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Heat transfer characteristics of flat plate finned-tube heat exchangers with large fin pitch

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Abstract

The objective of this study is to provide experimental data that can be used in the optimal design of flat plate finned-tube heat exchangers with large fin pitch. In this study, 22 heat exchangers were tested with a variation of fin pitch, number of tube row, and tube alignment. The air-side heat transfer coefficient decreased with a reduction of the fin pitch and an increase of the number of tube row. The reduction in the heat transfer coefficient of the four-row heat exchanger coil was approximately 10% as the fin pitch decreased from 15.0 to 7.5 mm over the Reynolds number range of 500–900 that was calculated based on the tube diameter. For all fin pitches, the heat transfer coefficient decreased as the number of tube row increased from 1 to 4. The staggered tube alignment improved heat transfer performance more than 10% compared to the inline tube alignment. A heat transfer correlation was developed from the measured data for flat plate finned-tubes with large fin pitch. The correlation yielded good predictions of the measured data with mean deviations of 3.8 and 6.2% for the inline and staggered tube alignment, respectively.

Keywords: Experiment; Heat exchanger; Finned tube; Geometry; Heat transfer coefficient

Echangeurs de chaleur à plaques à tubes ailetés à distance d'alignement importante

Mots clés : Expérimentation ; Échangeur de chaleur ; Tube aileté ; Géométrie ; Coefficient de transfert de chaleur

1. Introduction

With increasing emphasis on energy savings, extensive efforts are being made to enhance the heat transfer performance of heat exchangers. The heat exchanger performance often is limited by the air-side heat transfer coefficients, because those coefficients are naturally lower than the refrigerant-side values. Therefore, many active and passive methods have been developed to improve the air-side heat transfer performance to reduce heat exchanger volume and manufacturing costs. Air-conditioning system utilizes enhanced finned-tube heat exchangers with small fin pitch, while refrigerators or freezers use flat plate finned-tube heat exchangers with large fin pitch to obtain high performance and reliability during the frosting and

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Nomenclature

A	surface area (m^2)
A_c	minimum free flow area (m^2)
C_P	specific heat (kJ/kg K)
D_c	collar diameter
DF	dimensionless fin pitch (F/D_0)
D_h	hydraulic diameter (mm) ($D_h = 4A_cL/A$)
D_0	tube diameter (mm)
F	fin pitch (mm)
F_c	correction factor
G	mass flux (kg/m^2 s)
j	Colburn j-factor
h	heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
L	core depth (m)
LMTD	log mean temperature difference (K)
$\dot{m}$	mass flow rate (kg/s)
N	number of tube row
Pr	Prandtl number
$\dot{Q}$	capacity (W)
Re_{Dc}	Reynolds number based on collar diameter (GD_c/μ)
Re_{Dh}	Reynolds number based on hydraulic diameter (GD_h/μ)
Re_{Do}	Reynolds number based on tube diameter (GD_0/μ)
r	tube radius (mm)
T	temperature (°C, K)
x	flat plate length (cm)
U	overall heat transfer coefficient (W/m^2 K)

Greek symbols

δ	boundary layer thickness (cm)
μ	dynamic viscosity coefficient (Ns/m^2)
η_0	surface effectiveness

Subscripts

air	air
i	inside
in	inlet
loss	loss
mea	measured
ref	refrigerant
o	outside
out	outlet
pre	predicted

defrosting processes. Since most existing heat transfer data and correlations were developed for small fin-pitched heat exchangers, more comprehensive studies on heat exchangers with large fin pitches are required to develop a design tool for heat exchangers in refrigerators and freezers.

Experimental and numerical studies on the influence of fin pitches and tube alignment on the heat transfer performance have been conducted extensively in finned-tube heat exchangers with small fin pitches ($F < 8.7$ mm). Rich [1,2] studied the effects of both fin pitches and the number of tube rows for flat plate finned-tube geometry. He found that the Colburn j-factor decreased by 50% when the fin pitch was reduced from 8.7 to 1.23 mm. In addition, the Colburn j-factors became smaller for each successive row in the low Reynolds number range. Wang et al. [3,4] examined the effects of the number of tube rows, fin pitch, and tube diameter on the thermal hydraulic characteristics, and developed a correlation for finned-tube heat exchangers having plain fin geometry. They found that the effect of fin pitches on the Colburn j-factor was negligible for $N \geq 4$ and $Re_{Dc} \geq 2000$. Mon and Gross [5] studied the effects of fin pitches on four-row annular finned-tube bundles in staggered and inline arrangements. They found that the boundary layer and horseshoe vortices between the fins were substantially dependent on Reynolds number and the ratio of fin pitch to height. Saboya and Sparrow [6–8] measured the local heat transfer coefficients for one-row, two-row and three-row plate finned-tube heat exchangers using the naphthalene mass transfer method. Kim and Song [9] found that the total heat and mass transfer rate from the plate increased with F/D, and it reached saturation beyond $F/D = 0.5$.

Due to its performance and reliability under frosting and defrosting conditions, the continuous flat plate finned-tube with large fin pitch has been widely used in refrigerators. However, studies focusing on heat transfer characteristics on the air-side of flat plate finned-tube heat exchangers equipped with large fin pitches are very limited. The major objective of this study was to experimentally investigate the effects of fin pitches, the number of tube rows, and tube alignment on the heat transfer performance of flat plate finned-tube heat exchangers with large fin pitches. In addition, the present investigation aimed at developing a heat transfer correlation that can be used in the optimal design of evaporators for refrigerators and freezers.

2. Experimental setup and test conditions

A schematic of the experimental setup used to determine the characteristics of the heat transfer is shown in Fig. 1. The equipments were installed in a psychrometric chamber to provide a pre-controlled ambient temperature. The psychrometric chamber was maintained at 2 °C using an air-handling unit including a cooling coil, a heating coil, and a