Transient response of crossflow heat exchangers with one fluid mixed

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The transient responses of crossflow heat exchangers with one fluid mixed and the other fluid unmixed are analysed numerically in this paper. Two cases are investigated: one for the response to a unit step change in the inlet temperature of the stepped fluid being mixed, and the other for the stepped fluid being unmixed. It is found that the outlet transient responses of the unstepped fluid for the two cases investigated are the same, but the outlet transient responses of the stepped fluid are distinctly different. The effects of various governing parameters concerning the transient response are discussed in detail and shown in graphical form.

Keywords: *transient response, crossflow heat exchangers, mixed fluid*

Introduction

A heat exchanger is generally a part of a system that may be exposed to a number of planned or unplanned transients or start-ups and shutdowns in a certain time. The transient may also arise as a result of change in operating conditions, such as a change in inlet temperature or flow rates. The transient could produce such undesirable effects as reduced heat transfer performance, severe thermal stresses, and eventual mechanical failure. Thus it is important to predict the transient response of a heat exchanger, in addition to its steady-state performance. Furthermore, knowledge of the dynamic characteristics of heat exchangers is useful for the dynamic control of power plants and air-conditioning systems, as well as many other applications.

Several investigators have treated crossflow heat exchanger transients. Dusinberre[1] numerically obtained the transient behaviour of a crossflow heat exchanger with both fluids unmixed. Only one specific case was considered, and a parametric study was not attempted. Myers *et al*[2] analysed by an approximate integral method the transient response of a crossflow heat exchanger with one fluid mixed. They considered the fluid with a stepped inlet temperature as mixed, and the unstepped fluid as unmixed. Their solution is applicable under the condition that the thermal capacitance of the exchanger core is very large compared with the thermal capacitance of fluids contained in the heat exchanger. This condition is generally satisfied if both fluids are gases.

Yamashita *et al*[3] used a finite difference method to analyse the transient response to a step change in the inlet fluid temperature of crossflow heat exchangers with both fluids unmixed. They illustrated the effects of governing parameters concerning transient response by varying each one, with the others fixed at the same constant values. By using the Laplace transform method, Romie[4] analysed the transient response of a gas-to-gas crossflow heat exchanger with neither gas mixed. The excitation was a step change in the inlet temperature of either gas side.

The purpose of this paper is to present the transient response of a crossflow heat exchanger with one fluid mixed and the other fluid unmixed for a unit step change in the inlet temperature of either the mixed or unmixed hot fluid. The responses are found by use of a finite difference method similar to that described in Refs 3 and 5. Unlike the work in Ref 2, our solutions presented here are not limited to the large wall capacitance condition.

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Manuscript received 29 October 1986 and accepted for publication on 12 January 1987

0142-727X/87/030182-05$3.00

Mathematical analysis

General formulation

The crossflow heat exchanger analysed is shown schematically in Fig 1. The hot fluid is taken as the stepped fluid. Fig 1(a) shows the case when the hot fluid is mixed and the cold fluid is unmixed, while Fig 1(b) shows the case when the hot fluid is unmixed and the cold fluid is mixed.

On the basis of the same idealizations and notations as those made by Shah[6], applying the energy balance on the control volume indicated in Fig 1(a), the three governing differential equations for the step change in the inlet temperature of the hot fluid being mixed are given by

$$\bar{C}_h\frac{\partial T_h}{\partial\tau}+C_h\frac{\partial T_h}{\partial(y/L_h)}+(\eta_o hA)_h\left(T_h-\frac{1}{L_c}\int_0^{L_c}T_w\,dx\right)=0 \tag{1a}$$

$$\bar{C}_c\frac{\partial T_c}{\partial\tau}+C_c\frac{\partial T_c}{\partial(x/L_c)}-(\eta_o hA)_c(T_w-T_c)=0 \tag{1b}$$

$$\bar{C}_w\frac{\partial T_w}{\partial\tau}-(\eta_o hA)_h(T_h-T_w)+(\eta_o hA)_c(T_w-T_c)=0 \tag{1c}$$

In order to reduce the number of parameters, it is convenient to define the following dimenionless quantities:

$$\bar{x}=x/L_c,\qquad \bar{y}=y/L_h,\qquad \tau^*=\tau/\tau_{d,c}$$

$$\bar{C}_w^*=\bar{C}_w/\bar{C}_{min},\qquad C^*=\frac{C_{min}}{C_{max}},\qquad \tau_d^*=\frac{\tau_{d,h}}{\tau_{d,c}},\qquad N_c=\frac{UA}{C_c} \tag{2}$$

$$\tau_d^*=\frac{\tau_{d,h}}{\tau_{d,c}},\qquad R^*=\frac{(\eta_o hA)_h}{(\eta_o hA)_c},\qquad N_{tu}=\frac{UA}{C_{min}},\qquad N_h=\frac{UA}{C_h}$$

where C_{min} is the minimum of C_h or C_c, and C_{max} is the maximum of C_h or C_c. $\tau_{d,h}$ and $\tau_{d,c}$ denote the hot and cold fluid dwell time, respectively. UA is defined as

$$\frac{1}{UA}=\frac{1}{(\eta_o hA)_h}+\frac{1}{(\eta_o hA)_c} \tag{3}$$

The other symbols are defined in the Notation. Note that the wall resistance and fouling resistances are included in the hot and cold side thermal resistances.

In terms of these dimensionless quantities, Eqs (1) become

$$\tau_d^*\frac{\partial T_h}{\partial\tau^*}+\frac{\partial T_h}{\partial\bar{y}}+(1+R^*)N_h\left(T_h-\int_0^1 T_w\,d\bar{x}\right)=0 \tag{4a}$$

$$\frac{\partial T_c}{\partial\tau^*}+\frac{\partial T_c}{\partial\bar{x}}-\left(\frac{1+R^*}{R^*}\right)N_c(T_w-T_c)=0 \tag{4b}$$

$$\bar{C}_w^*\frac{\partial T_w}{\partial\tau^*}-\frac{1+R^*}{R^*}N_{tu}[R^*(T_h-T_w)-(T_w-T_c)]=0 \tag{4c}$$

Similarly, applying the energy balance on the control volume indicated in Fig 1(b), the three resulting equations for the stepped fluid being unmixed are as follows.

$$\tau_d^\ddagger \frac{\partial T_h}{\partial \tau^*}+\frac{\partial T_h}{\partial \bar{y}}+(1+R^*)N_h(T_h-T_w)=0 \tag{5a}$$

$$\frac{\partial T_c}{\partial \tau^*}+\frac{\partial T_c}{\partial \bar{x}}-\left(\frac{1+R^*}{R^*}\right)N_c\left(\int_0^1 T_w\,d\bar{y}-T_c\right)=0 \tag{5b}$$

$$\bar{C}_w^*\frac{\partial T_w}{\partial \tau^*}-\frac{1+R^*}{R^*}N_{tu}[R^*(T_h-T_w)-(T_w-T_c)]=0 \tag{5c}$$

Since Eqs (4) and (5) are linear and homogeneous, it is adequate to consider the initial conditions for T_h, T_w and T_c to be zero throughout the heat exchanger. Consequently, T_h, T_w and T_c represent the temperature differences above the initial values of these temperatures. The input to Eqs (4) and (5) is a unit step change in the inlet temperature of the hot fluid.

To facilitate numerical integration of Eqs (4) and (5), the characteristic coordinates X, Y, Γ suggested by Yamashita[3] are used, where

$$X=\bar{x}, \qquad Y=\bar{y}, \qquad \Gamma=\tau^*-\bar{x}-\tau_d^\ddagger\bar{y} \tag{6}$$

Then Eqs (4) become

$$\frac{\partial T_h}{\partial Y}+(1+R^*)N_h\left(T_h-\int_0^1 T_w\,dx\right)=0 \tag{7a}$$

$$\frac{\partial T_c}{\partial X}-\left(\frac{1+R^*}{R^*}\right)N_c(T_w-T_c)=0 \tag{7b}$$

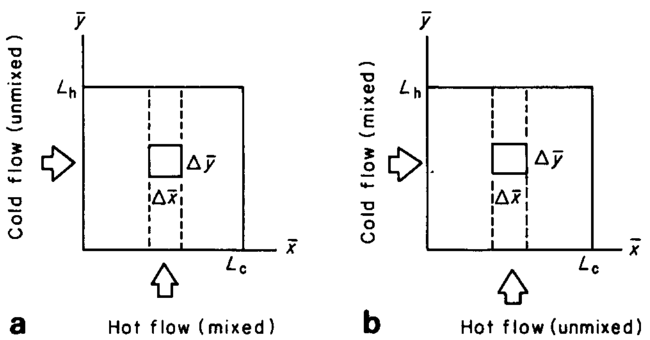

Figure 1 Schematic diagrams of crossflow heat exchanger with one fluid mixed

$$\frac{\partial T_w}{\partial \Gamma}-\frac{1+R^*}{R^*}N_{tu}[R^*(T_h-T_w)-(T_w-T_c)]=0 \tag{7c}$$

The initial conditions are transformed as follows:

at $\Gamma=-\bar{x}-\tau_d^\ddagger\bar{y}$;

$$T_h=T_w=T_c=0 \tag{8}$$

Similarly, Eqs (5) can be transformed in terms of the characteristic coordinates. The resulting equations are omitted here. Eqs (7) subjected to the initial conditions of Eqs (8) and to a unit step change in the inlet temperature of the hot fluid are solved by a finite difference method similar to that described in Refs 3 and 5. The finite difference equations are formulated along the plane $\tau^*=$constant. Grid spacings are taken so as to satisfy the relations $\Delta\bar{y}/\Delta\tau^*=1$, $\Delta\bar{v}/\Delta\tau^*=1/\tau_d^\ddagger$. Additional details can be found in Wang[7].

Large wall capacitance $\bar{C}_w^*$ solution

For the case of $\bar{C}_w^*$ being very large (approaching infinity), using a new time variable $\xi=\tau^*/\bar{C}_w^*$, substituting it into Eqs (4) and neglecting the terms $(\tau^*/\bar{C}_w^*)(\partial T_h/\partial\xi)$ and $(1/\bar{C}_w^*)(\partial T_c/\partial\xi)$, then the basic equations for the stepped fluid being mixed can be simplified to the following:

$$\frac{\partial T_h}{\partial \bar{y}}+(1+R^*)N_h\left(T_h-\int_0^1 T_w\,d\bar{x}\right)=0 \tag{9a}$$

$$\frac{\partial T_c}{\partial \bar{x}}-\left(\frac{1+R^*}{R^*}\right)N_c(T_w-T_c)=0 \tag{9b}$$

$$\frac{\partial T_w}{\partial \xi}-\frac{1+R^*}{R^*}N_{tu}[R^*(T_h-T_w)-(T_w-T_c)]=0 \tag{9c}$$

The initial conditions become

$$\xi=0, \qquad T_w=0 \tag{10}$$

Similarly, the basic governing equations for the stepped fluid being unmixed, Eqs (5), also can be simplified as $\bar{C}_w^*$ approaches infinity. It should be noted that the transient parameters $\tau_d^\ddagger$ and $\bar{C}_w^*$ disappear in the above equations.

Eqs (9) also are solved numerically by a finite difference method. Details also can be found in Wang[7]. For the case of $\bar{C}_w^*$ is very large, Eqs (9) can be calculated more accurately with a smaller number of nodes than by directly solving Eqs (7), without approximating the basic equations. The range of

Notation

A	Heat transfer surface area on one side of heat exchanger (with a subscript c or h)
C	Flow stream heat capacity rate (with a subscript c or h)
C_p	Specific heat
C^*	Heat capacity rate ratio $\equiv C_{min}/C_{max}$
$\bar{C}$	Flow stream heat capacitance $\equiv MC_p$
C_{min}	Minimum of C_h or C_c
C_{max}	Maximum of C_h or C_c
$\bar{C}_w$	Heat capacitance of the wall
$\bar{C}_w^*$	Wall heat capacitance ratio $\equiv \bar{C}_w/\bar{C}_{min}$
h	Heat transfer coefficient
L	Heat exchanger length for fluid flows
M_w	Mass of the fluid in the heat exchanger at any instant of time
M	Mass of heat exchanger core
N	Number of heat transfer units $\equiv UA/C$
N_{tu}	Number of heat transfer units $\equiv UA/C_{min}$
R	Ratio of thermal resistances $\equiv(\eta_o hA)_h/(\eta_o hA)_c$
T	Temperature (with a subscript c, w or h)
$\bar{T}_{h,o}$, $\bar{T}_{c,o}$	Mean outlet temperature of the hot fluid and cold fluid, respectively
U	Overall heat transfer coefficient
u	Fluid mean axial velocity
X, Y	Characteristic coordinates defined in Eq (6)
x, y	Cartesian coordinate
$\bar{x}, \bar{y}$	Dimensionless distance: $\bar{x}\equiv x/L_c$, $\bar{y}\equiv y/L_h$
Γ	Characteristic coordinate defined in Eq (6)
ε	Mean outlet transient response
η_o	Total efficiency of the surface on one side of the heat exchanger
ξ	Dimensionless time variable $\equiv\tau^*/\bar{C}_w^*$
τ	Time variable
τ_d	Dwell time
τ^*	Dimensionless time variable $\equiv\tau/\tau_{d,c}$
$\tau_d^\ddagger$	Dwell time ratio $\tau_{d,h}/\tau_{d,c}$

Subscripts

c	Cold fluid side
h	Hot fluid side
w	Solid wall
o	Outlet

Superscripts

1	Case of stepped fluid mixed
2	Case of stepped fluid unmixed

validity of the large wall capacitance approximation is evaluated in a later section.

Mean outlet transient response

The mean outlet transient response for each fluid is defined as

$$\varepsilon_{h,o}=\frac{\bar{T}_{h,o}(\tau^*)-\bar{T}_{h,o}(0)}{\bar{T}_{h,o}(\infty)-\bar{T}_{h,o}(0)}$$

$$\varepsilon_{c,o}=\frac{\bar{T}_{c,o}(\tau^*)-\bar{T}_{c,o}(0)}{\bar{T}_{c,o}(\infty)-\bar{T}_{c,o}(0)} \tag{11}$$

where $\bar{T}_{h,o}$ and $\bar{T}_{c,o}$ denote the outlet mean temperature of the hot fluid and cold fluid, respectively. They are calculated as follows:

$$\bar{T}_{h,o}=\int_0^1 T_{h,o}\,d\bar{x}$$

and

$$\bar{T}_{c,o}=\int_0^1 T_{c,o}\,d\bar{y} \tag{12}$$

Note that $\varepsilon_{h,o}$ and $\varepsilon_{c,o}$ represent the ratio of the corresponding fluid temperature change from its initial value to its ultimate value at time is equal to infinity. It is noted that $\varepsilon_{h,o}=\varepsilon_{c,o}=0$ at $\tau^*=0$, and $\varepsilon_{h,o}=\varepsilon_{c,o}=1$ as $\tau^*\to\infty$.

Discussion of results

Eqs (4) and (5) show that the mean outlet temperature responses $\varepsilon_{h,o}$ and $\varepsilon_{c,o}$ are dependent upon six dimensionless groups: τ^*, τ_d^*, $\bar{C}_w^*$, R^*, N_{tu} and C^*, where τ^*, τ_d^*, $\bar{C}_w^*$ are transient groups and N_{tu}, C^* and R^* are steady-state groups. The effects of each parameter on the transient responses are discussed below. We consider one of five groups (τ_d, $\bar{C}_w^*$, R^*, N_{tu} and C^*) as a parameter, the value of the other four groups being unity. To distinguish the two cases investigated, a superscript 1 is added to the results for the case of the stepped fluid being mixed, and a superscript 2 is added for the results of the case of the stepped fluid being unmixed.

Figs 2 and 3 show the local temperature as a function of time and position for the stepped fluid mixed and unmixed, respectively, in the cases of $\tau_d^*=\bar{C}_w^*=C^*=R^*=N_{tu}=1.0$. The solid lines, dashed lines and chain lines indicate T_h, T_w and T_c, respectively. The symbols A, B and C indicate the position

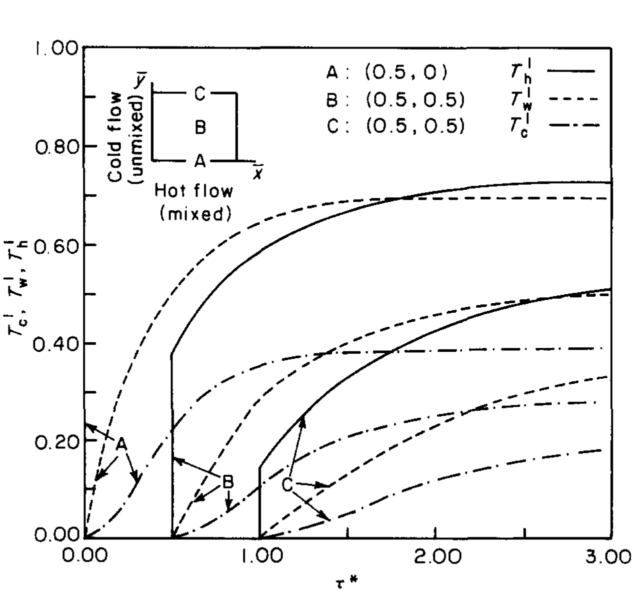

Figure 2 Variation of local temperature with time for stepped fluid mixed

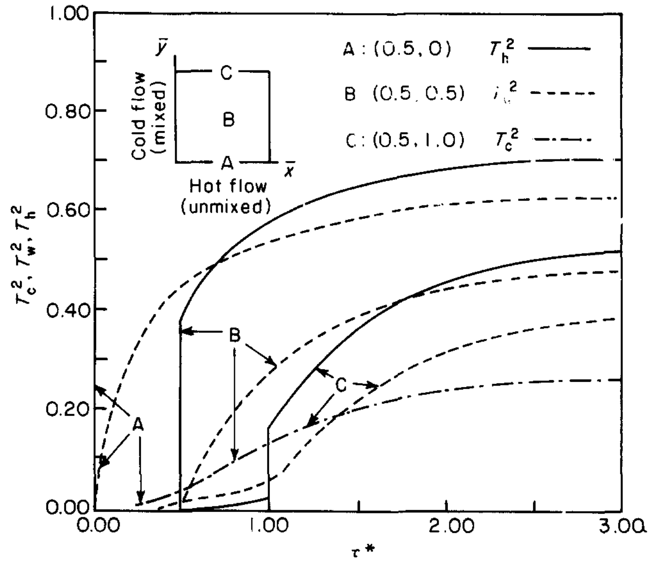

Figure 3 Variation of local temperature with time for stepped fluid unmixed

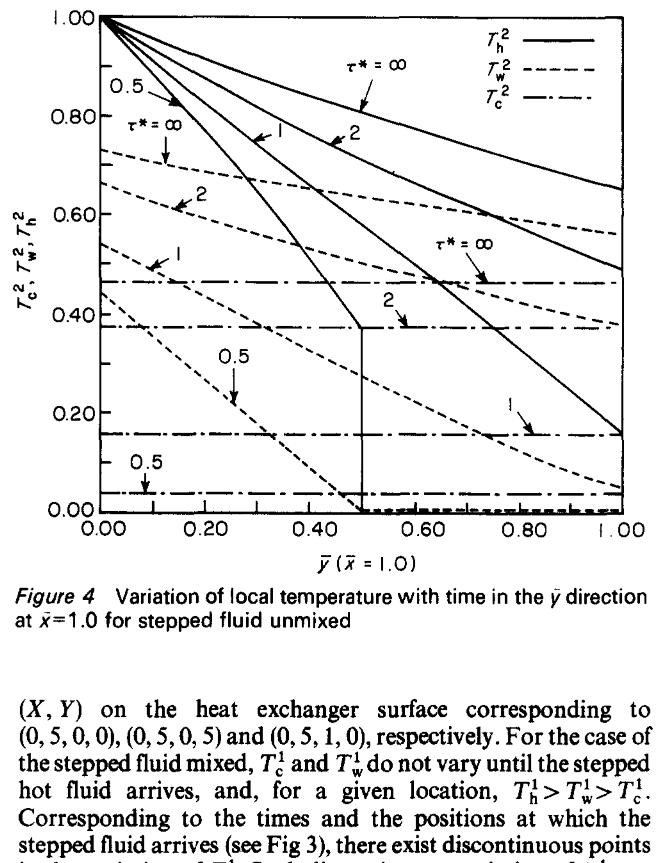

Figure 4 Variation of local temperature with time in the $\bar{y}$ direction at $\bar{x}=1.0$ for stepped fluid unmixed

(X, Y) on the heat exchanger surface corresponding to (0, 5, 0, 0), (0, 5, 0, 5) and (0, 5, 1, 0), respectively. For the case of the stepped fluid mixed, T_c^1 and T_w^1 do not vary until the stepped hot fluid arrives, and, for a given location, $T_h^1>T_w^1>T_c^1$. Corresponding to the times and the positions at which the stepped fluid arrives (see Fig 3), there exist discontinuous points in the variation of T_h^1. Such discontinuous variation of T_h^1 can also be calculated by using the analytical expression $T_h^1=\exp[-(1+R^*)N_h\bar{y}]$. This expression is found as follows. When the stepped hot fluid reaches the location $\bar{y}$ at time $\tau=y/u_h$, where u_h is the mean axial velocity of the hot fluid, the hot fluid sees $T_w^1=\tau_d^*(\partial T_h/\partial\tau^*)=0.0$ in Eqs (4). Consequently, Eq (4a) can be directly integrated from $T_h^1(\bar{y}=0)=1.0$ to $T_h^1(\bar{y})$. On comparing the results calculated from the analytical expression with those calculated from the finite difference method, the agreement is better than three decimal places. This excellent agreement establishes confidence in our numerical technique.

For the case of the stepped fluid unmixed, on account of mixing in the cold fluid, T_w^2 and T_c^2 may vary before the stepped flow arrives, and it is interesting that $T_h^2<T_w^2<T_c^2$ at location C for $\tau^*<1.0$. Fig 4 shows the variations of the local temperature in the direction $\bar{y}$ at $\bar{x}=1.0$ for the stepped fluid unmixed. It is seen that the temperature cross (ie $T_c^2>T_h^2$) may occur at location $\bar{y}<0.5$ for $\tau^*=0.5$. However, as $\tau^* \geqslant 1.0$ the hot fluid has reached the outlet, and the temperature cross disappears.

The effects of the dwell time ratio τ_d^* are shown in Fig 5. The transient parameter τ_d^* is a dwell time ratio of the hot fluid to the cold fluid. The outlet transient responses $\varepsilon_{c,o}^1$ and $\varepsilon_{c,o}^2$ are shown in the upper half of Fig 5, while $\varepsilon_{h,o}^1$, $\varepsilon_{h,o}^2$ are shown in the lower half. One can see that the smaller the value of τ_d^*, the shorter is the response time. This is because, for smaller τ_d^*, the value of $\tau_{d,h}$ is also small such that the hot fluid flows through the exchanger in a short time period. It should be noted that there is little difference between $\varepsilon_{c,o}^1$ and $\varepsilon_{c,o}^2$, for any value of τ_d^*, and there is an increasing difference between $\varepsilon_{h,o}^1$ and $\varepsilon_{h,o}^2$ as τ_d^* becomes larger.

In addition, as shown later, our numerical works also demonstrate $\varepsilon_{c,o}^1(\tau^*, \tau_d^*, \bar{C}_w^*, R^*, N_{tu}, C^*)=\varepsilon_{c,o}^2(\tau^*, \tau_d^*, \bar{C}_w^*, R^*, N_{tu}, C^*)$. Thus the outlet transient responses of the unstepped fluid for the two cases considered are the same. Recently, Romie[8] has indicated that, for counterflow heat exchangers, the normalized exit temperature history of unstepped fluid will be the same irrespective of which fluid of the exchanger is the stepped fluid. Moreover, he has also shown that this finding should apply to exchangers of all configurations. Our numerical work confirms this conclusion.

Fig 6 shows the effects of wall heat capacitance ratio $\bar{C}_w^*$. It is seen that the larger the wall capacitance ratio, the longer is the response time. The curve of $\varepsilon_{h,o}^1$ is nearly coincident with that of $\varepsilon_{h,o}^2$ in the case of $\bar{C}_w^*=20$ as shown in the lower half of Fig 6. In

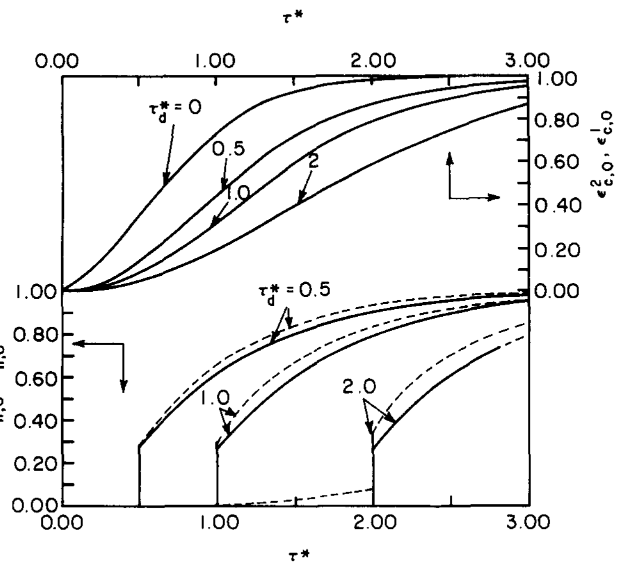

Figure 5 Effect of τ_d^*

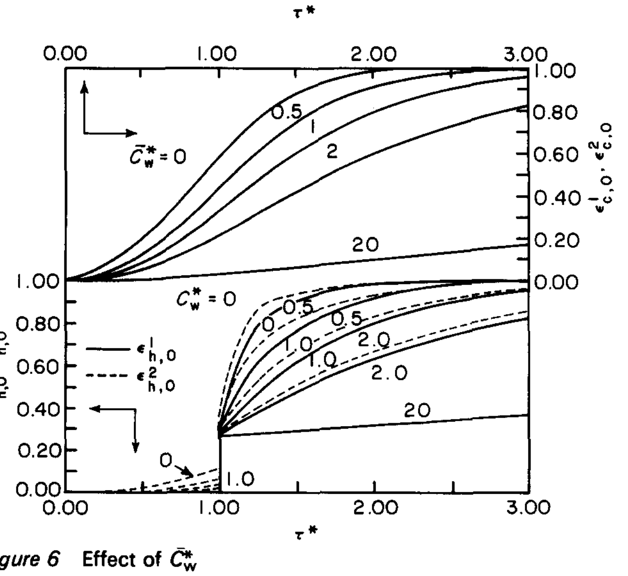

Figure 6 Effect of $\bar{C}_w^*$

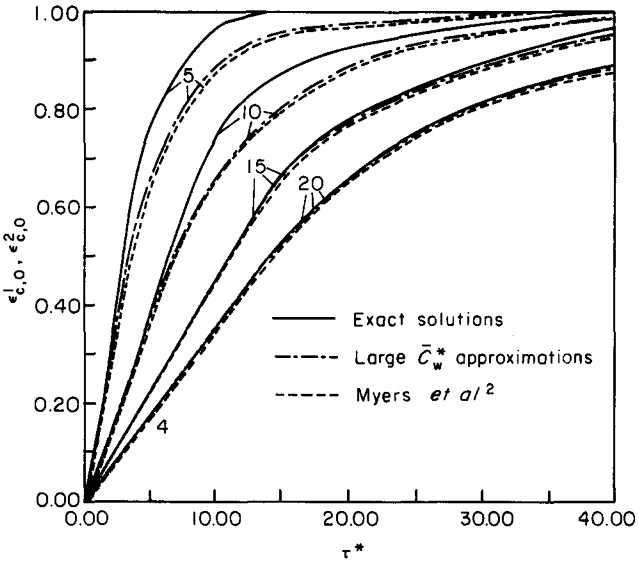

Figure 7 Comparison of exact solutions, large $\bar{C}_w^*$ approximations and Myer's solutions[2]

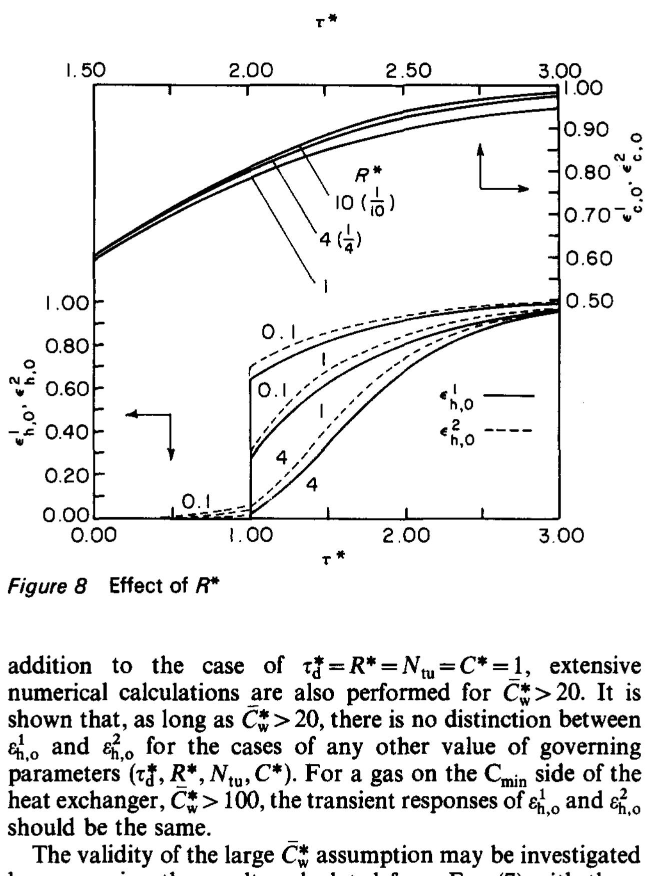

Figure 8 Effect of R^*

addition to the case of $\tau_d^*=R^*=N_{tu}=C^*=1$, extensive numerical calculations are also performed for $\bar{C}_w^*>20$. It is shown that, as long as $\bar{C}_w^*>20$, there is no distinction between $\varepsilon_{h,o}^1$ and $\varepsilon_{h,o}^2$ for the cases of any other value of governing parameters ($\tau_d^*, R^*, N_{tu}, C^*$). For a gas on the C_{min} side of the heat exchanger, $\bar{C}_w^*>100$, the transient responses of $\varepsilon_{h,o}^1$ and $\varepsilon_{h,o}^2$ should be the same.

The validity of the large $\bar{C}_w^*$ assumption may be investigated by comparing the results calculated from Eqs (7) with those calculated from Eqs (9), approximating the basic equations. The comparisons are shown in Fig 7 for $\bar{C}_w^*=5$, 10, 15 and 20. Myers's approximate integral solutions[2], indicated by dashed lines, are also included for comparison. It appears that the large $\bar{C}_w^*$ approximation agrees well with the exact solution when $\bar{C}_w^* \geqslant 20$. It is worthy of remark that the computer times for solving the large $\bar{C}_w^*$ approximations are only 10% of those for solving the basic equations.

The effects of thermal resistance ratio R^* on the outlet transient responses of the cold fluid are shown in the upper half of Fig 8. It is seen that the result for the case of $R^*=1.0$ reaches the steady state most slowly. An increase or decrease in the value of R^* reduces the response time. The effects of R^* on $\varepsilon_{h,o}^1$ and $\varepsilon_{h,o}^2$ are shown in the lower half of Fig 8. The larger the value of R^*, the longer is the response time. Further, the effects of R^* on $\varepsilon_{h,o}^2$ are more pronounced than on $\varepsilon_{c,o}^2$.

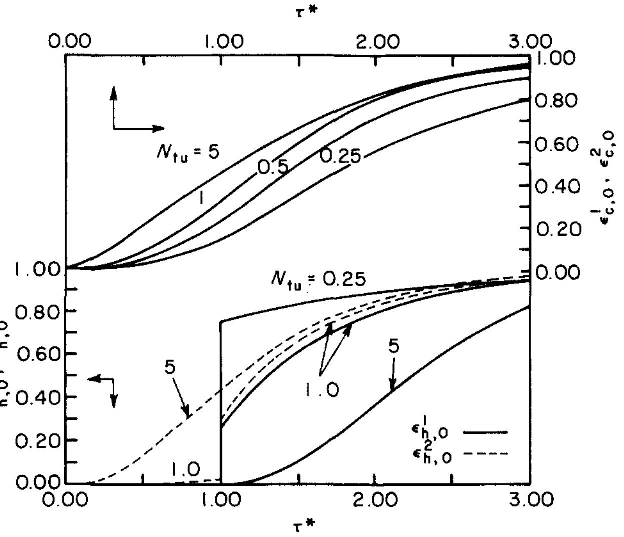

Figure 9 Effect of N_{tu}

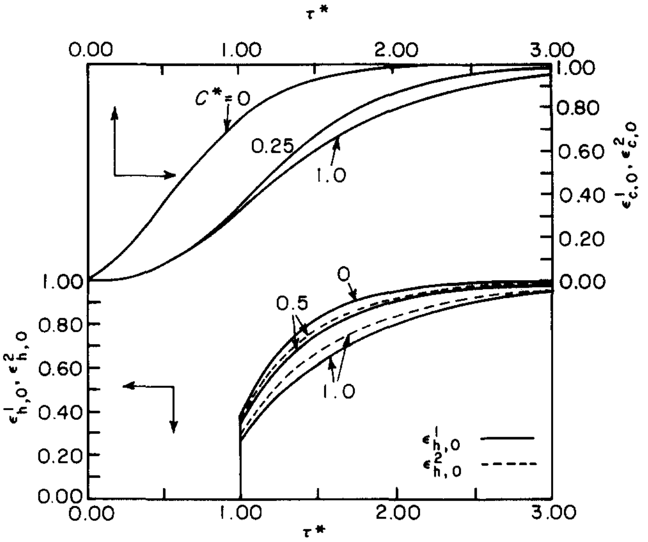

Figure 10 Effect of C^*

Fig 9 demonstrates the effects of the number of heat transfer units N_{tu}. An increase in the value of N_{tu} reduces the response time of $\varepsilon^1_{c,o}(\varepsilon^2_{c,o})$ and increases the response time of $\varepsilon^1_{h,o}$. For the case of the stepped fluid being unmixed, since the temperature cross may occur, there is no definite rule for the effects of N_{tu} on $\varepsilon^2_{h,o}$.

The effects of the capacitance rate ratio C^* are shown in Fig 10. The smaller the value of C^*, the faster the curved lines approach the value in the steady state for both $\varepsilon_{h,o}$ and $\varepsilon_{c,o}$. For the case of $C^*=0$ (condenser/evaporator), as indicated by London *et al*[9] and Myers and coworkers[2,5], the transient behaviours of any exchanger, however arranged, are the same and only dependent upon three parameters: N_{tu}, C^* and R^*. It is observed from Fig 10 that our numerical results for the case of $C^*=0$ agree with this conclusion.

Conclusions

The conclusions of this investigation can be summarized as follows.

(1) The outlet transient responses of a crossflow heat exchanger with one fluid mixed and the other unmixed have been presented for a wide range of the governing parameters τ^*_d, C^*_w, R^*, N_{tu} and C^*, and are shown in graphical form.
(2) The outlet transient responses of the unstepped fluid for the two cases investigated are the same, but the outlet transient responses of the stepped fluid are distinctly different.
(3) For the case of C^*_w very large, the numerical solutions calculated from the large wall capacitance approximation by the finite difference method are more accurate with smaller mesh points than by directly solving Eq (9) without approximating the basic equations. Therefore, the computer times needed can be significantly reduced. This approximation is regarded as sufficently valid for $C^*_w \geq 20$.

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