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Performance evaluation of convective heat transfer enhancement devices using exergy analysis

R. C. PRASAD and JIHUA SHEN

Department of Engineering, University of New Brunswick, P.O. Box 5050, Saint John, N.B., Canada, E2L 4L5

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Abstract—A new criterion for evaluating the effectiveness of a passive heat transfer augmentation device is proposed. This method is based on exergy analysis and can be applied to forced convection heat transfer systems, such as fluid to fluid heat exchangers. Use of heat transfer augmentation devices reduce the exergy destruction due to heat transfer across a finite temperature difference. However, the exergy destruction due to fluid flow friction increases. The net exergy destruction resulting from the above effects, thus, forms the evaluation criterion in the present work. Application of this technique to a tubular heat exchanger with wire-coil insert is presented.

INTRODUCTION

HEAT TRANSFER enhancement devices are commonly employed to improve the performance of an existing heat exchanger or to reduce the size and cost of a proposed heat exchanger. The enhancement technique improves the convective heat transfer coefficient and results in an increased rate of heat transfer per unit area. Various types of such devices have been developed and tested in the past [1, 2]. Several criteria for evaluating the effectiveness of these devices have been proposed. The method proposed by Bergles *et al.* [3] compares the performance of augmented surface heat exchanger to meet a defined objective, such as maximizing heat load or reducing surface area. The method, however, does not allow the assessment of two or more objectives simultaneously. This method does not take into account the conservation of energy. Bejan and co-workers [4–7] proposed an evaluation technique based on the second law of thermodynamics. Irreversibility or entropy generation associated with an augmentation device is used as an evaluating criterion. These methods do not include the effect of variation in fluid temperature similar to that present in tubular heat exchangers. Perez-Blanco [8] and Nag and Mukherjee [9] modified Bejan's entropy generation criterion by including the fluid temperature variation along heat transfer passage.

In the present work, an evaluation method based on exergy analysis is proposed. Using the principles of the first and second laws of thermodynamics, the exergy destruction, i.e. loss of availability, is evaluated in a non-dimensional form. The analysis includes the effect of fluid temperature variation along the length of a tubular heat exchanger. The proposed evaluation technique has been applied to a tubular heat exchanger using a wire-coil insert as an enhancement device.

EXERGY ANALYSIS

A tubular heat exchange system in which heat is transferred across a separating wall from/to a fluid stream flowing through the tube is considered here. The tube wall is assumed to be at a uniform temperature T_w which is maintained by an imaginary large heat sink/source around the tube. Fluid properties are assumed constant. A schematic diagram of the tubular heat exchanger system is shown in Fig. 1.

The specific flow availability (or flow exergy) is defined by

$$\psi = h - h_0 - T_0(s - s_0). \tag{1}$$

If $\psi = f(T, P)$, then $d\psi$ can be expressed as:

$$d\psi = \left(\frac{d\psi}{\partial T}\right)_p dT + \left(\frac{\partial \psi}{\partial p}\right)_T dp. \tag{2}$$

Using the definition of entropy, specific heat and the first law of thermodynamics, the following thermodynamic relations are obtained:

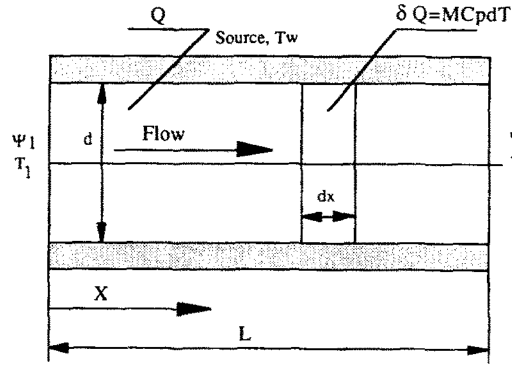

FIG. 1. Schematic diagram of a single-tube heat exchanger.

NOMENCLATURE

c	specific heat [J kg^{-1} K^{-1}]
CPI	coils per inch
d	diameter of pipe [m]
f	friction factor
h	heat transfer coefficient [W m^{-2} K^{-1}]
M	mass flow rate [kg s^{-1}]
Nu	Nusselt number
p	pressure [N m^{-2}]
Q	heat [J s^{-1}, W]
Re_d	Reynolds number based on d
s	entropy [J kg^{-1} K^{-1}]
SBr	pseudo Brinkman number
T	temperature [K]
v	specific volume [m^3 kg^{-1}]
V	velocity [m s^{-1}]
w	perimeter of the duct or tube [m]
x	distance along the length of heat exchanger [m].

Greek symbols

γ	defined in equation (14c)
ρ	density [kg m^{-3}]
τ	defined in equation (15c)
ψ	specific flow-exergy [J kg^{-1}]
Ψ	flow-exergy [J]
$\Delta\Psi$	flow-exergy destruction [J]
$\Delta\Psi^*$	dimensionless flow-exergy destruction.

Subscripts

0	reference state
1	inlet
p	constant pressure
w	wall
x	at x.

$$\left(\frac{\partial h}{\partial T}\right)_{\mathrm{p}} = c_{\mathrm{p}} \tag{3}$$

$$\left(\frac{\partial s}{\partial T}\right)_{\mathrm{p}} = \frac{c_{\mathrm{p}}}{T} \tag{4}$$

$$\left(\frac{\partial s}{\partial p}\right)_{\mathrm{T}} = \frac{1}{T}\left(\frac{\partial h}{\partial p}\right)_{\mathrm{T}} - \frac{v}{T}. \tag{5}$$

Partial derivative of equation (1) with reference to T gives

$$\left(\frac{\partial \psi}{\partial T}\right)_{\mathrm{p}} = \left(\frac{\partial h}{\partial T}\right)_{\mathrm{p}} - T_0\left(\frac{\partial s}{\partial T}\right)_{\mathrm{p}}$$

which, with equations (3) and (4), reduces to:

$$\left(\frac{\partial \psi}{\partial T}\right)_{\mathrm{p}} = c_{\mathrm{p}}\left(1 - \frac{T_0}{T}\right). \tag{6}$$

The partial derivative of equation (1) with reference to p and equation (5) together give

$$\left(\frac{\partial \psi}{\partial T}\right)_{\mathrm{T}} = \left(\frac{\partial h}{\partial p}\right)_{\mathrm{T}} - T_0\left[\frac{1}{T}\left(\frac{\partial h}{\partial p}\right)_{\mathrm{T}} - \frac{v}{T}\right]. \tag{7}$$

Equation (7) reduces to

$$\left(\frac{\partial \psi}{\partial p}\right)_{\mathrm{T}} = \frac{vT_0}{T} = \frac{T_0}{\rho T} \tag{8}$$

for liquids, such as water, for which the change in enthalpy is small for the pressure drop in a tubular exchanger. The present analysis is, thus, limited to tubular heat exchangers with liquid as heat exchanging fluid.

Combining equations (7) and (8) into equation (2) yields

$$\mathrm{d}\psi = c_{\mathrm{p}}\left(1 - \frac{T_0}{T}\right)\mathrm{d}T + \frac{v}{T}T_0\,\mathrm{d}p. \tag{9}$$

In Fig. 1, the exergy destruction over a differential element of length $\mathrm{d}x$ is given by

$$\mathrm{d}\Psi = M\psi_x - M\psi_{x+\mathrm{d}x} - \delta Q\frac{T_{\mathrm{w}} - T_0}{T_{\mathrm{w}}}$$

$$\mathrm{d}\Psi = -M\,\mathrm{d}\psi - \frac{T_{\mathrm{w}} - T_0}{T_{\mathrm{w}}}\delta Q \tag{10}$$

which, with equation (8) reduces to:

$$\mathrm{d}\Psi = MT_0\left[c_{\mathrm{p}}\left(\frac{1}{T} - \frac{1}{T_{\mathrm{w}}}\right)\mathrm{d}T - \frac{\mathrm{d}p}{\rho T}\right]. \tag{11}$$

The fluid temperature $T(x)$ is obtained from the solution of the following differential equation formulated by energy balance over the differential element in Fig. 1:

$$\delta Q = -Mc_{\mathrm{p}}\,\mathrm{d}T = hw(T - T_{\mathrm{w}})\,\mathrm{d}x \tag{12}$$

which can be rewritten as

$$\frac{\mathrm{d}T}{\mathrm{d}x} + \frac{hw}{Mc_{\mathrm{p}}}T - \frac{hw}{Mc_{\mathrm{p}}}T_{\mathrm{w}} = 0. \tag{13}$$

The solution for $T(x)$ from equation (13) is

$$T(x) = T_{\mathrm{w}} + \Delta T_1\,\mathrm{e}^{-\gamma x} \tag{14a}$$

where

$$\Delta T_1 = T_1 - T_{\mathrm{w}} \tag{14b}$$

and

$$\gamma = \frac{hw}{Mc_{\mathrm{p}}}. \tag{14c}$$

Integrating equation (11) with $T = T(x)$ from equation (14), for $x = 0$ to L, an expression for net exergy destruction is obtained. The non-dimensional exergy destruction ($\Delta\Psi^*$) is represented by [10]

$$\Delta\Psi^* = \left[\tau(1-e^{-\gamma L})+\ln\left(\frac{1+\tau\,e^{-\gamma L}}{1+\tau}\right)\right] + \left[\frac{f\,Re\,SBr}{8\,Nu}\ln\left(\frac{1+\tau\,e^{-\gamma L}}{(1+\tau)\,e^{-\gamma L}}\right)\right] \quad (15a)$$

where

$$\Delta\Psi^* = \frac{\Delta\Psi}{MT_0c_p} \quad (15b)$$

$$\tau = \frac{\Delta T_1}{T_w} \quad (15c)$$

$$SBr = \frac{\mu V^2}{kT_w} \quad (15d)$$

with

$$-\frac{dp}{dx} = \frac{f\rho V^2}{2d}. \quad (15e)$$

RESULTS AND DISCUSSION

An experimental study [10] was carried out using a counterflow concentric tube heat exchanger (Fig. 2) which provides approximately 2.2 m long test section. The wire-coil insert for heat transfer enhancement was assembled inside a copper tube of 14 mm ID which is concentric with the outer copper tube of 26 mm ID. The circulation system provided a counterflow condition in the test section. In this experiment, a 5 HP turbine pump circulated hot water through the inner tube from a 100-gallon stainless steel tank which was maintained at a temperature of 30–50°C by an 18 kW electric heater. The hot water exchanged heat in the test section with the cold water in the annular space at 10–20°C pumped by a 3 HP multistage turbine pump. The cold water leaving the test section was cooled by a coil-in-coil cooler before returning to the cold water reservoir, another 100-gallon stainless steel tank. The experimental conditions maintained were such that the temperature change in the annular section was relatively small (<4°C over 2.2 m length) to approximate the constant temperature condition under which equation (15) is applicable. Over an experimental range of Reynolds number (Re_d = 35 000–92 000), Nu and f were measured for a 14 mm diameter tube with wire-coil insert (wire diam. = 0.813 mm, coil pitch = 2.82–8.47 mm) as shown in Figs. 3 and 4. Equation (15) was used to calculate exergy destruction. Figure 5 shows the dimensionless exergy destruction as a function of Re for this case. Clearly a thermodynamically optimum is seen in Fig. 5 in the vicinity of $Re \cong 60\,000$. The exergy destruction in case of a smooth tube is also shown in Fig. 5.

The dimensionless exergy destruction $\Delta\Psi^*$ for augmented tubes initially decreases with increase in Reynolds number. In this region, the reduction in exergy destruction due to enhanced heat transfer more than offsets the increase in exergy destruction due to increased flow friction. However, as Reynolds number increases, the loss of availability (exergy) due to fric-

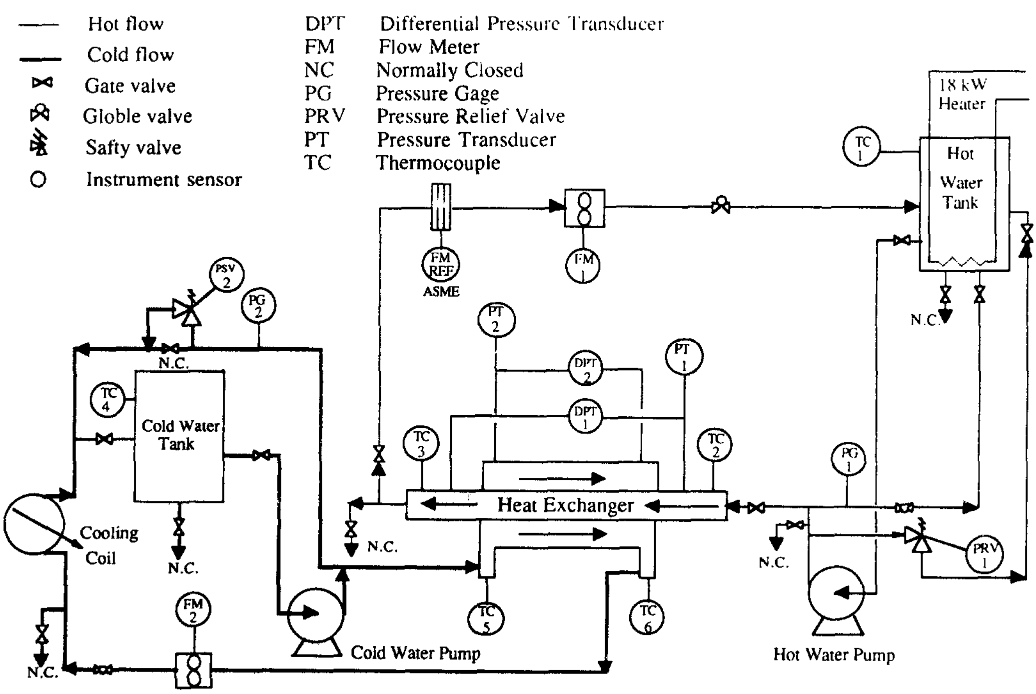

FIG. 2. Schematic diagram of experimental set-up.

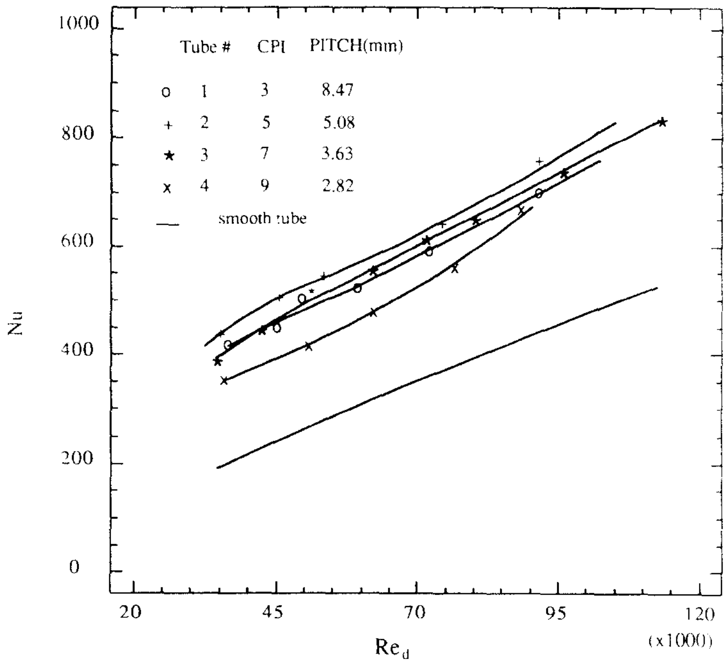

FIG. 3. Nusselt number with coiled-wire inserts (wire diameter = 0.813 mm).

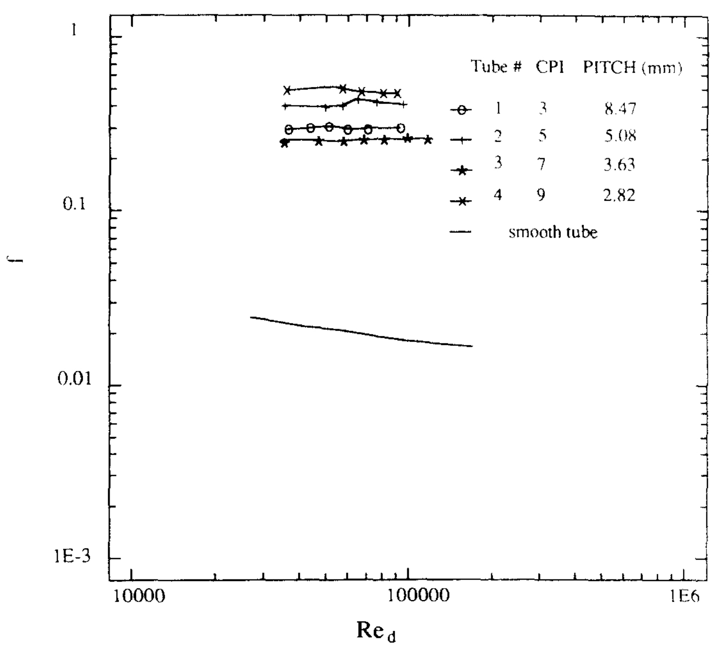

FIG. 4. Friction factor with coiled-wire inserts (wire diameter = 0.813 mm).

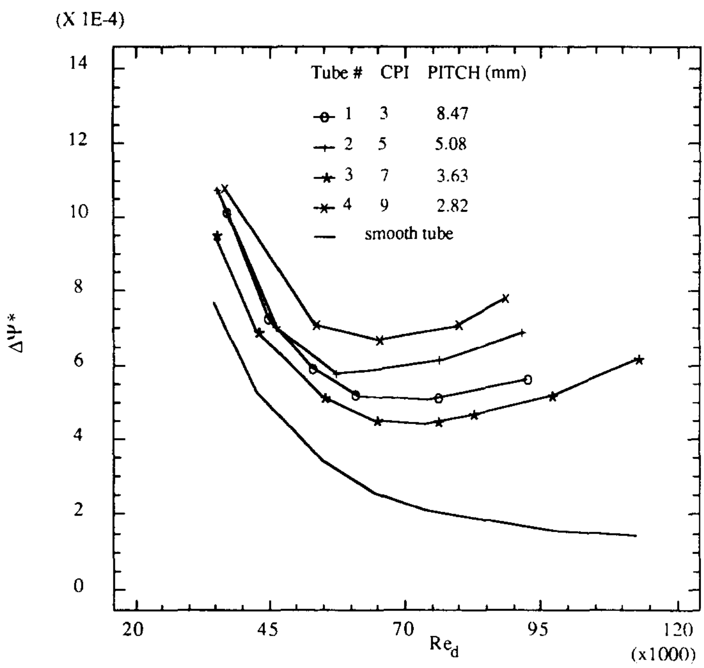

FIG. 5. Dimensionless exergy destruction for coiled-wire inserts (wire diameter = 0.813 mm).

tion increases and, beyond the optimum *Re*, eventually exceeds any reduction in exergy destruction due to improved heat transfer. The combined effect, thus, results in a minimum for $\Delta\Psi^*$ corresponding to optimum conditions in these tests.

The present method, thus, permits an evaluation of exergy destruction in a tubular heat exchanger. By minimizing this destruction, a thermodynamically optimum can be determined (a) for the operation of a given heat exchanger or (b) in the selection of an enhancement device. It should be noted that the optimization of the heat exchanger alone would not guarantee the optimum for an overall system in which the heat exchanger is one of several subsystems. The analysis, however, does provide a very useful method to evaluate and compare the performance of various heat transfer enhancement devices.

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