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Convection heat transfer in a shell-and-tube heat exchanger using sheet fins for effective utilization of energy

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ABSTRACT

Convection heat transfer in a shell-and-tube heat exchanger using sheet fins is numerically investigated. Heat and mass transfers in the heat exchanger are modeled under steady state to estimate the heat transfer rate and the pressure drop for various geometries of the heat exchanger. Based on the numerical results, the Nusselt number and pressure drop are formulated for practical applications. For convenience, similar expressions to those of conventional shell-and-tube heat exchangers, that is, the functions of dimensionless numbers such as the Reynolds number, are derived. In these equations, the geometry of the heat exchanger, fin efficiency, and contact thermal resistance are included as major factors. On formulating the equation for the overall heat transfer rate, it is found that the heat transfer coefficient for the heat exchanger with a fin does not correspond to the combination of the heat transfer coefficient of bare tube surface and the fin. This is because the heat exchange area is substantially limited especially at the narrow space between the tube and the fin. A correction factor for the substantial heat transfer area is therefore introduced. These formulated equations are helpful for installing sheet fins in manufactured heat exchangers. Using the formulated equations, effective conditions to enhance heat transfer rate by the fin are established, taking into account the increase in pressure drop.

1. Introduction

Effective utilization of energy is one of the most important topics relating to limited fossil-fuel resources. While much energy is consumed in industrial processes, large amounts of waste heat at low temperatures are released. To recover waste heat effectively, improvement of heat exchangers is still needed even though they have been studied for a long time. There are two main methods to improve the overall heat exchange rate, the extension of the heat transfer area and increasing the heat transfer coefficient. In the former method, fins are commonly used [1]. In the latter case, vortex generators are used to generate local turbulent flow [2]. These methods successfully improve the overall heat transfer rate and can be used for newly manufactured heat exchangers. However, there are not enough methods to improve the heat transfer rates of heat exchangers that are already in use.

In our previous study, an exchangeable sheet fin was developed for manufactured shell-and-tube heat exchangers. Fig. 1 shows a schematic of a heat exchanger with a sheet fin. The fin is zigzagged through the tube bank. When fin degrades under corrosive conditions, it can be replaced by a new one using low-cost materials such as stainless steel. Thus, corrosive-resistant materials such as titanium need not be used. The relationship between the overall heat transfer rate and contact thermal resistance was discussed in [3].

As seen in Fig. 1, fluid flow in the shell side seems to be stabilized by the fin, which might result in a decrease in the heat transfer coefficient. However, the overall heat transfer rate is given by not only heat transfer coefficient but also heat exchange area. In addition, stabilization of the fluid flow by the fin might not be a disadvantage from the viewpoint of the pressure drop, which is an important parameter in evaluating a heat exchanger. To investigate these matters, it is helpful to create expressions for estimating the heat transfer rate and pressure drop. It would also be helpful in practical use. While the heat transfer area can be calculated by the geometry of the heat exchanger, the heat transfer coefficient depends on the operating conditions such as the flow. Consequently, convection heat transfer in a shell-and-tube heat exchanger using a fin is numerically investigated in this study.

2. Numerical

2.1. Model description

Fig. 2 shows the schematic of the computational domain. The fin is shown by the dashed lines in the figure. It is assumed that

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Nomenclature

A	heat transfer area (m^2)
a	constant in Eq. (3)
b	constant in Eq. (3)
C_p	specific heat (J/kg K)
C_f	constant in Eq. (3)
C_H	constant in Eq. (5)
$C_1, C_2, C_{1\varepsilon}, C_{3\varepsilon}$	constants in the transport equation for the dissipation rate in Table 1
c	constant in Eq. (5)
D_0	tube diameter (m)
d	coefficient in Eq. (5)
E	internal energy (J)
G_b	generation of the turbulence kinetic energy because of buoyancy
G_k	generation of turbulence kinetic energy because of the mean velocity gradients
G_{max}	maximum fluid mass velocity in the tube bank (kg/m^2 s)
h	heat transfer coefficient (W/m^2 K)
k	turbulent energy (m^2/s^2)
l	distance (m)
l_p	tube pitch (m)
N	fin parameter
N_L	number of tube lines in the flow direction
Nu	Nusselt number
n	normal direction
P	pressure (Pa)
Pr	Prandtl number
Q	heat transfer rate (W)
R_c	thermal contact resistance (m^2 K/W)
Re	Reynolds number
T	temperature (K)
U	overall heat transfer coefficient (W/m^2 K)
V	flow velocity (m/s)
V_{max}	maximum flow velocity (m/s)

Greek symbols

β	correction factor defined in Eq. (6)
δ	fin thickness (m)
ΔT_{lm}	logarithmic mean temperature difference (K)
ε	turbulence kinetic energy dissipation rate, (m^2/s^3)
θ	half angle of contact part between tube and fin (rad)
λ	thermal conductivity (W/mK)
μ	viscosity (Pa s)
μ_t	coefficient of eddy viscosity (m^2/s)
ν	kinetic viscosity (m^2/s)
σ	dimensionless tube pitch
σ_k	turbulent Prandtl number for turbulence kinetic energy
σ_ε	turbulent Prandtl number for the turbulence kinetic energy dissipation rate
$\bar{\bar{\tau}}$	stress tensor (Pa)
ϕ	correction factor for the substantial heat transfer area
Ψ	coefficient of Eq. (5)

Subscripts

a	air
f	fin
h	heat transfer fluid
inlet	inlet
L	longitudinal direction
T	transversal direction
t	heat transfer tube
w	wall surface of the heat transfer tube
x	x-direction
y	y-direction
0	no-fin case
1	normal to the flow direction
2	flow direction

the tube is very long. Thus, the wall effect in the longitudinal direction of the tube can be ignored. In addition, the number of tube rows in the flow direction is ten in terms of development of the flow in shell side. Two tube rows were considered in the computational domain.

Calculations under steady-state turbulence were carried out assuming an incompressible fluid. For the governing equations, mass, energy, and momentum conservation equations were used for the fluid. A realizable k–ε turbulence model [4] was used. A heat conduction equation for the fin was also implemented. As boundary conditions, the wall function known as the enhanced wall treatment [5] was applied to the solid surface. The temperature at the tube surface was constant at T_w. For the top and bottom of the computational domain, a periodic boundary condition was applied. As a boundary condition between the tube wall and fin, the contact thermal resistance between the tube and fin R_c was included and given by the following equation:

$$\frac{1}{R_c}(T_w - T_f) = -\lambda_f \frac{\partial T_f}{\partial n}, \tag{1}$$

where T_f, λ_f, and n are the temperature of the fin, thermal conductivity of the fin, and normal direction, respectively.

Calculations were carried out under the following conditions based on the experiment in the previous study [3]. The diameter of the heat transfer tube D_0 was 10.5 mm. The upstream and downstream sections were 1200 mm and 400 mm from the tube bank, respectively, to reduce the effect of velocity and temperature distributions on the flows at the entrance and exit of the tube bank. The inlet fluid temperature T_0 was 293 K. The temperature of the tube wall surface T_w was 333 K. The thermal conductivity λ_a and specific heat of the fluid $C_{p,a}$ were 0.0242 W/m K and 1.006 kJ/kg K, respectively. The maximum flow velocity between the tubes, u_{max} $(=\{l_{p,y}/(l_{p,y} - D_0)\}u_0)$ was set to 7.3–21.9 m/s, which is

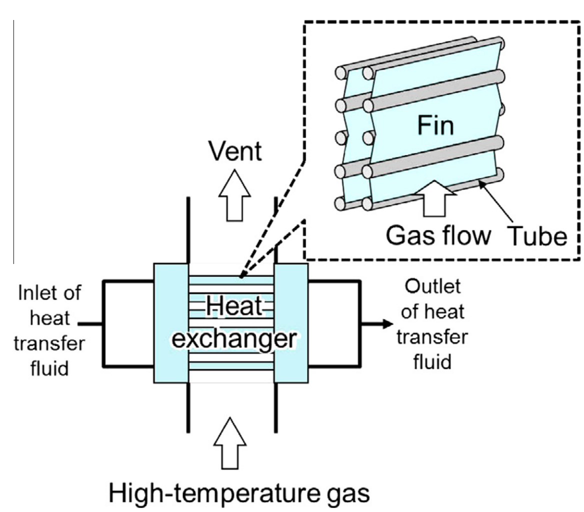

Fig. 1. Schematic diagram of heat exchanger with a sheet fin.