Energy Conversion and Management 85 (2014) 470–476

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Experimental study of thermal–hydraulic performance of cam-shaped tube bundle with staggered arrangement

Hamidreza Bayat [a,*], Arash Mirabdolah Lavasani [b], Taher Maarefdoost [a]

[a] Young Researchers and Elite Club, Central Tehran Branch, Islamic Azad University, Tehran, Iran.
[b] Department of Mechanical Engineering, Central Tehran Branch, Islamic Azad University, Tehran, Iran

ARTICLE INFO

Article history:
Received 20 February 2014
Accepted 2 June 2014
Available online 22 June 2014

Keywords:
Heat exchanger
Tube bundle
Experimental heat transfer
Cam-shaped tube
Cross-flow

ABSTRACT

Flow and heat transfer from cam-shaped tube bank in staggered arrangement is studied experimentally. Tubes were located in test section of an open loop wind tunnel with two longitudinal pitch ratios 1.5 and 2. Reynolds number varies in range of $27,000 \leqslant Re_D \leqslant 42,500$ and tubes surface temperature is between 78 and 85 °C. Results show that both drag coefficient and Nusselt number depends on position of tube in tube bank and Reynolds number. Tubes in the first column have maximum value of drag coefficient, while its Nusselt number is minimum compared to other tubes in tube bank. Moreover, pressure drop from this tube bank is about 92–93% lower than circular tube bank and as a result thermal–hydraulic performance of this tube bank is about 6 times greater than circular tube bank.

1. Introduction

Study of flow and heat transfer around single and multiple bluff bodies has wide engineering applications such as heat exchangers, cooling towers, and electronic cooling. There are several authors who published books about flow and heat transfer phenomena around bluff bodies such as Kays and London [1], Hoerner [2], Zukauskas and Ulinskas [3], Zukauskas and Ziugzda [4], and Zdravkovich [5,6].

Traub [7] reported that turbulence grids lead to an enhancement in heat transfer of plain tube bundles. Stanescu et al. [8] found that increasing Re_D decreases the optimal spacing of cylinder to cylinder. Wilson and Bassiouny [9] suggested to choose longitudinal pitch ratio $a \leqslant 3$ for circular tube bank, in order to have best performance and compactness. The studies of Mandhani et al. [10] showed that decreasing value of porosity and increasing values of Prandtl and Reynolds numbers, average value of Nusselt number of circular tube bundle increases. Yoo et al. [11] found that average Nusselt number of second and third tubes in staggered tube bank is higher than first tube. Gupta et al. [12] optimized coil finned tube heat exchanger, by choosing a suitable mean diameter of shell and appropriate clearance for a given fin diameter. Hassan [13] found that in a small tube bundle for decreasing pressure the pitch over tubes should be widened.

One of the aspects in studying flow and heat transfer from multiple bodies is in heat exchangers where reducing pressure drop and increasing heat transfer is of interest to many scientists. There are several studied about flow and heat transfer from non-circular tubes [14–22]. Rocha et al. [14] showed that compare to circular tubes plate fin heat exchangers, elliptic one performed better due to lower pressure drop and higher fin efficiency. Matos et al. [15,16] also found that elliptic tubes perform more efficiently than circular one. Ibrahim and Gomma [17] concluded that elliptic tube bank at zero angle of attack has the maximum thermal performance. Ibrahim and Moawed [18] found that in an elliptic tubes with longitudinal fins, the position of fin on elliptic tubes, effects on friction factor and heat transfer. Bouris et al. [19] reported that in in-line tube bank, deposition rate for elliptic-shaped tubes is 73% lower than circular tubes. Nouri-Borujerdi and Lavasani [20,21] experimentally measured flow and heat transfer characteristics around single cam-shape tube. Moawed [22] experimentally investigated forced convection from outside surface of helical coiled tube.

Furthermore, several authors used vortex generator in order to increase thermal performance of heat exchanger [23–25]. Joardar and Jacobi [23] reported that adding vortex generator enhanced heat transfer with modest pressure drop penalties. However, Wu and Tao [24] and Wu et al. [25] showed that it is possible to enhance heat transfer with reduction in pressure drop by using longitudinal vortex generator.

Compare to other works on literature streamlined-shaped tube bundle, has higher thermal–hydraulic performance and need less pumping power due to low hydraulic resistance. Because cam-shaped tube compare to circular tube has lower drag coefficient [20,21] and higher heat transfer of staggered tube bundle compared

* Corresponding author. Tel.: +98 937 1681530.
E-mail address: hrb.mech@gmail.com (H. Bayat).

http://dx.doi.org/10.1016/j.enconman.2014.06.009
0196-8904/

Nomenclature

C	circumferential length (mm)
C_D	drag coefficient
$c_{p,i}$	pressure coefficient
d	small diameter (mm)
D	large diameter (mm)
D_{eq}	equivalent diameter, $D_{eq} = C/\pi$ (mm)
f	friction factor
h	heat transfer coefficient (W m^{-2} K^{-1})
j	Colburn factor, $Nu/(Re \cdot Pr^{1/3})$
k	thermal conductivity (W m^{-1} K^{-1})
L	tube length (cm)
l	distance between centers (mm)
$\dot{m}$	mass flow rate (kg s^{-1})
N_L	number of transverse rows
P	pressure (Pa)
$\dot{Q}$	heat transfer rate (W)
S_D	diagonal pitch, (m)
S_L/D_{eq}	longitudinal pitch ratio
S_T/D_{eq}	transverse pitch ratio
Re_{eq}	Reynolds number, $(U_\infty D_{eq}/\upsilon)$
Re_D	Reynolds number, $(U_{max} D_{eq}/\upsilon)$
Nu	Nusselt Number, (hD_{eq}/k)
T	temperature (K)
U	velocity (m s^{-1})
U_{max}	maximum velocity, $(\frac{S_T}{S_T - D} U)$ (m s^{-1})
$\dot{V}_w$	volume flow rate (L s^{-1})
Greek	
i	density (kg m^{-3})
υ	fluid kinematic viscosity (m^2 s^{-1})
η	thermal–hydraulic performance
σ	$A_{free\ flow\ area}/A_{frontal\ area}$
θ	hole angle (degree)
Subscripts	
ave.	average
cam	cam-shaped tube
eq	equivalent
i	inlet
o	outlet
s	surface
w	water
∞	free stream

to in-line tube bundle, the purpose of this study is to experimentally investigate the flow and heat transfer characteristics around cam-shaped tube bundle in staggered arrangements subject to cross flow of air.

2. Experimental setup

The cross section profile of the cam-shaped tube is represented in Fig. 1. These tube are comprised of two circles with two arcs segments tangent to them and are made of commercial steel plate with 0.7 mm of wall thickness. Identical diameters of tubes are equal to d = 8 mm, D = 16 mm and distance between their centers is l = 15.75 mm.

A test tube with length of 31 cm was made, in order to measure drag coefficient of cam shaped tube in tube bank. To measure the static pressure on the tube surface by using a digital differential pressure meter, fourteen holes with diameter of 1 mm were drilled on the surface of test tube. Four test tubes with length of 22 cm were made for measuring heat transfer. In order to decrease heat transfer from these surfaces the two ends of test tubes were insulated by using elastomeric thermal tube insulation.

Fig. 2 shows fourteen cam-shaped tubes located at wind tunnel test section. The space between two tandem tubes is defined by longitudinal pitch S_L and the space between side-by-side tubes is defined by transverse pitch S_T. In this study transverse pitch ratio is S_T/D_{eq} = 1.25 and longitudinal pitch ratios are S_L/D_{eq} = 1.5 and 2. Fig. 3 shows an open circuit low speed wind tunnel where the experiments were performed. A pitot static tube is used to measure the free stream velocity in front of the frame cross section. The air velocity varied from 9 to 15 m/s by controlling a variable speed motor.

To heat up the tubes, a pump circulates hot water between a tank and the tubes. An electric heating element supplies the hot water and a control valve regulates the hot water at the tube inlet. Water temperature is measured at the inlet and outlet of the tubes using type-k thermocouple wires and saved at interval times of one second by using data logger. A glass tube flow meter measures the flow rate with 1% uncertainty in full-scale flow. A steady state condition is reached between 5 and 15 min, depending on the ambient temperature and free stream velocity, and then data collection is started.

To estimate the pressure drag and heat transfer from the cam shaped tubes compared to that of a circular tube with various cross sections, it is important to select an appropriate reference length. D_{eq} is the diameter of an equivalent circular tube whose circumferential length is equal to that of the cam-shaped tube. Based on Fig. 1, the equivalent diameter is obtained by $D_{eq} = P/\pi$ = 22.44 mm where P is perimeter of cam shape tube.

For understanding flow characteristic better, Reynolds number is defined with two equations. First, for comparing heat transfer from each tube in tube bank with single tube in crossflow, Reynolds number is calculated by $Re_{eq} = U_\infty \cdot D_{eq}/\nu$. Second, since the speed of fluid varies along its path in tube bank, a reference velocity base on minimum free area available for fluid flow is being used for calculating of $Re_D = U_{max} \cdot D_{eq}/\nu$. There are two correlations for

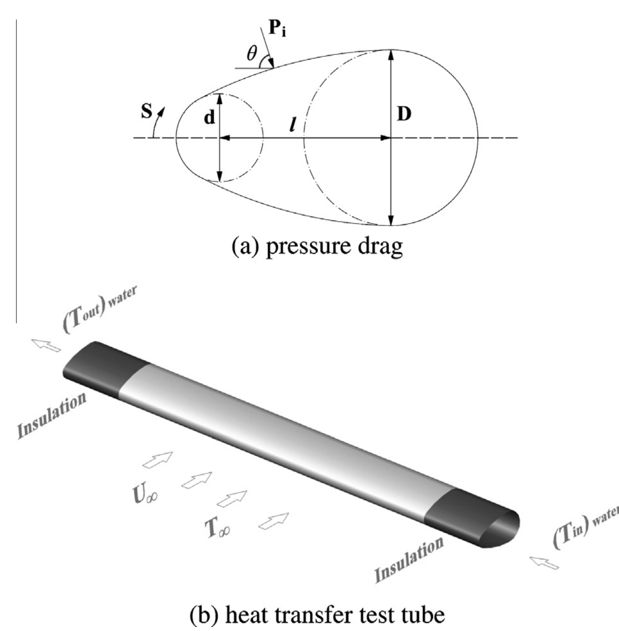

Fig. 1. Schematic of a cam-shaped tube: (a) pressure drag, (b) heat transfer test tube.