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Numerical study of mixed convection heat transfer for vertical annular finned tube heat exchanger with experimental data and different tube diameters

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ABSTRACT

The commercial software along with the inverse method and the experimental data is used to obtain the mixed convection heat transfer and fluid flow characteristics of the single-tube vertical annular finned tube heat exchanger with different tube diameters. Various flow models along with near-wall treatments are introduced to investigate their effects on the obtained numerical results. The inverse method of the finite difference method along with the experimental data is first applied to determine the fin temperature and the heat transfer coefficient for a smaller tube. Afterwards, the air temperature and velocity profiles, the fin temperature distribution and the heat transfer coefficient on the fins are determined by the commercial software along with various flow models. More accurate results are obtained if the resulting heat transfer coefficient and the fin temperature at the measurement locations are respectively close to the inverse results and the experimental data. In addition, the comparison between the obtained air velocity pattern and the existing experimental pattern is desirable. An important finding is that the choice of appropriate flow models, near-wall treatments, friction factor, y^+ value and the number of grid points varies with air velocity and tube diameter to obtain more accurate results. The heat transfer coefficient increases with increasing air velocity and fin spacing. However, the friction factor increases as the fin spacing increases and the air velocity decreases. The two vortices are symmetrical in the wake region behind the tube and rotate in the opposite direction. Increasing the number of grid points may not necessarily obtain more accurate results. The proposed correlation of the heat transfer coefficient is closer to the obtained inverse and numerical results than the existing correlation. To our knowledge, few researchers have used the similar method to investigate this problem in the open literature.

1. Introduction

Annular finned tube heat exchangers are commonly used in industry. In the design of such heat exchangers, it is necessary to note the interaction of local heat transfer and flow distribution between two adjacent fins. Many numerical and experimental methods have been proposed to obtain the heat transfer and fluid flow characteristics of such heat exchangers. Sparrow and Samie [1] and Rosman et al. [2] applied the experimental method combined with the heat-mass transfer analogy to determine the local and average heat transfer coefficients of plate fin and tube heat exchangers. However, the temperature of all tubes is assumed to be the same in Refs. [1,2]. Rocha et al. [3] and Saboya and Saboya [4] adopted the naphthalene sublimation technique and heat-mass analogy to determine local and global heat and mass transfer coefficients on elliptical and circular fins of one-row and two-row tubes and plate-fin heat exchangers. Sung et al. [5] also used the naphthalene sublimation technique to obtain the local mass transfer coefficient on the two transverse annular fins attached to the cylinder in cross flow. Hu and Jacobi [6] applied the naphthalene sublimation technique to obtain the local heat and mass transfer coefficients on the transverse annular fins in the crossflow for Reynolds number from 3300 to 12,000. Watel et al. [7] investigated the effect of fin spacing and flow velocity on the forced convective heat transfer of a single annular-finned tube by using Particle Image Velocimetry (PIV) and infrared thermography. They got a valuable correlation between the average Nusselt number on the fins, the dimensionless fin spacing and the Reynolds number. Their results

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Nomenclature

A_f	lateral surface area of the fin, m^2
c_p	specific heat of air, kJ/(kg K)
d_0	outer diameter of the circular tube, mm
D	outer diameter of the annular fins, mm
f	friction factor defined in Eq. (27)
g_i	gravitational acceleration component, m/s^2
h	local heat transfer coefficient on fins, $W/(m^2\ K)$
$\bar{h}$	average heat transfer coefficient on fins, $W/(m^2\ K)$
$\bar{h}_b$	heat transfer coefficient under the situation of T_0, $W/(m^2\ K)$
$\bar{h}_j$	heat transfer coefficient in the jth sub-fin region, $W/(m^2\ K)$
k	turbulent kinetic energy
k_a	thermal conductivity of air, W/(m K)
k_f	thermal conductivity of fins, W/(m K)
N	number of sub-fin regions
N_r, N_θ, N_{zf}	number of grid points on the fins in the r, θ and z directions
N_t	total number of grid points
Nu_d	Nusselt number, $Nu_d = \bar{h}d_0/k_a$
N_{za}	number of grid points between two adjacent fins in the z direction
p	pressure (Pa)
Pr	Prandtl number, $\Pr = \rho c_p v/k_a$
Pr_t	turbulent Prandtl number
Q	total heat rate dissipated from the entire fin, W
Ra	Rayleigh number, $Ra = \frac{g\beta(T_0-T_\infty)S^3}{v\alpha}\left(\frac{S}{D}\right)$
Re_d	Reynolds number, $Re_d = V_a d_0/v$
S	fin spacing, mm
S_{ij}	mean strain rate tensor, $S_{ij} = (\partial u_i/\partial x_j + \partial u_j/\partial x_i)/2$
T	fin temperature, K
T_a	air temperature, K
T_j	measured fin temperature at the jth measurement location, K
T_0	fin base temperature, K
T_∞	ambient air temperature, K
t	fin thickness, mm
u_i	air velocity component, m/s
V_a	frontal air velocity, m/s
x, y, z	spatial coordinates, m
y^+	dimensionless wall distance

Greek symbols

β	volumetric thermal expansion coefficient
ε	viscous dissipation rate of turbulence kinetic energy
v	laminar kinematic viscosity, m^2/s
v_{eff}	effective kinematic viscosity, m^2/s
v_t	turbulent kinematic viscosity, m^2/s
ρ	air density, kg/m^3

Superscripts

a	in the air region
f	in the fin region
mea	measured data

indicate that a reduction in fin spacing leads to a reduction in heat transfer for a fixed Reynolds number. Şabin et al. [8] employed the PIV technique to investigate the flow structure in plate fin and tube heat exchanger model consisting of a single cylinder located between two parallel plates with duct height-to-cylinder diameter ratio of 0.365 for the Reynolds numbers of 4000 and 7500.

It is seen from Refs. [8–13] that the very complex three-dimensional (3D) flow characteristics can appear in the plate finned-tube heat exchanger. The flow characteristics in the flow passage of the heat exchanger are strongly influenced by the presence of both the cylinder and the fins. These complex flow patterns caused by horseshoe vortices at the corner junction are of interest for the study of local heat transfer enhancement mechanism. The flow accelerates around the heated horizontal annular finned tube in a cross flow and forms a low-velocity wake region behind the tube. The boundary layer above the heated horizontal tube begins to develop in front of the tube and increases in thickness along the circumference of the tube. Thus, the forced convective heat transfer coefficient is the highest in the upstream region of the fins and is the lowest in the wake region of the fins, as shown in Refs. [14–16]. This means that the heat transfer coefficient on the fins is non-uniform. Thus, a more accurate estimate of the heat transfer coefficient on the fins is an important task for high-performance heat exchanger equipment. Reliable results can help designers develop more advanced heat exchanger technology to improve its performance. More importantly, the availability of numerical results obtained from commercial software should be based on experimental temperature data and reliable heat transfer coefficient obtained from inverse methods or correlations.

Torokoshi and Xi [9] applied the SIMPLE algorithm to investigate the 3D unsteady laminar and thermal fields of the staggered two-row plate finned tube heat exchangers. Mon and Gross [10] used 3D numerical study to investigate the effect of fin spacing on four-row in-line and staggered annular-finned tube bundles. The heat transfer and fluid flow characteristics are predicted using the commercial software FLUENT. The flow between the fins is considered to be laminar, but the rest of the bundle is treated as a turbulent region. The upstream boundary of the computational domain is located at 1.2 times fin diameter from the center of the first row while the downstream boundary is set at 3.6 times the fin diameter from the centerline of the last row. It is found from Ref. [10] that the one-dimensional heat conduction equation is assumed. The effect of the total number of grid points on the resulting numerical results is also considered to be small. Bilirgen et al. [11] used FLUENT to determine the heat transfer coefficient and pressure drop for a single row of annular finned tubes in cross-flow at a uniform velocity of 4.5 m/s only in the x direction. A grid independence study is performed to determine adequate grid density. The grid density is gradually increased until the relative change in pressure and velocity fields is less than 5%. In this case, the grid point is considered to be sufficient to provide an accurate solution. It is found in Refs. [10,11] that from 50,000 to 99,000 cells and from 66,000 to 121,000 cells are used to discretize the entire computational domain, respectively. Also, the fluid flow between the two adjacent fins is considered to be laminar, but the rest of this region is considered to be a turbulent region. Mon and Gross [10] and Bilirgen et al. [11] did not compare the numerical results obtained with the experimental data. Buyruk et al. [12] presented the numerical and experimental study of the laminar flow and heat transfer around a tube in cross-flow at low Reynolds number from 120 to 390. It is found from Refs. [12,17] that the buoyancy effect is not considered.