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Flow characteristics and thermal performance in chevron type plate heat exchangers

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ABSTRACT

We report an unsteady numerical analysis using large-eddy simulation (LES) to investigate the fluid flow in chevron-type plate heat exchangers. The flow in the heat exchanger consisted of a streamwise direction component and furrow direction components, and this complex flow structure induced turbulence, even at low Reynolds numbers. In the turbulent regime, the flow periodically oscillated, inducing a secondary flow and a vortex pair. The friction factor f and Colburn factor j were investigated for various geometrical parameters, i.e., $30° \leqslant \beta \leqslant 60°$ and $2.0 \leqslant p/h \leqslant 4.4$. The results showed that both f and j increased with increasing chevron angle β, and both increased as the ratio of the chevron pitch to the chevron height p/h decreased. We found that to achieve optimum performance, the following parameters should be used: $\beta = 30°$ with laminar flow and $\beta = 60°$ with turbulent flow; furthermore, $p/h = 2.0$ should be used in both laminar and turbulent flows.

1. Introduction

Various plate configurations have been investigated to improve the performance of compact heat exchangers. In particular, chevron-type plate heat exchangers have been used in various industrial fields because of their favorable performance and durability. Repetitive corrugations in chevron-type plate heat exchangers lead to turbulent flow, which increases the heat transfer rate. In addition, since a chevron-type plate heat exchanger has repetitive contact points, the plates are highly durable, which leads to a large maximum allowable pressure. However, there is a large pressure drop between the inlet and outlet of the heat exchanger due to the turbulent flow. To understand the characteristics of the heat transfer and pressure drop, a detailed analysis of the flow patterns in the heat exchanger is required.

Focke and Knibbe [1] and Gaiser and Kottke [2] experimentally investigated flow patterns in chevron-type plate heat exchangers with different chevron angles using flow visualization. Dovic et al. [3] investigated the flow characteristics using flow visualization with chevron angles of $\beta = 28°$ and $\beta = 61°$. However, these studies only examined the macroscopic flow patterns; local details of the flow in the heat exchanger were not considered. Dovic et al. [3] reported friction factor f and Nusselt number Nu correlations using a mathematical model. Focke et al. [4] also suggested f and Colburn factor j correlations as functions of the chevron angle. Muley and Manglik [5], Okada et al. [6], Heavner et al. [7], and Khan et al. [8] reported f and Nu correlations based on experimental investigations. However, while these studies considered the effects of the chevron angle, the effects of the chevron pitch and height were not examined.

Many numerical investigations of the fluid flow in chevron-type plate heat exchangers have been carried out. Ciofalo et al. [9] conducted a numerical analysis using a standard k–ε turbulence model, a low Reynolds number turbulence model, direct numerical simulation, and large-eddy simulation (LES); they reported that numerical analysis using LES provided the best agreement with the experimental data. Blomerius et al. [10] investigated the characteristics of self-sustained oscillatory flow and the frequency spectrum of the flow as a function of the Reynolds number using an unsteady numerical analysis. However, a study of the local flow pattern with respect to time was not carried out. Pelletier et al. [11] performed numerical analyses using a Reynolds-averaged Navier–Stokes (RANS) model under steady-state conditions, and reported that the results with a constant heat flux boundary condition were more accurate than those with a constant temperature boundary condition. Jain et al. [12] investigated the temperature and velocity distributions in a corrugated channel using a realizable k–ε turbulent model with steady-state conditions.

There remains a lack of studies investigating the local details of the fluid flow in chevron-type plate heat exchangers. Furthermore, most reports have emphasized the effects of the chevron angle

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Nomenclature

C_p	specific heat at a constant pressure (J/kg K)
Cr	Courant number, $u\Delta t/\Delta x$
D_{eq}	equivalent diameter (m)
f	friction factor, $\frac{\Delta p}{2\rho u_c^2}\left(\frac{D_{eq}}{L_m}\right)$
f_s	frequency (1/s)
G	filter function
h	chevron height (m) heat transfer coefficient (W/m^2 K)
j	Colburn factor, $\frac{h}{\rho u_c C_p}\mathrm{Pr}^{2/3}$
JF	JF factor, $\frac{j/j_{\mathrm{ref}}}{(f/f_{\mathrm{ref}})^{1/3}}$
L_w	plate width (m)
Nu	Nusselt number, hD_{eq}/k
p	chevron pitch (m) pressure (Pa)
Pr	Prandtl number
q_j	subgrid-scale heat flux (W/m^2)
Re	Reynolds number based on equivalent diameter, $\rho u_c D_{eq}/\mu$
Str	Strouhal number, $f_s h/u_c$
T	temperature (K)
t	time (sec)
u, v, w	velocity (m/s)
x, y, z	coordinate

Greek symbols

β	chevron angle (deg)
Δ	delta
δ_{ij}	Kronecker delta
λ	thermal conductivity (W/m K)
μ	dynamic viscosity (kg/m s)
μ_t	subgrid-scale turbulent viscosity
ρ	density (kg/m^3)
σ_{ij}	stress tensor
τ_{ij}	subgrid-scale stress
τ	shear stress
ϕ	variables

Subscripts

c	cross-sectional area
i, j, k	coordinate
ref	reference
t	turbulent

only. Very few studies have considered the effects of the chevron pitch and chevron height.

We investigate the local details of the fluid flow in chevron-type plate heat exchangers using large-eddy simulation. The friction factor and Colburn factor were studied as functions of various geometrical parameters, and we conducted performance evaluation of the heat exchanger considering the heat transfer coefficient and pressure drop.

2. Chevron geometry and problem formulation

Fig. 1 shows a schematic diagram of the chevron-type plate heat exchanger. Sinusoidal corrugations were constructed with various chevron angles β, pitches p, and heights h, and were repeated in the flow direction (i.e., the x-axis). Because the corrugation directions of the upper and lower plates were opposite, repetitive contact points existed between the plates. We considered the chevron angle β and the ratio of the chevron pitch to the chevron height p/h as dimensionless geometrical parameters. The operating conditions and geometrical parameters are listed in Table 1, and reflect those typically used in industrial applications. Reynolds numbers were randomly selected within this range. The characteristic length of the channel was the equivalent diameter D_{eq} [5,13], which is defined as follows:

$$D_{eq} = 2h \quad (\because h \ll L_w) \tag{1}$$

2.1. Numerical domain and boundary conditions

Fig. 2 shows the numerical domain used to investigate the fluid flow and heat transfer. Because the geometry shown in Fig. 2 is repeated in the streamwise (x-axis) and spanwise (z-axis) directions, the unit cell shown in Fig. 2 was used as the numerical domain, and periodic boundary conditions were employed in both the streamwise and spanwise directions. The pressure gradient in the streamwise direction was used to investigate the flow characteristics. No-slip boundary conditions were applied at the walls, and constant heat flux boundary conditions were used at the top and bottom surfaces in order to examine the convective heat transfer characteristics.

2.2. Governing equations

Blomerius et al. [10] reported that unsteady self-sustained oscillatory flow occurs in the channel of chevron-type plate heat exchangers. Therefore, we used a large-eddy simulation (LES) to investigate the time-dependent fluid flow. For the numerical analysis, following assumptions were applied.

(1) The flow is three-dimensional, time-dependent, and incompressible.
(2) The working fluid is water with constant properties.
(3) Natural convection and radiation are neglected.

The governing equations for the LES were the spatially filtered time-dependent three-dimensional Navier–Stokes equations. Eddies that were larger than the filter size (i.e., the grid size in this study) were resolved using the Navier–Stokes equations, whereas eddies that were smaller than the filter size were modeled. A filtered variable, denoted by the over-bar, is defined by [14]

$$\bar{\phi}(x) = \int \phi(x')G(x,x')dx' \tag{2}$$

where $G(x,x')$ is the filter function, which determines the scale of the resolved eddies. The filtered Navier–Stokes equations are as follows:

Continuity:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho \bar{u}_i) = 0 \tag{3}$$

Momentum:

$$\frac{\partial}{\partial t}(\rho \bar{u}_i) + \frac{\partial}{\partial x_j}(\rho \bar{u}_i \bar{u}_j) = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial \sigma_{ij}}{\partial x_j} - \frac{\partial \tau_{ij}}{\partial x_j} \tag{4}$$

where σ_{ij} is the stress tensor by molecular viscosity and τ_{ij} is the subgrid-scale stress, i.e.,

$$\sigma_{ij} = \left[\mu\left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right)\right] - \frac{2}{3}\mu\frac{\partial \bar{u}_i}{\partial x_i}\delta_{ij} \tag{5}$$

$$\tau_{ij} = \rho\overline{u_i u_j} - \rho\bar{u}_i\bar{u}_j \tag{6}$$