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Mathematical modeling and experimental study of heat transfer in a low-duty air-cooled heat exchanger

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ABSTRACT

Many plate fin and tube heat exchangers operate under low loads when the flow rate of air and water is low. The flow regime in tubes can change from laminar through transitional to turbulent. In this paper, much attention was paid to determine a semi-empirical correlation for the Nusselt number on the water-side in the transitional and turbulent range when the Reynolds number Re_w is higher than 2300. A new relationship for the friction factor in the transitional flow regime was proposed. The friction factor in the transitional flow range $2300 \leqslant Re_w \leqslant 3000$ was obtained by linear interpolation of the friction factor for $Re_w = 2300$, and $Re_w = 3000$. The influence of the formula for determining the water-side friction factor in the transitional flow regime on the Nusselt number was shown. Based on experimental data, heat transfer correlations were determined for the air and water-side Nusselt numbers for the low velocity of air and water. The semi-empirical correlation for the Nusselt number on the water-side derived by integrating the Reynolds averaged energy conservation equation for turbulent flow agrees well with the empirical correlation for the Nusselt number. The heat flow rate from hot water to cold air was calculated as a function of the water flow rate using a numerical model of the heat exchanger with the correlation for the water-side Nusselt number developed in the paper to compare it with the results of the measurements. The results of the numerical simulation agree very well with the results of the measurements.

1. Introduction

Finned heat exchangers are used when one fluid is a gas and the liquid flows inside the tubes. The fins are attached to the tubes. Water heaters, evaporators, and superheaters in heat recovery steam generators (HRSG) are usually made of finned tubes. Heat exchangers with tubes finned by individual or continuous fins are the essential equipment in dry cooling systems used to cool water in power stations [1]. Plate fin and tube heat exchangers (PFTHE) are widely used as car radiators and car air heaters. Also, PFTHEs are very widespread in ventilation, and air conditioning systems. The intensive development of renewable energy sources favors the development of PFTHEs. The exchangers in such installations are characterized by low load operation when velocities of the air flowing perpendicular to the tube axis and liquid flowing inside the tube are low. Thermal and hydraulic design theory of single-phase PFTHEs for steady-state operation are presented in detail in a book by Shah and Sekulić [2]. Thermo-hydraulic fundamentals, design, and fabrication of heat exchangers, including PFTHEs, are described in the book of Kuppan [3]. Various techniques of heat augmentation in PFTHEs are thoroughly discussed by Webb [4]. Many enhanced airside surfaces such as various wavy plate fins and louvered fins are discussed. Also, different techniques for intensifying the heat transfer inside the tubes are presented.

In the design calculations of heat exchangers, the log mean temperature difference (LMTD) method is most commonly used. In the performance calculations of PFTHEs, the ε-NTU or P-NTU methods are widely used for non-iterative calculations of the temperature of both fluids at the outlet of the PFTHEs. The symbols ε and P denote the effectiveness of the heat exchanger, and the NTU denotes the number of transfer units. Formulas and diagrams for $\varepsilon(\mathrm{NTU})$ and $P(\mathrm{NTU})$ for PFTHEs with various flow arrangements with a different number of passes are presented by Shah and Sekulić [3] and Kuppan [4]. The outlet temperatures of both fluids can easily be calculated using the formulas for calculating ε or P for a given NTU value. If the exchanger flow system is complicated and there are no formulas for calculating the effectiveness ε or P, then the numerical model of the PFTHE exchanger can be constructed using the method developed by Taler [5]. Heat transfer correlations for calculating heat transfer coefficients on both air and water side are needed in the thermal calculations of heat exchangers using engineering methods such as LMTD, ε-NTU, and P-NTU. These correlations are also required in the finite difference model developed in [5].

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Nomenclature

A	area (m^2)
A_{cin}, A_{co}	inner and outer cross section area of the oval tube (m^2)
A_f	heat transfer area of the fin (m^2)
A_{mf}	area of the bare tube between two adjacent fins (m^2)
A_m	mean surface of the tube, $A_m = (A_o + A_{in})/2$
$A_{\min}$	minimum free flow frontal area on the air side (m^2)
c	specific heat (J/(kg·K))
cc	cubic centimeter (cm^3)
$c_p\|_0^T$	mean specific heat at constant pressure at the temperature interval with the limit 0 and T (J/(kg·K))
A_{in}, A_o	inner and outer surface of the bare tube (m^2)
d_a	hydraulic diameter of air flow passages (m)
$d_{in,\min}$, $d_{in,\max}$	minimum and maximum inner diameter of the oval tube, respectively (m)
$d_{o,\min}$, $d_{o,\max}$	minimum and maximum outer diameter of the oval tube, respectively (m)
d_w	hydraulic diameter on the liquid side, $4A_{in}/P_{in}$ (m)
h	convective heat transfer coefficient (W/(m^2·K))
h_a, h_w	the air- and water-side heat transfer coefficient (W/(m^2·K))
h_o	effective heat transfer coefficient considering fin efficiency based on the outer surface area of the bare tube (W/(m^2·K))
k	thermal conductivity (W/(m·K))
k_t	thermal conductivity of the tube material (W/(m·K))
(k)	iteration number
L	tube length (m)
L_{ch}	length of the heat exchanger (m)
m	number of searched parameters
$\dot{m}$	mass flow rate (kg/s)
$\dot{m}_a$	air mass rate in the automobile radiator (kg/s)
$\dot{m}_w$	water mass rate in the automobile radiator (kg/s)
n	number of data sets
N_g, N_l	number of transfer units for the air and water side, respectively
$Nu_{m,q}$	mean Nusselt number for laminar flow in the tube for constant heat wall flux
$Nu_{m,q,1}$	mean Nusselt number for hydrodynamically and thermally fully developed flow in the tube with uniform heat flux
Nu_w	water-side Nusselt number, $h_w d_w/k_w$
Nu_a	air-side Nusselt number, $h_a d_a/k_a$
p_1	pitch of tubes in plane perpendicular to flow (height of the fin) (m)
p_2	pitch of tubes in the direction of flow (width of the fin) (m)
P_{in}, P_o	inner and outer perimeter of the oval tube, respectively (m)
Pr	Prandtl number, $\mu c_p/k$
$\dot{Q}$	heat transfer rate (W)
Re_a	air-side Reynolds number, $w_{\max} d_a/\nu_a$
Re_w	liquid-side Reynolds number, $w_w d_w/\nu_w$
s	fin pitch (m)
$t_{n-m}^{\alpha/2}$	quantile of the Student's t distribution for the confidence level $1-\alpha$ and $n-m$ degrees of freedom
T	temperature (°C or K)
T'_{am}, T''_{am}	mean inlet and outlet temperature of the air (°C)
T'_w, T''_w	water inlet and outlet temperature, respectively (°C)
$T''_{w,1}$, $T''_{w,2}$	water outlet temperature from the first and the second tube row in the upper (first) pass, respectively (°C)
$T''_{w,3}$, $T''_{w,4}$	water outlet temperature from the first and the second tube row in the lower (second) pass, respectively (°C)
T_{wm}	outlet temperature of the water downstream the first pass (°C)
$(T''_w)^{meas}$	measured water temperature at the outlet of the heat exchanger (°C)
U_o	overall heat transfer coefficient that is referred to the outer surface area of the bare tube (W/(m^2·K))
$\dot{V}_w$	water volume flow rate at the inlet of the heat exchanger (L/h or m^3/s)
w_0	average frontal flow velocity (air velocity before the heat exchanger) (m/s)
$w_{\max}$	mean axial velocity in the minimum free flow area (m/s)
W_{ch}	thickness of the heat exchanger, $W_{ch} = 2p_2$ (m)
x, y, z	Cartesian coordinates
x^+	dimensionless coordinate, $x^+ = x/L_{ch}$
y^+	dimensionless coordinate, $y^+ = y/p_2$

Greek symbols

δ_f	fin thickness (m)
δ_t	tube wall thickness (m)
Γ	Gamma function
μ	dynamic viscosity (Pa·s)
η_f	fin efficiency
ν	kinematic viscosity (m^2/s)
ξ	Darcy–Weisbach friction factor
ρ	fluid density (kg/m^3)
ρ_{am}	mean air density in the heat exchanger (kg/m^3)
ρ_{aw}	mean water density in the heat exchanger (kg/m^3)

Subscripts

a	air
in	inner
l	liquid
p	at constant pressure
w	water

Superscripts

$meas$	measured
$+$	dimensionless
$-$	mean
$'$	inlet
$''$	outlet

In the paper by Fahmy and Nabih [6], the LMTD method was used to evaluate the air-flow rate in finned air-cooled heat exchanger used for propane pre-cooling in an LNG (liquefied natural gas) installation. The Sieder and Tate correlation for turbulent tube flow [7] was applied for the calculation of the heat transfer coefficient at the tube inner surface and a power-type relationship for evaluating the air-side heat transfer coefficient.

The correlations on the air-side are typically determined experimentally for different wavy fins by maintaining a turbulent fluid flow inside the tubes. The relationships for determining the tube-side heat transfer coefficient are not determined experimentally but taken from the literature for turbulent flow of liquid in long straight tubes. The Gnielinski correlation [9] for the turbulent flow was used in the article by Li et al. [10] and by Wen et al. [11] to evaluate the waterside heat transfer coefficient. The formula proposed by Filonienko [12] was used for calculating the Darcy–Weisbach friction factor ξ_w appearing in the Gnielinski relationship. The disadvantages and advantages of determining only air-side correlation, assuming that the correlation for