THERMAL TRANSIENTS IN PARALLEL FLOW HEAT EXCHANGERS

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The control of systems incorporating heat exchangers necessitates an understanding of their response to variations of flow rates and/or entering fluid temperatures. Therefore, many analytical[1] and experimental[2] studies have dealt with this subject. Most models used to evaluate such transients assume that some parameters (the convective heat transfer coefficients, the effectiveness, etc) are known and remain constant throughout the transient operation. To avoid this assumption, we proceed with a numerical solution of the partial differential conservation equations. Here we analyse the response of a parallel flow heat exchanger following a change of one entry temperature.

MATHEMATICAL MODELING AND SOLUTION METHOD

The system under study consists of two coaxial tubes of length L. Their interior and exterior radii are R_1, R_2 for the small tube and R_3, R_4 for the big one. The outside surface is adiabatic while the two fluids, moving in the direction of increasing Z, enter the system with uniform but different temperatures T_{ie} and T_{oe}. We assume that the fluids are Newtonian, incompressible and that body forces are negligible. We also consider that the flow rates are constant and that the flow fields are developed. On the other hand, the temperature field is two-dimensional and varies with time.

The non-dimensional energy conservation equation for the fluids and the solid walls is:

$$\nabla^2 (\Gamma\ T) + S = \lambda\ (\partial T/\partial t)$$

The expressions of the diffusion coefficient Γ, the source term S and the constant λ are given in Table 1. D_H is the hydraulic diameter of the annulus, $D_1=2R_1$, Pe is the Péclet number, α is the thermal diffusivity, V_{av} the average velocity, k the molecular conductivity and k_{eff} the effective conductivity (sum of the molecular and turbulent conductivities). The indices i and o refer to the inside (hot) and outside (cold) fluids respectively.

The analytical expressions for the velocity profiles in the circular and annular regions are known for both laminar and turbulent flows[3]. For laminar conditions $k_{eff}=k$ and, therefore, $\Gamma=1$ for the fluids. For turbulent conditions, the radial distribution of the effective viscosity is calculated by applying the integral form of the axial momentum equation between two arbitrary cross sections. The corresponding distribution of k_{eff} is then found by setting the turbulent Prandtl number equal to 1.

The boundary conditions at the inlet (Z=0), for t<0, are : T=1 for $0<r<0.5$, T=0 for $R_2/D_1<r<R_3/D_1$ and $\partial T/\partial Z=0$ for $0.5<r<R_2/D_1$ and $R_3/D_1<r<R_4/D_1$. For all t, at the exit (Z=L/D) : $\partial T/\partial Z=0$ while at the axis (r=0) and at the outside surface ($r=R_4/D_1$) : $\partial T/\partial r=0$.

The solution was obtained using CONDUCT[4], a code based on the control volume approach. An implicit formulation was adopted for the time derivative. Several numerical tests were performed to ensure that the results are grid independent. The adopted grid consists of 22 radial and 52 axial non-uniformly distributed nodes. Their density is greater near the inlet and near the fluid-solid interfaces where temperature gradients are most important. Validation was obtained by comparing numerical predictions for steady state with analytical results and for unsteady state with experimental results[5].

RESULTS AND DISCUSSION

The results are for a water-to-water heat exchanger with copper tubes and R_1=15 mm, R_2=16 mm, R_3=25 mm, R_4=30 mm, L=3 m. The flow rates are such that $Re_i=Re_o=5000$. For these conditions the ratio k_{eff}/k varies from 1 to 40 in the central tube and from 1 to 400 in the annulus. Fig. 1 shows the initial temperature profiles at four axial positions. Very close to the inlet the temperature of each fluid is essentially uniform (T_i=1, T_o=0). Further downstream the temperature of the hot fluid decreases while that of the cold fluid increases. However, since the flow rate of the latter is much greater, the corresponding temperature change is quite small. From t=0 the temperature of the hot fluid increases to 1.2. Fig. 2 shows the temperature profiles shortly after the application of this step change. The temperature of the hot fluid at Z=0.27 and Z=10.8 is considerably higher than those in fig. 1. On the other hand, the temperatures at Z=34 have changed little while at Z=98.3 they are identical to those in fig. 1. At that instant therefore, the hot front is approximately at Z=34. Later (fig. 3) the profiles at the first two cross sections have not changed from those in fig. 2. They have reached the steady state corresponding to the new entry conditions. At this instant, the temperatures at Z=98.3 have started increasing, indicating that the hot front has now reached this position.

Subsequently, this heat exchanger was submitted to the following conditions corresponding to laboratory tests[6] : Re_i=1876, Re_o=414, inlet temperatures for t<0 : $T_{ie}=T_{oe}$=13.1 °C, inlet conditions for t>0 : T_{ie}=53.1 °C, T_{oe}=13.1 °C. The outlet temperature of the cold fluid was computed numerically and from an empirical formula[6]. The latter approximates the response by a constant, during an initial delay period t_r, followed by an exponential decay characterized by a single time constant τ. The experimental values for these parameters are t_r =50 s and τ=48 s while the numerical results are t_r =56 s and τ=48.2 s. In view of the relatively good agreement between these results, we have performed a parametric study by varying some physical and operational characteristics of the system. Figs. 4 and 5 respectively show the effect of the length L and of the flow rates (or, equivalently, the Reynolds numbers) on the outlet temperature of the cold fluid. We note that t_r and τ increase when L increases and when the flow rates decrease.

CONCLUSION

The transient temperature field in a parallel flow heat exchanger has been calculated numerically assuming fully developed hydrodynamic conditions. This approach uses fewer assumptions than published analytical studies. It shows the influence of physical and operational characteristics on experimentally defined parameters that describe the transient response of heat exchangers.

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			S
			S
			$\bar{V}_i \frac{\partial T_i}{\partial z}$
			$\bar{V}_o \frac{D_1}{D_h}\frac{\partial T_o}{\partial z}$
			0
		α_o	0

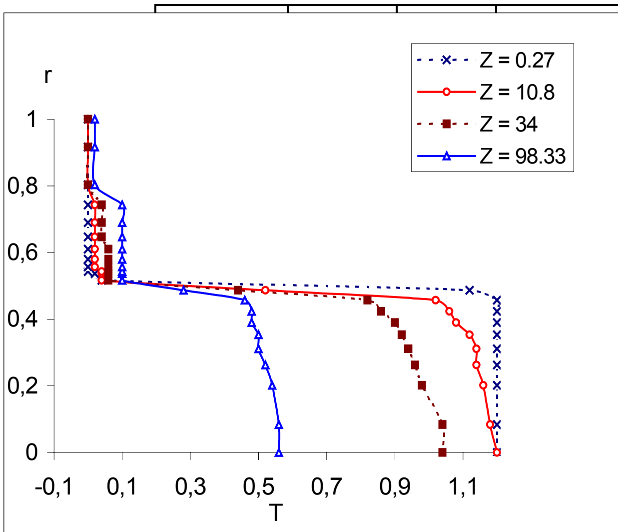
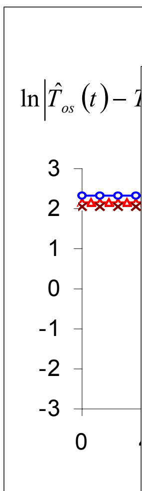


Fig. 3 Temperatures at t=48 s, $Re_i=Re_o$ =5000

Table 1. Diffusion coefficients and source terms



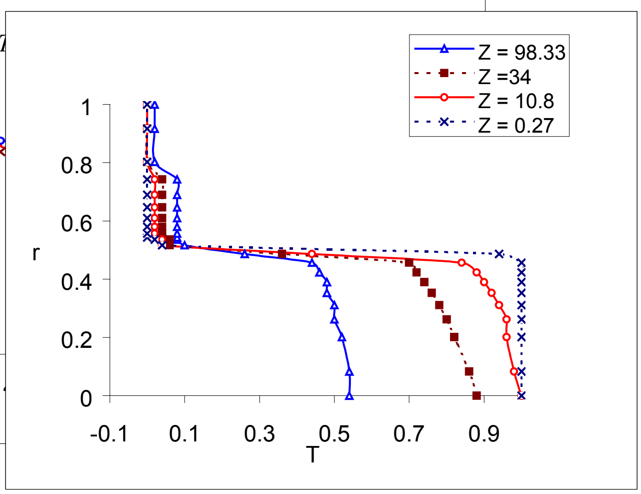


Fig. 4 : Outlet temperature of cold fluid

Fig. 1 Temperatures at t=0, $Re_i=Re_o$=5000

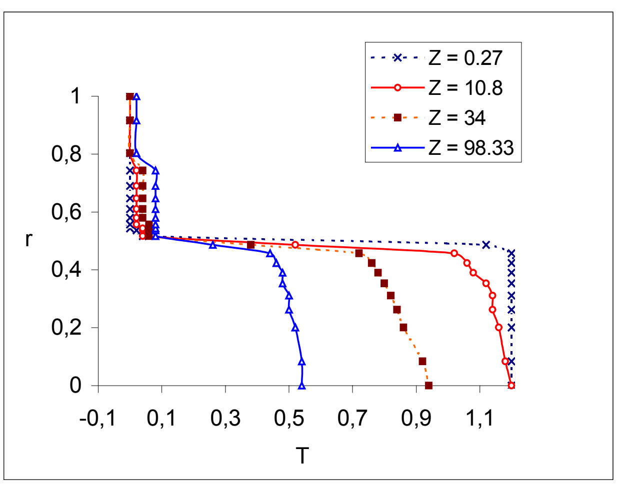


Fig. 2 Temperatures at t=2.1 s, $Re_i=Re_0$=5000

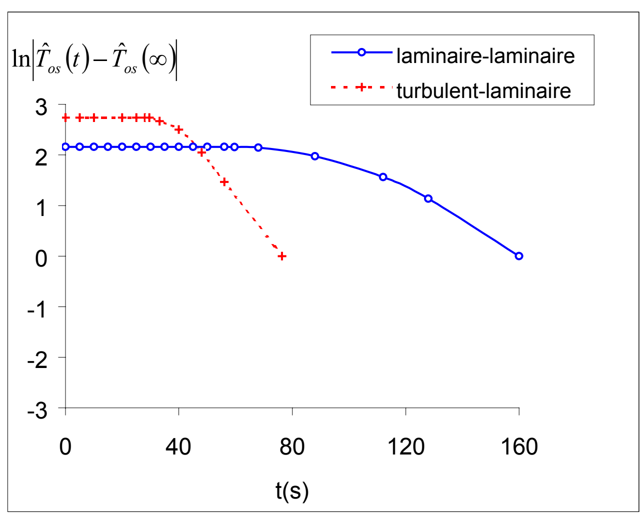


Fig. 5 Outlet temperature of cold fluid