where U_h represents the overall enthalpy transfer coefficient, and Δi_{lm} the log-mean enthalpy difference. For a counterflow heat exchanger [21, 22] it is given by

$$\Delta i_{lm} = \frac{(i_{h,i} - i_{c,o,st}) - (i_{h,o} - i_{c,i,st})}{\ln\left(\dfrac{i_{h,i} - i_{c,o,st}}{i_{h,o} - i_{c,i,st}}\right)} \tag{13}$$

where $i_{c,i,st}$ is the enthalpy of the hot gas (air–water mixture) were it to be saturated at the temperature $T_{c,i}$. Similarly, $i_{c,o,st}$ would be the saturation enthalpy at $T_{c,o}$. These enthalpies can be referred to as the *fictitious saturation enthalpies*, and are based on the assumption that $i = a + bT_w$ (Fig. 2 shows these temperatures and enthalpies schematically).

With this representation of the simultaneous heat and mass transfer, the overall enthalpy transfer coefficient is related to the outside enthalpy transfer coefficient, h_h, by

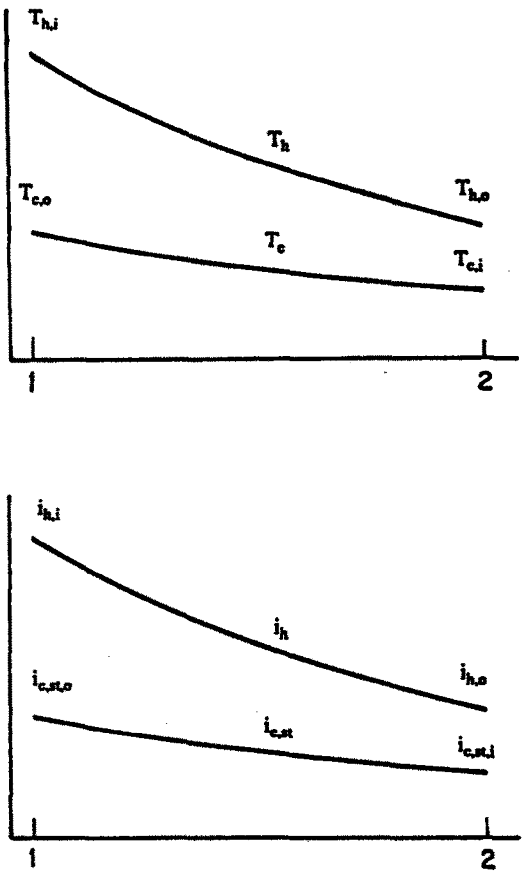

FIG. 2. Counterflow heat exchanger temperature/enthalpy diagram.

$$U_h = \frac{1}{b\dfrac{A_o}{A_i}\left(\dfrac{1}{\bar{h}_i}+F_i\right)+b\dfrac{A_o}{A_i}\dfrac{r_i}{k_w}\ln\left(\dfrac{r_o}{r_i}\right)+\dfrac{1}{\eta_o}\left(\dfrac{1}{\bar{h}_h}+F_o\right)} \quad (14)$$

where

$$b = \frac{di_{st}}{dT_w} \approx \frac{i_{c,o,st}-i_{c,i,st}}{T_{c,o}-T_{c,i}}. \quad (15)$$

Different approaches have been employed for determining the fin efficiency. McQuiston [23] suggested that the fin efficiency of an extended surface with combined heat and mass transfer is given by taking the M parameter in equation (7) to be

$$M^2 = \frac{h_o P}{kA}\left(1+\frac{ni_{fg}}{c_p}\right) \quad (16)$$

where

$$n = \frac{1}{2}\left[\left(\frac{\omega_{c,o,st}-\omega_{h,i}}{T_{c,o}-T_{h,i}}\right)+\left(\frac{\omega_{c,i,st}-\omega_{h,o}}{T_{c,i}-T_{h,o}}\right)\right]. \quad (17)$$

In deriving equations (16) and (17) the saturation state along the fin was assumed to vary with the fin temperature. While some variation may occur under some circumstances, e.g. for highly curved condensate surfaces, for the condensation of liquid metals (where the interface resistance comes into play), for non-uniform species concentrations, or for very high vapor velocities (where a free stream pressure gradient may be important); for the condensation of water, the saturation state along a fin would be essentially constant.

Burmeister [24] showed that for one specific configuration, the approximation of a single (average) heat transfer coefficient could be appropriate for condensation on fins. This alternate approach introduces some inherent errors, and the semi-empirical approach of McQuiston may be fortuitously more accurate. However, McQuiston's approach complicates the data reduction, interpretation, and computation. Furthermore, in the present study, differences in η_f computed by both methods were typically of the order of 3%, hence, the use of an average heat transfer coefficient in lieu of the McQuiston approximation is justified.

While the differences between these alternate methods of calculating fin efficiency had a minor influence on the results, it was found that significant differences in data representation resulted if property data were evaluated at different temperatures. Bump [25] suggested the following temperatures to be representative of the average hot and cold fluid temperatures in a counterflow heat exchanger

$$T_{h,ave} = T_{h,i} - \left(\frac{(T_{h,i}-T_{h,o})(T_{h,i}-T_{c,o})}{(T_{h,i}-T_{h,o})-(T_{c,o}-T_{c,i})}\right) + \left(\frac{(T_{h,i}-T_{h,o})}{\ln\left(\dfrac{(T_{h,i}-T_{c,o})}{(T_{h,o}-T_{c,i})}\right)}\right) \quad (18)$$

and

$$T_{c,ave} = T_{c,o} - \left(\frac{(T_{c,o}-T_{c,i})(T_{h,i}-T_{c,o})}{(T_{h,i}-T_{h,o})-(T_{c,o}-T_{c,i})}\right) + \left(\frac{(T_{c,o}-T_{c,i})}{\ln\left(\dfrac{(T_{h,i}-T_{c,o})}{(T_{h,o}-T_{c,i})}\right)}\right). \quad (19)$$

All data were reduced employing equations (18) and (19) for the temperatures at which property data were determined. The only exception being that the latent heat of condensation was evaluated at the mean dew point temperature of the condensing water.

TEST RESULTS

Experiments were conducted with the heat exchanger fully dry, i.e. no condensation, or fully wet, i.e. the entire heat exchanger below the dew point. 'Fully wet' is a qualified descriptor because the surface

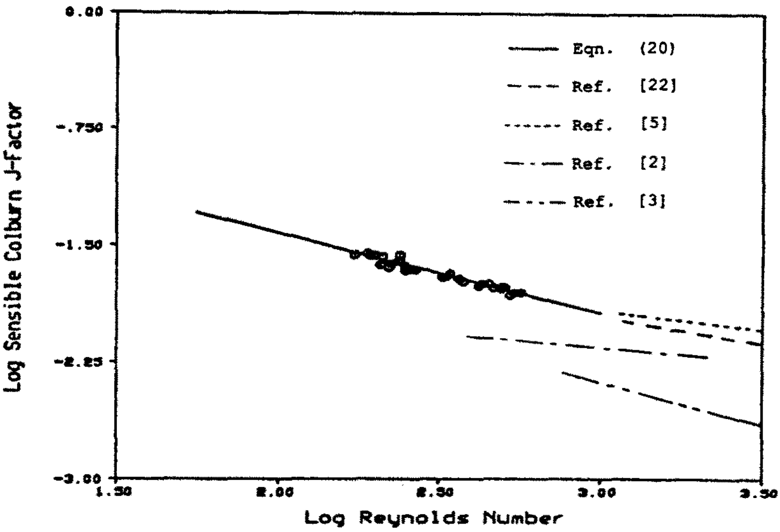

FIG. 3. Sensible Colburn J-factor for non-condensing case (heat transfer only).

may or may not be completely covered, depending on the mode of condensation. The results of these experiments are presented below. The data presented were reduced as outlined in the preceding section. The data reduction implementation has been improved over that of Idem *et al.* [3] through improved property evaluation, average temperature representation, and fin efficiency calculation.

Fully dry heat exchanger

Experimental results for the dry (baffled) heat exchanger are presented in Fig. 3. This figure shows the sensible Colburn J-factor vs Re. A least squares curve fit to the data of Fig. 3, yields the following correlation:

$$J = 0.396\, Re^{-0.510} \tag{20}$$

which is shown along with Elmahdy [22] for an unbaffled configuration, Idem *et al.* [3] for an unbaffled configuration, Idem *et al.* [2] for the 'T-baffled' heat exchanger, and Idem and Goldschmidt [5] for a heat exchanger typical of the one used in this report, but at higher Reynolds numbers.

Fully wet heat exchanger

Data for the case where the heat exchanger was everywhere below the dew point are shown in Figs. 4 and 5, along with the correlations due to Idem and Goldschmidt [5] which are appropriate at higher Re. The following correlations fit the data:

$$J = 0.046\, Re^{-0.206} \tag{21}$$

and

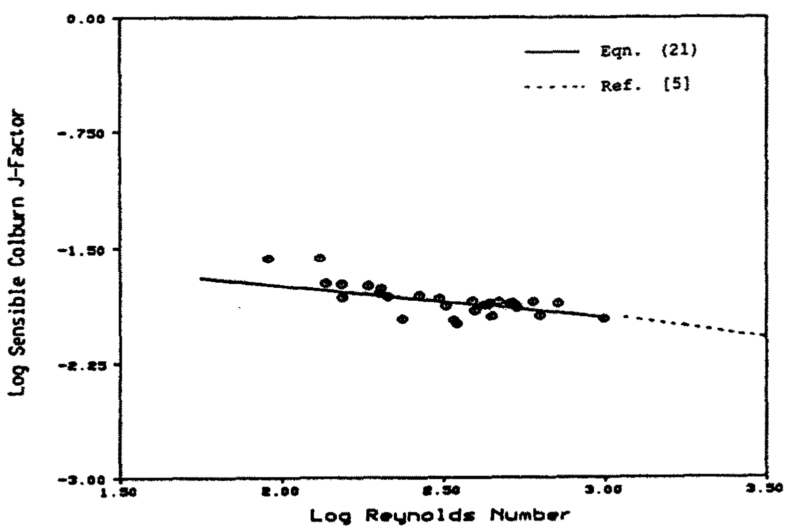

FIG. 4. Sensible Colburn J-factor for condensing case (simultaneous heat and mass transfer).

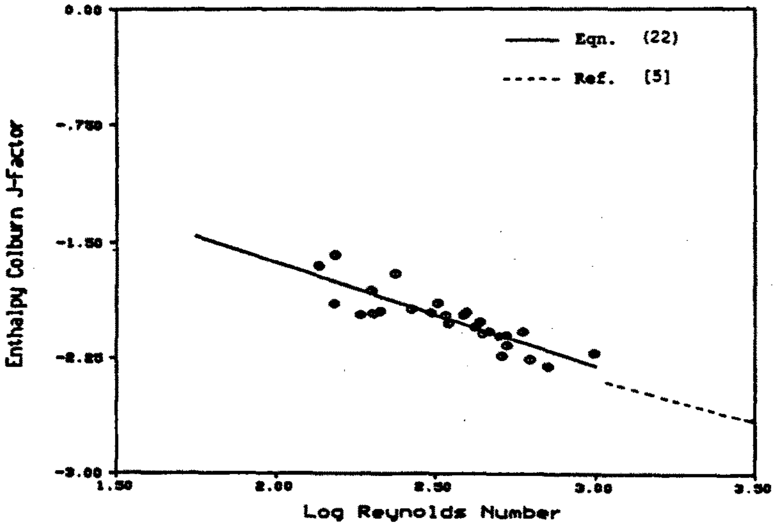

FIG. 5. Enthalpy transfer Colburn J-factor for condensing case (simultaneous heat and mass transfer).

$$J_i = 0.515\,Re^{-0.670}. \tag{22}$$

The correlations given here characterize the heat and mass transfer performance of the integral high-fin, condensing heat exchanger at low Reynolds numbers.

DISCUSSION

The trends in the data reported here are consistent with those observed (at higher Reynolds numbers) by Kays and London [1], Elmahdy [22], Idem *et al.* [2], and Idem *et al.* [3]. The slope of the J-factor vs Re line is consistent with those earlier investigations, and the magnitude is of the correct order. However, one particularly interesting effect is now notable at these lower Reynolds numbers.

In Fig. 6 it is evident that if the sensible heat transfer data from the non-condensing and condensing cases are compared, there is a 'cross-over' at a Reynolds number of about 1200. Although this effect is not obvious in the data of Idem *et al.* [3], it is suggested if one extrapolates those correlations to lower Reynolds numbers. A similar behavior was also reported by McQuiston [26], where a cross-over was observed as fin spacing was reduced.

Previous investigators postulated that the observed enhancement of sensible Colburn J-factor under wet over that of dry conditions (observed at Reynolds numbers above the values now reported) was due to the condensate on the heat exchanger disrupting the flow and acting as a turbulator. An attempt to duplicate this effect by attaching gypsum to a dry heat

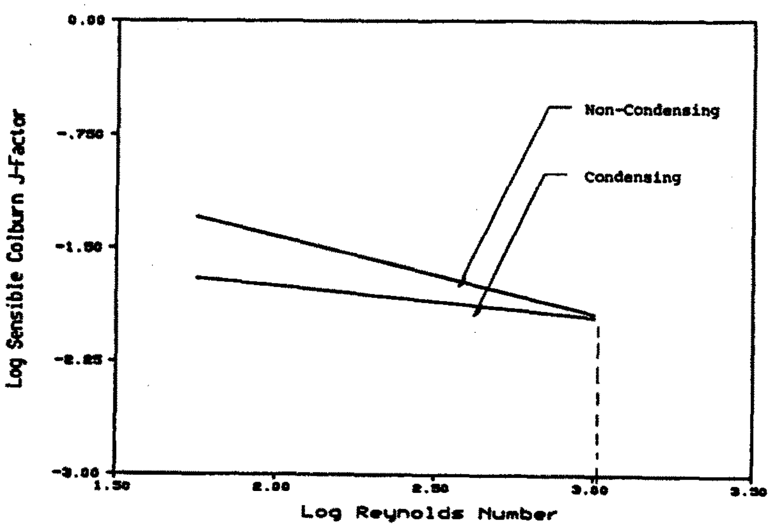

FIG. 6. Comparison of sensible Colburn J-factors dry to wet.

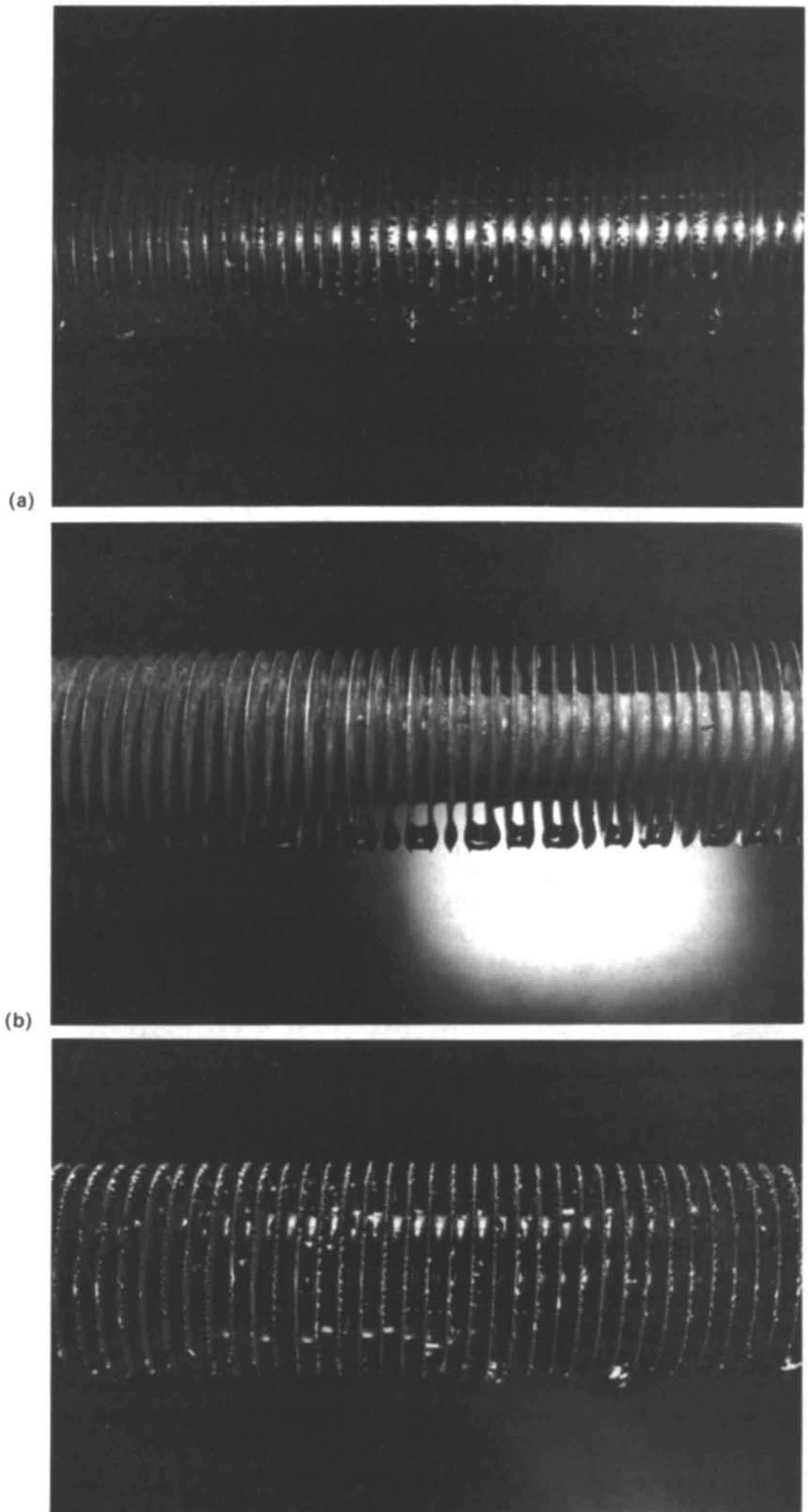

FIG. 7. Condensate retention on integral high-finned tubes: (a) as extruded; (b) previously fired (subjected to combustion chamber environment); (c) coated with Teflon-like protective material.

exchanger was reported by Bettanini [27]. However, as one may expect the physics at low Reynolds numbers are dominated by other mechanisms. One plausible hypothesis is that the cross-over is due to deleterious effects of condensate retention. Retained condensate between the fins, and around the baffles, reduces the effective area for sensible heat transfer. Alternately, this could be envisioned as an equivalent fouling resistance due to the retained condensate.

At low Reynolds numbers the vapor shear on the condensate, and pressure drop through the heat exchanger, will be reduced. As such, it is much more likely that surface tension forces will be capable of retaining condensate on the heat exchanger. Likewise, for close fin spacing surface tension forces are more important. Although McQuiston [26] reported no observed retention, this may have been due to his inability to see between the fins of his experimental heat exchanger. Visual studies undertaken by the current authors show that the integral high-finned tubes, such as those used in this study, are capable of retaining condensate under varying rates of condensation, and with varying surface conditions. Retention on three differently prepared surfaces is shown in Fig. 7. Figure 7(a) corresponds to an as manufactured surface, Fig. 7(b) oxidized (previously installed in a combustion chamber), and Fig. 7(c) with a corrosion resistant coating.

From the visual studies some very crude estimates can be made which buttress the mechanism put forth above. Consider a model of retained condensate as shown in Fig. 8. The weight force on this retained condensate would be

$$F_w = mg \approx \rho_l \alpha\beta\gamma\pi g. \tag{23}$$

The retaining surface tension force on four lengths of 2α could be approximated as

$$F_s = 8\alpha\sigma_w. \tag{24}$$

A drag force on a cross-sectional area given by $2\alpha\gamma$, due to the flowing vapor, could be crudely approximated by

$$F_d = C_D \rho_v \alpha\gamma V^2. \tag{25}$$

From equations (23)–(25) the approximate velocity required to shed this retained condensate can be found. Using $C_D = 1.0$, the properties of water, and assuming α between 0.508 cm (0.20 in.) and 0.635 cm (0.25 in.), β between 0.381 cm (0.15 in.) and 0.508 cm (0.20 in.), and γ equal to the fin spacing of 0.0368 cm (0.145 in.) gives velocities between 6.54 m s^{-1} (21.5 ft. s^{-1}) and 1.85 m s^{-1} (6.07 ft. s^{-1}). This would imply changes in retained condensate for $600 \leqslant Re \leqslant 1200$, possibly explaining the 'cross-over' value of 1200, and lending support to the possibility that retained condensate is responsible for the observed sensible Colburn J-factor behavior.

CONCLUSIONS

Heat and mass transfer have been characterized for a baffled, integral high-finned tube, two-pass, counterflow heat exchanger operating at low and moderate Reynolds numbers. Correlations are given in equations (20)–(22).

The sensible Colburn J-factor for condensing heat exchangers is higher than that for the non-condensing case, at high Re (for this particular case, for $Re \gtrsim 1200$). However, at low Re the dry sensible Colburn J-factor (i.e. without condensation) is higher. Condensate retention is proposed as a possible mechanism for this behavior. For the configuration studied, condensate retention was observed to occur over varied condensate rates, and surface preparations.

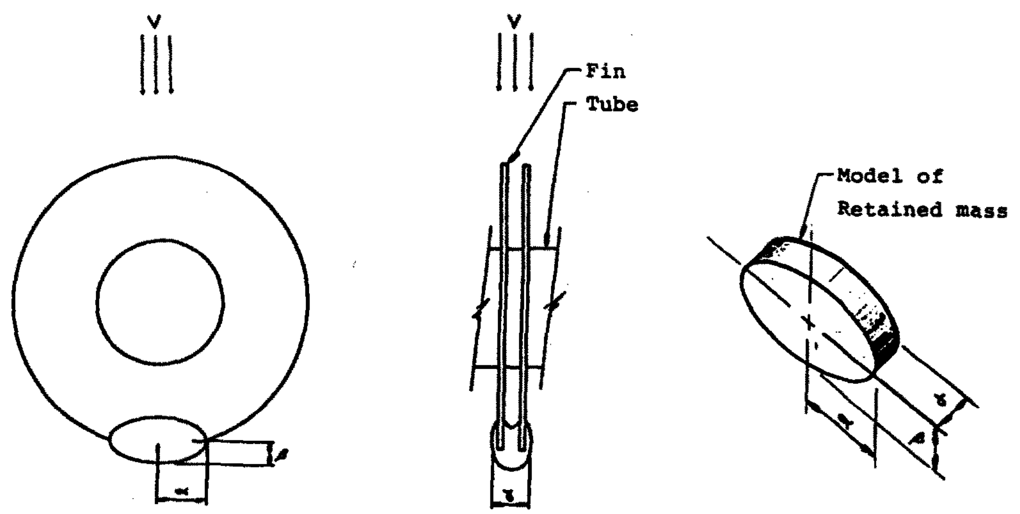

FIG. 8. Model of retained condensate.

Although the low Reynolds number sensible Colburn J-factor is deleteriously affected by condensate retention, the overall heat transfer rates are much higher when condensation occurs. For example, at a Reynolds number of about 1200, the total heat transfer with condensation may be as much as six times greater than that realized with a dry heat exchanger. Undoubtedly, research in this area will continue. Additional data, as well as an example in which the concepts presented are applied in an actual system are provided in ref. [28].

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MESURES DE TRANSFERT DE CHALEUR ET DE MASSE A PETIT NOMBRE DE REYNOLDS POUR UN ECHANGEUR THERMIQUE AVEC CONDENSATION, A CONTRE COURANT, TUBES AILETES ET BAFFLES

Résumé—On caractérise les performances en transfert de chaleur et de masse d'un échangeur à condensation. L'étude concerne un échangeur à contact indirect, globalement à contre-courant, avec tube aileté en cuivre et avec baffles. Les résultats de transfert de chaleur et de masse sont corrélés et on fait une comparaison avec d'autres études. On observe un comportment intéressant du transfert de chaleur sensible aux faibles nombres de Reynolds dans les cas de condensation et de non condensation, et on présente une explication physique de ces observations.

MESSUNG DES WÄRME- UND STOFFÜBERGANGS BEI KLEINER REYNOLDS-ZAHL IN EINEM GEGENSTROM-WÄRMEÜBERTRAGER MIT KONDENSATION

Zusammenfassung—Die Wärme- und Stoffübertragung in einem Kondensations-Wärmeaustauscher wird beschrieben. Die Untersuchung wurde an einem Rippenrohr-Wärmeaustauscher aus Kupfer mit Umlenkblechen bei indirektem Kontakt, direktem Austausch und Gegenstrom durchgeführt. Die Meßwerte der Wärme- und Stoffübertragung werden korreliert und mit anderen Untersuchungen verglichen. Es werden interessante Phänomene und Gegensätzlichkeiten beim Wärmeübergang mit kleiner Reynolds-Zahl für Fälle mit und ohne Kondensation beobachtet. Ein Mechanismus zur Erklärung der Physik dieser Beobachtungen wird vorgeschlagen.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ПРОЦЕССОВ ТЕПЛО- И МАССОПЕРЕНОСА ПРИ МАЛЫХ ЗНАЧЕНИЯХ ЧИСЛА РЕЙНОЛЬДСА В ПРОТИВОТОЧНОМ КОНДЕНСАТОРЕ-ТЕПЛООБМЕННИКЕ С ОТРАЖАТЕЛЯМИ И ОРЕБРЕННЫМИ ТРУБАМИ

Аннотация—Представлены тепло- и массообменные характеристики конденсатора-теплообменника. Исследование проводилось на медном противоточном рекуперативном теплообменнике с отражателями и оребренными трубами. Получены корреляции данных тепло- и массообмена и проведено сравнение с результатами других исследований. Отмечены интересные различия протекания процессов теплообмена, чувствительного к значениям числа Рейнольдса в конденсирующихся и неконденсирующихся газах, а также высказано предположение о механизме, лежащем в основе наблюдаемых процессов.