Colburn model. If the nonlinearities are pronounced, the three point method of Roetzel as presented in this paper is not always a better choice compared to the Colburn method. The surface area predicted by the Roetzel method can be considerably higher or lower compared to the exact numerical results.

In most shell-and-tube heat exchangers, the shell-side flow undergoes changes from crossflow to parallel-flow between and across baffle plates. Thus, the shell-side coefficient undergoes cyclic and not necessarily monotonic changes through the exchanger. In this regard, data set #5 was selected to demonstrate this influence on the surface area requirement. From the review of the results for a single data set #5 in Table 3, no general conclusions regarding which method of averaging U is the best can be drawn. However, it appears that the surface area predicted is quite accurate using the Colburn method, as was the case with data set #3. However, further study is needed to investigate the influence of the cyclic behavior of h or U on the surface area requirement for many multipass shell-and-tube heat exchangers.

Based on the findings of this work, it appears that none of the approximate methods considered here will accurately predict the exchanger surface area requirement when the variation of the heat transfer coefficient on one of the two-fluid sides is highly nonlinear and the corresponding thermal resistance is controlling. If the nature of variation in h is unknown, as in practical applications, the reliability of the approximate methods is even more questionable. The best approach is to conduct the numerical integration to take into consideration the actual variation of the heat transfer coefficient. In many applications, the heat transfer coefficient on one fluid side may not be controlling (i.e., the heat transfer coefficients on both sides may be of the same order of magnitude). In this case, the conclusions derived from the example of this paper are not necessarily applicable. Hence, the best solution is to conduct exact numerical integration to take into account the variations in the heat transfer coefficient on one or both fluid sides.

It should be emphasized that the most published correlations are not accurate to better than 10–15 percent, and also the variation in fluid properties and fouling factors may introduce additional uncertainties in h or U. Hence, even the "exact" numerical method may not have real validity. The approximate methods of overall heat transfer coefficient averaging may or may not have poor accuracy. In light of many commercial computer programs available for the design and analysis of heat exchangers that use some method of averaging U values, the results presented in this paper provide some guidelines on the averaging methods, particularly which ones should be avoided. However, unfortunately, none of the approximate methods came out superior to others in terms of accuracy and reliability.

Conclusions

The methods for incorporating the influence of nonuniformity in the overall heat transfer coefficient in conventional design procedures have been presented. Various definitions of mean overall heat transfer coefficients have been introduced (see Table 1) depending on the pertinent nonuniformity effect. The influences of the length effect (developing thermal boundary layer influence), the temperature effect (changes in thermophysical properties due to fluid temperature variations), and the combined effect have been included in the analysis by defining an area average mean overall heat transfer coefficient, a temperature average mean overall heat transfer coefficient and their combination. A step-by-step procedure is presented for evaluation of the mean overall heat transfer coefficient.

Five different methods of averaging U were compared to determine how they rank in accuracy in predicting exchanger surface area requirement for an example where the heat transfer coefficient on one fluid side was controlling; hence, the variations in h corresponded to the variations in U. For a significant nonlinear variation in h, none of the five methods yielded accurate results. The only plausible method in such a case is the numerical approach. The overall heat transfer coefficients using various reference temperatures in most cases underestimated the heat transfer surface and *should be avoided.*

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Effect of Fouling on Temperature Measurement Error and a Solution

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The objective of the present study was to investigate the effect of fouling on the accuracy of temperature measurement. When a thin scale layer is deposited on a temperature-measuring probe, the temperature reading can be in error by several degrees in spite of ±0.1 K resolution of the probe. This erroneous temperature reading can pose a serious problem in the evaluation of heat exchanger performance and in the operation of an automated process control system.

Nomenclature

A = heat transfer surface area

C_p = heat capacity

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D_H = hydraulic diameter
D_s = inside shell diameter
$d_{t,o}$ = outside tube diameter
Q = heat transfer rate
R_f = fouling factor
$T_{c,in}$ = inlet temperature of cold water
$T_{c,out}$ = outlet temperature of cold water
ΔT_{LMTD} = log-mean-temperature-difference
U = overall heat transfer coefficient based on outside tube diameter

Introduction

It is well known that fouling causes performance degradation of heat exchangers (Taborek et al., 1972; Suitor et al., 1977; Bott, 1995). One of the most common techniques to monitor the performance degradation is to measure the inlet and outlet temperatures of cold and hot streams in the heat exchanger. However, the effect of thermocouple or thermister probe fouling on temperature measurements has not been carefully considered (ASTM, 1981; Kerlin and Shepard, 1982).

When hard water is used as a cooling stream in a heat exchanger, the water becomes locally supersaturated as it is heated inside the heat exchanger (Cowan and Weintritt, 1976). Thus, precipitation fouling occurs inside the heat exchanger.

Since the temperature of the water leaving the heat exchanger is substantially greater than the inlet water temperature, a temperature-measuring probe located immediately after the heat exchanger can experience a serious fouling problem. As water is heated and passes through the heat exchanger, a number of small particles of submicron to micron size are formed, as dissolved mineral ions precipitate due to temperature changes and flow disturbances inside the heat exchanger (Cowan and Weintritt, 1976; Fan, 1997). Thus, both precipitation and particulate fouling can occur on the surface of the temperature-measuring probe.

When one considers the flow around a temperature-measuring probe outside a heat exchanger, it is often an impingement type of flow. Thus, the surface of the probe becomes an ideal site for scale build-up. The objective of the present study was to investigate the effect of fouling on the use of a temperature-measuring probe. In particular, the fouling of thermocouple probes will be examined in the context of performance losses of a heat exchanger due to fouling.

Experimental Method

Figure 1 schematically shows the test facility, which consists of two reservoir tanks, two pumps, two flow meters, the primary heat transfer test section made of a concentric tube heat exchanger, and a plate-and-frame heat exchanger.

The primary heat transfer test section is made of two concentric copper tubes, i.e., a single-tube counterflow heat exchanger. The shell diameter (ID) was D_s = 0.01684 m and the tube diameter (OD) was $d_{t,o}$ = 0.01275 m, leading to a hydraulic diameter of D_H = 0.00409 m. The diameter of both the inlet and outlet connecting tubes was 0.0109 m. The location of the thermocouple probe relative to the outlet connecting tube is shown in Fig. 1. Of note is that the high velocity at the connecting tube (i.e., Re = 5710) ensured a good mixing for accurate outlet temperature measurement.

The thermocouple used in the present study was Omega Model TMTSS-125G-6 (grounded copper-constantan T type). The thermocouple junction was encased in a 304 stainless steel sheath with a stem diameter of 0.0032 m. Calibration was carried out at 273 and 373 K, confirming the manufacturer's claim of the resolution of ±0.1 K. Hot water discharged from the heat exchanger impinged on the tip of the thermocouple probe longitudinally. The temperature of the hot stream entering the concentric heat exchanger was maintained at approximately 367 K using a plate-and-frame heat exchanger where high-pressure steam provided by the Philadelphia city steam network was used.

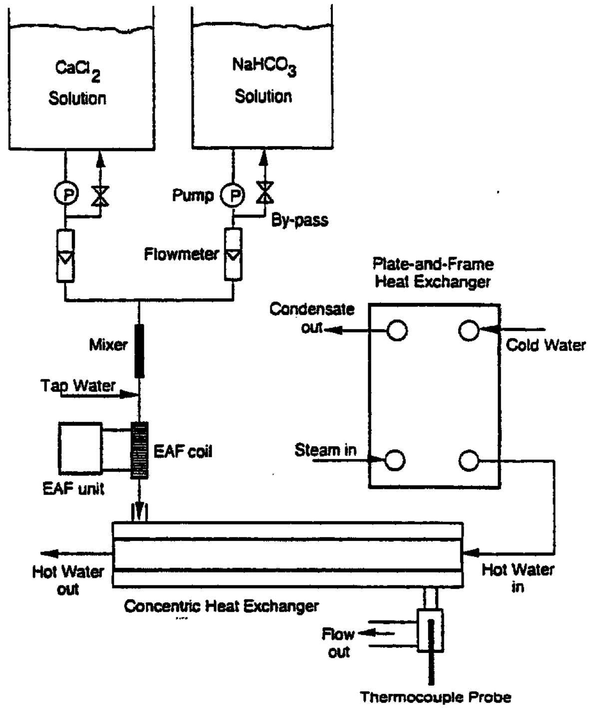

Fig. 1 Schematic diagram of the present test setup with a concentric tube heat exchanger as the primary test section. EAF coil for electronic anti-fouling treatment is used in a feed pipe to the heat exchanger.

Since the hardness of tap water available in Philadelphia is approximately 150 mg/L as $CaCO_3$, it is not suitable for fouling experiments. Therefore, hard water was prepared in our laboratory. The test solution was prepared by adding 0.01 M calcium chloride ($CaCl_2$) and 0.02 M sodium bicarbonate ($NaHCO_3$) to tap water such that the hardness of the test solution was equivalent to 1000 mg/L as $CaCO_3$. The resulting hard water caused severe scaling inside a pump, and one pump was lost on nearly every test due to the pump scaling. In order to avoid scaling in the pump, the calcium chloride and sodium bicarbonate solutions were prepared in two separate reservoir tanks and mixed using a static mixer as shown in Fig. 1.

The inlet temperature of the hard water was maintained at 302 K, whereas the outlet temperature was maintained at approximately 327 K. A flow rate of 2.7×10^{-5} m^3/s was used, resulting in a Reynolds number of 1870 inside the heat exchanger (i.e., in the shell side) and a Reynolds number of 5710 in the outlet connecting tube where the temperature-measuring probe was located. It is of note that the viscosity of the hard water was evaluated at a shell-side average temperature of 316 K and an outlet average temperature of 323 K in Reynolds number calculations. The velocities in the shell side and outlet connecting tube were 0.28 m/s and 0.29 m/s, respectively.

The overall heat transfer coefficient, U, was calculated using the log-mean-temperature-difference (LMTD) (Incropera and DeWitt, 1996), which was obtained from the four temperatures measured at both the inlet and outlet of the cold and hot streams. Fouling factor, R_f, was calculated using the usual definition:

$$Q = UA\Delta T_{LMTD} = \dot{m}_c C_p (T_{c,out} - T_{c,in}) \quad (1)$$

$$U = \frac{\dot{m}_c C_p (T_{c,out} - T_{c,in})}{A\Delta T_{LMTD}} \quad (2)$$

and

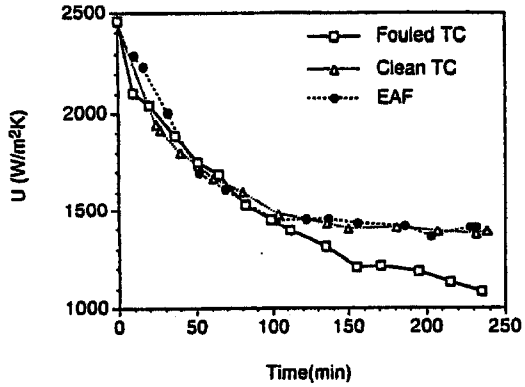

Fig. 2 Overall heat transfer coefficient versus time at a flow rate of 2.7 × 10^{-5} m^3/s for three different cases: with fouled thermocouple probe, with clean thermocouple probe, and with electronic anti-fouling (EAF) device

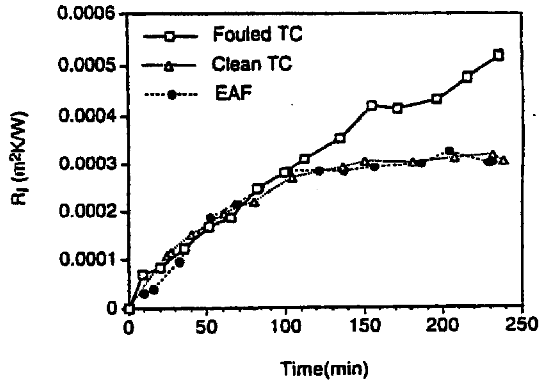

Fig. 3 Fouling factor versus time at a flow rate of 2.7 × 10^{-5} m^3/s for three different cases: with fouled thermocouple probe, with clean thermocouple probe, and with electronic anti-fouling (EAF) device

$$R_f = \frac{1}{U(t)} - \frac{1}{U_{\text{clean}}} \tag{3}$$

where U_{clean} is the overall heat transfer coefficient corresponding to the clean state, i.e., at $t = 0$. The errors estimated from the uncertainty analysis for Q, U, R_f, and A (heat transfer surface area) were 2.5, 3.0, 8.0, and 1.1 percent, respectively. In all test cases shown in the present paper, the heat balance between the cold and hot streams was within 5 percent.

Results and Discussion

Figure 2 presents the changes in the overall heat transfer coefficient, U, as a function of time for three different cases. The U value corresponding to the clean initial state was measured with tap water at a flow rate of 2.7 × 10^{-5} m^3/s, resulting in a value of 2460 W/m^2K. When the concentric heat exchanger was clean, the outlet temperature of the cold stream was 331 ± 0.1 K at a flow rate of 2.7 × 10^{-5} m^3/s.

As the heat exchanger continued to foul due to the use of the hard water, the outlet temperature measured with a "fouled" thermocouple probe decreased to 317.5 ± 0.1 K at the end of the 4 hour test, resulting in a U value of 1100 W/m^2K.

When the test was repeated and the thermocouple probe was manually cleaned every time we measured the outlet temperature, it decreased only to 320.2 ± 0.1 K at the end of the 4 hour test, rendering a U value of 1410 W/m^2K. The results shown in Fig. 2 clearly depict the effect of the thermocouple probe fouling on the estimation of the performance degradation of a heat exchanger. In other words, when one monitors a heat exchanger fouling with a fouled temperature-measuring probe, one significantly overestimates the fouling in the heat exchanger.

A fouled temperature-measuring probe gives a substantially lower temperature reading than a clean one. The error in the outlet temperature measurement depends on the magnitude of fouling on the surface of the probe. It can also cause an automated process control system to respond erroneously in such a way that further fouling can take place in both the heat exchanger and the probe.

Another identical test was carried out with an electronic anti-fouling (EAF) device. The outlet temperature was measured without cleaning the probe in this case. At the end of the 4 hour test the outlet temperature and the U value obtained with the EAF device were found to be exactly identical to the values obtained with the manually cleaned probe, as shown in Fig. 2.

Since the operating principle of the electronic anti-fouling device was introduced elsewhere (Fan, 1997; Fan and Cho, 1997a, 1997b; Cho et al., 1997a, 1997b), it will be described here only briefly. The EAF technology uses a square-wave current signal to create time-varying magnetic fields in a solenoid wrapped around a pipe. Subsequently, the time-varying magnetic field creates an induced electric field inside the pipe, a phenomenon that can be described by Faraday's law. The induced electric field, which oscillates with time, provides the necessary molecular agitation to charged mineral ions such that dissolved mineral ions such as calcium and bicarbonate collide and precipitate. Subsequently, the precipitation fouling is prevented on the surface of a temperature probe or heat exchanger.

Figure 3 shows the fouling factor calculated from the overall heat transfer coefficient as a function of time. The fouling factor obtained with the fouled temperature probe includes fouling in both the heat exchanger and temperature probe. The actual fouling that took place in the heat exchanger had to be significantly less, as demonstrated by the comparison with the case obtained with the manually cleaned temperature probe. In other words, fouling factors can be greatly overestimated if a temperature probe is fouled.

One may ask why fouling should affect temperature measurement under a steady-state condition. In order to illustrate the point, Fig. 4 shows a sketch of a typical thermocouple mounting. Since the fluid temperature is often substantially greater than the ambient temperature, heat flows from inside to outside through the probe sheath. When there is no scale deposit on the probe, the thermocouple junction temperature is equal to the fluid temperature. However, as the probe becomes surrounded by a scale layer, a temperature gradient across the scale layer due to heat flow is introduced. In other words, the junction temperature will be consistently lower than the fluid temperature. The temperature error due to the scale layer will depend primarily on the thickness and thermal conductivity of scale layer. In addition, for a heat exchanger located outdoors, the outside air temperature decreases in the winter, increasing the temperature difference between fluid and the outside air. Hence, the temperature error due to the scale layer will be larger in the winter than in the summer.

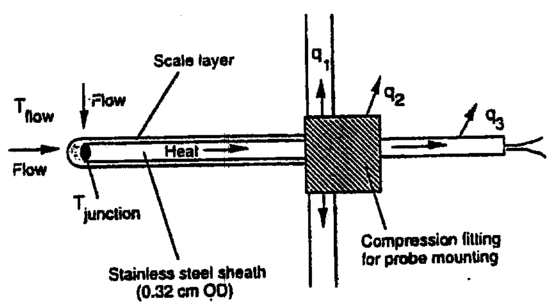

Fig. 4 Sketch of thermocouple probe mounting. Arrows indicate the direction of heat flow. q_1 = conduction heat loss throughout tube wall, q_2 = convective heat loss from fitting, and q_3 = convective heat loss from probe stem.

Conclusions

The present study was conducted in order to investigate the effect of fouling of a temperature-measuring probe on the accuracy of temperature measurement. When the temperature probe is fouled, the temperature reading can be in error by several degrees. Subsequently, this erroneous temperature measurement can overestimate the performance degradation of a heat exchanger caused by fouling.

A new electronic anti-fouling (EAF) device was found to prevent the fouling in a temperature probe. With the EAF device, the temperature measurement error caused by the probe-fouling can be avoided.

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Application of an Equivalent Single-Tube Model for Predicting the Frequency-Response Characteristics of Multitube Two-Phase Condensing Flow Systems With Thermal and Flow Distribution Asymmetry

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Introduction

This paper is concerned with multiple, in-tube condensing flow systems involving complete condensation, the transient characteristics of which are important in a broad spectrum of energy transport and conversion processes. Sufficiently large flow excursions or oscillations could substantially affect the performance and control of the processes taking place, cause damage to mechanical equipment and components, and endanger the safety of such systems.

Most practical applications of in-tube condensation involve multiple-tube geometries, consisting of parallel channels connected to common headers. To the best knowledge of the authors, there do not appear to be any theoretical models or experimental data in the archival literature pertaining to the frequency-response characteristics of multitube condensing flow systems. Therefore, the model presented in this paper represents the results of merging a theoretical model capable of predicting the frequency-response characteristics of single-tube condensing flow systems (Bhatt and Wedekind, 1980) with a model capable of predicting the transient behavior of multitube systems (Wedekind and Bhatt, 1989).

Equivalent Single-Tube Model

Wedekind and Bhatt (1989) demonstrated a method of approximating an n-tube condensing flow system with an Equivalent Single-Tube Model (ESTM), based on the System Mean Void Fraction (SMVF) Model developed earlier. The concept of the ESTM is to obtain an equivalent single-tube system time constant, τ_s, which, in its generalized form for an n-tube system, is a weighted average of the time constants for each of the individual tubes. In the present model, the flow distribution and heat flux for each individual tube are allowed to be different; however, the tubes are of identical geometry. Thermodynamic properties of the two-phase mixture are assumed to be the same in each tube, and evaluated at a mean condensing pressure. The spatially averaged heat flux for each tube is assumed to be time-invariant.[4] Also, the inlet flow quality entering each tube is assumed to be the same and equal to unity; $x_{i,1} = x_{i,j} = 1$. Using the ESTM, based on the SMVF model, the frequency-response characteristics of the outlet liquid flow rate for an n-tube condensing flow system can be approximated utilizing the results from single-tube studies carried out earlier (Bhatt and Wedekind, 1980). Thus, the gain and phase shift characteristics, G_m and Φ_m, respectively, can be modeled in terms of the liquid-to-vapor density ratio, (ρ/ρ'), the equivalent system time constant, τ_s, and the angular frequency, ω; thus,

$$G_m = \left\{\frac{1 + (\rho/\rho')^2(\omega\tau_s)^2}{[1 + (\omega\tau_s)^2]}\right\}^{1/2} \quad (1)$$

$$\Phi_m = \tan^{-1}\left\{\frac{[(\rho/\rho') - 1](\omega\tau_s)}{1 + (\rho/\rho')(\omega\tau_s)^2}\right\} \quad (2)$$

where

$$\tau_s = \sum_{j=1}^{n} \gamma_j\tau_j = \tau_1 \sum_{j=1}^{n} \gamma_j\beta_j \quad (3)$$

The time constant for each individual tube, τ_j, involves the vapor density, ρ', the heat of vaporization, $(h' - h)$, and the spatially averaged heat flux, $\bar{f}_{q,j}$. It also involves the tube cross-sectional area, $A_{t,j}$, periphery, P_j, and the system mean void fraction, $\bar{\alpha}$; thus for tube 1:

$$\tau_1 = \frac{\rho'(h' - h)\bar{\alpha}A_{t,1}}{\bar{f}_{q,1}P_1} \quad (4)$$

where

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[4] The water flow rate was sufficiently high such that the heat transfer coefficient on the water side was greater than that on the refrigerant side; thus, the wall temperature of the copper tubing would be essentially constant. The small transients in the condensing temperature, due to small transients in the condensing pressure, would result in a near-time-invariant heat flux.