Entropy generation minimization of combined heat and mass transfer devices

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ABSTRACT

This paper details a simple procedure by which the entropy generation in simultaneous heat and mass exchange devices can be minimized. Effectiveness for these devices is defined and a new parameter, 'modified heat capacity rate ratio' is introduced. It is found that the entropy generation of a combined heat and mass exchange device is minimized (at constant value of effectiveness) when the modified heat capacity rate ratio is equal to one irrespective of the value of other independent parameters. Several typical examples of the cooling towers have been studied to illustrate this concept. A practical application of the concept is also illustrated using a humidification–dehumidification desalination system.

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1. Introduction

Devices in which simultaneous heat and mass exchange occur are commonly used in the power and refrigeration industries. Cooling towers used in steam or combined cycle power plants are a typical example. Humidifiers and dehumidifiers used in HVAC systems are also examples. To optimize the performance of systems containing such devices, the irreversibilities of individual devices must be minimized [1,2]. Since the total irreversibility of a system is the sum of the component irreversibilities, this procedure improves the system performance.

A few researchers [2–4] have previously attempted to optimize the second law design of heat and mass exchange devices by studying the sources of irreversibilities at the local level. Carrington and Sun [4] presented the following expression for the volumetric entropy generation rate as a sum of the three components that arise in combined heat and mass transfer processes. These include a mass transfer component, a heat transfer component and a coupled heat and mass transfer component, neglecting the irreversibilities due to fluid flow.

$$\dot{S}_{gen}''' = \frac{k}{T^2}\cdot(\nabla T)^2 + \frac{2\rho^2\overline{R}D'}{M_A M_B c}(\nabla T)\cdot(\nabla x_A) + \frac{\rho^2\overline{R}D_{AB}}{M_A M_B x_A x_B c}\cdot(\nabla x_A)^2 \tag{1}$$

The first term on the right-hand side of Eq. (1) is the heat transfer component of the total irreversibility. In heat exchangers (without phase change or mass transfer, for example) this is the only term that causes the irreversibilities, other than the fluid flow irreversibilities. Hence, by balancing the stream-to-stream temperature difference in the heat exchangers we can minimize entropy production for a fixed effectiveness. In the following section it is shown how the temperature balance minimizes entropy production in a heat exchanger. However, the same is not true for a combined heat and mass exchange device, since in some situations the second and third term (related to mass diffusion irreversibilities) in Eq. (1) can play a major role. Which component dominates the irreversibility depends on how steep the temperature and mass concentration gradients (x) are.

To exactly evaluate the contribution of each of the terms in Eq. (1) on total irreversibility one has to perform a volume integral that requires knowledge of the local temperature and mass concentration profiles throughout the device. Hence, it is useful to apply a simple control volume approach to identify the key variables governing the process. This paper develops a control volume procedure that facilitates minimization of the entropy generation in simultaneous heat and mass exchangers. The method uses an energy-based effectiveness and modified heat capacity rate ratio, which is suitable for on-design analysis of adiabatic two-stream components within a cycle.

2. Heat exchangers

Several researchers have previously studied entropy production in heat exchangers [5–12]. In this section, we draw upon this

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Nomenclature

Symbols

c	Total molar concentration [mol/m^3]
$\dot{C}$	Heat capacity rate [W/K]
c_p	Specific heat capacity at constant pressure [J/kg K]
D'	Thermal diffusion coefficient [m^2/s K]
D_{AB}	Diffusion coefficient [m^2/s]
$\dot{H}$	Total enthalpy rate [W]
h	Specific enthalpy [J/kg]
h_a	Specific enthalpy of moist air [J/kg of dry air]
h_{fg}	Latent heat of evaporation [J/kg]
HCR	Heat capacity rate ratio [-]
k	Thermal conductivity [W/m K]
$\dot{m}$	Mass flow rate [kg/s]
P	Absolute pressure [N/m^2]
$\dot{Q}$	Heat rate [W]
R	Gas constant [J/kg K]
$\bar{R}$	Universal gas constant [J/mol K]
s	Specific entropy [J/kg K]
s_{fg}	Specific entropy change of evaporation [J/kg K]
$\dot{S}_{gen}$	Entropy generation rate [W/K]
$\dot{S}'''_{gen}$	Entropy generation rate per unit volume [W/m^3 K]
T	Temperature [°C]
x	mass concentration [kg/kg]

Greek

Δ	Difference or change
ε	Component effectiveness [-]
ϕ	Relative humidity [-]
ω	Absolute humidity [kg/kg of dry air]
ρ	Density [kg/m^3]

Subscripts

a	Moist air
c	Cold stream
ct	Cooling tower or humidifier
da	Dehumidifier
da	Dry air
h	Hot stream
i	Inlet
max	Maximum
min	Minimum
o	Outlet
sat	Saturated
v	Vapor
w	Water

Superscripts

ideal	Ideal condition
mod	Modified

literature to develop an understanding of how entropy production can be minimized for heat exchangers. Conservation equations for a counterflow heat exchanger shown in Fig. 1 are as follows. It is assumed that there is no phase change in either the hot or the cold fluid streams and that the heat exchanger has no heat loss to the environment.

Energy conservation is expressed by.

$$\dot{m}_c(h_{2,c} - h_{1,c}) = \dot{m}_h(h_{2,h} - h_{1,h}) \tag{2}$$

and if the specific heat capacities are constant.

$$\dot{m}_c c_{p,c} \cdot (T_{2,c} - T_{1,c}) = \dot{m}_h c_{p,h} \cdot (T_{2,h} - T_{1,h}) \tag{3}$$

Here, we have assumed that both the streams are incompressible and that the pressure change is negligible between the inlet and outlet. The Second Law gives,

$$\dot{S}_{gen} = \dot{m}_c \cdot (s_{2,c} - s_{1,c}) + \dot{m}_h \cdot (s_{1,h} - s_{2,h}) \tag{4}$$

$$\dot{S}_{gen} = \dot{m}_c c_{p,c} \cdot \ln\left(\frac{T_{2,c}}{T_{1,c}}\right) + \dot{m}_h c_{p,h} \cdot \ln\left(\frac{T_{1,h}}{T_{2,h}}\right) \geq 0 \tag{5}$$
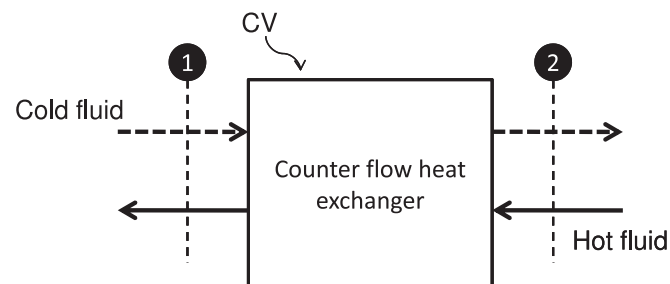

Fig. 1. Control volume for a counterflow heat exchanger.

The heat exchanger effectiveness is defined in the usual fashion as actual to maximum heat transfer:

$$\varepsilon \equiv \frac{\dot{Q}}{\dot{Q}_{max}} \tag{6}$$

$$\dot{Q}_{max} = (\dot{m} \cdot c_p)_{min} \cdot (T_{2,h} - T_{1,c}) \tag{7}$$

We define the heat capacity rate ratio (HCR) as.

$$HCR \equiv \frac{\dot{m}_c \cdot c_{p,c}}{\dot{m}_h \cdot c_{p,h}} \equiv \frac{\dot{C}_c}{\dot{C}_h} \tag{8}$$

Using these equations we can write the entropy generation in two possible ways,

Case I, $\dot{C}_c < \dot{C}_h$:

$$\frac{\dot{S}_{gen}}{\dot{C}_c} = \frac{1}{HCR} \cdot \ln\left\{1 - \varepsilon \cdot HCR\left(1 - \frac{T_{1,c}}{T_{2,h}}\right)\right\} + \ln\left\{1 + \varepsilon\left(\frac{T_{2,h}}{T_{1,c}} - 1\right)\right\} \tag{9}$$

Case II, $\dot{C}_h < \dot{C}_c$:

$$\frac{\dot{S}_{gen}}{\dot{C}_h} = HCR \cdot \ln\left\{1 + \frac{1}{HCR}\varepsilon\left(\frac{T_{2,h}}{T_{1,c}} - 1\right)\right\} + \ln\left\{1 - \varepsilon\left(1 - \frac{T_{1,c}}{T_{2,h}}\right)\right\} \tag{10}$$

Hence, it can be seen that entropy production depends on these parameters:

$$\frac{\dot{S}_{gen}}{\dot{C}_{min}} = f\left\{\left(\frac{T_{2,h}}{T_{1,c}} - 1\right), HCR, \varepsilon\right\} \tag{11}$$