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The harmonic response of counter-flow heat exchangers - Analytical approach and comparison with experiments

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ABSTRACT

The dynamic behavior of counter flow heat exchangers is studied using the harmonic response. To solve the set of partial differential equations the complex notation is used so that these equations become ordinary differential equations. In an appendix it is shown that assuming a uniform temperature in a cross section of the separating wall is correct for the example used as comparison. Experimental results dealing with a double pipe heat exchanger are compared with results obtained by assuming the correlations to be used for the convection heat transfer coefficients. The comparison is satisfactory when the excitation is applied to the inner tube fluid. It is less accurate when the excitation is applied to fluid flowing in the annulus. Finally it is shown that the residence time of the fluids is linked to a peak in the response curves.

1. Introduction

Fouling detection of heat exchangers is a major subject to insure optimal performances of industrial installations. It has been recently shown [1] that the study of the dynamic behavior of heat exchangers is a promising way to detect fouling. It can be noted that it is also necessary to get accurate models of heat exchangers when they are part of a more complex system to be studied in transient states. Depending on the time range required for the application (seconds to minutes for the step response, or hours to many days for monitoring), and the time available to model the heat exchanger, many techniques have been proposed, from the simplest to the most sophisticated.

For very short term characterizations, Computational Fluid Dynamics is sometimes used to solve the Navier-Stokes and energy equations [2,3]. This makes possible the detailed modeling of the geometry of the heat exchanger and of the fluid flow. But the computational time is very long and not suited for long term predictions needed for fouling detection.

For short term characterizations, an easier way to estimate the transient behavior of a heat exchanger is to numerically solve the partial derivative equations using a finite difference technique without solving the Navier-Stokes equations for the fluid as done in CFD [4–7]. In this case a simplified geometry is considered but the equations to be solved are simpler. The time span can be much longer than for the CFD computations, for a given computation time. But usually the time step needed to get accurate results is still not compatible with fouling detection.

For a long term characterization of the behavior of a heat exchanger, a standard approach is based on the Laplace transform and a numerical inverse procedure [8–10]. Flow maldistribution can be taken into account using this approach [11–13]. Once these models are determined a very simple computation is necessary to get the long term response in the Laplace space in a very short time. The main issue is then the inverse procedure.

When modeling a heat exchanger as a lumped system, Simulink® based models [14,15] and fuzzy observers [16] can be also used for long term characterizations; but the models are not very “flexible” and the number of “cells” has to be well defined as shown in Ref. [17].

Also based on physical equations, the state space models are efficient [18,19] and are more “compact” but more difficult to tune. Coming from the automatic control community, techniques such as neural networks [20–22] and Linear Parameter Varying models [23,24] use only the inlet and outlet variables (temperatures and flow rates) to model heat exchangers as black boxes. Once they have been tuned, the computational time is very short. Hence, all these models can be used for long term characterizations.

Very close to the Laplace transform approach, the study of the frequency response is presented [25] for a very restrictive case: no separating wall and Number of Transfer Units equal in both streams.

With the aim of getting a very sensitive tool for fouling detection,

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Nomenclature

A	Convective area per unit length [m^2/m]
B	Simplifying function [−]
Bi	Biot number [−]
C	Constant in a correlation [−]
c	Specific heat [J/(kg.K)]
D	Diameter [m]
E	Complex excitation [K] or [°C]
e	Thickness [m]
f	Simplifying function [(J/kg)/m]
f_{peak}	Frequency of the first peak [Hz]
g	Simplifying function [(J/kg)/m]
H	Simplifying function [m.(K/W)]
h	Convection heat transfer coefficient [W/(m^2.K)]
i	Spatial index or complex number $i = \sqrt{-1}$
K	Harmonic Fourier number [−]
k_θ	Coefficients in a transfer matrix (Eq. (15)) [−]
k_μ	Coefficients in a transfer matrix (Eq. (15)) [K/(kg/s)]
L	Length [m]
M	Mass per unit length [kg/m]
$\dot{m}$	Mass flow rate [kg/s]
N	Number of points for the spatial discretization
n	Constant in a correlation [−]
Nu	Nusselt number [−]
P	Point for the discretization
p	Constant in a correlation [−]
Pr	Prandtl number [−]
$\Re$	Real part of a complex variable
Re	Reynolds number [−]
T	Temperature [K] or [°C]
t	Time [s]
x	Abscissa [m]
y	Dummy variable
Z	Coefficient in a matrix

Greek symbols

α	Simplifying function [−]
β	Simplifying function [−]
Δx	Length of the cells used for the discretization [m]
δ	Simplifying function [K/(kg/s)]
Φ	Complex heat flux [W/m^2]
φ	Heat flux [W/m^2]
γ	Simplifying function [K/(kg/s)]
η	Simplifying function [−]
λ	Thermal conductivity [W/(m.K)]
$\dot{\mu}$	Complex mass flow rate [kg/s]
θ	Complex temperature [K] or [°C]
ρ	Density [kg/m^3]
ω	Angular frequency [rad/s]

Subscripts

0	At
c	Cold fluid or cold side of the separating wall
e_w	At $x = e_w$
h	Hot fluid or hot side of the separating wall
j	Spatial index
in	At the inlet or inner diameter
k	Dummy subscript
max	Maximum
out	At the outlet or outer diameter
w	Separating wall

Upper scripts

—	Mean value of a variable with respect to time
~	Fluctuating part of a variable with respect to time

the lock-in technique has recently been introduced in the study of heaters and heat exchangers [26–28]. The main advantage of this technique over the previous ones is the fact that the "disturbances" applied to the process lead to temperature variations that are in the range of the measurement noise; much smaller than what is needed for the other methods. To efficiently use this technique, it is necessary to choose the excitation frequency. So, the aim of the present work is to study the harmonic response of clean counter flow heat exchangers and to compare computed results to experimental data [29] in order to validate the developments, and then to show that it is possible to determine a well adapted excitation frequency very easily. In the present work the particular case of sinusoidal fluctuations of the inlet temperatures and flow rates are considered.

2. Developments

In this paper, the most efficient heat exchanger type is studied: counter flow heat exchangers. A simple but efficient model is a tube-in-tube heat exchanger. Noting x the location along the tube axis, the standard partial differential equations are:

$$\text{For the hot side: } M_h c_h \frac{\partial T_h}{\partial t} + \dot{m}_h c_h \frac{\partial T_h}{\partial x} = h_h A_h (T_w - T_h) \tag{1}$$

$$\text{For the cold side: } M_c c_c \frac{\partial T_c}{\partial t} - \dot{m}_c c_c \frac{\partial T_c}{\partial x} = h_c A_c (T_w - T_c) \tag{2}$$

$$\text{For the separating wall: } M_w c_w \frac{\partial T_w}{\partial t} = h_h A_h (T_h - T_w) + h_c A_c (T_c - T_w) \tag{3}$$

This assumes that the temperature in the separating wall is uniform in a cross section of the separating wall. This is equivalent to assume that the temperature difference between the two sides of the separating wall is small, i.e. the thermal resistance of the separating wall is negligible due to the high thermal conductivity and the small thickness of this wall. It is also assumed that the longitudinal conduction in the tube wall is negligible. The dimensionless axial conduction parameter as defined in Refs. [30–32] is about 1.35 10^{-5} showing that the ineffectiveness has not to be taken into account. From a more practical point of view, in the comparisons shown hereafter, the axial heat rate in the tube is much lower than 0.01% of the heat rate exchanged in the heat exchanger. This is due to the fact that the temperature difference between the inlet and the outlet in the tube wall is close to 10 °C for all comparisons.

It is considered that both fluids present mass flow and inlet temperature fluctuations. The amplitudes of these fluctuations are small compared to the average values. It is then possible to assume that the thermophysical characteristics of the fluids are constant. The highest angular frequency considered in this study being lower than 1 Hz, it is also considered that the correlations used to compute the convection heat transfer coefficients are valid for steady states as well as for transient states that are considered as quasi-steady.

Each variable is split into a mean value over time and a fluctuating value:

$$y(x,t) = \bar{y}(x) + \tilde{y}(x,t)$$

$$\text{This leads to : } \frac{\partial y(x,t)}{\partial t} = \frac{\partial \tilde{y}(x,t)}{\partial t}, \quad \frac{\partial y(x,t)}{\partial x} = \frac{d\bar{y}(x)}{dx} + \frac{\partial \tilde{y}(x,t)}{\partial x}$$