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Performance of bare-tube bundle having small diameter tube: With and without partial bypass☆

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ABSTRACT

The present study experimentally investigates the air-side performance of the staggered bare tube bundle having a small diameter tube in association with the partial bypass design. A total of ten arrangements are made and tested with corresponding surface reduction of 5%, 10%, and 20%, respectively. The measured heat transfer and pressure drop results of the conventional bare tube bundle agree well with the existing predictions. For the typical tube bundle, both heat transfer rate and heat transfer coefficient (HTC) first increase with the tube row and peaks at the fourth row. The HTC then remains almost unchanged when the tube row is further increased. Through the partial bypass designs, the front part of the tube bundle suffers some heat losses for having some fewer available surfaces but the rear part of the tube bundle will achieve a higher heat transfer rate for a higher temperature difference. The phenomenon of losing heat transfer in the front part but gaining performance in the rear part becomes more pronounced with the surface area reduction. For the partial bypass design with the same area reduction, it appears that the vertical reduction designs are marginally better than those lateral reduction designs. For comparisons of the effective thermal resistance subject to the same pumping power. It is clearly seen that the convectional design still outperforms all partial designs. As the pumping power becomes larger, the performance of the "partial bypass" designs are slowly and gradually improved to close to the traditional tube bundle. The difference is comparative small with the 5% and 10% "partial bypass" design. This trend indicates that the "partial bypass" is more beneficial at the operating condition of higher pumping power especially for the 5% surface reduction design.

1. Introduction

The air-cooled heat exchangers are widely used in many kinds of industrial applications such as air conditioning and refrigeration systems as well as process and power industry as coolers, heaters, evaporators and condensers. In practice, the air-side thermal resistance comprises around 70–95% of the total thermal resistance. Hence, most studies had focused on reducing air-side resistance and the mostly commonly used method is via adding surface area, i.e. fin-and-tube heat exchangers. However, this may also accompany with enormous pressure drop penalty and severe fouling problem. Hence an alternative way is to increase the air side heat transfer coefficient via enhanced geometries such as wavy, slit, louver, and vortex generator geometries, and some ingenious designs regarding to the fin patterns were reviewed by Wang [1,2]. In addition to the fin-and-tube heat exchangers, air-cooled heat exchangers can also take the form as bare-tube bundle. Without fins, this kind of heat exchanger has a much lower heat transfer performance; however, it becomes prominent if the air flow conditions are operating in worse environments where serious fouling or corrosion prevails. Besides, it is also difficult to clean those heat exchangers equipped with dense fins. Based on the above considerations, bare-tube bundle is regarded as reliable heat exchangers in particular applications, and there were many researches reporting the air-side performance, e.g. Grimson [3], Zukauskas and Ulinskas [4], and Hausen et al. [5] and some empirical correlations in terms of Nusselt number and friction factor were proposed. In addition to experimental works, Khan et al. [6] used the integral method of boundary layer to derivate the average Nusselt number of the tube bundle whose tube wall temperature was is constant. Gnielinski [7] had developed a prediction correlation of air-side heat transfer coefficient applied to both staggered and inline arrangements. His equation for staggered arrangements is listed in the following:

$$Nu_{bundle} = \left(1+\frac{2}{3P_L}\right)\left(0.3+\sqrt{Nu_{lam}^2+Nu_{turb}^2}\right) \quad (1)$$

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Nomenclature

A	Heat transfer area (m^2)
c_n	Coefficient
C_p	Specific heat at constant pressure (J/kg K)
D	Tube outside diameter (m)
D_i	Tube inside diameter (m)
f	Friction factor
F	Correction factor
h	Heat transfer coefficient (Wm^2/K)
HTC	Heat transfer coefficient (Wm^2/K)
k	Thermal conductivity (W/m K)
$LMTD$	Logarithmic mean temperature difference (K)
$\dot{m}$	Mass flow rate (kg/s)
Nu	Nusselt number
N	Number of rube rows
Pr	Prandtl number
P_L	S_L/D
P_T	S_T/D
Q	Heat transfer rate (W)
Re	Reynolds number
T	Temperature (K)
U	Overall heat transfer coefficient (Wm^2/K)
V_{fr}	Frontal velocity (m/s)
ρ	
μ	Dynamic viscosity (Pa s)
ν	Kinematic viscosity (m^2/s)
ΔP	Pressure drop (Pa)

Subscripts

a	Air side
avg	Average
c	Cold side fluid
h	Hot side fluid
i	Inlet
lam	Laminar flow
max	Maximum
o	Outlet or outer surface
s	Wall
t	Total
$turb$	Turbulent flow
w	Water side

where

$$Nu_{lam} = 0.664\sqrt{Re_\Psi}Pr^{1/3}, \tag{2}$$

$$Nu_{turb} = \frac{0.037Re_\Psi^{0.8}Pr}{1+2.443Re_\Psi^{-0.1}\left(Pr^{\frac{2}{3}}-1\right)}, \tag{3}$$

$$Re_\Psi = \frac{VL}{\Psi\nu}\left(10 < Re_\Psi < 10^6\right), \tag{4}$$

$$L_c = \pi D/2, \tag{5}$$

$$\text{when } P_L \geq 1, \Psi = 1-\frac{\pi}{4P_T}, \tag{6}$$

$$\text{when } P_L < 1, \Psi = 1-\frac{\pi}{4P_TP_L}. \tag{7}$$

If the bundle containing less than ten tube rows, the prediction equation is modified as the following form:

$$\frac{1+(N-1)\left(1+\frac{2}{3P_L}\right)}{N}\left(0.3+\sqrt{Nu_{lam}^2+Nu_{turb}^2}\right) \tag{8}$$

where N is the total number of the tube row. All properties in above equations are based on the average of inlet and outlet temperature. After obtaining Nu_{bundle}, the air-side heat transfer coefficient can be found out by the given equation:

$$h = Nu_{bundle}\frac{k}{L_c}. \tag{9}$$

Some other very comprehensive studies concerning the prediction of air-side heat transfer coefficient and pressure drop in tube bundle had been presented by Zukauskas [8,9]. The heat transfer correlations applicable for the staggered tube bundle:

$$\text{when } Re_{D,max} = 1 \sim 500$$
$$Nu = 1.04c_n\left(Re_{D,max}\right)^{0.4}Pr^{0.36}\left(\frac{Pr}{Pr_s}\right)^{0.25}, \tag{10}$$

$$\text{when } Re_{D,max} = 500 \sim 10^3$$
$$Nu = 0.71c_n\left(Re_{D,max}\right)^{0.5}Pr^{0.36}\left(\frac{Pr}{Pr_s}\right)^{0.25}, \tag{11}$$

$$\text{when } Re_{D,max} = 10^3 \sim 2\times10^5$$
$$Nu = 0.35c_n\left(Re_{D,max}\right)^{0.6}Pr^{0.36}\left(\frac{Pr}{Pr_s}\right)^{0.25}\left(\frac{S_T}{S_L}\right)^{0.2}, \tag{12}$$

$$\text{when } Re_{D,max} = 2\times10^5 \sim 2\times10^6$$
$$Nu = 0.031c_n\left(Re_{D,max}\right)^{0.8}Pr^{0.4}\left(\frac{Pr}{Pr_s}\right)^{0.25}\left(\frac{S_T}{S_L}\right)^{0.2}, \tag{13}$$

where Pr_s is based on the average temperature on the tube wall. The remaining properties are based on the average temperature of air-side inlet and outlet.

However, the pressure drop correlations of Zukauskas [8,9] were applicable only to equilateral-triangle staggered and inline arrangement. Besides, some deviations of the predicted equations are encountered if the number of tube rows is less than eight or when the Reynolds number is lower than 1000. Wang et al. [10] and Lee et al. [11] had made correction in these ranges. Similar correlation was proposed and made available from the literature [12].

Note that the forgoing studies were generally based on comparatively larger diameter (normally D ≥ 10 mm) for primarily industrial applications. Hence, the first objective of this study is to examine the applicability of the previous correlations when a small diameter tube is employed. In addition, the present study also aims at valuation of the possibility of so-called "partial bypass" concept in the multi-row bare-tube bundle subject to air-cooling. The "partial bypass" is taken from Wang et al. [13] and Wang [14] who proposed a concept for heat transfer augmentation of fin-and-tube heat exchangers by increasing the effective temperature difference between the air side and tube side especially at the rear part of the heat exchangers. The basic concept can be depicted in Fig. 1. The idea is to reduce the number of tubes in the front part of the heat exchanger, thus the heat transfer area is reduced due to part of the inlet stream directly bypasses the front part of heat exchanger. However, all the air flow gathers to pass through the rear part of the heat exchanger. Although the heat transfer performance of the front part of heat exchanger will be reduced by the loss of the heat transfer area, the heat transfer performance of the rear part is increased substantially due to effective temperature rise of the rear part of the heat exchanger. The idea made use the fact that the heat transfer